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VOL.31, NO.2, SPRING 2000

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Edited by

YOSHISUKE KISHIDA

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This is the 107th issue since its maiden issue in the Spring of 1971

EDITORIAL

Thirty Years of AMA Service

AMA's maiden issue went into circulation in April 1971 - a full 30 years now of playing the role of exponent and advocate agricultural progress via the adoption of farm mechanization. The initial issue carried the sub-title " Agricultural Mechanization in South East Asia ", a rather narrow confine of geographical coverage which is why the next sub-title's issue was changed to "Farm Mechanization in Asia" through a popular request by readers and co-editors to spread the geographical coverage. But as expansion would have it, the sub-title was once more enlarged in 1981 to cover not only Asia but Africa and Latin America as well. This latest change kept the identity AMA intact as the second "A" in AMA stands for Asia, Africa and Latin America.

All these years, in my capacity as publisher of this journal, I constantly drew support and encouragement from the increasing numbers of co-editors and world-wide readership. To all of them, I express a vote of thanks and gratitude even as I anticipate their continued support in AMA's continuing effort to promote agricultural mechanization. AMA's role has to be in the continuing effort because, 30 years ago, we were already faced with many difficulties in promoting agricultural mechanization such as the development of suitable machineries and implements for local use, establishing after-service system, spare parts supply, inadequate government policies to encourage farm mechanization promotion and manufacturing farm machineries priced enough for most farmers to afford, among other considerations. That AMA has made substantial impact in easing these difficulties is not going to be easy to measure qualitatively and quantitatively. Rather, suffice it to say that there has been vast improvements over time. Among the so-called developing countries 30 years ago, a considerable number have attained rapid industrial growth as to be renamed developed countries. However, many other countries remain in a state of developing to this day. I hold the view that the adoption of farm mechanization by these developing countries should substantially contribute towards the effective use of limited natural resources. This is essentially the reason why for 30 years now, I never cease to rally the support of agricultural engineers and technologists to help AMA fulfill its mission in eradicataing poverty through farm mechanization.

Yoshisuke Kishida

Chief Editor

Tokyo, Japan
April 2000

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AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

Vol.31, No.2, Spring 2000

Package of Improved Implements for Sunflower Production in Maharashtra, India

by

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Abstract

In order to mechanize the sunflower cultivation the use of implements such as the jyoti multicrop planter, multipurpose hoe, phule sunflower thresher (pedal operated) are necessary for obtaining increase in yield. The field data reveal that there is saving in time, labour and other inputs like seed and fertilizer and at the same time there is increase in yield when these implements are properly used.

The use of bullock-drawn multipurpose hoe for inter-cultivation resulted in efficient weeding, tilling and earthing operations with minimum damage to the standing crop. By using the phule sunflower thresher (pedal operated), it was observed that there was 100% threshing with cleaning efficiency of 95.5%. Further, there was a reduction in labour savings in the costs of operation.

Thus the overall effect of the use of a package of improved implements has shown encouraging results as regards savings in time and labour with an increase in yield over the traditional method. It is observed that by using the package there was about 52.75% of savings in cost of operation over the traditional method. The yields are also increased by 20.45% over the average yield by the conventional method of sunflower cultivation.

Introduction

Sunflower is one of the most important oil seed crops grown in all seasons in Maharashtra. The total area planted to the sunflower in the State during (1993-94) was 5.72 lakh ha and total yield was 3.58 lakh tones. The area planted to sunflower is increasing day by day. This crop can be grown under rainfed and irrigated conditions. Sunflower oil is also a good source of edible oils.

At present most of the farmers use local implements and follow indigenous practice in performing various field operations. The implements used by most of them are inefficient in so far as quality of work and quantum of work done in a limit time is concerned. This paper gives the details of the package of improved implements for use in sunflower production.

Materials and Methods

Sowing of Seeds and Fertilizer Application

Land is prepared by using indigenous plough or mould board plough after a firm and smooth seed bed is prepared by using a cultivator and a harrow. With a view to making proper placement of the seeds, proper placement and efficient use of granular fertilizer and to maintain correct seed rates as per recommendation, the Agricultural Engineering Research Centre, Pune has developed a bullock-drawn Jyoti multicrop planter.

The jyoti multicrop planter

The jyoti planter **(Fig.1)** is useful for planting the seeds of sunflower, groundnut, soybean, safflower, maize, jowar, gram etc. Simultaneously, a dose of fertilizer in granular form, 5 cm to the side and 5 cm below the level of seeds is applied. The implement carries three seed hoppers, on each fitted an individual coulter (furrow opener). Mixed crop sowing is possible with such arrangement. For planting and sowing different seeds, plastic seed

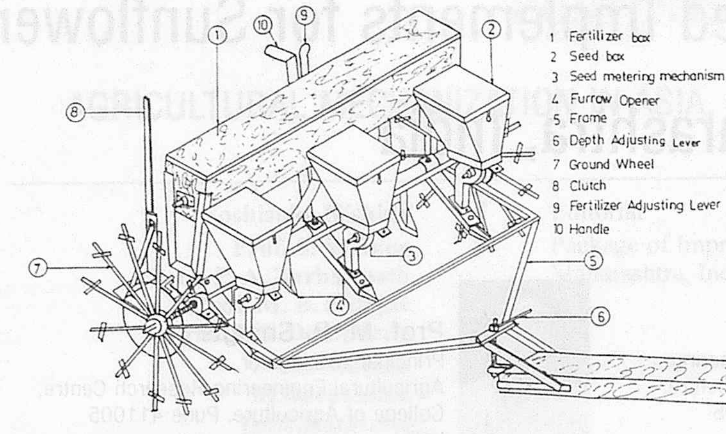

Fig.1 Jyoti planter(bullock-drawn).

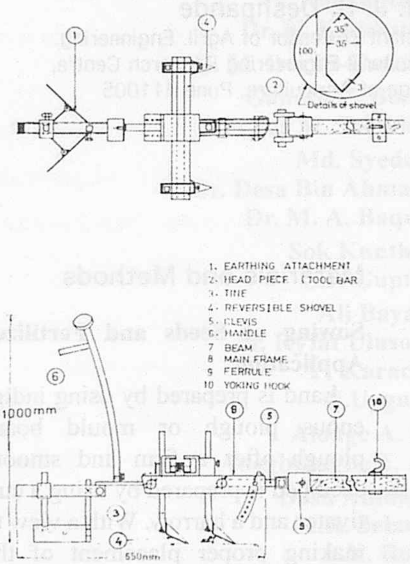

Fig. 2 Multipurpose hoe.
(reversible shovel+earthing attachment)

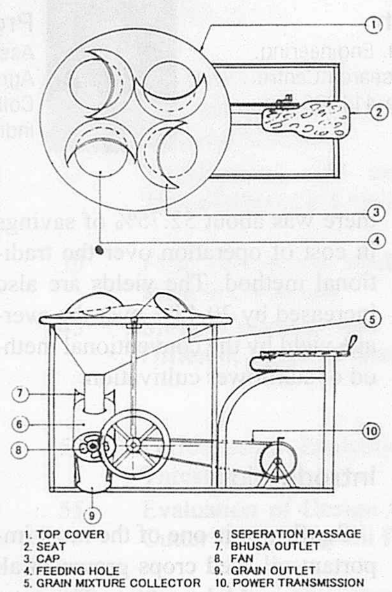

Fig. 3 Phule sunflower thresher.

Table 1. Field Performance of the Improved Implement in Planting Sunflower Seeds

Particular	Sowing	
	Jyoti Planter	Local
Av. depth of operation (cm)	5.9	5.0
Av. draft (Kgf)	75.25	54.75
Seed rate obtained (Kg/ha)	7.25	11.75
Field efficiency (%)	75.27	62.72
Time requirement (h/ha)		
Machine	3.77	4.58
Man	3.77	4.58
Bullock	7.54	9.19
Cost of operation (Rs/ha)	76.49	77.36

plates (rotors) carrying cells or grooves of different sizes and numbers are prepared. For each crop a separate rotor is required. The clutch is provided for disengaging the power to the seed and fertilizer metering mechanisms at the time of turning and transportation. The clutch operation avoids the wastage of valuable seeds and fertilizers.

The Jyoti Multicrop planter is animal-drawn, three row planter with the arrangement for adjustment for the depth of planting. The weight of the machine is 80 Kg. The hopper capacity for seed is 12 Kg. (4 Kg. × 3 boxes) and for fertilizer is 25 Kg. The row spacing can be adjusted from 22.5 to 45 cm. The field capacity is 1.5 to 2.2 ha/day. The draft required is 70-75 Kgf.

The field trials reveal that the field efficiency of the Jyoti Planter is 75 percent as against 62 percent of local seed drill. The cost of operation is near about same, i.e., 76.49 Rs/ha by Jyoti planter and 77.36 Rs/ha local method. The advantage with the use of the planter is that, precision placement of seed and fertilizer can be maintained. The results are summarized in **Table 1**.

Multipurpose hoe

The Agricultural Engineering Research Centre (AERC) at Pune developed a multipurpose hoe (**Fig.2**) for weeding, weed collection, soil tilling and earthing up in the row crop. The multipurpose hoe is provided with a main frame, head piece, tines, tine with clamps, shovels and sweeps. The width of the implement can be adjusted from 22.5 to 45 cm. The weight of the implement is 14 kg without beam. Three multipurpose hoes can be used per pair of bullocks. One person per hoe is required for holding two operative hoes in the right position. The maximum mulch depth, maximum tilling height and maximum earthing height is observed as 3.5 cm, 7.5 cm and 5 to 8 cm respectively. The field capacity is observed as 3.5 ha/day.

The cost of operation of the multipurpose hoe is 109.02 Rs/ha as against 185.26 Rs/ha by the local method. The field efficiency of the multipurpose hoe was 77% as against 63.71% by the local hoe.

Table 2. Results of Field Performance of Improved Implements for Interculturing in Sunflower

Particular	Sowing Multipurpose hoe	Local hoe
Av. tilling depth obtained (cm)	7.05	2.93
Av. earthing height obtained (cm)	6.25	3.0
Weed population /m^2		
Before test	21.0	18.8
After test	5.0	6.2
Plant damage (%)	4.52	9.64
Weeding efficiency	79.19	67.02
Av. draft (Kgf)	36.75	26.5
Field efficiency (%)	77.00	63.71
Power requirement (h/ha)		
Machine	6.87	12.03
Man	6.87	12.03
Bullock	4.58	8.08
Cost of operation (Rs/ha)	109.02	185.26

Table 5. Cost of Operation (Rs/ha)

Operation	Improved	Local
Planting	76.49	77.36
Weeding	109.02	185.26
Threshing	222.07	600.00
Total	407.58	455.04
Benefits (Rs/ha)		52.75
Benefits (%)		

The test results are summarized in **Table 2.**

The sunflower thresher (pedal-operated)

Sunflower threshing is a tedious and time-consuming operation. The AERC developed a manually-operated hold-on-type thresher (**Fig.3**) consisting of four feeding holes, top cover, blowing fan, threshing wheel, etc. The power is taken through a cycle chain to the threshing wheel and blowing fan. Four persons can work on the thresher at a time. They hold the heads on the revolving threshing wheel until the seeds are detached. Of the four persons one pedals the thresher. The sunflower head is not crushed or broken. Grains / seeds obtained are clean and free from straw. The weight of this machine is 42 kg. Grains / seeds can be obtained in one hour. The feed rate is 160 kg/ha. The percentage of broken grains is nil. The percentage of blown grain is 1.2%. The threshing efficiency is 100%. The cleaning efficiency is 95.5%. The cost of operation was Rs. 24.62 per quintal as against Rs. 50 per quintal by manual threshing. The results are given in **Table 3.**

Results and Discussions

The yield obtained by using the improved implements are 9.65 q/ha as against 7.5 q/ha by local practices. As such, there was an increase in yield by 20.45% in using the improved implements as against the conventional method.

The economic analysis shows that the cost of operation by using the improved implements was Rs. 407.58 as against Rs. 862.62 by using local implements. Therefore, there is a savings of Rs. 455.04 per ha by using the package improved implements.

Table 3. Results of Field Performance of Pedal-operated Sunflower Thresher

Particular	Average Results Pedal operated	Manual
No. of persons used	4	4
Feed rate(Kg/h)	134.75	75
Broken grains (%)	Nil	Nil
Blown grains(%)	1.33	2.98
Threshing efficiency (%)	100	100
Cleaning efficiency (%)	95.76	95.5
Output capacity (Kg/ha)	44.60	20.0
Cost of operation (Rs/Q)	24.62	55.00

Table 4. Effect of Improved Implements on Yield of Sunflower

Year	Yield (Q/ha) Imp.	Local	Percentage increase in yield	Average (%)
1992	9.65	7.5	22.32	20.45
1991	9.30	7.4	20.43	20.45
1990	9.34	7.6	18.62	20.45

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Design and Development of A Trencher

by

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Abstract

A tractor-drawn trencher suitable for 35 - 45 HP tractor was designed and developed for opening a rectangular trench of 300 × 300 mm. The design and development of the tractor-drawn trencher and its components were carried out to optimize the trenching parameters. The functional components of the trencher were designed and developed keeping in view the practical trenching conditions. The developed unit consists of a share, bar share, Y-base, mould board, standard, side plates, frame and three-point hitch system. The share, bar share and mould boards were fitted to a Y-base. This assembly is fitted to standard which, in turn, is connected to the frame by means of the side plates. A safety pin is provided between the standard and side plates to protect the unit from over load. The two bottoms are placed in a line such that one operates behind the other. The front bottom throws the soil towards right side and forms a vertical wall at the left side of the trench and opens the trench up to 15 cm. The rear bottom opens the trench to a further depth of 15 cm, throws the soil towards left side and forms a vertical wall on the right side of the trench.

Introduction

The trend in the farm machinery market in India is increasing every year and about 160,000 four wheel farm tractors were produced during 1994-95 (Jain, 1995). India has emerged as the world's largest tractor manufacturer and consumer (Baruah, 1995). As a global power in the international tractor industry, India must ensure efficient utilization of such a heavy recurring investment on the agricultural tractors for a complete range of matching implements. With this trend, India must graduate into a crop specific mechanization to meet the needs of nationally important crops (Jain, 1995).

Modernization of the Indian agriculture requires appropriate, improved farm implements and machinery for ensuring timely field operations, effective application of various crop production inputs, and for optimal utilization of human, animal and mechanical power sources. Reduction in drudgery of farm operations, cost effectiveness, eco-friendlyness and employment potential are the other important advantages of using farm implements and machinery. The purchase and use of tractors is being encouraged by the govermment by way of subsidy to marginal and small farmers.

Review of Literature

Palmer *et al.* (1979) designed and evaluated an under-water pipe line trenching plough, for constructing a 1.5m deep under-water trench, using the principle of the long beam forestry plough to give accurate depth control. Jeong *et al.* (1987) designed and developed a rotary trencher for digging holes for fruit trees, planting mulberry bushes, etc. It was driven by a tractor P.T.O. with a centre drive. The blade diameter and driven velocity were 640 mm and 546 rpm, respectively.

Few trenching machines are available in foreign countries for digging the utility trenches for oil, gas and water pipe lines, drainage ditches, sewers, cables and foundations as well as for the construction and road-building jobs (Borshchov *et al.*, 1988). Continuous - action trenching machines; ladder-type, wheel-type and drain laying trenching machines are some of the types under continuous action trenching machines. Some of the trenching machines which are used for drainage purposes are backhoe, endless-chain trencher-slanted boom and drain-tube plow.

These heavy duty-earth moving trenching machines are not suitable for Indian farming conditions due to fragmented land holdings, changed soil-crop-machine parameters which are complex and are non-cost effective for local adoption. As such there is no implement or low cost machinery operated by the four wheeled tractors in the range of 35 to 45 HP, which can be easily offered to the farming community.

Materials and Methods

Conceptual Design of Trencher

The basic concept involved in the design of a trenching machine is that the two trench forming bottoms are placed one behind the another such that they follow the same trench path. The front and rear working bottoms of the trencher consist of trench cutting, soil lifting and throwing components. The two bottoms are placed in such a way that the front bottom will open a trench to a depth of 15 cm from the surface and the rear bottom will open a trench, 15 cm deeper than the front bottom in a normal soil and operating conditions. Hence the rear bottom of the trencher is placed 15 cm lower than the front bottom. The front bottom share first penetrates from ground surface of the soil and cut a normal 30 cm width of trench. The front mould board lifts and throw the soil towards the right side of trench by creating a clear vertical trench wall at the left side of the trench. The rear bottom follows the same trench path but at different depth by further cutting and throwing the soil towards the left side, by creating a clear vertical wall at the right side of the trench. Thus the left and right side vertical walls of the trench are formed by the front and rear bottoms, respectively.

Most of the agricultural research work under moisture conservation, afforestation, agro-forestry, horticulture, drainage, irrigation, fertilizer or soil amendment/mulching studies in trenches shows that the trench dimension of 30 × 30 cm would be commonly adopted.

The draft requirement of the plough is given by:

$D = d \times b \times k$ (Naik, 1962)

where,

D - draft requirement, kg
d - depth of cut, cm
b - width of cut, cm
k - soil resistance in kg/cm^2, which is approximately 0.5 to 0.7 for light soi, 0.7 to 0.9 for medium soils and 1.0 to 1.3 for very heavy soils.

$D = 30 \times 30 \times 1.2 = 1080$ kg

From the conceptual design, trench cross sectional area, opened by either front or rear bottom

$= 15 \times 30$ cm.

Hence, the maximum draft acting on either front or rear bottom of the trencher is 540 kg

$D_r = D_f = (1080) / 2 = 540$ kg

Trencher Parts

The trencher mainly consists of the following parts.

i. Share - trench cutting
ii. Mould board-soil lifting and throwing
iii. Y-base- share and mould board supporting
iv. Standard - suppoting the trench bottom
v. Side plates - suppoting the standard
vi. Frame assembly - main parallel hitch bars, oblique bracing bar, drawbar hitch and top link hitch

i. Design of trench cutting share

One of the important functional parts of the trencher is the share. The function performed by the share is that of cutting off trench slices from the trench bottom and raising them to the mould board portion which lifts and throws the soil. For proper penetration, cutting and raising of soil on the soil lifting mould board, the blade should be designed based on the soil characteristics.

a. Different angles of trencher share

When the share is positioned in cutting position, it forms a spatial wedge and forms three different angles. For any common type of mould boards like cylindrical, cylindroidal, helical or semi-helical, the values of load angle (α), cutting angle (γ) and setting angle (θ_0) are taken as 15, 22 and 40 degrees, respectively, (Krutikov, 1961).

b. Length of share (l)

The selected setting angle (θ_0) is 40 degrees. Hence length of blade (l) required to cut a 30 cm trench width (b) is $\text{Sin}(\theta_0) = b/l$ and $l = 46.6$ cm

c. Thickness of share (t)

Under the worst conditions of operation it is assumed that the total draft is distributed as follows :

i. 60 to 75 per cent for soil cutting
ii. 10 to 15 per cent for soil lifting/inversion
iii. 15 to 25 per cent for pushing the soil forward.

Hence, the maximum draft for soil cutting by either front or rear share is 405 kg:

$= 540 \times 0.75$ kg $= 405$ kg

According to Johnson (1960) and Kepner *et al.* (1987) an object striking the share line will produce a clockwise or counter clockwise moment, when contact is made close to the point of the share. The center of resistance lies at a distance equal to $3/4^{th}$ size of the plough from the share wing.

Maximum bending moment,

$M = 405 \text{ kg} \times 3/4\ (30) \text{ cm} = 9112.5 \text{ kg cm}$

Shares are mostly constructed from high carbon (hot rolled) steel and it has an allowable bending stress (f_b) values of 32,500 kg/cm^2

Using flexural equation:

$M/I = f_b/y$

where,

$I = 1\,t^3/12 = 46.6 \times t^3/12$, cm^4

$= (3.89\,t^3)$, cm^4

$y = t/2$

where,

t = thickness of the share

$\therefore\ 9112.5/\ 3.89t^3 = 32500/(t/2)$

and taking factor of safety as 4,

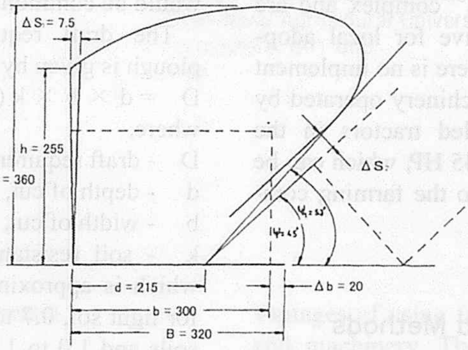
$t = 0.78$ cm

The thickness of the share 't' was taken as 8 mm with 466 × 90 mm flat.

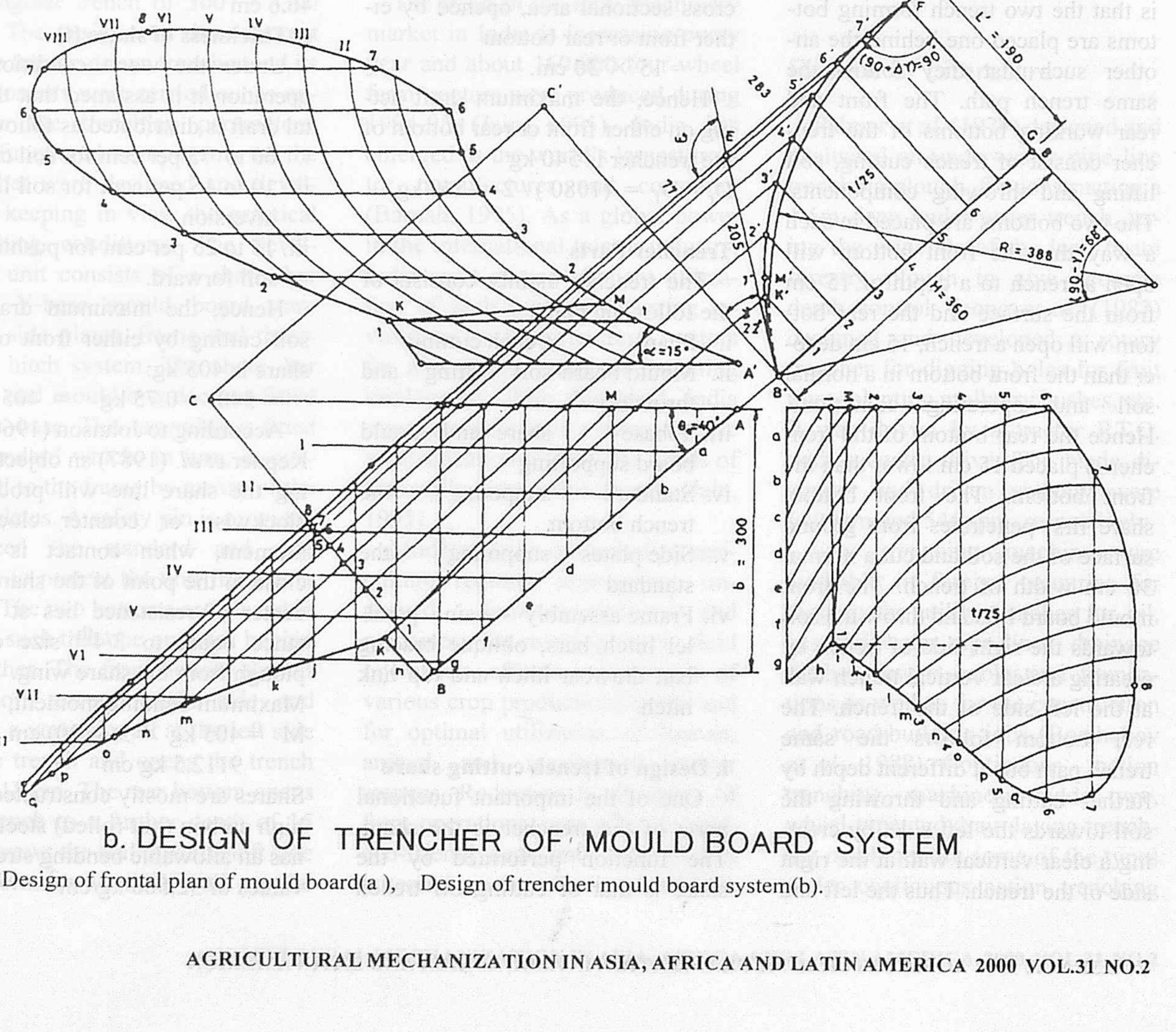
The share was mounted to the front lower edge of the Y-base. An adjustable bar point was mounted to the lateral surface of the Y-base by a tapered key, so that the (setting angle, θ_0) edge of the share should exactly coincide with bar point. The share point is the part of the cutting portion, which takes the greatest load while trench cutting in hard and stony conditions and, therefore, bar point is provided with a material reserve, so that removing and resharpening it can be done when it becomes dull.

ii. Design of trencher soil lifting mould board system

The trencher mould board system should perform a complete lifting and throwing of soil slice away from a trench with least resistance. Hence a cylindrical type of mould board design was adopted. The design details of frontal plan of mould



a. DESIGN OF FRONTAL OF MOULD BOARD



b. DESIGN OF TRENCHER OF MOULD BOARD SYSTEM

Fig. 1 Design of frontal plan of mould board(a), Design of trencher mould board system(b).

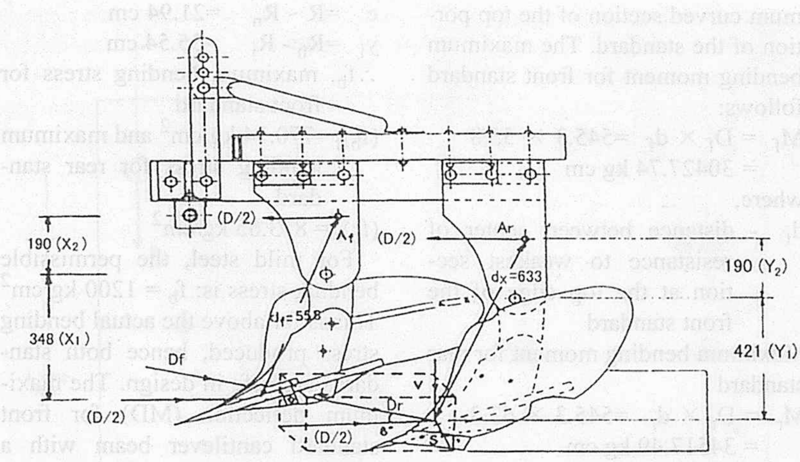


Fig. 2 Forces acting on the standard.
Note: Values in the parentheses are for the rear standard.

the width of share (B):

$B = b + \Delta b$ (Bosoi *et al.*, 1987)

where,

b - structural width of the trench slice

Δb - +2 to +4 cm for standard mould board

The height of the breast edge of the mould board `h' depends on the width of the furrow slice tillage depth `d' and on the forward speed`v'.

$h = b + \Delta h_1 + \Delta h_2$

where,

Δh_1 -0.1 b to -0.2b for hard and grass lands

Δh_2 - 0 cm, when v < 7 kmph

$h = 30 + (-0.15 \times 30) + 0$

$= 25.5$ cm

The entire height of the mould board (H) was calculated from the formula :

$H = \sqrt{d^2+b^2+\Delta h_2+\Delta h_3}$

where,

Δh_2 - as in the above formula

Δh_3 0 to -3 cm, for non-lea mould board

d trench depth

In medium tillage (tillage depth d = 18 to 24 cm and width of furrow/ trench slice, b = 25 to 35 cm) the ratio of `b/d' in structural assumption of plows is 1.4 to 1.5.

So,

b/d = 1.4 andd = 21.5 cm

$\therefore H = \sqrt{(21.5)^2 + (30)^2 + 0 + (-1.5)}$

$= 36$ cm

The breast edge of the mould board falls from the perpendicular by

ΔS_1 =0.5 to 1 cm

ΔS_1 is taken as 0.75 cm

To determine the lower edge of the mould board, the line of the profile of an inverted trench slice was plotted on the front plan. For this purpose, at a distance `d' (21.5 cm) from the trench wall, a line was plotted at angle ψ calculated from the formula.

ψ = arc (sin d/b)=arc (sin 21.5 / 30.0) $\cong 45°$

The lower edge of the mould board was determined by a line passing over the same point as the profile line of the trench slice, inclined at angle ` ψ_1' which amount to,

$\sin \psi_1 = (d + \Delta d)/b$ (Bernacki *et al.*,1972)

where,

$\Delta d = 2.5$ cm

$\therefore \psi_1 = \arcsin [(21.5 + 2.5)/ 30]$

$= 53°$

The upper edge of the mould board was plotted by a convex line in such a manner that the highest point of the mould board may achieve a greater distance from the trench wall than the width of the trench slice. In order to obtain the complete profilogram of the mould board on plan and elevation, different projections are drawn from the frontal plan and parabolic curvature as suggested by Bernacki *et al.* (1972).

The value of L' is calculated by the relation, $L' = L \sin \theta_0$

But L/H = L/36 = 0.8, L = 2.9 cm

$\therefore L' = 29 \sin 40^0 = 19$ cm

The two mould boards were fabricated from designed profilagram by adopting a standard wooden model from a mould board contour device technique. Among the two fabricated mould boards, one was exactly a mirror image of the other so as to lift and throw the cut soil on either side of the trench, i.e., front mould board will throw towards right side and the rear mould board towards the left side of the trench.

iii. Y-base

The Y-base is bottom component which holds the share, bar point and mould board. A standard type of sturdy Y-base as adopted by Massey-Furguson (TAFE) Ltd. was selected for fabrication. The top curvature of the front and rear Y-bases are exactly similar to the curvature of their respective mould boards. The front and rear Y-bases are exactly a mirror image of one another.

iv. Design of standard

The standards were designed in such a way that they can take up the resultant of useful and parasitic soil forces during the operation of the trencher bottom. The useful soil forces acting upon a trencher bottom are those resulting from the operations of cutting, lifting, and throwing soil slice. These soil forces practically, always introduce a rotational effect on the bottom (Clyde, 1961). The parasitic forces include, those that act upon the side and bottom (including friction) of the Y-base (Ma-

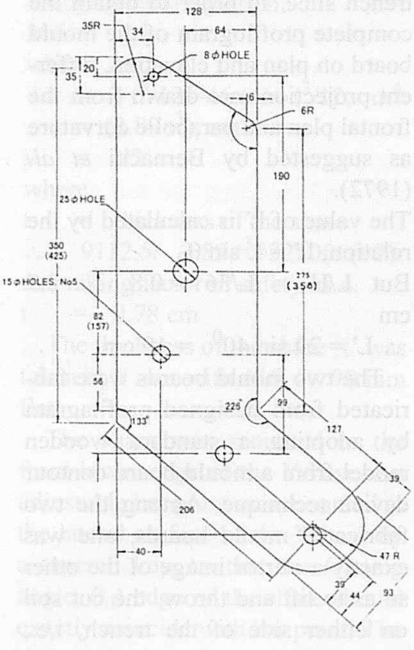


Fig. 3 Front standard details of trencher.

cluding friction) of the Y-base (Maheshwari and Gupta, 1960). The standard connects the Y-base at its one end and to the frame through side plates by means bolts and nuts and a safety shear pin at its other end. Both the front and rear standards were mounted as cantilevered element as illustrated in **Fig.2**. The total maximum soil resistance force, acting on either front or rear portion is 540 kg.

Based on the analysis made by Clyde (1940) and Maheshwari and Gupta (1960) on mounted plow, in hard penetration the resultant force is longitudinally and laterally inclined at an angle to the horizontal (draft) force. Both these angles may be taken as 8 to 10°. The corrected values of D_r and D_f are 540 / cos 8° = 545.3 kg.

The point of intersection of forces acting on the mould board surface is called the center of resistance of the bottom. Under average soil conditions, the center of resistance is more or less in the middle of the width of the furrow/trench slice at height of 1/3 of the trench slice thickness from the side of the trench bottom. The critical section occurs at the maximum curved section of the top portion of the standard. The maximum bending moment for front standard follows:

$M_f = D_f \times d_f = 545.3 \times 55.8$
$= 30427.74$ kg cm

where,

d_f - distance between center of resistance to weakest section at the top edge of the front standard

Maximum bending moment for rear standard

$M_r = D_r \times d_r = 545.3 \times 63.3$
$= 34517.49$ kg cm

where,

d_r - distance between center of resistance to weakest section at the top edge of the rear standard

Considering a failure of the standards in bending, the standards were considered as a curved beam subjected to a bending moment M. The moments acting on the front and rear standard are shown in **Fig. 2**. The general expression for the calculation of bending stress at any fiber at a distance of y from the neutral axis is given by :

$f_b = [M / A. e][y / (R_n - y)]$

where,

A - area of cross section

e - distance from the centroidal axis to the neutral axis = $R - R_n$

R_n - radius of curvature of the neutral axis

R - radius of curvature of the centroidal axis

y - distance from the neutral axis to fibre under consideration.

The maximum bending stress always occurs at the inside fibre and the simplified bending stress equation is given by:

$f_{bi} = M. y_i / A. e. R_i$

where,

y_i - distance from the neutral axis to the inside fibre = $R_n - R_i$

R_i - radius of curvature of the inside fibre = 2.5 cm

$A = w \times t = 2.0 \times 12.8 = 25.6 cm^2$

$R_n = t/[\log_e (R_o/R_i)]$ for rectangular section

$R_n = 12.8/[\log_e(3.5/2.5)] = 38.04$ cm

$R = R_i + t/2 = 2.5 + (12.8/2) = 8.90$ cm

$e = R \sim R_n = 21.94$ cm

$y_i = R_n - R_i = 35.54$ cm

$\therefore f_b$, maximum bending stress for front standard

$(f_b)_f = 770.14$ kg/cm^2 and maximum bending stress for rear standard

$(f_b)_r = 873.65$ kg/cm^2

For mild steel, the permissible bending stress is: $f_b = 1200$ kg/cm^2 This is far above the actual bending stress produced, hence both standards are safe in design. The maximum deflection (MD) for front standard cantilever beam with a point load acting at the free end is given by :

$MD = M_A d_f^3 / 3 E I$

where,

$I = Wt^3 / 12$

E - modulus of elasticity

For both front and rear standard

$I = (1/12) \times 2.0 \times (12.8)^3 = 349.525$ cm^4

Then, maximum deflection of the front standard

$MD_f = 0.045$ cm

Maximum deflection of the rear standard

$MD_r = 0.066$ cm

The design details of both front and rear standards are shown in **Fig.3**.

Permissible deflection

= Span / 325

Permissible deflection for front standard

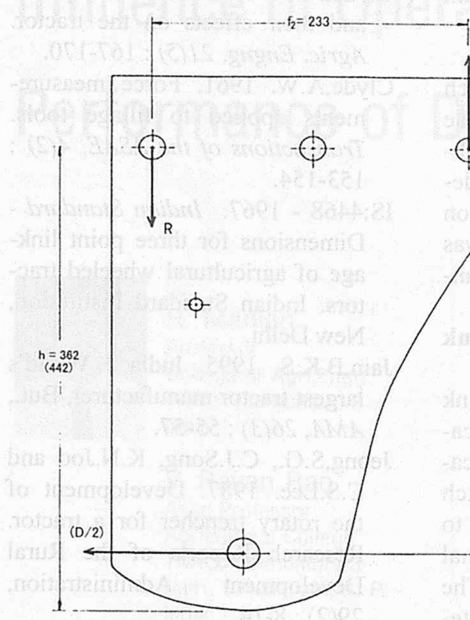
= 55.8 / 325 = 0.1717 cm

Permissible deflection for rear standard

= 63.3 / 325 = 0.1948 cm

Thus the maximum deflection produced in front and rear standards were lower than the permissible deflection, hence both standards are safe.

