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AMA

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

VOL.31, NO.3, SUMMER 2000

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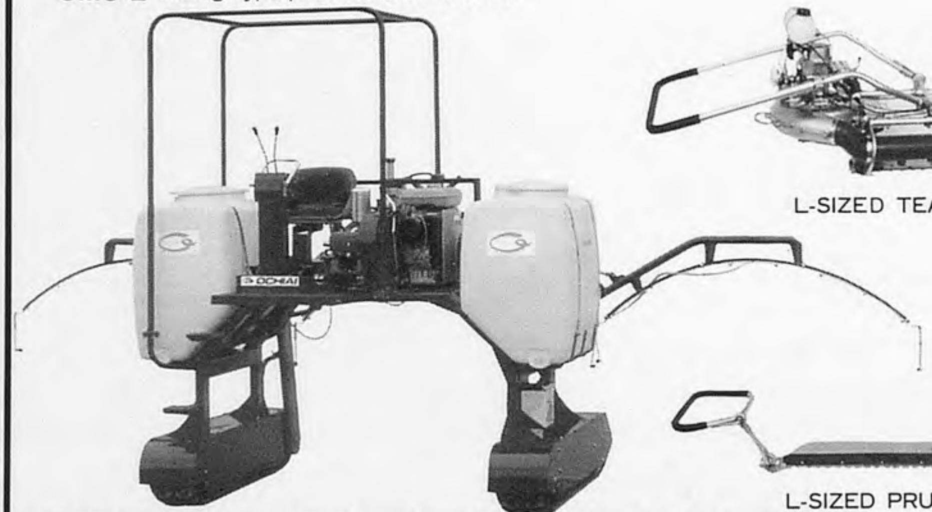

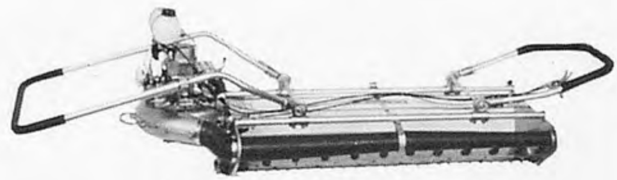

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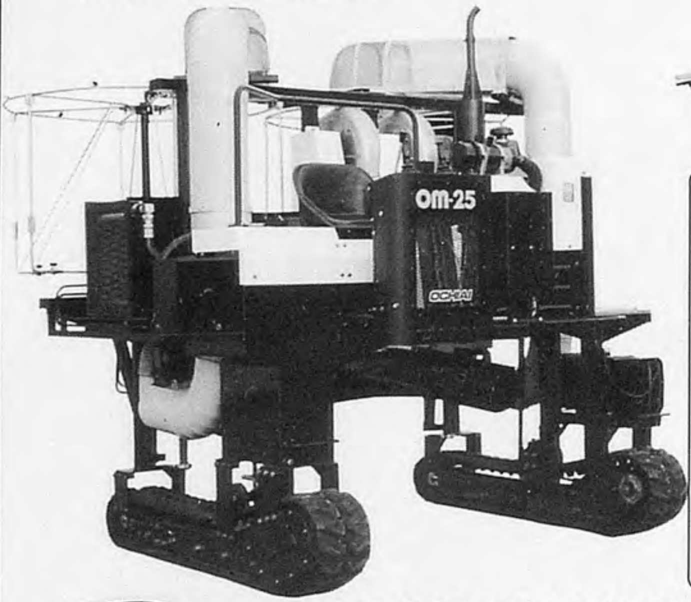

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CONTRIBUTORS WANTED

This publication, published quarterly, has an objective to promote agricultural mechanization in developing countries. Its readers consist of so many people in various fields such as farmers, dealers, manufacturers, researchers, government officials, students, etc. not only in Asia but also in the whole world. To enrich contents and to reflect many opinions, we want contributors for **Agricultural Mechanization in Asia" Africa and Latin America**. Articles, comments, investigations, reports and so on will be received with open arms. If you hope to contribute, contact us without delay.

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EDITORIAL

Some Contrasting Irony of Fate

The mass media recently flashed around the world that some African countries, particularly Ethiopia, are in the grip of the deadly AIDS infection even as many international organizations are undertaking rescue efforts to save the infected people and helping in the mounting refugee problem as a result. It is rather an irony of fate that many third world countries still have to suffer this crisis on top of their continuing difficulties or providing adequate food and shelter to the poorest of the poor.

Many ironies of fate are in stark contrast up to the present time such as: the dwindling resources world-wide and the ever widening gap between the rich and the poor: like the dreadful overweight problem among the children and adults alike in many advanced countries in contrast to their emaciated and underfed counterparts in many poor countries; and the display on TV of well-off families having their meals in air-conditioned dining rooms in the face of hunger and malnutrition in many parts of the world. To say the least, food and potable water, the most essential elements of health and life itself, are still in short supply in many countries.

As we head into the 21st century, we are witness to the accelerating progress in science and technologies. For example, the human genome has been reported recently that it has already been broken by scientists. And yet, it is rather sad to realize that the adequate supply of food has yet to be met in many parts of the world. As human population increases year after year, natural resources, in contrast, continues to diminish.

What all this portends is that in the next century, we have very little chance to raise agricultural productivity level through the development and spread of technologies. And no one can deny that the use of appropriate farm machineries not only release farmers from the back-breaking farm chores but more importantly, to raise land productivity.

As the AMA since has been promoting improved farming practices, hence improved crop yields through the use of farm machineries, we reiterate, at the expense of being repetitious, must be adopted in Asia, Africa and Latin America. We will keep rallying agricultural engineers to join AMA in this crusade.

Yoshisuke Kishida
Chief Editor

Tokyo, Japan
July 2000

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Development and Evaluation of Loading Car for Assessment of Drawbar Performance of Power Tiller

by
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Abstract

The requirement for Indian farming system demands the development of a power tiller as a versatile power source for various farming operations which primarily depend on the drawbar output of the power tiller. For evaluating the drawbar performance of a power tiller under varying operating conditions, the prime requirement is the controlled and precise loading of the power tiller through drawbar. A loading car consisting of power transmission system and hydraulic loading system was developed to apply drawbar load precisely at any desired level. It is capable of applying a drawbar load accurately from 40 to 200 kg at an increment of 5 kg. The forward speed and slip was measured by means of a revolution counter fixed to the wheel axle of the power tiller. The fuel consumption was measured by using a fuel measuring device. The drawbar pull developed by the power tiller was monitored from the hydraulic dynamometer mounted between the power tiller and the loading car. The drawbar horse power and the drawbar specific fuel consumption were computed. The drawbar horse power of 7.46 kW power tiller was found to be very low which varied from 0.30 to 0.65 kW and 0.20 to 0.50 kW in untilled and tilled soil conditions, respectively. The minimum point of drawbar specific fuel consumption(1.1 kg/kW hr) coincided with the maximum drawbar power (0.65 kW).

Introduction

Power tillers were introduced primarily as a source of farm power to prepare seedbeds with rotary tiller and for transportation. The requirement for Indian farming system demands the development of a power tiller as a versatile power source for operation in small farms such as ploughing, levelling, ridging, inter-cultivation, root crop harvesting, etc. These operations primarily depend on the drawbar output of the power unit. The power tillers at present have limitation in their use for traction work due to low DBHP per BHP of the engine. For evaluating the drawbar performance of a power tiller under varying soil conditions, the prime requirement is the controlled and precise loading of the power tiller through drawbar. The existing loading devices are specially designed for four wheeled tractors and the operation is usually in the higher horse power range. They are capable of applying drawbar load in the range of 300 to 500 kg. In this paper the development of a loading car and the evaluation of the same for assessing the drawbar performance of power tiller are reported.

Review of Literature

Devnani (1981) reported the development of an Iowa horse testing car. A gear pump was mounted and run by a wagon on which the loading machine was constructed. Tomar (1985) reported that the drawbar horse power available with the power tillers is only 10 to 20 per cent of the brake horse power, whereas in the case of a tractor it is 30 to 60 per cent. Dogan Erdogan (1987) developed a loading car by replacing the engine of a tractor

Fig. 1 Loading car.

with a hydraulic pump. The pump apparatus was equipped with controlling elements such as flow variant valve and directional valve. Erdogan also measured the tractive performance of four small tractors under field conditions and reported that the drawbar force of the tractors on the stubble fields with hard surface was higher than the one on the fields with soft surface. Srivastava and Pandey (1992) reported the design, development and evaluation of an animal loading car for draftability studies of animals in the draft range of 30 - 500 kg. The braking effect on the loading car is achieved through the control of outlet pressure of two gear pumps run by traction wheels of the car. Hydraulic pumps and control valves ensured precise and reliable load settings.. Pudijono and Macmillan (1995) measured the drawbar performance of a 5.6 kW power tiller on firm and soft soils and observed that 30 per cent decrease in maximum drawbar horse power in soft soil when compared to firm soil. Suresh Narang and Varshney (1995) evaluated a 6.71 kW power tiller for draft and drawbar power on tar roads at different gear positions and concluded that the wheel slip increased with the increase in draft and drawbar power of the power tiller. The drawbar power increased with the operation of the power tiller in high gears and at high engine speed. The specific fuel consumption decreased with an increase in drawbar power and was less in high gears.

Materials and Methods

Concept and Development of Loading car

The loading car developed for assessing the drawbar performance of power tiller consists of two systems *viz*., power transmission system and hydraulic loading system.

Power transmission system

When the loading car is pulled by a power unit at certain speed the tractive force is converted into a rotary mechanical power. This power is utilized to run a hydraulic pump. For conversion of tractive force into rotary power, the chassis, gear box and wheels of an old power tiller were used. The chassis along with the transmission system were taken from an old 'ISEKI' make power tiller. The engine of the power tiller was removed from the chassis. On the chassis, a rectangular box frame of size 590 × 350 mm, made of 40 × 6 mm mild steel 'L' angle was mounted vertically near the gear box. Dead weights were placed inside the frame work to avoid skidding of the drive wheel. Adjoining the weight box on the chassis frame, a gear pump with base plate was installed (**Fig. 1**). In the front end of the chassis, a 220 mm dia caster wheel with swiveling action was provided for adjusting the horizontal component of pull to be at zero degree always with horizontal and for better manoeuvrability and stability.

For proper balancing, adequate dead weights were also placed on the handle of the loading car. The loading car was mounted on dual wheels of the same size to increase the wheel track for improved stability in the field condition. The additional wheel was attached to the wheel hub assembly with the help of a flange coupling fabricated for this purpose. The output rotary power was obtained at the loading car clutch pulley shaft The linkages meant for gear shifting were modified such that it could be easily controlled by the operator.

Hydraulic loading system

The hydraulic system is to utilize the mechanical power available from the tractive system to run a hydraulic pump. By using a positive displacement pump driven by the ground wheels of the loading car, the load can be controlled by regulating the outlet opening of the pump. This is an accurate and easy method of varying the load . When hydraulic method of loading using a positive displacement pump is employed, the smoothness of flow and the speed ratio of the drive and driven mechanisms are the main limiting factors which control the uniformity and range of loading.

Gear pump

For recirculation of the hydraulic fluid, a 'DOWTY' make gear pump was used. The specification of the gear pump is shown in **Table 1**.

The pump was mounted on a rectangular base plate of size 300 × 150

Table 1. Specification of the Gear Pump

Particulars	Unit	Technical details
Theoretical displacement,	cm^3/rev	9.46
Nominal delivery at 1500 rpm,	lit/min	12.7
Maximum pressure,	kg/cm^2	214
Maximum speed at maximum pressure,	rpm	3500
Minimum speed at maximum continuous pressure,	rpm	500
Overall dimensions	mm	94.3 × 44.5 × 17.3
Inlet and outlet size	mm	13

Fig. 2 Power tiller and loading car during field test.

mm, made of 6 mm m.s. sheet. The base plate was fixed on the chassis frame of the loading car in the front portion, adjoining the weight box. A cast iron B type 'V' groove pulley was provided on the pump shaft. A 410 mm diameter cast iron 'V' groove pulley fixed to the clutch pulley shaft is the output shaft of the tractive system. The drive from the clutch pulley was transmitted to the pump pulley of 100 mm diameter through a V-belt transmission. For ensuring proper tension of the belt, slots were provided on the base plate for moving the pump in or out.

Hydraulic reservoir and control systems

The oil reservoir is of 30-litre capacity made of GI sheet and fitted in front of the weight box assembly, supported by mild steel 'L' angle. To minimize vibration, which can be transmitted to the pump by the rigid pipe lines, flexible high pressure hoses were used. The suction and the return lines were fixed on the top of the reservoir. The pump suction line draws oil from a point which is 100 mm above the tank bottom, to present sludge deposits from entering the pump. The suction line was provided with a strainer with standard infiltration of 125 micron. The suction and delivery lines were free from sharp bends to prevent excessive working head. The system was designed to prevent entry of air. A positive head was maintained as far as possible.

To make the loading effect of the device, independent of speed, a 'POLYHYDRON' make pressure regulated relief valve was provided in the return line. It was mounted on the top of the oil reservoir. The pressure relief valve is a direct active valve of guided poppet design with cushion arrangement for greater stability and noise control. The pressure can be adjusted with a set screw and lock nut. A pressure gauge was also provided in the return line to monitor the outlet pressure of the hydraulic fluid.

Draft measuring system

The function of this unit was to sense the drawbar load and indicate its magnitude. For measuring the draft, a hydraulic dynamometer of 0 - 500 kg capacity was used. The weight balancing of the power tiller was done by adding 40 kg of dead weight on the power tiller. A hitch bracket assembly in the form of a rectangular frame of 185 × 120 mm size made of 6 mm mild steel sheet was fabricated and fitted in front of the loading car.

A dynamometer mounting frame consisting of two parts was fabricated. One part was fitted to the drawbar of the power tiller and the other part was fitted to the front hitch bracket assembly of the loading car (**Fig. 2**). The dynamometer connects the two parts which were moving in a telescopic arrangement. The dynamometer was calibrated.

Evaluation of Loading Car

Soil condition

For evaluating the loading car and to asses the drawbar performance of power tiller two types of soil predominant in the region of study viz., black clay loam soil and red loam soil were selected. The mechanical analysis of the two soils indicated that the black clay loam soil had a composition of 34, 10 and 36 per cent of clay, silt and fine sand respectively and the red loam soil had 18, 16 and 34 per cent of clay, silt and fine sand. For primary and secondary tillage operations the power tiller has to be operated in unploughed and ploughed condition of cultivable land. Hence the following two track conditions were selected:

Untilled condition- The soil was left as fallow land for six months after the harvest of the previous crop.

Tilled condition - The fallow land was ploughed twice with tractoR drawn cultivator before conducting the experiment.

The soil properties such as moisture content(MC), bulk density and cone index were measured during the test. The MC of the soil was maintained in the range of 5 to 7 per cent. The bulk density measured at 10 cm depth defined the track condition of the soil. Its value in untilled and tilled conditions in sandy clay loam soil was 1.46 and 1.26 gm/cc, respectively, and the value for sandy loam soil is 1.50 and 1.35 gm/cc, respectively. The cone index value for the untilled and tilled condition in sandy clay loam soil was 9.32 and 2.31 kg/cm^2 and its value for sandy loam soil is 10.12 and 2.89 kg/cm^2, respectively.

Gear position

The power tiller is controlled and operated in the field by an operator who walks behind it. The normal walking speed of the operator in the field is around 2.5 kmph. For effective and efficient manoeuvrability in the field, the power tiller has to be operated at a forward speed of about 2.5 kmph only, which occurs in the gear position range of Low I to Low III. Hence, the low range of gear positions viz., Low I, Low II and Low III were selected for the study. The maximum forward speed corresponding to the above gear position are 1.2, 1.9 and 3.3 kmph respectively.

Drawbar load

The power tiller is mainly used for primary and secondary tillage operations like ploughing with a single bottom mould board plough, harrowing with a cultivator, bunding with a terracer blade. These operations, done with power tiller operated equipments require a draft ranging from 75 to 150 kg. Hence the trial was conducted at four different levels of drawbar load of 60,

Fig. 3 Wheel revolution counter and fuel measuring device.

90, 120 and 150 kg in untilled condition and it was limited to three levels in tilled condition due to higher slippage beyond 120 kg.

Field Test and Measurements

During the field test with the test power tiller, the following parameters were measured: DBP, Wheel slip, Fuel consumption, and Forward speed

Drawbar pull (DBP)

For conducting the field test, the power tiller and the loading car were connected by the hydraulic dynamometer with its mounting frame. The power tiller was operated at the recommended level of 3/4th rated speed of the engine (1500 rpm). By adjusting the height of the castor action wheel provided in front of the loading car, the horizontal component of pull was adjusted to be at zero degree with horizontal. The loading car was pulled by the power tiller in the field and the pressure relief valve was operated by opening the lock nut and rotating the screw for the application of drawbar load to the power tiller. The drawbar load applied was monitored by using the dynamometer gauge. The operation of pressure relief valve was continued till the desired level of drawbar load of 60, 90, 120 and 150 kg was indicated in the dynamometer gauge. The power tiller and the loading car during the test are shown in **Fig. 2**.

Wheel slip

The slip of the power tiller was measured by monitoring the number of revolutions of the wheel over a distance of 30 meters under load and zero load conditions. The revolutions made by the wheel was fed to a revolution counter through a speedometer cable. The revolution counter was mounted at appropriate height for easy observation. The wheel revolution counter is shown in Fig 3. The slip was calculated by using the following formula:

$S = (n_1 - n_0) / n_0 \times 100$

where,

s =Wheel slip, per cent

n_1=Number of revolutions of wheel under load condition for a distance of 30 m

n_0=Number of revolution of wheel under no load condition for a distance of 30 m.

Fuel consumption

For measuring fuel consumption of the power tiller engine, a device consisting of a 50-ml burette was provided between the fuel tank and the fuel injection system of the engine. A separate fuel tank of 500-ml capacity and the burette with control knobs at the bottom rigidly fixed on the two sides of a wooden board were mounted on the power tiller near the clutch assembly with necessary supports (**Fig 3**). Fuel consumption was measured by recording the time required and quantity of fuel consumed during 30-m run and then calculating the fuel consumption on an hourly basis, as given below:

$W_f = 36\ (V_f / t)\ 10^{-4}$

where,

W_f=Fuel consumed, m^3/hr

V_f=Volume of fuel consumed for 30 m run, cm^3

t = Time taken for 30 m run, sec

Drawbar specific fuel consumption (DSFC)

The drawbar specific fuel consumption of power tiller was calculated by using the following expression:

$DSFC = (W_f / DBHP)_f$

where,

DSFC=Drawbar specific fuel consumption, kg/kW hr

W_f =Fuel consumed, m^3/hr

Éœ$_f$ =Density of fuel, kg/m^3

Forward speed

The forward speed of operation was calculated by observing the distance traveled and the time taken:

$S = L / t$

where,

S=Forward speed of operation, m/sec

L=Distance traveled, m

t=Time taken, sec

Drawbar horse power (DBHP)

The DBHP for each run was calculated by using the following formula:

DBHP= (DBP)S 75 × 0.746

where,

DBHP=Drawbar horse power, kw

DBP = Drawbar pull, kg

S = Forward speed of operation, m/sec

Results and Discussion

The DBHP and DSFC were the parameters used for the assessment of the drawbar performance of power tiller by using the loading car. Hence the calculated values of DBHP and DSFC are plotted as a function of slip. The performance curves of the power tiller in untilled and tilled conditions of black clay loam and sandy loam soils under different gear positions are drawn and presented in **Figures 4 to 7**.

From the figures it is observed that the drawbar horse power increased with the increase of slip upto a range of 25 per cent and then decreased at all the gear positions tested in untilled condition of both the soils whereas the same occurred in the slip range of more than 25 per cent in tilled soil condition. The drawbar horse power increased with the increase in gear position in tilled and untilled soil conditions. It will be

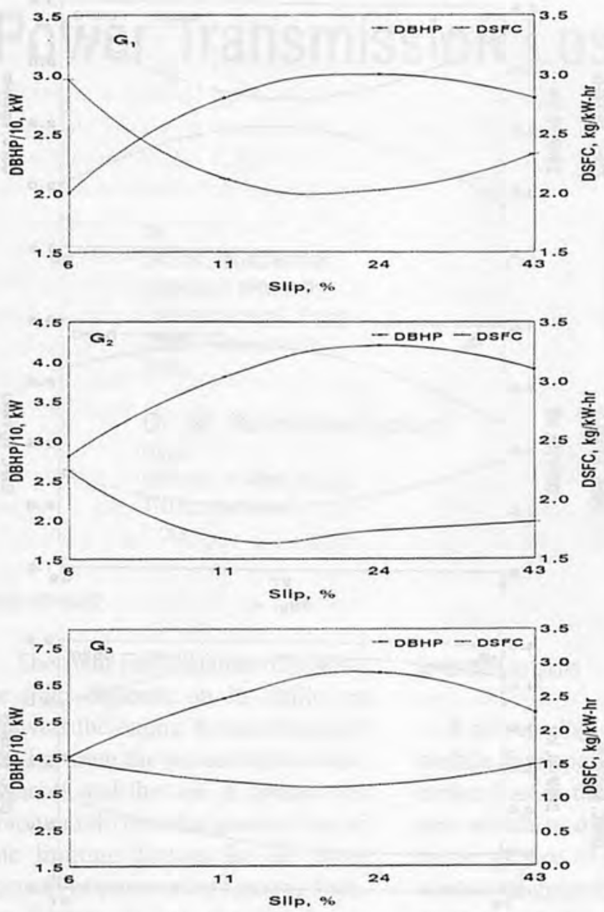

Fig. 4 Drawbar performance in untilled black clay loam soil.

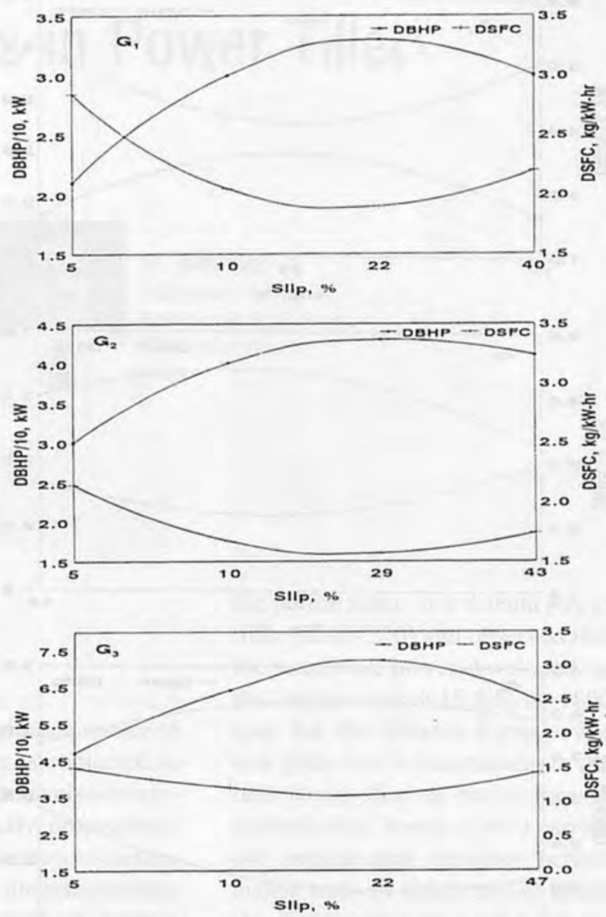

Fig. 5 Drawbar performance in untilled sandy loam soil.

Table 2. Performance Values of Power Tiller at 15 per cent Slip

Track condition	Black clay loam soil			Sandy loam soil		
	Gear position			Gear position		
	G_1	G_2	G_3	G_1	G_2	G_3
i.Tilled soil						
DBHP, kW	0.20	0.31	0.48	0.27	0.36	0.50
DSFC, kg /kW hr	3.20	2.70	2.00	2.70	2.20	1.60
ii. Untilled soil						
DBHP, kW	0.30	0.43	0.67	0.33	0.43	0.65
DSFC, kg /kW hr	2.00	1.70	1.20	1.80	1.50	1.10

noted that the drawbar specific fuel consumption decreased with the increase in slip up to a range of 25 per cent in untilled soil condition and then increased at all the gear positions in the untilled condition of both the soils whereas the same occurred in the slip range of more than 25 per cent in tilled soil condition. The performance values at 15 per cent slip on the test tracks is computed from the figures as specified by the standard test codes and presented in **Table 2**.

From the table it is inferred that the drawbar horse power availability of the power tiller varied from a minimum of 0.20 kW to a maximum of 0.65 kW. From the values furnished in **Table 1**, the variation in DBHP and DSFC values at 15 per cent slip between tilled and untilled soils were computed and the performance of the power tiller in tilled soil was compared with that of the performance in untilled soil at different gear positions. The per cent reduction in DBHP developed was 18 to 33, 16 to 19 and 23 to 28 per cent in tilled soil condition at G_1, G_2 and G_3, respectively. The corresponding per cent increase in DSFC values were 60 to 130, 20 to 170 and 33 to 100. On further analysis regarding the decrease in values of DBHP and increase in values of DSFC among the soil types, it is concluded that the performance in tilled sandy loam soil is better than that of black clay loam soil.

Conclusions

The conclusions drawn from the results of the study are :

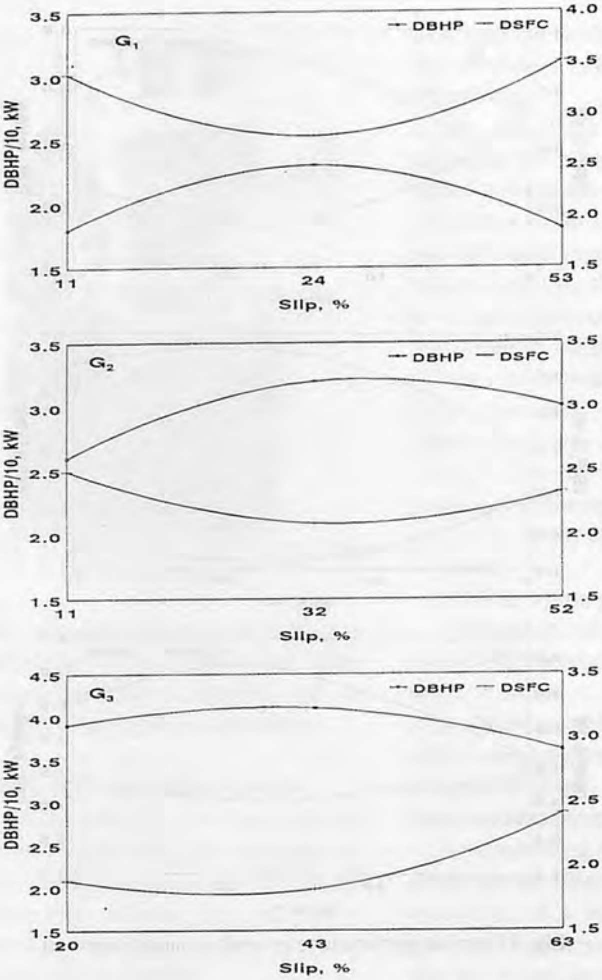

Fig. 6 Drawbar performance in tilled black clay loam soil.

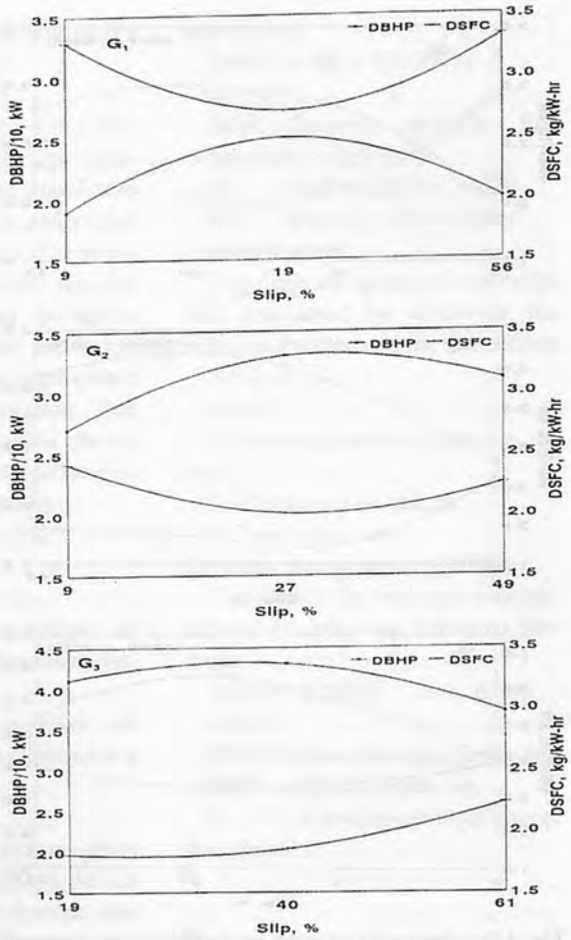

Fig. 7 Drawbar performance in tilled sandy loam soil.

1.A loading car for precise and controlled application of drawbar load to the power tiller was developed. It is capable of applying load from 40 to 200 kg.

2.The drawbar horse power developed by the power tiller varies from a minimum of 0.20 kW to a maximum of 0.65 kW.

3.The performance in tilled soil is marked by a decrease in DBHP of 18 to 33, 16 to 19 and 23 to 28 per cent and increased in DSFC values of 60 to 130, 20 to 170 and 33 to 100 at G_1, G_2 and G_3 respectively.

4.The point of occurrence of maximum drawbar horse power coincided with the minimum drawbar specific fuel consumption.

5.The drawbar performance of power tiller was better in sandy loam soil as compared to black clay loam soil.

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Power Transmission Loss in Power Tiller

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Abstract

The field performance of a power tiller depends on its ability to convert the engine power transmitted between the ground drive components and the soil in contact for traction into drawbar power. One of the limiting factors for the slow growth of power tillers among Indian farmers is low drawbar horse power availability due to substantial power loss. Hence an investigation was undertaken to determine the power loss in a 7.46 kw power tiller. Specially fabricated test rigs were used to measure the maximum power of the engine and to measure the axle horse power available at different gear positions and at 50, 75 and 100 per cent of maximum engine power (power load). The extent of transmission loss was quantified and the conclusions drawn are: (i) the transmission loss increased with an increase in engine speed in all gear positions; (ii) the higher the gear position, the lower is the loss for a given engine speed; (iii) the transmission loss increased linearly with an increase in engine speed; and (iv) the loss varied from 22 to 67, 25 to 65, 33 to 70 per cent of the maximum engine power at 50, 75 and 100 per cent power load respectively.

Introduction

A power tiller used as a source of mobile farm power is subjected to power loss in the transmission system which is directly proportional to the degree of surface-to-surface contact of gears in the transmission system and the extent of heating and churning of transmission oil. The field performance of a power tiller depends mainly on its ability to convert the power transmitted between the ground drive components and the soil in contact for traction into drawbar power. Each type and make of power tiller exhibit characteristics both in isolation and as a part of a system based on the working condition of that particular engine and work environment. Substantial power loss occurs during transmission of power from the engine to ground drive components and its conversion into tractive power. The drawbar performance and the tractive efficiency of the power tillers could be determined only after ascertaining the extent of power loss.

Review of Literature

Garg (1965) carried out tests on the performance of a Kubota power tiller (KMB 200) and observed that the maximum power developed by the engine was 8.15 kw at 1800 rpm, but the drawbar horse power was quite low in comparison to the belt power due to heavy loss in transmission. Verma (1967) carried out engine and drawbar performance tests on Krishi power tillers (V - 500) and observed that the engine did not produce the specific quantum of power. The governor sensitivity also seemed to be poor. The drawbar performance showed that the tractive efficiency was not satisfactory due to heavy loss in transmission and during the conversion of power into drawbar by the wheels. McCarthy and Kolozsi (1974) measured the transmission loss in a tractor and reported that in a given gear, the loss increases with engine speed, within either the high or low gear range: the higher the gear, the higher is the loss for a given engine speed. Over the engine speed range of 1000 - 2000 rpm the loss may be approximated by a straight line. Bhole and Tiwari (1977) analyzed the distribution of power produced by the engine of the power tiller to the different outlets separately and reported that the higher the gear, the lower is the loss for a given engine speed. The transmission loss varied from 7.3 to 63.0

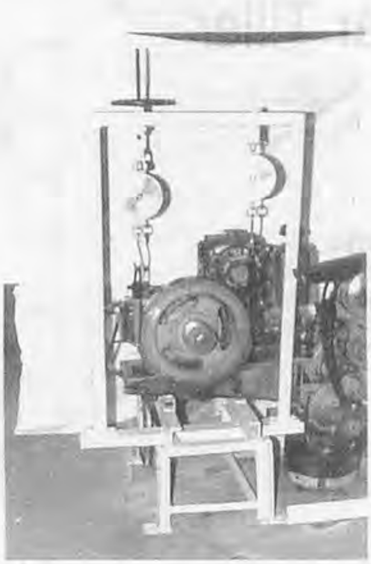

Fig. 1 Engine power testing.

