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CONTENTS

DESIGN AND OPERATING EFFECTS OF PARAMETERS ON DRAUGHT OF MODEL CHISEL FURROWERS IN AN ARTIFICIAL SOIL

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ABSTRACT

A study was conducted to determine the effects of design and operating parameters on the draught of model chisel furrowers in an artificial soil in a laboratory soil bin. The design parameters considered were slide and nose angles while operating parameters were soil depth and tool travel speed. The effects of the interactions of depth and speed on slide and nose angles were evaluated. The influences of depth and speed on the interactions of slide and nose angles were also determined. Results showed that draught increased with increases in depth, speed, slide and nose angles. Soil depth gave draught values that were slightly higher than that of tool speed. The effect of depth on the interactions of slide and nose angles resulted in higher draught than with speed. The slight differences recorded in draught of the model tools showed that the tool design parameters were a suitable combination and the tools could perform well in a good soil if depth and speed are carefully selected.

KEYWORDS: Chisel furrower, draught, soil bin, artificial soil, slide angle, nose angle.

1. INTRODUCTION

The design of tillage tools to accomplish different jobs is a very complex engineering work. This is based on the fact that different crops require different soil preparations. Different soil conditions also require different tillage operations. Shape and size are usually the first parameters to be considered in the design of any tillage tool. The shape influences the pattern of soil movement and final soil conditions while the size determines the power required to pull the tool through the soil. Although the designer can control tool shape and size, tools cannot perform optimally without proper combination and orientation of the tool design parameters. Tillage tools of different shapes and sizes have been designed and constructed for the purpose of soil manipulation. One of the most important performance criteria in tillage tool design is the force needed to pull the tool through a given soil (Gill and Vanden-Berg, 1967). Several researchers (Bukhari et al., 1990; Harrison and Bai, 1990; Oni et al., 1992; Biswas et al., 1993; Salokhe and Pathak, 1993; Hann and Giessibl, 1998; Onwualu and Watts, 1998; Shrestha et al., 2001) have investigated the relationships between tool geometry, tool travel speed and depth of tool engagement, soil failure pattern and volume of soil upheaval. The results reported by each of these researchers varied with tool geometry, operational parameters and soil conditions at the time of the experiments.

Shirin et al. (1993) evaluated the effects of disc and working parameters on the performance of a disc plough in a clay soil in the field. Experiments were conducted at disc angles of 40°, 45° and 50°; tilt angles of 17°, 20° and 23°, and average forward speeds of 3, 4 and 5 km/h. They reported that at 17° tilt and 45° disc angles, the specific draught decreased with an increase in soil moisture content for all the ploughing speeds. Draught decreased when disc angle was increased from 40° to 45°. There was a sharp increase in draught when the disc angle was changed from 45° to 50° in all cases. Increases in disc angle increased disc penetration and the width of cut. At all disc angles and speeds, there was a marked decrease in specific draught with an increase in tilt angle.

Fielke (1996) studied the effect of the cutting edge geometry of tillage implements on tillage forces, soil failure and soil movement below the tillage depth in the field and in a laboratory soil bin. He used experimental sweeps that were standardized with 400 mm width; 32 mm lift height, 10º rake angle and 70º sweep angle. He reported that increasing the speed of operation consistently increased the draught but had little effect on the vertical force. An increase in cutting edge height increased the forward and downward movement of soil at the cutting edge, and an increasing draught and vertically

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upward forces accompanied this. The sharp cutting edge of the tillage tool minimized the draught and vertically upward forces and it also gave a minimum soil disturbance below the tillage depth. Mckyes and Maswaure (1997) evaluated the effects of the geometric parameters of flat tillage tools on their draught, cutting efficiency and loosening of a moist clay soil. They observed increases in draught with width, depth and rake angle of the tools. Alï Suhaibani and Al- Janobi (1997) investigated the effects of speed and depth on the draught of a chisel plough, an offset disk harrow, a mouldboard plough and a disk plough on a sandy loam soil in the field. The authors observed a significant increase in draught for all the implements with an increase in depth. The specific draughts of the four tillage implements were also affected significantly by speed and depth.

Mouazen and Nemenyi (1999) estimated the draught and vertical forces, soil deformation and normal pressure distribution on four geometrically similar subsoilers using the finite element method. The four subsoilers studied have a combination of a vertical shank with 15°, 23° and 31° inclined chisels, and a 75° rake angle shank with 15° inclined chisel. They reported that a subsoiler with a shank angle of 75° and a chisel angle of 15° had the least draught. Manian et al. (2000) conducted studies in a soil bin containing black clay loam and sand in order to assess the draught, vertical and lateral components of reaction in discs at different moisture levels. The variables used were disc angle (40, 44 and 48º), tilt angle (16, 20 and 24º), disc diameter (51, 56 and 61 cm) and forward speed (4, 7 and 10 km/h). Soil moisture content, speed and disc diameter affected the draught, vertical and lateral components of reaction in both soils at varying degrees. The 16º tilt angle setting resulted in lower draught and vertical reaction components as compared to the 24º and 30º settings. Disc angles set at 44º gave minimum draught than at 40º and 48º. The vertical reaction component decreased as the disc angle increased.