The standards were fabricated from 20 mm thick M.S. flat section in which the front standard had a total length of 510 mm and 128 mm width at top vertical portion and 93 mm at the lower bent portion. A rear standard consisted of almost the same dimensions except, for an extra height of 75 mm at the top vertical portion. At the top a safety



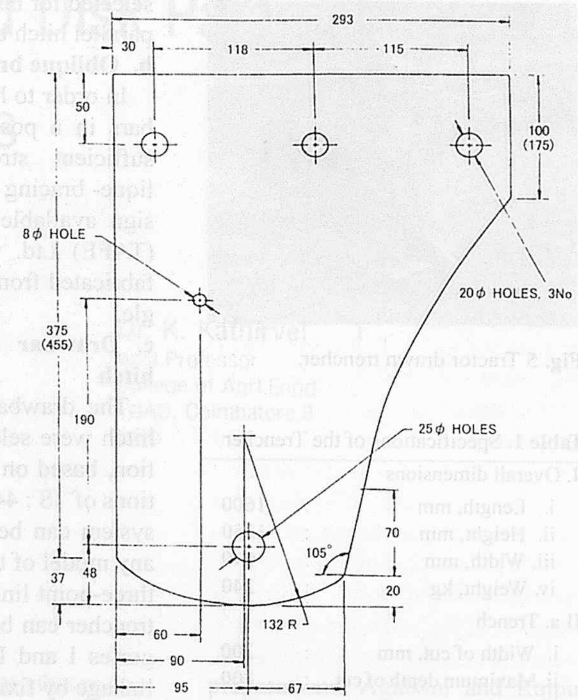


Fig. 4a Details of kinematics on front side plate.
Note: Values in the parentheses are for the rear side plate.

Fig. 4b Front side plate details of trencher.

shear pin through side plates was also provided for the safety of the trencher bottoms against hard objects in the soil.

v. Design of side plates

Side plates are the intermediate component, which connects the bottom of the trencher to the frame through the standard. Kinematic details of the front and rear side plates is shown in **Fig. 4a**.

a. Front side plates

Since, there are two side plates, the maximum draft acting on them is equally shared. So the maximum draft acting on one side plate is,

$(D_f/2)$=270 kg

Side plates are designed with regards to its bending strength in relation to the maximum draft $D_f/2$. Bending stress amounts to

$f_b = (D_f/2)\, l_1/Z$

where,

Z - the cross section index (section modulus) of one side plate

For rectangular cross section, at X-X axis,

$Z_{xx} = h\, t^2 /6 = 262.088\ cm^3$

$\therefore f_b = (270 \times 27.7)\ 262.088$

$= 28.536\ kg/cm^2$

Taking factor of safety as 4, then $f_b = 114.145\ kg/cm^2$. The design details are shown in **Fig. 4b**.

b. Rear side plates

$Z_{xx} = 1.2\,(44.2)^2\ /6 = 390.728\ cm^3$

$\therefore f_b = (270 \times 35.2)\ /\ 390.728$

$= 24.324\ kg/cm^2$

Taking factor of safety as 4, then $f_b = 97.295\ kg/cm^2$

For MS, permissible bending stress,

$f_b = 1200\ kg/cm^2$

This is far above the actual bending stress produced in the front and rear side plates. Hence the design is safe. For the fabrication of the front and rear side plates (two each), 12 mm thick, MS sheet was used. Basically, the dimensions of front and rear side plates were similar except, the rear side plate which was 75 mm longer than the front one. The design details are shown in **Fig. 4b**.

c. Design of safety shear pin

A shear pin is provided for the safety of the trencher bottoms to avoid the damage due to big stones/boulders. From **Fig.4a**. draft force acting at the pin section:

$D_f = (D/2)\, x_1 / x_2$

$= (540 \times 34.80)19$

$= 989.05$ kg for front and

$D_r = (D/2)\, y_1\ /\ y_2$

$= 1196.52$ kg for rear

The diameter of the shear pin is calculated to meet the above shearing forces.

$d_f^2 = 4(D_f)\ /\ (\pi f_s)$

Taking, low carbon steel (hot rolled) as a shear pin material (shear stress of 4690 kg/cm^2)

$d_f^2 = 4\,(989.05)\ /\ (\pi \times 4690)$

$d_f = 0.27$ cm for front side plates and,

$d_r^2 = 4\,(D_r)\ /\ (\pi f_s)$

$d_r = 0.32$ cm for rear side plates

Considering the factor of safety as 2, 0.8 cm diameter shear bolts were used for both the side plates.

vi. Frame assembly

a. Main parallel hitch bar

These are designed on the basis of maximum drawbar pull of a trac-

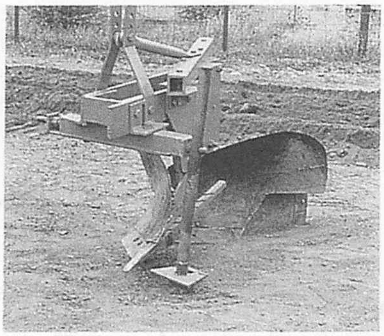


Fig. 5 Tractor drawn trencher.

Table 1. Specifications of the Trencher

I. Overall dimensions		
i. Length, mm	:	1600
ii. Height, mm	:	1250
iii. Width, mm	:	940
iv. Weight, kg	:	240
II a. Trench		
i. Width of cut, mm	:	300
ii. Maximum depth of cut, mm	:	300
b. Mould board componets		
i. Front		
Type	:	parabolic
Length, mm	:	910
Height, mm	:	390
ii. Rear		
Type	:	parabolic
Length, mm	:	850
Height, mm	:	490
III. Power required		
Tractor, hp	:	35-45

withstand for tension failure against the maximum pull .

f_t = (DBP) / (w-d)t

where,

d - maximum diameter of hole on the bar

w - depth of bar

t - thickness of bar

f_t - allowable tensile stress (for MS, ft = 3150 kg/cm^2)

The depth of bar (w) and maximum diameter of hole (d) were taken at 13cm and 1.8cm, respectively. For MS a factor of safety as 12 to 60 is recommended for shock load (Mahadevan and Reddy, 1984). Considering a factor of safety as 40, then the thickness, t, becomes 1.86 cm. So 130 × 20 mm MS flat was selected for both right and left main parallel hitch bars.

b. Oblique bracing bar

In order to hold the parallel hitch bars in a position and to provide sufficient strength, a sturdyoblique- bracing bar similar to the design available at Massey- Furguson (TAFE) Ltd. was adopted. It was fabricated from 75 × 10 mm L-angle.

c. Drawbar hitch and top link hitch

The drawbar hitch and top link hitch were selected for the fabrication, based on the design specifications of IS : 4468 - 1967. The hitch system can be directly mounted to any model of tractors by the normal three-point linkage attachment. The trencher can be mounted with categories I and II of the three- point linkage by fixing appropriate pins.

Conclusion

A complete schematic diagram of the designed and developed tractor drawn trencher is shown in **Fig.** 5. The design specifications of the trencher is given in **Table 1**.

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Influence of Operating and Disk Parameters on Performance of Disk Tools

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Abstract

Studies were conducted in soil bin containing black clay loam and sand in order to assess the draft, vertical and lateral components of reaction in disks at different moisture levels. The selected variables are: disk angle-40, 44 and 48 degrees, tilt angle-16,20 and 24 degrees, disc diameter-51, 56 and 61 cm, and forward speed- 4,7 and 10 kmph.

The drafts and lateral reaction components show high values in black clay loam. The vertical reaction decreased with an increase in soil moisture content (MC) in both soils. The increase in draft in black clay loam soil was less when compared to sand with increase in soil moisture. In sand, some sinkage of disks was observed when the MC was increased. The MC range between 13 to 17.18 per cent may provide good working condition in black clay loam soil. At 7 kmph speed, the drafts and the soil reactions were observed to be balanced, which suggests the speed about 7 kmph may be the right operating speed.

Larger size and heavier (56 and 61cm) disks gave satisfactory results in black clay loam soil at low MC level as they showed low upward vertical reaction component which helped in penetration and in sand 51 cm was disk found workable. Disk angles set at 44 degrees were found to justify for the working of disk tools, since the drafts observed were much at lower and higher disk angles. The 16 degrees tilt angle results in lower draft and vertical reaction component (better penetration) as compared to 30 and 24 degrees.

Introduction

Tillage is the major event in the process of crop production which consumes nearly 30 to 35 per cent of the total energy requirements. Disc tools play a prominent role in tillage and under certain conditions they are reported to be advantageous over other implements used for the purpose, as they roll into the soil instead of sliding. Creery and Nichols (1956) reported that the disk is a major tillage tool which is considered to be the best implement for a wide variety of soil preparations. Agarwal and Rajput (1964-65) observed that disk tools can be used in adverse soil conditions such as lack or excess of MC of the soil presence of roots, stones and hard pan in the subsoil. Because of rolling action their unit draft is low if working under similar soil conditions. They concluded that the draft of disk tools is influenced by the factors such as speed of operation, soil type, moisture content, depth of penetration, disk size, concavity, disk angle and tilt angle.

Though tillage has its advantages, because of its high energy consumption, soil compacting tendency and soil erodibility, a new concept, namely; minimum tillage came into existence and a need arises to evaluate the existing tillage tools for their soil handling capacity and energy requirements in the present day of energy crises.

Review of Literature

Gordon (1941) concluded that draft will be minimum at about a 45 degree disk angle. Higher values of draft at greater disk angles may be due to greater throw of the soil. At smaller angles the draft tends to increase because of greater contact area. Penetration is improved by increasing the disk angle since the upward vertical reaction decreased considerably. Hordon (1941) reported that the draft increases with an increase in the angle of inclination. For better penetration in hard soils the angle of inclination should be reduced to 18 - 20 degrees as is the most effective angle. Reed (1948) stated that disks are generally operated at 42 to 45 degrees with the plane of the disk to the direction of travel. He also observed that disks are tilted backward at an angle of 15 to 25 degrees from the vertical position. Leon Lyles and Wardruff (1962) concluded that MC has definite effect on tillage operations and 15 to 23 per cent moisture content is the optimum range.

According to Agarwal and Rajput (1964-65) the draft of disk plough increased when the disk was set above or below 45 degrees in the direction of travel. Minimum draft was observed at 45 degrees. At smaller angles the convex side of the disk contacts more with the furrow wall which increases friction. At angles above 45 degrees the width of cut and the throw of the soil result in increased draft. They also observed that the draft increases as the tilt angle increases within 15 and 25 degrees and the soil lift is higher causing the disk to penetrate deeper. Not much soil is thrown sideways at higher tilt angles due to the tendency of the disks form flatter. Penetration was better at smaller angles within 15 to 25 degrees and 18 to 20 degrees being the most effective angles. For clay soils tilt angles should be increased since for sticky soils, larger angles turn better slice.The optimum speed of operation of the disks was about 6 km/hr.

Horrison (1977) concluded that the draft of the disk plough was reduced when the disk angle was increased from 35 to 45 degrees.

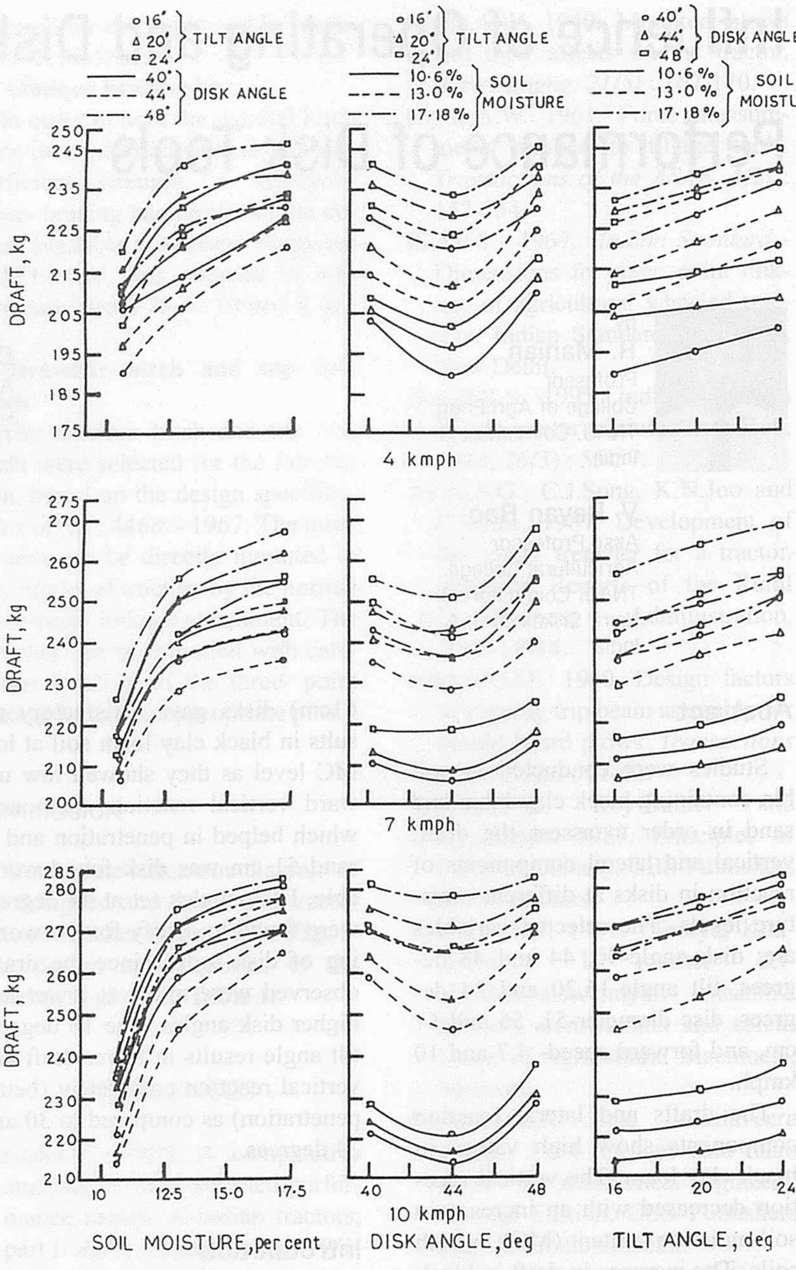

Fig. 1 Effect of soil moisture, disk angle, tilt angle and forward speed for 51cm disk diameter in black clay loam soil.

Alakra (1987) reported that a disk angle of 50 degrees and tilt angle of 18 degrees to be appropriate for average ploughing conditions. Panigrabi (1990) observed that the depth of cut was maximum at 16 degrees tilt angle and increase in soil MC increases the depth of penetration. Shirin et al. (1993) reported that an increase in the soil MC and tilt angle, and decrease in the forward speed, reduced the specific draft requirements of the disc plough. A 45-degree disk angle setting showed less specific draft requirement compared to settings of 40 and 50 degrees.

Materials and Methods

The experiments were conducted at the tillage and traction bins, College of Agricultural Engineering, Tamil Nadu Agricultural University, Coimbatore, India.

Disk Attaching Frame

A frame shown in **Fig. 1** is suitable for attaching the plough disks under test with provisions to operate the disk at different depths of operation, disk angles and tilt angles, have been made. The frame was secured to the soil bin test trolley by means of bolts and nuts. The frame consists of angular section, revolving unit, and plough body. The variables chosen for evaluating the disks were: soil, soil MC, speed of operation, disk diameter, disk angle and tilt angle.

Two types of soils were selected for conducting the trials. Soil reactions acting on the plough bodies were found to vary with the type of soil. Since most of the soil comes under the classification between clay and sand, a typical Coimbatore soil with more clay content, namely; black clay loam soil and pure sand were taken for this study. The mechanical analysis and the physical properties of the black clay loam soil is given below:

Mechanical analysis

Clay	:	39.15%
Silt	:	9.32%
Fine sand	:	16.26%
Coarse sand	:	32.78%

Physical Properties

Apparent specific gravity	:	1.36 %
Pore space	:	60.12 %
Water holding capacity	:	58.6 %

Soil Moisture Content

For the purpose of evaluation of plough disks the MC of soil has been kept as a variable and the reactions were noted at different moisture levels as follows: sandy soil: 5.0, 8.0 and 10.6 per cent and black clay loam soil: 10.60, 13.00 and 17.18 per cent.

Speed of Operation

Agarwal and Rajput (1963-64) stated that the optimum speed of operation of disks is about 6 km/hr. Hence the speed of operation was taken as a parameter in this study. The speeds selected for this experiment were: 1.4 kmph, 7 kmph and 10 kmph. All three speeds are well within the range of tractor operation for working with disk ploughs.

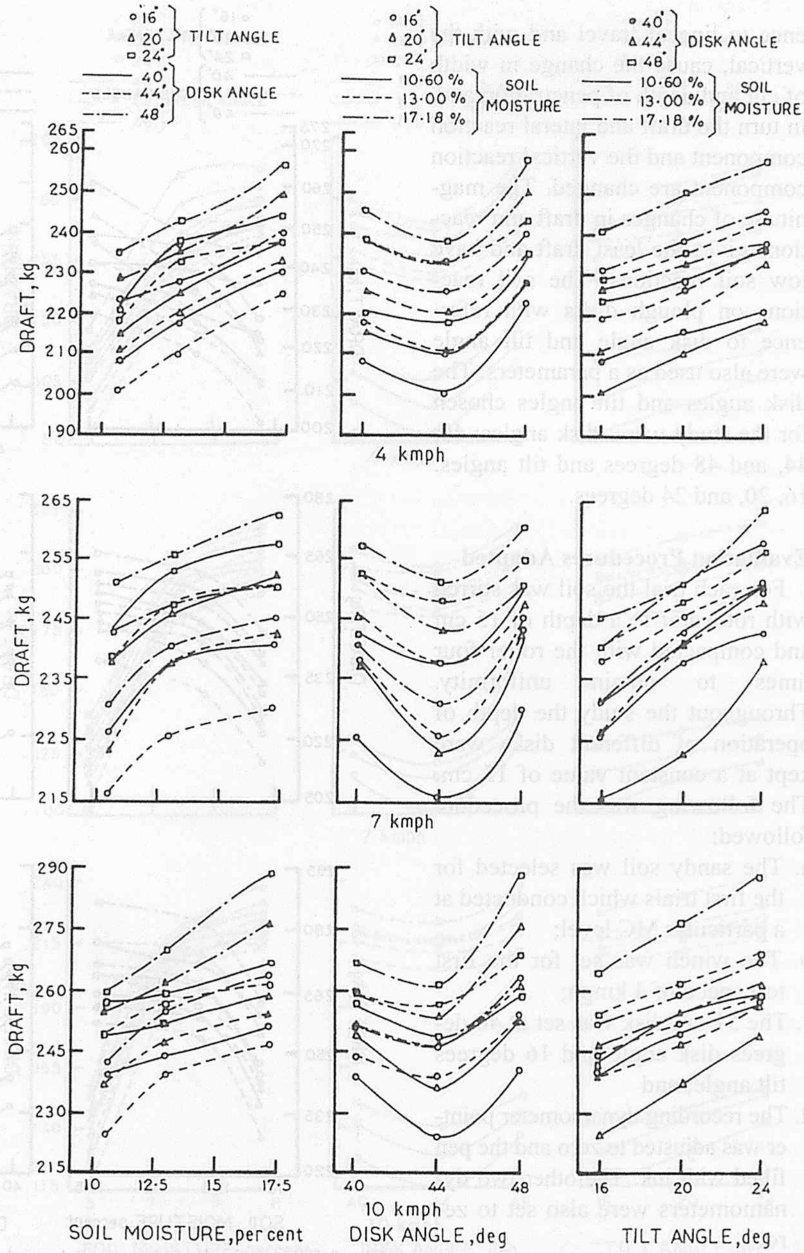

Fig. 2 Effect of soil moisture, disk angle, tilt angle and forward speed for 56cm disk diameter in black clay loam soil.

Disk Diameter

Gordon (1941) and Reed (1948) reported that large diameter disks require lesser draft and also penetration will be high as they have more weight. In India the most common manufactured and used disks used in the study were: 51 cm dia - 6.0 cm concavity - weight 5.00 kg; 56 cm dia 7.5 cm concavity - weight 9.00 kg; and 61 cm dia - 8.5 cm concavity weight 9.5 kg.

Disk Angle and Tilt Angle

It has been reported that the fixing (set up) of the disk with reference to line of travel and with the vertical, cause the change in width of cut and depth of penetration and, in turn the draft and lateral reaction component and the vertical reaction component are changed. The magnitude of changes in draft and reactions gives the least draft and have low soil reactions. The soil reactions on plough disks with reference to disk angle and tilt angle were also used as a parameters. The disk angles and tilt angles chosen for the study were: disk angles: 40, 44, and 48 degrees and tilt angles: 16, 20, and 24 degrees.

Evaluation Procedures Adopted

For each trial the soil was stirred with rotovator to a depth of 15 cm and compacted with the roller four times to obtain uniformity. Throughout the study the depth of operation of different disks were kept at a constant value of 15 cm. The following was the procedure followed:

a. The sandy soil was selected for the first trials which conducted at a particular MC level;
b. The winch was set for the first test speed of 4 kmph;
c. The 51-cm disk was set at 40 degrees disk angle and 16 degrees tilt angle; and
d. The recording dynamometer pointer was adjusted to zero and the pen filled with ink. The other two dynamometers were also set to zero.
e. The trials were conducted until the following observations: pull recorded in the recording dynamometer; side draft; vertical reaction; depth of furrow; and width of furrow.
f. The soil was worked with a rototiller and compaction roller (four times) to bring the soil to original condition.
g. Keeping the speed and disk angle constant, the trial was conducted by changing the tilt angle to 20 degrees and 24 degrees at the same moisture content.

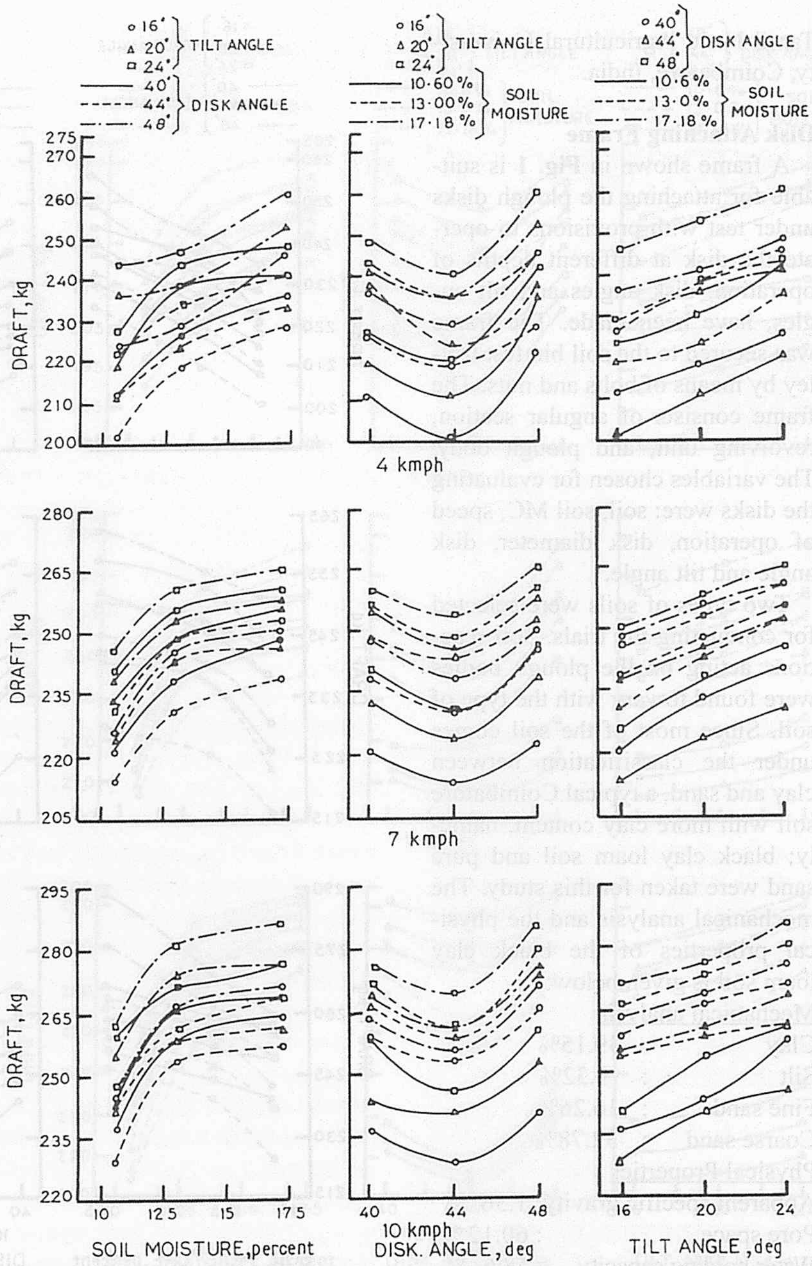

Fig. 3 Effect of soil moisture, disk angle, tilt angle and forward speed for 61cm disk diameter in black clay loam soil.

h. Steps (e) and (f) were repeated.
i. Keeping the speed constant, the tests were repeated by changing the disk angle to 44 degrees and 48 degrees for the tilt angles 16 degrees, 20 degrees and 24 degrees at the same MC level.
j. The same procedure was repeated at 7 kmph and 10 kmph speed repeating steps (e) and (f) after each trial.
k. The soil was watered for changing the MC level and the trials were conducted repeating the steps (b) to (j). Thus the MC levels were varied.
l. The experiment was repeated (steps b to k) for 56 cm and 61 cm disks.
m. The trolley was shifted to other portions of the soil bin containing black clay loam soil and the experiment was repeated as outlined in steps (b) to (l).

Soil Moisture Content

The percentage of soil MC was calculated by the use of gravimetric method on dry basis.

The sample was dried at 105°C for about 24 hours until the weight became constant as follows:

Moisture per cent =[(WM - Wd) / Wd] × 100

where,

WM - weight of wet soil sample

Wd - weight of dried soil sample

Soil Compactness

The soil compactness was determined by using the 'Vicksherg' penetrometer which consists of a sensitive proving ring of 100 kg capacity; extension piece connected to the bottom of the proving ring; and a detachable penetration cone (30° and 5.8 sq.cm. cross sectional area) attached at the base of the extension piece. The penetration load for the proving ring dial was from a calibration chart. The test was conducted at zero level and 15 cm depth from the top. The soil compactness was in terms of soil resistance to penetration.

Example:

Load from calibration chart= 10 kg

Cross sectional area of the cone= 5 cm^2

Therefore soil resistance = 10/5 = 2.0 kg/cm^2

The soil compactness was maintained at a constant value throughout the study by using the compaction roller.

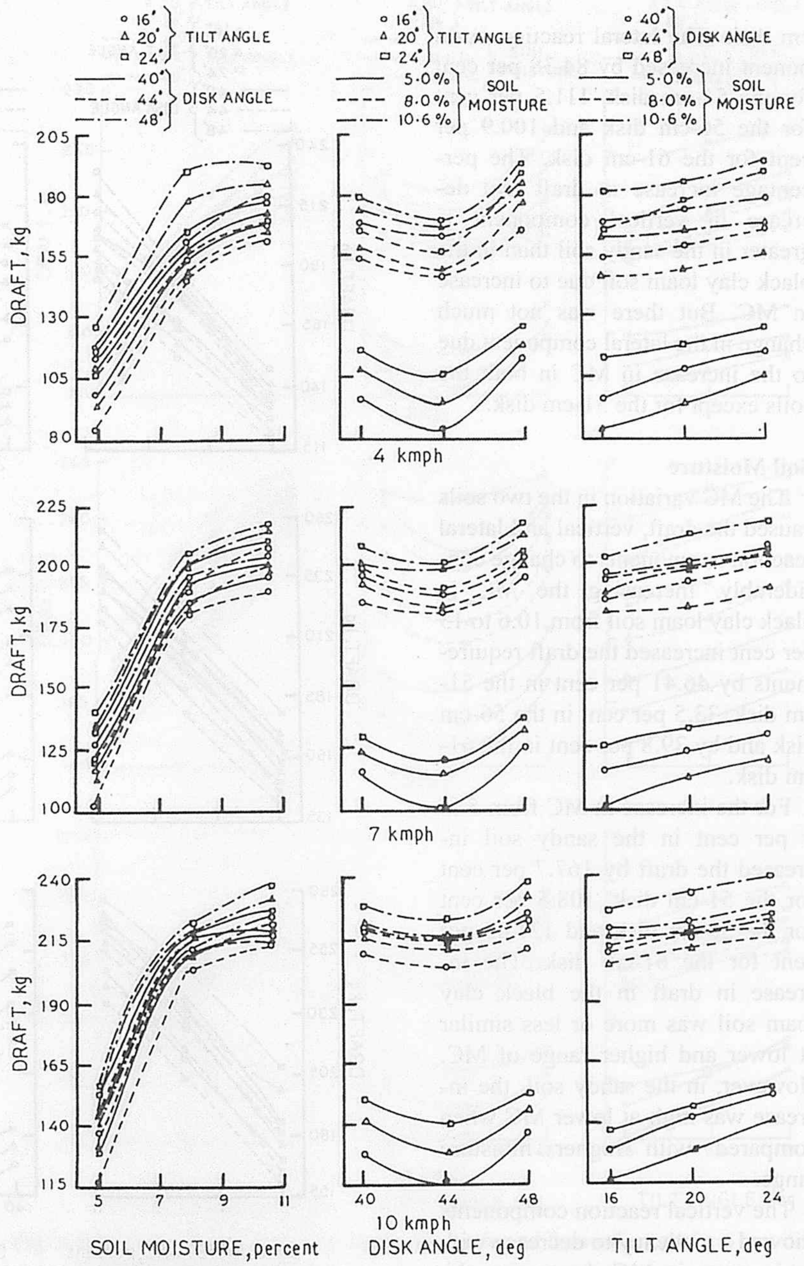

Fig. 4 Effect of soil moisture, disk angle, tilt angle and forward speed for 51cm disk diameter in sandy soil.

Results and Discussion

Black Clay Loam Soil

The performance of 51,56 and 61cm disks in black clay loam is shown in figures 1,2 and 3. In this soil, increasing in MC from 10.5 to 17.18 per cent increased the draft by 50.39 per cent for the 51-cm disk, 30.32 per cent for the 56-cm disk and 42.28 per cent for the 61-cm disk (**Fig. 2**). The upward (+ve) vertical reaction component decreased by 90.80 per cent for the -51-cm disk, 92 per cent for the 56-cm disk and 104.93 per cent for the 61-cm disk, which indicates that as the MC is increased the penetration is deeper. The lateral component increased by 112.3 per cent for the 51-cm disk, 103.7 per cent for the 56-cm disk and 96.30 per cent for the 61-cm disk.

Sandy Soil

The performance of 51, 56 and the 61-cm disks in black clay loam is shown in **Figures 4**, **5** and **6**. In this soil, an increase in MC from 5 to 10.6 per cent, increased the draft by 185.6 per cent for the 51-cm disk, 132.9 per cent for the 56-cm disk and 129.5 per cent for the 61-cm disk (**Fig. 3**). The upward vertical reaction component decreased by 115.8 per cent for the 51-cm disk, 153.8 per cent for the 56-cm disk and 157.21 per cent for the 61-

cm disk. The lateral reaction component increased by 84.38 per cent for the 51-cm disk, 111.5 per cent for the 56-cm disk and 100.9 per cent for the 61-cm disk. The percentage increase in draft and decrease in vertical component is greater in the sandy soil than in the black clay loam soil due to increase in MC. But there was not much change in the lateral component due to the increase in MC in both the soils except for the 51-cm disk.

Soil Moisture

The MC variation in the two soils caused the draft, vertical and lateral reaction components to change considerably. Increasing the MC in black clay loam soil from 10.6 to 13 per cent increased the draft requirements by 46.41 per cent in the 51-cm disk, 33.5 per cent in the 56-cm disk and by 39.8 per cent in the 61-cm disk.

For the increase in MC from 5 to 8 per cent in the sandy soil increased the draft by 167.7 per cent for the 51-cm disk, 108.5 per cent for the 56-cm disk and 121.45 per cent for the 61-cm disk. The increase in draft in the black clay loam soil was more or less similar at lower and higher range of MC. However, in the sandy soil, the increase was high at lower MC when compared with higher moisture range.

The vertical reaction components showed a tendency to decrease with an increase in MC. Increasing the MC from 10.6 to 13 per cent in the black clay loam soil, the vertical reaction component decreased by 89.8 per cent for the 51-cm disk, 92.25 per cent for the 56-cm disk and 87.55 per cent for the 61-cm disk. When the MC was increased in the same soil from 13 to 17.18 per cent, the vertical reaction decreased by 89.26 per cent for the 51-cm disk, 106.68 per cent for the 56-cm disk and 114 per cent for the 61-cm disk.

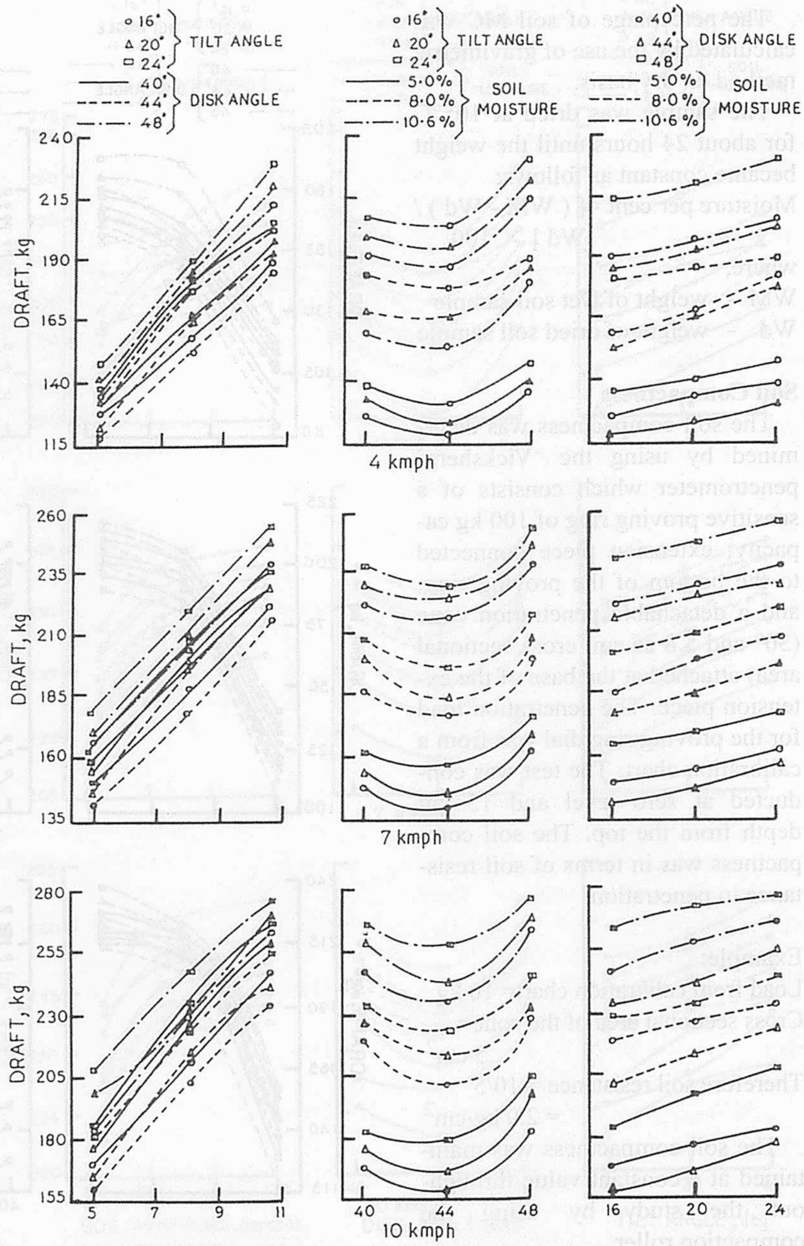

Fig. 5 Effect of soil moisture, disk angle, tilt angle and forward speed for 56cm disk diameter in sandy soil.