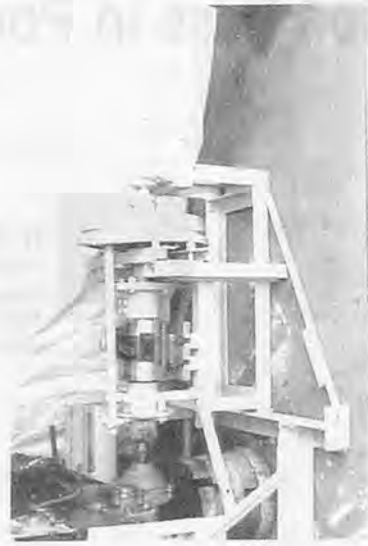

Fig. 2 Torque transducer with breaking device.

per cent for different gear positions.

Materials and Methods

Engine Power Test Rig

The maximum power of the power tiller engine was measured by using a test rig consisting of a mechanical brake drum type dynamometer. The test rig consists of a 1160 × 760 mm rectangular frame as shown in **Fig. 1**. Two spring balances were used for measuring the applied load. A 50-kg capacity spring balance was rigidly fixed at one end of the top member and another 100-kg capacity spring balance was fixed to a screw rod and wheel assembly provided at the other end of the top member of the rectangular frame, so that the spring balance can be moved up and down by rotating the wheel. The bottom hooks of the two balances were connected to two rings provided at the ends of a flat belt. The inner side of the belt was reinforced with asbestos metal brake lining of size 590 × 62 × 7 mm. The fly wheel of the power tiller engine was used as a brake drum which runs in frictional contact with the brake lining.

The wheels were removed and the power tiller was jacked up. The front chassis of the power tiller was fixed on a mounting bed, specially fabricated for this purpose. The chassis was rested on two channel sections of size 75 × 40 mm, placed across the length of the chassis. To the hitch bracket assembly of the power tiller, a 6-mm mild steel plate of size 400 × 210 mm was attached and extended downwards. The mild steel plate was rigidly welded to 690 mm square frame made of 75 × 6 mm 'L' angles, placed on the ground. The engine test rig frame was fixed on the fly wheel side of bed on which the chassis was mounted. The engine speed was measured by using a 'LUTRON DT-2236' make digital photo-cum-contact tachometer, having an accuracy of ± 0.05 per cent.

Measurement of BHP

Full load

For measuring the maximum horse power of the engine, the engine was run at no load for about 30 minutes so that the engine achieves a steady speed. At first, to obtain the maximum engine power and the corresponding idle speed, different values of idle speed were set by changing the throttle position. In each throttle position, from the idle condition, the breaking load on the fly wheel was gradually increased by operating the dynamometer till the point of stable operation of the engine was observed. The speed of the crank shaft was measured by using the tachometer and the load indicated from the two spring balances were recorded. This procedure was repeated till the full open throttle position was reached. The observations were recorded.

To determine the horse power and the corresponding engine speed, the engine was gradually loaded from the idle speed corresponding to maximum power. As the load was increased gradually, the reduction in speed and the corresponding load indicated by the spring balances were observed. This process was continued up to a stage beyond which any further increase in load reduces the engine speed to such an extent that the engine appears to be installed. After this, further load was not applied. The test was conducted at an ambient temperature of 26°C and at an atmospheric pressure of 762 mm of mercury which was with in the range of general requirements of ISI test code of tractors. The BHP was calculated as detailed below and the value was noted.

$$BHP = (2\pi N_1 T_1 / 4500) \times 0.746$$

where,

BHP = Engine brake horse power, kw

N_1 = Crank shaft speed, rpm

T_1 = $(F_1 - F_2)$ r = Torque produced by the engine, kg m

F_1 = Load registered in fixed spring balance, kg

F_2 = Load registered in movable spring balance, kg

r = Radius of fly wheel, m

Part load

The power performance of a power tiller at part load is an important consideration for good power and machinery management. Potential fuel economy was demonstrated

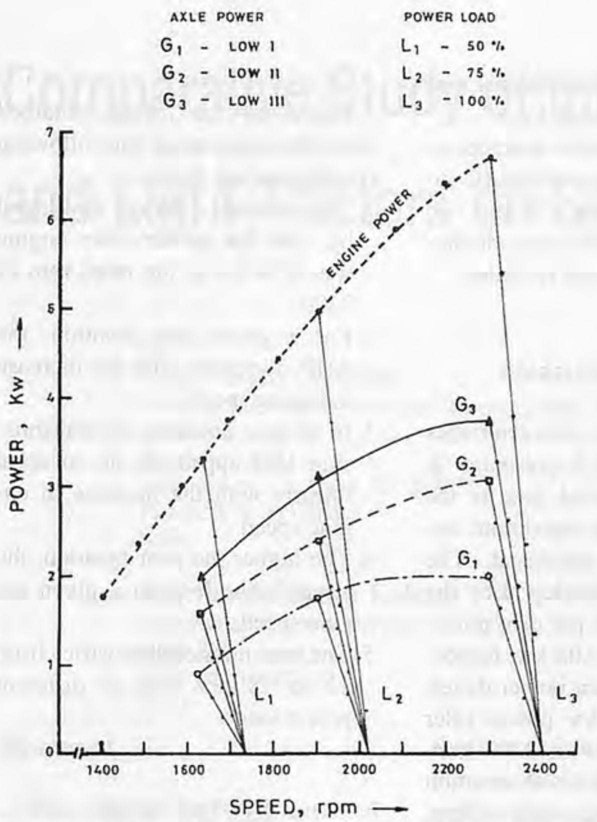

Fig. 3 Engine and axle power of power tiller at low gear positionth.

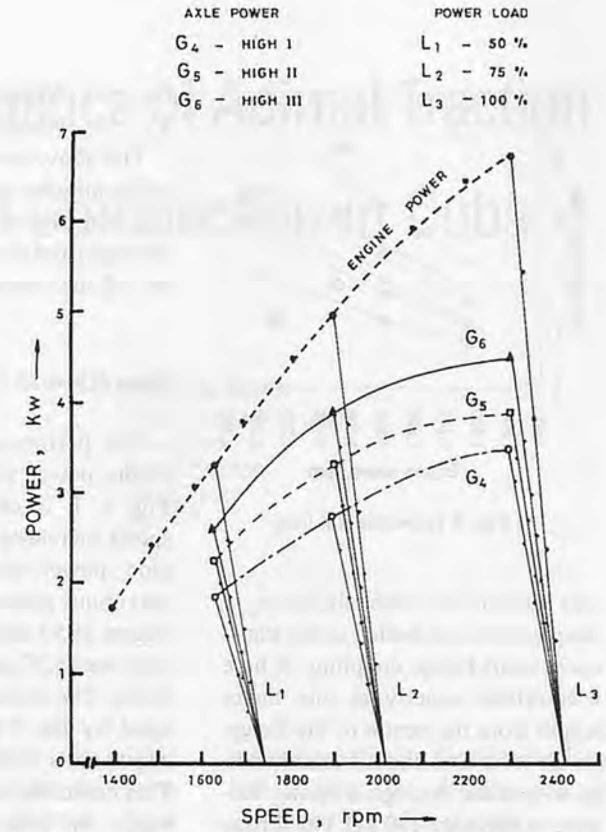

Fig. 4 Engine and axle power of power tiller at high gear position.

by considering partial load on tractors (Rickets, 1961). Different power load can be obtained by use of the governor setting at various levels of maximum power produced by the engine. Three levels of power load were selected for the study based on the maximum power produced by the power tiller: 100, 75 and 50 per cent of maximum power.

For the desired three levels of power load, the corresponding throttle positions were marked on an aluminium disc and was fixed near the governor of the engine. The maximum idle speed of the engine for the corresponding three levels of 100, 75 and 50 per cent of power was also determined. The part load performance of the engine at 50 and 75 per cent of maximum power was measured by repeating the above procedure and all the observations were recorded.

Axle Horse Power (AHP)

For measuring the power available at the wheel axle, a test rig was fabricated to mount the torque transducer and breaking device, **Fig. 2**. The torque transducer was connected to a load indicator for the display of the sensed torque at various speeds.

Development of test rig

A test rig for mounting the torque transducer and the braking device was fabricated. It consists of a bed for mounting the torque transducer between the wheel axle of the power tiller and the driven member which is a brake drum. The bed consists of a rectangular frame of size 710 × 250 mm made of 50 × 6 mm m.s. 'L' angles. It is supported on all the sides with 'L' angle of size 25 × 6 mm.The torque transducer was attached to the wheel axle hub at one end and the other end was attached to the brake drum shaft with the help of flange couplings. The braking device consists of a brake drum of tractor, mounted on the bed with one end of the brake drum shaft resting on bearing with housing and the other end connected to transducer shaft flange coupling. The power tiller wheel axle, transducer shaft and brake drum shaft were aligned perfectly. To make the test rig as an integral unit of the power tiller, the wheel test rig frame provided on the right side was rigidly connected to the engine test rig frame. The wheel axle was loaded by applying braking load through the brake drum with the help of the brake cam shaft. For gradual and uniform loading and unloading, a 20 mm mild steel square rod was attached to the brake pedal shaft. One end of the rod was fixed to the brake pedal shaft and the other end was attached to a crane with a helical spring of size 19 mm outer diameter and 3 mm wire diameter. The square rod was also connected to the engine test rig frame with a spring for smooth operation and release of brake shoes during unloading.

Calibration of torque transducer

Before making observation with the torque transducer, calibration

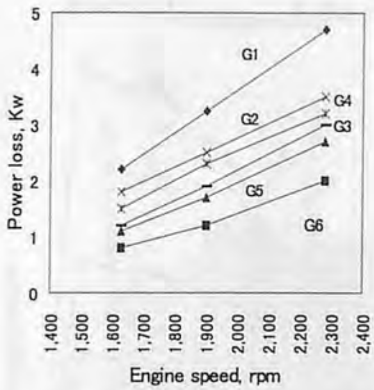

Fig. 5 Transmission loss.

was carried out statically using a simple lever arm bolted to the transducer shaft flange coupling. A hole was drilled exactly at one meter length from the centre of the flange on the lever arm and it was connected to a crane through a spring balance of capacity 100 kg. The torque transducer was calibrated by applying a known torque ranging from 5 kg m to 100 kg m at an increment of 5 kg m and the corresponding readings on the digital load indicator were noted. The calibration chart is given in **Fig. 3**.

Measurement of axle horse power

The power tiller engine was run at the maximum idle speed corresponding to 50 per cent power load. After selecting the gear position low I, when the engine was running at the maximum idle speed, the braking load was applied gradually and the power tiller was allowed to run for about 5 minutes to bring it to steady speed. The digital load indicator reading and the corresponding wheel axle speed were recorded. This process was repeated until a further increase in torque applied, reduced the engine speed to such an extent that the engine appears to be stalled. The horse power available at the wheel axle was calculated by using the following expression.

$AHP = (2\pi N_2 T_2 / 4500) \times 0.746$,

where,

AHP = Axle horse power, kw

N_2 = Wheel axle revolutions, rpm

T_2 = Wheel torque, kg m

The above procedure was repeated for all other gear positions (G_2 to G_6) and power loads (75 and 100 per cent) and the AHP was calculated. All the values were recorded.

Results and Discussion

The performance characteristics of the power tiller is presented in **Fig. 3**. It is observed that as the speed increased the maximum engine power also increased. The maximum power developed by the engine at 50 and 75 per cent power load was 3.37 and 5.02 kw, respectively. The maximum power developed by the 7.46 kw power tiller engine was 6.74 kw at 2300 rpm. This coincided with the observation made by other scientists (Garg, 1965 and Verma, 1967). The AHP at the selected power loads for low gear positions (G_1 to G_3) and high gear positions (G_4 to G_6) are shown in **Figs. 3** and **4** respectively. It is observed that for a given gear position, as the engine speed is increased the axle horse power developed was high.

The transmission loss of power from the engine to the axle of the power tiller is computed by subtracting the wheel axle power from the corresponding engine power. The transmission loss at different gear position is presented in **Fig. 5**. It is inferred that in all gear positions, the transmission loss appears to increase linearly with the increase in engine speed. It is also observed that higher the gear position, lower is the loss for a given engine speed. The transmission loss varied from 33 to 70 per cent, 25 to 65 per cent and 22 to 67 per cent of the engine power at 100 per cent, 75 per cent and 50 per cent power loads respectively.

Conclusions

Based on the results obtained from the experiment, the following conclusions are drawn:

1. The maximum power developed by 7.46 kw power tiller engine was 6.74 kw at the rated rpm of 2300.
2. For a given gear position, the AHP increases with an increase in engine speed.
3. In all gear position, the transmission loss appear to an increase linearly with the increase in engine speed.
4. The higher the gear position, the lower is the loss for a given engine speed.
5. The transmission loss varies from 22 to 70 per cent at different power loads.

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Comparative Study of Influence of Animal Traction and Light Tractors on Soil Compaction in Cuba

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Abstract

The aim of the work reported here was to determine the degree and intensity of soil compaction produced by work animals and light tractors, both commonly used in Cuba for soil tillage.

The following parameters were determined: animal weight; draft force produced; front and rear hoof area; number of steps; and area of soil compacted during different tillage operations.

The tractors used were YUMZ-6M and MTZ-80 of Soviet Union manufacture. Both are wheeled tractors and have engine powers of 44.1 and 55.8 kW, respectively. They are the most widely used models in Cuban agriculture and their effect on soil compaction was determined for the most frequently affected operations.

The results show that for traditional draft animal, powered soil preparation (plowing twice and four passes with a spike-toothed harrow) a pair of oxen will compact 5699 m^2 ha^{-1} (or 57%) with a pressure of 281.44 kPa. The YUMZ-6M and MTZ-80 tractors equipped with a three disc plow, effecting double plowing (a second pass at 90 to the first), and four passes with a disc harrow resulted in the following compaction per hectare: front wheels 15922 m^2; rear wheels 31984 m^2. The total area of soil compacted by the tractors was 47976 m^2 ha^{-1}, or 480% with a pressure under the front tyres of 214.5 kPa and 102 kPa under the rear tyres.

Introduction

Soil compaction can result in loss of available soil fertility due to the effect on vital soil factors such as movement of air and water, moisture content, porosity and the general capacity of the soil to produce crops. Inns and Kilgour (1978) estimated that, for healthy root growth, a soil porosity of 50% is needed with half the pore space filled with water leaving an air space of 25%. As a general rule, plant growth and crop yield will suffer appreciably if the air space is reduced below 10% to 15%.

Espinosa (1967) and Sielka Lal and Eliodoro (1986) explained that when soil is compacted by the passage of animals, tractors, machinery and implements, the macro-and micro-pores are reduced with the resulting difficulty for microbiological activity, the development of soil fauna and of crop roots.

It could be thought that the problem would be resolved by soil pulverizing machinery to restore the normal bulk density and so re-establish the retention and circulation of gases and water. Nevertheless, although there is usually a positive crop yield response when damaged soil is pulverised, this does not usually achieve levels equal to those achievable with originally non-compacted soils. With each successive year of tillage on a given soil, the deleterious effects of compaction tend to be cumulative.

In the specific case of sugarcane in Cuba there is a high incidence of machinery traffic from soil preparation through weed control cultivation, fertilizer application, harvest and transport to the mill. This can produce soil compaction to such an extent that the number of cuts can be reduced five-fold or more in permanent plantations. Martín *et al.* (1994) investigated the levels of compaction achieved in a range of soils under cane production as a result of field work. They confirmed that the level of compaction increases with an increase in the number of

Table 1. Soil Pressures Exerted by Different Equipment in Cuba

Equipment	Weight, kN	Traction	Contact Area, m^2	Soil Pressure, kPa
Cane combine KTP-1	48	Pneumatic tired wheels	0.24	200.0
Cane trailer KTC2 (empty)	35	Pneumatic tired wheel	0.30	116.7
Cane trailer KTC2 (loaded)	108	Pneumatic tired wheel	0.30	360.0
Soil compacting roller	-	roller	-	294-392

Note: *From Sielka and Eliodoro, 1986.*

Table 2. Soil Penetration Resistance at Different Depths before and after Soil Loading

Moisture content %	Mean penetrometer resistance at three depths, kPa 7 cm	25 cm	35 cm	Soil loading
27.4	3080	2844	2775	before
	3736	3089	2844	after

Source: Sielka Lal and Eliodoro, 1986.

machinery passes (but it should be noted that the greatest damage is done by the first pass of the wheels or hooves, with up to 70% to 90% of soil compaction occurring with the first of multiple passes [Ashburner y Sims, 1984]). Sielka Lal and Eliodoro (1986) compared the compaction of soil as a result of wheelings from a cane combine and cane trailer compared with a soil compacting roller (**Table 1**).

The distance between sugar cane rows in Cuba is 1.4 - 1.6 m so that the KTP-1 combine (which is a single row harvester) will pass 63-72 times per hectare, compacting 5670 to 6480 m^2 at a pressure of approximately 200 kPa. The trailer employed travels alongside the combine to form a synchronised unit. Once filled, the trailer returns between the rows of cut cane to the road and the mill. This means that the trailer will have passed twice between the rows, compacting a total of 12 600 m^2 with a pressure of between 116 and 360 kPa. As yields increase, so does the soil damage. With a yield of 50 t ha^{-1} the combination will cover 900 - 1000 m to fill the 6-7 tonne capacity trailer, reduced to half that distance with a yield of 100 t ha^{-1}.

The traffic of all these machines and equipment is concentrated in the inter-row space and, when the effect of the tractor towing the trailer is added (Table 4) it can be deduced that the combined effect can be considerably greater than that produced by the rollers used for highway compaction.

A further negative aspect associated with soil compaction is the resistance to penetration of soil acting implements. This can result in an increase in the energy requirement for the job and an increase in the number of passes required to achieve the soil manipulation necessary to ensure healthy crop development. **Table 2** gives an indication of the increase in penetration resistance as a result of soil loading.

Soil compaction has a direct effect on soil erosion as infiltration is reduced and run-off increased. The principal measures to reduce compaction, and so reduce the risk of hydraulic erosion, are to reduce ground pressure by reducing vehicle weight and tyre pressure and increasing ground contact area reducing the number of machinery passes; and maintaining soil cover (Kamprath *et al.*, 1979).

Given the potential overuse of heavy farm machinery in Cuban agriculture and the consequent risks of soil compaction and erosion, the present study proposed to measure the degree and intensity of compaction with wheels and the hooves of draft animals during soil preparation.

Materials and Methods

Location, Soil Characteristics and Animal Breeds Used

The work (plowing and "crossing" with the same plow) was done using two pairs of oxen (*yuntas*), one Holstein and the other Zebu on the compacted red ferralitic soils of the *Instituto Superior de Ciencias Agropecuarias de la Habana,* in Havana Province. The work was repeated with two further *yuntas* (one *Criollo* and the other cross-bred) on brown carbonaceous soils on the farm of Livan Socarrás, Bahía Honda, Pinar del Río Province.

Animal Parameters Evaluated

The following characteristics were noted: breed; age; body length (withers to tail root); thoracic circumference; height; hoof circumference; length of pace; number of paces per 100 m; and number of paces per hectare.

Using these data the following calculations were made:

$P_c = P_t^2 \times L \times 92.46$ m
$V = s/T$ m s^{-1}
Where:
P_c = ox body weight, kg
P_t = thoracic circumference, m
L =Body length (withers - tail root), m
V = forward speed, m s^{-1}
s = distance covered, m
T = time taken, s

Draft Force

Draft force was estimated by assuming that an ox can sustain a force equivalent to 10% of its body weight throughout its working day.

Determination of Support Area of Animals

Hoof-prints were taken by placing the hoofs on paper placed beneath them, the area was then measured with a planimeter. Prints were made of both front and rear hooves and the support area determined in all cases. When walking, oxen support their weight on one front hoof and the rear hoof on the opposite side, the front hooves support 55-60% of the animal's weight whilst the rear hooves exert 55-60% of the draft force. The support surface of the walking animal is the sum of the area of one front and one rear hoof; when standing the body weight is sustained by all four hooves and the support area is the sum of all hoof areas.

Pace Length, L_p

The distance between the centers of the prints of the front and rear hooves was measured. Measurements were taken for both right and left feet and for each working animal. During hard work when the draft requirement is increased, the pace length is shortened; in addition, with increasing hours of work the pace length is also reduced. Consequently, information was recorded on the type of work, pulling resistance and number of hours worked.

Number of Hoof-prints per 100 m of Work, N_{p100}

The number of steps per 100 m is calculated by first calculating the mean pace length of the *yunta*. As each animal has two rows of hoof-prints, the number of prints in a 100 m run is four times this value.

$N_{p100} = 100/L_p \times 4$ m

Where:

N_{p100} = number of hoof-prints per 100 m

L_p = mean pace length, m

Number of Hoof-prints per hectare, P_{ha}

Given that a hectare is 10000 m^2, and N_{p100} is the number of hoof-prints per 100 m, the number of passes per hectare, N_v, is given by:

$N_v = 10000/100 \div B$

$= 100/B$

$P_{ha} = N_{p100} \times N_v$

Where:

N_v = Number of passes per hectare

B = Working width of the implement, m

P_{ha} = number of hoof-prints per hectare

N_{p100} = number of hoof-prints per 100 m

Area of Hoof-prints per hectare, S_{ha}

$S_{ha} = P_{ha} \times A_c$ m^2

Where:

S_{ha}= area trodden per hectare m_2

P_{ha}= number of hoof-prints per hectare

A_c= mean hoof contact area m^2

Pressure Exerted on Soil by Working Animals

The pressure exerted on the soil as a result of working animals is due to two factors: the weight of the animal and the vertical component of the pulling force exerted (**Figure 1**).

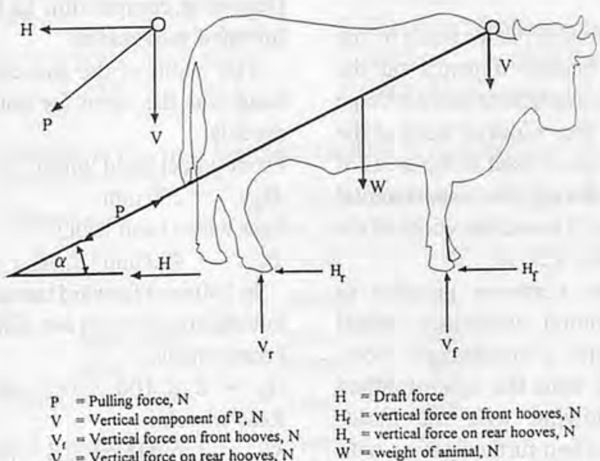

Fig. 1 Forces acting on a working animal.

Although with the animal at rest the front hooves support the greater weight, as the pulling force increases as a result of increased external loading, weight is transferred progressively to the hindquarters.

As the animal moves forward its weight is borne on one front hoof and the opposite rear hoof. The animal's weight is supported by the contact area of these two hooves and the ground pressure is equivalent to this weight plus the vertical component, V, of the pulling force P. Assuming that draft bovines exert a sustainable pulling force of approximately 10% of their body weight, V is $0.1 \times W \sin\alpha$. The ground pressure exerted by an animal is P_s, where:

$P_s = W + 0.1 W \sin\alpha / (A_d + A_t)$

Where:

P_s= Ground pressure exerted by the animal, kPa

W= Weight of animal, N

A_d= Contact area of front hoof, m^2

A_t= Contact area of rear hoof, m^2

Methodology of Evaluation of Soil Pressure with Tractors

Calculation of Weight Distribution of Tractor

The tractors used in the study were models YUMZ-6M and MTZ-80 of 44.1 and 58.8 kW respectively, and approximately 14 kN drawbar force. These tractors were manufactured in the ex-USSR and are the most commonly used in Cuban agriculture.

For two-wheel drive tractors the weight of the tractor acting on the driving wheels directly influences the capacity for traction and is usually approximately 60% W. Single-axle traction will develop approximately 60% of the power available at the pto. In the case of four-wheel drive tractors the whole weight of the tractor is available to produce traction and the power available at the tractor drawbar is approximately 65-

70% of the available pto power.

Two-wheel drive tractors

$W_d = W \times 30 / 100$ N

$W_t = W \times 70 / 100$ N

Where:

W_d = Tractor weight acting on the front wheels, N

W_t = Tractor weight on rear wheels, N

W = Tractor weight, N

Four-wheel drive tractors

$W_d = W \times 40 / 100$ N

$W_t = W \times 60 / 100$ N

Area of Ground Pressure in 100 m of Forward Travel

$G_d = 2L_c \times B_d$ m^2

$G_t = 2L_c \times B_t$ m^2

Where:

G_d = Area of front wheelings

G_t = Area of rear wheelings

L_c = Length of run (100 m in this case)

B_d = Width of front tire contact area

B_t = Width of rear tire contact area

Number of Passes per hectare, N_v

$N_v = 100 / B$ m

Where:

B = Implement width, m

Area of Tractor Wheelings per hectare, G_{Tha}

$G_{dha} = G_d \times N_v$ m^2 ha^{-1}

$G_{tha} = G_t \times N_v$ m^2 ha^{-1}

$G_{tha} = G_{dha} + G_{tha}$ m^2 ha^{-1}

Where:

G_{dha} = Area of front tractor wheelings, m^2 ha^{-1}

G_{tha} = Area of rear tractor wheelings, m^2 ha^{-1}

G_{tha} = Total area of tractor wheelings, m^2 ha^{-1}

Soil Pressure

$P_{rd} = P_d / S_d$ m^2

$P_{rt} = P_t / S_t$ m^2

Where:

P_{rd} = Ground pressure due to front wheels, kPa

P_{rt} = Ground pressure due to rear wheels, kPa

S_d = Front wheel contact area, m^2

S_t = Rear wheel contact area, m^2

Results and Discussion

Draft Animals

Animal weight and hoof contact area

The draft oxen were selected according to the average sizes and weights in the provinces of Havana and Pinar del Río (604 kg). The animals used did not differ significantly from this mean. Also, the hoof contact area was similar for all the animals studied and for both front and rear hooves. The mean value was 0.011 m^2.

Number of steps per 100m, N_{p100}

The right-hand side animal of the *yunta* made a mean of 301 steps per 100 m whilst plowing and cross-plowing; the mean for the left-hand side animal was 271 steps over the same distance. The difference was significant for both *yuntas*.

The explanation could be that the right hand side animal walks in the furrow bottom at a lower level than the left-hand ox. The looser soil conditions and the increased load on the head result in shorter steps, and, therefore, an increased number to maintain the speed of work.

Number of hoof-prints per hectare, P_{ha}

The number of passes made by the *yunta* per hectare depends on the width of the implement and the shape of the plot. The width of work of the moldboard plow used in the areas of study and during the experimental work was 0.32 m and the width of the spike harrow, 1.50 m.

The most common practice in soil preparation comprises: initial plowing with a moldboard plow; two passes with the spike-toothed harrow; plowing with the moldboard plow; two further passes with the harrow. In some cases, depending on the crop to be planted, a further plowing and spike-harrowing may be given.

The number of passes for plowing and "crossing" (N_{v1} ha^{-1}) was 625; for the four passes with the harrow the number of passes (N_{v2} ha^{-1}) was 226. The number of hoof-prints for this practice was: P_{ha1} = 357 638 for plowing; P_{ha2} = 152 592 for harrowing; P_{haT} = 510 230 for the entire practice of soil preparation.

Area trodden per hectare, S_{Tha}

The trodden area per hectare is the sum of the number of hoof-prints per hectare (P_{haT}) and the mean hoof contact area (A): 569 923 m^2. This means that the *yunta* treads on 57% of the area during conventional soil preparation.

Soil pressure, P_s

$P_s = W + 0.1W \sin\alpha / 2A$

Ground pressure averaged 281.55 kPa which is considered high and is one of the reasons why excessive soil manipulation should be avoided, especially under conditions of high soil moisture content.

Effect of Light-Wheeled Tractors on Soil Compaction

Tractor weight over front and rear wheels

The weight distributions of the two tractors evaluated is given in **Table 3**.

Degree of compaction in 100 m of forward movement

The width of the soil compacted band was the same for both tractor models:

Front wheel band width:

B_d = 200 mm

Rear wheel band width:

B_t = 400 mm

In 100 m of forward travel, the following areas of soil are compacted:

Front wheels:

$G_d = 2 \times 100 \times 0.2 = 40$ m^2

Rear wheels:

$G_t = 2 \times 100 \times 0.4 = 80$ m^2

Number of passes per hectare

The number of passes per hectare depends on field shape and implement width. Assuming a plot of 100 m × 100 m, the ADI-3 disc plow has a working width of 0.75 m and the number of passes per hectare, N_{va} = 133.3. The disk harrow has a working width of 3 m and requires N_{vg} = 33.3 passes per hectare.

The traditional practice of preparation of clay soils consists of the following operations:

1. Plowing with the ADI-3 disk plow
2. Two passes with a 909 kg disk harrow
3. Plowing at 90E with the ADI-3
4. Two further passes with the disk harrow, or one pass with the harrow and one with a ridger

The number of passes per hectare for this menu of tillage is:

N_{va} =268 passes per hectare for the two plowings

N_{vg}= 133 passes per hectare for the four diskings

This gives a total of 401 passes per hectare for the total operation (N_{vT}).

Table 3. Weight Distribution of Two Test Tractors, kN

Weight Distribution	Tractor YUMZ-6M	MTZ-80
On front axle	11.46	9.72
On rear axle	26.75	22.46
Total	38.21	32.1

Table 4. Ground Pressure Under Front and Rear Tires of Two Test Tractors, kPa

Ground Pressure under:	Tractor YUMZ-6M	MTZ-80
Front tires	253.9	203.8
Rear tires	102.0	85.3

Degree of Compaction per hectare

The area of soil compacted by the two wheelings is:

Front wheels:

G_{dha}= 15992 m^2 ha^{-1}

Rear wheels:

G_{tha} = 31984 m^2 ha^{-1}

Total compacted area:

G_T =47976 m^2 ha^{-1}

This means that the area compacted by the tractor wheelings using this soil preparation practice amounts to 480% (approximately) of the area cultivated.

Ground pressure due to the tractor without implements

Table 4 shows the ground pressures due to the unloaded tractors.

Comparing the ground pressures exerted by the tractors' front wheels with that of the *yunta* it can seen that the differences are not great. However, the rear tractor wheels exert a far lower pressure due to the wider tires and greater ground contact area. **Table 5** summarizes the results.

Conclusions

The results achieved lead to the following conclusions:

1. The tractors used in the study compact an area of soil 8.4 times greater than that compacted by the *yunta* whilst engaged in the conventional soil preparation practices.
2. The *yunta* creates greater ground pressure, 1.1 times that of tractor front wheels and 3.3 times that produced by the tractor rear wheels. However, the cumulative effect of the *yunta* is much less, as less area of soil surface is compacted.
3. An increase in implement width used with both animals and tractors for soil preparation would reduce the soil compaction produced. In addition the use of reduced tillage techniques and conservation tillage will reduce the level of compaction and its negative effects on soil fertility.

Table 5. Summary of the Areas Compacted and the Ground Pressures Produced by Tractors and Draft Oxen

Power Source		Area Compacted per hectare M^2 Ha^{-1}	(%)	Ground Pressure, kPa
Draft oxen (*yunta*)		5699	57	281
Tractor Yumz-6M, 44.1 kw	Front wheels	15992	160	254
	Rear wheels	31984	320	102
	Total	47976	480	
Tractor MTZ-80, 58 kw	Front wheels	15992	160	204
	Rear wheels	31984	320	85
	Total	47976	480	

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Effects of Tillage System and Traffic on Soil Properties

by

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Abstract

The effects of seven different tillage systems and subsequent wheel traffic (22 kN axle load) on the physical and mechanical properties of soil were examined on a typical Central Anatolian clay loam soil. Both tillage and field traffic influenced the physical and mechanical properties of the soil significantly except the insignificant effect of traffic on moisture content. The compaction status of the soil generally increased proportionally with the increased loosening of soil during tillage. Wheel traffic effects on chisel systems were minimal when compared with the same effects on conventionally tilled soil. Soil stress occurred during wheel passage was highly correlated with the soil strength confirming that soil strength is more sensitive to wheel traffic effects than bulk density. Also, both tillage and traffic induced differences in aggregate size.

Introduction

Soil compaction is generally considered as the increase in soil bulk density resulting from loads applied by field machinery. When a soil is compacted, there is a re-arrangement of soil particles and a reduction in pore volume, particularly the large pores (Godwin, 1990). The primary impact of compactive forces on soil is to change soil strength, bulk density, porosity and pore sizes (Anonymous, 1992). These changes influence the movement of water and air throughout the soil and plant growth.

The susceptibility of soil to compaction depends upon soil conditions at the time of loading, the load applied, the manner of loading and the number of loadings (Erbach, op. cit.). Soil contact pressure influences surface soil compaction whilst total axle load subsoil compaction (Anonymous, 1992). Generally, compaction depth increases proportionally with the increase of the vertical force from the tractor at the same moisture content (Erbach op. cit.). Although changes from crop to crop, 0.9 - 1.5 MPa soil strength restricts root growth. Changes in bulk density, cone index and surface elevation caused by traffic were found to depend upon tillage systems (Erbach, op. cit.).

Bauder et al., (1985), analysed the effects of wheel traffic during crop production (tillage, sowing, spraying and harvesting) after four different tillage systems (chisel, moldboard plough, no-till and ridge-till) and found that tillage system had no effect on porosity and gravimetric water content while both tillage and traffic had significant effects on bulk density and penetration resistance down to a depth of 22 cm.