Natsis et al. (2002) investigated the influence of foreploughshare and disc coulter on tillage quality and tractor fuel consumption in a clay soil in the field. The treatments consisted of tillage with mouldboard plough equipped with both disk coulter and foreploughshare; tillage with mouldboard plough equipped with foreploughshare without disk coulter, and tillage with mouldboard plough but without disk coulter or foreploughshare. The authors reported that tillage with no foreploughshare or disk coulter produced the least soil draught while tillage with foreploughshare and no disk coulter gave the highest soil draught. Tillage with disk coulter reduced both soil draught and tractor fuel consumption. Although tillage with disk coulter and foreploughshare did not give the least soil draught, the best tillage quality was obtained at this treatment.

Despite the fact that several works have been done on the effects of design and operating parameters on numerous model tools, none was on model chisel furrowers. The objective of this study was therefore to determine the effects of design and operating parameters on the draught of model chisel furrowers in an artificial soil.

2. MATERIALS AND METHODS

The soil bin facility used for this experimental study was located in the Tillage and Traction laboratory in the Department of Agricultural Engineering, University of Ilorin. The soil bin was 5.6 m long, 0.44 m wide and 0.58 m deep. The transmission from a single-axle tractor was used as source of power for soil processing and tool carriage. The carriage (Fig. 1) was equipped with a rotary tiller, soil leveller, roller compactor, tool bar frame and tool bar. The details of the design and construction of the soil bin and the carriage accessories were presented by Mamman (2002). The rotary tiller was used for soil pulverization after each test run. The roller compactor was used to compact the pulverized soil in the soil bin to a uniform density. The tool bar carries the model tool and the transducer. A 3.73 kW, three-phase electric motor served as the prime mover for the carriage and the rotary tiller.

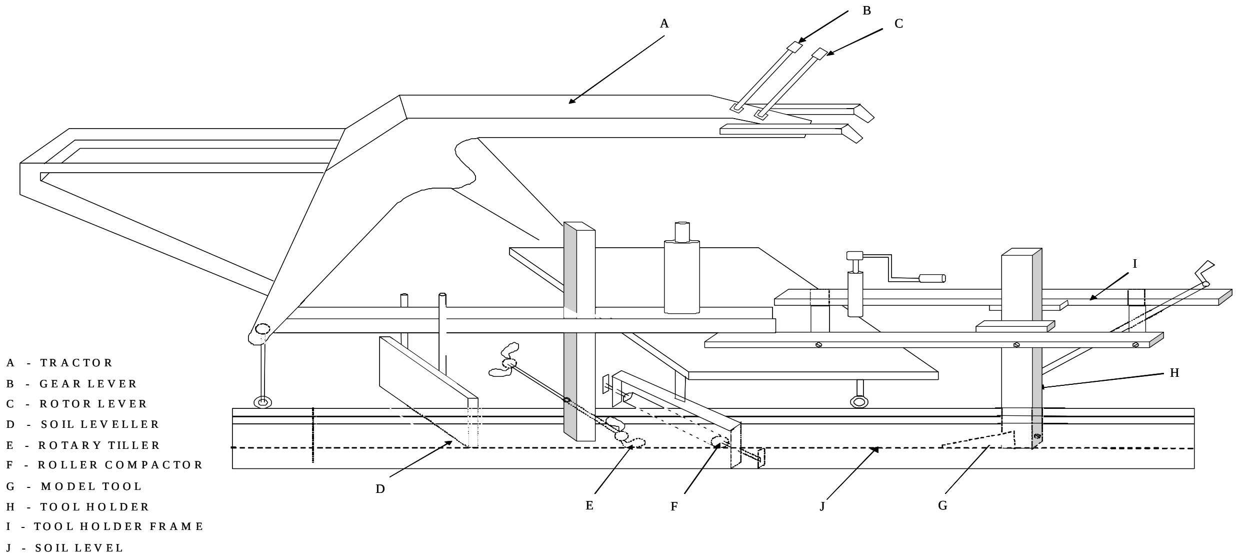

Fig.1: Schematics of a single-axle tractor showing soil processing carriage and model tool arrangements in the soil bin.

The transducer used for measuring the draught of the model tools consisted of two electrical resistance strain gages, two-140q fixed resistors and 5V power supply. One of the strain gages was mounted on one face of the tool bar in direction of travel (tension) and the other one was mounted on the opposite face of the tool bar (compression). The transducer was connected to a Wheatstone bridge. The Wheatstone bridge was connected to an amplifier, which was in turn connected to an S-T plotter. Details of the design and construction of the transducer and amplifier was presented by Mamman (2002).