When the MC was increased from 5 to 8 per cent in the sandy soil, the vertical reaction component decreased by 102.11 per cent for the 51-cm disk, 137.3 per cent for the 56-cm disk and 151.13 per cent for the 61-cm disk. In the same soil when the MC was increased from 8 to 10.6 per cent, the vertical reaction decreased further by 134.5 per cent for the 51-cm disk, 164.3 per cent for the 56-cm disk and 178.5 per cent for the 61-cm disk. The percentage decrease in the vertical reaction component in the black clay loam soil was similar at both the MC ranges. In the sandy soil, the percentage decrease was greater at high moisture levels.

Increase in soil MC increased the lateral reaction component. For the MC increase from 10.6 to 13 per cent in the black clay loam soil, the lateral reaction component increased by 83.6 per cent in the 51-

cm disk, 76.8 per cent in the 56-cm disk and 92.6 per cent in the 61-cm disk. In the same soil, when the MC was increased from 13 per cent to 17.18 per cent the lateral reaction increased by 75 per cent in the 51-cm disk, 65 per cent in the 56-cm disk and 47.1 per cent in the 61-cm disk. Soil moisture increase in the sandy soil from 5 to 8 per cent caused lateral reaction component increase by 58.27 per cent for the 51-cm disk, 103.8 per cent in the 56-cm disk and 96.2 per cent in the 61-cm disk. When the MC was increased from 8 to 10.6 per cent in the same soil, the increase in lateral reaction was 80.8 per cent in the 51-cm disk, 87.15 per cent for the 56-cm disk and 71.4 per cent for the 61-cm disk. The trend in increase in lateral reaction component was similar in both soils at the corresponding MC levels.

Speed of Operation

Increasing the speed of operation, increased draft and lateral reaction components. The upward vertical reaction component decreased as the speed of operation is increased. Increasing the speed of operation from 4 to 10 kmph in black clay loam soil, increased the draft by 29.7 per cent for the 51-cm disk, 22 per cent for the 56-cm disk and 17.45 per cent in the 61-cm disk. For the same range of increase in speed from 4 to 10 kmph in sandy soil, the draft is increased by 89.2 per cent for the 51-cm disk, 85.9 per cent for the 56-cm disk and 70.8 per cent for the 61-cm disk. Increase in speed increased the draft in both soils but was 3 times in the sandy soil to that of the increase in draft in black clay loam soil.

With an increase in speed from 4 to 10 kmph, the lateral reaction component increased by 75 per cent for the 51-cm disk, 77.03 per cent for the 56-cm disk and 60.6 per cent for the 61-cm disk in black clay loam soil. In sandy soil for the

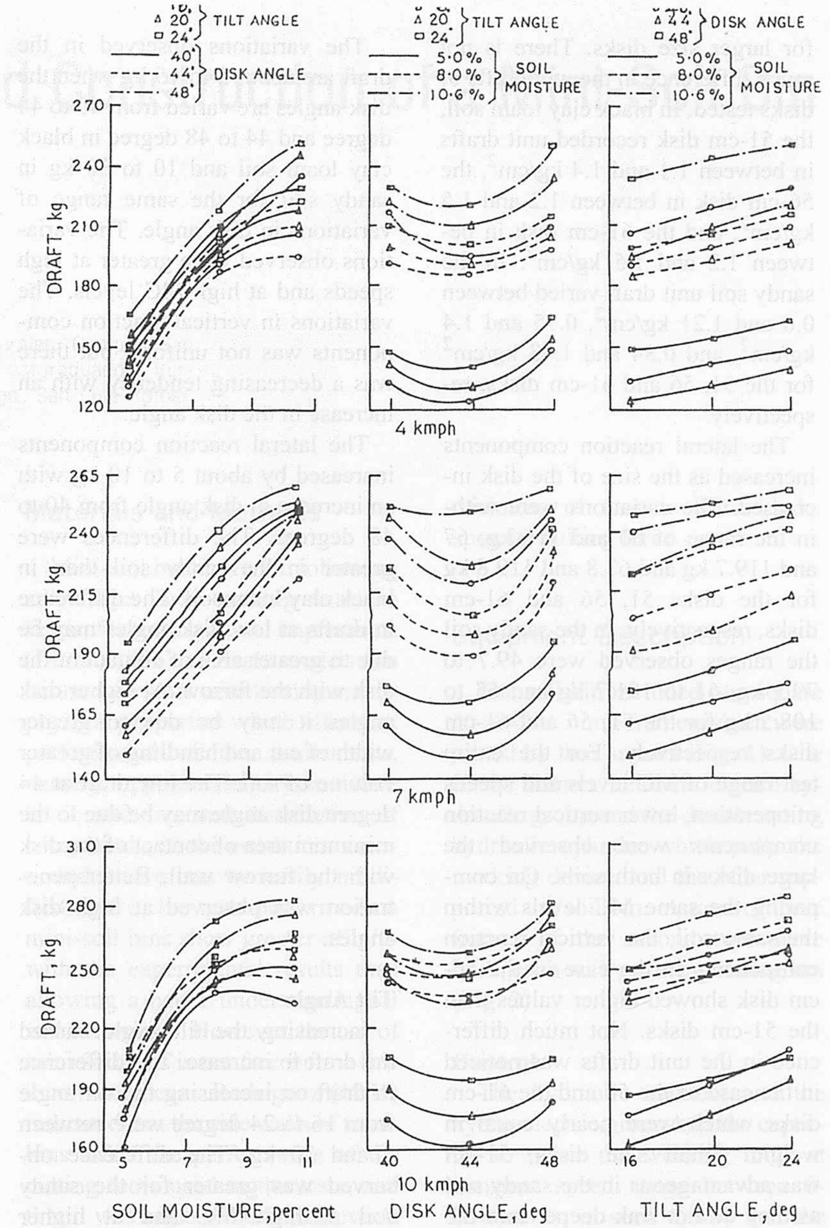

Fig. 6 Effect of soil moisture, disk angle, tilt angle and forward speed for 61cm disk diameter in sandy soil.

same range of speeds the lateral reaction component increased by 58.4 per cent for the 51-cm disk, 62.2 per cent for the 56-cm disk and 45.4 per cent for the 61-cm disk. The increase in lateral component due to an increase in speed was greater in the black clay loam soil than in the sandy soil. As the speed increased from 4 kmph to 10 kmph, the upward vertical reaction component decreased. In the black clay loam soil, the decrease was 84.2 per cent for the 51-cm disk, 108.3 per cent for the 56-cm disk and 109.4 per cent for the 61-cm disk. In the sandy soil, the upward vertical reaction component increased by 131.7 per cent for the 51-cm disk, 107.3 per cent for the 56-cm disk and 113.2 per cent for the 61-cm disk.

Disk Sizes

High values of draft and lateral reaction component are recorded for larger size disks. There is not much difference in the unit drafts of disks tested. In black clay loam soil, the 51-cm disk recorded unit drafts in between 1.1 and 1.4 kg/cm^2, the 56-cm disk in between 1.2 and 1.5 kg/cm^2, and the 61-cm disk in between 1.2 and 1.5 kg/cm^2. In the sandy soil unit draft varied between 0.6 and 1.21 kg/cm^2, 0.75 and 1.4 kg/cm^2, and 0.84 and 1.43 kg/cm^2 for the 51, 56 and 61-cm disks, respectively.

The lateral reaction components increased as the size of the disk increased. The variations were within the range of 60 and 116 kg, 67 and 119.7 kg and 67.8 and 119.8 kg for the disks 51, 56 and 61-cm disks, respectively. In the sandy soil the ranges observed were 49.7 to 79.1 kg, 61 to 101.7 kg and 65 to 108.5 kg for the 51, 56 and 61-cm disks respectively. For the entire test range of MC levels and speeds of operation, lower vertical reaction components were observed the large disks in both soils. On comparing the same MC levels within the same soil, the vertical reaction components in the case of the 56-cm disk showed higher values than the 51-cm disks. Not much difference in the unit drafts was noticed in the case of the 56 and the 61-cm disks which were nearly equal in weight. Small size disks, 51-cm was advantageous in the sandy soil as they do not sink deeper into the soil.

Disk Angle

Drafts observed were minimum at 44 degrees than observed at the 40 and 48 degrees angles. Slight increase was draft is noticed at the 48 degree disk angle over the 40 degree disk angle. The lateral reaction components showed a tendency to increase with the increase in disk angle. The vertical reaction components decreased as the disk angle increased. This implies that the penetration is easy at large disk angles.

The variations observed in the draft are about 5 to 15 kg when the disk angles are varied from 40 to 44 degree and 44 to 48 degree in black clay loam soil and 10 to 30 kg in sandy soil for the same range of variations in disk angle. The variations observed were greater at high speeds and at high MC levels. The variations in vertical reaction components was not uniform but there was a decreasing tendency with an increase in the disk angle.

The lateral reaction components increased by about 5 to 10 kg with an increase in disk angle from 40 to 48 degrees. The differences were greater in the sandy soil than in black clay loam soil. The difference in drafts at low disk angles may be due to greater area of contact of the disk with the furrow. At higher disk angles it may be due to greater width of cut and handling of greater volume of soil. The low draft at 44 degree disk angle may be due to the minimum area of contact of the disk with the furrow wall. Better penetration was observed at high disk angles.

Tilt Angle

Increasing the tilt angle caused the draft to increase. The difference in draft on increasing the tilt angle from 16 to 24 degree were between 5 and 30 kg. The difference observed was greater for the sandy soil at high MC and at higher speeds. The vertical reaction components showed high upward values for high tilt angles which means that the penetration is poor at higher tilt angles. The differences found were not uniform. The smaller disk showed greater upward values in the case of the black clay loam soil at low MC levels. The values noted at higher speeds were highly variable. The lateral reaction components showed decreasing values with an increase in the tilt angle. The difference is about 5 kg to 10 kg with an increase in tilt angle. The minimum values of draft were observed at the 16 degree angle of inclination.

Conclusion

Soil: The drafts and lateral reaction components showed high values in black clay loam. The vertical reaction decreased with an increase in soil MC in both soils. At high moisture levels, the increase was diminishing slightly in the black clay loam. The tendency to increase was continuous in the sandy soil.

Soil moisture: The increase in draft in black clay loam soil was less when compared to the sandy soil on increasing soil MC. The percentage decrease in vertical reaction component in the black clay loam soil was the same for 10.6 to 13 per cent and 13 to 17.18 per cent in soil MC levels. In the sandy soil, greater sinking of disks was observed when moisture is increased. In both soils the lateral reaction increased with an increase in soil MC. The soil MC ranged between 13 and 17.18 per cent which may provide good working condition in black clay loam soil.

Speed of operation: At 4 kmph speed the drafts and lateral reaction components observed were low for a particular disk at a particular MC level, but the observed vertical reaction components showed more positive values. At 10 kmph the drafts and the lateral reaction component were greater and the vertical reaction component showed erratic variation showing a tendency to move towards the negative side. At 7 kmph speed the drafts and the soil reactions were observed to be balanced which suggests the speed about 7 kmph may be the right operating speed.

(Continued on page 38)

Development and Construction of a Mini-Soil Bin

by

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Abstract

The development and construction of a mini-soil bin was addressed in order to study in laboratory the mechanical behaviour of agricultural soils handled under different implements. The implement was attached to the cylinder rod through a dynamometer (to measure the horizontal and vertical forces and their momentum), and its operating speed was controlled by a flow regulating valve (variable orifice). The mini-bin consisted of two structures: one of them held the electro-hydraulic group and the other a double-acting hydraulic cylinder which pulled the working device. The soil bin was placed on the latter.

Introduction

The soil bins are very useful tools to analyze the tearing of the soil through the action of tilling implements, and the compacting processes due to the movement of wheels on the soil. The high cost of a duct on which actual tilling implements could be tested has stirred a growing interest toward using small bins or mini-bins which were utilised to test scale models of actual small, geometrically simple working tools. The main objective of this study was to develop and build a mini-soil bin.

Materials and Methods

Mini-soil bins have the following advantages over making trials in fields, for testing actual implements (Durán, 1996): a) Scale models can be used on soils with homogeneous physical properties which affect the performance of the implements, namely; apparent density, moisture level, and size and distribution of compounds; b) the conditions mentioned above can be easily modified, and c) the tests performed on mini-soil bins show greater affinity with the experimental results thus allowing a better understanding of soil behaviour. However, the use of mini-soil bins is not free from a few inconveniences. Rodríguez (1981) indicated that the side walls of mini-bins affect the performance of the cutting tools used in the tests, and that it was impossible to achieve in mini-soil bins the soil structure found in natural conditions.

Payne (1956) was the first who used a mini-soil bin to study soil behaviour under the action of a tilling implement. Osman (1964) built a bin with glass walls to observe the soil deformation which occurs just before it breaks when pushed by a blade. Darmora and Pandey (1995) used a mini-bin to evaluate the behaviour of different types of furrow-making blades, used in gush planters. Fielke (1996) used a mini-soil bin to study the effect of the cutting angle and the edge thickness of a wide brim blade, upon its pulling force.

Equipment Description

The soil bin included two square 40 × 40 × 3 mm frames which were joined by four screws (**Fig. 1**). One of the frames held the electro-hydraulic group and the double acting hydraulic cylinder which pulls the working device, and the other supported the soil bin. A total of three frames of the second type were built, each with a corresponding bin (**Fig. 2**).

Controls

The hydraulic cylinder and the hydro-electric group had the capacity to move the working tools at constant speed range between 0-350 mm/s, in such a way that the movement would be uniform, and there would be no vibration. The linear speed of the cylinder rod was controlled through a flow regulating valve and its course through a directional electro-valve. The possible rotation of the cylinder rod

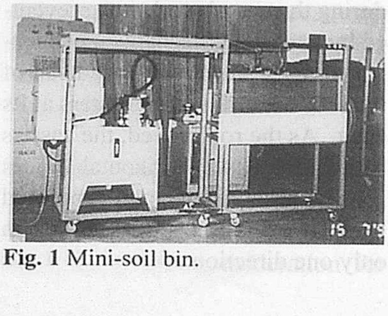
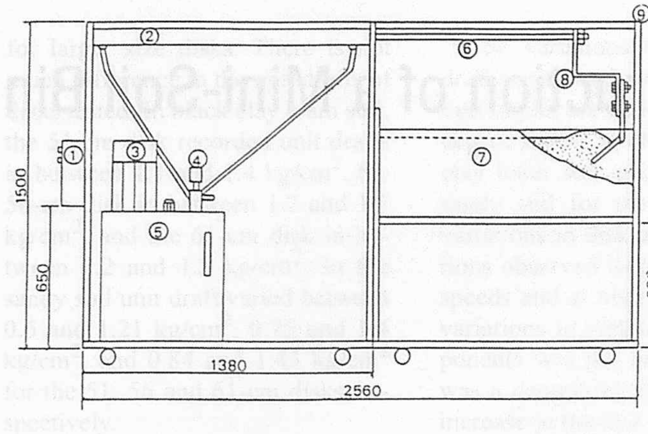
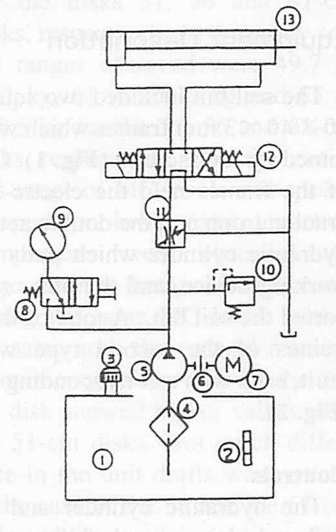


Fig. 1 Mini-soil bin.



1. Electric control
2. Hydraulic cylinder
3. Motor
4. Directional valve
5. Oil tank
6. Ram
7. Bin
8 Dynamometer
9. Structure

Fig. 2 Mini-soil bin with dimensions.



1. Oil tank
2. Oil temp level
3. Air breather
4. Suction line filter
5. Pump
6. Drive coupling
7. Motor
8. Relief valve
9. Pressure gauge
10. Solenoid valve
11. Caudal regulator
12. Electro-valve
13. Cylinder

Fig. 3 Schematic of hydraulic circuit.

during the actual work was prevented by a guiding mechanism consisting of a "T" attached to the head of the rod, which had two casters at its ends. As the rod moved, the casters rolled along two horizontal braces parallel to the frame, and supported by it over the rod, thus yielding in only one direction.

The inside diameters were 63 mm for the hydraulic cylinder and 36 mm for the rod. The rod tolerated a maximum axial force of 20 kN, and its effective run was 1250 mm.

Figure 3 shows a complete diagram of the hydraulic circuit which activated the cylinder. The gear pump had a 54 liter/min flow which was placed in the interior of a 120 l oil tank. The pump was activated with a 9.5 kW three-phase motor, which rotated at 1500 rev/min. The oil propelled by the pump travelled through the flow regulating valve (with regulating capacity of 0-46 l/ min.) before getting to the four-mode, three-position directional electro-valve which guided the dual function cylinder.

There are two advantages in the use of this type of cylinder. First, the working tool can return to its initial position after performing the working run, and second, the run can be stopped at any moment. The hydraulic circuit was completed by a regulating pressure valve at 0.85 kPa set pressure, and a manometer with a range between 0-1.6 kPa.

Bin Design

The bin was 1000 mm long, 400 mm wide and 400 mm deep. Its structure was made of 30 × 30 mm per side, 3 mm thick, steel angles

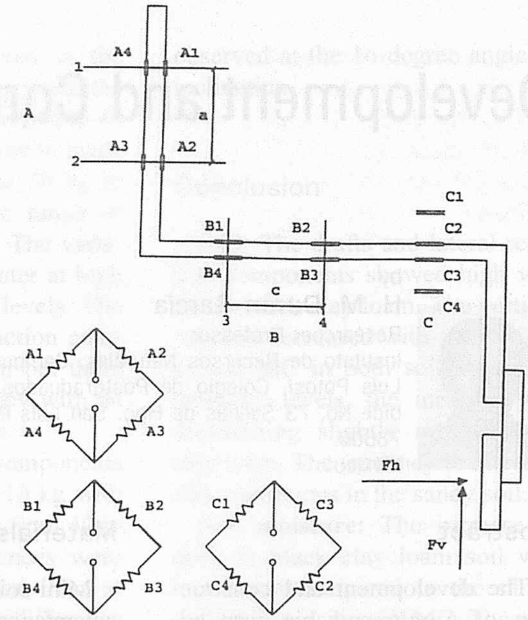


Fig. 4 The load cell.

which supported the acrylic side walls of 15 mm thick.

Design, construction and calibration of the dynamometer

This device was built using an F-111 (0.2% C) "L" shaped piece of steel, with maximum elasticity of 250 MN/m^2. **Figure** 4 shows the way three Wheatstone bridges (A, B, and C) were positioned on the "L". The cross section of the dynamometer was a rectangle, 10 mm high and 60 mm long. The dynamometer measured the horizontal and vertical forces and their momentum.

All the Wheatstone bridges had four Cu-Ni strain gages with self-temperature-compensation and a 2.1 factor for an operating range between 10 and 100^{o}C.

The strain gages on bridges A and B were used as differential arms. Thus, they measured the force value independently from the element to which it was applied. On bridge C the gages had four active arms to measure the momentum originated by the force.

(Continued on page54)

Development of a Tractor Front-mounted Pineapple Plant Dressing Machine

by

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Abstract

A pineapple plant dressing machine, for mounting at the front of a tractor, was developed using the concept of dressing the plant top material by rotating circular saw blades. Four blades, arranged in a row, were used with a specially designed power transmission system at the front of the tractor. The cutting width can be adjusted to cater to the variation of row-to-row distance in different plantations. The developed machine was tested in actual field conditions. The performance was found to be satisfactory with an average effective field capacity of 0.45 ha/hr. The field and cutting efficiencies were found to be about 80 and 95 per cent respectively. The fuel consumption was found to be 2.68 liters per hour with cost of operation at US$3.6 per hour or US$8 per hectare against US$19 per hectare for manual dressing. The break-even point for this machine was 126 hectare in a year. The pay-back period for the machine was calculated at three years. Due to the simplicity of the design, it will be possible to fabricate this machine in local workshops.

Introduction

Pineapple is one of the world's most unique and exotic tropical fruits. It has been grown in Thailand for several centuries. It is an important and popular tropical fruit that has a high export demand. Currently Thailand is the largest producer of pineapple in the world (FAO, 1996).

The vegetative reproduction of pineapples is through the crown, slips, shoots, suckers, or ratoons. In Thailand, farmers usually transplant shoots. For healthy shoots, it is essential to dress the pineapple plants. Moreover, dressing may help to store the total energy in the shoots, generally extracted by the roots, after harvesting. This would produce healthy shoots that can reduce the crop cycle and prove beneficial to the farmers, reducing the production cost and enhancing the quality of pineapple, thereby increasing earnings.

Pineapple is a labour-intensive crop and a long knife is used to dress the plant. Due to shortage of labour and the need to cultivate more because of the export orientation of the business, the farmers engaged in pineapple cultivation favour mechanization, at least for a few operations. An attempt was made to develop a pineapple plant dressing machine (Bora, 1997).

Materials and Method

In order to develop this dressing machine one has to consider critically the pineapple plants characteristics and the principles of plant cutting devices. Pineapples are always propagated vegetatively for commercial production. Suckers, shoots, slips, crown or butts are used for vegetative propagation. All these planting materials have considerable resistance to desiccation and may be stored for several weeks before planting (Purseglove, 1988).

The cutting process follows many steps before the final separation of the cut materials (Persson, 1987). Oduori (1988) designed and developed a revolving knife-type sugarcane basecutter using four smooth blades. Shearer et al. (1991) developed and tested a cut-off saw mechanism for selective harvest of broccoli. The cut-off mechanism, consisting of a hydraulic motor and saw blade mounted on a hydraulically actuated swing arm, was attached to the right side of the tractor between the front and rear wheels. Roy (1996) developed a power tiller operated pineapple plant dressing machine with two rotary saw blades. However, the machine could perform well only in short plant heights and was subject to operator drudgery.

Machine Fabrication

The details of cultivation practices followed on local plantations were taken into consideration for the design of this machine. Generally the shoots are planted on both sides of a flat ridge at a distance of 60 cm. Two ridges are separated by a furrow 70 cm wide on the top. The ridge is generally 15 cm high and the pineapple plants are dressed at a height between 25 and 35 cm.

This machine was mounted on a tractor because of large power output, operator comfort, higher field capacity and more area coverage per unit of time. The basis for front mounting the dressing machine was that the tractor wheels were expected to press the scattered cut leaves into the furrows which would act as mulching as well as reduce soil loss. The machine had four cutter blades placed in a single row and it was mounted at the front of the tractor. It was driven by the tractor engine through a belt and pulley mechanism which drove a disc clutch on a flywheel. The flywheel was connected to the differential gearbox through a shaft. The differential gear ran two bevel gears on its two sides. Each bevel gear drove a chain sprocket on each side with the help of a coupling and vertical shaft. Each chain sprocket ran the outer shaft on either side to rotate the cutter blade. The pineapple plants were dressed by the action of rotation of the blades.

The cutter blades were of 35 cm diameter with 80 teeth. Each blade was connected to a vertical shaft through a hub which gave mechanical support to the blade. The direction of rotation of each blade was such that the cut leaves were thrown into the furrow. The frame of the machine was fabricated by mild steel hollow square tubing. The position of the frame could be adjusted according to the height of cut. The chain casing was sealed to store lubricating oil.

Performance Test and Economic Evaluation

The performance of the developed machine was evaluated through field capacity, field efficiency, cutting efficiency and fuel consumption in the field as per test procedures given by RNAM (1995). The economy of operation of a machine is an important factor for its adaptation by farmers, especially in developing countries (Hunt, 1963). The cost of operation was estimated to predict the adaptability of this machine. It can be calculated through the number of hours the machine is utilized in a year. The break-even point was also calculated to see the area at which it will be economical to own the machine and the pay-back period during which the farmer gets back the capital invested in the machine.

Testing and Development

The pineapple plant is dressed manually in Thailand and this test was undertaken to see whether the developed machine would be able to replace the present manual dressing. The test was undertaken at the SAICO plantation in Rayong. During the preliminary testing, the problem of wrapping of the blade shaft was faced and was checked by blocking the gap between the frame and the blade with metal plates on the hub. A belt was worn due to the excessive tension and took a long time to replace it. Moreover, the burning of a clutch plate due to overload was also encountered in the field test.

Despite these problems, the farmers were satisfied with the performance and cutting action of the machine. Based on the results obtained during its testing, the machine was modified. A schematic diagram of the machine is shown in **Fig. 1**. For the modified design, the pulley-belt transmission system was discarded and a direct power transmission shaft was used to transmit power from the engine to the flywheel. This was done in order to avoid the difficult and time-consuming repair of the pulley-belt system. For this version of the machine, a bigger clutch plate with a bigger flywheel was used to overcome difficulties faced by the first prototype. The flywheel with the clutch plate transmitted power to the differential gear. The differential gear transmitted power to the bevel gears on each side through a shaft. The two bevel gears on each side transmitted power to the outer bevel gear through a sliding shaft connected with the help of a coupling. This helped to vary spacing of the outer blades. A bigger blade of 46 cm was used to increase the

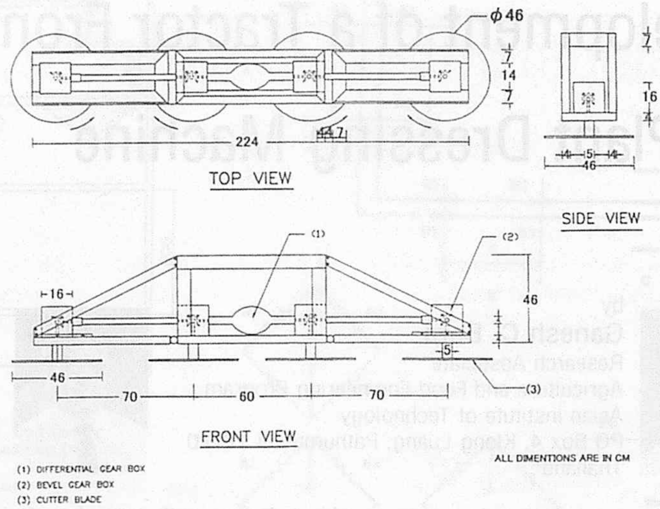

Fig. 1 A schematic of machine.

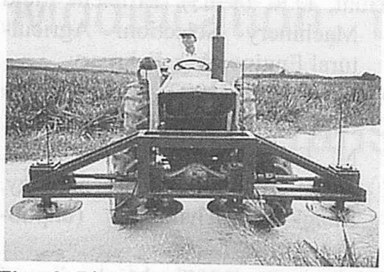

Fig. 2 Pineapple plant dressing machine.

cutting width and thereby the cutting efficiency.

The spacing between the inner blades was kept at 60 cm. The outer blades were kept at a distance of 70 cm, but they could slide inside by 10 cm if the row spacing changed. The frame was made out of mild steel hollow bars and it was in three pieces. The outer two portions can be detached from the inner portion of the frame to facilitate transportation of the machine, off-farm operation of the tractor, mounting the machine or allowing work with only two blades or the topography or shape of the plantation requires.

The modified machine was tested in the field. The problem of cutting height and proper throwing of cut leaves was faced due to smaller gaps between the blades and the frame. The hub was modified to increase the gap for proper throwing of the leaves and to decrease the height of cut to 30 cm. Four pieces of iron rod, each of 1.5 cm diameter, were fitted to each gearbox as a marker to enable the operator to know the position of the cutting blade. **Fig. 2** shows a machine mounted on the tractor. The rotation of the blades were such that the cut leaves were thrown in the furrow and the tractor wheels ensured the pressing of the cut leaves into the furrow. After dressing of the pineapple plants, the outer sections at both ends of the frame can be detached along with the blades for safe road transport of the tractor.

Results and Discussion

Field Performance

The field tests were conducted on seven plots of varying sizes. Tests were conducted at different forward and engine speeds thereby with various blade speeds. The theoretical field capacities, effective field capacities, field efficiencies and cutting efficiencies for each replication were calculated and the fuel consumption in each trial was recorded.

Though the cutting width of the machine was constant for all the trials, the theoretical field capacity varied from plot to plot because of the different forward speeds varying from 0.54 to 0.61 ha/hr with an average theoretical field capacity of 0.57 ha/hr. On the other hand, the effective field capacity varied from 0.41 to 0.53 ha/hr with an average effective field capacity of 0.45 ha/hr. This is because of the varied field sizes and shapes and different times lost in turning the tractor.

The field efficiency for these plots was quite high, ranging from 74.58% to 88.14%, with an average of 79.89%. The variation was due to the number of passes the tractor had to make in different plots and the availability of space for efficient turning of the tractor. The cutting efficiency varied from 94.68% to 96.12% with an average of 95.11%. The higher cutting efficiency was due to the size of the blades, which covered almost all plants compared to the previous small blades. Fuel consumption ranged from 2.50 l/hr to 2.87 l/hr with an average of 2.68 l/hr. The fuel consumption depended upon continuous work for a long time and would decrease with length of operating time. The height of cut was satisfactory and varied from 29 to 32 cm at an average of 30.4 cm.

Fig. 3 shows the trend of theoretical and effective field capacities with the forward speeds. It was observed that an increase in forward

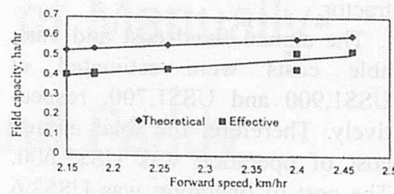

Fig. 3 The relationship of the oretical and effective field capacity with forward speed.

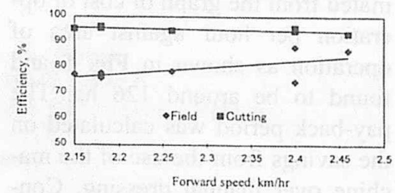

Fig. 4 The relationship of the cutting and field efficiency with forward speed.

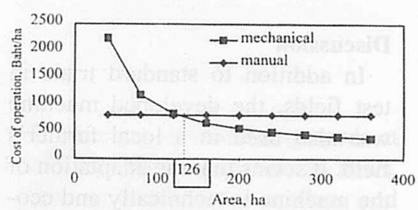

Fig. 5 Cost per hectare against area, showing break-even area.

speed resulted in an increase in field capacity. The values of the theoretical field capacities as expected, were higher than the effective field capacities. **Fig. 4** shows the relationship between the cutting and field efficiencies, and the forward speed. Cutting efficiency did not show any significant change with the change in speed. However, field efficiency increased with speed.

Apart from parameters of the machine performance, the economy of operation is an important factor which will determine its adaptation by the farmers. Based on the assumption that the tractor is fully dedicated to the machine with 1000 hours of annual use, and taking standard rates for fixed costs and variable costs, an economic analysis was performed to estimate the cost of operation. The cost of the machine was estimated at US$10,000, including the cost of a tractor.

The annual fixed cost and variable costs were estimated at US$1,900 and US$1,700, respectively. Therefore, the total annual cost of operation was US$3,600. The cost of operation was US$3.6 per hour or US$8 per hectare.

The break-even point was estimated from the graph of cost of operation per hour against area of operation as shown in **Fig. 5** and found to be around 126 ha. The pay-back period was calculated on the savings from the use of the machine over manual dressing. Considering 1000 hours of annual use, the pay-back period was calculated at 2.04 years.

Discussion

In addition to standard trials in test fields, the developed machine was also used in a local farmer's field. It seems that the adaptation of the machine is technically and economically feasible. Considering the simple design, it is possible that the machine can be fabricated in local workshops.

For the developed machine, it was found that the cutting performance and field capacity were satisfactory for the farmers. The machine can dress an average of 0.45 hectare per hour in comparison to manual dressing of 0.02 hectare per hour per person. Considering eight hours of work per day, the machine can cover 3.6 ha per day whereas this area needs 22.5 man-days to cover it by manual dressing at a rate of 0.16 ha per man-day. Economically, the machine would be viable looking at its per hour cost of operation which is estimated at US$3.6 per hour only and needs US$8 to dress a hectare of pineapple plantation. On the other hand, manual dressing needs US$19 to dress the same area.

The migration of agricultural labour to the industrial sector greatly necessitated the development of such a machine to replace the time-consuming and expensive manual dressing. In the industrial sector, the workers have lighter and easier jobs in contrast to the back-breaking job of pineapple plant dressing. Even the small farmer who does all the operations himself finds difficulties with this job. This machine will not only replace the drudgery of human labour but also save time and money for the farmer enabling him to take more interest in plantation and other related works.

Conclusions

The cutting action of the developed pineapple plant dressing machine was satisfactory. The use of this machine would be advantageous as it could place the cut leaves into the furrows with uniform cutting height. Moreover, the machine is technically and economically efficient to replace manual dressing. The design of the machine is not very complicated which will enable artisans to manufacture this it in local workshops. The present study tried to incorporate all the parameters for developing this machine. Still it has to go through various developmental processes necessary for its adoption in varied field conditions. Hydraulic power may be used to vary the height of cut during operation as well to rotate the cutting blades. This will eliminate the mechanical drive.

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Modification, Test and Evaluation of Manually-Operated Transplanters for Lowland Paddy

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Abstract

Experiments were conducted for the improvement of the IRRI-designed, 6-row manually-operated rice transplanter to adapt in rice producing countries. After modification, a 5-row prototype was developed and both IRRI and BRRI transplanters were evaluated compared with hand transplanting method. At full load the weights of IRRI and BRRI transplanters were 37 kg and 30 kg, respectively. The dragging force of the BRRI transplanter was reduced by 25-30% compared with IRRI transplanter. The wooden skid of the IRRI transplanter was replaced by a skid made of G.I. sheet in BRRI transplanter which made it durable, light weight and incurred less sliding resistance. The circular configuration of the picker finger of the IRRI transplanter was modified to a semi-circular one, which improved its cutting action. The effective field capacities of BRRI transplanter, IRRI transplanter and hand transplanting methods were 0.0191 ha/hr, 0.0155 ha/hr and 0.0023 ha/hr, respectively. As a result, a 20% increase in working capacity was achieved with the BRRI transplanter compared to the IRRI transplanter. The field efficiencies of BRRI and IRRI transplanters were 78.90% and 76.83%, respectively, but that of hand transplanting method was 91.40%. The cost of machine transplanting with wooden frame, bamboo frame, plastic frame, plastic tray and nylon rope and nursery seedlings were US$ 60.96/ha, US$ 54.98/ha, US$ 58.39/ha, US$ 109.76/ha and US$ 47.56/ha, respectively. On the other hand, the hand transplanting cost with wetbed nursery seedling was US$ 93.18/ha. Therefore, machine transplanting with any of the nursery seedlings methods except plastic tray, was more profitable compared to hand transplanting method.