Soil stress during field traffic is the leading cause for the increment in soil bulk density and strength after compaction. Gruber and Tebrügge (1990), used a ground pressure measurement system for evaluating the compaction stress of conventional, reduced and no-till systems. They stated that, the highest ground pressure and tire sinkage depth were obtained in conventional tillage systems and susceptibility of subsoil compaction is much lower for reduced tillage systems due to the even distribution of wheel loads. Wheel traffic generally results in increased bulk density and soil strength, nevertheless soil strength was considered as a more sensitive indicator of wheel traffic effects than bulk density (Hill and Montalvo, 1990; Bicki and Siemens, 1991). No-tilled soil had a greater soil strength than conventionally tilled soil, however, wheel traffic had a smaller effect on no-tilled soil than conventionally tilled (Hill and Montalvo, op. cit.). When a no-tilled soil becomes compacted, it has sufficient soil strength to carry traffic without further compaction. It has also been reported that when a soil has developed sufficient load-carrying capacity, it can resist further changes in bulk density or total porosity. In such a soil, wheel traffic may change the pore distribution slightly by elastic deformation (Anonymous, 1992). Ngunjiri and Siemens (1995), investigated the effects of different wheel traffic patterns during ploughing and reported that wheel traffic was found to increase soil bulk density and cone index to a depth of 30 cm with the most important impacts in the 0-15 cm depth. A study attempted to determine soil cone index variability under no-till, conventional and reduced tillage systems (Manor et al., 1991), indicated that much of the variability in cone index under field conditions is caused by tillage and traffic.

Erbach et al. (1992), reported that plough loosens soil more than chisel, however, after some natural effects and secondary tillage recompaction of soil was greater in plough systems than chisel.

Çarman (1996), studied the effects of field traffic patterns causing from three types of tillage on the variation of soil bulk density, penetration resistance and tire sinkage depth. Tillage systems using; conventional tillage (ploughing to a depth 22 cm followed by two disking to a depth of 12 cm), rotary tilling twice to 15 cm depth and no-till. In this study, the tire sinkage depth and increments in bulk density and penetration resistance due to field traffic were the highest in conventionally tilled plots and the least in no-tilled.

An important step towards understanding and controlling compaction is the ability to predict compaction. Prediction of soil compaction requires a mathematical description of the relating forces that causes compaction to the resulting strain or changes in bulk density and soil strength. Thus, there is a great need about the compaction susceptibility knowledge and data for the effects of different kinds of tillage systems on soil properties.

The objective of the work reported here was to evaluate the susceptibility of compaction for different tillage systems, including conventional, reduced and stubble mulch tillage. Tillage systems were compared with respect to the physical conditions and strength of soil as well as rut depth and ground pressure. Beyond this, an actual dependence among soil strength, bulk density and ground pressure in the upper 200 mm of the soil assessed.

Materials and Methods

Site, Location and Soil

Experiments were carried out at the research farm of Ankara University, 40 cm south of Ankara, Turkey where the average rainfall was 376 mm. The site altitude is 1050 m and the soil is an imperfectly drained clay loam soil of the Central Anatolian series with 21% clay, 58% silt and 21% sand. Soil pH averaged 7.83 and organic matter 1.13%. The experiment was arranged in three blocks each consisting of seven tillage treatments. The plots were 25 m wide and 75 m long. There were wheat stubble on plots.

Field Operations

A 65 kW, two-wheel drive tractor weighed to 32.4 kN was used for both tillage and traffic treatments. The rear axle was fitted with 13.6/12-36 bias-ply single tires, inflated to 160 kPa while the front axle was fitted with 7.50-16 bias-ply single tires, inflated to 240 kPa.

A total of seven tillage systems were performed in this study as follows:

S1.Chisel + disc harrow (chiseling to a depth of 28 cm followed by two disking to a depth of 13 cm);
S2.Chisel + tooth harrow (chiseling to a depth of 28 cm followed by two harrowing to a depth of 10 cm);
S3.Chisel + cultivator-tooth harrow combination (chiseling to a depth of 28 cm followed by two cultivator-tooth harrow combination to a depth of 12 cm);
S4.Moldboard plough + tooth harrow (Ploughing to a depth of 22 cm followed by two disking to a depth of 13 cm);
S5.Moldboard plough + disc harrow (Ploughing to a depth of 22 cm followed by two harrowing to a depth of 10 cm);
S6.Rotary tiller with horizontal axis-rotary harrow combination to a depth of 13 cm; and
S7.Rotary tiller with vertical axis-rotary harrow combination to a depth of 13 cm.

Traffic treatments were performed twice, after tillage with 1.5 m/s forward speed and 22 kN rear axle load without considering any crop rows to be planted (Montalvo, 1990; Reeves et al., 1992).

Soil Characterization

Soil bulk density, moisture content, penetration resistance and shear strength were measured after tillage and wheel traffic treatments. All measurements related with soil physical properties and strength were performed in respect to row position rather than randomly within each plot. There were six transects in each plot of which measurement points for both wheel tracks and non-wheel tracks were selected (Manor et al. 1991). Some initial physical conditions of the soil is shown in **Table 1**.

Bulk density was measured on undisturbed soil cores from both wheel tracked and non-wheel tracked interrows of each plot to the depth of 0-30 cm. Each were sectioned into 5-cm depth increments. Soil moisture content was determined gravimetrically from bulk density samples. Total porosity was calculated from the saturation water content and bulk density assuming a

Table 1. Initial Conditions of the Experimental Plots

Property	Depth (cm)		
	0-10	10-20	20-30
Moisture content (%)	20.075	24.702	25.333
Dry bulk density (Mg/m^3)	0.960	1.097	1.162
Air porosity (%)	44.428	31.472	26.663
Total porosity (%)	63.714	58.571	56.085
Vane shear strength (kPa)	41.333	124.708	168.750
Penetration resistance (kPa)	1037	1575	2035

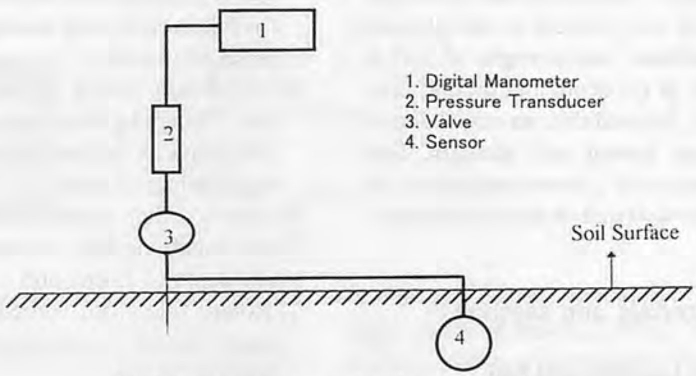

Fig. 1 Pressure gauge used in the experiments.

particle specific gravity of 2.65. Percentage air voids was calculated from the ratio of volume of air to the total volume of soil. All of these parameters were calculated after oven drying of cores at 105 °C.

The changes in soil strength resulting from tillage and traffic treatments were from measurements of shear vane and cone penetrometer. A vane borer with a 16x32 mm bladed vane and measurement range of 0-260 kPa was used for shear strength measurements. On each of six transects within a plot, vane shear strength was measured down to a depth of 30 cm at 5 cm depth increments. Penetration resistance was measured with a hand-operated recording type cone penetrometer which had a 30° steel cone of 1 cm^2 base area. The penetrometer was pushed vertically into the soil with a uniform speed down to a depth of 30 cm and penetration resistance was recorded at each 1 cm depth interval. Measurements were made on six transects of each plot.

Vertical stress in soil under wheel load was measured with a developed pressure gauge (Gruber and Tebrügge, op. cit.) as shown in **Fig.1**. The pressure remaining after wheel passage were considered. All vertical stress measurements were performed at 10 and 20 cm depths.

In each plot, sinkage of rear tire was measured after traffic treatments with a profile meter. The profile meter consisted of vertical metal rods sliding through a 100 cm long iron bar at a regular spacing of 2.5 cm. The bar was placed across the wheel tracks to measure the shape of the depression (Çarman op. cit.).

Five-kg soil was sampled from each plot and later sieved with a rotary sieve to determine size distribution of aggregates. The mean weight diameter was calculated as the sum of the product of aggregate diameters in each size and the fractional weight of aggregate in that size (Adam and Erbach 1992). The mean weight diameter was computed for each tillage treatment in wheel tracks and non-wheel tracks.

The effects of various tillage system and wheel traffic effects on bulk density, total porosity, moisture content, percentage of air voids, shear resistance, penetration resistance, soil stress, tire sinkage and mean aggregate weight diameter were evaluated by analysis of variance with tillage as the main effect, wheel traffic as the first split plot and depth as the second split plot. Comparison of mean values was accomplished using least significant differences (LSD) at $\alpha \leq 0.05$.

Results and Discussion

Table 2. Summary of ANOVA, Indicating the Effects of Tillage, Wheel Traffic (WT) and Soil Depth on Physical and Mechanical Properties of Soil

Item	Bulk density	Total porosity	Moisture content	Percent air voids	Shear strength	Penetration resistance	Soil stress	Sinkage depth	Mean weight diameter
Tillage	**	**	**	**	**	**	**	**	**
Depth	**	**	**	**	**	**	**	-	-
Tillage × depth	**	**	**	**	**	**	**	-	-
WT	**	**	ns	**	**	**	-	-	**
Tillage × WT	**	**	ns	**	ns	ns	-	-	*
Depth × WT	**	**	ns	**	**	**	-	-	-
Tillage × depth × WT	**	**	ns	**	ns	ns	-	-	-

** : significance at the 0.01 probability level.
* : significance at the 0.05 probability level.
ns : non-significant at the 0.05 probability level.
\- : not considered as in the analysis of variance.

Table 3. Mean Comparisons of Dry Bulk Density as Affected by Tillage and Traffic

Item	Depth (cm)	Dry bulk density (Mg/m^3) •						
		S1	S2	S3	S4	S5	S6	S7
After tillage	0-5	0.83 ab	0.84 a	0.80 cd	0.81 bc	0.81 bc	0.77 e	0.79 de
	10-15	0.98 a	0.99 a	0.97 ab	0.91 c	0.93 bc	0.95 b	0.95 b
	20-25	1.08 d	1.07 d	1.07 d	1.14 a	1.15 ab	1.16 bc	1.17 c
After traffic	0-5	0.93 a	0.93 a	0.92 a	0.92 a	0.92 a	0.93 a	0.92 a
	10-15	1.04 ab	1.05 a	1.04 ab	1.03 ab	1.04 ab	1.02 b	1.03 ab
	20-25	1.08 b	1.08 b	1.07 b	1.16 a	1.17 a	1.17 a	1.16 a

• Numbers in a row within a depth range followed by the same letter are not significantly different at 0.05 confidence level ($LSD_{0.5}$ = 0.0222 Mg/m^3).

Table 4. Mean Comparisons of Percentage Air Voids as Affected by Tillage and Traffic

Item	Depth (cm)	Air voids (%)•						
		S1	S2	S3	S4	S5	S6	S7
After tillage	0-5	56.4 e	55.5 e	57.9 d	58.4 cd	58.9 c	61.3 a	60.0 b
	10-15	42.7 de	41.9 e	43.4 d	49.0 a	47.0 b	45.9 c	45.7 c
	20-25	32.5 a	33.0 a	33.4 a	28.6 b	29.0 b	26.5 c	25.9 c
After traffic	0-5	51.3 cd	50.6 d	51.7 bc	52.8 a	53.2 a	53.1 a	52.7 ab
	10-15	39.4 de	38.4 f	39.0 ef	42.1 a	40.2 cd	41.5 ab	40.9 bc
	20-25	32.3 a	32.9 a	32.8 a	27.5 b	27.4 b	26.0 c	26.2 c

• Numbers in a row within a depth range followed by the same letter are not significantly different at 0.05 confidence level ($LSD_{0.5}$= 0.981 %)

Table 5. Mean Comparisons of Vane Shear Strength as Affected by Tillage and Traffic

Item	Depth (cm)	Vane shear strength (kPa)•						
		S1	S2	S3	S4	S5	S6	S7
After tillage	0-10	14.0 a	13.6 a	9.00 ab	7.6 ab	9.0 ab	4.6 b	6.5 b
	10-20	68.3 b	71.6 b	61.5 c	49.8 d	52.0 d	96.6 a	102.6 a
	20-30	142.8 b	143.1 b	140.8 b	175.1 a	177.1 a	172.8 a	178.1 a
After traffic	0-10	23.6 a	24.1 a	17.8 abc	18.8 abc	21.1 ab	13.5 c	15.3 bc
	10-20	85.6 b	89.5 b	78.5 c	71.5 c	72.1 c	108.6 a	110.3 a
	20-30	150.1 b	146.5 b	147.5 b	178.5 a	179.1 a	179.5 a	183.3 a

• Numbers in a row within a depth range followed by the same letter are not significantly different at 0.05 confidence level ($LSD_{0.5}$= 6.687 kPa).

Table 6. Mean Comparisons of Penetration Resistance as Affected by Tillage and Traffic

Item	Depth (cm)	Penetration resistance (kPa)•						
		S1	S2	S3	S4	S5	S6	S7
After tillage	0-10	563 a	555 a	453 b	341 c	381 bc	265 c	351 b
	10-20	1046 c	1023 c	900 d	775 e	826 de	1240 b	1343 a
	20-30	1851 b	1865 b	1835 b	2058 a	2035 a	2070 a	2090 a
After traffic	0-10	731 a	720 a	603 b	553 bc	581 b	470 c	546 bc
	10-20	1305 bc	1275 c	1160 d	1091 d	1131 d	1376 ab	1441 a
	20-30	1923 b	1891 b	1898 b	2095 a	2096 a	2120 a	2115 a

• Numbers in a row within a depth range followed by the same letter are not significantly different at 0.05 confidence level ($LSD_{0.5}$= 92.86 kPa).

A summary of the ANOVA indicating the effects of tillage, wheel traffic and depth on various physical and mechanical properties are shown in **Table 2**. Tillage, traffic and depth had significant effects ($p<0.01$) on soil bulk density, total porosity, percentage air voids, vane shear strength and penetration resistance. Wheel traffic areas were characterized by greater mean values of bulk density and soil strength.

There was not any significant effect of traffic on moisture content despite the significant effects of tillage and depth. Tillage had also significant effect on soil stress, tire sinkage depth and aggregate size ($p< 0.01$). Soil stress was significantly influenced by and aggregate size by traffic.

At 0-5 and 10-15 cm depths the lowest bulk density after tillage was obtained in S6 and S4, respectively, whereas the greatest in S2 (**Table 3**). For all systems the lowest bulk density was obtained at 0-5 depth ranging between 13.8-23.5% of the initial dry bulk density. At 10-15 depth, this variation was between 10.1-20.5%. After wheel traffic, dry bulk densities at all depths except 20-25 cm increased significantly. At 0-5 and 10-15 cm depths the highest increase in bulk density was occurred in the plots tilled with S6 and S4, respectively, whereas the lowest tilled with chisel systems S1 and S2.

When compared with the values obtained after tillage, the increments varied between 10.7-20.2% at 0-5 and 5.3-13.7% at 10-15 cm depth. These results support the idea that the impact of traffic on compaction is greater when soil is loosened very much during tillage. Inevitably, conventional tillage systems are more susceptive to compaction during secondary tillage and field traffic than chisel systems owing to more loosening during primary tillage. This argument is supported by Bauder et al., op. cit. and Erbach et al., op. cit.

The largest differences between the porosity of the plots (not shown) after tillage and after traffic were similar with bulk density. In contrast, the minimum changes were in S1 and S2 within tillage depth of 15 cm. At the depths below 20 cm, tillage did not have any significant effect except chisel systems as these systems loosen the soil down to 28 cm. Traffic treatments did not affect the porosity below 20 cm except in conventionally tilled plots.

The mean comparisons of moisture content exhibited significance for tillage system and depth (**Fig. 2**). This does not comply with the results of Bauder et al., op.cit and Hill and Montalvo, op.cit stating that tillage did not have significant effect on gravimetric water content. Tillage reduced the moisture content especially down to 15 cm (**Table 5**) ranging between 5.0-8.12% at 0-5 cm and 4.0-6.55% at 10-15 cm with the largest decrease in reduced systems and lowest in chisel systems.

The percentage of air voids indicated somewhat similar trend with total porosity (**Table 4**). The largest percent of air voids after tillage occurred in S6 and S4, respectively, at 0-5 and 10-15 cm and in chisel systems at 20-25 cm depth. All systems were affected from traffic at 0-15 cm depth of which conventional and reduced till were higher. As well as that, conventional tillage systems were influenced significantly from traffic at 20-25 cm in means of decreasing percentage air voids. Theses observations does not support an earlier hypothesis that tillage does not affect the air voids (Bauder et al., 1985).

Vane shear strength and cone penetrometer measurements exhibited approximately similar variations due to tillage and traffic (**Table 5, 6**). The lowest soil strength after tillage were obtained in S6 at 0-10 cm, in S4 at 10-20 and in chisel systems at 20-30 cm depth ranges. On the other hand, the lowest reductions were observed S1 and S7, respectively, for 0-10 and 10-20 cm depths. Increments of soil strength at 20-30 cm depth in the systems except S1, S2 and S3 indicated formation of hard pan below the tillage depth due to soil not having been deep tilled (Bicki and Siemens, op. cit.).

The impact of wheel traffic on the strength properties of differently tilled plots was significant down to 20 cm depth. S1 and S2 were affected quite less than the other systems by wheel traffic (Erbach et al., op. cit.). The increments of soil strength in S6 and S7 at 10-20 cm were restricted because of the resisting non-tilled soil layer below 15 cm. However during the traffic treatment, the tilled layer over 15 cm was easily compacted and increased the average soil strength of 10-20 cm. When compared with the values obtained after tillage, the highest increment was obtained in S4 and considering the depth ranges independently, highest increments were occurred in S6 and S4, respectively, for 0-10 cm and 10-20 cm depth range. The fact that differences in the soil strength at the 20-30 cm after traffic were not significant supports the hypothesis that most of the effect of tillage and wheel traffic on soil properties and mechanical impedance is confined to the 0-22 cm soil depth which is essentially the depth of commonly used tillage practices.

Comparisons of vertical soil stress occurred during the process of compaction at 10 and 20 cm depth are shown in Fig. 3. The highest values were in S6 at 10 cm whereas in S4 and S5 at 20 cm depth. In contrast, the lowest values were in S1 and S7, respectively, for 10 and 20 cm.

Considering these changes, regression analysis were performed about the dependence of vane shear strength, penetration resistance and bulk density values obtained after tillage upon values of after compaction and soil stress during compaction. The results are shown in **Table 7**. According to the indicated regression equations in both 10 and 20 cm depths, the dependencies of shear strength and penetration resistance to soil stress were more reliable than bulk density. This result supports the opinion of wheel traffic generally resulted in increased

Table 7. Regression Analysis for the Variation of Vane Shear Strength, Penetration Resistance and Bulk Density due to Soil Stress at 10 and 20 cm Depth

Parameter	Regression equation		R^2	Significance level
Shear strength	K= -24.109 + 1.994 KT + 0.666 B	(10 cm)	0.952	**
	K= 38.008 + 0.726 KT - 0.193 B	(20 cm)	0.976	**
Penetration resistance	PT= 145.011 + 0.913 PT + 1.978 B	(10 cm)	0.972	**
	PT= 719.284 + 0.552 PT - 2.627 B	(20 cm)	0.961	**
Dry bulk density	HT= 1.052 - 0.114 HT - 0.0009 B	(10 cm)	0.557	ns
	HT= 0.692 - 0.341 HT - 0.00145 B	(20 cm)	0.789	*

K:Shear strength after traffic (kPa), KT:Shear strength after tillage (kPa).
P:Penetration resistance after traffic (kPa), PT:Penetration resistance after tillage (kPa).
H:Dry bulk density after traffic (Mg/m^3), HT:Dry bulk density after tillage (Mg/m^3).
B: Vertical soil stress (kPa).
**: Significant at the 0.01 probability level (p<0.01).
ns: Non-significant at the 0.05 probability level (P>0.05).

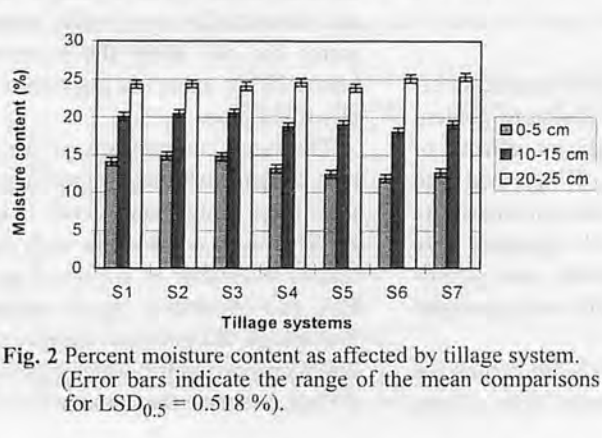

Fig. 2 Percent moisture content as affected by tillage system. (Error bars indicate the range of the mean comparisons for $LSD_{0.5}$ = 0.518 %).

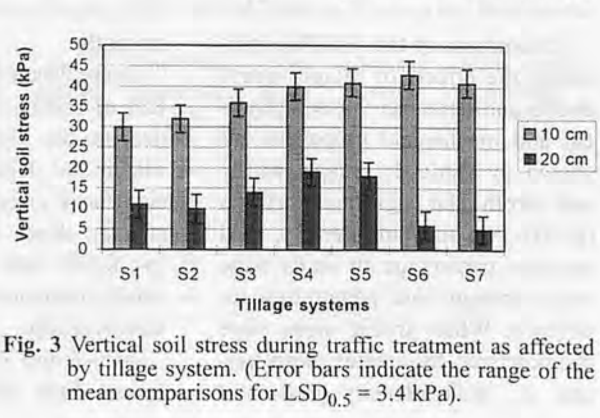

Fig. 3 Vertical soil stress during traffic treatment as affected by tillage system. (Error bars indicate the range of the mean comparisons for $LSD_{0.5}$ = 3.4 kPa).

bulk density and soil strength with soil strength being a more sensitive indicator of wheel traffic effects than bulk density (Hill and Montalvo, 1990).

Tire sinkage depth during compaction treatment also exhibited difference among differently tilled plots (**Fig. 4**). The average highest tire sinkage depth was occurred in S4 whereas smallest in S1 system. Tire sinkage depth was found to be closely related with soil conditions after tillage and became deeper when the strength and bulk density was relatively low down to the depth of approximately 20 cm.

The mean weight diameter of aggregates after tillage exhibited statistical difference for both tillage system and wheel tracks during tillage (**Fig 5**).

In non-wheel tracks the biggest and smallest diameters were found in S2 and S6, respectively. This trend was the same for wheel tracks. However, the difference between these two paths was maximum in S4 and S5 having the increase of average 27.1%. On the contrary, smallest increases between 15.4-17.4% were observed in reduced tillage systems. Use of cultivator instead of disc resulted in smaller aggregate size when S1 and S3 compared supporting the hypothesis of Adam and Erbach (1992).

Conclusions

Both tillage and field traffic influenced the physical and mechanical properties of the soil significantly except the insignificant effect of traffic on moisture content. As well as that, tillage system appeared to be an effective factor on the degree of compaction during field traffic. Wheel traffic effects on chisel systems were minimal when compared with the same effects on conventionally tilled soil. The upper 10 cm layer compaction was maximum in rotary tilled soil owing to highly pulverisation of the soil along tillage depth. Traffic negated the effects of tillage on the surface mostly down to 20 cm. The wheel traffic induced changes in soil physical properties seem to be larger and more prevalent for conventionally tilled soils. However, neither tillage system nor wheel traffic (22 kN rear axle load) adversely changed soil physical properties to such an extent that detrimental conditions for plant and growth should be encountered except some significant vane shear strength changes below the tillage depth of conventional and reduced till systems. Negative changes appeared in reduced tillage systems owing to their shallow depth of tillage. But, the use of chisel systems was found to be more reasonable to alleviate the effects of wheel traffic. Too much loosening of the soil had negative effects during field traffic. The compaction status of the soil generally increased proportionally with the increased loosening of soil during tillage. As the inevitable result of this, systems S1 and S2, resisted compactive forces more than other systems.

Soil stress occurred during wheel passage was highly correlated with the soil strength confirming that soil strength is more sensitive to wheel traffic effects than bulk density. Also, both tillage and traffic induced differences were exhibited in aggregate size of differently tilled plots.

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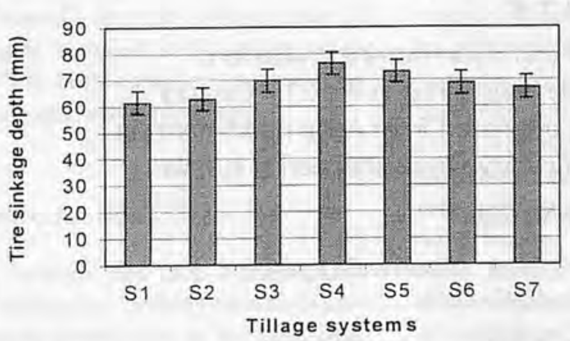

Fig. 4 Tire sinkage depth during traffic treatment as affected by tillage system. (Error bars indicate the range of the mean comparisons for $LSD_{0.5}$ = 4.304 mm).

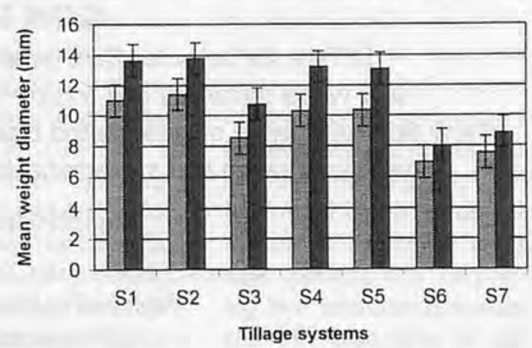

Fig. 5 Mean weight diameter as affected by wheel and non wheel tracks during traffic. (Error bars indicate the range of the mean comparisons for $LSD_{0.5}$ = 1.058 mm.)

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Effect of Pre-soaking of Sorghum Seed on The Performance of Two Animal-Drawn Planters

by

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Abstract

The effect of mechanised planting of soaked sorghum seeds was investigated. Two planters, the Safim (with horizontal rotating seed plate) and Sebele (with fixed plate and gravity drop) were used. Seeds were soaked for 6, 12 , 18 , 24, 30 and 36 h and thereafter put through a planting operation. The Safim planter recorded a reduction in the number of seeds were dropped as the soaking time increased. Completely damaged seeds reduced while partially damaged seeds were increased. The Sebele planter, on the other hand, showed a decrease in the number of seeds dropped with soaking time while both complete and partially damaged seeds increased.

Introduction

Drought and crop establishment are the most limiting factors to sorghum production in the semi-arid tropics. Whilst there is not much that could be done to control drought, there are many interventions which are possible to improve on crop establishment. Normal planting conditions in Botswana are uncertain, and often result in long delays to shoot emergence (5-14 days). This period is too long and establishment is most of the time compromised. Seeds which spend too much time in the soil risk damage to their emerging shoots because of the arid and hot environments. One of the interventions possible is seed soaking (priming). This concept has already been used in Botswana and elsewhere with good results. The concept stems from the idea that seeds do not need soil to germinate but water and warmth. If seeds are soaked in water at the optimum temperature, for some length of time, the germination process is activated and is usually complete in 24-72 h (Mortlock,1989). The operating principle is that the quicker the plumule reaches the soil surface, the more chances it has for emerging before a soil surface crust develops or the shoots are damaged. Harris and Tisdall (1996) found that the time taken by sorghum seeds to germinate at 30 °C, decreased as soaking time increased. Emergence was increased by as much as 23%, if seeds were soaked for only 6 h. Similar results were observed by Singh et el., (1984), when they found that soaking sorghum seeds for 12 h shorter flowering time by 4 days. Radford and Nielsen (1985) also observed that soaking sorghum seeds reduced time of emergence. Soaking time was found to peak or reach saturation point at 36 h irrespective of soaking temperature (Oguntunde and Adebawo, 1989).

Since soaking of seeds activates germination and is subsequently beneficial to emergence, the effect of mechanically planting these seeds is by and large unknown, more so that soaked seeds are likely to be vulnerable to damage even after mopping and drying. The study set out to investigate if seed soaking would render seeds vulnerable to mechanical damage and whether such planting will be possible using two common Botswana planters. The objectives of this study were, therefore:

a) To determine the amount of damage of soaked seeds attributable to each of the planters; and

b) To determine the optimum seed soaking time leading to the least seed damage for each of the planters.

Materials and Methods

The *Sorghum bicolor* seed (cv Segaolane), commercially supplied by the Seed Multiplication Unit (S.M.U) of Botswana was used during the experiment. It was first cleaned of debris and stalks by blowing air through the seed with a fan. Then it was passed through a 3.175mm seed screen to make sure that all the seed, which were used was of uniform size. All the seeds which fell through were discarded as 'small' seed. Seven packets of seed (800 g) were prepared for the experiment in perforated plastic bags. Six of the packets were then put into a water bath at 27 ℃ for 6, 12, 18, 24, 30 and 36 h, respectively, as treatments. The control was not soaked. On removal from the water bath and after the designated time frame, the seeds were wiped dry with a blotting paper. Each was then divided into two equal portions and put into the planters. The planters in question were the **Safim**, using the horizontal rotating seed plate (plate hole diameter 4.85mm) with a knocker and the **Sebele standard**, using the fixed orifice (hole diameter 6.15mm) gravity drop with an agitator as seed metering mechanisms. These two planters were chosen because they are recommended to farmers by the Ministry of Agriculture. The planters were jacked up to ensure smooth rotation of the drive wheels. Damage to the seeds was determined by rotating the planter drive wheels 10 times with 5 replications. Each time the number of completely damaged seeds (completely split), partially damaged seeds (bruised but likely to germinate) and undamaged seeds were counted for each of the planters. To identify completely damaged and partially damaged seeds visually, a Luxo light magnifier was used to good effect because it made seeds bigger and even a bruised coat was visible. The percent moisture content (dry basis) of the seeds at the time of testing was determined by oven drying a small sample. At the same time seed size was measured for thickness (t), width (W) and length (L) from a sample of 20 seeds using a micrometer screw gauge (error=0.01) (**Fig. 1**). The experiment was conducted at the Botswana College of Agriculture, Content Farm, Sebele.

An Analysis of Variance was performed on the data to determine if the average percentage of damaged seeds was any different between treatment means. Since the data was on the count of damaged seeds and expressed as a percentage, it was necessary to perform Arc Tine Transformation before analysis. The

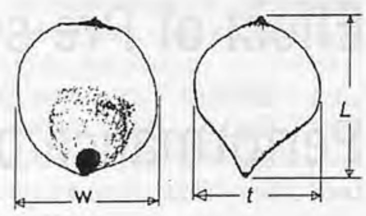

Fig.1 Illustration of a sorghum seed showing the principal dimensions.

Duncan's Multiple range test was done to separate the treatment means using SAS statistical package. (SAS, 1987)

Results and Discussion

As expected, soaked seeds absorbed water and their size generally increased. Length increased by a maximum of 20.5% with 24h soaking. Thickness increased by 12.0% from unsoaked to 36 h of soaking, whilst width increased by 14.2% to a high of 3.874mm after 24 h of soaking (**Fig. 2**).

Moisture content of seeds ready for planting showed a clear rise from 9.68% for unsoaked to a high of 59.74% for seeds soaked for 30 h (**Fig. 3**). Water uptake increased at a decreasing rate showing that initial uptake is more since the seed was dry. Also, after about 24 h of soaking, the seed seem to have reached its soaking capacity.

Safim Planter

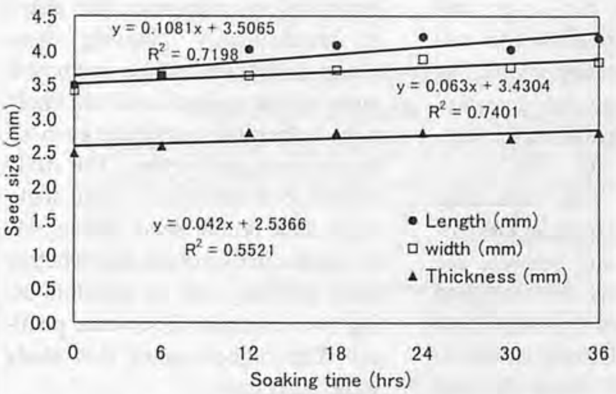

Fig. 2 Relationship between soaking time and seed size.

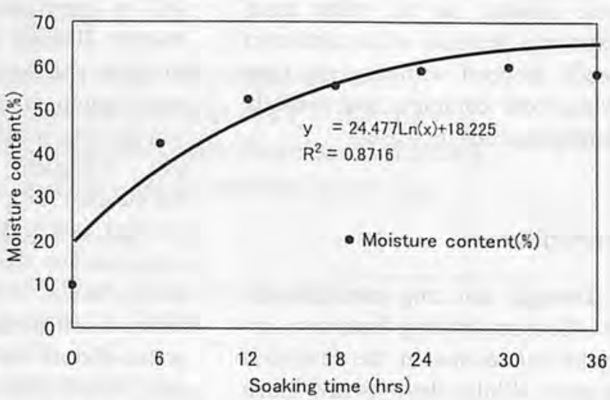

Fig. 3 Relationship between seed soaking time and moisture content.