Introduction

Rice is an important crop and staple food in most of the countries in Asia and the Pacific, and is generally grown under wetland condition. It may be planted in a number of ways, i.e., broadcasting seeds at random, planting seeds in rows, planting seedlings at random and planting seedlings in rows. The work may be done either manually or mechanically. In many of the rice producing countries, more than 90 percent of the ricecrop are transplanted. In Bangladesh, out of 14.4 million hectares, about 9.17 million hectares are under cultivation. The cropping intensity in the country at present is 171%. Rice is grown in almost 80% of the arable land. About 90% of the paddy crop is grown in transplanted condition and the rest is direct-seeded.

In China, Shen (1934) observed that transplanted rice gave higher yield than direct-seeded rice. The reason was that transplanting provides a better stand and more effective weed control. In India, Ramiah and Hanumontha (1936) showed that higher and more stable yield was obtained from transplanted rice than direct-seeded rice. Ghose et al. (1960) also reported that in most provinces of India, transplanted rice yielded 10 to 20 percent higher than broadcast rice. In the Philippines, Bautista (1938) reported that trans-

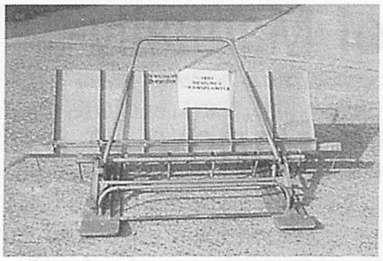

Fig. 1 IRRI-designed transplanter.

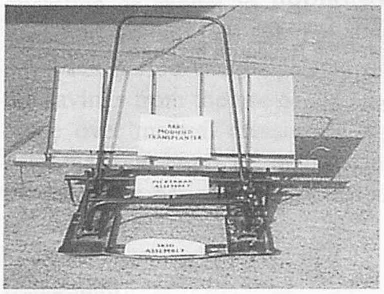

Fig. 2 BRRI-modified transplanter.

planted rice produced 15 to 20 percent more yield than broadcast rice. Besides, it has some added advantages compared to direct seeding, i.e., better water and weed control, uniform ripening and less lodging. Studies at IRRI (1970) showed increased yield in transplanted paddy compared to direct seeding with normal tillage practices. Devasundrarajah (1971) reported that there are two clear advantages in transplanting system of crop establishment, i.e., the transplanted paddy occupies the field for lesser time than the direct seeded paddy and facilitates the control of weeds. Human stress and drudgery are too much in rice transplanting. Islam (1993) conducted a study on transplanting paddy in the Sreepur area of Gazipur district, Bangladesh. He found that 400-450 man-hr/ha was necessary for hand transplanting in rows, but in the case of random transplanting, the labour requirement was 300-350 man-hr/ha. In recent years, due to increasing employment opportunities in several developmental activities and expanding educational facilities for rural youths, there is often an acute labour shortage at the time of transplanting. As a result, timeliness in this operation is not achieved and hence production goes down. Therefore, the manually-operated rice transplanter developed by the IRRI was modified and evaluated in order to adapt it among the farmers in the rice growing countries.

The objectives of the study were: to modify the IRRI-developed manually-operated TR4 transplanter to adapt in the rice producing countries; to determine field performances of IRRI-designed and BRRI-modified transplanters compared with hand transplanting method and to compared the economics of machine transplanting with different seedling raising methods compared to hand transplanting.

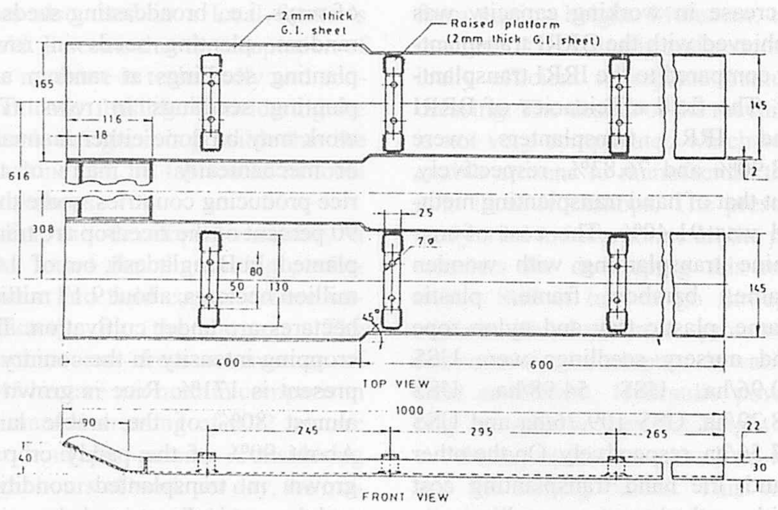

Fig. 3 BRRI-modified transplanter skid.

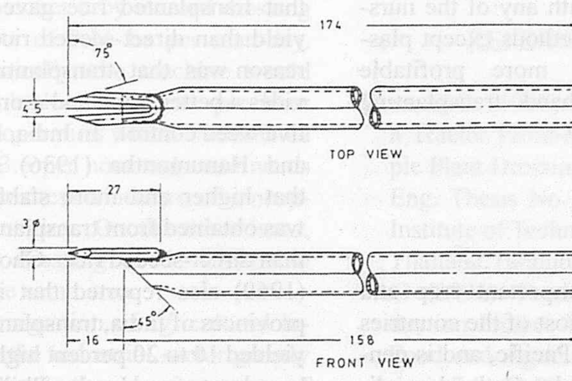

Fig. 4 IRRI TR4 transplanter picker configuration.

Materials and Methods

Modification of IRRI TR4 Transplanter

In order to reduce the weight of the transplanter, the number of rows of the IRRI prototype was reduced from 6 to 5 in the BRRI prototype (**Fig. 1** and **Fig. 2**). The overall weight of the modified transplanter was 21 kg, however, that of the previous prototype was 24 kg. The wooden skid of the existing prototype was replaced by a skid made from G. I. sheet in order to reduce adhesion force between the soil and the skid (**Fig.3**). Another advantage of the metal skid was that it did not absorb water. In the case of the wooden skid, there was a possibility of absorbing water and that could in-

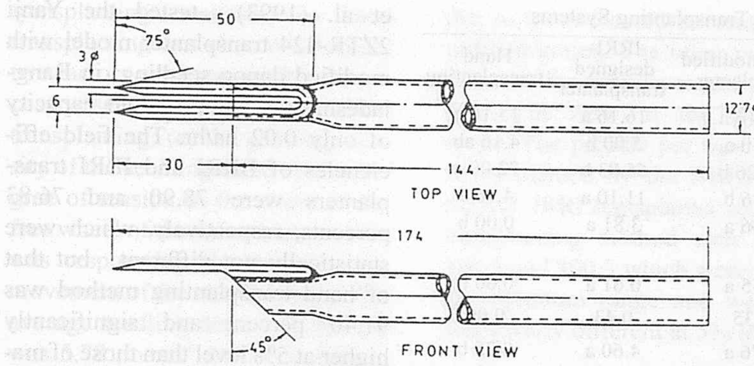

Fig. 5 BRRI-modified transplanter picker configuration.

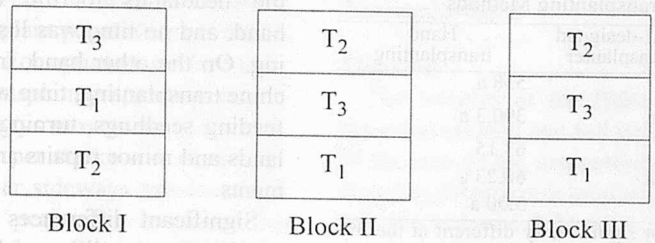

Fig. 6 Layout plan of experiment.

Table 1. Time Utilized in Transplanting by Different Methods and Their Respective Field Performances

Operation	Time consumed (hr/ha)		
	BRRI-Modified Transplanter	IRRI-designed Transplanter	Hand Transplanting
Planting	41.66 (79.30)	49.75 (76.80)	402.77 (90.86)
Turning	1.58 (3.00)	2.88 (4.45)	
Feeding seedlings	6.25 (11.9)	6.35 (9.81)	40.52 (9.14)
Adjustment/ repair	3.04 (5.80)	5.80 (8.94)	
Total	52.53 (100)	64.78 (100)	443.29 (100)
Effective field capacity (ha/hr)	0.0191 a	0.0155 a	0.0023 b
Theoretical field capacity (ha/hr)	0.0242 a	0.0200a	0.0025 b
Field efficiency (%)	78.90 b	76.83 b	91.40 a

crease the overall weight of the machine. The round cross section of the picker fingers were modified to a semi-circular one in order to improve the seedling picking efficiency (**Figs. 4** and **5**).

Field Test of Transplanters

Experiments were conducted at the Bangladesh Rice Research Institute (BRRI) farm in silty clay loam soil to evaluate the performances of BRRI modified transplanter and IRRI designed transplanter compared with the hand transplanting method. The treatments were as follows:

T_1 = Transplanting by BRRI modified transplanter

T_2 = Transplanting by IRRI designed transplanter

T_3 = Hand transplanting in rows

Design of Experiment

For the field test of the transplanters, a randomized complete block (RCB) design was used, and the treatments were replicated thrice in each block. The experimental layout is presented in Fig. 6.

Experimental Procedure

Prior to the field test, both transplanters were tested in the laboratory to confirm the workability of all the functional components. The seedlings were raised in a modified dapog bed near the main field and at the age of 14 to 16 days, they were cut into slices of 19 cm × 40 cm to feed in the transplanter tray. The main field was prepared with ploughing and laddering, and left for 24 hours for settling. In the following day the transplanting operations were done by hand and machine as per treatments and layout plan. In the machine transplanted fields, 2 - 3 cm standing water was ensured as it was necessary for easy trafficability and smooth release of the seedling into the ground. The effective field capacities and field efficiencies were calculated from the collected data. Moreover, the data of planting distance, planting depth, standing angle, number of seedlings per hill, missing hills, buried and damaged hills were recorded in order to work out the planting accuracy. The crops were managed with irrigation, drainage, weeding, fertilizer and in-

Table 2. Working Accuracy Parameters of Different Transplanting Systems

Parameter	BRRI-modified transplanter	IRRI-designed transplanter	Hand transplanting
Planting distance, hill to hill (cm)	15.16 a	16.16 a	15.10 a
Planting depth (cm)	3.70 a	5.00 b	4.16 ab
Standing angle (°)	66.26 b	65.83 b	72.00 a
Seedling/hill (Nos)	8.16 b	11.10 a	4.16 c
Missing hill at planting in 50m^2 (%)	3.66 a	3.81 a	0.00 b
Buried hill in 50m^2 (%)	0.55 a	0.61 a	0.00 b
Damaged hill due to picker injury in 50m^2 (%)	0.35	0.43	0.00
Missing hill after fifteen days in 50m^2 (%)	4.76 a	4.60 a	0.25 b

In a row, means followed by a common letter are not significantly different at the 5% level by DMRT

Table 3. Crop-Cut Data of the Fields of Different Transplanting Methods

Parameter	BRRI-modified transplanter	IRRI-designed transplanter	Hand transplanting
Tiller/m^2	575 a	570 a	538 a
Panicle/m^2	385.7 a	380.3 a	390.3 a
Filled grain/panicle	65.95	65.48	63.15
Percent filled grain (%)	72.48 a	69.57 a	66.23 a
Grain yield (ton/ha)	3.09 a	2.95 a	3.00 a

In a row, means followed by a common letter are not significantly different at the 5% level by DMRT

Table 4. Cost of Seeding Production and Transplanting by Different Methods

Method of seedling production	Cost of Seedling production (US$/ha)	Cost of frame (US$/ha)	Cost of transplanter operation (US$/ha)	Total cost (US$/ha)
Wooden frame nursery	27.21	18.92	14.83	60.96
Bamboo frame nursery	27.77	12.38	14.83	54.98
Plastic frame nursery	27.20	16.36	14.83	58.39
Plastic tray nursery	44.89	50.04	14.83	109.76
Nylon rope nursery	27.04	0.87	19.65	47.56
Wet bed nursery*	19.32	0.00	73.86	93.18

*- Hand transplanting was done with wetbed nursery seedling.

secticide application. Finally they were harvested, threshed, cleaned and dried and the yields were expressed in ton per hectare at 14% moisture content. The data recorded from the experiment were subjected to analysis of variance and DMRT by using IRRISTAT package.

Results and Discussion

Field Performance of Transplanters

The effective field capacities of the BRRI modified transplanter, IRRI-designed transplanter and hand transplanting method were 0.0191 ha/hr, 0.0155 ha/hr and 0.0023 ha/hr, respectively, (**Table 1**). The results reveal that about 20% working capacity was increased in the BRRI modified transplanter compared to the IRRI-designed transplanter although the difference was not statistically significant. The transplanting capacities of BRRI or IRRI transplanter were almost 7 to 8 times greater than that of the hand transplanting method, and the results were statistically significant at 5% level. Islam et al. (1993) tested the Yanji 2ZTR-424 transplanter model with modified dapog seedling in Bangladesh, and found a field capacity of only 0.02 ha/hr. The field efficiencies of BRRI and IRRI transplanters were 78.90 and 76.83 percents, respectively, which were statistically not different, but that of hand transplanting method was 91.40 percent and significantly higher at 5% level than those of machine transplanting. In the case of the hand transplanting, the time was lost only in carrying seedlings from the headlands to the operators' hand, and no time was lost in turning. On the other hand, in the machine transplanting, time was lost in feeding seedlings, turning at headlands and minor repairs and adjustments.

Significant differences in planting depth, standing angle of the planted seedlings and number of seedlings per hill were observed among the three planting methods. The highest planting depth (5 cm) was observed in the IRRI transplanter, however, the lowest depth (3.7 cm) was observed in the BRRI transplanter. For seedling establishment, 3 to 4 cm planting depth was good enough. Considering the planting depth, BRRI modified transplanter was good as it could handle smaller seedlings. The standing angles of the planted seedlings between BRRI and IRRI transplanters were mostly identical (65 to 67 degree), but in the hand transplanting the standing angle was significantly higher (72 degree) than those found in machine transplanting (**Table 3**).

The average seedling per hill in all three transplanting methods were found to be 8.16, 11.10 and 4.16, respectively. The number of seedling per hill in the machine transplanting was significantly higher than that of hand transplanting. The optimum number of seedling per hill ranged from 4 to 5. Therefore, a considerable wastage of seedling in machine transplanting compared to hand transplanting was observed. Mainly two factors were responsible for excessive seedlings per hill in machine transplanting: (a) the growing density of the seedlings; and (b) the degree of tension in the chain of the free wheel of the transplanter which was responsible for the sideways movement of the seedling trays. The average seedling density in the mat was 5.78 no/cm^2, but according to the recommendation of the Yanji transplanter company it was necessary to maintain a density of 3.8 to 4 seedlings/cm^2 in the mat to obtain a required plant population per hill in the field. For optimum performance of the transplanter approximately 1 cm sideways movement of the tray assembly per stroke was necessary. The higher sideways movement of the tray per stroke increased the number of seedlings per hill. The missing hills were 3.66% and 3.81% in BRRI and IRRI transplanters, respectivel,y which were mostly identical. But in the case of hand transplanting, there was no missing hills recorded, implying that the hand is a better device for rice transplanting, though it is very laborious, tedious and arduous.

The rice yield from the fields planted by all three transplanters were 3.09, 2.95 and 3.00 ton/ha, respectively and not significantly different at 5% level (**Table 3**). Irrespective of the planting method the rice yield per unit area was low, due to the increased percentage of unfilled grains. The other yield indicators like tiller/sq.m and panicle per sq.m were in the optimum range. The panicle per sq.m in the fields planted by the BRRI transplanter, IRRI transplanter and hand transplanting method were 385.7, 380.3 and 390.3 which were also in the optimum range and were not statistically different at 5% level.

Mechanical Performance of Transplanters

The Skid

The weights of the IRRI transplanter at no load and full load were 24 kg, and 37 kg, respectively. The dragging force at full load exceeded 20 kgf which was difficult to furnish by an average size labour (**Fig. 8**). The weights of the BRRI transplanter at no load and full load were 21 kg and 30 kg, respectively. The dragging force at full load was limited to 13 kgf which was affordable by an average size labour (**Fig. 7**). Compared to the wooden skid of the IRRI transplanter, the metal skid of the BRRI transplanter became durable, light weight and incurred less sliding resistance between the puddled field and the skid.

The Seedling Picker

In the IRRI transplanter, the picker fingers were rounded in configuration. Therefore, the fingers did not get any cutting action with the mat, as a result of which the seedling blocks were deformed rather than being cut with sharp corners. In order to increase the cutting action the BRRI transplanter was provided with semi-circular picker fingers which rendered rectangular seedling block with sharp edge and good soil bearing index.

Economic Consideration

Machine transplanting costs with wooden frame, bamboo frame, plastic frame, plastic tray nylon rope and nursery seedlings were US$ 60.96/ha, US$ 54.98/ha, US$ 58.39/ha, US$ 109.76/ha and US$ 47.56/ha, respectively, (**Table 4**). The highest cost in plastic tray method was basically due to the high initial cost for the trays. Approximately US$ 625.00 was necessary to purchase 1100 trays needed for the production of seedlings for one hectare of land. On the other hand, the lowest cost in nylon rope nursery method was due to the low cost of the nylon ropes compared with other methods. The hand transplanting cost with wetbed nursery seedling was US$ 93.18/ha. Therefore, machine transplanting with any of the nursery seedling methods except plastic tray, was more profitable compared to hand transplanting method.

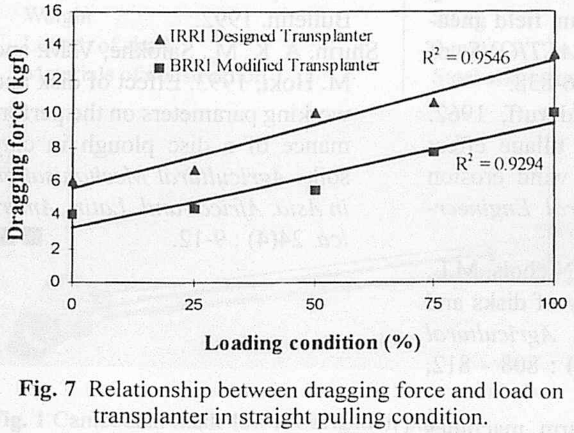

Fig. 7 Relationship between dragging force and load on transplanter in straight pulling condition.

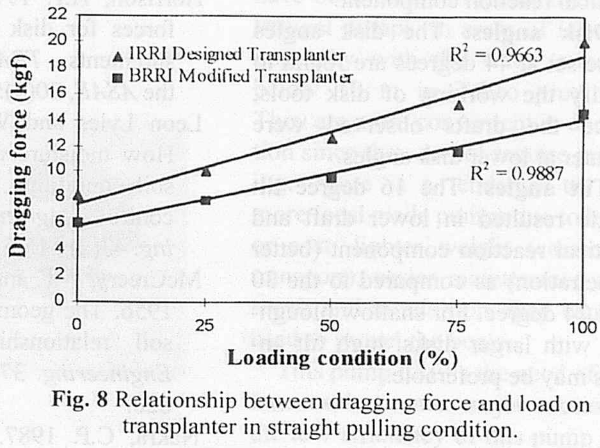

Fig. 8 Relationship between dragging force and load on transplanter in straight pulling condition.

Conclusions

About 30% less dragging force was obtained for the BRRI-modified transplanter compared with the IRRI-designed transplanter, irrespective of the degree of loading, mode of pulling, and field water depth. The semi-circular configuration of the seedling picker of the BRRI modified transplanter increased the cutting action which helped detach rectangular seedling block with minimum energy. About 20% increase in the working capacity of the BRRI modified transplanter was achieved over the IRRI-designed transplanter.

Recommendation

Rice transplanting by the BRRI-modified transplanter with either wooden or plastic frame nursery seedlings may be recommended for the farmers in the rice growing countries as it was found more profitable compared to hand transplanting method.

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(Continued from page 26)

Influence of Operating and Disk Parameters on Performance of Disk Tools

Size of disks: Large size and heavier (56 and 61) disks gave satisfactory results in black clay loam soil at low MC level as they showed low upward vertical reaction component which help in penetration. In the sandy soil, the 51-cm disk was workable since large- sized disks showed high negative values for the vertical reaction component.

Disk angles: The disk angles were set at 44 degrees are found to justify the working of disk tools, since the drafts observed were greater at lower disk angles.

Tilt angles: The 16 degree tilt angle resulted in lower draft and vertical reaction component (better penetration) as compared to the 30 and 24 degree. For shallow ploughing with larger disks, high tilt angles may be preferable.

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Field Testing and Modification of a Low-lift Irrigation Pump Used in Cambodia

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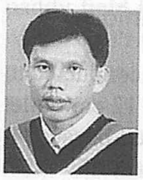

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Abstract

A local made low lift irrigation pump of 12 cm diameter and 3 m long discharge pipe was evaluated for the performance under six different speeds from 1250 to 2500 rpm and three total static heads of 1, 1.5 and 2 m. The maximum efficiency was 49.2, 47.1 and 44.3% at speed of 2250 rpm for all total static heads, respectively. When the speed increased the maximum percentage increase in efficiency was at speed of 2250 rpm for all total static heads.

The mechanical power loss was found as high as 45.01% of power input at speed of 2200 rpm. The high mechanical loss was due to high friction between drive shaft and nylon and wooden support bearing, deflection, misalignment and corrosion of drive shaft. Total power loss accounted for 60% of power input.

A new hollow carbon steel drive shaft was developed to obtain lower mechanical power loss and freedom from torsional vibrations. Compared to the original shaft the test results show that power requirement of newly developed drive shaft is reduced by 27.06 and 47.83% at speed 1250 and 3000 rpm, respectively.

The main specifications of this pump are as follows:

Type	Inclined shaft low lift pump
Power transmission	Rigid direct coupling
Power Source	Combustion engine of 3-12 hp
Diameter of impeller	200 mm
Diameter of discharge pipe	120 mm
Number of vanes	2
Weight	20 kg
Length of shaft	2 m
Materials of construction	Steel and wood

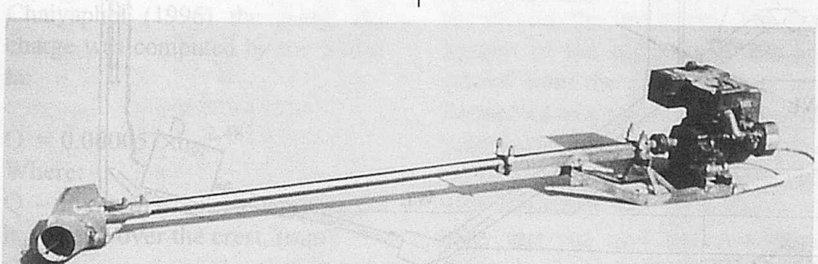

Fig. 1 Cambodian-made low lift irrigation pump.

Introduction

For majority of Cambodian people, rice is the staple food. Limited water availability is the major constraint in Cambodia at present. Utilization of water pumps will hopefully solve the problem of inadequate infrastructure and water requirement for crop irrigation.

The low lift irrigation pump of inclined type (**Fig. 1**) with engine output ranging from 3 to 12 hp have been used to replace the centrifugal pumps, in areas of low topography with the water column below 10 m in Takeo province. They are more convenient in operation since they do not require priming. There is a tendency to use more local made pumps due to lower cost, lighter weight (easier to transport), easier maintenance and repair and in addition, they can be used to propel the boats.

This pump fulfills the need of the farmers, but some major causes of the low efficiency of this pump are as follows:

- The contact area between drive shaft and nylon rope bearing is long and drive shaft is small (**Fig.2**), which leads to high mechanical loss, deflection and vibration.
- Surface of pump casing is not smooth and painting leads to high skin friction loss and corrosion. Strainer is not used to cover the inlet hole, which allows foreign materials to enter the casing. The connection between pump outlet and discharge pipe is not close enough, it is simply inserted and fixed by the rubber and rope. This leads to pressure loss.
- Chord length of impeller, first increases and then decreases with increasing radius whereas in well-designed pumps the chord length increases continuously from hub to tip. The vanes were not well finished and the hub was slightly eccentric. The space between the vane tips and the casing is large which leads to leakage losses around this space.

In recent years, local pump makers have developed their facilities in making some pumps based on foreign design. But most of them have not been tested adequately. There is no design standard published locally. No information on pump testing and materials choice for components have been published. The users almost know nothing about the actual performance curves of the pumps, so they are not able to choose the suitable operating region for the pumps. This explains why most pumps, especially the local made pumps, have very short operating life and low efficiency.

Hence testing and improvement in design are required to improve the efficiency and adequate information about performance characteristics of the pump must be given to local pump makers as well as pump users.

Experimental Technique

The experiment was conducted to test a low lift pump of 12 cm outlet diameter. It was selected because it was an optimum size and most commonly used by the farmers in this area. The 18 cm diameter two-blades impeller was equipped with a driving shaft (diameter 12 mm and 2 m length). The length of dis-

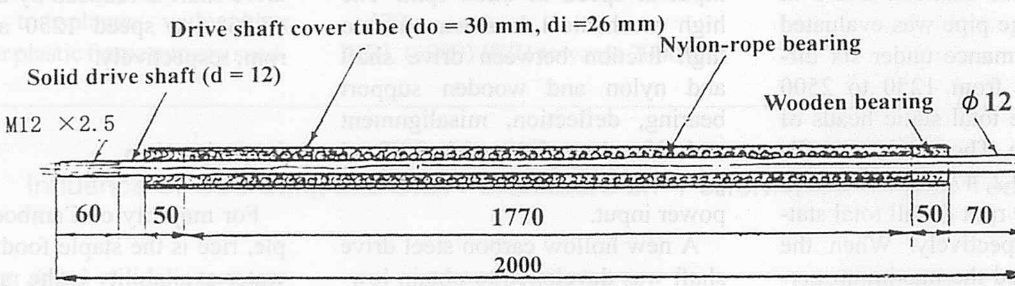

Fig. 2 Schematic diagram of original drive shaft.

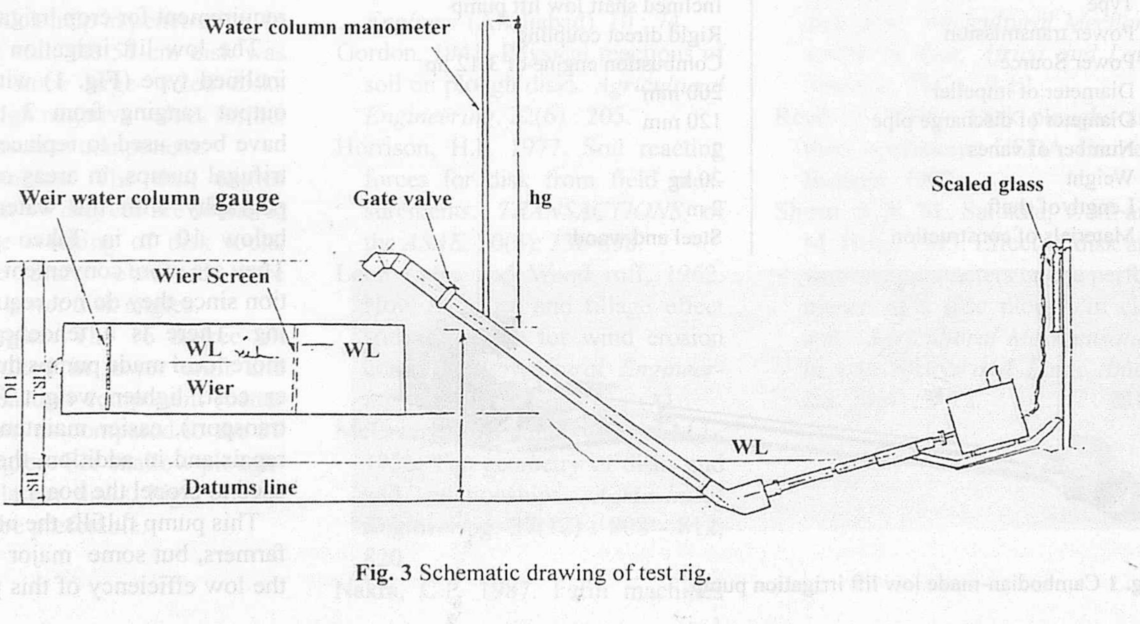

Fig. 3 Schematic drawing of test rig.

charge pipe was 3 m. Three replications were made for each set of condition and mean values of rotational speed, discharge pressure head, height of water over weir crest and fuel consumption rate were measured, recorded and used in the analysis. Testing arrangement was in the closed-loop method as shown in **Fig. 3.** and rig was installed on a pond in low-lying paddy field in Prey Kabas District of Takeo Province, Cambodia. The pump was carefully installed to submerge at sufficient level of 500 mm under water surface in order to prevent efficiency loss due to vortex during the test. Proper care was also taken to provide adequate clearance for the pump intake from all sides of the pond including height between the bottom of pump and the bottom of pond.

The pump speed was measured by using contact tachometer touching an intermediate driven shaft, which was built parallel to the pump drive shaft. They connected each other through no. 29 V-belt through 1:1 pulleys with 80 mm diameter. Six different speeds 1250, 1500, 1750, 2000, 2250 and 2500 rpm were measured and recorded directly from the tachometer indicator.

Pump discharge rate was measured by means of a 90° triangular weir. It was made of 4 mm steel sheet with 2 m long, 1 m wide and 0.8 m high **(Fig. 4)**. The head over the weir crest was measured by means of water column gauge on the upstream side at a distance of 0.45 m from the weir. According to, Chaiyaphol (1996) the pump discharge was computed by the formula:

$$Q = 0.000057 \times h_w^{2.48}$$

Where:

Q - Pump discharge rate, (l/sec)

h_w - head over the crest, (mm)

The pressure head was measured by a piezometric opening and a water column manometer with a diameter of 5 mm and height of 5 m. It was inserted on the side of discharge pipe at distance of 2 m from the inlet to measure static pressure of discharge pipe. During each operating condition, corresponding variation of water in a water column manometer was recorded and the discharge pressure head was computed by the following formula:

$$h_d = (W_g/W_w) \times h_g$$

Where:

h_d - Discharge pressure referred to the center of the discharge opening of the pump, expressed by the meter of liquid column height, (m)

W_g - Specific weight of gauge liquid, (kgf/l)

W_w - Specific weight of the lifted water, (kgf/l)

h_g Vertical indication of water column manometer, (m), (EBARA, 1993)

Total head was computed by the following formula:

$$H = z + h_d + (V_d^2/2g) + h_l$$

Where:

H - Total pump head, (m)

z - Elevation of pressure gauge above datum line, (m)

V_d - Mean velocity of liquid lifted passing through the sectional area at the side of discharge pressure tapping, (m/s)

g - Gravitational acceleration, (m/s^2)

h_l - Total head losses, (m), (EBARA, 1993; Ali, 1992)

A 6-hp gasoline engine was used for driving the test pump. The fuel system of the engine was disconnected from the fuel tank, but was connected to a scaled cylinder glass with 0-250 ml range through a plastic tube. The fuel consumption rate was measured for 10 minutes of each test run and test condition. Pump power input is the product of fuel equivalent power and engine efficiency.

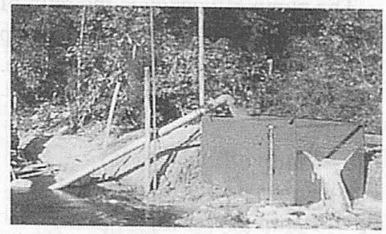

Fig. 4 Measurement of discharge by 90° V-notch weir.

$$P_s = \eta_e \times [(HV \times FC \times W_f) / K_{fe}]$$

Where:

Ps - Power input, (kW)

HV - Heating value of fuel = 47600 kJ/l

FC - Fuel consumption rate, (l/h)

W_f - fuel specific gravity (gasoline) = 0.735

K_{fe} - Unit constant = 3600

η_e - Engine efficiency (η_e = 0.25 for gasoline engine),(Goering, 1992).

The mechanical power losses were measured by rotating the shaft without impeller at the same arrangement as in the pump performance test. Friction losses between the rotating shaft and water were assumed to be the same as in pumping condition. For each set run, the no-load fuel consumption rate was measured and recorded by using the same measuring instruments and time as was done in the pump performance test. The mechanical power loss in the pump was computed as:

$$P_m = \eta_e \times [(HV \times FC_{no\text{-}load} \times W_f)/K_{fe}]$$

Where:

P_m Mechanical power loss, (kW)

$FC_{no\text{-}load}$ - Fuel consumption rate at no-load condition, (l/h)

The pump power output is the rate of energy input to the liquid and was defined by the relation:

$$P_w = [(W_w \times Q \times H)/102]$$

Where:

P_w - Water power output, (kW)

Q - Pump discharge rate, (l/s)

H - Total head, (m)

W_w - Specific weight of water, (kgf/l)

102 - Conversion factor

The pump efficiency is the ratio of water power output to the pump power input, expressed in percentage.

$Eff = (P_w/P_s) \times 100$

Where:

Eff - Pump efficiency,(%)

P_w - Pump power output, (kW)

P_s - Pump power input, (kW)

Modification of drive shaft

A new drive shaft is a hollow carbon steel supported by two anti-friction bearings at both ends (**Fig. 5**). It was developed to obtain low mechanical power loss and freedom from torsional vibrations. It is made of hollow steel shaft in order to reduce weight and increase the stiffness of the shaft. The shaft diameter based upon the bending moment alone is,

$d = (32M/3.14S_t)^{1/3}$

Where:

M - Maximum bending moment

S_t - Allowable tensile or bending stress

$S_t = [(32Md_o)/(3.14(d_o^4 - d_i^4)]$

Where:

d_o - Outside diameter

d_i - Inside diameter

For more stability the first of fundamental frequency should be about 1.3 to 1.5 time away from the operating speed. The general formula that holds for the vibrating beam of different modes of natural frequency for SI units can be expressed as:

$n_{nf} = 670B(IE/wl^4)^{1/2}$

Where:

n_{nf} - Natural frequency, (cycle/min.)

B - A factor value which depends on where beam is fixed

I - The area moment of inertia of the beam cross-section with respect to the bending axis, (m^4)

E - The modulus elasticity of the beam material

For carbon steel shaft

$E = 200 \times 10^9$ N/m^2

w - The mass per unit length = W/gl, (kg/m)

l - The length of the beam, (m), (Chaiyaphol, 1996).

From the calculation the new hollow carbon steel shaft was specified with outside diameter of 25 mm and inside diameter of 12.4 mm. The shaft is coupled to the pump and driven by a 10-hp gasoline engine. It rotated up to 3000 rpm with satisfactory operation. The pump speed and no-load fuel consumption were measured, recorded and used for calculation of the mechanical power loss as done in performance test.

Results and Discussions

Test Results and Effect of Total Static Head and Speed on Pump Performance

The summary of results of performance test in **Table 1** shows that the maximum efficiency were 49.2, 47.1, and 44.3%. for all total static head of 1, 1.5 and 2 m, respectively. At these efficiencies discharge were 23.5, 21.9 and 20.2 l/s, the total head were 2.86, 3.52, and 3.91 m and the power input were 1.34, 1.60, and 1.75 kW, respectively. Further increase in speed resulted in decreased efficiency. Therefore, at 1.5 and 2 m total static head the efficiency becomes zero when the speed is equal to or below 1250 rpm.

When the total static head was increased from 1 m to 2 m the discharge was gradually decreased by 6.8 to 52.7%. In order to maintain constant capacity of this pump at high total static head the pump needs to be operated at a high speed, since there is higher leakage loss when operating at higher total static head. However, high differential pressure or pressure drop through the clearance between the impeller edge and pump casing caused high leakage loss. The total head was increased by 11.6 to 33.1 %. It was clear that a change in total static head had an effect on total head of a test pump, since it changed the elevation of the discharge point. The kinetic energy imparted by the rotation of the impeller was wasted by the eddy and hydraulic loss due to skin friction in pump casing and piping system. Part of the energy was also transformed into the potential energy in order to lift water up to the required total static head. Input power was increased by 8.98 to 28.4%. This is due to increase of static pressure on lifting water in discharge pipe and pumping system which lead to increased pressure on rotating impeller i.e., disk friction loss and leakage loss through the clearance

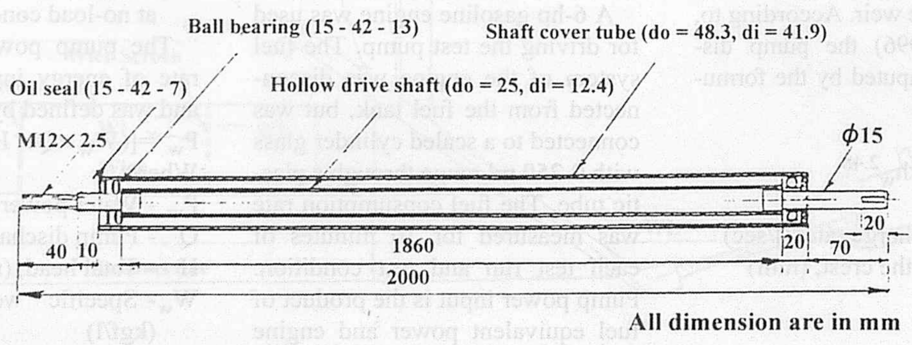

Fig. 5 Schematic diagram of developed dive shaft.

Table1. Summary Results of Performance Test at Different Total Static Heads and Speeds

N (rpm)	Q (l/s)			H (m)			Ps (kW)			Eff (%)		
TSH (m)	1	1.5	2	1	1.5	2	1	1.5	2	1	1.5	2
1250	8.80	0.00	0.00	1.23	1.46	1.84	0.44	0.50	0.51	24.2	0.00	0.00
1500	12.7	9.77	6.00	1.66	1.95	2.39	0.67	0.79	0.80	30.90	23.60	17.50
1750	17.00	15.20	10.80	2.21	2.50	2.90	0.92	1.04	1.09	40.10	35.80	28.10
2000	21.80	18.90	15.90	2.63	3.04	3.39	1.16	1.28	1.41	48.30	44.10	37.60
2250	23.50	21.90	20.20	2.86	3.52	3.91	1.34	1.60	1.75	49.20	47.10	44.30
2300	24.70			2.89			1.45			48.20		
2400		22.40			3.48			1.65			46.3	
2500			19.80			3.77			1.88			39.10

between impeller edge and pump casing. The degree of inclination of the pump drive shaft also had a significant effect on power input. The pump efficiency was decreased by 3.75 to 43.1%, respectively. However, at the lowest speed the efficiency decreases 100%. The decrease of efficiency was due to the decrease in discharge, but input power was highly increased with corresponding increase in total static head. However, the efficiency of transformation of energy within the impeller was mainly dependent upon correctly designed surfaces which gradually change in areas and surface finish.

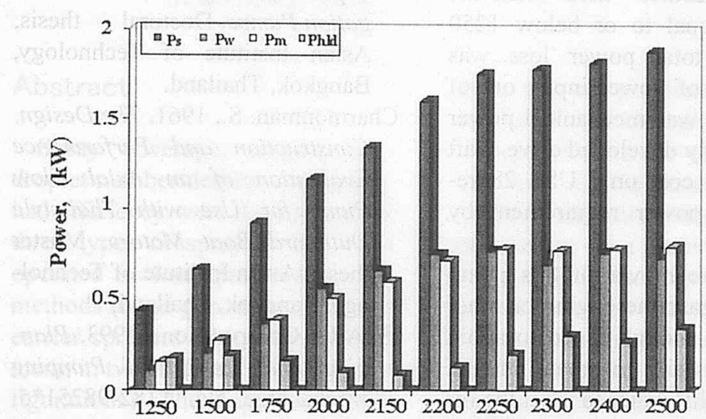

Fig. 6 Power loss of Cambodian-make low lift irrigation pump (Original pump)

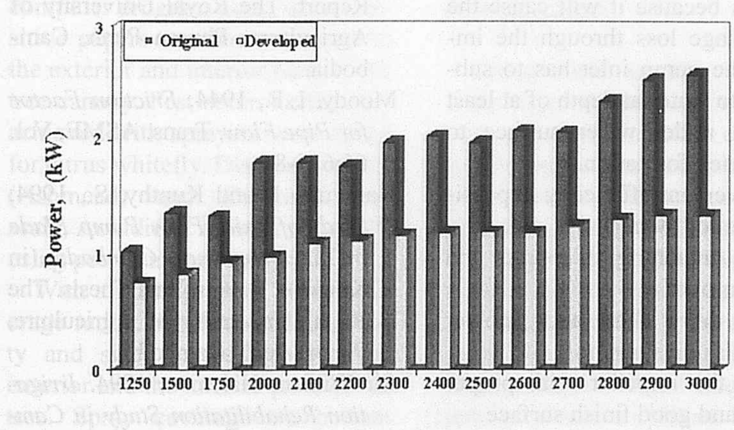

Fig. 7 Comparison of mechanical power loss of developed and original drive shaf.

When the speed increased the maximum increase in percentage of total head and power input was at speed of 2300, 2400, 2500 rpm and at shutoff condition for all total static heads. It was observed that a change in total static head had no significant effect on total head, especially when the pump was operated at the same speed. It had significant effect on this pump due to higher leakage loss. Hence, to maintain a constant discharge, i.e., to reduce leakage loss at high total static head, it was necessary to operate the pump at high speed. Increase in power input was due to increased static pressure in pump and piping system lead to the increase in gyroscopic action of rotating masses, i.e., drive shaft, coupling and impeller. A non-straight 2 m long shaft of 12 mm diameter caused a high friction in the bearing leading to an increase of torsional load. The highest increase in efficiency was at speed of 2250 rpm for all total static heads. It is concluded that the best operation speed for this pump is 2250 rpm.

Power Losses of Original Pump

The experimental results in Fig. 6 show that at speed of 1250 and 2500 rpm the mechanical power loss was 0.15 and 0.8 kW respectively.

The percentage of mechanical power loss was as high as 45.01% of the power input at speed of 2200 rpm. With further increase in speed the mechanical power loss changed randomly. This high mechanical loss was due to a great deal of friction between a long drive shaft of 2 m and two wooden support bearings at both ends and nylon rope at the middle in almost the entire length of the drive shaft. Moreover, the degree of shaft misalignment and inclination of shaft while in operation affected the friction. During the test the inclination of shaft was 15°, 20° and 30° at total static head 1, 1.5 and 2 m respectively , which had a significant effect on the mechanical friction. Even a small eccentricity in shaft can cause a higher mechanical friction in the bearing and result in an increase of torsional load. The total power losses due to hydraulic, disk friction and leakage varied from 6.41 to 39.63% of power input. The total of all power losses accounted for about 60% of the pump power input resulting in low efficiency of this pump.

Modification of Drive Shaft

The experimental result in **Fig. 7** show that the mechanical power loss of new developed drive shaft is 0.76 and 1.35 kW at speed 1250 and 3000 rpm respectively. The mechanical power loss of the original drive shaft is 1.04 and 2.58 kW at the same speeds. Compared to the original shaft, the test results show that power requirement of newly developed drive shaft was reduced by 27.06 and 47.83 % at speed 1250 and 3000 rpm respectively. When the pump was tested, it vibrated at speed lower than 1750 rpm. This vibration was mainly result of fluctuations in engine running speed, non-straight shaft. With further increase in speed, it rotated freely and satisfactorily.

Conclusions and Recommendation

This pump was first tested for pumping water at six different speeds and three different total static heads of 1, 1.5 and 2 m. The tested pump had a discharge pipe of 120 mm diameter and length of 3 m.

The maximum efficiency were 49.2, 47.1, and 44.3%. for total static head of 1, 1.5 and 2 m respectively. At these efficiencies discharge were 23.5, 21.9 and 20.2 l/s, the total head were 2.86, 3.52, and 3.91 m and the power input were 1.34, 1.60, and 1.75 kW, respectively. Further increase in speed resulted in decreased efficiency. Therefore, at 1.5 and 2 m total static head the efficiency becomes zero when the speed is equal to or below 1250 rpm. The total power loss was about 60% of power input, out of which 45% was mechanical power loss. A newly developed drive shaft with added cost only US$ 28 reduced the power requirement by 47.83%.

The prime mover of this pump should be gasoline engine or other motor with enough speed to avoid leakage loss and to create enough pressure to lift water to meet the requirement of total static head. Operation speed should be 2250 rpm to get a high efficiency. The total static head should not be higher than 2 m because it will cause the high leakage loss through the impeller. The pump inlet has to submerge into water at depth of at least 500 mm under water surface to avoid vortex formation.

Improvement efficiency is possible by the following:

- Proper matching of pump and prime mover.
- Improve drive shaft and power transmission system.
- Select the impeller with proper shape and good finish surface.
- Painting inside surface of casing and piping.
- Keeping the delivery pipe as close to the ground as possible.
- Use of efficient auxiliaries such as PVC suction and delivery pipe and improved piping system.

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Spray Coverage and Citrus Pest Control Efficiency with Different Types of Orchard Sprayers

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Abstract

Spray coverage, droplet size, droplet number density and biological efficiency were determined for three types of sprayers that were operated in four-spray application methods (hand gun, air-carrier, air-carrier sprayers with tower attachment; in two different working configurations). The experiment was conducted in a Washington. navel orange orchard. Spray coverage and droplet size of sprays were determined by an image processing computer program using water sensitive papers attached on leaves in the exterior and interior parts of the tree canopy. Pest control efficiency achieved with sprayers were tested for citrus whitefly, Dialeurodes citri (Ashmead) and California red scale, Aonidiella aurantii (Maskell) using petroleum oil sprays.

With all the sprayers, spray coverage ratios, droplet number density and sizes varied between the exterior and the interior parts of the tree. Spray coverage on bottom zone of the tree was higher than the coverage on the top zone of the tree. A mean coverage ratio of 100% was not provided by all the sprayers. Good biological efficiency was achieved for citrus whiteflies with all the tested sprayers. For the California red scale, there appeared to be a better control of the pest in the exterior than in the interior parts of the tree canopy. However, the California red scale is mostly found in the inner parts of the tree, where the air-carrier sprayers achieved lower control than the hand gun applications.

Introduction

Citrus trees usually harbor a wide range of insect and mite pests that are often controlled chemically. The application requirements of pesticides are influenced by the characteristics of the tree, the specific pest problem, and the characteristics of the pesticide and its formulation. Citrus trees are evergreen, densely foliated, and with large canopies. These tree characteristics create a difficult situation for spray application. In addition, target pests may be located outside or inside the canopy, on upper or lower surface of leaves on fruit, or other parts of the tree (Carman,1975). Pests such as California red scale, Aonidiella aurantii (Maskell) (Hom., Diaspididae) are infesting all plant tissues like stem, twigs, leaves and fruits, especially at the interior of the citrus tree. The citrus whitefly, Dialeurodes citri (Ashmead) (Hom., Aleyrodidae) always infest the lower surface of the leaves in both exterior and interior of the tree canopy. For controlling scale insects many growers apply petroleum oil (white oil) since it is very cheap, less phytotoxic, and cause less damage on beneficial insects compared with other insecticides. To achieve a high control efficiency, a thorough coverage of the target pest must be obtained if the petroleum oil is applied (Carman,1974).

In Turkey, like in many other citrus-producing countries, pesticide application in citrus is performed manually using hydraulic sprayers with one or two hand guns, or air-carrier sprayers. Citrus growers have to spray the citrus trees at least two to five times a year depending on pest pressure. In applications by a sprayer with hand gun, generally spray application volumes vary from 20 to 50 l per tree (4 000-10 000 l/ha for 200 trees per ha) depending on the size and volume of trees. The hand gun spray application involves much more utilization of active ingredients and more drift and, in addition, has very low work capacity.

Many researchers have tested different types of air-carrier sprayers to improve spray application efficiency in citrus trees. Generally, spray deposition (Whitney and Salyani, 1991; Bayat et al.,1994), drift (Bayat et al.,1994; Salyani and Cromwell, 1992), air-carrier performance (Salyani and Hoffmann, 1996), and application cost (Munraro et al., 1988; Whitney, 1968) of sprayers were determined. Juste et al., (1990) used different spray application techniques to control citrus aphids and Lepidosaphes sp. in Spain. However, they considered only the deposition on target and not the coverage rate. Sometimes, high coverage on the target is more important than high deposition in application of some pesticides such as petroleum oils. Few studies have been carried out on spray coverage level (Carman, 1975; Furness and Pinczeweski, 1985) of sprayers in citrus spraying.

In order to determine the spray coverage, different methods and droplet collecting surfaces have been used. Furness and Pinczewski (1985) compared the performances of spray application of different types of sprayers in citrus. They used a fluorescence tracer to detect droplet number density and spray coverage obtained on leaf using a microscope and ultra-violet light source. Giles et al., (1989) and Salyani et al., (1987) profited from water sensitive papers (WSP) as a droplet collecting surfaces in their air-carrier applications. The former researchers used color changes on the card by using a tristimulus colorimeter. The latter researchers used the size and numbers of droplet and the coverage on WSP using a video image analysis system. Franz (1993) developed a software to convert image data to spot counts, sizes, and coverage information on the WSP and Kromekote cards. A hand-held optical scanner (scanning a resolution at 400 pixel) was used for the images on the cards.

The object of this research was to determine spray coverage, droplet number density and sizes on the citrus leaves, and to determine the efficiency in controlling the citrus whitefly and California red scale achieved with different types of orchard sprayers in citrus.

Materials and Methods

Citrus Trees

The spraying experiments were conducted in a 16-year old Washington navel orchard at the Agriculture Faculty, Çukurova University. Tree density was 231 trees/ha with a spacing of 7 × 6 m, all rows were approximately closed, and the trees had a height of 4.5 to 5.0 m,

Sprayers and Operatio Conditions

Four spraying methods were used with three sprayers. The sprayers were given a reference of S1 to S3.

S1 was a trailed type hydraulic sprayer (Baþman Co., model 1000 l, Adana-Turkey) with two hand guns with nozzles of a 2.5 mm orifice diameter operated manually, and had a 1000 l tank capacity with a piston pump.

S2 was a trailed type air-carrier sprayer (Tecnoma Co.,Fludair model 800, Epernay-France) with 10 cone type nozzles (Tecnoma type,disc/core;23/H5) which have a 2.3 mm orifice diameter, provides 16 500 m3/h air capacity at 40 m/s at a p.t.o. speed of 540 rev/min with a axial fan, and also had a 800 l tank with a piston pump (**Fig. 1**). Air discharge width and length in each spraying site were 8 × 70 cm,

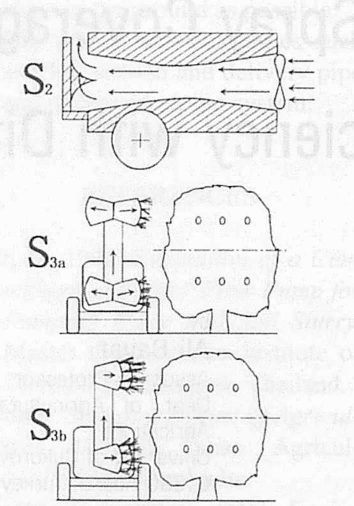

Fig. 1 Schematic view of sprayer S_2, and configurations of sprayer S_3 (S_{3a}, S_{3b}).

respectively.

S3 was a trailed type air-carrier sprayer (Ayaz Co., model 1000 l, Adana-Turkey) with tower attachment and four individually adjustable spraying heads (**Fig. 1**), two of them for spraying the bottom zone of tree and the other two for the top zone of tree. S3 was operated in two spraying configurations (S3a, S3b). In S3a configuration, the spraying heads were directed in both right and left spraying position at the same time. In S3b, all spraying heads were turned to one-side (the right hand side). S3 had 16 cone nozzles, having an adjustable height of turbulent chamber like hand gun nozzle, with a 1.5 mm orifice diameter, a 1000 l tank with a piston pump, provides 5 700 m3/h air capacity at 70 m/s in lower and 3 225 m3/h air capacity at 40 m/s in the upper heads at a p.t.o. speed of 540 rev/min with a radial fan of a 75 cm diameter. Air discharge width and length per head were 7 × 32 cm, respectively. Total air capacity for four heads was 17 850 m3/h

Air-carrier sprayers were trailed at the ground velocity of 3.2 km/h. The maximum application volume was aimed for air carrier-sprayers by using a high pressure and low

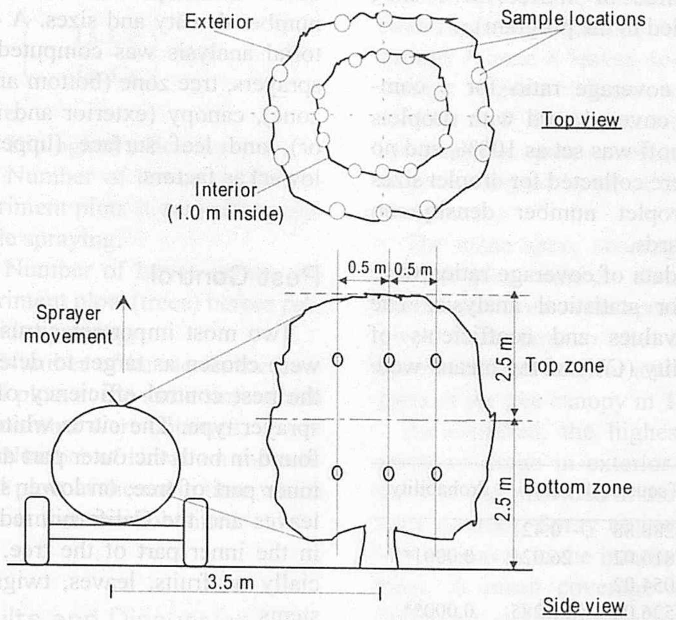

Fig. 2 Schematic view of tree canopy, sampling locations, tree zones and sprayer position.

ground velocity and existent nozzle sizes in possibility. The other operation conditions are given in Table 1. During the orchard operations, wind velocity was 0.5 to 1.0 m/s, air temperature was 28 to 300C, and relative humidity was 65 to 70 %.

Spray Coverage, Droplet Number Density, Droplet Size

Spray coverage, droplet number density and droplet size (VMD, NMD) on leaves were determined using water sensitive paper (WSP, Spraying Systems Co., Wheaton, IL) placed at different zones and locations of the trees. The same trees were sprayed one week later again, including the same treatments with petroleum oil in the recommended dosage in September 1995.

For all tests, a randomized bloc design with four replications was used. Each sprayed tree was assumed as one replicate. Four trees were sprayed for each treatment. Before spraying, each tree was divided into two zones (bottom and top) in all directions (east, west, south, north) and in both interior and exterior of canopy (**Fig. 2**). WSP (76 × 26 mm) were stapled onto the upper on lower surfaces of selected leaves at each sampling location. In each zone, WSP were attached to 12 leaves in the exterior, and 12 leaves in interior parts of tree canopy. Thus, a total of 48 locations were used on each tree. After attaching the WSP, only water was sprayed with each sprayer depending on trial plan. Afterwards, WSP were collected separately and placed in keeping boxes.

Spray coverage and droplet sizes were determined by an image processing computer program (ÖLÇÜM 1 for Windows) developed at the Ege University (Izmir, Turkey), Computer Science Department. Before using this program, a scanner (Scanjet 3P, Hewlett-Packard Co.) set to a resolution of 600 pixels per 25 mm (42.5 μm/pixel). Each WSP was scanned separately to create a TIFF file. In program, The TIFF file may be manipulated with different filters and threshold values (0...255) depending on user experience for better images. Overlapped droplets were eliminated for measuring the droplet size, but included for measuring covered area with droplets. After this operation on the computer monitor, stains of droplets were converted to green color. Following the processing an image, droplet number density and sizes were given by the image program. Droplet number density determined with program was not used because some stains were eliminated due to overlapped and noncircular droplets. In this study the real droplet number density (droplet no./cm^2) was determined using a microscope by counting droplets for 1 cm^2 sections of each WSP. Droplet number density was not determined on WSP with a much larger numbers

Table 2. Working Conditions for the Tested Sprayers

Sprayer	Pressure (bar)	Droplet sizes# (μm) VMD	NMD	CH$^{\xi}$	Application volume (l/ha)	Volume (l/tree)	Oil* dose (cc/tree)
S_1	40	380	152	2.50	4760	20.6	309.0
S_2	20	235	118	1.99	1380	6.0	90.0
S_{3a}	45	283	110	2.57	2016	8.7	130.5
S_{3b}	45	307	132	2.32	4032	17.5	262.5

* Porkan (Em)70, Sandoz, Concentration; 1.5 l/100 l water.
# VMD: Volume median diameter, NMD: Number median diameter; Data obtained with the laboratory facilities on WSP with a spatial droplet sampling attachment for one nozzle without air discharged.
ξ VMD/NMD; Coefficient of droplet homogeneity.

than 300 droplets/cm^2. When this program was tested for a known coverage and droplet sizes, the errors for droplet size and coverage were ± 5-8 %. This error rate was not considered in this work. The following equations were used to calculate the spray coverage ratio;

$$Sc = Ca/A * 100 \quad (1)$$

Where;

Sc = Spray coverage ratio (%),

Ca = Area covered with droplet stains (cm^2), and

A = Area of WSP, (19.76 cm^2, loaded in the program).

The coverage ratio for a completely covered card with droplets like runoff was set as 100%, and no data were collected for droplet sizes and droplet number density on these cards.

The data of coverage ratios were used for statistical analysis. The mean values and coefficients of variability (CV) of the means were used in comparison for droplet number density and sizes. A 4-factorial analysis was computed with sprayers, tree zone (bottom and top zone), canopy (exterior and interior), and leaf surface (upper and lower) as factors.

Pest Control

Two most important citrus pests were chosen as target to determine the pest control efficiency of each sprayer type. The citrus whitefly is found in both the outer part and the inner part of tree, on lower side of leaves and the California red scale in the inner part of the tree, especially on fruits, leaves, twigs, and stems.

Before application, alive whitefly larvae and California red scales per leaf and fruit were determined in the laboratory. The same counts also were conducted on the control trees. Afterwards, Porcan mineral oil with a concentration of 1500 cc per 100 l water was sprayed on trees with the different sprayers.

Four days after spraying, leaves and fruits were collected on the treated trees according to the sampling scheme in Chapter spray coverage, droplet number density and droplet size. Then, the biological efficiency was calculated using the

Table 2. Analysis of Variance for Coverage Data

Sources of variation	Degrees of freedom	Sum of squares	F value	Probability
Replications	3	288.88	0.421	
Sprayers	3	17816.02	26.021	0.0001**
Error 1	9	2054.02		
Tree canopies (exter. inter.)	1	15526.08	99.285	0.000**
Sprayer × canopies	3	2326.80	4.959	0.0182*
Error 2	12	1876.54		
Tree zones (top bottom)	1	2943.65	46.700	0.0000**
Sprayer × zones	3	2366.03	12.512	0.0000**
Canopy × zones	1	753.44	11.953	0.0020**
Sprayer × canopy x zones	3	863.57	4.566	0.0114*
Error 3	24	1512.78		
Leaf surfaces	1	9173.18	148.087	0.0000**
Sprayer × leaf surface	3	942.59	5.0722	0.0039**
Canopy × leaf surface	1	94.58	1.526	0.2226
Sprayer × canopy × leaf surface	3	461.129	2.481	0.0722
Zone × leaf surface	1	12.171	0.196	0.0000**
Sprayer × zone × leaf surface	3	542.309	2.918	0.0435*
Canopy × zone × leaf surface	1	39.57	0.638	0.0000**
Sprayer × canopy × zone × leaf sur.	3	525.94	2.830	0.0482*
Error 4	48	2973.33		

*, **Significant at P< 0.05, 0.01, respectively.

Table 3. Mean Spray Coverage Ratios, Droplet Number Density on Water Sensitive Papers and Coefficients of Variability (Cv)* in Chosen Zones and Parts of the Canopy

		Spray coverage (%)				Droplet number density (no./cm^2)			
Sprayer	Leaf surface	Bottom zone		Top zone		Bottom zone		Top zone	
		Exterior	Interior	Exterior	Interior	Exterior	Interior	Exterior	Interior
	upper	84.38	70.63	73.00	65.31	>300	>300	>300	>300
S_1	lower	69.93	.47.81	74.00	56.25	>300	202	>300	282
	CV(u/l),%	11/9	28/27	6/7	18/26	-	-/18	-	-/31
	upper	71.25	67.25	52.81	40.63	>300	>300	271	214
S_2	lower	57.19	60.32	50.94	16.63	278	289	235	103
	CV(u/l),%	5/22	4/12	18/30	20/75	-/33	-/28	33/28	25/62
	upper	60.63	36.67	66.94	20.00	>300	185	>300	128
S_{3a}	lower	36.88	14.42	35.39	6.17	189	98	172	52
	CV(u/l),%	12/14	6/8	9/44	8/39	-/27	12/22	-/54	19/65
	upper	80.81	56.25	78.44	49.06	>300	295	>300	208
S_{3b}	lower	65.31	35.63	56.88	19.38	>300	179	290	158
	CV(u/l),%	5/10	9/14	7/13	40/51	-/29	35/48	-/39	27/42

* CV=(standard deviation of observations in the mean /mean) × 100.
(u/l) CV for upper / CV for lower surface.

Henderson-Tilton equation:

$$T = \left(1 - \frac{Ts.Ce}{Te.Cs}\right) \cdot 100 \qquad (2)$$

Where;

T = Biological efficiency (%),

T_s = Number of larvae on the experiment plots (trees) after pesticide spraying,

T_e = Number of larvae on the experiment plots (trees) before pesticide spraying,

C_s = Number of larvae on the control plots at the same time which T_s number were obtained,

C_e = Number of larvae on the control plots at the same time which T_e number were obtained.

Results and Discussion

Spray Coverage, Droplet Number Density and Droplet Sizes

Variance analysis of the data of spray coverage ratios at different locations obtained with sprayers is given in **Table 2**.

As shown in the variance analysis, sprayers, tree canopies, tree zones and leaf surfaces all have had a significant effect on coverage at $P < 0.01$. In addition, interactions between sprayer × canopy × zones, canopy × zone × leaves, sprayer × canopy × zone × leaves were significant at $P < 0.05$. According to these results, spray coverage on the target is affected by all of the variation sources.

The mean spray coverage ratios and droplet number density on leaves achieved with sprayers are given according to the different zone in height, exterior and interior parts of the tree canopy in **Table 3**.

As expected, the highest mean spray coverage in exterior part of the tree was provided at the bottom zone of trees. Spray coverage was low on leaves in the interior part of trees. A mean coverage ratio of 100% was not achieved for any of the sprayer. However, Carman (1977) illustrated that for petroleum oil sprays, a 100% spray coverage must be provided on targets for effective control of California red scales. The CV in coverage ratios for air-carrier sprayers were lower on upper surface than on the lower surface of leaves. The lowest CV for coverage was achieved with sprayer S_2 on lower surface of leaves in the bottom zone of the tree. Droplet number density was higher than 300 droplets/cm^2 at many locations for the hand gun sprayer. Droplet number density greatly varied for air- carrier applications. The CV for the droplet number density was higher than the values of coverage's CV. The source of this variability was probably due to the droplet count the only on 1 cm^2 of WSP while for the coverage the whole paper was considered.

Droplet sizes at the upper and lower surface of leaves in various locations are given in **Table 4**. The mean droplet number density due to elimination of some droplets which overlapped or noncircular stains on the WSP was 57 for sprayer S_1, 72 for sprayer S_2, 45 and 68 droplets/cm^2 for S_{3a} and S_{3b} configuration of sprayer S_3, respectively.

The volume median diameters were higher for exterior than for interior parts of trees. All sprayers achieved high values of VMD on upper surface of the leaves. But the VMD was lower in the inside parts of the trees. The reason for this sit-

Table 4. Mean Volume Median and Number Median Diameters (µm) on Water Sensitive Papers in Different Zones and Locations

Sprayer	Leaf surface	Bottom zone Exterior VMD	Bottom zone Exterior NMD	Bottom zone Interior VMD	Bottom zone Interior NMD	Top zone Exterior VMD	Top zone Exterior NMD	Top zone Interior VMD	Top zone Interior NMD
S_1	upper	342	154	290	135	302	145	245	141
	lower	328	147	280	137	285	138	248	144
S_2	upper	215	113	185	110	189	105	174	121
	lower	205	118	182	108	185	117	177	113
S_{3a}	upper	209	121	205	107	180	107	192	122
	lower	204	115	189	111	185	111	189	133
S_{3b}	upper	298	114	240	118	215	108	208	110
	lower	259	111	235	121	190	113	204	108

Table 5. Mean Spray Coverage, Droplet Densities (No./cm^2) and Volume Median Diameter (µm) in All Sampling Locations

Sprayer	Parts of tree canopy: Exterior	Parts of tree canopy: Interior	Parts of tree canopy: Ratio (Ex./In.)	Average coverage ratios*	Average** VMD	CV$^{\Phi}$ (%) for VMD
S_1	75.34	60.00	1.25	67.70a	290	11.80
S_2	58.05	46.20	1.25	55.20b	189	7.43
S_{3a}	49.96	19.31	2.58	34.60c	194	5.42
S_{3b}	70.36	40.08	1.75	52.10b	231	15.13

Φ CV=(standard deviation of observations for all the target surfaces in the mean /mean) × 100.

* Means followed with same letter in column are not significantly different using Duncan's multiple Range at the 0.05.

** Data obtained from Table 4.

uation may be droplet vaporization due to the long distance between sprayer and target, and settlement of small droplets on target due to the relatively low air-carrier velocity with respect to air velocity at the outside part of tree canopy. Sprayer S_1 and S_{3b} provided bigger droplet sizes on target than the sprayer S_2 and S_{3a} (**Table 4**).

Table 5 shows the mean coverage ratios, volume median diameters and coefficient of variability for VMD using data obtained from **Table 3** and **4**.

The maximum spray coverage was achieved with sprayer S_1 in both parts of tree canopy. However, this was not statistically different for sprayer S_2 and S_{3b} that provided the same coverage on leaves. The lowest coverage was achieved with S_{3a} Spray coverage of S_2 in the interior of tree was high compared to sprayer S_3. Considering the results of Randall (1985), one of the basic reasons of this situation is the relatively high ratio of the air volume/velocity of sprayers S_2. The air volume/velocity of sprayer S_2 and S_3 were 0.11 and 0.09, respectively, whereas the average air velocity at upper and lower heads for S_3 was calculated at 55 m/s. The highest average VMD was achieved with sprayers S_1 and S_{3b}. Sprayer S_2 accomplished the same coverage ratio (exterior/interior) with sprayer S_1. Sprayers S_1 and S_{3b} provided higher CV for VMD than sprayers S_2 and S_{3a} on WSP. The high volume application of sprayer S_1 and S_{3b} could have this variation (**Table 5**).

Pest Control Efficiency

As shown in **Table 6**, all the equipment achieved a considerable high biological efficiency ratios on the exterior parts of the tree. However, variation coefficients were high for all treatments, especially for the interior parts of the trees. This indicates an uneven distribution and coverage of the petroleum oil sprays in the interior canopy. Hence, the biological efficiency of the different treatments matches well with results obtained with WSP (**Table 3**). Because of the high variation in the insect counts no further statistical analysis was made.

S_{3b} application of air-carrier sprayer with tower achieved a better control ratios of whitefly in both parts of the tree compared to sprayer S_2 and S_{3a}. Since, only two or three larvae per leave were found after spraying, a number under economic injury threshold of whitefly, all equipment could be used to control whitefly. Here, we recommend to use sprayer S_2 because the application volume and dose per tree was lower compared to the sprayer S_1 and S_3.

Biological efficiency ratios of California red scale were high on the exterior part of tree canopy in both leaves and fruits, but low in the interior parts of tree canopy, especially when spraying with air-carrier sprayers. It has remembered that population of this pest is mainly higher in the interior parts of the tree canopy because the multiplying of this pest begins in interior parts of the tree. The control efficiency on fruits was somewhat lower for S_2 compared with the S_1 and sprayer S_3. Considering alive California red scale per fruit (3 to 6) in the interior part of the trees after spraying, air carrier sprayers were not effective in controlling this pest using petroleum oil sprays. Consequentially, air carrier sprayers used in this study do not present themselves as an alternative to hand gun sprayers in controlling California red scale. For more efficiency, as a first step, spray application volume and air capacity of air- carrier sprayers should be increased, because the California red scale is a stationary pest and thus requires a high coverage for effective control.

Conclusions

(1)All sprayers achieved high coverage ratios on leaves in the exterior parts of the tree canopy. Spray coverage on upper surface of leaves was higher than that of lower surfaces in all applications. The ratio of exterior/ interior of coverage ratios of hand gun and standard type air-carrier sprayer was the same.

(2)Spray coverage ratios of hand gun sprayer on citrus leaves were

Table 6. The Biological Control Efficiency of Target Pests with Sprayers (Unit:%)

Sprayer	Whitefly control leaves Exter.	Inter.	Average*	California red scale control leaves Exter.	Inter.	Average*	fruits Exter.	Inter.	Average*
S_1	90.3	69.9	80.1	96.7	85.2	90.9	87.5	75.5	81.5
	(35)$^{\xi}$	(42)		(26)	(18)		(22)	(45)	
S_2	70.8	47.7	59.2	91.4	48.6	70.0	60.7	35.8	48.2
	(28)	(35)		(18)	(47)		(23)	(52)	
S_{3a}	68.2	41.6	54.9	72.9	38.4	55.6	72.2	37.6	54.9
	(48)	(55)		(45)	(72)		(31)	(67)	
S_{3b}	82.4	55.1	68.7	84.0	50.3	67.1	91.3	45.4	68.3
	(36)	(44)		(23)	(58)		(18)	(71)	

ξ CV = (standard deviation of observations in the mean /mean) × 100.

higher than that of air-carrier sprayers although one of them have had similar application volume per hectare. Although the standard type air-carrier sprayer had only about 1/3 of the application volume of S_{3b} configuration of air-carrier sprayer with tower, the same coverage ratio was provided on leaves.

(3)Droplet number density was higher than 300 droplets/cm^2 for hand gun sprayer at many target locations. Air-carrier sprayers provided a somewhat lower droplet number density on leaves at many locations compared to the hand gun sprayer. The VMD achieved with all sprayers were higher in the exterior leaf surfaces of tree canopy than in the interior parts. The average VMD on WSP for all the used sprayers ranged between 189 and 290 µm.

(4)Air-carrier sprayers can be an alternative to hand gun sprayer in controlling whitefly using petroleum oil sprays, but they have low efficiency for controlling California red scale, particularly on fruits at the interior part of tree.