Table 1. Comparisons of Means for Completely Damaged Seeds

Duncan Group	Average	Replications	Treatment
A	11.36	5	Control
B	3.71	5	6 h
B	3.56	5	30 h
B	1.70	5	12 h
B	1.63	5	24 h
B	1.63	5	18 h
B	0.00	5	36 h

Table 2. Comparison of Means for Partially Damaged Seeds

Duncan Group	Average	Replications	Treatment
A	16.68	5	36 h
A	16.26	5	30 h
A	16.24	5	12 h
AB	13.66	5	18 h
B	10.19	5	24 h
C	0.00	5	6 h
C	0.00	5	Control

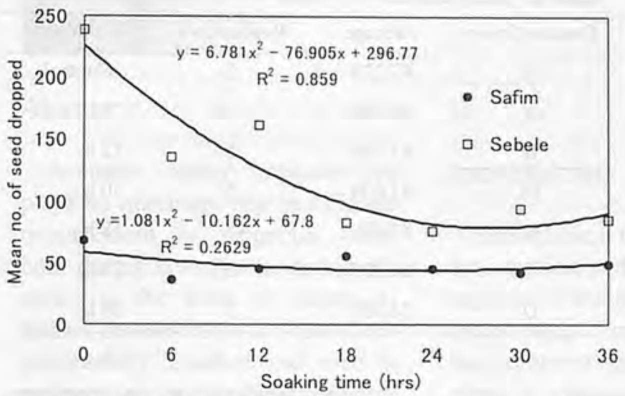

Fig. 4 Relationship between soaking time and number of seed dropped.

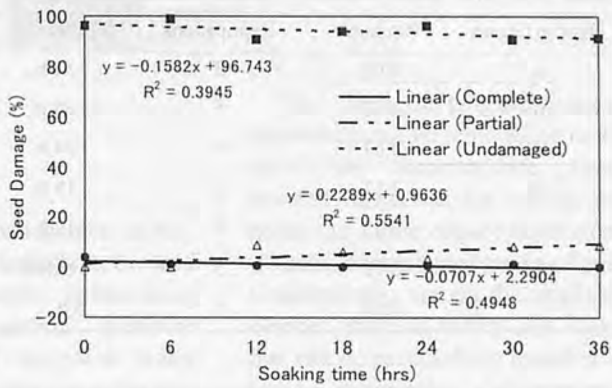

Fig. 5 Relationship between soaking time and seed damage for Safim planter.

The number of seeds dropped by the **Safim** planter decreased with soaking time from a high of 67 seeds for unsoaked to a low of 36 seeds for the 6 h soaking treatment (**Fig. 4**). Soaked seeds though larger in size were still small enough to pass through the seed plate hole (4.85mm). The decrease in the number of seeds dropped could, therefore, be explained in part to the reduced flow of seeds. The length of seeds at 4.191mm (24 h soaking, **Fig. 2**) though smaller than the plate diameters was quite large and meant seeds needed to squeeze through.

The planter recorded a drop in the number of completely damaged seeds with soaking time. The control at 3.85 % complete damage was significantly higher than all the soaked treatments (**Table 1**). Partial damage to seeds increased for the planter. The control and the 6 h soaking treatment were significantly lower than the rest of the soaking treatments. The 36, 30 and 12 h treatments were also significantly higher in partial damage than the rest (**Table 2**). The overall picture for the planter was that the shorter the soaking time, the higher the percentage of undamaged seeds (**Table 3** and **Fig. 5**).

Sebele Planter

A much more drastic decrease in the number of seeds dropped was observed for Sebele planter, from 238 seeds for the unsoaked to a low of 76 seeds for the 24 h soaking treatment (**Fig. 4**). For this planter, it could have been possible for two seeds to fall through at the same time before soaking. However, the possibility would have diminished markedly after soaking as evidenced by the increase in thickness as well (**Fig. 2**). The planter was slightly different from Safim in that both the complete and partial damage increased with soaking time (**Tables 4** and **5**).

Both the control and the 6 h soaking treatment were significantly lower than the 36 h soaking treatment for complete damage. The 36 h soaking treatment was significantly higher than all the other treatments for partial damage. The control, 6 and 12 h soaking treatments were all significantly lower than the 30 h soaking treatment for partial damage (**Table 5**). Overall an increase in seed soaking time, resulted in more seed damage for this planter (**Table 6** and **Fig. 6**).

Conclusions

For the Safim planter, sorghum seed soaking significantly reduced splitting (complete damaged) in seeds, partial damage in the form of bruising was, however, increased with seed soaking. The overall picture though was that seed damage in total (splitting and bruising) was increased due to soaking. Results

Table 3. Comparison of Means for Undamaged Seeds

Duncan Group	Average	Replications	Treatment
A	86.29	5	6 h
B	79.13	5	24 h
B	78.19	5	Control
B	75.77	5	18 h
B	73.30	5	36 h
B	73.20	5	12 h
B	72.95	5	30 h

Table 4. Comparison of Means for Completely Damaged Seeds

Duncan Group	Average	Replications	Treatment
A	7.96	5	36 h
AB	6.69	5	24 h
AB	5.98	5	12 h
ABC	5.56	5	18 h
BC	3.94	5	6 h
BC	3.69	5	30 h
C	2.42	5	Control

Table 5. Comparison of Means for Partially Damaged Seeds

Duncan Group	Average	Replications	Treatment
A	8.66	5	36 h
B	4.67	5	30 h
BC	2.67	5	24 h
BC	2.57	5	18 h
C	0.92	5	12 h
C	0.72	5	Control
C	0.00	5	6 h

Table 6. Comparisons of Means for Undamaged Seeds

Duncan Group	Average	Replications	Treatment
A	87.278	5	Control
A	86.060	5	6 h
B	83.748	5	12 h
BC	83.004	5	30 h
BC	82.606	5	18 h
C	81.206	5	24 h
D	78.050	5	36 h

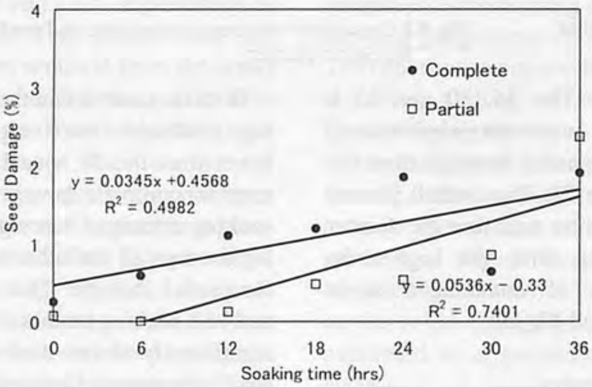

Fig. 6 Relationship between soaking time and seed damage for Sebele planter.

also indicated that soaking beyond 6 h might lead to significant amounts of seed damage.

On the other hand, sorghum seed soaking significantly caused seed splitting or cracking in the case of Sebele standard planter. Partial damage or seed bruising was increased with time of soaking. In general, seed soaking for 6 h seem to be acceptable, beyond which seed splitting and bruising may become significant.

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Double-Throated Flume: A Suitable Water Measuring Device for Rectangular Lined Channels

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Abstract

Accurate water measurement plays an important role in efficient management of irrigation water both during conveyance and application in the field. A variety of flumes developed so far, have been successfully installed and used in unlined channels for discharge measurement to a desirable accuracy. These devices have however, offered many problems when attempted to install in the lined channels. The problems are distortion of watercourse cross section, borrowing of soil from adjoining fields, water pollution by sediments and consequent problems of watercourse maintenance emerging out of sediment inflow.

The double-throated flume (DTF) is a recent innovation that offers opportunities of overcoming the above mentioned problems during installation and flow measurement in lined rectangular channels. The device has been tested for single throat width of 2 to 10 cm (4 to 20 cm for double throat), upstream head range of 28 to 60 cm and discharge range of 31 lps to 127 lps as calibrated against Cut-throat Flume measurements. The average correlation coefficient between the measured and the observed discharge ranges between 0.97 and 0.98 showing a high degree of predictability of the designed Double-throated flume.

Introduction

The efficient management of water supplies, particularly in the arid regions of the world, is becoming increasingly important. Accurate measurement of irrigation water plays a primary role in achieving high efficiency and operating the irrigation system on economical and scientific basis.

A number of water measuring devices such as Parshall Flume, Cut-Throat Flume, BCW Flume, Trapezoidal Flume and Weirs have been developed to measure water flow in open channels (Parshall 1926, Skogerboe 1969, Replogle 1978 and Replogle and Bos 1982). These devices can be installed and used in unlined watercourses after necessary alteration of the watercourse section, blocking the sides with soil borrowed from adjoining fields and stopping the leaks beneath and around the flume. After removal of flume from the channel, most of the blocking soil gets washed with water into the channel, and, therefore, causes a source of sediments in tertiary irrigation conveyance system. The necessity of watercourse remodelling, borrowing of soil and consequent addition of sediments present serious problems in water measurement at farm level.

The lined rectangular watercourses cannot be remodelled or altered to accommodate these devices. Moreover, the leakage beneath the flume cannot be stopped without external application of soil. Consequently, among the available devices, there is hardly any flume that can be successfully installed in lined rectangular watercourses without getting into problems mentioned above. Increasing number of lined watercourses as a result of water management development program in the country, necessitates that a suitable water measuring device for lined rectangular channels be developed to overcome the above mentioned problems associated with existing devices.

Methodology

The design of a flow measuring device such as double-throated flume (DTF) requires the establishment of a control section by providing a smooth constriction in the sides of the channel. It creates critical flow conditions at or near the constriction produced by a converging and a diverging section. However, unlike conventional flumes, the flow cross section of DTF lies between the outer wall of flume and channel wall on both sides of the flume as shown in **Fig.1**.

Fig. 1 The double-throated flume installed and operating in a watercourse.

Utilizing the concept of E-Y relationship as applied to specific energy, the general flow equation describing the critical depth - discharge relationship, as given by Henderson, 1971 was used in the design.

Design of DTF

As the double-throated flume was considered for use in lined rectangular watercourses, the design specifications of the existing lined watercourses were reviewed to provide basis for determining an appropriate size of the flume. Consequently, built design specification data of 14 watercourses were collected in the field. The resulting average depth of watercourses ranged from 32 to 61 cm, width from 48 to 72 cm and discharge from 55 to 195 lps.

The design parameters of the DTF were throat width, lengths of the converging and diverging sections, angles of convergence and divergence, height and width of flume at inlet/outlet sections, location of upstream (h_a) and downstream (h_b) gauges and the construction material. A width adjustment mechanism was provided to facilitate testing of flume at various ranges of the observed discharge and width of the reviewed watercourses.

Keeping in view the observed maximum depth of watercourse, the design height of the flume was kept as 61.5 cm. The flat bottom of the flume was provided with rubber lining to seal against any possible leakage beneath the flume. The locations of upstream and downstream gauges were fixed at the same distances from throat as provided for the cut-throat flume. The design specifications thus determined for DTF are shown in **Table 1**. The flume was fabricated locally using 14 gauge G.I. sheet metal.

Experimental Setup

The DTF was calibrated in the main watercourse of the university against the observed discharge from cut-throat flume measuring 20.3 cm × 91.4 cm size which was installed about 42 m downstream of DTF. The discharge through the watercourse was varied and the upstream head (h_a) and downstream head (h_b) for each setting of throat width of DTF were recorded. The flow through cut-throat flume was also measured simultaneously under steady state conditions. The throat width of DTF was varied from 2 to 10 cm which allowed the discharge to vary from 28 to 127 lps. Using the observed upstream head values, the corresponding discharge values through DTF were predicted by using the following models selected for the study:

Model 1: $Q = Ah^B$ (1)

Model 2: $Q = A + B\,h$ (2)

Model 3: $Q = Ae^{Bh}$ (3)

Model 4: $Q = A + B \log h$ (4)

Where:

A, B = constants of correlation

h = represents upstream head (h_a)

To represent the effective head (h) under submerged flow conditions, the following models were also tested:

$h = (h_a - h_b) / S$ (5)

$h = (h_a - h_b) / -\log S$ (6)

$h = h_b/h_a$ (7)

Table 1. Design Specifications of Double Throated Flume

Specification	Measurement,cm	
Linear Length of Flume	Min. 132	Max. 147
Width	Min. 39	Max. 63
Max. length of Converging Section	53	
Max. length of Diverging Section	94	
Height	61.5	
Location of h_a from Throat	38	
Location of h_b from Throat	78.9	

Table 2. Correlation Coefficients Between the Model Predicted and Observed Discharge Values

Throat Width	Correlation Coefficients			
(cm)	Model 1	Model 2	Model 3	Model 4
2	0.94	0.97	0.92	0.97
3	0.94	0.97	0.91	0.97
4	0.94	0.97	0.92	0.97
5	0.94	0.97	0.92	0.97
6	0.93	0.97	0.91	0.97
7	0.94	0.97	0.91	0.97
8	0.94	0.97	0.91	0.97
9	0.94	0.97	0.91	0.97
10	0.95	0.98	0.92	0.98

Results and Discussion

The head-discharge data thus collected were analyzed using software "STATPAC" to find the best fit for each of the throat width. Under submerged flow considerations, the statistical analysis of the predicted and observed discharge data resulted in low values of correlation coefficient (from 0.026 to 0.89) from Eq. 5,6 and 7. In addition, using the upstream head values only, the prediction models resulted in higher degree of correlation between depth and discharge as compared to the submerged flow considerations for effective head. A definite explanation for such a be-

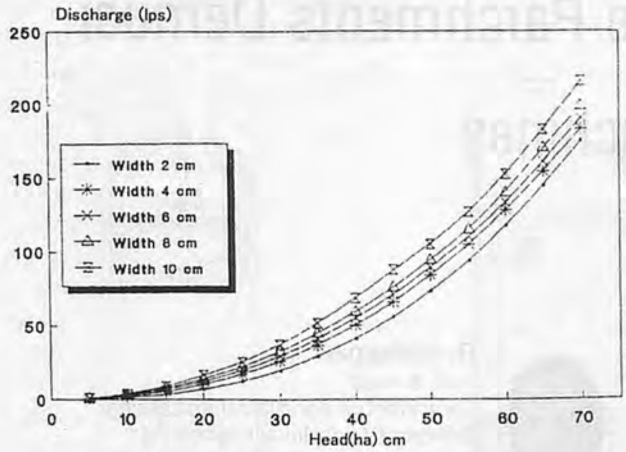

Fig.2 Head and predicted discharge relationship.

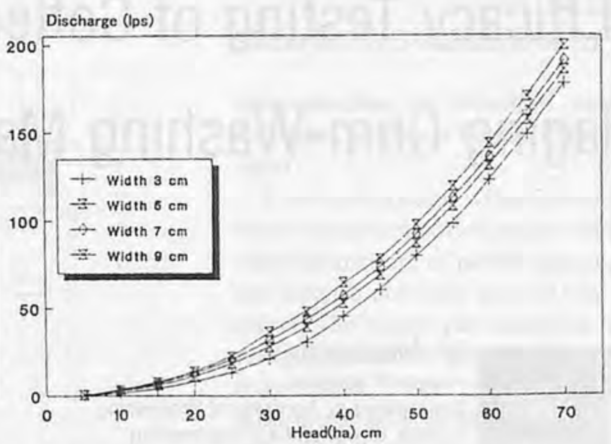

Fig. 3 Head and predicted discharge relationship.

haviour was not known. Further investigations are required to verify the calibration under different hydraulic conditions and explore possible reasons for such changes in behaviour.

Based on the upstream head data, the correlation coefficients between the observed and predicted discharge resulting from the statistical analysis of the data and the prediction models (Eq. 1 to 4), are shown in **Table 2**.

The values of correlation coefficient achieved with model 1 ranged from 0.93 to 0.95 and with model 3 ranged from 0.91 to 0.92 for various width settings of the DTF. The models 2 and 4 gave correlation coefficients ranging from 0.97 to 0.98.

Evidently, models 2 and 4 predicted the measured discharge more closely to the observed discharge values as compared to models 1 and 3. In addition, the coefficient of determinant were also high for models 2 and 4.

Table 3. Regression Coefficients of Prediction Model for Various Throat Widths

Throat Width (cm)	A	B	R
2	0.003	2.60	0.97
3	0.007	2.40	0.97
4	0.007	2.39	0.97
5	0.009	2.38	0.97
6	0.010	2.30	0.97
7	0.020	2.20	0.97
8	0.020	2.17	0.97
9	0.020	2.13	0.97
10	0.020	2.10	0.97

Although, the estimated standard deviation resulting from model 4 was high, the variance remained small as compared to model 2. Thus, model 4 yielding the highest correlation coefficient was selected for calibration and developing rating curves for the double-throated flume. The presented calibrations are based on upstream head (ha) although the submerged flow conditions prevailed during observations.

The regression coefficients (A and B) as well as the correlation coefficient (R) resulting from the selected prediction model (model 4) for developing the rating tables / curves for DTF using the observed data are shown in **Table 3**. For almost all settings of throat widths, the discharge was correctly predicted by the model for the initial part of the tested head range (28 to 45 cm), over predicted for the mid range (39 to 45 cm) and under predicted for the last part of the tested head range (47 to 60 cm).

Using the regression coefficients as given in **Table 3**, the predicted discharge for each width of throat and a head range of 0.5 to 71 cm were determined and the rating curves were developed as shown in **Figs.2** and **3**. The reported depth-discharge ratings were, however, developed on the basis of data from one watercourse, therefore, the device needs to be tested further for a wider range of watercourses and field conditions before it can be standardized.

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Efficacy Testing of Coffee Parchments Demucilaging Cum-Washing Machines

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Abstract

Two models of demucilaging cum washing mechanisms, namely; auger type and brush type were developed and evaluated with Arabica and Robusta parchments. In the auger type, the highest washing efficiencies of 84.9 percent and 95.3 percent for Arabica and Robusta, respectively, were achieved at 40 rpm, 200 kg/hr and 0.44 lps of auger speed, feed rate and water flow rate, respectively. The highest washing efficiencies of 100 percent and 85.7 percent for Arabica and Robusta, respectively, at 620 rpm, 25 kg/hr and 0.03 lps of brush speed, feed rate and water flow rate, respectively, were achieved in the brush type.

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Introduction

Coffee is used as a refreshing beverage at least since two hundred years ago. The fresh coffee beans are obtained from ***Coffea arabica*** L. and ***Coffea robusta*** L. Demucilaging of coffee beans is the second stage of the process in wet method after pulping. The presence of mucilage leads to the quality deterioration and has to be removed thoroughly by the process of washing. For washing, huge quantity of water is being used at estate level. This water requirement has to be used sparingly as its availability is limited at estate level. By combining the two processes, namely; demucilaging and washing and developing the suitable mechanisms will reduce the water requirement considerably. The physical properties like bulk density, true density, coefficient of static friction of the wet parchment with mucilage are important in the design of such mechanisms and handling systems for demucilaging and washing.

Gumbe (1989) determined the bulk density of coffee parchment by weighing known volume of coffee parchment. The porosity was calculated by fitting the voids of known volume of coffee parchment with a known volume of water. The density and porosity of parchment were 440 kg/m^3 and 43.7 percent, respectively. Menon (1989) observed that the pulped parchment coming out of the pulper was fed directly to the aqua washer, which removed the mucilage by friction. Thus, the parchment coming out of the machine was free from mucilage and washed for further drying.

Raoeng pulper and demucilager simultaneously pulped, removed the mucilage and washed the coffee. It essentially consisted of a long, cylindrical, perforated casing with a current of water running through it under pressure in which a fluted cylinder ran rapidly about 400 to 500 runs/min. The coffee beans, which were carried along with the water, were compressed between the cylinder and its casing. The shredded pulp and the mucilaginous material were separated from

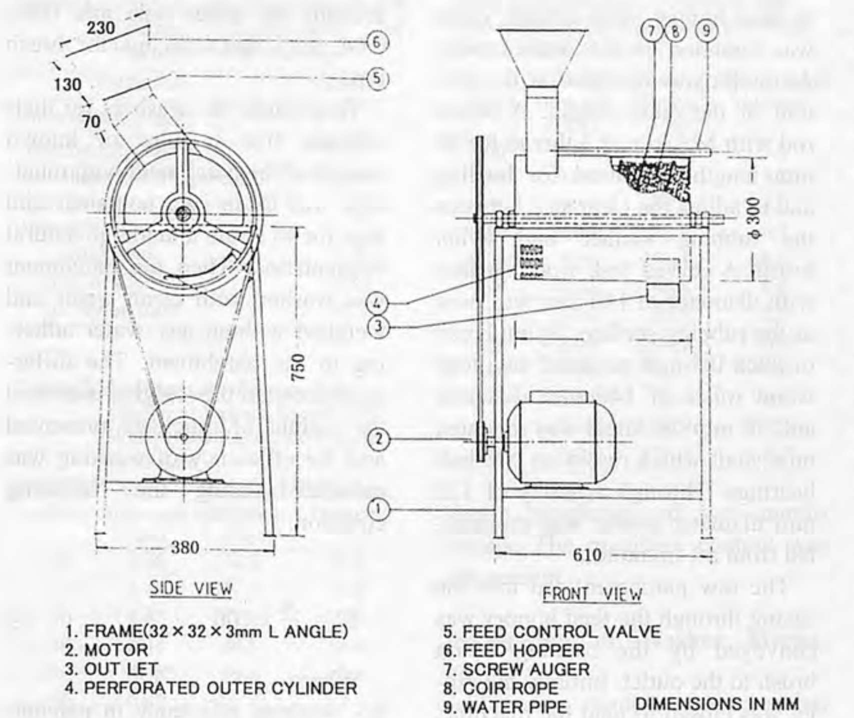

Fig.1 Power operated auger type washer.

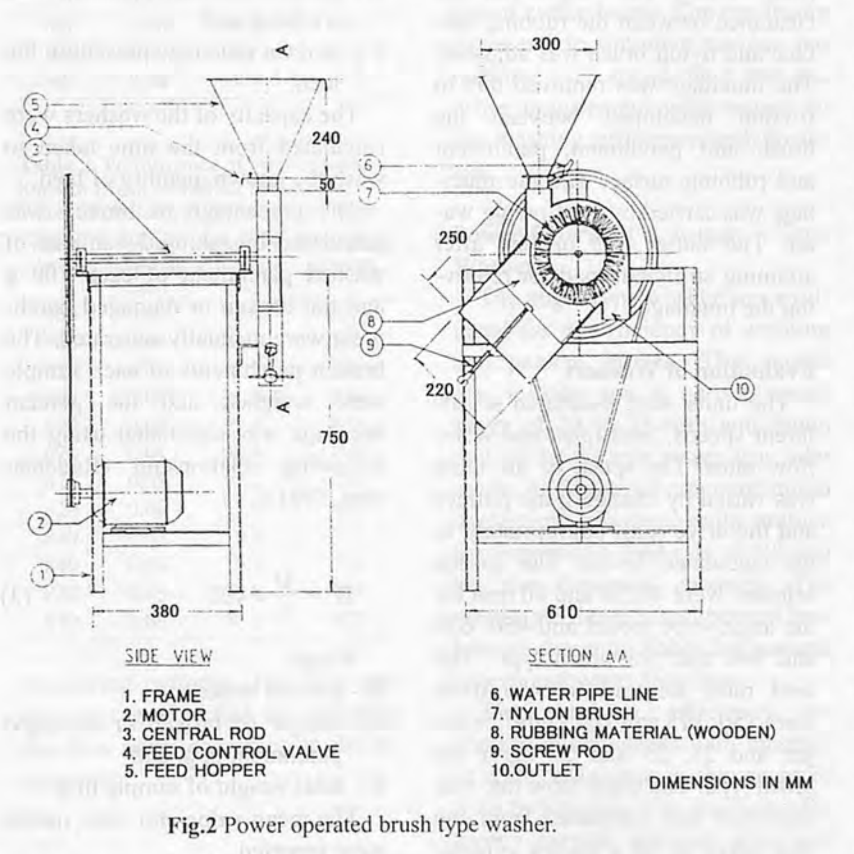

Fig.2 Power operated brush type washer.

the beans and washed away by the water. It needed 309 m^3 of water for 750 to 3000 kg of beans per hour and the energy required was 8 to 25 horse power.

Materials and Measurements

Determination of Densities and Friction Coefficient for the Parchment

The Arabica beans were obtained from the nearby coffee estates and were the mixture of partly ripened, just ripened and over ripened condition. The beans were stored in a refrigerator at 5°C and used in the experiments. The beans were pulped manually for the determination of bulk density, true density and friction coefficient.

The bulk density of the parchment with mucilage was determined from the mass of the known volume of the parchments. The true density was calculated using the true volume determined by the water displacement method. The coefficient of static friction for the parchments against mild steel surface and coir rope wound surface was determined using a friction apparatus. This apparatus consisted of a material holding container, frictionless pulley and a loading pan and was similar to the one used for neemnut (Visvanathan *et al.*, 1996) and soybean (Sreenarayanan *et al.*, 1988). From the weights in the pan (H) and the weight of the parchment with mucilage (W) the static coefficient of friction (f) was calculated using the formula:

$$f = H/W \quad (1)$$

The experiment was repeated 5 times for different surfaces like mild steel, mild steel wound with coir rope of 6 mm diameter along the direction of movement of sample holder and the average value for each surface was reported.

Development of Demucilaging Cum-Washers

Two models of demucilaging cum-washing machines were designed and developed aiming to minimize the cost and water requirement. Both models were operated with 1 hp motor.

Auger Type

This model consisted of an auger drum, outer cylinder with perforations, feed hopper, outlet with closing mechanism and a frame as shown in **Fig.1**. The auger drum was made to a diameter of 210 mm and 380 mm length. An auger with a pitch of 70 mm and 30 mm depth was developed on this drum with mild steel plate of 6 mm thick.

Coir rope of 3 mm diameter was wound over the auger drum to develop a rough surface. The auger drum was mounted on two bush bearings made of brass on either side. The outer cylinder was made of a mild steel perforated sheet of 20 SWG thick and had the perforations of 25 × 3 mm size. The diameter and length of the outer cylinder were 32 cm and 43 cm, respectively. A feed hopper and outlet with loading mechanism were provided opposite to each other along the length on the outer cylinder. The unit mounted on a frame of mild steel angle.

The raw parchment fed was conveyed by the auger drum and simultaneously water was applied through the water pipe. Slowly the outlet was opened until the feed rate coincided with the outlet. The mucilage was removed by friction developed between the parchments, the coir rope and parchment and parchment with the outer cylinder. The mucilage removed was washed away by the water and discharged through perforations provided on the outer cylinder.

Brush Type

This model consisted of a feed roller with nylon bristles, loading mechanism, outlet, strainer, water inlet, central rod, ball bearings and stand as shown in **Fig.2**.

The outer casing was made of 3 mm thickness mild steel sheet to 260 mm diameter with 4 cm diameter hole at the centre on either side. A feed hopper with control valve was mounted on the outer casing. An outlet was mounted at the bottom of the outer casing. A screw rod with M 12 × 1.5 thread for 80 mm length was used for loading and to adjust the clearance between the rubbing surface and nylon brush. A curved teak wood surface with diameter of 140 mm was used as the rubbing surface. Nylon brush of thick 0.5 mm mounted on a teak wood roller of 140 mm diameter and 50 mm thickness was mounted on a shaft which rested on two ball bearings. Through a pulley of 125 mm diameter power was transmitted from a 1 hp motor.

The raw parchment fed into the casing through the feed hopper was conveyed by the rotating nylon brush to the outlet. Initially the outlet was closed to load the machine, simultaneously water was also applied. Using the screw rod the clearance between the rubbing surface and nylon brush was adjusted. The mucilage was removed due to friction developed between the brush and parchment, parchment and rubbing surface and the mucilage was carried by the flowing water. The outlet was opened after attaining sufficient load for removing the mucilage.

Evaluation of Washers

The units were evaluated at different speeds, feed rates and water flow rates. The speed of the units was varied by changing the pulleys and the drive shaft appropriately to the calculated levels. The speeds selected were 40, 50 and 60 rpm for the auger type model and 450, 620 and 840 rpm for brush type. The feed rates selected for the trials were 150, 175 and 200 kg/hr for auger and 21, 23 and 25 kg/hr for brush type. The water flow rate was regulated and calculated from the time taken to fill a known volume. The various levels of water flow rates used in the trials were 0.38, 0.40, 0.42 and 0.44 lps (litres per second) for auger type and 0.03, 0.04, 0.05 and 0.06 lps for brush type.

To evaluate the washers for their efficacy five samples of known weight of the parchment with mucilage was taken in a container and kept for 48 hours to undergo natural fermentation. Then the parchment was washed with clean water and weighed without any water adhering to the parchment. The difference between the weights indicated the weight of mucilage presented and the efficiency of washing was calculated using the following equation:

$$E = \frac{A}{F} \times 100 \qquad (2)$$

Where,

E - washing efficiency in percentage,

A - percent mucilage removed after washing, and

F - percent mucilage present in the feed.

The capacity of the washers were calculated from the time taken to wash the known quantity of feed.

The percentage of broken was calculated by taking 3 samples of washed parchment of each 100 g and the broken or damaged parchment were manually separated. The broken parchments of each sample were weighed and the percent breakage was calculated using the following relationship (Rademachar, 1981):

$$B = \frac{D}{S} \times 100 \qquad (3)$$

Where,

B - percent broken,

D - weight of broken or damaged parchment in g, and

S - total weight of sample in g.

The mean values for each model were reported.

The water requirement was calculated by measuring the time required to fill the known volume in a

Table 1. Densities, Mucilage Content and Friction Coefficients of Parchment with Mucilage

Particulars	Value
Bulk density, kg/m^3	681.37
True density, kg/m^3	856.57
Mucilage content, percent	20.09
Coefficient of static friction against:	
i. Mild steel	0.42
ii. Coir rope	0.80
iii. Nylon rope	0.32

Table 2. Performance of Auger Type Washer at Various Speeds and Water Flow Rates at 200 kg/hr Feed Rate

Speed (rpm)	Water flow rate (lps)	Washing efficiency (%)	Percentage broken
40	0.38	74.9	10.4
40	0.40	79.6	10.4
40	0.42	82.8	10.4
40	0.44	84.5	10.4
50	0.38	69.6	12.1
50	0.40	73.3	12.2
50	0.42	79.3	12.2
50	0.44	79.5	12.2
60	0.38	70.2	16.7
60	0.40	74.6	16.8
60	0.42	76.1	17.2
60	0.44	78.9	17.4

Table 3. Performance of Washer with Smooth Brush at Various Speeds and Water Flow Rates at 25 kg/hr Feed Rate

Speed (rpm)	Water flow rate (lps)	Washing efficiency (%)	Percentage broken
450	0.03	82.6	3.1
450	0.04	82.3	3.2
450	0.05	82.1	3.5
450	0.06	81.9	3.6
620	0.03	99.9	0.0
620	0.04	98.4	0.0
620	0.05	98.2	0.0
620	0.06	97.9	0.0
840	0.03	79.8	7.8
840	0.04	79.3	7.9
840	0.05	78.8	8.0
840	0.06	78.2	8.2

measuring cylinder. The flow rate of water was varied by adjusting the flow valve according to the requirement.

Results and Discussion

Table 1 shows the value of different properties of wet parchments. The mucilage content was 20 percent.

Table 4. Performance of Washer with Hard Brush at Various Speeds and Water Flow Rates at 25 kg/hr Feed Rate

Speed (rpm)	Water flow rate (lps)	Washing efficiency (%)	Percentage broken
450	0.03	82.7	3.2
450	0.04	82.5	3.8
450	0.05	82.2	4.0
450	0.06	82.0	4.1
620	0.03	100.0	0.0
620	0.04	98.5	0.0
620	0.05	98.2	0.0
620	0.06	98.1	0.0
840	0.03	98.1	8.0
840	0.04	79.4	8.1
840	0.05	78.9	8.2
840	0.06	78.3	8.2

Evaluation of Washer Mechanisms

The two models of the washer mechanisms developed were evaluated for their efficiency and breakage of coffee beans. The results are discussed to optimize various parameters viz., speed, feed rate and water requirement with respect to the washing efficiency and breakage.

Power-Operated Auger Type Washer

The auger type washer was evaluated for its efficiency of washing and percent broken. This model was initially run at various speed range of 20 to 75 rpm and found that 40 to 60 rpm range was adequate. At this speed range the initial evaluation was done with arabica parchment at a feed rate of 200 kg/hr, the designed capacity. The washing efficiency and percent broken are shown in **Table 2** at various speeds and water flow rates.

The washing efficiency increased at all speeds with increase in water flow rate in the range of 0.38 to 0.44 lps. The washing efficiency ranged between 69.6 and 84.5 percent in the speed range of 40 to 60 rpm. The highest washing efficiency was 84.5 percent at auger speed of 40 rpm with water flow rate of 0.44 lps.

The percent broken at speeds of 40 and 50 rpm at all water flow rates are around 10.4 and 12.1, respectively. However, at the speed of 60 rpm the percent broken was in the range of 16.7 to 17.4. At all water flow rates the percent broken increased with increase in speed.