(5)The efficiency of standard type air-carrier sprayer may be improved by increasing the application volume and air capacity. Unfortunately, sprayers with higher air capacity than the ones used in this study are not used and not available in Turkey

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Performance Evaluation of a Locally Developed Grain Thresher - II

by

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Abstract

A grain thresher for maize and cowpea designed and constructed at the Institute of Agricultural Research and Training (IAR&T), Ibadan was evaluated both in the workshop and on the field for its performance. Tests were carried out to determine the effects of speed on the shelling output, cracking and grain losses at three moisture content levels. Test results showed that the rate of shelling increased as the speed was increased. Grain losses decreased as the speed was increased while more breakages were recorded at high speed. It was further observed that the rate of shelling, grain losses and breakages decreased as the moisture content increased.

Introduction

The importance of maize cannot be overemphasized as it constitutes the main ingredient for most livestock feeds in Nigeria. There are many problems confronting Nigerian farmers in their production practices, among which is shelling. Most farmers in the country still operate at subsistence level. The maize produced are shelled by hand or by beating sacks stuffed with kernels-on-cobs. These threshing processes are time wasting and energy consuming.

Most of the few shelling machines available in the country today are imported ones which the local farmers cannot afford to buy due to their high cost and complexity in design and operation. The available shellers are, therefore, found mostly in the research institutes, universities and commercial farms.

A simple, inexpensive small hand-operated maize sheller was designed and fabricated at the International Institute for Tropical Agriculture (IITA) in 1981 to replace the tedious hand shelling. Alonge (1986) reported the various types of shellers available such as small hand-operated, pedal-operated, power maize shellers, tractor- operated, combine cylinder type and picker-shellers. Oni and Ali (1986), in their studies, noted that overall threshing efficiency was inflenced by cylinder speed, crop ear size, the two factor interaction of speed and variety, and , feed rate and ear size. Sanhar and Panwar(1974) studied machine crop variables influencing shelling of maize in India while Sharma and Devnani(1980) carried out threshing studies on soyabean and cowpea, also, in India. Their studies showed that threshing efficiency increased with the increase in cylinder tip speed but decreased with the increase in feed rate and concave clearance.

This paper presents the performance test of an Institute of Agricultural Research and Training engine-operated maize sheller which was designed and contructed to thresh grains. It also examines the effect of speed, moisture content on the threshing efficiency of the sheller.

Description of the Thresher

Figure 1 shows an schematic diagram of the thresher. It consists of shaft supported by two ball bearings mounted on either side of the frame/support for delivery housing. The shaft also carries the cylindrical drum and and the shelling members which are of shelling heads and rectangular rods. The rectangular rods and shelling heads are arranged in three rows and spaced at a distance of 6mm from each other. The rectangular rods are responsible for moving the cobs towards the end of the cylinder casing while the shelling heads do the shelling of the maize kernels on cobs. The sheller has upper and lower cylinder casing. The latter cylinder casing has many holes drilled into it and the former casing covers the cylinder drum. At the side of the former casing, the hopper is located. The hop-

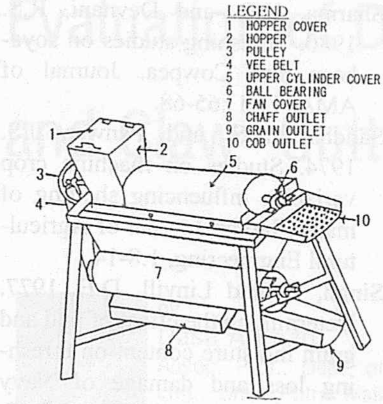

Fig. 1 Schematic diagram of an engine operated grain thresher.

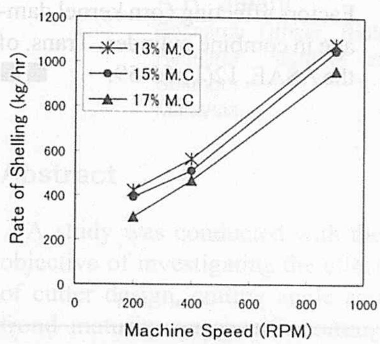

Fig. 2 Effect of machine speed on the rate of shelling of maize.

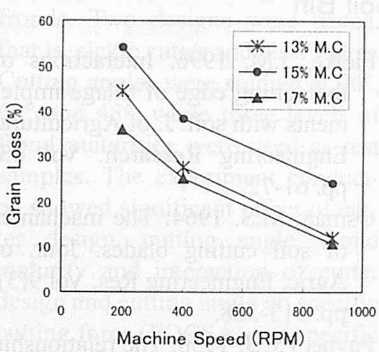

Fig. 3 Effect of machine speed on grain loss for maize.

per has a cover hinged to it at the mouth. The hopper cover helps to prevent grain loss during shelling.

Materials and Method

The sheller was operated by an 5-hp(3.73kW) engine. Three speeds were used during the testing. These speeds were obtained by adjusting the pulley. Three levels of moisture content were obtained,

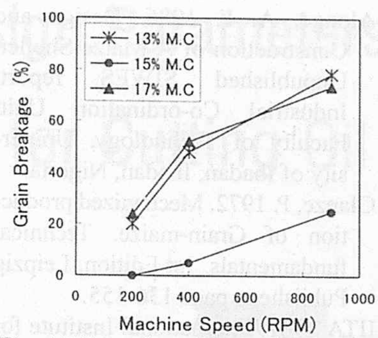

Fig. 4 Effect of machine speed on grain breakage for maize.

i.e., 13%, 15% and 17%. The moisture content of the grain was determined by air oven method and universal moisture meter. For each test, a known weight of maize grains were fed at a time. The shelling time for each test was recorded by a stop watch. The shelled grain was collected at the outlet. The percentage grain loss during the shelling was determined. The process was replicated twice for all the combinations of speed and moisture contents.

Breakage was evaluated by thoroughly examining the kernels for fissures, cracks or breaks in the seed coat. The broken or damaged seed coat were separated from the whole kernels. Grain damage which can be defined as the percentage of all broken grains was evaluated.

Results and Discussion

Table 1 shows that the threshing shelling efficiency of the thresher varied from 71% to 95.6% for the laboratory and field performance evaluation. **Figure 2** shows the effect of speed on the rate of shelling. In all cases, the higher the speed, the higher the rate of shelling at the moisture levels evaluated. The lower the speed, the lower the rate of shelling. The shelling rate, however, decreased with an increase in moisture content of the grain. Onwueme (1979) noted that harvested maize contains 15% to 20% moisture content MC. He also affirmed that the crop could be dried after harvesting to a moisture content of 12% to 13% before storage.

Figure 3 shows the effects of machine speed on grain loss at the three moisture levels investigated. Generally, the lowest percentage grain loss occurred at a moisture content of 13%. However, the the highest loss occurred at moisture content of 15%. This due perhaps to the physical and morphological ear properties of the grain. The varieties of maize grown was observed to exert an influence on the level of losses and degree of husking (Glanze, 1972).

Figure 4 shows that maximum breakages and cracks occurred at the highest moisture level while breakages and cracks were minimum at the intermediate range of moisture content of 15%. The low grain damage at 15% MC could be the change in grain properties. At low MC, the grain becomes hard and brittle. As soon as a force is applied through the speed rotation, it splits into pieces. At high MC, the grains are easily compressed with lower force. **Figure 4** also shows that the higher the speed, the more the damage to the grains. This is because increasing the speed in-

Table 1. Performance Tests of the Grain Thresher

Moisture Content (%)	Workshop		Field
	Shelling efficiency		Shelling efficiency
	I (%)	II (%)	III (%)
13	94.4	93.4	95.6
15	81.0	80.0	84.0
17	72.8	71.0	74.0

creased the frequency of impact between the crop and the shelling members and, consequently, rubbing of the grains was more severe.

Conclusions

Tests for the grain thresher revealed the following:

1. The grain breakage increased with an increase in moisture content of the grains;
2. As the speed of the machine was increased, the grain or kernel damage also increased;
3. Threshing efficiency decreased with an increase in MC. About 95.6% threshing efficiency was obtained at 13% MC level; and
4. The highest percentage grain loss was obtained at 15% MC level.

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(Continued from page 28)

Development and Construction of a Mini-Soil Bin

The dynamometer was calibrated with an electronic selector and a high speed analog/digital converter. An SA64 and an Ad32, 3.0- version data collecting system were used. These systems were installed in a 486 SX, 33 MHz computer, which can process a maximum of 20,000 samples/second.) The dynamometer could be loaded horizontally or vertically because of the way it was attached to the end of the cylinder rod. Its loads were registered in three different channels, one for each the horizontal and vertical loading positions, and another for the momentum. An important factor in calibrating the dynamometer was that there must not be any interference from channel to channel.

After the dynamometer had been attached a specific force was applied to it in order to calibrate the channel which registered the momentum. The value obtained increased according to the length of the lever.

Conclusions

A mini-soil bin was built as a simple and inexpensive alternative, suitable for universities, as well as for education and research institutions, in developing countries.

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Evaluation of Design Parameters of Sickle Cutter and Claw Cutter for Cutting Oil Palm Frond

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Abstract

A study was conducted with the objective of investigating the effect of cutter design, cutting angle and frond maturity on specific cutting force and energy requirement per unit cut area for cutting oil palm fronds. Two designs were tested, that is, sickle cutter and claw cutter. Cutting angles were studied at 90°, 60° and 45°, while three levels of frond maturities were used as test samples. The experiment conducted showed significant effect of cutter design, cutting angle, frond maturity and interaction of cutter design and cutting angle on specific cutting force (FOCSA) and specific cutting energy (ENCSA) requirement for cutting oil palm fronds. The maximum FOCSA for sickle and claw cutter were 12.18kg/cm^2 and 22.9kg/cm^2, respectively, while the maximum ENCSA for sickle and claw cutter were 65.41kg-cm/cm^2 and 115.5kg-cm/cm^2 respectively. This indicated that the sickle cutter required 46% less FOCSA and 43% less ENCSA compared to that of claw cutter. It was found that increasing the cutting angle would result in high FOCSA and ENCSA requirements. The trend was found similar to frond maturity in that the mature the frond, the higher the FOCSA and ENCSA required to accomplish the cutting.

Introduction

Malaysia is currently the world's largest producer of palm oil. Statistics reveal that the total cultivated area of oil palm in the country was 55,000 ha which produced about 92,700 tones of palm oil per annum in 1960. In 1995, the total area planted to oil palm increased to about 2.358 million ha producing about 7.6 million tones of crude palm oil, which was about 64% of the total world's production. It is predicted that by the year 2005, Malaysia may produce 10 million tones of crude palm oil and 2.5 million tones of palm kernel (Dato' Khalid, 1996). However, the Malaysian palm oil is now facing competition not only from other oil and fat industries but also from other palm oil producing countries. Rising competition in the world market, declining in price, and shortage of labour are some of the factors influencing the well being and future of oil palm industry. The high production cost of palm oil is mainly due to the high labour cost and the labour cost in the plantation is about 30-35% of the total production cost (Turner and Gillbank, 1982). The acute labour shortage problem can be minimized through mechanization. At present, most of field operations, viz. land clearing, manuring, spraying and fresh-fruit-bunch (FFB) transporting are already being mostby mechanized except for FFB harvesting. This harvesting operation is still done manually, requires a large number of labour and, therefore, more effort is needed. For comparison, **Table 1** shows the extent of mechanization in oil palm estates based on a survey carried out by Malek (1993).

At present, cutting of fronds and harvesting of FFB for short palms

Table 1. Extent of Mechanization in Oil Palm Estate

Operation	Extent of mechanization (%)
FFB cutting	0
In field transportation	35
Mainline Loading	59
Weding	36
Fertilizer application	39

Note: The above study was based on a survey of 485 estates in Peninsular Malaysia.

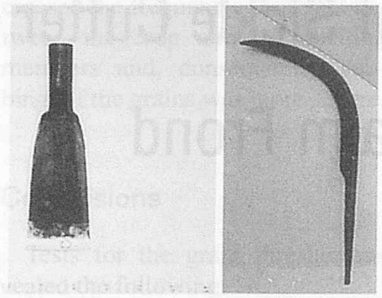

Fig. 1 Sample of chisel(left) and siickle(right) available in the market

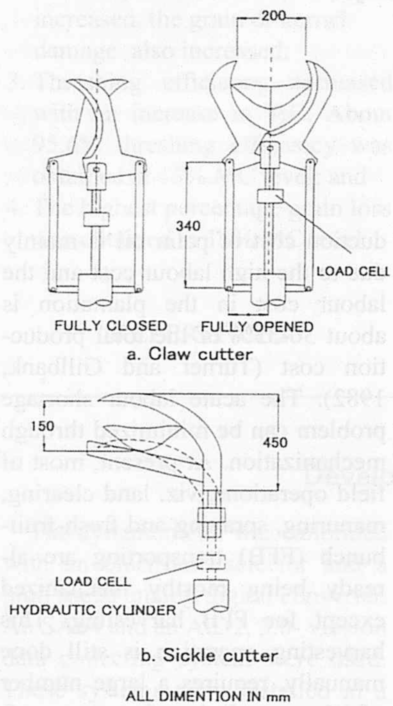

Fig. 2 Cutting test rig for (a) Claw cutter (b)Sickle cutter.

is normally done by using a chisel fixed to a short steel pole, and for tall palms (>2.5 m) a sickle attached to a bamboo or aluminum pole is used (**Fig. 1**). There are several disadvantages in using these manual tools. Obviously, energy for cutting comes mainly from the harvester, and it can only be reduced by the tool sharpness and self skill. Thus, the harvester who is handling such tools should be strong enough to maintain his energy throughout the day. It is observed that generally the harvesters would not be able to maintain their endurance for the whole day and they normally stop in the afternoon (Jelani, 1997). Over the past 10 years, many researchers (Hadi, 1994; Razak et al., 1995; Rahim et al., 1988 and Ahmad, 1990) have been trying to develop machines and tools to improve the field operation's efficiency. Their efforts in developing an effective cutting device showed some convincing results. However, economically, they still could not compete with the existing manual tools. Thus, if a mechanical tool that requires less energy' is available for the cutting operation, the harvesters would be able to work longer hours and, consequently, increase their daily productivity with cost-effectiveness. Therefore, this study was carried out to formulate and develop a technology for cutting fronds and fruit bunches easily and efficiently. The design requirements of the device were set to lower cutting force, fast in cutting action and easy to handle. A test rig was designed and developed to study the effect of the parameters (cutting force and energy) for a sickle with countershear cutter and a claw cutter.

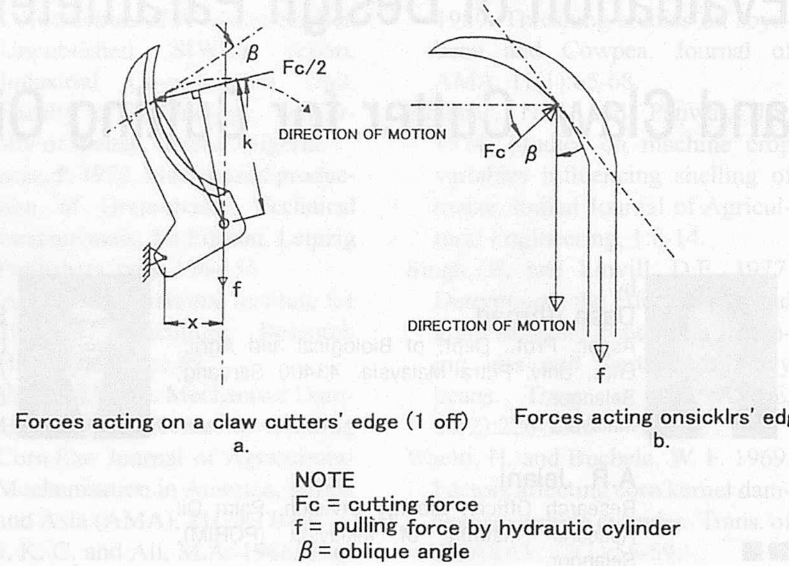

Fig. 3 Forces acting on blades edges: (a)Claw cutter, (b)Sickle cutter.

Testing Methods for Cutters

Claw Cutter (Scissors)

The configurations and dimensions of the blade and its set-up are shown in **Fig. 2a**. The blade was made of high carbon steel and weighs about 0.6 kg with thickness of 3 mm (1 off). Its total length and width were 31.7 cm and 15.5 cm, respectively. The cutting edge was designed to have a curvature of 17 cm radius so that it can grasp and cut the frond effectively. The edge angle (α) was designed to be at 10°, and its oblique angle (β) was kept constant at 24.2° in all positions. Both blades were joint by pivot. The pivot was connected to a hydraulic pusher rod to enable the blade to perform. A load cell was located in the middle of this rod to measure the pulling force required during cutting. Two linkages were connected to the blade. These linkages were arranged vertically to overcome the effect of linkages angle on the force applied. **Fig. 3a** shows the forces acting on the blade (one off). During the experiment, the frond would be located at a distance of about 23 cm from the pivot. This distance was selected because it was the point of contact of the frond edge in actual cutting in the field. The distance was kept constant for all tested samples of fronds.

For a claw cutter, cutting forces provide by two edges of two blades must be considered as to accomplish the cutting. Therefore, the pulling force sensed by the load cell was the resultant force required by the blades to accomplish the cutting. The cutting force requirement is equal to resistance force given by the material. Assuming the friction force at pivot, Z, is much smaller as compared to the pulling force (f), thus the following equation represents the maximum cutting force requirement at the cutting point (by taking moment about the point O, **Fig. 3a**):

$Fc_{max} = 2f(x)/k$ (1)

where:

Fc_{max} = maximum cutting force by two edges, kg
Fc = cutting force, kg
f = force sensed by the load cell, kg
k = perpendicular distance from pivot to the line of Fc (set at 23 cm)
x = horizontal distance from the pivot to the linkages, cm

The maximum cutting force arose when the direction of Fc was on the horizontal line, that is when the value of x was 10 cm. Substituting the values of k and x, and also the force sensed by the load cell (f) into Eq.1, the maximum force for cutting frond was obtained.

Slicing Cut (Sickle)

An ordinary sickle (popularly used by the harvesters) was used for the study. The edge angle was 10° and the thickness was 3 mm. The sickle was made of hardened steel with heat treatment. A countershear to react the cutting force was put at 15 cm from the end tip of the sickle and it was installed at 15° with respect to the horizontal line. The end of the sickle was connected to the hydraulic pusher rod to enable the sickle to perform the cutting operation. A load cell was placed in the middle of the rod to sense the pulling force required during cutting. **Fig. 2b** shows the configuration and dimensions of the blade and also the experimental set-up. During the test, the edge of frond was located at the beginning of the sickle curve to get the effect of slice cutting and this point was kept constant for all tested fronds.

By solving the involved forces vertically **(Fig. 2b)**, the maximum cutting force is given by the following equation:

$f = (Fc)\cos\beta$ (2)

$Fc = f/\cos\beta$

$Fc_{max} = f$ when $\beta \Rightarrow 0$

where:

Fc_{max} = maximum cutting force, kg
Fc = cutting force, kg
f = force sensed by the load cell, kg
β = cutting oblique angle, deg

Test Rigs

An experimental test rig for measuring the force and energy requirement in cutting oil palm frond was designed and developed. **Figs. 4** and **5** show the experimental rig setting-up for sickle with claw cutter and countershear, respectively. Test rigs were built with mild steel frames. The cutters were actuated by a 4-tone hydraulic cylinder run by a single phase Enerpac Hydraulic Pumpset. The linear velocity of the pusher rod was maintained at 0.6 m/s and this velocity was determined by a specially designed stop watch (Jelani, 1997). As the two cutters were having countershear as to balance the cutting force, thus the velocity of cutting was not a great significance. The force applied through the hydraulic cylinder was sensed by the load cell (Kyowa LT-500 KF) and the signal was amplified by the amplifier (Kyowa WGA-710A).

Experimental Procedures

Cutting force, cutting energy and the effect of cutting on the material are influenced by the knife movement with respect to the material and the countershear. This is also influenced by the knife orientation to its direction of motions. Cutting force is the resultant of the stresses applied on the material (Persson, 1987; Wieneke, 1972 and O'Dogherty, 1981). The cutting energy is

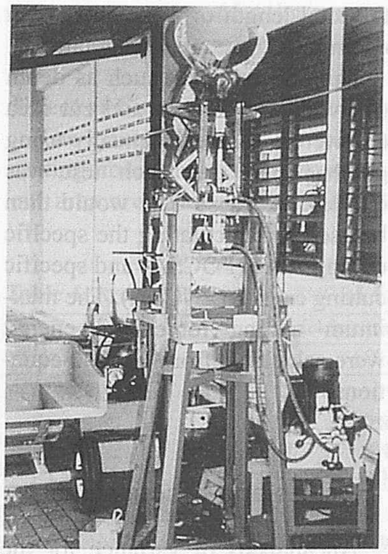

Fig. 4 Experimental set-up of Claw cutter.

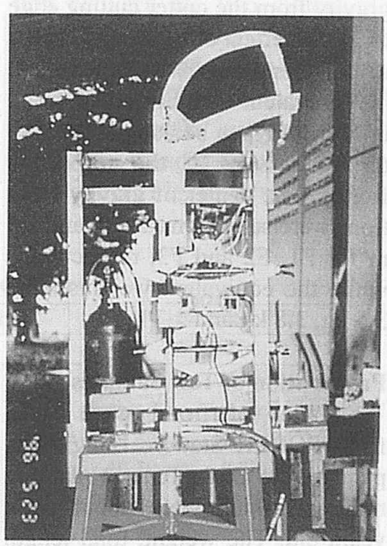

Fig. 5 Experimental set-up of Sickle cutter.

the product of cutting force and the depth of cut or the cut area. Tests were done to evaluate the effect of design (claw and sickle cutters), cutting angle (90°, 60° and 45°) and frond maturity (F1, F2 and F3) on the cutting force and energy requirement. The results were useful for the designer in designing the cutting tool.

The parameters investigated were as follows:

1. Cutter design (T) - methods of cutting, viz. claw (T1) and sickle cutter (T2).
2. Cutting angle (S) - it is the angle of knife with respect to fronds longitudinal, viz. S1 = 90°, S2 = 60° and S3 = 45°.
3. Frond maturity (F) - (i) F1= the second frond below the ripe fruit which is considered as the most matured frond compared with others, (ii) F2 = frond above the ripe fruit, and (iii) F3 = frond above F2. All fronds were taken from palms of similar age and variety.

In this study, one end of the frond sample was fixed and allowed to move vertically with regards to the pivot, so the cutting angle could be varied. An equal weight (half of frond total weight) was put at the end of the frond to get the effect of bending. The cutter was then forced downwards by using the hydraulic cylinder to cut the material into two halves. The force required for cutting was measured by the load cell which was placed in between the cutting device and the hydraulic pusher rod. The reading of the load cell was amplified by the amplifier which recorded the maximum value of the cutting force. The actual value of the cutting force is the difference between the force sensed by the load cell under load and without load. The maximum cutting force was then determined using equations 1 and 2 for claw cutter and sickle, respectively.

Constant Parameters

Table 2. Equation for Maximum Cutting Force and Energy

	Claw cutter	Sickle
Max. cutting force, Fc_{max} (kg)	2f(x)/k	f
Max. sp. cutting force FOCSA (kg/cm^2)	2f(x)/k.A	f/A
Max. sp. cutting energy, ENCSA (kg-cm/cm^2)	FOCSA × d	FOCSA × d

where:
A = cutting area, cm^2
d = depth of cut, cm

Table 3. Analysis of Variance for sp. FOCSA and ENCSA per Unit Cut Area

Source of variation	df	ANOVA SS	Mean Squares	F value	Pr>f
FOCSA					
Design (T)	1	685.80	685.80	112.00*	0.0001
Cutting angle (S)	2	867.10	433.80	70.80*	0.0001
Frond Maturity (F)	2	418.70	209.40	34.20*	0.0001
T*S	2	192.60	96.30	15.70*	0.0001
T*F	2	8.74	4.37	0.71	0.4925
S*F	4	15.40	4.00	0.65	0.6264
T*S*F	4	11.13	2.78	0.45	0.7686
FOCSA					
Design (T)	1	15925.2	15925.2	53.50*	0.0001
Cutting angle (S)	2	16103.7	8051.9	27.00*	0.0001
Frond Maturity (F)	2	16120.8	8060.4	27.10*	0.0001
T*S	2	8756.7	4378.3	14.70*	0.0001
T*F	2	201.2	100.6	0.34	0.7140
S*F	4	868.6	217.1	0.73	0.5741
T*S*F	4	181.3	45.3	0.15	0.9615

Note:*significant at 1% level; speed of cutting = 0.6 m/s; α = 10°; and distance of center of gravity of fronds: F1 = 2 m; F2 = 2.3 m; F3 = 2.3 m

There were three parameters that were set constant. These parameters were the blade edge angle (α) which was set at 10°, speed of cutting which was maintained at 0.6 m/s, and the distance of center of gravity from the cutter cutting edge to get the effect of bending as in the actual operation. In the experiment, an equivalent weight of half-frond only would then be hung at the center of gravity of the respective fronds. The center of gravity locations were set at 2 m, 2.3 m and 2.3 m for F1, F2 and F3, respectively, while the equivalent weights were 3.5 kg, 3.8 kg and 3.7 kg.

Preparation of Test Samples

Samples of frond were taken from PORIM Station, Bangi. The palms were DxP (Tenera) planting material which were about eight years old. The weight, total length and center of gravity of samples were recorded just after they were cut. In this exercise, the sample of fronds were hung with a rope to get the center of gravity. A spring balance was located in the middle of the rope to record the total weight of frond. The center of gravity and the total length of frond were then determined by a measuring tape.

The effects of cut such as depth (d), width of cut (w) and cut area (A) were measured manually using graph paper after each test was completed. These data would then be used for calculating the specific cutting force (FOCSA) and specific cutting energy (ENCSA). The maximum cutting force and energy were calculated from **Table 2** equations:

Results and Discussions

An analysis of variance for the specific cutting force (FOCSA) and specific cutting energy (ENCSA) is

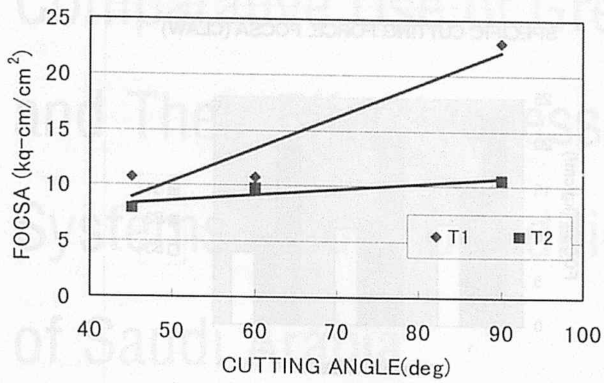

Fig. 6 Effect of cutting angle on FOCSA.

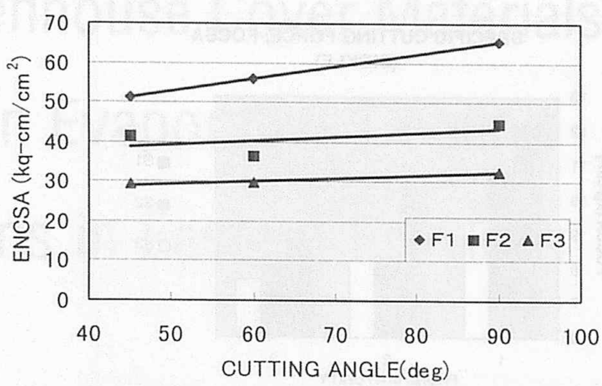

Fig. 7 Effect of cutting angle on ENCSA for Sickle cutter.

shown in **Table 3**. It indicates significant effects of design, cutting angle, frond maturity and the interaction of design and cutting angle. Other interactions were found not affecting the FOCSA and ENCSA. Maximum FOCSA was obtained through the interaction of T2S1F1 (22.9kg/cm^2), while the minimum FOCSA was from T1S3F3 (7.7kg/cm^2) interaction. The maximum FOCSA for the sickle and claw cutter were 12.18kg/cm^2 and 22.9kg/cm^2, respectively. For ENCSA, the maximum value was from the interaction of T2S1F1 (115.52kg-cm/cm^2) and the minimum value was achieved through the interaction of T1S3F3 (29.27kg-cm/cm^2). The maximum ENCSA for sickle and claw cutter were 65.41kg/cm-cm^2 and 115.5 kg/cm-cm^2, respectively.

Effect of Cutters

The effect of design on specific cutting force and energy per unit cut area are presented in **Table 4**. The average specific cutting force and energy were minimum for the sickle cutter, i.e., 9.36kg/cm^2 and 42.98kg-cm/cm^2, respectively, compared to 14.4kg/cm^2 and 67.27kg-cm/cm^2, respectively, for claw cutter. Based on the experiments, the sickle cutter (T1) offers a lower cutting force and energy to that of the claw cutter (T2).

Table 4. Average FOCSA and ENCSA for Two Cutters

	Sickle cutter (T1)			Claw cutter (T2)		
	90°	60°	45°	90°	60°	45°
FOCSA (kg/cm^2)						
F1	12.18	10.80	9.36	22.90	15.22	11.17
F2	10.60	8.42	8.27	20.60	11.56	10.69
F3	8.82	8.11	7.70	19.70	9.80	7.98
ENCSA (kg-cm/cm^2)						
F1	65.41	55.97	51.28	115.20	78.04	57.43
F2	44.75	36.42	41.46	94.40	54.53	52.62
F3	32.59	29.73	29.28	80.60	37.77	34.51

Effect of Cutting Angle

The relationships of cutting angle on the specific cutting force and energy per unit cut area are shown in **Figs. 6** and **7**. In the case of both cutters, the higher cutting angle would result in higher cutting force and energy. The difference in the slope shows that there is an interaction between the two cutters (T1 and T2) and cutting angle (T*S). The specific cutting force and energy did not differ so much at 45°, but as the cutting angle increased, there was a rapid increase for the claw cutter **(Fig. 6)**. It was also noticed that increasing cutting angle from 45° to 90° increased the cutting force by 24% for the sickle and 111% for the claw cutter. Similarly, the cutting energy increased to 17% for the sickle and 110% for the claw cutter due to increase in cutting angle.

The related regression equations between cutting angle and the specific cutting force for the sickle cutter and claw cutter for the most matured frond (F1) are as follows:

For sickle cutter,

FOCSA = 0.0603 (S) + 6.8614

where,

r^2 = 0.9596

For claw cutter,

FOCSA = 0.26 (S) - 0.47

where,

r^2 = 0.9998

Effect of Frond Maturity

Figs. 8 and **9** illustrate the effect of frond maturity on the specific cutting force and energy per unit cut area. The FOCSA and ENCSA required to accomplish the cutting increased as the frond matures. Mature fronds seemed to be a bit difficult to cut due to the strength of the fibers becoming harder as the frond matures.

Conclusions

This study showed that the cutter design, cutting angle and frond ma-

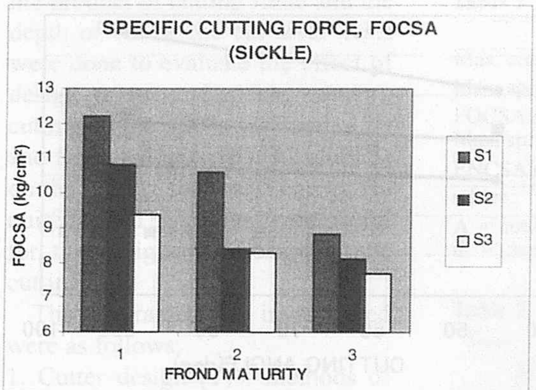

Fig. 8 Effect of frond maturity on FOCSA for Sickle cutter.

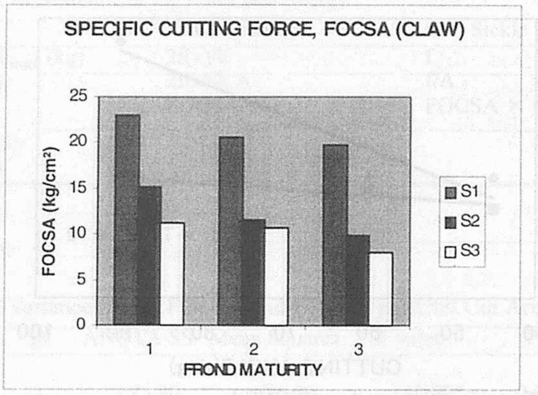

Fig. 9 Effect of frond maturity on FOCSA for Claw cutter.

turity had significant effects on the specific cutting force and energy per unit cut area for cutting fronds (p-value =0.0001). The sickle cutter requires about 9.36 kg/cm^2 compared to 14.4 kg/cm^2 cutting force required by the claw cutter which was about 35% less. Similarly, the cutting energy follows the same trend. The cutting angle has significant effect on the cutting force and energy. The lower the cutting angle (45°), the lower the cutting force and energy requirement. Therefore, in actual cutting operation in the field, the harvester has to bring the pole closer to the palms' trunk to get lower cutting angle, to reduce the cutting force requirement. On the other hand, higher cutting force and energy are required to accomplish the cutting of matured fronds.

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Comparative Use of Greenhouse Cover Materials and Their Effectiveness in Evaporative Cooling Systems Under Conditions in Eastern Province of Saudi Arabia

by

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Abstract

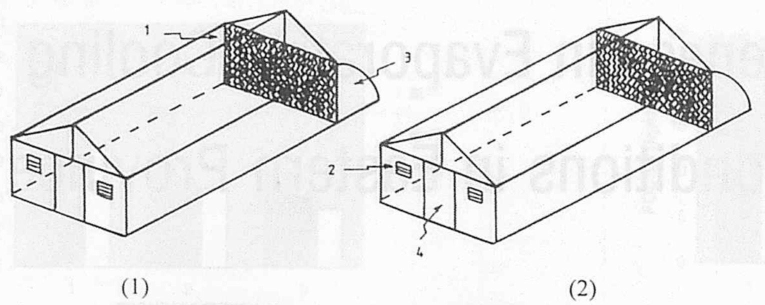
Two experimental gable-even-span greenhouses were designed, constructed and employed to vegetate and produce a cucumber crop during the summer of 1997. One of these greenhouses was covered by 0.8 mm thick fiberglass reinforced plastic. The other one was covered by 0.1 mm thick double layers polyethylene sheets. Two identical evaporative cooling systems (cooling pads and extracting fans) were designed, built and used to reduce the ambient air temperature inside the two greenhouses. These systems of cooling were automatically operated according to the ambient air temperature inside the greenhouse using a differential thermostat.

Results of this research work reveal that the evaporative cooling system inside the fiberglass greenhouse was more efficient than that inside the polyethylene greenhouse due to the difference in intensity of solar radiation inside the greenhouses. Consequently, the fiberglass greenhouse enhanced the rate of growth and increased the fresh yield of cucumbers by 32.87% compared with the polyethylene greenhouse.

Introduction

Protected cropping production is mainly limited due to constraints imposed on the crop by environmental parameters; nutrient elements, pathological factors, and others. Environmental parameters are generally recognized to have a major impact on the production of protected cropping. These parameters include ambient air temperature, air relative humidity, intensity of light, intensity of solar radiation, and intensity of plants per unit area of ground surface. Temperatures of ambient air inside greenhouse are frequently 11 °C higher than those outside in spite of open ventilators. Detrimental effects of high temperatures are typified by loss of the following: stem strength, leaf area, seed insemination, fruit set, size of fruit, and delay of flowering (Buxton et al.,1985).