Further evaluation at water flow rate of 0.44 lps, the speed and feed rate of 40 rpm and 200 kg/hr resulted in the highest efficiency of 84.5 percent was achieved. The efficiency was least at 150 kg/hr feed rate at 60 rpm speed.

Power-Operated Brush Type Washer

The brush type washer fitted with smooth and hard brushes were evaluated. This was initially run at various speed range of 300 to 1000 rpm and found that 450 to 840 rpm is suitable. At this speed range the initial evaluation was done with arabica parchment at four levels of water flow, keeping the feed rate at 25 kg/hr.

Performance of Washer with Smooth Brush

The brush made of 0.35 mm thick bristles was termed as smooth brush.

The performance of this type washer was evaluated for its efficiency of washing and the percent broken and the results are shown in **Table 4** for the various speeds and water flow rates at 25 kg/hr feed rate.

The washing efficiency decreased at all speeds with increase in water flow rate from 0.03 to 0.06 lps. The washing efficiency ranged from 78.2 to 99.9 percent for the speed range of 450 to 840 rpm. The maximum efficiency of 99.9 percent is achieved at 620 rpm with the water flow rate of 0.03 lps. The least value of washing efficiency is achieved at 840 rpm with the water

Table 5. Performance of Washers for Robusta Coffee

Washer	Speed (rpm)	Feed rate (kg/hr)	Water flow rate (lps)	Washing efficiency (%)
Auger type	40	200	0.44	95.3
Brush type (Hard brush)	620	300	0.04	85.73

flow rate of 0.06 lps.

The percent broken in this brush type model with smooth brush was maximum of 8.2 at 840 rpm with the water flow rate of 0.06 lps. There was no broken at 620 rpm speed at all the water flow rates. The variation in percent broken is much less in 450 and 840 rpm for these water flow rates.

Performance of the Washer with Hard Brush

The brush made of 0.5 mm thick bristle was termed as hard brush. The performance of the washer fitted with hard brush was evaluated for washing efficiency and percent broken and are given in **Table 4** for the various speeds and water flow rates at 25 kg/hr feed rate.

The washing efficiency decreased at all speeds with increase in water flow rate in the range of 0.03 to 0.06 lps. The washing efficiency ranged from 78.3 to 100 percent for the speed range of 450 to 840 rpm at these water flow rates. The maximum efficiency of 100 percent was obtained at the speed of 620 rpm with the water flow rate of 0.03 lps and the minimum value of washing efficiency was obtained at a speed of 840 rpm with water flow rate of 0.06 lps.

The percent broken in this brush type washer was maximum of 8.22 at the speed of 840 rpm with water flow rate of 0.06 lps. There was no broken at 620 rpm at all the water flow rates. But the percent broken is not varied widely at 450 rpm at all the water flow rates as similar to the speed 840 rpm. The increase in water flow leads to increase in percent broken at the speeds of 450 and 840 rpm but it is negligible.

Evaluation of Washers with Robusta Parchment

The auger type and brush type washers were evaluated with Robusta parchment at their respective optimum operating conditions found suitable for Arabica parchment. The performance of washers with Robusta parchment is shown in **Table 5**.

The auger type washer achieved a washing efficiency of 95.3 percent with no breakage. During the trials with brush type washer with smooth bush, it was observed that the mechanism was not able to remove the mucilage and washing the parchment effectively. The mechanism with hard brush was able to demucilage and wash the parchment by the repeated pass of the washed parchment. It was required to pass the parchment for 2 times because of the more sticky nature of mucilage in Robusta coffee. A washing efficiency of 85.7 percent was obtained at this stage.

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Modification and Evaluation of a Self- Propelled Reaper for Harvesting Soybean

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Abstract

Soybean crop was introduced in Madhya Pradesh about 20 years back. Today Madhya Pradesh is known for highest production of soybean in India. As this is not a traditional crop, shortage of labour is being felt at the time of harvest. Moreover, due to the shattering nature of the crop, timely harvesting of the crop is essential. A self-propelled reaper was earlier introduced in South India for harvesting rice by CIAE-IRRI Industrial Extension Project. The reaper was modified by reducing the height of the cutter bar knife and speed of cross conveyer belt and evaluated for harvesting soybean in Madhya Pradesh. The machine was evaluated for three crop years with three different crop varieties. The average plant height for the three crop varieties; Punjab-1, Durga and MACS 58, were 64.5, 64.1 and 80.6 cm, respectively. The average grain moisture content ranged between 11.4 to 13.1%. The average field capacity, fuel consumption and stubble height were 0.206 ha/h, 0.747 l/h and 8.8 cm, respectively. The average stubble loss, shattering loss, header loss and total machine loss were 0.45, 2.28, 2.73 and 3.57%, respectively of the grain yield. The height of cut had direct bearing on increased stubble loss. The shattering loss was the major problem that accounted for 63.8% of the total farm machinery loss.

Introduction

Soybean has become an important crop over the years in India. As this is not a traditional crop, shortage of labour is being felt at the time of harvest. Due to the shattering nature of the crop, timely harvesting of the crop is essential. The dominant crop characteristics affecting harvesting are low set pods, easily shattered pods, lodging and variation in grain moisture content. Thus, mechanical harvesting of soybean is necessary to ensure timeliness of harvesting operation. Efforts need to be concentrated on development of harvesting machines so that the harvesting losses could be minimized. Lamp et al (1961) reported that over 80% of all the losses accounted for header losses of which 55 per cent was due to shattered loss, 28 per cent stalk and lodged loss and 18 per cent stubble losses. The efficient design of harvesting equipment for soybean calls for reducing gathering losses. Low cutting height must be achieved to get under low set pods. The results of one study conducted at Iowa State University (ASAE, 1977) has indicated that harvesting losses increased as the cutting height was increased. The harvesting losses doubled as the height of cut increased from 90 mm to 165 mm. The study further revealed that the least shattered loss was associated with the lowest blade speed, sharpest blade, increasing moisture content and reduced stem displacement during cutting.

There have been considerable breakthrough in the design of header assembly ever since the mechanical harvesting was introduced for soybean. The introduction of flexible floating cutterbar and automatic height control has been quite effective in achieving consistent and low cutting height and thereby reducing header losses. The challenge to reduce losses continues to centre around gathering devices. Over crowding of crop and higher forward speed with the conventional 76.2 mm knife section led to introduction of narrow pitch cutter bar. The use of air-jet guards on the flexible floating cutter bar helped in considerable reduction of header losses.

In India the use of reapers and combine harvesters is progressively increasing. Harvesting with a combine can be done only after the crop dries up to a moisture content suitable for threshing. At this stage the losses due to shattering increase. This situation can be avoided by harvesting at a higher moisture content with a reaper and, subsequently, threshing the crop after sun drying. Yadav and Yadav (1985) reported development of tractor-operated reaper for soybean. Tractor front mounted as well as side mounted reapers are being used for harvesting soybean also. But both designs put a limitation on lowest achievable height of cut which is required for harvesting the crop with low set pods. To meet the demand of small and medium farmers the self propelled reaper introduced by CIAE-IRRI Industrial Extension Project for rice harvesting has been adapted for harvesting soybean. The reaper is operated by an operator walking behind it and works on the principle of vertical conveying and windrowing. The reaper has been modified by reducing the speed of the cross conveyor belt and minimum height of cutterbar. The performance of the reaper was evaluated for harvesting three varieties of soybean for three seasons. Two small-scale manufacturers in Madhya Pradesh have started the production and sale of the reaper for harvesting soybean.

Materials And Methods

Salient Features of Vertical Conveyor Windrower

The vertical conveyor reaper windrower has certain features which enable it to have an edge over other designs of harvesters. The plants are conveyed in a vertical orientation and the earheads or pods have little contact with the moving parts of the machine which helps in minimizing the shattering loss. The plants are laid on the ground in a clear windrow maintaining the direction of tillers perpendicular to direction of travel which facilitates manual collection and tying of the crop.

Fig. 1 Self-propelled reaper harvesting soybean.

Construction of the Machine

The self-propelled vertical conveyor reaper windrower (**Fig.1**) is powered by a 6-hp light weight diesel engine. The main parts of the machine are crop row dividers, standard cutter bar with 76.2-mm pitch of knife sections, upper and lower vertical conveyor belts, steel lugged wheels and power transmission system. The effective cutter bar width is one metre. The total mass of the reaper with 6 hp Greaves Lombardini diesel engine is about 160 kg. The crop row dividers guide the standing crop, the star wheels direct the crop towards cutter bar and help in slightly lifting the crop after it is cut, turn it at 90° and help in conveying by the two lugged conveyor belts. The two lugged flat belts convey the cut crops towards the right side of the machine keeping the crop in vertical orientation with the help of wire springs. At the end, the crop is discharged and laid on the ground in the form of windrow. The power is transmitted from the engine to top transmission shaft through a V-belt and pulley giving drive to the steel ground wheel through chain and sprocket. The top transmission shaft, in turn drives vertical shaft of the cutter bar through a V-belt and pulley giving reciprocating motion to the cutter bar. With the help of chain and sprocket, the drive is taken for the shaft driving the pulleys of two conveyor belts. The top conveyor belt drives the star wheels of the crop row dividers. Pressure springs are fitted below the star wheels to keep the cut crop in upright position while it is being conveyed out of the machine. A brief specifications of the reaper are given in **Table 1**.

Evaluation Procedure

The evaluation of the self-propelled walk-behind vertical conveyor reaper was carried out under different varieties and crop conditions of soybean. The crop conditions were defined by crop variety, plant height, plant population, row-to-row spacing, straw grain ratio, moisture content and crop yield. The test was conducted to assess area harvested, fuel consumption, operational speed, working width, stubble height and losses.

The harvesting losses were due to header, conveying and machine loss. Prior to the determination of losses, Pre-harvest losses were determined with reference to the loose grain or pod fallen on the ground prior to the operation of the machine. The samples were drawn from an area of 1 m^2 and the results were expressed in kg/ha. The header loss consisted of shattering loss, stubble loss, lodged loss and stalk loss. Shattering loss refers to the beans free of pods or in pods free of stalk while stubble loss refers to the beans remaining in pods attached to the stubble. Lodged loss includes beans remaining in pods attached to stalks which are not cut or cut at heights greater than the stubble. Stalk loss amounts to beans remaining in pods attached to stalks which are cut but not conveyed to the windrow. For purposes of assessment of losses, lodged loss was included in stubble loss and the stalk loss was considered along with shattering loss. These losses were determined based on samples drawn from 1 m^2 area. The results were expressed both in kg/ha and as percentage of average crop yield. The conveying loss is attributed to the loss caused by the gathering and discharging devices. The loss was determined by collecting samples of loose grain and pods on a piece of cloth placed along the direction of motion just outside the crop being cut. The crop after cutting was allowed to fall on the cloth in the form of windrow and the sample for

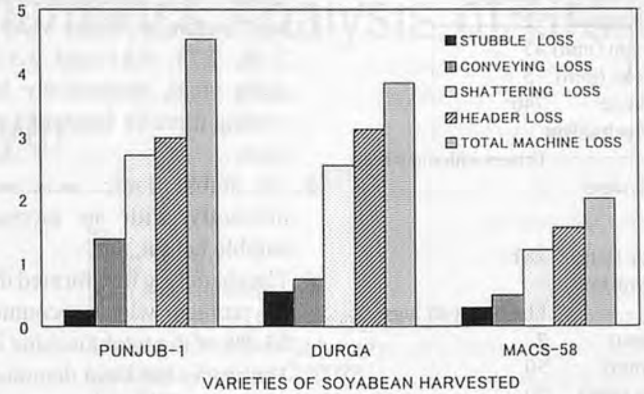

Fig. 2 Average losses in harvesting soybean.

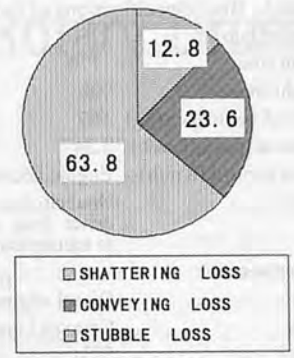

Fig. 3 Average distribution of harvesting losses in soybean crop.

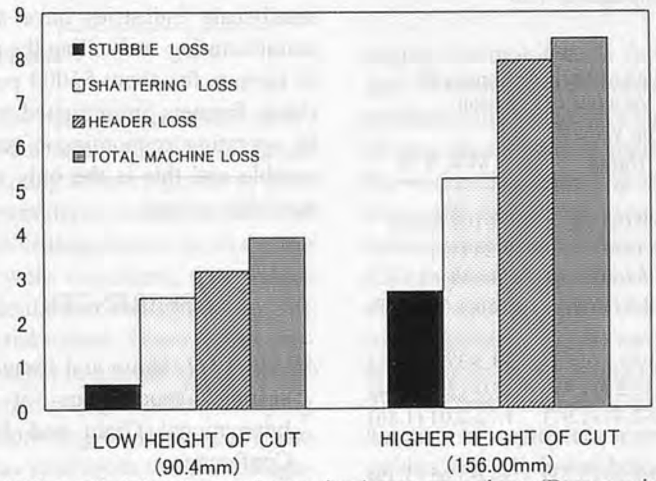

Fig. 4 Effect of height of cut on harvesting losses on soybean (Durga variety).

one-metre test run was drawn. The results were expressed both in kg/ha and as percentage of average crop yield. For specification and determination of various performance characteristics including losses, reference was drawn from relevant Indian Standards (BIS, 1981) and RNAM test codes and procedures (RNAM, 1983).

Results and Discussion

Field Performance and Crop Factors

The results of harvesting studies on three different varieties of soybean conducted during three crop years are summarized in **Table 2**. The plant height varied from 28 to 78 cm, 31 to 99 cm and 55 to 109 cm with average values of 64.5 cm, 64.1 cm and 80.6 cm for the three varieties Punjab 1, JS 72-280 (Durga) and MACS 58, respectively. The average values of straw moisture content were 11.7%, 36.1% and 34.5% while the grain moisture content was 13.1%, 11.4% and 11.4%, respectively, for Punjab 1, Durga and MACS 58,. The corresponding values of effective field capacity were 0.246, 0.180 and 0.217 ha/h and the forward speed of the machine were 3.18, 3.23 and 3.39 km/h for the three varieties stated above. The hourly fuel consumption was measured at 0.77, 0.78 and 0.68 l/h while the fuel consumption on area basis was 3.13, 4.35 and 3.13 l/ha in harvesting three varieties Punjab 1, Durga and MACS 58, respectively. It was possible to achieve low cutting height with the machine and corresponding values of stubble height were 86.5, 90.4 and 84.0 mm for the three soybean varieties.

The average values of grain losses for the three crop varieties are shown in **Fig. 2**. The total machine losses were recorded at 4.35, 3.86 and 2.01% of grain yield for Punjab-1, Durga and MACS 58 varieties of soybean. An overall distribution of various estimate of the harvesting losses is shown in **Fig. 3**. The stubble loss, conveying loss and shattering loss represented 12.6, 23.6 and 63.8%, respectively of total machine loss.

Effect of Stubble Height on Harvesting Losses

It is important to maintain low cutting height in soybean crop in order to minimize harvesting losses, particularly stubble losses due to low pod-setting characteristic of the crop. The effect of height of cut on harvesting losses is shown in **Fig. 4**. It is evident that as the average stubble height increased from 90.4 mm to 156.0 mm, the stubble losses increased from 0.58% to 2.66% of grain yield. The shattering losses also increased from 2.54% to 5.22% due to greater contact of pod-bearing crop canopy with the moving conveyor belt of the machine.

Conclusions

The one-metre self-propelled

Table 1. Brief Specifications of Self-propelled Vertical Conveyor Reaper

Length (mm)	2180	Effective width (mm)	927
Width (mm)	1170	Minimum height (mm)	45
Height (mm)	900	Length of stroke (mm)	75
Mass of the reaper (kg)	160	Strokes pr minute	740
Forward speed (Km/h)	3.24	**Gathering Mechanism**	
Speed control (Km/h)	Engine accelerator	Type	Drivers with star wheels
Clutch	One belt clutch in the driver from engine to transmission	No. of row dividers	4
Power source		Star wheel dia. (mm)	283
Type	Diesel engine	**Cross conveyor belt**	
Make	Greaves Lombardini	Type	Flat belt with lugs
Model	523	No. of belt (mm)	2
Power (hp)	6.0	Size of belt (mm)	50
Rated Speed (rpm)	1800	Height of lugs (mm)	50
Cutterbar		Lug spacing (mm)	125
Type of cutter bar	Reciprocating	Linear belt spped(m/s)	1.26

Table 2. Test Results of Summary One-metre Self-propelled Vertical Conveyor Reaper Windrower for Harvesting Different Varieties of Soybean

Item	Crop Variety		
	Punjab-1	Durga	MACS 58
Crop Factors			
Plant height (cm)	28-78 (64.5)	31-99 (64.1)	55-109 (80.6)
Plant population (No./m row)	13-34 (26)	6-53 (17)	9-24 (17)
Number of pods per plant	23-82 (44)	9-135 (60)	25-64 (44)
Straw-grain ratio	0.92-1.59 (1.16)	1.5-2.0 (1.69)	1.56-1.59 (1.57)
Moisture content, wb(%)			
Stem	6.7-17.0 (11.7)	16.9-56.1 (36.1)	30.2-37.9 (34.5)
Grain	11.2-14.8 (13.1)	9.4-13.2 (11.4)	11.0-12.2 (11.4)
Crop yield (t/ha)	1.42-1.73 (1.61)	1.43-2.41 (1.97)	1.72-2.01 (1.86)
Field Performance			
Forward speed (km/h)	3.09-3.21 (3.18)	2.33-4.00 (3.23)	3.12-3.65 (3.39)
Effective field capacity (ha/h)	0.219-0.256 (0.246)	0.142-0.216 (0.180)	0.200-0.234 (0.217)
Fuel consumption			
l/h	0.52-0.95 (0.77)	0.65-1.00 (0.78)	0.65-0.70 (0.68)
l/ha	2.01-3.74 (3.13)	3.08-6.34 (4.35)	3.00-3.26 (3.13)
Stubble height(mm)	60-135 (86.5)	50-160 (90.4)	50-120 (84.0)
Losses			
Pre-harvest loss (kg/ha)	0-74.0 (25.3)	0-77.0 (21.6)	0-8.0 (3.2)
Shattering loss (%)	1.38-3.88 (2.70)	1.04-5.02 (2.54)	0.81-2.03 (1.20)
Stubble loss (%)	0-0.96 (0.28)	0-1.96 (0.58)	0-0.76 (0.36)
Header loss (%)	1.38-4.23 (2.98)	1.20-5.46 (3.12)	0.80-2.79 (1.56)
Conveying loss (%)	0.40-2.65 (1.37)	0.04-1.87 (0.74)	0-0.88 (0.45)
Total machine loss (%)	1.96-5.32 (4.35)	1.72-5.58 (3.86)	1.06-3.09 (2.01)

NOTE: *Figures in parentheses represent average values*

reaper, introduced for rice harvesting in southern India was adapted for soybean harvesting by reducing the speed of the cross conveyor belts and by lowering the minimum height of the cutterbar. The reaper, when evaluated for harvesting Punjab 1, Durga and MACS 58 varieties of soybean, with average plant heights of 64.5, 64.1 and 80.6 cm, respectively gave the following results when grain and straw moisture contents were in the range of 11.4 to 13.1% and 11.7 to 36.1%, respectively:

1. The average values of effective field capacity, fuel consumption, forward travelling speed and stubble height were 0.206 ha/h, 0.747 l/h, 3.27 km/h and 87.6 mm, respectively;
2. The stubble loss, shattering loss, header loss, conveying loss and total machine losses were 0.45, 2.28, 2.73, 0.84 and 3.57% of grain yield, respectively in harvesting three varieties of soybean crop;
3. The stubble losses increased significantly with an increase in stubble height, and
4. The shattering loss formed the major problem which accounted for 63.8% of the total machine loss.

The reaper has been demonstrated to the farmers in Madhya Pradesh in India for harvesting soybean. Some small-scale industries have started manufacturing and selling the reaper to farmers for about $1000 per machine. Farmers are attracted to it as its operating economics is very favorable and this is the only option available to them.

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Kinematics Analysis of Grains in a Rotary Drum Dryer

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Abstract

On the basis of theoretical analysis of the forces applied to grains and moving locus of grains in a rotary drum dryer, residence time and contact heating time of grains in the dryer were calculated. The values of calculation conformed to the measured values. These results provided reliable foundation in theory and practice for further studying and improving the dryer. They also serve as references to other designers of similar type of dryer.

Introduction

In order to reduce the mildew losses of grains, a conduction type continuous-flow rotary drum dryer with high temperature short time drying was developed in the light of present economic and technical conditions of the vast countryside in developing countries. The dryer is efficient, inexpensive, simple and multi-purpose. The high temperature direct conduction drying technology was used in this dryer. So, the heat efficiency and productivity depends to a large extent on the residence time (τ_1) and contact heating time(τ_2) of grains in the dryer. Grain is a kind of live material and highly sensitive to temperature. We must guarantee that τ_1 and τ_2 are in accord with the demands of drying technology. A few scholars[1,2] studied the determination of residence time of particular material in a drum dryer sometime ago. However, the following problems were met in their studies: (1) The most complicated motion of materials in drum dryer, that is the slide on the cylinder wall and guides, was neglected; (2) It was considered that τ_1 had nothing to do with the structure parameters, such as diameter of cylinder, number of guides, etc.; (3) It was held that τ_1 had nothing to do with the cylinder rotation speed and cylinder inclination. These problems led to the great difference between the calculated and measured results.

To counter the problems above, the forces applied to grains and the actual moving locus of grains in rotary drum dryer were analyzed theoretically in this paper, and τ_1 and τ_2 which tallied with the actual situation were calculated. These results supply reliable foundation in theory for further studying and improving the drum dryer. Moreover, the drum dryers were applied widespread in all kinds of fields of the food industry, agricultural product processing industry, chemical industry, metallurgical industry, building materials, light industry, etc.

Structure of the Dryer

The drum dryer used in this analysis and test is shown in **Fig. 1**. The cylinder rotates at a normal speed to optimize the contact duration between the grains and the hot cylinder surface. β is the angle between cylinder axis and the horizontal plane (β=1.5). Four groups of guides were riveted on the inner surface of the cylinder at α inclination along the cylinder axis. There were 5 guides in every group

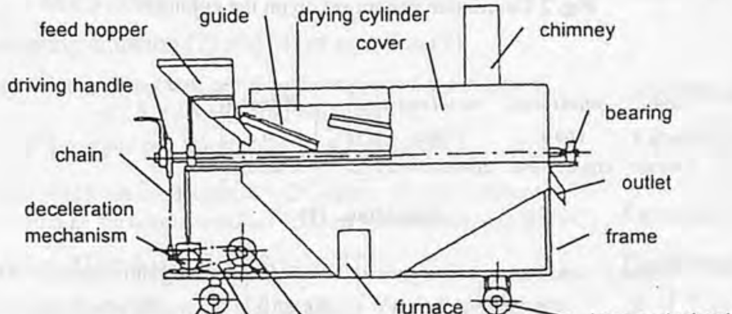

Fig. 1 Schematic drawing of dryer.

of guides. The functions of the guides were to achieve a more uniform loading and a more aggressive mixing of grains, and to maintain the designed residence time. After being fixed to the cylinder, the guide was a spiral surface with very big pitch, and which was always perpendicular to the cylinder surface approximately. In order to simplify the calculation, the guide was approximated to a plane.

Kinematics Analysis

The motion of grains in the drum dryer was very complicated, as it was influenced by a lot of factors such the properties of grains (variety, moisture content, inner friction outer friction, and so on). To study on the motion of single grain was very difficult and meaningless. In this paper, a pack of grains, which was held by a unit length of guide, was regarded as a motion particle.

To a single grain, rolling resistance is smaller than sliding resistance and easier than sliding. However, to a pack of grains sliding is its main motion, because of the great adhesive force among grains, which includes interaction attraction among molecules, capillary force of adhesive water, surface tension of free water, electrostatic attraction, and so on, and friction among grains , which is much larger than the friction between steel plate and grains.

Forces Applied to Grains in Drum Dryer

The coordinate systems ($OXYZ$ and $O_1X_1Y_1Z_1$) are shown in **Fig. 2**.

Where, the angle between Y axis and the horizontal plane is β. $O_1X_1Y_1Z_1$ had reached its final position by a sequence of two relative rotations and a translation, a rotation α about axis X in the clockwise direction followed by a rotation ωt about axis Y in the counter clockwise direction, then a translation of coordinate original point from O to O_1.

Therefore, the relationship between $OXYZ$ and $O_1X_1Y_1Z_1$ was present as follows:

Equation. (1)

Where, R--radius of cylinder.

The forces acted on a pack of grains include gravity (G), reacting force of guide (N_1), reacting force of cylinder (N_2), friction between grains and guide(N_1f), friction between grains and cylinder (N_2f), and centrifugal inertia force ($P=M\omega^2R'$).The gravity could be resolved to 3 components below in the $O_1X_1Y_1Z_1$ coordinate system.

Equation. (2)

The forces act on grains were provided with two different situations:

(1)When $N_2 \neq 0$, the grains would not move along X_1 or Z_1 direction. So

$$\begin{aligned} &\Sigma F_{X1} = G_{X1} + N_2 - M\omega^2R'=0 \\ &\Sigma F_{Y1} = G_{Y1} + N_2f - N_1f \qquad (3) \\ &\Sigma F_{Z1} = G_{Z1} + N_1 = 0 \end{aligned}$$

Where, $R' = R - a/3$; a--the height of guide; ω--angular velocity of cylinder; f--sliding friction coefficient between grains and steel plate (f=0.54[3]); M--the mass of a pack of grains.

(2)When $N_2 = 0$, the grains would not move along Z_1 direction. So

$$\begin{aligned} &\Sigma F_{X1} = G_{X1} - N_1 f\cos\delta \\ &\Sigma F_{Y1} = G_{Y1} - N_1 f\sin\delta \qquad (4) \\ &\Sigma F_{Z1} = G_{Z1} + N_1 = 0 \end{aligned}$$

Where, δ --the angle between X_1 direction and the actual moving direction(S) of grains $\delta = arctg(dY_1/dX_1)$.

Motion of Grains in Drum Dryer

The motion of grains in a drum dryer could be divided to four different stages:

(1)Uniform circular motion ($A_1 \sim A_2$). Grains are elevated with the rotation of cylinder under the action of guides.

(2)Compound motion ($A_2 \sim A_3$). At the same time of uniform circular motion, grains slide along the guides.

(3)Projectile motion ($A_3 \sim A_0$). After being elevated to a certain height, grains break away from the guides and do a projectile motion.

(4)Slide ($A_0 \sim A_1$). In this stage, grains always contact with the cylinder surface, and slide on the

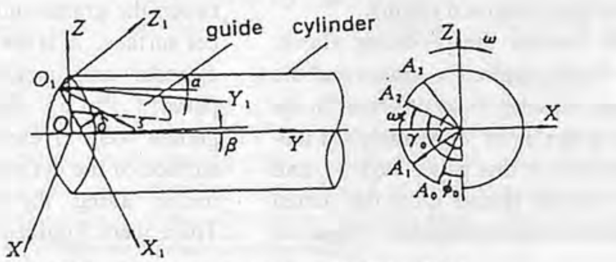

Fig. 2 Coordinate system set up on the cylinder.

$$\begin{bmatrix} x_1 \\ y_1 \\ z_1 \end{bmatrix} = \begin{bmatrix} \cos\omega t & -\sin\omega t\sin\alpha & -\sin\omega t\cos\alpha \\ 0 & \cos\alpha & -\sin\alpha \\ \sin\omega t & \cos\omega t\sin\alpha & \cos\omega t\cos\alpha \end{bmatrix} \begin{bmatrix} x \\ y \\ z \end{bmatrix} - \begin{bmatrix} -R\cos\omega t \\ 0 \\ R\sin\omega t \end{bmatrix}$$

Equation. (1)

$$\begin{bmatrix} G_{X1} \\ G_{Y1} \\ G_{Z1} \end{bmatrix} = \begin{bmatrix} \cos\omega t & -\sin\omega t\sin\alpha & -\sin\omega t\cos\alpha \\ 0 & \cos\alpha & -\sin\alpha \\ \sin\omega t & \cos\omega t\sin\alpha & \cos\omega t\cos\alpha \end{bmatrix} \begin{bmatrix} 0 \\ Mg\sin\beta \\ -Mg\cos\beta \end{bmatrix} = \begin{bmatrix} Mg\sin\omega t\cos(\alpha+\beta) \\ Mg\sin(\alpha+\beta) \\ -Mg\cos\omega t\cos(\alpha+\beta)) \end{bmatrix}$$

Equation. (2)

cylinder surface, until grains begin to do uniform circular motion after grains drop to the guides.

Uniform circular motion

The motion equations were as follows:

$$X = -R\cos\omega t$$
$$Z = R\sin\omega t \qquad (5)$$
$$2n\pi + \phi_0 - 90° \leq \omega t \leq 2n\pi + \gamma_0 - 90°$$
$$(n=0, 1, 2, 3...)$$

Where, ϕ_0--the starting boundary of uniform circular motion;

γ_0--the stopping boundary of uniform circular motion, that is, the starting boundary of the slide along guides.

Compound motion

The fixed reference system was fixed in the frame and a pack of grains was regarded as a motion particle. So, the relative motions of the motion particle are the slides along cylinder surface and guides, including the radial slide and the axial slide, and the involving motion is the uniform circular motion.

In the light of the D'Alembert's principle, the equation of compound motion for the grains could be written in the form.

$$F + N - Ma = 0 \qquad (6)$$

Where: $a = a_r + a_e + a_k$, a_r--the relative acceleration; a_e--the involving acceleration; a_k--the Coriolis acceleration.

Involving Motion

The involving motion equation was the same as equation (5), but $2n\pi + \gamma_0 - 90° < \omega t \leq 2n\pi + \gamma - 90°$ (n=0, 1, 2, 3...). Where: γ ---the angle between radial direction and the negative direction of Z axis when grains were breaking away from the guide.

Relative Motion

By way of theoretical analysis, it could be proved that grains could not slide only in radial direction or in axial direction when the rotation speed of cylinder was lower than 20 (r/min)(as space is limited, the process of proof was left out). Because the highest rotation speed of this drum dryer was 12.05r/min, the radial slide and axial slide of grains had to happen simultaneously.

The following equations could be obtained if we project the equation (6) to X_1 and Y_1 direction respectively: **Equations. (7, 8** and **9).**

In terms of boundary condition, we knew that $t=T_0$, $x_1=0$, $y_1=0$, $dx_1/dt=0$, $dy_1/dt=0$, at the starting boundary of slide. It could be proved,

$$tg\delta_0 = \left.\frac{dy_1}{dx_1}\right|_{t=T_0} = \left.\frac{d^2y_1}{d^2x_1}\right|_{t=T_0} \qquad (10)$$

On the basis of the sufficient and necessary condition ($d^2x_1/dt^2 \geq 0$ and $d^2y_1/dt^2 \geq 0$) when grains could slide, solving equations (8), (9) and (10) simultaneously with the aid of a computer, we could obtain δ_0, T_0 and γ_0. Where, δ_0 is the angle between X_1 direction and the actual moving direction(S), T_0 is the time when grains start to slide on the guides (**Table 1**).

Given, $A=g/\omega\cos(\alpha+\beta)$, $B=g/\omega\sin(\alpha+\beta)$, and integration of equations (8) and (9) led to

Equations. (11, 12, 13, and **14)**

Where C_1, C_2, D_1, and D_2 were integral constants, which could be solved in terms of the boundary conditions.

The slide locus of grains on the guide was a curve, and the δ was changing with the time. However, in view of the time that grains slid on the guide was very short, and the variation range of δ was very small, δ was considered as a fixed value (given, $\delta = \delta_0$) to simplify the calculation. Then, we could find: T_1 (the time that grains broke away from guide), the slide distance along Y axial directions, γ, and the original condition of the projectile motion-- the position of grains(x_1, y_1, z_1), the velocity of grains(dx_1/dt, dy_1/dt, dz_1/dt), and the acceleration of grains(d^2x_1/dt^2, d^2y_1/dt^2, d^2z_1/dt^2) when $t=T_1$. The original condition could be transformed to the position (x_0, y_0, z_0), velocity(v_{xo}, v_{yo}, v_{zo}), and acceleration(a_{xo}, a_{yo}, a_{zo}) in the $OXYZ$ coordinate system by means of equation (1).

Projectile motion

Projectile Motion Locus

Given $t_1 = t - T_1$, the parameter equations of projectile motion locus of grains could be expressed in the following form:

$$x = (g\sin\beta + \alpha_{xo})t_1^2/2 + v_{xo}t_1 + x_o$$
$$y = (g\cos\beta + \alpha_{yo})t_1^2/2 + v_{yo}t_1 + y_o \qquad (15)$$
$$z = \alpha_{zo}t_1^2/2 + v_{zo}t_1 + z_o$$

The Turned Angle(ϕ) of Cylinder While a Pack of Grains Held and Projected Once

The stopping boundary of projectile motion was exactly the starting boundary of slide, that is the intersection point of space curve showed by equation (15) and the inner surface of cylinder ($x^2+z^2=R^2$). In this

$$\Sigma F_{X1} - M(d^2x_1/dt^2) - M\omega^2R' - 2M\omega(dy_1/dt)\sin\alpha\cos\omega t = 0$$
$$\Sigma F_{Y1} - M(d^2y_1/dt^2) = 0 \qquad \textbf{Equation. (7)}$$

Substituting equation (2) and (4) in equation (7)

$$(d^2x_1/dt^2) = g\sin\omega t\cos(\alpha+\beta) - gf\cos\omega t\cos(\alpha+\beta)\cos\delta - \omega^2R' + 2\omega(dy_1/dt)\sin\alpha\cos\omega t \qquad \textbf{Equation. (8)}$$

$$(d^2y_1/dt^2) = g\sin(\alpha+\beta) - gf\cos\omega t\cos(\alpha+\beta)\cos\delta \qquad \textbf{Equation. (9)}$$

$$(dx_1/dt) = (2B\sin\alpha - A)\cos\omega t + (2C_2\sin\alpha - fA\cos\delta)\sin\omega t + 0.5fA\sin\alpha\sin\delta\cos 2\omega t + 2\omega tB\sin\alpha\sin\omega t - \omega^2R't + C_1 \qquad \textbf{Equation. (11)}$$

$$(dy_1/dt) = -fA\sin\delta\sin\omega t + \omega Bt + C_2 \qquad \textbf{Equation. (12)}$$

$$x_1 = 1/\omega(4B\sin\alpha - A)\sin\omega t - 1/\omega(2C_2\sin\alpha - fA\cos\delta)\cos\omega t + \frac{1}{4\omega}fA\sin\alpha\sin\delta\sin 2\omega t - 2Bt\sin\alpha\cos\omega t - 1/2\,\omega^2R't^2 + C_1t + D_1 \qquad \textbf{Equation. (13)}$$

$$y_1 = 1/\omega fA\sin\delta\cos\omega t + 1/2\,\omega Bt^2 + C_2t + D_2 \qquad \textbf{Equation. (14)}$$

way, we could obtain T_2 (the time while a pack of grains was during the projectile motion), and the position A_0 (x, y, z) when the projectile motion stopped, and the angle between radial direction and negative direction of Z axis[$\phi_0 = arctg(x/z)$].

ϕ could be solved by the following equation: $\phi = \gamma - \phi_0 + \omega T_2$

The number of times which a pack of grains was held and projected by guides when cylinder turned one circle, was described by the equation:

$p = m/INT(m\phi/360+1)$

Where, m--the number of guides in every group of guides; INT--integer function.

Calculation and Test Verification of Residence Time

Residence Time (τ_1)

The probability when grains were held by guides would vary with the variation of feeding quantity. But all the grains had the identical probability if every condition was kept the same.

$$\kappa = \sum_{i=1}^{m} (V_i/V)$$

Where , V_i--the volume of grains which were held by a unit length of ith guide ;

V--the volume of grains which were piled up in a unit length of cylinder;

k--the probability when grains were held by guides.

Because the structure of all the guides were the same, the volume of grains which were held by different guide were equal under the stable working situation, i.e., $V_1 = V_2 = ... = V_i = ... V_m$.

Therefore, $k = mV_i/V$ (Generally, $mV_i \leq V$). This could lead to

$Q = INT(l/(k \cdot y)+1)$

$\tau_1 = 60 \times Q/(n \cdot p) \quad (s)$

Where, Q--the number of times , which a pack of grain was held by guides;

l--the length of cylinder(m);

n--the rotation speed of cylinder (r/min).

Contact Heating Time (τ_2)

Except the time when grains slid on guides and done the projectile motion , grains would always contact with the cylinder. So, the contact heating time could be defined by the following equation:

$\tau_2 = \tau_1 - Q \times (T_1 + T_2 - T_0)$

Comparing the calculation results with the test results, we found that calculation results of residence time tallied with the test results. Besides, the contact heating time was reduced with the increase of cylinder rotation speed, i.e., that the heat efficiency was reduced and the heat consuming rate was increased with the increase of cylinder rotation speed. Moreover, the lower of the cylinder rotation speed was the more distinct of such kind of variation. However, ϕ was small when the cylinder rotated at a lower speed, which would result in insufficient mixing and no-nuniform drying of grains. Therefore, the cylinder rotation speed could neither be very high nor be very low, which completely conformed to the test results[4].

Table 1. Results of Calculation and Test

Rotation speed(r/min)	1.95	4	7	10	12.05
T_0 (s)	1.999	0.982	0.573	0.415	0.354
γ_0 (°)	23.384	23.573	24.083	24.872	25.571
T_1 (s)	2.433	1.320	0.850	0.658	0.581
γ (°)	28.470	31.672	35.690	39.460	42.011
T_2 (s)	0.253	0.255	0.258	0.254	0.252
ϕ (°)	87.779	101.552	117.391	129.899	138.037
τ_2 (s)	325.349	123.405	84.142	55.071	42.213
Calculation value of τ_1(s)	348.718	140.000	102.857	66.00	52.282
Test value of τ_1(s)	307.4	152.3	113.9	61.4	51.3
Error(%)	13.44	-8.08	9.70	7.49	1.91

Conclusions

(1)The motion pattern and forces acted on grains in drum dryer obtained by the theoretical analysis tallied with the actual situation very well.

(2)The structural and technical parameters had a bearing on residence time and contact heating time , which must be considered when residence time and contact heating time were calculated.

(3)The residence time obtained by theoretical analysis conformed to the test result. The theoretical analysis method was also for the references of other designers to design similar type of dryer.

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Evaluation of Drying Methods and Storage Conditions for Quality Seed Production

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Abstract

To prevent quantitative and qualitative seed losses, seed drying for safe storage is important. High temperature and relative humidity (r.h.) are undesirable conditions for seed storage in tropical countries. This study was conducted to evaluate a range of seed drying methods and storage conditions, with a view to selecting an appropriate method(s) for use in humid tropical countries.

Pea seeds at three initial moisture contents (m.c.) of 23.8, 18.0 and 14.5% were dried to 10% m.c. before storage. The performances of four different drying methods: artificial (Kiwi Mini) dryers set at 30°C or 45°C, natural sun drying, and in-bin natural ventilation drying were tested. Dried seeds were stored under 2 conditions: open storage at 20.5°C and 55% r.h., and closed storage at 25°C and 90% r.h. for 20, 40, and 60 days. Deterioration of the seed due to storage conditions and drying methods used was determined by assessing effects on seed germination.

Germination did not differ significantly among the selected drying methods. Conductivity, a measure of seed vigour, was also not affected by drying methods. Germination after open and closed storage for 20 days was not different from that immediately after drying, although differences appeared after 40 days storage. However, closed and open storage for 60 days significantly reduced germination to 33 and 50%, respectively.

Introduction

To avoid loss of seed germination and vigour, drying is often necessary for successful storage of grain/seed until it is used for processing or sowing (Hill, 1996). There are many drying systems based on two basic drying methods: natural and artificial drying. Natural drying in the open air is a method commonly used in developing countries. Grain is spread out on a concrete or brick floor under the sun and is manually turned to obtain uniform drying. However, losses in quality due to sudden rain and hygiene issues often occur. Ventilation is a more convenient method based on blowing heated or unheated air through the seed bulk. Unheated air is quite effective if the r.h. of the air is lower than the equilibrium moisture content of the seed to be dried. As natural drying is highly dependent on weather, it is difficult or impossible during the rainy season, and when the costs of quantity and quality losses due to birds and fungal contamination are taken into account, losses can be considerable.

Artificial drying uses heated air that is mechanically forced through or over the seed mass. However, high drying temperatures will kill the embryo. At the initial stages of drying, when the m.c. of the seed is high, high levels of moisture evaporate from the surface. When the seed m.c. decreases following evaporation, this enhances seed temperature, causing death of the embryo. High temperature drying, or drying too quickly or excessively, can dramatically reduce viability (Bewley and Black, 1986) and vigour (Hampton, 1990). This is because high temperature causes case-hardening whereby the surface of the grain dries out rapidly, sealing the moisture within the inner layers (Abe et al., 1992). This leads to mould growth during storage and a reduction in germination. In order to ensure the viability of seeds, Justice and Bass (1978) have recommended that the seed temperature during drying should not exceed 40 - 43°C.

Maintenance of seed m.c. at a safe level for storage is essential because seed m.c. will come to an equilibrium with the surrounding air during storage. The purpose of seed storage is to preserve and maintain the high germination capacity with high vigour of seed from harvest to planting time. Degradation of seed during storage depends principally on a combination of environmental factors. Temperature and moisture have a direct influence on the speed of development of insects and microorganisms. After drying and placing in storage, seed m.c. still may change depending upon the fluctuation of temperature and r.h. of the surrounding air as storage environment affects directly the seed m.c. that indirectly affects seed quality by the activity of storage fungi.

The best storage atmosphere is dry and cold. This helps retain the viability and vigour of many seed types. Temperature and humidity of the storage atmosphere should be kept relatively low to minimize seed deterioration. In most cases, the lower the seed m.c. and storage temperature, the longer the viability. Harrington (1973) reported that in the range of seed m.c. between 5 and 14%, seed longevity doubled when seed m.c. was reduced by 1%, or when temperature was lowered by 5°C. However, according to Roberts and Ellis (1977), longevity would double for every 2.5% reduction in seed m.c. and 6°C reduction in temperature.

Islam (1984) showed that a storage temperature of 28°C and 84% r.h. drastically reduced germination of pea seed within 8 weeks and seeds completely lost viability after 13 weeks of storage. Sangakkara (1988) dried soybean seeds at 30°C and found that germination was lost completely after 8 weeks with seeds stored at 20°C, 90% r.h. Castillo (1992) also found that pea seeds decreased their germination relatively rapidly after 1 month and completely lost germination after 3 months when they were stored at 25°C, 95% r.h.

This research was undertaken to evaluate changes in seed quality (germination, vigour) as affected by drying techniques and drying temperature, and storage condition, including temperature and r.h. Such a study was considered important for developing specifications for farmers and the commercial sector involved in seed production in humid tropics.

Materials and Methods

Seed Preparation

Peas (*Pisum sativum* L.) were hand harvested at 28.0% m.c. and were immediately spread on an indoor concrete floor to avoid further deterioration before being machine threshed at 24.0% m.c. All the seeds were threshed at a drum speed of 680 rpm by a 'Seed Master' thresher and cleaned by spiral separation. Pure seeds (ISTA, 1996) were then mixed by hand and divided into three lots for experiment. Lot 1 with seed m.c. of 23.8% was used to dry immediately. Lots 2 and 3 were spread on a floor in a large shed at a depth of 5 cm for 3 and 10 days with regular turning using a plastic spade to reduce moisture content to 18.0% and 14.5%, respectively. Seeds of each lot were tested for germination, m.c. and conductivity before drying.

Experimental Treatments

Natural sun drying

Pea seeds were spread in a thin layer of 2.5 cm on a concrete floor in the open under the sun using an area of 0.69 m × 0.40 m. Every 4 hours, a sample was taken to test if the seeds had reached 10% m.c. The seeds were then collected, put into a plastic bag, and placed in a sealed plastic box to prevent rewetting overnight. The seeds were exposed to the sun each day until they reached the desired m.c. level.

In-bin natural ventilation drying

A wooden box 0.69 m long × 0.40 m wide × 0.30 m high was constructed to dry pea seed by natural air ventilation. The pea seeds were spread in a shallow layer of 2.5 cm on the surface area which was similar to that of natural sun drying. The difference in size between the wide intake (exposed to wind direction) and narrow outlet forced air to circulate within the box, and remove moisture from the seeds. Every 4 hours, a sample was taken to test if the seeds had reached 10% m.c. As for natural sun drying, the seeds were collected, put into a plastic bag, and placed in a sealed plastic box to prevent rewetting overnight. The seeds were placed in a ventilated bin during day time until they had reached the desired m.c.

Artificial drying

Two Kiwi Mini dryers (Massey University-designed mini-scale laboratory dryers), were used for this experiment. A built-in fan in each dryer was used to force ambient air to pass through an electrical heater, and through the seeds. Two drying temperatures, 30°C and 45°C, were used for drying, to compare the effect of these two temperatures regimes on energy consumption, time taken, and on quality of the seed.

Storage Conditions

Once the seeds reached 10% m.c., seed germination and conductivity were immediately determined. Each seed lot after drying was halved. One-half of each seed lot was placed onto a wiremesh above a solution of 30% glycerine and 70% distilled water, closed in a plastic box and put into a 25°C room to establish 90% r.h. condition. The other half of each seed lot was put into a paper bag and placed in an open environment in a laboratory under conditions of 20.5°C and 55% r.h. After 20, 40 and 60 days of storage, the seeds from both storage conditions were tested for seed m.c., germination, conductivity and hollow heart percentage.

Parameters and Measurement Procedures

Seed moisture content

The seed m.c. was determined by using a gravimetric method. For samples of seed with a m.c. higher than 17%, seed m.c. was determined in two stages (ISTA, 1996). For the first stage, two sub-samples of at least 25 g each were placed for 30 minutes into a oven set at 130°C, and then kept exposed separately in a laboratory room for 2 hours, by which time the seed m.c. had reduced to less than 17%. The seed moisture loss at this stage was then determined. For the second stage, these two sub-samples were ground separately and two sub-samples of less than 10 g each were put into a 130°C oven for 1 hour as stated by ISTA (1996) for pea. The second stage was also used to determine the m.c. of any pea seed lot with a m.c. of less than 17%. If the difference in m.c. between two sub-samples exceeded 0.2%, the determination was repeated.

Seed germination

The between-paper (BP) method was used for the germination test for peas. Four replicates of 50 seeds each from seed sample were rolled in damp papers and placed in a 20°C room for 5 and 8 days counts (ISTA, 1996). Assessment of seed germination was based on the number of normal, abnormal seedlings and remained seeds (ISTA, 1993).

Conductivity

The conductivity of the seed was measured by using a conductivity meter Model CDM-83. The procedure used was as described by Hampton (1995).

Results and Discussion

Effect of Drying Methods

Seed germination

Germination did not differ significantly among the different drying methods (**Table 1**). Germination of seed dried at 45°C was similar to that dried at 30°C. Similar results were found by Mian (1983) who dried sweet corn seeds at 30 and 50°C, and found that there were no significant differences in germination, and Gane et al. (1984) who suggested that for pea seed below 24% m.c., the drying temperature should not exceed 49°C. In the present experiment, as pea seeds were spread on a concrete floor for sun drying, heat was also expected to be conducted and might have affected seed embryos by direct contact with material bearing heat for a relatively long period. This did not affect seed germination immediately after drying, but might affect seed viability in storage.

Seed conductivity

Seed dried at 30°C had a lower conductivity ($18.6\ \mu S.cm^{-1}.g^{-1}$) than seed dried at 45°C (Table 1). A drying temperature of 45°C might have been expected to damage seed membranes (Seyedin et al. 1984), and, therefore, increase the electrical conductivity of the seed. However, the conductivity of all

Table 1. Seed Germination and Conductivity When Dried by Different Drying Methods

	Drying methods				
	30°C	45°C	Sun	Vent*	LSD
Germination, %	78	75	73	73	6.6
Conductivity, $\mu S.cm^{-1}.g^{-1}$	18.6 a	21.7 b	21.2 ab	20.6 ab	3.0

* Vent = Natural Ventilation.
Values with the same letter in rows are not significantly different at $P < 0.05$.

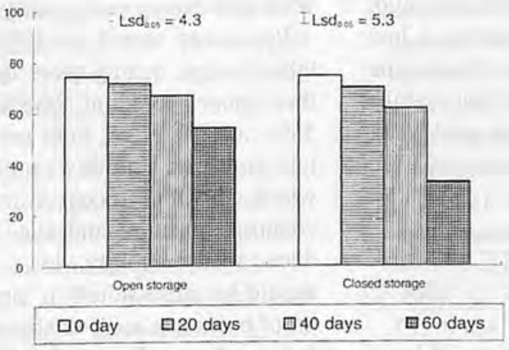

Fig. 1 Effects of storage conditions and time on seed germination.

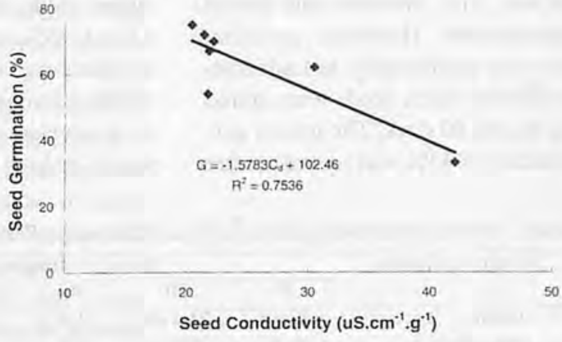

Fig. 2 Relationship between seed conductivity and germination during storage.

seed lots after drying by the different methods was less than 24 $\mu S.cm^{-1}.g^{-1}$. Based on the seed vigour, grades suggested by Gane et al. (1984), all pea seeds lots would have been considered to be of high vigour.

Effect of Storage Time and Storage Conditions

Seed germination

The time of storage in open conditions had a major effect on germination. Seed germination immediately after drying and after 20 days of storage was similar. Germination was only 75% immediately after drying because on extremely wet summer, sprouting had occurred prior to harvest (Thuy, 1998). However, germination was reduced after 40 and 60 days of storage, respectively, (**Figure 1**). The slight reduction in germination during open storage agreed with Castillo (1992) who stored pea seeds from 2 cultivars (Pania and Princess) in different storage conditions for different length of times. His results showed that seeds that were open stored in ambient condition for 18 months had a slight reduction in their germination. The germination of soybean seeds after drying at 45°C (Sangakkara, 1988) decreased from 93 to 69% when stored in 40% r.h. at 20°C for 2 to 12 weeks.

The seeds stored in closed conditions after 20 days had no reduction in germination. (**Figure 1**). This might be due to the fact that this storage time was not long enough for seed m.c. increases and mould development. However, germination was significantly and adversely affected when seeds were stored for 40 and 60 days. The lowest germination of 33% was recorded after 60 days of storage. This was because at this stage, seeds gained an equilibrium m.c. of 22% at 25°C, 90% r.h. Such storage conditions are favourable for mould development (Priestley, 1986).

Seed conductivity

Seed conductivity, an indicator of the seed deterioration process, did not increase after 20 days in open storage. However, conductivity increased slightly to 21.8 $\mu S.cm^{-1}.g^{-1}$ after 40 days, and remained stable until 60 days of storage (**Table 2**). On the other hand, conductivity of the seeds stored in closed conditions was significantly affected by the length of storage. It increased slightly from 20.4 immediately after drying to 22.2 after 20 days of storage, then rapidly increased to 30.5 and 42.1 $\mu S.cm^{-1}.g^{-1}$ after storage for 40 and 60 days, respectively. Castillo (1992) had earlier found that an increase in storage time caused increasing leachate conductivity.

The seed m.c. is increased in high r.h. storage conditions, and this lead to changes in quantity of membrane phospholipids, and the formation of free radicals which can result in membrane damage (Powell, 1986). This result was similar to that of Castillo (1992) who had shown that after 12 months storage, the conductivity of pea seeds increased from an average of 22.0 to 28.0 $\mu S.cm^{-1}.g^{-1}$ in ambient open storage, and dramatically increased to 45 $\mu S.cm^{-1}.g^{-1}$ when seeds were stored at a high r.h. of 95%, despite having a low temperature of 5°C. Singkanipa (1996) similarly found that viability in soybean and wheat seed when stored at high temperature of 30°C and 95% r.h. decreased as conductivity increased.

Table 2. Seed Conductivity ($\mu S.cm^{-1}.g^{-1}$) as Affected by Periods of Storage

Storage conditions	Period of storage (days)				
	0	20	40	60	LSD
Open	20.4 a	21.4 ab	21.8 b	21.7 b	1.2
Closed	20.4 a	22.2 b	30.5 c	42.1 d	1.7

Values with the same letter in rows are not significantly different at $P < 0.05$

An inverse relationship (**Figure 2**) of r = - 0.87 between germination and conductivity during storage supported the view that pea seed conductivity was a good predictor of storage life. Conductivity seemed to have a high relationship with field emergence of pea seeds after storage (r = - 0.82 and - 0.76 with and without irrigation, respectively), than any of other laboratory vigour methods (Castillo 1992).

Conclusions

Natural sun drying may be satisfactorily adopted for small scale seed drying. However, the operation would depend upon the weather, require large spaces for drying and labour to spread, stir and collect seeds. This may be appropriate for farmers with small farms with abundant manpower available. If labour was unavailable at peak times, as for commercial farms and state owned warehouses, fossil fuel energized dryers would be more suitable.

No differences in seed quality characteristics (germination, conductivity) were found between the selected drying methods. Artificial drying has advantages for seed production in terms of time and labour savings, though it can cause severe loss of seed quality and vigour if it is not carried out properly. Pea seeds can be dried at up to 45°C without reducing seed quality.

Pea seeds stored at 25°C and 90% r.h. lost quality more quickly than those stored at 20.5°C and 55% r.h. Therefore, high temperature and high humidity in storage, which commonly occur in tropical countries, must be controlled to reduce seed viability loss. Seed should be close-stored in moisture proof packages, such as aluminium foil or plastic film for small seed lots. In large scale operations, temperature and r.h. in silos must be kept low in order to maintain a safe seed moisture content throughout storage. Seed germination was linear-inversely related with seed conductivity during storage.

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An Anthropometry of Indian Female Agricultural Workers

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Abstract

The sample of unorganised female agricultural workers involved in different agricultural activities from the Western part of India, were selected in order to gather information about the body dimensions, which are commonly used in ergonomic design. Earlier anthropometric surveys carried out in the country were very few and specific to the male agricultural workers only. Therefore, 30 body dimensions necessary for the design of these equipment were identified and a sample study was conducted on 40 female farm workers in the age group of 18 to 50 years. Data has been analysed statistically and compared with those obtained for the agricultural workers from other parts of the country. The mean stature of West Indian female workers was observed 154.6(±6.18), while those for male workers from Eastern, Southern, Central, Northern and Western regions were 162.1(±5.80), 160.7(±6.00), 162.0(±4.95), 168.5(±6.84) and 164.4 cms, respectively. Body dimensions were compared with Americans, Germans and Japanese and a variation was found. Through some examples, effort was made here to illustrate the use of the data in the design of farm equipment.

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Introduction

In Indian agriculture, women have an important role in the operation of different agricultural equipment. The time has come when the tractor is also being operated by Indian women. The use of female anthropometric data along with those of the male can help in the proper designing equipment for better efficiency, safety and human comfort. With the advent of technology, disregard for the human factor is no longer possible and a knowledge of man's size and its variability has become progressively more critical in designing farm equipment and workplaces (Woodson and Conover, 1973).

Ergonomic dimensions correspond best to the orientation of the designed hardware which are registered in different positions and postures that simulate the real working postures and positions in a conventional form. Hence, to achieve better efficiency, human comfort and safety, it is necessary to design the equipment keeping in view the operators' capabilities and limitations. The use of female anthropometric data in the design of agricultural equipment is one of the forward steps in this direction. However, the designers are now developing ergonomic consciousness and if the anthropometric data of the target population are available, those could be used in the design process.

In Western countries a large amount of anthropometric data are available for reference (NASA, 1978). The anthropometric data bank assembled and maintained by the Aerospace Medical Research Laboratories, Dayton, Ohio, is the largest and most comprehensive single repository of raw anthropometric data in the world. However, it does not contain any data on the Indian (Asia) population. The most important requirement in the consideration of fe-

Table 1. Anthropometric Data of Indian Female Agricultural Workers

Body dimensions	Mean	SD	CV	Range	Percentile		
					5th	50th	95th
Stature	154.6	6.18	4.00	146.2 - 171.3	148.0	154.8	165.8
Body weight, kg	49.5	5.25	10.61	34.0 - 70.2	38.4	49.1	68.2
Grip strength, kg	24.3	2.66	10.95	15.1 - 37.5	18.3	25.4	31.6
Grip diameter (inside)	3.8	0.22	5.79	3.0 - 4.8	3.2	3.8	4.3
Waist circumference	88.3	5.07	5.74	67.1 - 96.9	72.5	87.4	91.4
Hand circumference	17.6	1.16	6.59	13.6 - 22.4	14.8	17.2	29.2
Eye height	145.3	5.05	3.48	133.5 - 158.2	135.2	145.6	156.4
Shoulder height	127.5	4.86	3.81	121.0 - 132.6	123.6	128.1	130.8
Elbow height	97.3	6.68	6.87	89.2 - 103.6	92.5	98.6	103.1
Metacarpal III height	63.7	2.34	3.67	58.8 - 70.1	60.6	64.3	68.4
Sitting height	76.9	2.45	3.19	68.6 - 81.4	70.3	77.2	80.1
Sitting eye height	65.2	2.10	3.22	60.0 - 71.5	61.5	65.0	69.6
Sitting shoulder height	47.4	2.62	5.53	42.8 - 51.6	43.5	46.9	49.8
Elbow rest height	17.8	1.38	7.75	14.1 - 20.2	15.7	17.7	18.0
Knee height	46.5	2.70	5.81	41.4 - 52.8	42.8	47.0	50.4
Popliteal height	39.8	2.14	5.38	36.0 - 41.5	37.1	39.6	40.2
Knuckle height	64.5	3.82	5.92	58.2 - 68.8	60.5	64.1	66.3
Buttock knee length	50.4	2.40	4.76	44.2 - 60.8	45.9	50.7	56.8
Buttock popliteal length	42.3	2.07	4.89	36.6 - 44.8	37.5	42.6	43.5
Functional leg length	91.8	3.81	4.15	85.4 - 104.2	88.7	92.0	101.8
Foot length	20.7	1.44	6.96	18.0 - 24.3	18.6	19.9	23.0
Shoulder elbow length	28.2	1.82	6.45	23.4 - 31.5	25.4	27.8	30.1
Fore arm-hand length	41.4	1.95	4.71	39.2 - 47.6	40.6	41.8	44.5
Hand length	16.5	1.28	7.76	14.5 - 19.1	15.5	17.2	18.4
Hand breadth	7.8	0.76	9.74	7.0 - 8.2	7.1	7.9	8.1
Arm reach from wall	76.4	3.63	4.75	66.8 - 90.2	70.5	76.9	82.1
Forward grip reach	66.7	4.02	6.03	59.7 - 81.1	62.8	67.0	71.5
Thigh clearance	9.2	0.96	10.43	8.0 - 14.5	8.8	10.0	12.4
Hip breadth (sitting)	36.4	2.10	5.77	27.2 - 40.0	30.6	37.1	38.6
Shoulder breadth	40.5	2.41	5.95	32.6 - 45.8	37.1	40.0	42.7

Note: Measuring unit is cm unless otherwise specified.

male anthropometric data in ergonomic designing is availability of ready data. The number of anthropometric surveys carried out in the country is very small and the dimensions included were specific to the requirement and specific to male workers only (Sen, 1964; Gupta et al, 1983; Gite and Yadav, 1989; Fernandez and Uppugonduri, 1992; Yadav, 1995; Yadav and Tewari, 1996; and Yadav et al, 1996). Yadav et al (1996) pointed out that there was considerable difference between the anthropometric data of Indians and Westerns. Yadav et al (1997) also highlighted the necessity of a cab for Indian tractors in order to increase export level as well as operator's efficiency. Variation was also reported within Indian males' data from different regions. Sen et al (1977) observed that the anthropometric dimensions of the general industrial workers (53% of which were agricultural workers) were very similar to those of other industrial workers. This paper reports an anthropometric data of Indian female agricultural workers as a reference for the ergonomic design and also help in modifications of agricultural equipment and machineries.

Methodology

Thirty anthropometric measurements were carried out which were considered useful for farm equipment design on 40 female workers who were chosen randomly among agricultural workers from Junagadh, Gujarat (India), including female students of the College of Agricultural Engineering and Technology at Gujarat. In order to measure the various body dimensions of the subjects, an anthropometer, a grip-size measuring device, a grip strength dynamometer and a bathroom weighing scale were used. The standard anthropometric definition of measurements and techniques were adopted from Damon et al., (1966), Pheasant (1986), Reobuck et. al., (1975) and Hertzberg (1968). The female observers were given enough practice to measure all the dimensions in a correct posture and in a precise manner. The subjects were asked to stand on the platform of the anthropometer, its arm was adjusted according to her height and measurement was recorded from the vertical scale. Similarly, other measurements were recorded in sitting and standing postures with the help of an anthropometer. The grip diameter was measured with the grip measuring device. The data recorded for the subjects was taken to be the mean of three readings.

Results and Discussion

Table 1 shows the various body dimensions and estimates of the mean, standard deviation, coefficient of variations, range and percentile values (5th, 50th and 95th). Slight variations are there between mean value and 50th percentile value. These female anthropometric data were compared with the Eastern, South, Central, Northern and Western Indian male workers as presented in **Table 2**. As evident from the Table, the male farm workers from the Northern region (Gupta et al., 1983) are taller and possess higher body weight than those of other regions of the country, including Indian female workers. The comparison of median values indicate that female Indians are smaller than male Indians in all body dimensions except hip breadth and waist circumference. The female anthropometric data were also compared with median

Table 2. Comparison of Indian Female Anthropometric Data and Those of Male Indians

Body dimensions	Female India	Eastern India [14]	Southern India [24]	Central India [34]	Northern India [44]	Western India [54]
Stature	154.6 (±6.18)	162.1 (±5.80)	160.7 (±6.00)	162.0 (±4.95)	168.5 (±6.84)	164.4
Body weight, kg	49.5 (±5.25)	53.6 (±6.73)	56.6 (±5.14)	49.3 (±5.95)	61.8 (±8.68)	54.7
Grip strength, kg	24.3 (±2.66)	30.2 (±5.45)	32.1 (±3.43)	36.2 (±10.4)	N A	N A
Grip diameter (inside)	3.8 (±0.22)	4.3 (±0.39)	N A	4.1 (±0.3)	N A	N A
Waist circumference	88.3 (±5.07)	81.3 (±4.87)	N A	83.1 (±4.48)	N A	84.3
Hand circumference	17.6 (±1.16)	19.2 (+1.40)	N A	19.9 (+0.80)	N A	N A
Eye height	145.3 (±5.05)	150.8 (±5.10)	149.7 (±6.1)	151.0 (±5.22)	N A	N A
Shoulder height	127.5 (±4.86)	131.2 (±4.80)	130.1 (±4.6)	134.6 (±4.87)	N A	N A
Elbow height	97.3 (±6.68)	101.88 (±3.80)	98.9 (±3.80)	102.6 (±2.89)	N A	N A
Metacarpal III height	63.7 (±2.34)	67.8 (±2.70)	N A	68.5 (±2.89)	N A	N A
Sitting height	76.9 (±2.45)	80.9 (±2.20)	79.1 (±4.00)	83.8 (±2.52)	N A	86.2
Sitting eye height	65.2 (±2.10)	71.4 (±2.00)	70.3 (±4.60)	73.9 (±2.62)	N A	N A
Sitting shoulder height	47.4 (±2.62)	53.4 (±2.12)	52.9 (±3.90)	55.7 (±2.08)	59.0	N A
Elbow rest height	17.8 (±1.38)	17.5 (±1.58)	15.4 (±0.60)	20.3 (±2.02)	21.0 (±1.86)	N A
Knee height	46.5 (±2.70)	51.5 (±2.87)	54.2 (±3.80)	50.9 (±2.70)	N A	N A
Popliteal height	39.8 (±2.14)	42.0 (±1.74)	47.1 (±3.5)	41.6 (±2.07)	41.8 (±2.60)	42.0
Knuckle height	64.5 (±3.82)	68.8 (±3.95)	68.0 (±5.3)	N A	N A	N A
Buttock knee length	50.4 (±2.40)	55.1 (±3.10)	51.6 (±2.90)	55.6 (±2.10)	N A	54.6
Buttock popliteal length	42.3 (±2.07)	46.2 (±2.28)	44.7 (±2.3)	46.6 (±1.75)	48.3 (±3.17)	45.6
Functional leg length	91.8 (±3.81)	97.6 (±4.24)	N A	96.9 (±5.51)	N A	N A
Foot length	20.7 (±1.44)	23.9 (±1.24)	21.9 (±2.00)	25.0 (±1.00)	N A	N A
Shoulder elbow length	28.2 (±1.82)	30.2 (±1.80)	29.6 (±2.20)	N A	N A	N A
Forearm-hand length	41.4 (±1.95)	44.6 (±1.96)	40.1 (±2.50)	45.9 (±2.00)	N A	N A
Hand length	16.5 (±1.28)	17.8 (±1.61)	16.4 (±1.40)	18.3 (±0.84)	N A	N A
Hand breadth	7.8 (±0.76)	N A	N A	8.3 (±0.36)	N A	N A
Arm reach from wall	76.4 (±3.63)	82.8 (+3.90)	N A	83.1 (+3.91)	N A	82.8
Forward grip reach	66.7 (±4.02)	73.7 (±4.46)	73.2 (±3.90)	70.9 (±3.21)	N A	N A
Thigh clearance	9.2 (±0.96)	12.9 (±2.24)	12.5 (±0.60)	13.4 (±1.32)	N A	N A
Hip breadth (sitting)	36.4 (±2.10)	31.7 (±2.30)	33.4 (±2.10)	30.8 (±1.56)	31.3 (±1.99)	N A
Shoulder breadth	40.5 (±2.41)	42.3 (±2.60)	N A	N A	N A	N A

Note: Measuring unit is cm unless otherwise specified N A.