Evaporative cooling systems for greenhouses or livestock houses are generally of two types, misting/fogging systems and pad systems. Detail descriptions of both types are given by Timmons and Baughman, 1983 and 1984, and Bottcher et al., 1989. These systems are normally evaluated in terms of an evaporative cooling or saturation efficiency, which is defined as the ratio of temperature drop provided by the system to the difference between dry and wet-bulb temperatures, or wet-bulb depression. Misting and fogging systems typically have relatively low evaporative efficiency compared to pad systems. Values of saturation efficiency range from 10% to 37% with an average of 23.5% for misting systems having water pressure ranging from 275 to 1380 kPa while its values range from 6% to 95% with an average of 77.5% for a pad system at the same range of water pressure (Critten, 1988). Fresh air is an important element in greenhouse production system. As the air inside the greenhouse is continuously moving, ambient air temperatures are uniformly distributed, humidity surrounding leaf surfaces is reduced, and carbon dioxide levels are thus decreased (Al-amri, 1997). The greatest values of cooling effect (13.2°C) and cooling efficiency (81.5%) were achieved with the greatest value of wet-bulb depression (16.2°C) and the lowest value of air relative humidity (26%) and



Greenhouse 1, covered by fiberglass *Greenhouse 2, covered by polyethylene*

Fig. 1 Schematic diagram of the experimental greenhouse. 1. crossfluted cellulose pads; 2. extracted fan; 3. shade of cooling system; 4. greenhouse door.

vice versa.(Abdellatif, 1993).

Many types of construction are being used successfully for greenhouses. Some may have advantages over others for particular applications, but there is no one best type of greenhouse. The optimal form of the greenhouse is constrained by choosing a high length -to- width ratio specifying a minimum height to the eaves of 1:5 m and a minimum rafter (roof) slope of 25°. The rafter slope ranges from 28-30° is mainly recommended. Below 22° rafter slope, condensation will drop from the cover rafter rather than running down the underside, thus damaging and encouraging diseases (Thomas, 1978). The greenhouse with a straight side wall and gable roof is the most common shape and has advantages in framing and in space utilization. The choice of cover materials are strongly dependent upon several factors: solar radiation flux incident, long wavelength radiation transmission, resistance to ultraviolet degradation, mechanical strength, coefficient of expansion, temperature limitations, heat loss, initial purchase price, and installation cost (Roberts, 1985; Henemann and Walker, 1987; Waker, 1979; Yang et al., 1989 and many others).

The major purpose of this investigation is to study the effect of greenhouse cover materials on the effectiveness of the evaporative cooling systems under conditions in the Eastern Province of the Kingdom of Saudi Arabia. Also, it intends to investigate, study, and estimate the most important dominant environmental parameters affecting cucumber crop production during the summer season.

Description of Experimental Greenhouses, Evaporative Cooling System and Experimental Facilities

Two experimental gable-even-span greenhouses were designed, constructed and utilized to grow and produce cucumbers during the summer of 1997. Each one had gross dimensions of 8.00 m long, 4.00 m wide, and 3.16 m height, with a net surface area of 32.00 m2 (**Fig. 1**). Water galvanized pipes (38.1 mm diameter) were used to form the structural frame of the two experimental greenhouses. To reduce both the side effects of wind blowing over the roof of the greenhouse and the intensity of solar radiation flux incident inside the greenhouse during the hot summer season, the rafters were sloped at 30o of the horizontal plane. Each gable roof had gross dimensions of 2.310 m long (rafter) and 1.16 m height. The height of each side wall was 2.00 m. The straight-side wall pipes were strongly connected to the concrete foundations in order to transfer gravity, uplift and overturning loads such as those from the crop, suspended equipment, and wind loads safely to the ground. In order to increase and maintain the durability of the structural frame and the polyethylene cover, twenty galvanized tensile wires were tied and fixed throughout the length of rafters and vertical pipes in each side of the polyethylene greenhouse. One of the two greenhouses was covered by 0.8 mm thick corrugated fiberglass reinforced plastic panels, and the other one was covered by 0.1 mm thick double layers of polyethylene sheets. These experimental greenhouses were orientated in an East-West direction. For the rest of this paper, the greenhouse covered by corrugated fiberglass reinforced plastic and the greenhouse covered by double layers of polyethylene sheets are referred to as greenhouse 1 and greenhouse 2, respectively.

The evaporative cooling system is mainly based on the process of heat absorption during the evaporation of water. It mainly consists of cooling pads and extracting fans. A crossfluted cellulose pad was mounted in a vertical fashion at the end of the greenhouse. A PVC pipe (0.5 inch diameter) was suspended immediately above the cooling pads. Holes were drilled each 10 cm long throughout the length of PVC pipe, and the end of this pipe was capped. To spread the water uniformly before it drops onto the cooling pads, a baffle was placed below the water pipe. A water sump was mounted under the pads to collect the water and return it into the water tank (600 liters), from which it could be recycled to the cellulose pads by means of the water pump. In order to bring the cold air onto the plants throughout the growth period, the cooling pads were located 20 cm above the ground surface of the greenhouse. Two extracting fans (single speed, direct driven, 50 cm diameter, and 3630 m^3 /hr discharge) were located on the leeward side of the greenhouse and the pads on the side toward the prevailing wind (opposite side of the extracting fans).The extracting fans were automatically operated using a differential thermostat. They switched "ON" when the ambient air temperature inside the greenhouse was equal or greater than 25°C, and switched "OFF" when the interior ambient air temperature was lower than 25°C. In order to prevent the accumulation of salt on the cellulose pads (mean reason of pad damage), potable water was usually used in the evaporative cooling system.

A drip irrigation system was employed throughout the experimental work. It mainly consisted of five components: main water supply tank, fertilizer tank, pipes, water pump, and drippers. A three thousand liter water supply tank (spherical form, 75 inch diameter) was located 5 m above the ground surface in order to provide an adequate head for maximum use rate of water. In order to mix chemical fertilizers with irrigation water before it passes through the drippers, an eighty liter fertilizer tank (cylindrical form) was placed in the line of the watering system. To distribute the irrigation water uniformly for the two greenhouses, a polyvinyl chloride (PVC) pipe (1.0 inch diameter) was employed as a main. Twenty drippers (0.375 inch diameter) were uniformly distributed alternatively (with 50 cm dripper spacing) throughout each row of plants using lateral rubber pipe (0.5 inch diameter, and 3.785 l/h discharge). The watering system was pumped and run only thirty minutes a day throughout the experimental work, in order to conserve the irrigation water.

Sixteen temperature sensors (on celsius scale) thermocouples were employed to measure the ambient air temperatures (dry and wet-bulb temperatures) at various points inside and outside the two greenhouses. Water temperature of the evaporative cooling system was measured using two thermocouples. Two hygrometer devices were placed inside the greenhouse to measure the air relative humidity. The meteorological data from a weather station of the university (at a distance of about 100 m) was utilized throughout the experimental work. All the sensors were connected to a data-logging system in order to test, display, and record the data throughout the research work.

Two hundred mini-cucumber seeds (NABIL F1 RS Variety) were sown in the nursery on May 14, 1997. The mini-cucumber seedlings were raised in 10 cm peat blocks and vegetated out at the four leaf stage. One hundred and sixty seedlings of cucumber were selected and planted inside the two greenhouses (80 plants for each greenhouse in four double rows, each single row having 1 plants) on the first of June 1997.

In the hot climate of the Eastern Province of the Kingdom of Saudi Arabia, evaporative cooling systems have been commonly employed to reduce the interior ambient air temperature of greenhouses. Evaporative cooling system efficiency (η) is normally defined as (ASHRAE, 1983):-

$$\eta = \frac{T_{odb} - T_{idb}}{T_{odb} - T_{owb}} \times 100 \ (\%) \quad (1)$$

or

$$\eta = \frac{T_{dd}}{T_{wd}} \times 100 \quad (\%) \quad (2)$$

where:-

T_{odb}= dry-bulb temperature of outside air, °C

T_{idb}= dry-bulb temperature of inside air, °C

T_{owb}= wet-bulb temperature of outside air, °C

T_{dd} = cooling effect, °C

T_{wd} = wet-bulb depression, °C

Data were stored in microcomputer files and analyzed using MICROSTAT STATISTICAL PACKAGE. An emperical equation for predicting air temperature inside the greenhouse was developed. A model for predicting mean air temperature of in leaving the evaporative cooling system for different exterior air relative humidities was also developed using regression analysis.

Results and Discussions

The experimental work was carried out during the summer months of 1997 (June to August, 92 days were recorded during this period). Data obtained from this research work are summarized in **Table 1**. It clearly shows that substantial decrease of ambient air temperature inside the greenhouses occurred when the air relative humidity outside the greenhouses was less than 10% and outside air temperature exceeded 46 °C. Under these conditions, the evaporative cooling system provided a cooling effect (air temperature difference between outside and inside the greenhouse) of 20 °C or more. The cooling effect and, consequently, the effectiveness of the evaporative cooling system were strongly affected by the wet-bulb depression (difference between dry and wet-bulb temperatures of outside air) that was affected mainly by air relative humidity. Therefore, all the data collected throughout the experimental work were examined to approach mathematical models (**Fig. 2**). The best fit equations relating the cooling effect (T_{dd}) to the wet-bulb depression (T_{wd}) for the two greenhouses were:-

T_{dd} (Greenhouse 1) =
$-5.6178 + 0.961\ (T_{wd})\ [R = 0.997]$

Table 1. Daily Average, Ambient Air Temperatures Outside (T_{ao}) and Inside (T_{ai}) the Greenhouses, Wet-bulb Depression (T_{wd}), Air Relative Humidity Outside the Greenhouses (R.H) and Effectiveness of Evaporative Cooling System (η)

Month	Greenhouse	T_{ao}	T_{ai}, C°	T_{wd}, C°	R.H., %	η, %
June, 1997	G1	33.70	24.59	15.04	24.92	52.94
	G2		27.06			41.98
July, 1997	G1	37.06	26.30	16.94	23.24	57.92
	G2		28.70			44.59
August, 1997	G1	38.14	26.45	18.58	17.90	67.10
	G2		28.60			52.00
Mean	G1	36.30	25.78	16.85	22.02	59.32
	G2		28.12			46.19

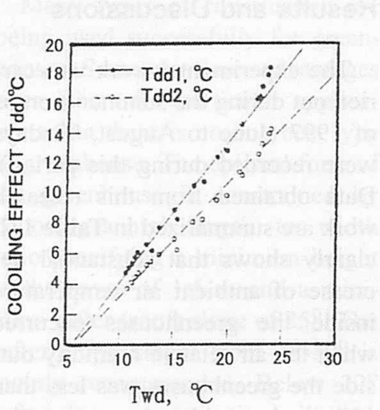


Fig. 2 Cooling effect (Tdd) versus wet-bulb depression (Twd).

T_{dd} (Greenhouse 2) =
-4.3404+0.746 (T_{wd}) [R = 0.997]

The greatest values of cooling effect (18.14 °C) and effectiveness of evaporative cooling system (77.33%) were achieved inside greenhouse 1 at the greatest value of wet-bulb depression (23.90 °C) and lowest value of air relative humidity (9.40%); while, the lowest value of cooling effect (3.69 °C) and effectiveness of cooling system (34.89%) occurred inside greenhouse 2 at the lowest value of wet-bulb depression (10.7 °C) and greatest value of air relative humidity (33.40%). For the duration of the experimental work, the wet-bulb depression was found to be directly related to the exterior ambient air temperature, air relative

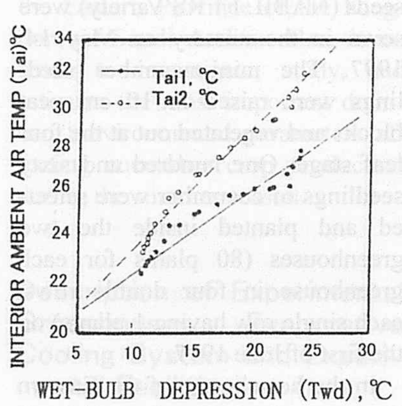


Fig. 3 Interior air temperature (T_{ai}) versus wet-bulb depression (T_{wd}).

humidity and saturation pressure of the air. As the exterior ambient air temperature is increased and the air relative humidity is decreased, the saturation pressure is thus increased making the cooling system more efficient. To examine the relationship between the interior ambient air temperatures (T_{ai}) for the two greenhouses and the exterior ambient air temperature (T_{ao}), air relative humidity (R.H.) and wet-bulb depression (T_{wd}), multiple regression analysis was used. This analysis showed that, the interior ambient air temperatures for the two greenhouses were significantly affected by the exterior ambient air temperature (R = 0.9996; P < 0.001), air relative humidity (R = 0.9995; P < 0.001) and wet-bulb depression (R = 0.9994; P < 0.001).

The multiple regression equations obtained were:-

T_{ai}(Greenhouse 1) = 28.485+0.674 (T_{ao})-0.350 (R.H.)-1.176 (T_{wd})
T_{ai}(Greenhouse 2) = 23.273+0.648 (T_{ao})-0.268 (R.H.)-0.773 (T_{wd})

The combined correlation coefficient between the interior ambient air temperature and these parameters for greenhouses 1 and 2 was 1.0. It also indicated that the differences between the intercepts and the slopes for the two greenhouses were significant (P < 0.001). The ambient air temperature for greenhouses 1 and 2 was plotted against wet-bulb depression (**Fig. 3**). Regression analysis revealed a highly significant linear relationship between these parameters for greenhouses 1 and 2.

It also showed that, the differences between the intercepts and the slopes were highly significant (P < 0.001). The regression equations obtained for the greenhouses were:-

T_{dd} (Greenhouse 1) =
19.260+ 0.387 (T_{wd}) [R = 0.995]
T_{dd} (Greenhouse 2) =
18.014+0.595 (T_{wd}) [R = 0.992]

The effectiveness of evaporative cooling system varied from time to time, day to day and from greenhouse to another throughout the experimental work according to the intensity of solar radiation flux incident, the air relative humidity, the stage of plant growth, the intensity of cucumber plants per square meter of ground, and the type of greenhouse cover. Although cold air just leaving the evaporative cooling system for the two greenhouses was at the same level of temperature, the ambient air temperatures inside the greenhouses were varied from one greenhouse to another throughout the air stream. Greenhouse 1 increased the temperature of cold air just leaving the cooling system by 2.6 °C, while greenhouse 2 increased the interior ambient air temperature by 4.9 °C.

This difference (2.3 °C) was due to the variation in intensity of solar radiation flux incident inside the two greenhouses, which was converted into thermal energy. Because, the polyethylene cover has an effective transmittance (84%) of solar and infrared radiation greater than that for fiberglass reinforced plastic (77%), it was continuously increased the ambient air temperature inside the greenhouse. Also, the plastic cover increased the ambient air temperature just after sunrise faster than the fiberglass cover. As the exterior air relative humidity is decreased and outside dry-bulb temperature exceeded 40 °C, the wet-bulb depression and the saturation pressure increased and the vapour pressure thus decreased making the evaporative cooling system more efficient and vice versa. Wet-bulb depression was plotted against air relative humidity (**Fig. 4**). Regression analysis revealed a highly significant linear relationship ($P < 0.001$) between these parameters. The best fit equation relating the wet-bulb depression (T_{wd}) to the outside air relative humidity (R.H.) was:

$$T_{wd} = 28.110 - 0.536(R.H.)$$
$$[R=0.996]$$

As the number of plants per square meter of greenhouse ground surface area is increased, the consumption rate of light energy in photosynthesis process and shading area inside the greenhouse are thus increased making the rate of heat exchange between the ground bare area and interior ambient air decrease. To assess the most important parameters affecting effectiveness of evaporative cooling system, the exterior ambient air temperature and wet-bulb depression were employed. Multiple regression analysis revealed a highly significant linear relationship ($R = 0.989$; $P < 0.001$) between the exterior ambient air temperature and effectiveness of evaporative cooling system. It also showed a highly significant linear relationship ($R = 0.916$; $P < 0.001$) between the wet-bulb depression and the effectiveness of the evaporative cooling system. The multiple regression equations obtained were

$$\eta\,(\text{Greenhouse 1}) = -9.054 + 2.350(T_{ao}) - 1.006\,(T_{wd})$$
$$\eta\,(\text{Greenhouse 2}) = -16.140 + 2.518(T_{ao}) - 1.726\,(T_{wd})$$

For the duration of the experimental work and due to the reasons discussed previously, the mean effectiveness of evaporative cooling system for the two greenhouses was 59.32% and 46.19%, respectively (**Fig. 5**). Thus, the fiberglass cover mainly increased the effectiveness of cooling process by 28.43%. The relationship between ambient air temperatures inside and outside the two greenhouses was plotted in **Fig. 6**.

The stem length of cucumber plants during this research work for the two greenhouses are plotted in **Fig. 7**. It clearly shows that the difference in stem length of cucumber plants varied from one week to another throughout the growing season. It also indicates that the greater stem length was obtained from greenhouse 1 because the ambient air temperature inside the greenhouse 1 was around the optimum temperature (25 °C), particularly at the critical period (from 10 am to 3 pm) during daylight (**Fig. 6**), while the ambient air temperature inside greenhouse 2 was on average 7.6 °C over the optimum temperature. Therefore, the weekly average rates of vegetative growth (stem length of plant) for the two greenhouses were 24.78 cm/plant/week and 21.67 cm/plant/week, respectively (**Fig. 7**). Thus, greenhouse 1 increased the average rate of vegetative growth by 14.35% (compared with greenhouse 2). This difference may be explained by the fact that the optimum ambient air temperature surrounding the cucumber plants enhances and increases the absorption rate of nutrient elements, photosynthesis process, and building of carbohydrates. As the green areas of leaves increased due to increase of vegetative growth rate, the biochemical processes in-

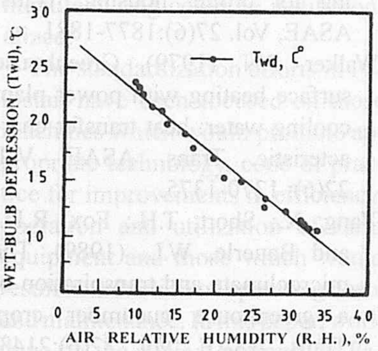


Fig. 4 Wet-bulb depression (T_{wd}) versus air relative humidity (R.H.).

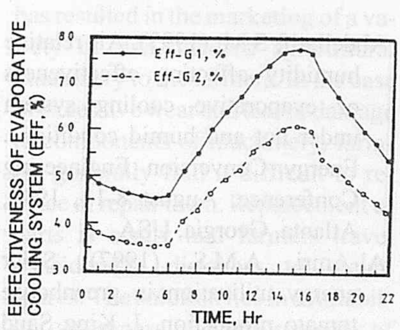


Fig. 5 Relationship between effectiveness of evaporative cooling system and time of day.

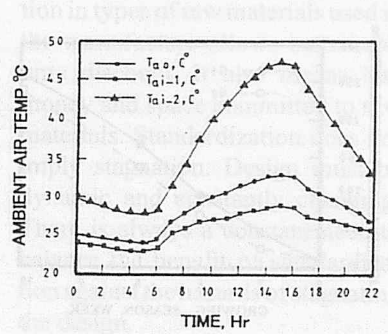


Fig. 6 Relationship between ambient air temperature inside and outside the greenhouses.

creased, making the process of photosynthesis more efficient. The ambient air temperature surrounding the cucumber plants played a vital role not only for vegetative growth, but also in influencing the vitality of seed insemination and, consequently, the number of fruit set per plant. As the ambient air temperature inside the greenhouse is increased over 30 °C, the death rate of seed insemination is increased, and the fruit set is thus decreased making the crop fruitful at a minimum level. Also, excess temperatures of greenhouse ambient air commonly caused loss in stem strength, and area of leaves, delay in flowering, loss of fruit size and increase in pathogenic organisms. Due to all reasons discussed above, the total fresh yield of cucumber crop for the two greenhouses during this research work was 385.6 kg (4.82 kg/plant) and 290.2 kg (3.628 kg/plant) respectively. Consequently, greenhouse 1 increased the average fresh yield of cucumber crop by 32.87%.

Conclusions

The importance of air temperature is demonstrated by the many different effects it has on the growth and development of protected cropping. Air temperature influences the rates of photosynthesis, respiration, and other metabolic processes. Also, differences in day/night temperature

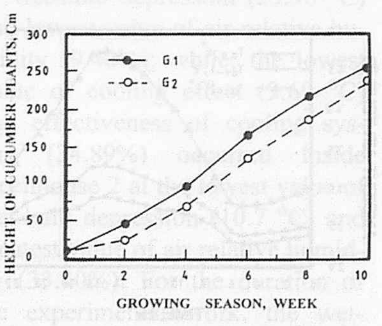


Fig. 7 Relationship between height of cucumber plants and growing season.

affect the balance between the yield and quality of greenhouse crops. The results of this experimental work clearly shows that the variations in exterior air relative humidity took place throughout the day just as air temperatures varied. Usually, the period of lowest air relative humidity was realized during the hottest time of the day, when the greatest degree of cooling was required. Fortunately, the evaporative cooling system at that time was more efficient. The exterior ambient air was continually cooled by evaporative process to approximately 2.6 °C over the wet-bulb temperature as it was passed through the wetted pads. This cold air removed the excessive heat from the plant zone as it was continually drawn throughout the greenhouse. The fiberglass cover (FRP) mainly increased the effectiveness of the cooling process by 28.43% compared with the polyethylene cover (PE). Because, the effective transmission of polyethylene cover (84%) is greater than that, of the fiberglass cover (79%), the intensity of solar radiation inside the plastic greenhouse was higher than that inside the fiberglass greenhouse, particularly at or around noon. Thus, the greenhouse covered by 0.8 mm of fiberglass reinforced plastic, on the average, increased the rate of vegetative growth by 14.35%, and fresh yield of cucumber crop by 32.87%.

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Farm Machinery Standardization

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Abstract

The efficiency and quality of farm work depends upon performance and capabilities of farm machines. To ensure quality, safety and profitability of farm machines, a performance criteria for products, materials and systems must be established so that interchangeability between two similar products manufactured by two or more organizations could be provided, variety of components required could be reduced, the degree of personal safety and the efficiency of engineering efforts could be increased, a sound basis for codes, education and legislation related to agro-industry could be developed, uniformity of practices among different countries could be promoted and specialized manufacturing could be institutionalized.

The standardization efforts in Pakistan have been focused on those machines which would promote appropriate technology, code of practice for improvements in efficiency, operation and utilization of farm equipment and those which would result in reduction of operating costs and maintenance. In this paper, work done on the subject not only in Pakistan but also at regional and international levels is briefly reported.

Introduction

Timeliness of farm operations is an essence of efficient agriculture and the use of appropriate farm machinery is a condition for achieving the optimum crop yields. Farm machines play a catalytic role in increasing agricultural productivity by ensuring the optimum utilization of inputs, reducing drudgery and costs of cultivation, in general, to improve management of an agricultural enterprise.

There are some 450 small-and-medium scale enterprises (SMSEs) producing farm machinery in Pakistan. They have been producing machinery for different farm operations, each following its own design specification depending upon the availability of local raw materials and customer requirements. This has resulted in the marketing of a variety of non-standard agricultural machinery to the farmers. In the case of excessive wear and tear or damage to components of machinery, farmers generally find it difficult to replace or repair them. Replacement of parts is costly and farmers travel considerable distances to secure spares. The solution lies in availability, awareness and enforcement of standards relating to farm machinery in the country. Standards define the general shape, dimensions, material of construction, minimum performance limits and test procedures, and these are vital in raising and ensuring quality and safe use of machinery. The producers and users are benefitted equally from standardization. The producer is facilitated to produce quality machinery with reduced production cost while the consumer receives quality and reliable machinery.

Standardization allows few tooling changeovers to manufacture batches of components as there are relatively limited components. It also helps in reducing the inventory needed by the manufacturer as there are lesser types of components in stock and takes less time to cycle through the entire product line. Parts standardization simplifies record-keeping because there are fewer components to track. Standardization in types of raw materials used in the manufacture allows better volume discount. It also means less money and space committed to raw materials. Standardization does not imply stagnation. Design must be dynamic and constantly changing. There is always a constant need to balance the benefit of standardization against the hazards of stagnating the design.

The efforts made at international, regional and national levels for the promotion of farm machinery standardization briefly follows:

Standardization at International Level

The first move for the harmonization of farm machinery standardization at the international level was made in 1929 by the then International Standards Association by setting up a Technical Committee ISA/23 with the Secretariat in Germany. This committee held one meeting each in 1930 and 1931 at Hague and Copenhagen, respectively. However, later the work of this committee came to an end due to World War II.

From 1943 to 1947, the international standardization activities were carried out by the United Nations Standards Coordinating Committee (UNSCC) with its 18 members to facilitate unification of standards at the global level.

In 1947, at the initiative of UNSCC, International Standards Organization (ISO) was formed for the development and promotion of standards in the world to facilitate international exchange of goods and services. Its Technical Committees ISO/TC 23 and ISO/TC 22-Automobile with their Secretariats in Portugal and France were allotted the subjects of farm machinery and tractors, respectively. In 1949, it was decided to have a separate section as ISO/TC 22T attached to ISO/TC 22. Simultaneously, the Food and Agriculture Organization (FAO) of the United Nations also took interest in the work and started its own efforts. During the second meeting of ISO/TC 22T held in October 1952, specific proposals such as Power Take Off (PTO) hitch connections, track width, tires, power lifts, pulleys and test code were made for the preparation of international standards. The task for development of standards on PTO, three-point linkage and tractor test code has been accomplished.

In 1971, it was decided to merge ISO/TC 22T with ISO/TC 23 and designate it as Agricultural Machinery and Tractors (now-renamed as Tractor and Machinery for Agriculture and Forestry) Technical Committee with its Secretariat in France so as to have better liaison with the work of tractor and other agricultural machinery. Eighteen sub-committees and a number of working groups were set up under ISO/TC 23 to deal with various aspects of agricultural machinery. The handbook No. 13 provides a number of international standards which have been brought out in a consolidated form. At present ISO, Geneva, International Electro-technical Commission (IEC), Geneva and Organization Internationale De Metrologie Legale (OIML), Paris are the main organizations for the development of international standards.

Standardization at Regional Level

There are different regional organizations engaged in the standardization activities related to farm machinery by virtue of their geographical grouping. These are the Arab Organization for Standardization and Metrology (ARSM) for Arabian Countries, African Regional Standards Organization (ARSO) for African countries, European Committee for Standardization (GEN) for countries in Europe and Pan American Standards Committee (CO-

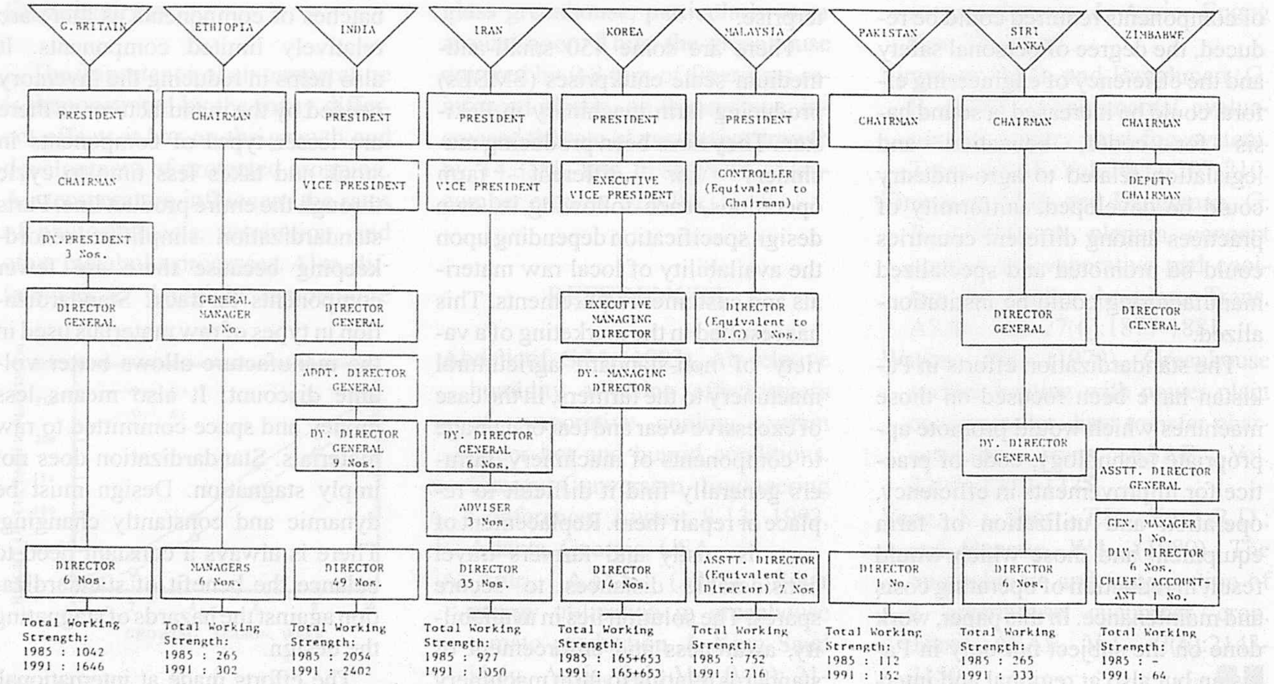

Fig. 1 Organaizational strengths of selected standards organizations.

PANT) for Latin America.

Regional standardization in the field of tractors was attempted by the then Organization of European Economic Cooperative (OEEC) which issued tractor test code in 1959. Later this was endorsed by the Organization for Economic Cooperation and Development (OECD). COPANT has also worked on the development of standards related to tractors.

For the promotion of standardization in Asian Countries, Asian Standards Advisory Committee (ASAC) was established in November 1967. ASAC played a significant role in this field until it became inactive when its parent organization i.e. Asian Development Council was discontinued as a part of the rationalization of the United Nations (UN) Economic and Social Commission for Asia and the Pacific (ESCAP) structure in 1974. More recently Regional Network for Agricultural Machinery (RNAM), an ESCAP Project, played an active part in the promotion and coordination of efforts related to farm machinery standardization amongst its participating countries. Twelve countries of the region participating in the project are: Bangladesh, India, Indonesia, Islamic Republic of Iran, Nepal, Pakistan, Peoples Republic of China, the Philippines, Republic of Korea, Sri Lanka, Thailand and Viet Nam.

RNAM plays a catalytic role in the identification, testing, development, manufacture, popularization and use of appropriate machinery, tool and equipment in the participating countries with a view to enabling small farmers to attain high levels of productivity and income. RNAM organized a regional workshop and a training course on the subject of farm machinery standardization in India during 1983 and 1984, respectively. RNAM has also formulated 18 test codes and procedures related to farm machinery. They provide the methodology, instrumentation, data sheets and reporting formats. **Figure 1** presents organizational strengths of some selected standards organizations of RNAM participating countries in comparison with Great Britain, Malaysia, Ethiopia and Zimbabwe.

Standardization at National Level

The Pakistan Standards Institution (PSI) is the apex national standards body in Pakistan. Activities of the PSI are: preparation, printing, sale of Pakistan Standards and registration of inspection agencies. PSI is administratively attached with the Federal Ministry of Science and Technology. It was established in 1951 as Directorate within the former Directorate General of Supplies and Development, Ministry of Industries as a result of the recommendation of the Pakistan Industrial Conference organized by the Government in December 1947. Later, it was made an autonomous body by registering under the Societies Registration Act of 1860 in 1958. Its status was given a legal cover in 1961 through the enactment of the Pakistan Standards Institution Certification Marks Ordinance 1961. This ordinance has also empowered the PSI to enforce Pakistan standards on compulsory basis. In order to control and regulate in inspection of goods exported out of, or imported into Pakistan. The Government promulgated an Inspection Agencies (Registration and Regulation) Ordinance, 1981. Under this ordinance, the PSI was declared as a competent authority to register inspection agencies.

The PSI has technical links with international and regional organizations involved in this field and it also acts as an agent for the procurement and sale of their standards in Pakistan. Up to December 1994, the PSI had established 3617 standards with the help of its eight different technical divisions, namely; Agriculture and Foods, Chemical, Civil Engineering, Electronics, Electro-technical, Mechanical, Textile, and Weights and Measures.

Standards Development Procedure

The PSI bases it standards formulation approach on an internationally recognized procedure to enhance the export trade of Pakistan. The standards are formulated in a systematic manner through various technical committees on the basis of proposals received at PSI headquarters on any specific subject. After investigation and careful study the need for the standard is established and the task is assigned to relevant committee. The committee explores and studies the subject and prepares draft. Some-

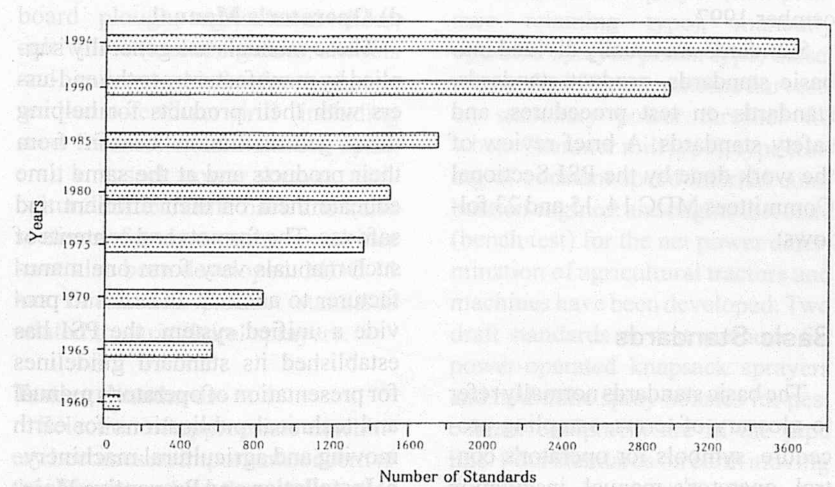

Fig. 2 Number of standards developed by Pakistan standards institution.

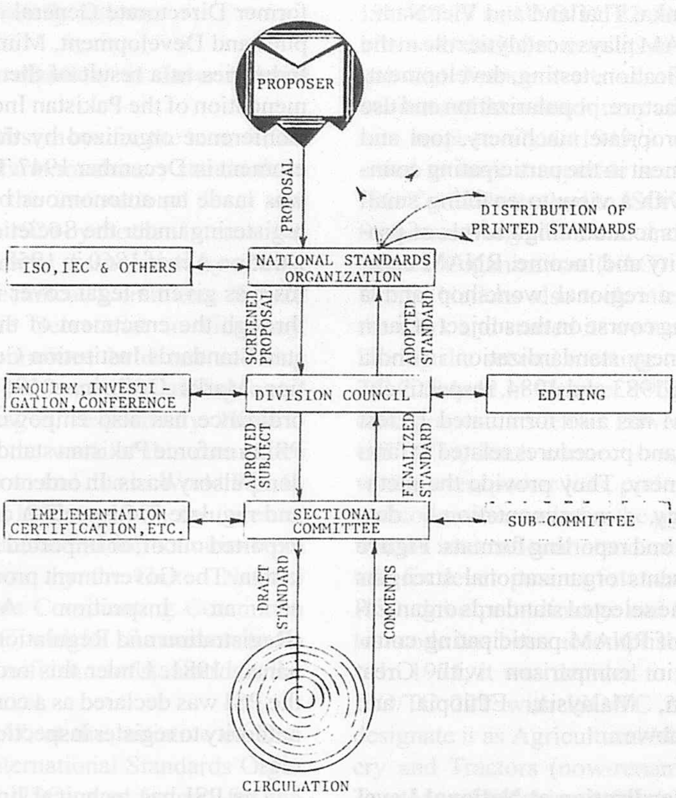

Fig. 3 Standards development procedure.

times the matter is entrusted to a sub-committee for spade work. During these stages, international standards are also consulted. The draft standard then prepared is discussed by the committee after having been circulated to the concerned quarters for seeking their comments, and is finalized. The finalized draft is then forwarded to the concerned divisional council for adoption as Pakistan Standard. The adopted standard is printed, gazette notified and put on sale. The PSI also reviews its adopted standards after about five years for their validity.