[14] Yadav et al (1996), [24] Fernandez and Uppugonduri (1992), [34] Gite and Yadav (1989) [44] Gupta et al (1983), [54] Sen (1964).

values of three different ethnic groups as presented in **Table 3**. This comparison indicates that the female Indians are very smaller than German, USA and Japanese males, in all body dimensions. This variation could be due to the differences in body build-up of Indian and Western females. Comparison of sitting height to stature ratio of female Indians with different population is presented in **Table 4,** which shows almost similar ratio of stature to sitting height among different populations. The ratio between stature and other body dimensions were also calculated and compared with the ratio given by Barkla (1961) for the British population (**Table 5**). According to Murrel (1975), there is a high probability that whatever the mean stature of a sample, any given body dimension of length will be very nearly a constant proportion of the stature. Thus, if the stature is known, dimensions not available in the sample can be observed by proportion.

Ergonomic Approach in Farm Equipment Design

Manually operated equipment are extensively used in Indian agriculture for various farm operations like digging, weeding and harvesting. Nag et al (1988) analysed the effect of sickle design on manual harvesting and the harvester. The performance of the study was justified by the claim that manual harvesting is a moderately heavy task which requires the worker to adopt many awkward postures. The design of the handles of these tools depends on the mode of operation, amount of effort required, and the anthropometric data of the working population, including female workers. The internal grip diameter for 5th percentile female Indians is 3.8 cm while that of male Indians varied from 4.1 to 4.3 cm. For the design

Table 3. Comparison of Female Indian Anthropometric Data with Three Different Ethnic Groups

Body dimensions	Indian	German[1*]	US[2*]	Japanese[3*]
Stature	154.6	174.5	175.5	165.8
Shoulder height	127.5	146.4	143.5	134.5
Sitting height	76.9	91.9	91.3	90.4
Sitting eye height	65.2	80.2	79.9	78.5
Elbow rest height	17.8	23.7	23.2	26.0
Popliteal height	39.8	45.4	43.1	40.2
Buttock leg length	91.8	N A	108.5	N A
Foot length	20.7	26.0	26.7	N A
Shoulder elbow length	28.2	N A	36.4	N A
Forearm hand length	41.4	N A	47.8	N A
Arm reach from wall	76.4	N A	84.6	N A
Thigh clearance	9.2	15.1	14.3	N A

Note: Measuring unit is cm unless otherwise specified.
[1*]Jurgens et al (1972), [2*]Hertzberg et al (1954), [3*]Yokohori (1972).

Table 4. Comparison of Sitting Height to Stature Ratio with Different Populations

Ethnic group (men)	Ratio	Source
Female Indian	0.4974	Present study
Eastern Indian	0.4991	Yadav et al (1996)
South Indian	0.4922	Fernandez and Uppugonduri (1992)
Central Indian	0.5173	Gite and Yadav (1989)
Western Indian	0.5243	Sen (1964)
German	0.5266	Jurgens et al (1972)
U S	0.5202	Hertzberg et al (1954)
Japanese	0.5452	Yokohori (1972)
South Eastern African	0.5096	Jurgens et al (1990)

Table 5. Some Body Dimensions Expressed as Proportions of Stature

Body dimensions	Proportion of stature	
	Barkla (1961)	Present study
Eye height	0.936	0.940
Shoulder height	0.811	0.825
Elbow height	0.608	0.629
Sitting height	0.525	0.497
Sitting eye height	0.477	0.412
Sitting shoulder height	0.340	0.307
Knee height	0.315	0.301
Popliteal height	0.245	0.257
Buttock knee length	0.342	0.326
Buttock popliteal length	0.280	0.274
Fore arm hand length	0.272	0.268
Hip breadth (sitting)	0.205	0.235
Shoulder breadth	0.260	0.262

Note: Measuring unit is cm unless otherwise specified.

of a handle, its diameter should not exceed the internal grip diameter. For animal drawn equipment the handle is one of the most important components with which the operator controls and guides the implement properly during field operations. If the height is too low, say, 61 cm, the operator has to bend excessively which strains the operator. If the height is greater, say, 93 cm the maneuverability of the implement is affected and operation will not be proper. The elbow height (standing) data is helpful for designing proper handle height. Elbow height (standing) for the 5th percentile female Indians is 92.5 cm which is less than those of the Indian male 5th percentile standing elbow height (ranged from 98.9 to 102.6 cm). The optimum handle height, however, depends upon type of equipment and will have to be determined separately through experiments. Similarly, proper power tiller handle height is also based on anthropometric data of the target population. Adjustable handle height is preferred but is not always practically possible.

During a grain thresher operation, a worker feeds the crop material in the thresher in standing posture. The maximum permissible height for the feeding chute should be decided considering the 5th percentile value of shoulder height. The placement of different controls in a tractor is a complex task for the designer and requires the anthropometric characteristics of the target population. The efficiency and comfort of the operator can be improved with properly designed tractor workplace. The dimensions of seat, location of controls and access/exit provisions are the parameters where anthropometric data can provide help in matching the workplace according to the user's capabilities and to the physiological reach of the operator. For design purposes, either one of the boundary value (5th or 95th percentile) or the mean value is used depending upon the dimensional element. Anthropometrically, seat height from foot rest to suit female Indians 5th and 95th percentile population would be within the range of 37.0 to 40.0 cms. While in the case of male Indians would be within the range of 41.6 to 47.1 cms. If the equipment is to be operated by women, the anthropometric data of female must be considered in the design along with men anthropometric data. The weight range of female Indians is 38.4 to 68.2 kg (5th & 95th percentile values). Therefore, female Indian agricultural workers anthropometric data may be useful in design of agricultural equipment and machines.

Conclusions

Ergonomics is still a new concept in Indian agriculture. This survey presents a useful compilation of anthropometric data of Indian female workers which could be used as a guide to designing and modifying the agricultural and industrial equipment suiting to the human capabilities and limitations of Indian population. In the case of large variations in the size, adjustable designs could be created and the anthropometric data could be used for designing the range of adjustment to be provided. It strongly recommended that extensive surveys be carried out by concerned parties in different regions of the country in order to generate necessary data for both females with males. The result should be useful for farm and industrial equipment designs.

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Entrepreneurship in Mechanized Agriculture Technology-Oriented Operations

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Abstract

The future prosperity of Africa's economies depends on sustainable growth in agricultural production and productivity coupled with protection of its natural resources and environment. Success in this will require employment of engineering technologies as major inputs and instruments for promoting growth at all levels of mechanization. This paper highlights the need and opportunities for promoting agriculture-led industrialization through mechanized-technology oriented entrepreneurship and argues that in order to achieve sustainable broad based improvement in agricultural production system, concerted effort and action must be put in place. Possible framework and areas for intervention for sustainable agricultural production and processing operations have been proposed. In general, the appropriate choice and subsequent proper use of mechanized inputs into agriculture has a significant effect on achievable levels of agricultural production and processing operations, the profitability of farming and the environment.

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Introduction

Most of the developing countries and, indeed Sub-Saharan Africa (SSA), have an economy strongly dominated by the agriculture sector which generates over 50% of GNP contributing to over 80% of trade in value; more than 50% of raw materials to industries and provides employment for majority of its people (Rukuni, 1995; Kaumbutho et al. 1995). Despite this domination backed with good policy documents and statements, agriculture in most of SSA countries is still underdeveloped. For example, in Tanzania, only 20% (i.e., 7 million ha) of the estimated 40 million ha of cultivable land is presently being cropped. Furthermore, over 30 - 40% of agricultural produce is still lost due to poor processing and storage methods (Hatibu and Simalenga 1991, 1993). There is, therefore, a high potential for lateral expansion of the agriculture sector at all levels. Low level of engineering technology inputs in agriculture has been cited as one of the main constraints hindering the modernization of agriculture and food production systems in Africa.

In its major review of economic development in Sub-Saharan Africa, the World Bank has noted that the transformation of economies of SSA requires transformation of agriculture (IBRD 1989). The sector must not only produce adequate food for a rapidly growing population (currently estimated at 3.1 percent per year) but also increase production of export crops in an increasing competitive market, and continue to be a major employer of an increasing labour force. The challenge of transforming African agriculture as cited by the World Bank report includes:

- Expanding its productive capacity for improving the living standard and achieving food security; and
- Increasing food production which has to grow at about 4 percent a year (from the current less than 2 percent in order to meet the demand of population growth) and to enable to raise incomes and meet Africa's import needs

The seriousness of the challenge facing African countries is reflected in **Figure 1**. Aggregate cereal demand and supply balances for African countries, assuming a continuation of present trends in economic, agricultural and population growth, show an increase in the required cereal imports from the current 9 million metric tons to 27 million metric tons by the year 2020 (Badiane and Delgado 1995).

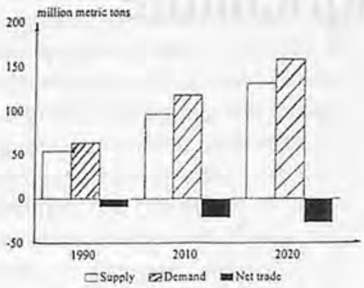

Fig. 1 Projection for cereal supply, demand and net trade in SSA (Source: Badiane and Delgado, 1995).

Thus Africa must set its target for long term sustainable agricultural growth of not lower than 4% a year. This will not be an easy task. During the last 30 years, agricultural production in SSA has risen by only 2% a year. Farm exports have declined, and food imports are increasing at about 7% a year (IBRD, 1989). Despite the rapid growth in food imports, severe food shortages are widespread; drought and famine have been common over the years and this damaging background of chronic food insecurity clearly shows that Africa's future is at stake! Therefore, the best minds must be put to work, the best policies and practices must be sought and a new sense of urgency must drive efforts on every level to accomplish the task.

One way of enhancing agricultural production is to encourage the use of agricultural mechanization technologies with its associated agro-processing industries. Despite the worldwide developments in agricultural machinery and implements and the increasing use of Draft Animal Power (DAP), agriculture in SSA is still carried out with old handtool technology (**Table 1**). Enhancing the use of agricultural mechanization technologies in agricultural operations is a crucial factor for increasing both the production and productivity of agriculture in the region.

A lot of information is available on various types of appropriate technologies, mechanization systems and proven research results from research institutions, but there is very low adoption of these technologies by both farmers, manufacturers and entrepreneurs. (Simalenga et. al 1990; Simalenga, 1997;. Haan 1994; Bliek and Veldhuizen, 1993; Starkey, 1990). Something must be wrong somewhere! Either our African farmers and technicians cannot cope with advanced technologies or the environment/circumstances are not conducive and well set for the uptake. Simalenga (1995,1996) and Mkomwa et.al. (1994) have cited some of the basic reasons as :

- Inadequate access to technology for mechanized operations;
- Lack of entrepreneurship drive and motivation;
- Inadequate finances;
- Ineffective policies; and
- Lack of participatory approaches in technology development and transfer.

However, it is interesting to note that similar statements on the need for promoting the use of agricultural mechanized inputs in order to modernize agriculture have been repeated over and over again for a number of years. In 1967 for example, De Wilde made the following statement in reviewing agricultural development in Africa. " ...The urgency of improving and increasing the equipment of African agriculture cannot be denied. Unfortunately, all the experience of the past has provided warnings of difficulties and few concrete guidelines for a more positive approach have been provided. In many cases, for instance, it is difficult to determine whether mechanization has failed because it was uneconomic, or because it suffered from certain technical and managerial problems that could have been avoided or overcome.."

Almost 25 years later, 60 African mechanization experts, meeting in Zaria, Nigeria recommended the following: "... If agricultural mechanization is to succeed in Africa, then there is a need for all concerned from policy makers, planners, donors and farmers to understand the role and consequences of agricultural mechanization in the entire farming system. The understanding should be in all aspects: technical, socio-economic, environmental, etc by careful analysis of past mechanization attempts (both failures and successes). In this context over the last 50 years, too often, costly and avoidable mistakes have been made again and again in agricultural mechanization by both national governments and donor agencies. Further, failed agricultural mechanization schemes have often been subjected to cursory economic reviews without detailed technical analysis required

Table 1. Sources of Power of Primary Land Preparation (% of Cultivated Land)

	Human Power (%)	Draft Animal Power (%)	Mechanical Power (%)
Sub-Saharan Africa	80	16	4
Botswana	20	40	40
Kenya	84	12	4
Tanzania	80	14	6
Zimbabwe			
South Africa	10	20	70
India	18	21	61
China	22	26	52

Source: COMSEC (1992)

Table 2. Small-Scale Enterprises in Zimbabwe and Distribution of Projects and Loans by Business Sector

Sector	Per Business Sector (%)	Distribution of loans (%)
Commercial	76.9	55.8
Industrial	16.0	19.6
Service	5.7	2.7
Construction	0.5	3.1
Agriculture	0.1	0.04
Other	0.7	18.8

Source: Matare (1993).

and at the same time, success cases have not been adequately substantiated and publicized.." (COMSEC 1991).

It is therefore, the objective of this paper is to provide some ideas of transforming SSA agriculture and discuss critical issues to be considered for sustainable agricultural mechanization and processing operations.

Key Issues in Agricultural Mechanization

Agricultural mechanization has been defined in a number of ways by different people, but perhaps, the most appropriate definition is "the process of improving farm labour productivity through the use of agricultural machinery, implements and tools." Mechanization, as Bishop and Morris (1992) clearly put it, is a key input in any farming system. It involves the provision and use of all forms of power sources and mechanical assistance to agriculture, from simple hand tool, to animal draft power and mechanical power. As a major agricultural production input, mechanization aims to achieve the following:

i) Reduction of drudgery in farming activities;
ii) Improvement of timeliness of farm operations;
iii) Accomplishment of tasks that are difficult to perform without mechanical aids; and
iv) Improvement of the quality of work and products.

One of the major reasons for the disappointing performance and contribution of agricultural mechanization to agricultural development has been the fragmented approach to mechanization issues. (Bishop and Morris 1992; Rijk 1989; Mrema and Odigboh, 1993). This often arises from poor planning and over-reliance on unpredictable or unsuitable aid-in- kind for many mechanization inputs and limited co-ordination within and between government and private sector agencies dealing with mechanization. Formulation of national agricultural mechanization strategy is, therefore, necessary where a holistic or system analysis approach is used in planning process which must consider all key players in the context of economic and cultural environment in which development is taking place. In most of Sub-Saharan Africa, no serious planning for sustainable mechanization has taken place. In many cases where mechanization has made a positive contribution to agricultural development, it has been by chance and not by design (Muchiri et.al. 1994).

Most of the African economies are now going through a very dynamic period as it changes from the confined socialist/centralized control economy to a more liberalized market economy. In a free market economy, the supply of mechanization inputs is demand driven, whereas in a centrally planned economy it is supply driven. Therefore, in a true free choice situation, governments should not make policies which stipulates by which means or by how much, agriculture will be mechanized. The type and degree of mechanization will have to be decided by the producer to best suit his/her business and meet his/her client's own particular circumstances. These changes will eventually lead to a situation where the individual will be the focus of policy, plan and development of most countries.

For these reasons, the individual will need to be more enterprising and self-reliant. And, therefore, one way of enhancing agricultural production through the use of mechanized inputs is to encourage trained professionals, especially agricultural engineers, to be directly involved in agricultural production with associated agro-processing industries. This will promote agriculture-led industrialization through enhanced entrepreneurship in agricultural operations.

Most of the current programmes in the region puts emphasis and support other small businesses-like traders, with little emphasis on agricultural technology oriented entrepreneurship. In this regard, there is a need to establish a mechanism in the region which will support and promote agricultural mechanized technology-oriented entrepreneurship in the region. Experiences in promoting small scale enterprises in Zimbabwe have shown that out of 1811 projects, a total of 77% are in the commercial sector and are mostly general dealers (traders) and do not require any specific skills and are relatively easy to manage. Some 16% of the projects are industrial projects dominated by grinding mills, metal fabrication plus a random distribution of bakeries. In nature, most of these projects are simple which do not require a high degree of technology, skill and capital. The third in line is service oriented projects at 5.7%. Transport business tends to dominate this sector. As can be seen in **Table 1**, agricultural-oriented technology entrepreneurship is still not developed with only 0.1% in business with loan assistance of only 0.04% (Matare, 1993).

Therefore, in order to achieve a

sustainable mechanization level in agricultural production and processing, Africa needs its own crop of entrepreneurs to seize the market and technical opportunities of the twenty-first century. The entrepreneurial potential of the African farmers, agricultural engineers, artisans and traders should be enhanced systematically as proposed in the following chapters.

Factors Limiting Mechanized Agriculture-Technology-Oriented Entrepreneurship

The following are some of the major factors limiting Agricultural oriented entrepreneurship in the Sub Saharan Africa:

1.Lack of initiative from agricultural engineers (professionals) to start up business (mainly due to conventional training and aspirations)

The nature of current training programmes in agricultural engineering in most higher learning institutions makes the grantes aspire to be employed rather than be self-employed. The situation is totally different with other professionals such as civil engineers or pharmacists who immediately, after graduating, are direct involved in their professionals by starting their own consulting firms and constructing companies or medical stores. The highly trained agricultural engineers (the professionals) totally lack business acumen, self-drive and confidence to start up agricultural technology oriented enterprises. There is a need to re-address our training approach for future generations.

2. Lack of adequate awareness (technology information availability and inadequate dissemination of research results)

Lack of relevant technical information in agriculture-technology oriented entrepreneurs i.e. information related to production processes, product designs, tools and equipment, is partly due to the passive attitude of not actively seeking such information. On the other hand, this is caused by ineffectiveness of technology institutions conducting research and development and negligence on part of suppliers of equipment to diffuse such information.

Small-scale producers have been found to rely largely on information from relatives and friends. They seldom visit official dealers of relevant equipment and do not consult technology institutions. A survey carried out by FIT (Haan, 1994) in Ghana, found out that only 10% of Technology-oriented entrepreneurs knew about the main technical institution in the country, Ghana Regional Appropriate Technology Industrial Service (GRATIS), while only 17% of the small entrepreneurs had heard about the national apex body, the National Board of Small Scale Industries (NBSSI).

3. Lack of effective policies and on entrepreneurship (infrastructure, taxation system etc.)

Existing physical facilities and local infrastructure such as power and water supply, roads and telecommunications are inadequate in most countries and are often a major bottleneck that limit the access to and transfer of new technologies to the small producers of the informal sector.

The Promotion of mechanized technology-oriented enterpreneurs should, therefore, go hand in hand with the development of other sectors in the national economy. It is also desirable to put in place deliberate policies such as reduction of customs tariffs on imported raw materials, low import duty and sales taxes in order to promote the technology-oriented enterpreneurs.

The governments role in informal sector development should always be to provide an enabling environment for the entrepreneurs to operate profitably. In Japan, for example, deliberate efforts and policies were formulated in 1956 to encourage and support small scale enterprises by: removing transaction constraints, facilitation of financial resources, provision of tax incentives and development of technologies (Kagami, 1995).

4. Inadequate co-operation between private sector and R&D institutions

Some of the reasons why research and development (R&D) activities and technology dissemination have not been very successful in relation to micro and small enterprises have been reported by Kyerematen 1993; Gamser and Jones, 1987. as follows:

i) R & D institutions set research goals without any comprehensive needs survey and analysis of the socio-economic development requirements, and particularly the needs of the private sector.

ii) Most R&D institutions end up with research only. No tangible results or little product development for commercialization.

iii) They tend to concentrate on the development of new technologies while ignoring the crucial needs for assimilating and adapting existing, especially imported technologies.

Most of the technology development projects often contain plans which stop with the successful operation of a pilot project. A pilot project, by its very nature, has a very limited impact. If the benefits which may be generated by a technology are to be shared more widely, and if the organization which has developed the technology is to get a return on its investment in research, then the use of the technology has to extend beyond the pilot project. Dissemination or diffusion must occur. The success of any R&D project should be: "how many en-

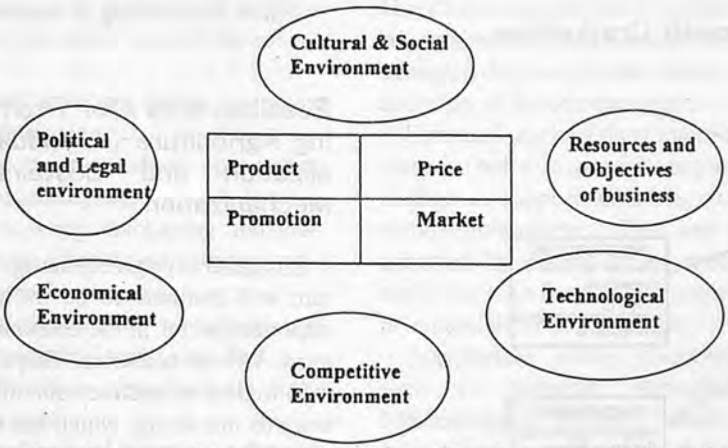

Fig. 2 The four controllable market variables within the uncontrollable environment.

trepreneurs have taken up the developed technology for commercial purposes".

5. Funding mechanism and high interest rates

It is clear that the small scale enterprises have for a long time been prejudiced by economic policies which have explicitly given privileged access to key resources to well established enterprises. Even under the current ESAP/ERP programmes, many of the restrictions have been impinged on the small scale sector. Allocation of finances to Small Scale Enterprises (SSEs) have already become more difficult. In fact the overwhelming majority of SSEs still remain outside of the banking systems when it comes to loans and they feel the ripple effects of the credit squeeze when it comes to reduce demand for goods and services.

The high cost of borrowing money from formal credit institutions up to 45% interest rates against the loaned sums presents problems even to those that succeeded in securing loans. This has resulted in a lack of adequate and appropriate financing for the agricultural-led industrialization. Without capital, the potential for mechanized agriculture cannot be harnessed. Thus the demand for sustainable mechanization is to establish an appropriate institutional financial framework which would enable private banks to concentrate on agriculture with special emphasis on financing the emerging mechanized-technology oriented entrepreneurs.

Table 3. Perceived Problems of MSEs, Zimbabwe Case Study (Unit%)

Perceived Problem	At Start-up (%)	During growth period (%)	Currently (%)
Market problems	23.3	24.0	20.5
Finance problems	31.9	20.0	25.3
Stock/raw materials	18.4	8.0	27.4
Tools/Machinery	7.2	16.0	7.0
Miscellaneous problems	9.8	10.0	4.6
Govt/regulatory problems	3.7	4.0	3.2
Location/space problems	2.5	10.0	4.9
Transport problems	1.2	6.0	4.8
Labour difficulties	0.6	2.0	1.7
Utilities problems	0.9	0.0	0.9
Total	100	100	100

Table 4. Perceived Problems of Micro Entrepreneurs at Time of Start-up (Reported as Percentage of Enterprises Reporting Problems)

Item	Botswana	Swaziland	Malawi	South Africa
Working capital, credit and finance	52.0	51.2	41.0	35.3
Markets and demand	16.7	21.6	23.5	28.1
Inputs, tools and machinery	6.0	9.0	13.5	6.9
Taxes, licences and other government regulations	2.0	4.0	5.5	10.9
Other problems	23.3	14.0	16.5	18.3

Source: Mead, 1994.

Market information

Marketing is a crucial factor in running any enterprise. Timing, packaging, storage, product quality and where to sell (market information) can all affect profitability of an enterprise. Marketing comprises of the knowledge (controllable variables) i.e., areas one needs to have answers before making decisions on the product itself, and the "Uncontrollable factors" which surrounds the whole business environment and can limit the profitability and expansion of the enterprise (**Fig. 2**).

In a country-wide survey of micro small-scale enterprises (MSEs) in Zimbabwe, McPherson (1991), came up with 10 broad categories of primary problems faced by their enterprises at three different points in time; when the enterprise was started, during periods of major growth (if any) and at the time the survey was conducted (**Table 2**).

The study showed that, when the enterprises began, 25% of the respondent reported having no prob-

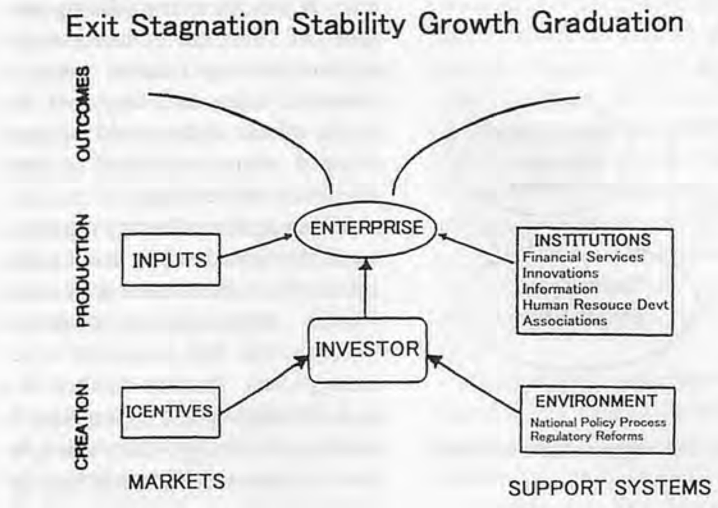

Fig. 3 Agents of change in entrepreneurship development. (After Steel, 1994)

lems. Of the group listing problems, 32% complained of finance difficulties. Within this group, 10% reported shortages of investment capital, while 22% cited operating capital constraints. Another 23% reported having market problems; in particular demand shortfall. Finally, 18% had difficulties involving raw materials or inputs, with most of these citing shortages rather than excessive expense.

In general, four categories of problems are most commonly cited in each time period: market problems, finance problems, difficulties involving stock or raw materials, and problems with tools and machinery. Not surprisingly, proprietors seem to be more constrained by demand and operating capital shortfalls, and by shortages of stock or raw materials when they are struggling to get their enterprises off the ground and at the current times than during times of rapid growth. It is also interesting to note that the four problems most frequently cited in **Table 2** above are the same as those cited by other entrepreneurs and small business people in Southern Africa (Fisseha and McPherson 1991; Mead, 1994). **Table 3** summarizes the major constraints at the start-up of mechanized technology oriented enterprises in 4 farm of Southern Africa.

Despite of an urgent need to promote agriculture production through mechanized technical oriented entrepreneurs, there is no quick and easy route to solve the above highlighted problems. However, efforts to improve the situation can be made on two fronts: improvement of business environment (policies, regulations on SSEs) and the provision of direct assistance to entrepreneurs (**Fig. 3**). The formation of private sector advisory groups that include representatives of government and private business might be a useful forum for reaching consensus on the changes that are needed.

As shown in **Figure 3** above, the forces of change can be found in six areas and may be grouped into a matrix with three columns:

1. The *Enterprise*, the central column which depends first and foremost on the investor.
2. The second column consists of the *Markets* in which firms operate: the incentives that determine costs and benefits (i.e. profits) and the inputs needed for the production process.
3. The third column is the *Support Systems* that are available to help enterprise solve problems and facilitate the working of markets.

Possible Areas for Promoting Agriculture-led Industrialization and Sustainable Mechanization

It is generally agreed that agriculture will continue to be the most important sector in the economy of most African countries. Therefore, it is logical to redirect our efforts towards this sector, which has been given lip service for a very long time. One way of promoting sustainable agricultural development is to promote private sector participation through entrepreneurs in mechanized agriculture operations and agro-business.

The following are some of the possible areas and opportunities for intervention in view of strengthening/initiating entrepreneurship in the region.

1. Manufacture of agricultural machinery, implements and parts

(With emphasis on small scale agriculture-led industries)

The main objective is to promote the manufacturing base of agricultural operations and processing technologies. The goal is to increase availability of agricultural machinery and equipment. Examples include:

a. Production of agricultural machinery;
b. Manufacture of tractor and animal drawn implements and spare parts;
c. Production of spraying and transport equipment;
d. Manufacture of crop processing, handling and packaging equipment (shellers, threshers, oil expellers etc.); and
e. Manufacture of fruit/vegetable processing and packaging equipment.

2. Agricultural processing industries and food technology systems

The main goal is to increase processing and storage capacity and marketing of agricultural produce. Some examples are as follows:

a. Fruit and vegetable processing and canning;
b. Livestock products processing and packaging;
c. Processing, packaging and marketing of commodity crops;
d. Animal feed production;
e. Apiculture (bee) products processing and packaging; and
f. Utilization of agricultural waste products.

3. Professional farming and provision of services to farmers and Agro-industries

The main objective is to develop scientific commercial and profitable farming for both local and external market which include:

a. Privately owned contract hire services for both tractor and draft animal power;
b. Horticultural crop production and processing; and
c. Training, business information and advisory services.

4. Marketing (supply) of technology-oriented hardware, inputs and services

Here the main goal will be to increase the supply of technologies, spares and services to farmers and agro-industries such as:

a. Trading, importation and distribution of farm machinery and other technologies;
b. Commercialization of R&D results e.g. renewable energy technologies, rice weeders; and
c. Provision for maintenance and repair services and supply of inputs and spares (dealership).

Conclusions.

As stated in previous sections, the problems for starting and sustaining mechanized technology-oriented entrepreneurs does not have a magic formula to solve them. It requires both re-orientation of donor assistance towards micro-enterprises, deliberate government intervention in creating enabling environment and a strong mechanism to make follow up and support the entrepreneurs.

Financial markets need major reform in order to provide required finance to support technology-oriented entrepreneurs. This can be achieved by setting up of special revolving loan fund or by provision of accessible credit facilities.

Appropriate policy framework such as National Agricultural Mechanization Strategy, needs to be in place so as to ensure that enabling environment is created and sustained to run business profitably. The strategy will also assist to identify the required infrastructure and other sectors of the national economy to be developed to support the agricultural development.

Education and training will need to be reviewed in line of the changing economic environment. Course such as business and capital management, and application of information technology in agri-business will have to be incorporate as well as entrepreneurship training in universities, colleges, and professional training courses.

A mechanism for information for business and market (networking) will need to be established for regional collaboration, promotion and publicity. The mechanism will also initiate the formation of data bank (especially for regional trade and tariffs, market opportunities etc). This will greatly assist regional networking for one-stop-information acquisition system.

The paper has attempted to show that agriculture is a viable economic activity which can enhance the quality of life and bring in added value from the use of natural resources. It thus deserves support in terms of investment. Development and modernization of Africa's agriculture and its agro-industries will depend to the large extent on the transformation of policies, education and entrepreneurship thrust. It is argued that for sustainable agriculture growth in the medium and long term, there is need to change policy from emphasis on small-scale peasant farmers to medium-scale African commercial farmers. Entrepreneurship and professionalism in agriculture are likely to be the main key to sustainable development of African agriculture.

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Tractor Workplace Design : An Application of Biomechanical and Engineering Anthropometry

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Abstract

Tractor-operator workplace design concepts are described from biomechanical and anthropometric approaches. The design parameters of the tractor workplace were determined using measured anthropometric dimensions from Eastern India adult male tractor operators. A comparison was made between the existing dimensions of five recent models of Indian tractors. The existing tractor workplace dimensions widely varied, which indicates the need for anthropometric dimensions to be considered in tractor operator workplace design to meet the physical needs of the tractor operators.

Introduction

Technical progress become profitable for both farmers and tractor manufacturers with successful tractor workplace development. In India a wide choice of competitive tractor models are now available to farmers, but the design changes in these tractors are not adequate due to economical reasons (Anon., 1990). Indian tractors need lots of improvements to get in line with world trends regarding operator workplace. Unlike highly sophisticated designs in Europe, North America and Japan (saturated markets), the two biggest nations of the world, China and India, could benefit from the use of medium technology in low power level tractor operation. The use of medium technology may lead to high annual growth in domestic sales (Renius, 1994). The design of modern tractor includes the consideration of human factors (Zander, 1969). These factors, when properly incorporated in design, allow the operator to perform many complex tasks with efficiency, safety, and a minimum of fatigue. Human factors include: riding, comfort, visibility, location and arrangement of controls, ease of operating controls, design for thermal comfort, sound and vibration control (Liljedahl et al. 1997, Pheasant, 1986 and Zander, 1969). Many factors need to be considered for the design of the tractor operator workplace. With the changing scenario there is need to provide new and efficient workplace designs of tractors.

The work space should not seriously hinder the movement of the operator in getting on and off the tractor and should allow easy and unhindered, access to various controls. Anthropometric dimensions are the initial data used to design the seat and tractor workplace parameters and these data should be only considered from user's population (Haslegrave, 1979 and Pheasant, 1986).

The relative positioning of brake and clutch pedals, foot rest, steering column and steering wheel, study was carried out by Lehmann (1958). He concluded that the optimal direction for applying pedal thrust was 70^{o} with vertical, with seat height of 200 mm, and the optimal lateral separation of legs for clutch and brake pedals varying between 80 to 120 mm. Gibben's (1970) survey of tractor operators indicated that access to the driving seat was awkward for one-third of the drivers. He reported that fatigue was experienced by 24% of the operators interviewed in the survey and approximately one-fifth of the drivers had medical complaints, back ailments being the most common. Improvements in

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tractor controls were suggested by 44 per cent of the operators.

Bottoms (1973, 83) reported optimum dimensions for a tractor seat and suggested that horizontal and vertical adjustments of the seat are necessary for variations in leg length and other tractor controls should be placed where they can easily be reached, and operated in such a manner that movement of the control will produce the desired movement of the vehicle. Matthews (1977) outlined the research methodologies and results that led to improvements in tractors by incorporating seat suspension to improve ride and better controls to reduce physical effort to operate. Moreover, these controls were better disposed within the workplace and better identified.

A systematic layout was described by Babbs (1979) to define the seating package and body reference points in vehicle design, as well as determining the adjustment envelope required to fit a given range of population size. The operator's anthropometric and biomechanical characteristics were emphasised by Liljedahl et al., (1997) in the design of tractor workplace. Anthropometric data of user population are only used for proper design of the operator's workplace to meet visibility and clearance functions. He also quoted the work done by Hasson et al., (1970) that safety, comfort and convenience must be considered in the design, location and construction of the operator's workplace. Tractor operator controls within the workplace should be conveniently and logically located and the workplace should fit both tall and short operators.

The behavioral and biomechanical aspects of the operator, viewed as a "comfort seeker" or as a discomfort by Oborne (1982). Keegan and Radke (1964) suggested that sitting posture which results the nearest approximation to the normal lumbar shape is one in which the trunk-thigh angle is about 115^{o} and the lumbar position of the spine is supported. Pheasant and Harris (1982) studied the biomechanical factors which influenced human strength in the operation of a pedal. The variables investigated were horizontal distances in front of the seat reference point (SRP), vertical distance above and below SRP, lateral distance from the midline, direction of thrust and the use of steering wheel for bracing.

A workplace design for the vehicle cab was proposed by Sanders and McCormick (1987). The important features were seat height, depth, and back angle; forward and backward adjustability of the seat; leg and knee clearance; location of hand and foot controls; and visual field (the driver should have a good view of the road and traffic environment). The brake pedal force for both leg positions (neutral and extended leg position) were greatest when the foot angle on the pedal was between 15^{o} and 35^{o} from the vertical, respectively. Grandjean (1988) suggested that the seats should be designed so that in both forward and backward sitting postures they provide support to the upper edge of the pelvis in order to assist in rotating the upper part. To accommodate tractor operators ranging from 5th to 95th percentile, it is easier to adjust the seat than to adjusting the controls. However, there must be a few compromises between hand and foot controls.

A study was conducted by Casey and Kiso (1991) in which 172 tractor operators from North America and Europe were involved in the evaluation of 69 different tractors produced by various manufacturers. Human factors and usability concerns were the primary focus of the evaluations. A knowledge-based system for optimum workplace design was described by Pham and Onder (1992). The system was constructed using a commercially available hybrid development tool. It was interfaced to a database of anthropometric data and an optimization programme. The optimization programme employed a genetic algorithm.

Yadav (1995) designed and developed a tractor operator workplace simulator for laboratory simulation of the existing Indian tractor workplace configurations. Control locations in existing Indian tractors were evaluated on the basis of ISO standards (ISO 4252, 1977 and ISO 4253, 1983). Special provisions were made to change the location of different controls such as seat, steering, foot pedal and hand controls. Studies were made in an environmental control chamber within the range of basic ride vibrations (0 - 20 Hz). Large variation was reported within the existing Indian tractors (Yadav, 2000). It was found that for Indian operators, the steering angle of 64.8^{o}, the foot pedals (clutch and brake) distance of 62.7 cm from seat reference point (SRP), and the hydaulic control lever distance of 16.7 cm from SRP were optimum values based on minimum energy expenditure rate (EER) and rated perceived exertion (RPE) scores (Yadav and Tewari, 1998).

The tractor seat and location of different controls are designed to accommodate the size and shape of at least 90% of the tractor driver population, based on the anthropometric data. Most Indian tractors are manufactured to suit the anthropometric measurements applicable to the countries where the tractors are designed (Anon., 1990).

Better design of seat and controls resulted in less tractor operator effort and stress (Dupuis, 1959). The efficiency and comfort of the operator is improved with properly designed tractor workplace (Lehmann, 1958; Bottoms, 1973, 83; Pheasant and Harris, 1982; Huang and Suggs, 1967 and Yadav and Tewari, 1998). The biomechanical and anthropometric design of tractor operator workplace is an effective method for improving operator comfort. Therefore, great emphasis needs to be placed on adapting operating controls to the physical needs of the human operator. Almost all the tractor operators in India are male.

Biomechanical Model of Seated Tractor Operator

The seated tractor operator was modeled as a biomechanical model consisting of straight links (representing bones) and joints (representing major articulations). A typical link-joint biomechanical model is presented in **Figure 1**. In this model hands and feet are not subdivided into their components, and the spinal column is represented by only one link. Clearly, such simplification does not represent the true design of the human body, but may be sufficient to represent mechanical properties. The determination of joint center-of-location is easy for simple joints, such as the hing-types in fingers, elbows, and knee. However, this is much more difficult for complex joints with several degrees of freedom such as in the hip or, shoulder.

Unfortunately, anthropometric dimensions are not measured between joints but usually between externally discernable landmarks, such as bony protrusions on the skeletal system. Hence, a major problem in developing a model depicting the human body is to establish the numerical relationship between standard anthropometric measure and link lengths. However, an attempt has been made to express different link lengths approximately in terms of anthropometric measurements (**Table 1**).

Placement of Tractor Operator Controls

Determining where the controls and displays should be placed within the tractor operator's workplace needs to be considered judiciously. Not only do aesthetics and styling need to be considered but also such factors as the operator's comfort and safety, proximity of the controls for easy limbs, avoidance of overloading the operator, ease of operator understanding and many other factors

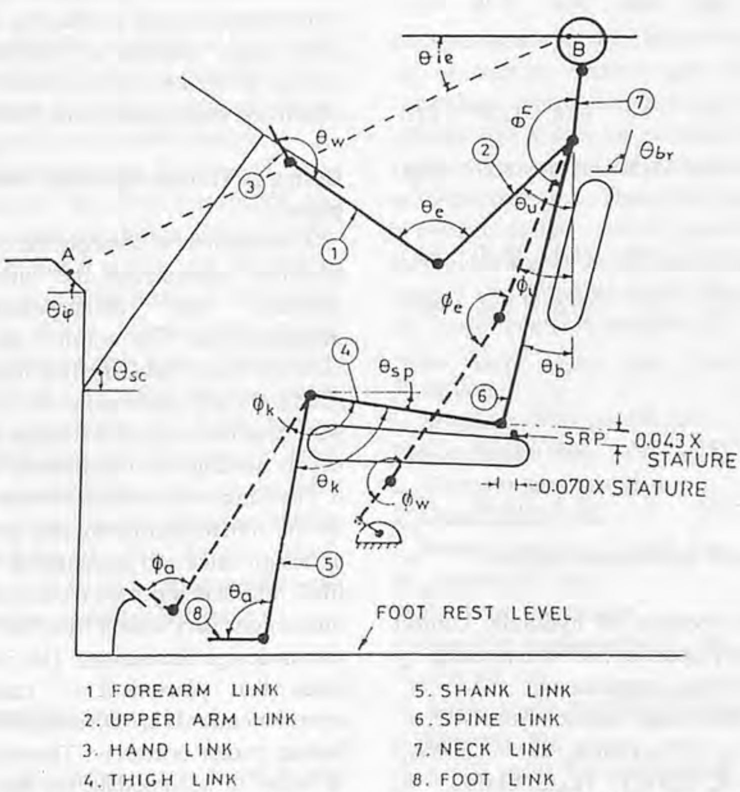

Fig. 1 Typical link-joint biomechanical model of seated tractor operator.

which are possibly not quantifiable. Human beings differ from each other in anatomical dimensions. In order to accommodate tractor operators ranging from 5th to 95th percentile, it is easier to adjust the seat than to adjust controls. However, there must be a few compromises between hand and foot controls.

Steering Wheel Position

The location of the steering wheel center is expressed by equations (**Fig. 1**):

Horizontal distance from SRP

$$= (L_{fa} - 0.5\ L_h)\ Sin(\theta_e - \theta_u) + L_e\ Sin(\theta_u) - H_s\ Sin(\theta_b) + (L_h/2)\ Sin(\theta_w + \theta_e - \theta_u - \pi) + 0.07\ H_{st} \quad ...(1)$$

Vertical distance from SRP

$$= H_s\ Cos(\theta_b) - L_e\ Cos(\theta_u) + (L_{fa} - 0.5\ L_h)\ Cos(\theta_e - \theta_u) + (L_h/2)\ Cos(\theta_w + \theta_e - \theta_u - \pi) + 0.043\ H_{st} \quad .... (2)$$

Foot Operated Controls

The clutch pedal is used very frequently. It has been observed that the force exerted on the clutch pedal is generally far greater than that which is necessary. To avoid this waste of energy, it seemed practical to fit the tractor with a bottom plate which is curved upwards in front in such a way that it provides a comfortable foot rest. Horizontal and vertical distance of foot control (brake and clutch pedal) from SRP in vertical plane are expressed as follows, considering the research finding and geometry of biomechanical model (**Fig. 1**):

Horizontal distance of foot control from SRP:

$$= L_p\ Cos(\theta_{sp}) + H_p\ Sin(\phi_k + \theta_{sp} - \pi/2) + (L_f/2)\ Cos(\phi_k + \theta_{sp} - \phi_a) + 0.07\ H_{st} \quad ...(3)$$

Vertical distance of foot control from SRP:

$$= - L_p\ Sin(\theta_{sp}) + H_p\ Cos(\phi_k + \theta_{sp} - \pi/2) - (L_f/2)\ Sin(\phi_k + \theta_{sp} - \phi_a) + 0.043\ H_{st} \quad ...(4)$$

Hydraulic Control and Gear Shift Lever

The hydraulic control and gear shift levers should be placed within arms reach of the tractor operator

Table 1. Range of Comfort and Angle Used in Design of Tractor Operator Workplace

Body member	Angle range, deg (Rebiffe,1969)	Angle used in design,deg	Comments
Back	(θ_b) 20 - 30	10	
Hips	(θ_h) 95 - 120	95	
Knee	(θ_k) 95 - 136	95	Foot resting on foot rest
	(ϕ_k)	115	Foot pedal operation
Ankle	(θ_a) 90 - 110	90	
	(ϕ_a)	90	
Upper arm	(θ_u) 10 - 45	45	For steering control
	(ϕ_u)	10	For hydraulic control
Elbow	(θ_e) 80 - 120	120	For steering control
	(ϕ_e)	165	For hydraulic control
Wrist	(θ_w) 170 - 190	170	For steering control
	(ϕ_w)	170	For hydraulic control
	(θ_{sp}) NA	3	
	(θ_{ls}) NA	25	
	(θ_n) NA	180	

Table 2. Placement of Tractor Operator Controls in Existing Indian Tractors

Parameters	T1	T2	T3	T4	T5	Mean	Cv*	Design
1 Steering wheel								
i) Steering column angle from horizontal, deg.	59.6	74.9	49.8	70.2	55.0	61.9	16.89	60.0
ii) Horizontal distance of steering wheel center from SRP, cm	60.5	52.5	61.9	67.3	60.2	60.5	8.76	56.2
iii) Vertical clearance of steering wheel center from SRP, cm	30.2	32.3	29.3	30.2	30.0	30.4	3.70	43.7
2 Foot control								
a) Clutch								
i) Horizontal distance from SRP, cm	75.9	52.5	64.5	59.0	67.1	63.8	13.77	63.3
ii) Vertical clearance from SRP, cm	-37.7	-32.4	-30.6	-32.6	-31.8	-33.0	8.27	-38.1
b) Brake								
i) Horizontal distance from SRP, cm	78.8	52.5	64.3	62.1	74.5	66.4	15.70	63.3
ii) Vertical clearance from SRP, cm	-38.3	-33.8	-30.4	-31.7	-30.7	-33.0	9.88	-38.1
3 Foot rest height from SRP, cm	-49.6	-48.6	-48.9	-50.2	-48.1	-49.1	1.66	-39.8
4. Hydraulic control lever								
i) Horizontal clearance From SRP, cm	11.2	11.5	19.5	29.6	20.0	18.4	41.18	23.0
ii) Vertical clearance from SRP, cm	-15.5	-17.8	2.2	-23.5	2.3	-10.5	114.24	-15.2

* Coefficient of variation.

Note: Distance measured below and rearward of SRP are considered negative.

with minimal change in his/her alert driving posture. It should specifically be ensured that the operator need not bend forward or sideways while operating these levers. The forward or sideways bending is likely to strain the spinal column. The location of hydraulic control with respect to SRP is expressed by following expressions:

Horizontal distance from SRP

$$=(L_{fa}-0.5L_h)\mathrm{Sin}(\phi_e-\phi_u)+L_e\mathrm{Sin}(\phi_u) - H_s\ \mathrm{Sin}(\theta_b) + (L_h/2)\ \mathrm{Sin}(\phi_w + \phi_e - \phi_u - \pi) + 0.07\ H_{st} \quad ...(5)$$

Vertical distance from SRP

$$= H_s\ \mathrm{Cos}(\theta_b) - L_e\ \mathrm{Cos}(\phi_u) + (L_{fa} - 0.5\ L_h)\ \mathrm{Cos}(\phi_e - \phi_u) + (L_h/2)\ \mathrm{Cos}(\phi_w + \phi_e - \phi_u - \pi) + 0.043\ H_{st} \quad ...(6)$$

Tractor Opeartor Workplace

Operator Workplace Dimensions of Some Existing Tractors

The tractor models which are under production and widely used in India were considered. The maximum pto power of the tractor models were 19.7, 30.1, 32.8, 40.0 and 32.2 Ps for T1, T2, T3, T4 and T5, respectively. Different dimensions of operator's workplace of five tractor models (T1, T2, T3, T4, and T5) from five Indian manufacturers were measured and compared. At the time of measurement the seat was positioned in the middle of its adjustment range, both vertically and horizontally and at the mid-point of its suspension travel. The comparison of dimensions and their mean and coefficient of variation are shown in **Table 2**. Wide variations are found in the case of dimensions such as steering column angle, position of hydraulic control lever, horizontal distance of clutch and brake pedal from SRP.

Design of Tractor Operator Workplace

The locations of controls were determined considering the biomechanical and anthropometric measurements. The comfort angle between links (**Table 1**), link length (**Table 3**) and anthropometric data were used to calculate the design values by making use of equations 1 to 6. The computed location of controls for the 50th percentile are also given as design value and presented in **Table 2**. The existing tractor workplace dimensions vary widely from the optimum design dimensions. This indicates that placement of tractor operator controls are not suitable for Indian tractor operators. Therefore, in order to accomodate the 5th to 95th percentile tractor operators,

Table 3. Length of Links in Terms of Anthropometric Measurement

Link	Link length in terms of anthropometric measurement (cm)
Forearm link	(Forearm hand length, L_{fa} - Hand length, L_h)
Upper arm link	(Shoulder elbow length, L_e)
Hand link	(Hand length, $L_h \times 0.5$)
Thigh link	(Buttock-popliteal length, L_p x 0.8)
Shank link	(Popliteal height, $H_p \times 0.8$)
Spine link	(Sitting shoulder height, H_s)
Neck link	(Sitting eye height, H_{eh} - Sitting shoulder height, H_s) $\times$ 0.5
Foot link	(Foot length, $L_f \times 0.5$)

(Unit : cm unless otherwise specified)

Table 4. Placement of Tractor Operator Controls and Seat

Parameters	Design value for percentile			Range
	5th	50th	95th	
1. Steering Wheel				
i)Horizontal distance of steering wheel center from SRP, cm	51.0	56.2	62.2	11.2
ii)Vertical clearance of steering wheel center from SRP, cm	42.2	43.8	44.6	2.4
2. Foot control				
a) Clutch				
i) Horizontal distance from SRP, cm	60.2	63.3	66.2	6.0
ii) Vertical clearance from SRP, cm	-35.8	-38.1	-40.5	4.7
b) Brake				
i) Horizontal distance from SRP, cm	60.2	63.3	66.2	6.0
ii) Vertical clearance from SRP, cm	-35.8	-38.1	-40.5	4.7
3. Foot rest height from SRP, cm	37.3	39.8	42.3	5.0
4. Hydraulic control lever				
i) Horizontal clearance from SRP, cm	23.6	26.1	29.2	5.6
ii) Vertical clearance from SRP, cm	-10.6	-13.7	-18.4	7.8

provision for vertical and horizontal adjustment in seat is desirable. This range of percentile is very commonly used in ergonomic design.

The computed location of controls for the 50th percentile are presented as design values in **Table 4** which shows that the difference in vertical distance of controls for the 5th to 95th percentile tractor operators stature varies from a minimum of 2.4 cm (for steering wheel) to a maximum of 7.8 cm (for hydraulic control). Therefore, in order to accommodate the 5th to 95th percentile tractor operator stature, provision of 7.8 (= 8) cm vertical and 11.2 (= 11) cm horizontal adjustment in seat is desirable.

Conclusions

Large variation was found in seat dimensions and the workplace control locations among different models of tractors under study. The workplace control locations of the tractors also widely varied from design locations suited to Eastern Indian tractor operators. Therefore, there is need to design a tractor operator workplace based on biomechanical aspects and anthropometric data of the tractor operator population.

Notations

L_e Shoulder elbow length, cm
L_f Foot length, cm
L_{fa} Forearm hand length, cm
L_h Hand length, cm
L_p Buttock popliteal length, cm
H_{eh} Eye height, cm
H_p Popliteal height, cm
H_s Shoulder height (sitting), cm
H_{st} Stature, cm
θ_a Ankle angle for foot resting on foot rest, deg
ϕ_a Ankle angle for foot resting on undepressed clutch/brake, deg
θ_b Back angle from vertical, deg
θ_e Elbow angle for operation of steering wheel, deg
ϕ_e Elbow angle for operation of hydraulic control lever in lowest position, deg θ_k Knee angle for foot resting on foot rest,deg
ϕ_k Knee angle for foot resting on undepressed clutch/brake pedal, deg
θ_u Upper arm angle from vertical for steering wheel operation,deg
ϕ_u Upper arm angle from vertical for hydraulic control lever operation in lowest position, deg
θ_w Wrist angle for steering wheel operation, deg
ϕ_w Wrist angle for hydraulic control lever position in lowest position, deg
θ_n Angle between neck and spine links, deg
θ_{sp} Seatpan angle from horizontal, deg
θ_{sc} Steering column angle with horizontal, deg
θ_{br} Back rest angle from horizontal, deg
θ_{ip} Angle of instrument panel from horizontal, deg
θ_{ie} Angle between line of sight for instrument panel from horizontal , deg

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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.

736

Tractor Workplace Design:An Indian Case Study: Rajvir Yadav, Associate Prof. College of Agril. Engng. and Tech,G.A.U. Campus Junagadh, 362001 INDIA, e-mail:fmp@gauju.guj.nic.in . V K Tewari, Associate Prof. Department of Agril. and Food Engineering, I.I.T, Kharagpur, 721302 INDIA.

An ergonomic study was conducted to evaluate Indian tractor workplace design. The design dimensions of tractor operator workplace designated as T1, T2, T3, T4 and T5 from tractor models TM1, TM2, TM3, TM4 and TM5, respectively, were measured and compared. Workplace dimensions like steering column angle with horizontal, foot pedal (clutch and brake pedal) horizontal distances from seat reference point (SRP), foot rest height from SRP and hydraulic control lever position with respect to SRP were measured. The coefficient of variation (CV) was found between 1.66 to 114.24% for these dimensions of the available Indian tractors. Uniformity should be maintained in placement of these controls to suit the operators capabilities and limitations and they should also be placed to the physical need of the operators.

812

Hydraulic Performance of A Raingun (PY_1-30) with 8-mm Diameter Nozzle: Muhammad Yasin, Dr. Shahid Ahmad, Senior Scientific Officer and Director, Water Resources Research Institute, National Agricultural Research Centre, Islamabad, PAKISTAN. Badruddin, Professor, Department of Agricultural Engineering, NWFP University of Engineering and Technology, Peshawar, PAKITAN. Asif Ali Bhatti, Senior Engineer, Water Resources Research Institute, National Agricultural Research Centre, Islamabad, PKISTAN.

A raingun (PY_1-30) with 8-mm diameter nozzle was operated under 13 different pressures ranging from 20 to 80 psi with an interval of 5 psi. Each set of data was replicated thrice. The water distribution pattern, discharge and radius of coverage were measured. The discharge of the raingun was 10.91 gpm at 20 psi pressure and 23.46 gpm at 80 psi pressure. The discharge increased with an increase in pressure. The radius of coverage was 17.5 m at 20 psi pressure and 24.5 m at 80 psi pressure. The radius of coverage increased with an increase in pressure but the increase in radius of coverage was not uniform nor regular. The relationships between pressure and discharge and pressure and radius of coverage were also determined.

The application rates of the rain gun nozzle under various pressures were determined. The maximum application rate of 3.14 mm/hr was 55 psi pressure and the minimum application rate of 2.27 mm/hr was 25 psi pressure. The relationship between pressure and application rate was also determined.

The coefficients of uniformity of the raingun were determined in four quarters by dividing the radius of coverage into four equal parts. The high coefficients of uniformity were found in the first quarter and the low coefficients of uniformity were found in the fourth quarter. The coefficients of uniformity were also determined at potential and effective radius of coverage.

814

Enhancement of Oil Extraction of Shea Butter Using an Optimised Dilution/Clarification Protocol, Nigeria: Olaoye, Joshua Olanrewaju, Agricultural Engineering Department, The Federal Polytechnic P. M. B. 55 Bida, Niger State, Nigeria.

The local method of extraction of shea butter oil was critically examined. Materials handling during processing procedure was identified as the major cause of poor oil quality; while poor oil recovery is due to poor clarification techniques. Experimentation was carried out on the effects of dilution rate and setting time on oil recovery in a two to power two factorial experiment. Results indicated an optimum dilution rate of 3:1 (3parts water to 1 part oil meal) for best oil recovery. The oil recovery for the new clarification dilution protocol showed a significant increase of 25% in oil recovery when compared with the traditional procedure.

816

Effect of Shade-drying and Solar-treatment on the Proximate Composition of Drought-resistant Cassava Plant: Dr. Anigbogu N. M., Dept of Animal Nutrition, Feed Milling Technology, Federal University of Agriculture, Umudike, P. M. B. 7267, Umuahia, Abia State, Nigeria.

The trial on the effect of processed and fresh samples on the proximate composition of drought-resistant cassava leaf stem and other principal components in Nigeria was studied. There were significant differences in proximate composition between processed and fresh samples ($P = 0.05$). In the trial samples, the level of protein in cassava leaf stem feed (for goats and sheep) was 52.8% higher over other principal components. The same trends were not observed in crude fiber, and nitrogen-free extract values for both over the leaves alone.

The levels of nitrogen free extract in cassava leaf stem feed was 22.84% higher than in processed leaves alone and 87.31% higher in processed tuber with peels. But in fresh cassava components, leaf stem feed was 86.12% to 91.94% over fresh tuber with peels and leaves, respectively. The ether extract contents and caloric value of the cassava leaf stem feed over other principal components was higher by 22.84% to 87.32% and 0.98% to 3.18%, respectively, in dry basis.

818

Application of Power Tiller in Agro-forestry, India: A.C.Varshney, Suresh Narang, Principal Scientist and Senior Scientist, Central Institute of Agricultural Engineering, Nabi bagh, Bhopal, India 462038

The traditional system of operating farms on agro-forestry with the help of human and animal power are time consuming and labour intensive, hence there is need to mechanize agro-forestry cultivation. Power tillers have a good potential for use in agro-forestry due to their narrow size, low centre of gravity and less turning radius. With this consideration, power tiller and matching equipment were used agro-forestry farming under rainfed conditions in Bhopal region of Madhya Pradesh. Subadul (Leucaena lcucocephala) as forest plantation and soybean, wheat, groundnut and pigeonpea as food crops were selected in agro-forestry system in view of their potential in the region. Power tiller trailer, pit digger, rotary tiller, seed-cum-fertilizer drill and cultivator can be successfully used for performing various operations in agro-forestry farming. The use of power tiller system was compared with traditional system of cultivation of soybean and wheat crops for four years. Subabul, plus groundnut-wheat combination gave a maximum profit of Rs.22,434.00 per hectare using power tillers after four years while the total net profit of Rs.18,345.00 per hectare was obtained from soybean and wheat food crops under traditional system.

820

Appropriate Solar Technology for Developing Countries: An Illustration for Candle Production: Dr.P.B.L.Chaurasia, Senior Scientist, Division of Agricultural Engineering and Energy Management, Central Arid Zone Research Institute, Jodhpur-342003, India

A simple solar technology has been developed and illustrated for candle production. The candle production by solar technology is a convenient process and has numerous advantages over the conventional candle making methods. This solar system generates employment among the rural people and suitable for developing countries. ■■

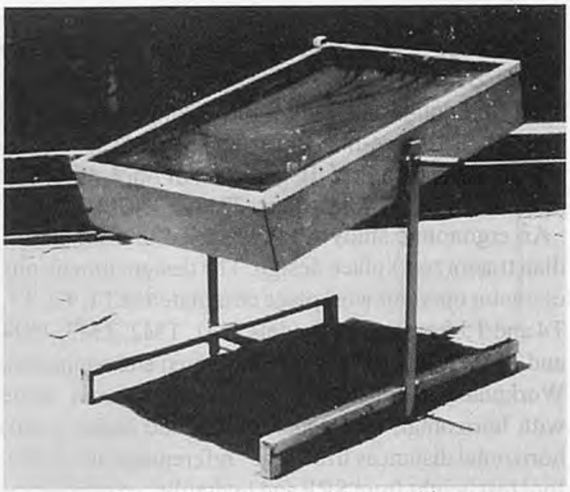

Fig. 1 Solar candle making machine.

Fig. 2 Back view of solar candle machine.

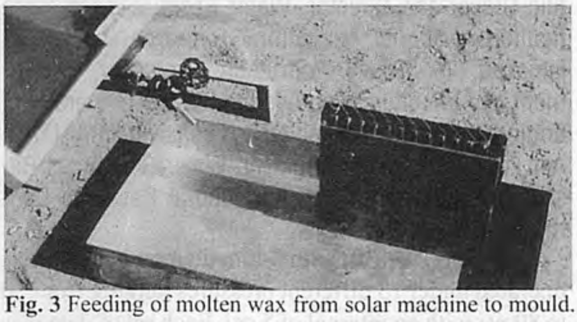

Fig. 3 Feeding of molten wax from solar machine to mould.

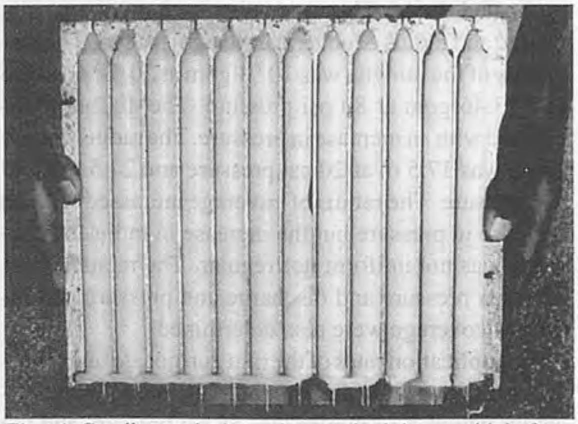

Fig. 4 Candle made by solar method in mould before detachment.

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Genetic algorithms, Fuzzy methods, Neuro-fuzzy methods, Post harvest operations, Control for precision farming, Harvesting systems, Quality and storage automation, Food processing control, Ergonomics, Instrumentation and measurements, Robotics for bio-production and bio-systems.

For further details, see:

http://www.creata-ipb.com/icaaa-ifac

CAPPT 2001-3rd IFAC/CIGR Workshop on Control Application in Post-Harvest and Processing Technology

October 3-5, 2001
Tokyo, Japan

Workshop Themes:

1) Optimum control, intelligent control, fuzzy control
2) Measurement and modeling of crop or product response for quality enhancement during storage or processing
3) Identification of control structures based on plant response
4) Intelligent, crop based decision criteria
5) Effect of product variability on control strategy
6) Compartment and instruments for control in processing and storage

For further details, contact:

Prof. S. Oshita, Graduate School of Agri. & Life Science, Dept. of Biological & Environmental Engineering, University of Tokyo; Yayoi 1-1-1, Bunkyo-ku, Tokyo 111-8657, Japan

6th ISTVS Asia-Pacific Regional Conference

December 3-6, 2001
Asian Institute of Technology
Bangkok, Thailand

Call for Papers

Organized by:

International Society for Terrain Vehicle Systems (ISTVS)

Asian Institute of Technology (AIT)

Japanese Society of Terramechanics (JST)

Topics for the Technical Sessions:

Papers are invited for oral presentation and to publish in the pro-

ceedings in the following areas:
1) Off-road vehicle dynamics
2) Soil-vehicle interactions
3) Modeling and simulations
4) Traction
5) Tests and evaluations
6) Conservation of terrain environment
7) Soil properties
8) Soil-tool interactions
9) Precision control

Deadlines:

Submission of Abstracts March 31, 2001

Confirmation of Acceptance April 28, 2001

International Agricultural Engineering Conference 2000

December 4-7, 2000
Bangkok, Thailand

Submission of Full Manuscripts July 31, 2001

Advance Registration August 31, 2001

For Information contact:

Prof. V. M. Salokhe, AASE Program, School of Environment, Resources and Development, Asian Institute of Technology, P. O. Box 4 Klong Luang, Pathumthani 12120, Thailand

Tel:(66-2) 524-5479
(66-2) 524-5450
Fax:(66-2) 524-6200
(66-2) 516-2126
E-mail: salokhe@ait.ac.th

The conference will be held from 4th to 7th December 2000 at the Asian Institute of Technology, Bangkok, Thailand. This conference will coincide with the 10th Anniversary Celebration of the formation of the Asian Association for Agricultural Engineering. Until today, over 100 abstracts have been received and the authors have been notified of the acceptance of their papers.

Other details of the conference are being published in the Final Announcement of the conference and mailed to all interested members. Please pass on this information to your colleagues. The conference details are also available on the AAAE Website:

http://www.aaae.ait.ac.th

The important dates to remember are:

Receipt of Full Papers: 31 July 2000

Pre-registration: 31 August 2000

International Symposium on Water and Environment

December 5, 2000
Bangkok, Thailand

As a part of the International Agricultural Engineering Conference, the International Symposium on Water and Environment will be held at AIT Center in Bangkok, Thailand.

The main objective of the symposium is to bring scientists, industry and policy-makers together to

discuss sustainability issues of agriculture production systems in Asian countries and face the environmental challenges of supplying adequate irrigation water to agriculture and protecting water quality from agricultural chemicals and animal waste with sound science and socially acceptable practices. This symposium will combine keynote and invited speakers.

For more details contact:

Ramesh Kanwar, Ph.D. Professor of Agri. and Biosystems Engineering 219, Davidson Hall, Iowa State University Ames, Iowa 50011

Tel: 515-294-4913
Fax: 515-294-2552
E-mail: rskanwar@iastate.edu

Agricultural Machinery Production in 1999

Italy

The year was positive for the compartment of tractors which closed '99 with production of 81,427 units and weight totalling 233,245 tons - a result which was substantially in line with'98 levels - with value at 3.766 trillion lire, up 1.8%. The production of incomplete tractors declined to 101,850 tons, down 9% under the '98 level,

but the value of output rose by about 1%.

The production of other agricultural machinery, including many types of self- powered and non-self-powered types and gardening machinery, came to a total of 537,672 tons for a gain of 0.5% and value of 6.018 trillion lire, down 0.4%.

In detail, 1999 was a good year for the manufacturers of machinery and equipment for gardening. In fact, output reached the value of 1.38 trillion lire to show an increase of 6% over the previous year. The gain came as a result of a sound trend on the domestic market, up about 5%, and a 6% increase in exports which accounted for 69% of production, for a value of about 960 billion lire.

About 40% of total production99 was taken up by the domestic market while the remaining 60% was shipped to foreign markets to confirm the strong export drive of this national mechanics sector. ■■

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

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b. are relevant: to the promotion of agricultural mechanization, particularly for the developing countries ;
c. have not been previously published elsewhere, or, if previously published are supported by a copyright permission ;
d. deal with practical and adoptable innovations by, small farmers with a minimum of complicated formulas, theories and schematic diagrams ;
e. have a 50 to 100-word abstract, preferably preceding the main body of the article ;
f. are printed, double-spaced, under 3,000 words (approximately equivalent to 6 pages of AMA-size paper) ; and those that
g. art: supported by authentic sources, reference or bibliography.
h. written on floppy disc.

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   vi) bibliography
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h. Express measurements in the metric system and crop yields in metric tons per hectare(t/ha) and smaller units in kilogram or gram(kg/plot or g/row).
i. Indicate by footnotes or legends any abbreviations or symbols used in tables or figures.
j. Convert national currencies in US dollars and use the later consistently.
k. Round off numbers, if possible, to one or two decimal units, e.g., 45.5 kg/ha instead of 45.4762 kg/ha.
l. When numbers must start a sentence, such numbers must be written in words, e.g., Forty-five workers..., or Five tractors..."instead of 45 workers...., or, 5 tractors.