Standards Developed

Although the subject of farm machinery standardization was taken up by the PSI in 1967, yet it has picked up its pace only during the last one and a half decades. The Mechanical Division of PSI has 30 Sectional Committees out of which MDC 14, MDC 15 and MDC 23 deal with standardization of farm machinery, plant protection equipment, and earth moving machinery, respectively. These committees consist of representatives drawn from industries, end-users, academic institutions, research, policy making and credit lending organizations. The MDC 14, 15 and 23 have developed 53, 10 and 18 standards, respectively, up to December 1997.

Standards are usually divided into basic standards, product standards, standards on test procedures, and safety standards. A brief review of the work done by the PSI Sectional Committees MDC 14, 15 and 23 follows:

Basic Standards

The basic standards normally refer to glossary of terms, sampling procedure, symbols for operator's control, operator's manual, installation and preventive maintenance. These are of paramount importance and are developed for general application. The PSI has brought out 21 basic standards related to agricultural machinery and are detailed as follows:

a) Glossary of Terms

These standards are developed with a view to providing authentic definitions of the terms commonly used in relation to agricultural machinery. The PSI has finalized its standards on this subject for earth moving machinery; tillage and intercultivation equipment; plant protection equipment; and, harvesting, threshing and forestry machinery.

b) Sampling

The PSI has developed a standard on method of sampling of agricultural machinery and tractors in order to carry out uniform evaluation of a lot of finished products for quality and acceptability reasons.

c) Symbols for Operator's Control

The use of symbols eliminates the language barriers as it enables operators to identify and understand the functions of different controls thus leading to better handling and safe operation of the machinery. Besides the symbols, their location in the machine and method of operation is equally important and complementary to each other. Three standards have been brought out by the PSI on this topic related to earth moving and agricultural machinery, including tractors.

d) Operator's Manual

These manuals are generally supplied by manufacturers to the end-users with their products for helping them get maximum benefit from their products and at the same time educate them on their efficient and safe use. The formats and contents of such manuals vary form one manufacturer to another. In order to provide a unified system, the PSI has established its standard guidelines for presentation of operator's manual and technical publications for earth moving and agricultural machinery.

e) Installation and Preventive Maintenance

Proper installation and maintenance guidelines are of extreme importance in order to exploit maximum potential of a machine to obtain trouble-free service and safe operation of machinery. Pakistan standards on code of practice for installation, operation and preventive maintenance of combine harvester and tractors have been developed.

Product Standards

This normally refers to standardizing the design of a product, i.e., its specifications. The exact and precise prescription of quality in terms of technical requirements is a standard specification. A specification generally prescribes product related terminology, raw material, dimensions and designs wherever necessary, performance and their methods of test and analyses. A product standard thus becomes a prerequisite for certification.

The PSI has prepared standards on some of the components of tractors such as power take-off; drawbar; attachment of mounted implements; mounting for front ballast weight; three point linkage (categories 1,2,3 & 1N) and, lynch pins; and, operator's work place access and exit dimensions, and seating accommodation dimensions. Pakistan standards on some of the tractor- and animal-drawn equipment and their parts such as mould board plough, agricultural discs, cultivators (including tines, shovels and sweeps), blade for scrapers, seed-cum-fertilizer drill (including seed feed roller and furrow openers), knapsack power sprayer, agricultural trailer, knife back and knife sections for harvesting machines have also been developed. MDC 15 has finalized 7 product standards related to agricultural sprayers.

Testing Standards

Selection of appropriate machinery becomes an important concern to the farmers when they start using machinery in place of hand tools commensurate with the pace of agricultural mechanization. Therefore, in order to furnish authoritative guides for farmers in selecting machineries, test for the technical evaluation of machines by official agencies become inevitable. On the other hand, such tests by official agencies are indispensable not only for the sake of trade by machinery manufacturers but also to ensure quality improvement of the machinery.

The testing of the machineries means a systematic determination of the functional performance, structural strength, durability, power requirements, capacity and external forces acting upon it by carrying the test under a wide range of conditions, both in the field and laboratory. One of the main contributions of PSI in the promotion of farm machinery standardization in Pakistan has been the development of standards on test procedures for determination of power (drawbar and PTO), hydraulic lift capacity, turning and clearance diameters, travel speed, noise level, visibility and field performance of agricultural tractors, drafts of which were prepared and proposed by the Testing and Standardization Section of the Farm Machinery Institute, NARC, Islamabad. The test codes on seed-cum-fertilizer drill, knapsack (pneumatic) compression sprayer (non-pressure retaining type), manually operated sprayer-piston type, diesel engine fuel filters, combine harvester, stationary power thresher for wheat (hammer mill type), type testing of constant speed internal combustion engines, and engine test code (bench test) for the net power determination of agricultural tractors and machines have been developed. Two draft standards on test methods for power-operated knapsack sprayers and hydraulic spray nozzles for pest control equipment are in the pipe line. Four standards on earth moving machinery, i.e., test methods for measurement of tool movement time, operator's field of view (Parts I and II) and for locating centre of gravity have been developed.

Safety Standards

The increasing application of modern technology in man-machine environment system has also resulted in accidents of serious nature whenever safety has been overlooked. Realizing the gravity and severity of accidents, many countries have had farm accidents studies to know the causes and frequency of accidents for effective counter measures. A survey conducted by the Farm Machinery Institute, NARC Islamabad revealed that the sprayers, tractors and threshers had been the prominent source of majority of farm accidents in Pakistan. A standard on provision of safety on farm implements has been publicized by the PSI. Noise level tests were included in the standards on test procedures for tractors and combine harvesters with a view to generating data for setting the safe limits.

The other relevant standards finalized by the PSI include earth moving machinery - access system and minimum access dimensions, performance of constant speed internal combustion engines for general purpose, the PS system of limits and fits general tolerance and deviations. The PSI has also finalized two standards on the grouping of service tools of earth-moving machinery depending upon the nature of work.

Implementation of Standards

The implementation of farm machinery standards in Pakistan is on a voluntary basis. Full benefits of standardization cannot be reaped unless standards are adopted and practiced. The production of agricultural implements and machines in the country is mostly undertaken by small-scale manufacturers who do not get information on standards. The first step is to make the standards available to them in Urdu (national language). The next step would be to persuade and assist the manufacturers to follow the set standards.

Problems and Challenges

The farm machinery standardization activity in Pakistan is fairly recent. Some work has been done and a good deal of work is yet to be done. The publicity and availability of standards is entirely the responsibility of the PSI. The responsibility of organizations engaged in machinery trade is to persuade manufacturers and the farmers to prefer and insist upon the purchase of machines conforming to Pakistan standards.

Gap at the manufacturer's end

The formulation of standards is based on the overall requirements of the country and does not necessarily include manufacturing details. However, details of raw material, testing methods, operational and maintenance practices etc. are essential. The Pakistan Agricultural Machinery and Implements Manufacturers Association (PAMIMA) can play an important role in bringing about a conscious change for adoption and popularization of agricultural machinery standards in Pakistan.

Gap at the user's end

Extension services are required from information dissemination point of view. The awareness of end-users as to current state of national standards is important. This can be made possible through setting up of a network of PSI inspectorate in various parts of the country. The induction of experienced and qualified manpower, including engineers in the industry should not be overlooked. This is required to maintain interest and upgrading of skills and knowledge. PSI can take up this responsibility through education and training.

Conclusion

Keeping in view the importance of standardization for improved performance of agricultural machinery due emphasis is needed for creating the necessary infrastructure, motivation and training programmes. The PSI carries a major responsibility to meet the challenge. Some efforts have also been made to rationalize those systems and equipment by providing suitable and timely standards. It is hoped that the formulation of standards would provide effective and economical application of farm machinery in the country.

Recommendations

a) Government should examine the possibility of providing incentives to the manufacturers and end-users in the adoption of machinery and equipment conforming to Pakistan Standards.

b) Quality raw material banks at the major farm machinery manufacturing centres in the country be established to enable the manufacturers to use recommended materials for agricultural machinery production.

c) In order to improve and maintain the quality of locally produced farm machinery there is a need for adoption of PSI standards through the establishment of a centre of excellence in the country. The centre is visualized as a specialized facility located near Lahore, possibly with regional satellites in north south of the country.

d) Particular attention should be given to timely printing and availability of Pakistan Standards for consultation.

e) The PSI should establish additional facilities, especially for dealing effectively with agricultural machinery and equipment on priority basis.

f) Hand tools standardization has been entirely neglected in Pakistan. The PSI should either assign this task to MDC 14 or create a separate Sectional Committee on the subject.

g) The Farm Machinery Institute, NARC Islamabad and Agricultural Mechanization Research Institute, Multan be accredited by PSI as its testing agencies.

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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.

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Tractive Efficiency Study of Tractor Tiers: R. Manian, Professor, College of Agrl.Engg. TNAU, Coimbatore 3, India. K. L. Gorkhali, Agrl.Inputs, Corporation, Nepal. Dr. K. Kathirvel, Asst.Professor, College of Agrl.Engg. TNAU, Coimbatore 3, India.

The best use of the tractor during agricultural operations is when it is operating at high tractive efficiency. This is essential in the present day of energy crisis. The operator is not having any control over the design parameters but he can manipulate the tire inflation pressure and ballast on tractive wheels under varying operating conditions to achieve best models. Experiments were conducted in soil bin consisting of black clay loam and sand with the widely used tires of size 11.2-28 and 13.6-28 in 35 to 45 hp tractors.

The tractive efficiencies of tires decrease in black clay loam soil and sand as tire inflation pressure is increased. The overall tractive efficiency of 11.2-28, 6 ply tire is found to be optimum at an inflation pressure of 0.625 kg/cm^2 when operating in black clay loam soil. A range of 0.50 kg/cm^2 to 0.75 kg/cm^2 inflation pressure in 13.6-28, 4 ply tire will lead to better performance in black clay loam soil. In sand, both tires give the highest tractive efficiency at lowest tested inflation pressures of 0.375 kg/cm^2 and 0.25 kg/cm^2, respectively, but tire life has also to be considered before operating at lowest inflation pressure to achieve higher tractive efficiency.

A ballast of 428 kg on 11.2-28, 6 ply tire increases tractive efficiency in black clay loam soil at moisture range of 11.0 to 16.12 per cent, whereas a ballast range of 275 to 325 kg is found to be adequate on 13.6-28, 6 ply tire for developing optimum tractive efficiency in black clay loam soil.

Increase in net traction and tractive efficiency occurs with an increase in soil moisture to a limit. Moisture range around 11.0 to 16.30 per cent in black clay loam soil is found to be the optimum for better performance by 11.2-28, 6 ply and 13.6-28, 4 ply tires. Lowered moisture than above range lowers the tractive efficiency of tires.

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Tractive Performance Evaluation of a Tractor–Drawn Trencher: M. Devananda, Ph.D. Student, College of Agrl.Engg. TNAU, Coimbatore 641003, India. R. Manian, Professor. Dr. K. Kathirvel, Asst.Professor.

A tractor-drawn trencher was developed for opening a rectangular trench of 30 × 30 cm, suitable for 35 - 45 hp tractors. The unit consists of two bottoms placed in line one behind the other. The front and rear bottoms operate at a depth of 0 - 15 cm and 15 - 30 cm, respectively. The two bottoms throw the removed soil in opposite directions and form vertical walls on each side of the trench. Field performance trials were conducted in black clay loam and red sandy loam soils.

The test results indicated that a maximum drawbar pull of 8.65 and 9.4 kN were needed at operating speed of 2.5 kmph in black clay loam and red sandy loam soils, respectively. At an optimum slip of 10 per cent, the maximum drawbar pull required were 6.90 in black clay loam and 7.25 kN in red sandy loam soils, respectively, at forward speed of 2.5 kmph. The optimum economical speed of operation and the corresponding fuel consumption were 2.25 kmph and 4.5 × 10^{-3} m^3/h in black clay loam and 2.17 kmph and 4.58 × 10^{-3} m^3/h in red sandy loam, respectively.

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A Plough Type Trencher Bottom Geometry – Design and Evaluation: R. Manian, Professor, College of Agrl.Engg. TNAU, Coimbatore.3, India. M. Devananda, Ph.D. Student. Dr. K. Kathirvel, Asst.Professor.

A study was undertaken to design and develop a tractor-drawn trencher bottom for opening a rectangular trench of 30 × 30 cm, suitable for 35 - 45 HP tractors. The developed unit consists of two bottoms placed in line one behind the other. The front and rear bottoms operate at a depth of 0 - 15 cm and 15 - 30 cm, respectively. The two bottoms throw the removed soil in opposite directions and form vertical walls one on each side of the trench. The trencher mould board shape was expressed in terms of mathematical equations. At each Y coordinates of the mould board, section 0 to 30 cm was parabola, except at 20 and 27.5 cm. From section 30 cm to tail end, the best fit curve equations varied from section to section. The cross sectional area of trench was increased by 3.6 and 3.7 percent in black clay loam and 12 and 16 percent in red sandy loam soils with increase in speed from 1.5 to 2.0 kmph and 2.0 to 2.5 kmph, respectively. The non-uniformity index for width and depth of trench cut were less than 5 percent at all the three speeds of operation. ■■

NEWS

American Society of Agricultural Engineers News Release for Immediate Release

America

ST JOSEPH, MICHIGAN-Agricultural engineering shared the limelight in Washington, DC, Tuesday when Agricultural Mechanization was named among the 20 Greatest Engineering Achievements of the 20th Century. The announcement was made on behalf of the National Academy of Engineering (NAE) by astronaut/engineer Neil Armstrong at a luncheon of the prestigious National Press Club. Ranked number seven on the list, Agricultural Mechanization stood alongside such other engineering accomplishments as Television, Automobiles, Computers, Safe and Abundant Water Supply, and-receiving the top citation-Electrification. Larry Huggins, President of the American Society of Agricultural Engineers (ASAE) and Associate Dean of Engineering, Purdue University, attended the luncheon and was very pleased with the announcement. "It is a tremendous honor for the work done by agricultural engineers to be recognized by the National Academy of Engineering," he said.

Huggins credits engineers' effort to mechanize agriculture with dramatic changes in farm productivity and labor. "Mechanization enabled much of the population to leave production agriculture and industrialize the world, and it has simultaneously improved the quality of diet and lowered food costs," he said. "The innovative designs and achievements of agricultural engineers have helped make the US the leading agricultural producer in the world."

VDI Professional Congress "The future of work" Working in global networks Discussion opened on World Engineers' Convention 2000,

June 19 21, 2000

Hanover, Germany

The vision that we can work with whom, when and where we want" will become reality in a new global industrial society with the help of modern information technology. This thesis is being introduced by the Programme Committee for the Professional Congress "The future of work", which will take place next year as one of five Professional Congresses associated with the World Engineers' Convention 2000 from 19 to 21 June during the World Exposition EXPO 2000 in Hanover. The organiser is the VDI The Association or Engineers in cooperation with the EXPO 2000 Hanover GmbH.

The theses for the Professional Congresses at the World Engineers' Convention 2000 will be published in Internet forums and mode available for discussion by the global technical community. Comments, suggestions and results of discussions from the forums will flow into the discussion during the Professional Congresses.

The Internet forums for the Professional Congress for the world Engineers' Convention 2000 can be reached under http://www.vcii.de/wec.

VDI Verein Deutscher Ingenieure. The Association of Engineers. L'Association des Ingenieurs P.O. Box 10 11 39. D-40002 Duesseldorf/Germany - Visitors: Graf-Recke-Str. 84. D-40239 Duesseldorf/Germany

Phone: +49 211 62 14-0. Fax +49 211 62 14-575. E-Mail: wec-expo2000@vdi.de. Internet: http://www.vdi.de

International Agricultural Engineering Conference 2000 and 10th Anniversary Celebration of AAAE

Bangkok. Thailand

After successful conferences in 1990. 1992, 1994, 1996 and 1998, the next conference will be from 4th to 7th December 2000 at the Asian Institute of Technology, Bangkok. Thailand. This conference will coincide with 10th Anniversary Celebration of the Formation of Asia Association for Agricultural Engineering. Till today, over 80 abstracts have been received and the authors are being notified of the acceptance of their papers.

Techono-Festa

To be held during International Agricultural Engineering Conference, 4-7 December 2000 On December 5, 2000

Background and Purpose:

Agricultural mechanization over the years has been at an incredible high rate in Asian countries with their economic developments. Techno-Festa will be specially designed for agricultural machinery industries, including domestic manufactures. The main purpose in to exchange technical information and relevant experience of professionals concerned of machine design and development as well as appropriate technologies depending upon the regional needs. Thus this will be an excellent opportunity for machinery manufactures to gain design and production information required for technical advancement, including small-to-medium-sized enterprises in Asian countries.

NEWS

The Top Ten Greatest Achievements of the 20th Century
Ranked by Niel Amstrong, the first moon walker and a Purdue University graduate

1. Electrification
2. Automobile
3. Airplane
4. Water purification and safety system
5. Electronics
6. Radio/Television
7. **AGRICULTURAL MECHANIZATION**
8. Computer
9. Telephone
10. Air conditioning and refrigeration

This conference will be held on

2nd International Conference on "Engineering Contributions to Food Security in Developing Countries"
Kumasi, Ghana

24-28 September 2000 at the University of Science and Technology, Kumasi, Ghana.

Topics include:

The official language will be English.

For further information, please contact:

Dr. S.K. Agodzo, Conference Secretary, GHAE, Agricultural Engineering Department University of Science and Technology, Kumasi, Ghana.

Tel: +233-51 60242/60328; Fax: +233-51 60137

E-mail: ustlib@libr.ug.edu.gh

EurAgEng International Conference on Agricultural Engineering into the Third Millennium, AGENG Warwick'2000
July 2-7, 2000
University of Warwick, 2000, UK.

Contact:

Alan Spedding

Royal Agricultural Society of England

Stoneleigh Park

Warwickshire CV8 2LZ, United Kingdom

Tel: +44-1203-696969

Fax: +44-1230-535732

Email: alans@rase.org.uk

The 3rd IFAC Workshop on Control Applications in Post Harvest and Processing Technology,
October 3-5, 2000
Tokyo, Japan

Contact:

Prof. Yasushisa Seo, Faculty of Agriculture

Graduate School of Agricultural and Life Science, University of Tokyo Yayoi 1-1 Bunkyo-ku, Tokyo 113-8657

Tel/Fax: +81-3-5689-8095

CIGR/JSAM Workshop pn Bio-Robotics, Information Technology and Intelligent Control for Bioproduction System
November 25-26, 2000
Osaka, Japan

Contact:

Prof. Naoshi Kondo, Faculty of Agriculture, Okayama University 1-1-1 Tsushimanaka, Okayama, Japan 700-8530

Tel/Fax: +81-86-251-8351

E-mail:nkondo@cc.okayama-u.ac.jp

International Agricultural Engineering Conference 2000,
December 4-7, 2000
Bangkok, Thaikand

Contact:

Ms. Tasana Sa-id, Agricultural and Food Engineering, Asian Institute of Technology, P.O.Box 4, Klong Luang, Pathumthani 12120, Thailand

Tel: (66-2) 524 5489/5245450

Fax: (66-2) 524 6200/5162126

E-mail: aaae@ait.ac.th

15th ISTRO Conference – "Tillage at the Threshold of the 21st Century: Looking Ahead"
July 2-7 2000, Fort Worth, Texas, United States.

Contact:

ISTRO-2000 Conference Dr. John Morrison USDA-ARS-GSWRL 808 East Blackland Road Temple, Texas USA 76502

Tel:+254 770 6507

Fax:+254 770 6561

E-mail: istro@brcsun0.tamu.edu

http://www.agen.tamu.edu/organizations/istro

☆ ★ ☆

Agri-Food 2000-Key to the Future

July 15-19, 2000
Winnipeg Convention Centre,Canada

A Joint Conference of the AIC, CIFEST and CSAE

For more information about Agri-Food 2000, contact:

Tel: +1-204-261-6139
Fax: +1-204-261-6565
E-mail: AF2000@mia.mb.ca

The Eighth International Symposium on Animal, Agricultural and Food Processing Wastes (ISAAFPW 2000)

Amrica

Sponsored by

The American Society of Agricultural Engineers

Purpose

This conference will concentrate on managing wastes and nutrients from agricultural production and food processing, including treatment processes and utilization, and environmental impacts, particularly to land and water.

Topic Areas

Nutrient Management
Waste Treatment System (composting, solid/liquid separation, SBR etc.)
Waste Utilization
By-Product Recovery
Water Quality Impacts
Food Processing Waste Treatment
Land Application
Mortalities
Aquacultural Waste Treatment
Recycling Nutrients

For other information contact: Brenda West, Director, ASAE Meetings & Conferences, 2950 Niles Road, St. Joseph, MI 49085-9659.

Voice: +1 (616) 428-6327(Eastern Time Zone)
Fax:+1 (616) 429-3852
Internet: west@asae.org

First International Swine Housing Conference

America

Sponsored by

The American Society of Agricultural Engineers

Purpose

The propose of this conference is to bring university, industry and swine production personnel together to share information, technological advancements and research findings to improve the swine industry.

Topic Areas

Housing Type
Structural Components
Ventilation and Environment Control
Animal Welfare
Feeding Systems
Energy Conservation
Equipment Selection
Site Selection
Consultants Role

For other information contact: Brenda West, Director, ASAE Meetings & Conferences, 2950 Niles Road, St Joseph, MI 49085-9659.

Voice: +1 (616) 428-6327(Eastern Time Zone)
Fax:+1 (616) 429-3852
Internet: west@asae.org

Second International Conference on Air Pollution from Agricultural Operations

America

Sponsored by:

The American Society of Agricultural Engineers

Propose

This symposium will concentrate on air quality outside the animal facility and air quality inside and outside other facilities such as grain elevators and animal feeding lots.

Topic Areas

Dust Control
Dispersion Modeling
Odors
Ammonia Emissions
Particulate Matter (PM 10 and PM 2.5)
Air Quality Regulations

For other information contact: Brenda West, Director, ASAE Meetings & Conferences, 2950 Niles Road, St. Joseph, MI 49085-9659.

Voice:+1 (616) 428-6327(Eastern Time Zone)
Fax:+1 (616) 429-3852
Internet: west@asae.org

The 11th International Conference And Exhibition on Mechanization of Field Experiments

July 10-14,2000
London/Chelmsford, UK

Organised by

Association of Applied Biologists (AAB) International Association on Mechanization of Field Experiments (IAMFE) Writtle College, Chelmsford, UK July 10-14, 2000

Objectives of the Conference and Exhibition

a. To give the participants an opportunity to study up-to-date field and laboratory machinery, equipment and instruments at exhibition and field demonstrations

b. To present results regarding the progress of mechanization of field experiments during the past four years by means of oral and poster presentations
c. To discuss and exchange experiences of handling field trials from planning to report with reference to management systems, training of personnel as well as selection, use and maintenance of appropriate mechanization levels for different countries
d. To stimulate cooperation and coordination of efforts regarding the supplying of information on testing, construction and use of machinery, equipment and instruments specially intended to meet requirements of field and laboratory experiments
e. To promote personal contacts between agronomists, plant breeders, agricultural engineers and others who are interested in mechanization and management of field experiments
f. To hold a General Assembly of IAMFE to discuss the philosophy, policy and future world of the organization

Further information about these events can be found on the Internet Location:http://www.rase.org.uk (see conferences) or by contacting Dr. Alan Spedding, Royal Agricultural Society of England, Stoneleigh Park, Warwickshire, CV8 2LZ, UK.
Fax: +44-1203-696900
E-mail: alams@rase.org.uk

9th Workshop of the Network General Theme: Technology Transfer

September 6-9, 2000
Palazzo Feltrinelli -University of Milan Gargnano (BS)- Italy

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A brief presentation of the workshop

The FAO network is organising every two years alternatively a Conference or a Workshop.

The 8th meeting was held in Rennes as a Conference with an attendance of 150 persons.

The 9th meeting will be structured as a workshop with a fixed number of papers related to "Technology transfer"

The five foreseen sessions will deal with:

Legislation and codes of good practice
Control of water and air pollution
Compost, manure, slurry and sludge management and treatment
Spreading of compost, manure, slurry and sludge
Transfer of technologies

For each session an invited speaker will present the state of the art of the particular topic.

At the end of the Workshop recommendations will be prepared both for developed and developing countries (animal production facilities and pollution abatement) to obtain, for both situations, a sustainable and environmental friendly production system.

Information on Abstracts Submission

The abstract of the scientific papers and posters proposed must be sent to the Secretariat either by mail or be e-mail: **FAX ARE NOT ACCEPTED**

The electronic form could be in any of the currently used word processors, but to ensure the impact of the paper, please include a .rtf file.

Address: Francesco Maria Tangorra

Insituto di Ingegneria Agraria Facolta di Agraria Via Celoria, 2 20133 Milano-Italy

E-mail: ramiran.2000@unimi.it

For Updated Information Please Consult the Web Site:

http://users.unimi.it/~fsangio/fao/ramiran2000.html

Information from the International IAMFE Centre

New Internet addresses to The International IAMFE Centre

Please note that, from January, 2000 our Internet addresses have been changed. The old addresses may still work for some time but will eventually be removed, so please update your bookmarks and address books. The new addresses are:

World Wide Web: http://www.iamfe.org

IAMFE/AAB UK '2000

Please find under separate cover the Final Announcement and registration forms for IAMFE/AAB UK '2000. You will also find information on the Internet under the IAMFE and AAB WWW pages. The address to the AAB home page is http://www.hri.ac.uk/aab. The Internet versions will be continuously updated with further program details.

The deadline for proposing as well as for submitting papers/posters is 25 March for the first draft. Papers/posters will be reviewed by editors (organizers) for each main topic (conference session). Instructions to Authors of papers and posters will also be available on the IAMFE and AAB Internet pages.

IAMFE/AAB UK '2000 will be our 11th International Conference/Exhibition. Do not miss this great event. Please make your registration as soon as possible in order to assure your residential accommodation at Writtle College. As the number of rooms is limited, additional hotel accommodations will be suggested.

Members of IAMFE or AAB receive substantial discounts on registration. Also members of IAMFE branches receive some discount on the registration fee. At the exhibition, IAMFE cat. 2 and 3 members get a 25% discount. Please not that we have reduced the exhibition prices compared to the First Announcement.

As was the case at IAMFE/ FRANCE '96, we offer IAMFE branches as well as countries/regions to set up National stands at the poster area of the exhibition. These stands are free of charge under the conditions that they are used for presenting a country's or regions' research, development and extension service involving agricultural field experimentation. Examples of small instruments/tools can be displayed. Commercial manufacturers/organizations are invited to the indoor/outdoor exhibition.

Persons who will need additional financial support in order to participate in the conference, may send an application to the conference secretariat at the AAB Office. The application should contain a very short CV and a specification of costs that would need to be covered. We have very limited possibilities to give support and thus we will have to make priorities. In general, only persons from countries with a difficult financial situation can be supported. Paying IAMFE members get priority before others. Support is normally given in the following priority order: 1) Support for registration fees, 2) Support for accommodation costs at Writtle College and 3) Support for some travel costs.

The 10th General Assembly at IAMFE/AAB UK '2000

On Thursday, 13 July (during IAMFE/AAB UK' 2000) the members of IAMFE will meet at our 10th General Assembly. This will be a very important meeting and we ask our members to submit proposals or ideas on any topic that we should discuss.

One important issue is the possible adoption of new statutes for IAMFE. With the IAMFE News 1'2000, to be mailed in May, we will enclose a proposal for new statutes as well as other proposals from the Executive Committee or from our members.

Retirement of Tien-Song Peng

Taiwan

Mr. Tien-Song Peng, AMA Co-Editor retired from the director of TAMRDC. As his successor, Dr. Lu Fu-Ming was inaugurated on April 1, 2000. He is concurrently the Chairman of Agricultural Machinery Engineering Department of National Taiwan University.

More Than 16 Trillion Lire

UNACOMA

Italia

Press Conference - Rome, March 10, 2000

The UNACOMA (Italian Farm Machinery and Earth-Moving Machinery Manufacturers Association) president presents the situation and present trend of agricultural mechanization.

Present assessments of production and the market for agricultural and earth moving machinery for 1999 point to total volume of 1,385,000 tons for a value of more than 16 trillion lire. The figures, discussed by UNACOMA President Aproniano Tassinari, are slightly below the results for 1998 (1,404,000 tons), which marked an all-time high for production, but confirm the sound trend of this sector of Italian mechanics in growth since 1995 and solidly in the Front ranks at the world level for production capacity and product quality.

In detail, tractor production should reach 320,000 tons, for a downturn of 7.5% compared to '98; agricultural machinery turned out should come to 530,000 tons, in line with '98 results, and earth moving machinery should hit 535,000 tons to show growth of2.3% over the previous year.

The decline in the tractor compartment was due to a fall-off of exports resulting from crisis conditions on international markets which had repercussions on all production sectors. The drop was partially offset by the growth of the domestic market and this positive trend nationally has been confirmed by definitive data provided by the Ministry for Transport on the registration of those vehicles requiring license plates.

In 1999, 34,679 tractors were registered for a 7.8% increase over '98; combine harvesters came to 816 units for a gain of 28.7% while trailers showed an increase of 6.2%, moving from 16,117 units in'98 to 17,122 in'99. The only figure on the decline was that for multi-purpose farm vehicles, off from 3,582 units in '98 to 3,299 in '98 for a downturn of 7.9%. ■■

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By Marie-Paz Yuste, Juan Gostincar

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By Jie Shen, Radhey Lal Kushwaha

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(U.S.A.)

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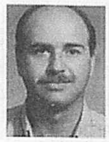
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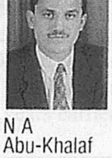
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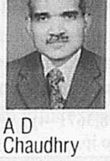
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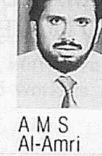
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i. Indicate by footnotes or legends any abbreviations or symbols used in tables or figures.
j. Convert national currencies in US dollars and use the later consistently.
k. Round off numbers, if possible, to one or two decimal units, e.g., 45.5 kg/ha 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.

BACK ISSUES

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AGRICULTURAL MECHANIZATION 1N ASIA, AFRICA AND LATIN AMERICA

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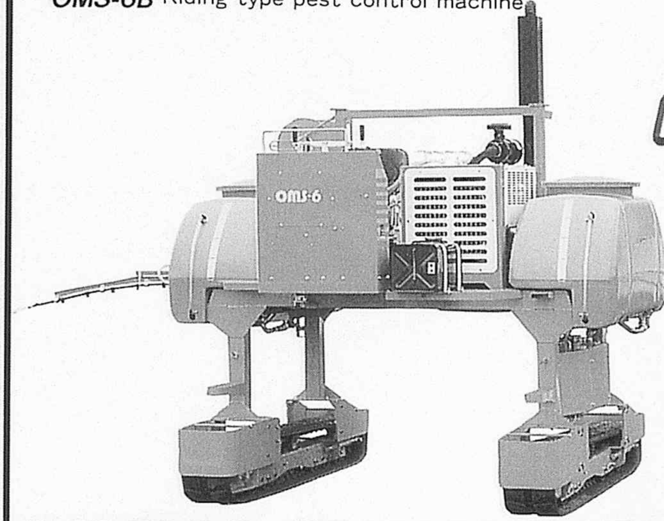
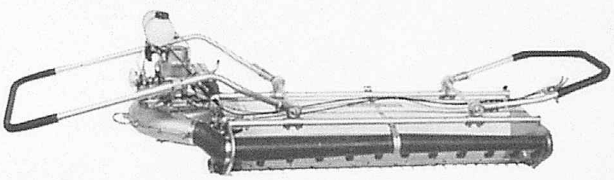
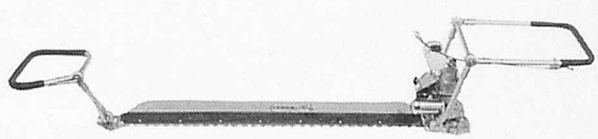
OCHIAIi is the top-ranking tea-leaf piker manufacturer in Japan. OCHIAI's products are used in tea-producing areas worldwide.

OMS-6B Riding type pest control machine





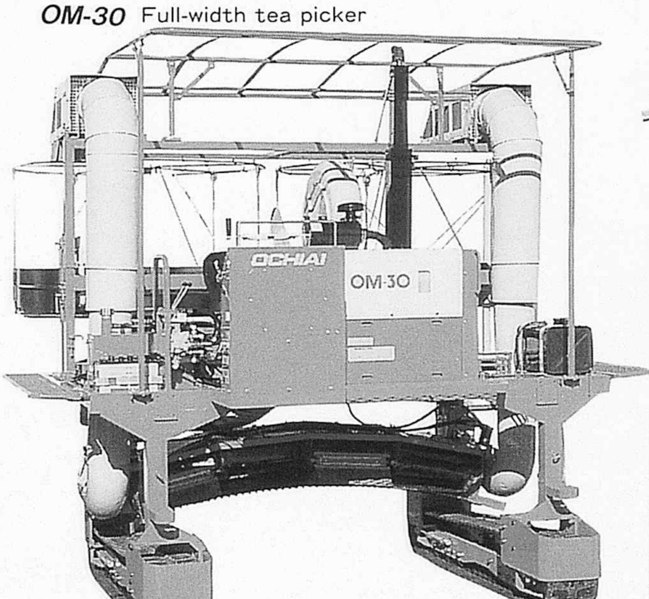
L-SIZED TEA PLUCKER *V-8*

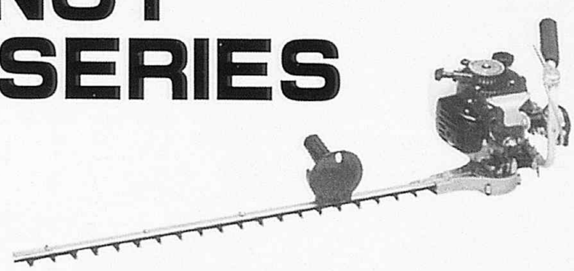


L-SIZED PRUNER *R-8*

HIGH-EFFICIENCY RIDING TYPE SERIES

OM-30 Full-width tea picker





ENGINE PRUNER *E-6*

GUIDE TO OCHIAI

- Succeeded in devising Japan's first automatic tea-leaf picker in 1959.
- Received the Director of the Board of Scientific Technology Award in 1967.
- During the intervening period (1959-1967) obtained a number of patents, as well as receiving a variety of awards and prizes in the domain of science and technology.
- The top-ranking tea-leaf picker and tea-tree trimmer producer, holding 60% of the shares in the same line of business in Japan, surpassing the other manufacturers in sales and product, and leading the related business worlds in its expansion and development.

OCHIAI HAMONO KOGYO CO LTD

OCHIAI-SHOJI CO., LTD.

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