The model tools were intended to create furrows in a vertisol (Typic Pellustert) for dry season sorghum (Masakwa) seedling transplanting. But because of the difficulty of getting workable moisture content, the soil was composed into an artificial soil by mixing with hydraulic transmission oil (Azolla 68 ZS). Particle size analysis of the original vertisol gave 5% sand, 35% silt and 60% clay (Ohu et al., 2001). The strength properties of the artificial soil were determined for cohesion (3.66 kPa), adhesion (3.90 kPa), soil angle of internal friction (4.17°) and soil-metal friction angle (22°).

Thirty-six geometrically similar model chisel furrowers were designed and fabricated for the study. The model tool design parameters were cutting edge height (H), slide angle (б), nose angle (ɔ), width (W) and height (h). All tools were 150 mm long and the cutting edge of each was 1 mm thick. The width and height of each model tool was proportional to its nose and slide angles, respectively. The geometric parameters of the model tools considered in this report were slide and nose angles. The slide angles used were 5, 10, 15 and 20° and the nose angles were 10, 20 and 30°. The operating parameters used for the investigation were soil depth and tool travel speed. Soil depths were 2.5, 5, 7.5 and 10 cm while tool speeds were 2, 5, 10 and 15 cm/s.

A 3x3x4x4x4 (3 levels each of cutting edge height and nose angle, and 4 levels each of slide angle, speed and depth) full factorial statistical design was used for data collection. The experiments were replicated two times. Analysis of variance of the mean values of draught was carried out using a statistical package; Genstat 5 and Least Significant Difference (LSD) tool was used to test the level of significance. The mean values of draught that were statistically significant were used for the interpretation of the results. The effects of the interaction of depth and speed on slide and nose angles and the effects of the interactions of slide and nose angles on draught as affected by depth and speed were reported.

3. RESULTS AND DISCUSSION

Figures 2-5 illustrate the response of draught of model tools to changes in slide angle, tillage depth and tool travel speed. At each level of speed of operation, draught increased with increases in slide angle and tillage depth. For all slide angles and tillage depths, draught increased with increases in tool travel speed. At a tool travel speed of 2 cm/s and tillage depth of 2.5 cm, a mean draught of 76.70 N and 117.64 N were recorded at 5º and 20º slide angles, respectively. For the same tool speed (2 cm/s) and tillage depth of 10 cm, a mean draught of 118.32 N and 156.89 N were obtained at 5º and 20º slide angles, respectively. At the highest tool speed considered in the study (15 cm/s) and at tillage depth of 2.5 cm, a mean draught of 122.83 N was recorded at 5º slide angle and 154.04 N was obtained at 20º slide angle. For the same tool speed (15 cm/s) and tillage depth of 10 cm, a mean draught of 159.34 N and 188.35 N were obtained at 5º and 20º slide angles, respectively.

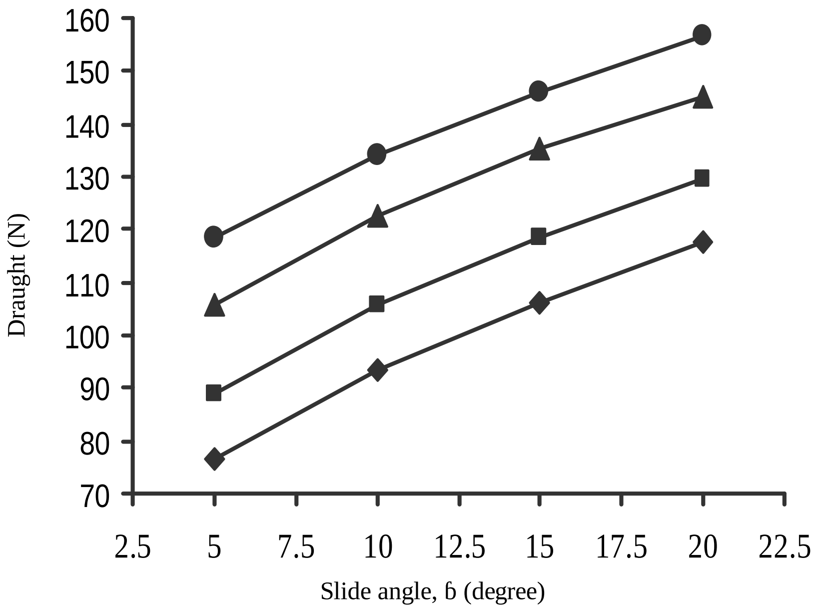

Fig. 2: Influence of slide angle and depth on draught at a speed of 2 cm/s: d=depth; Dž d_1= 2 cm; ý d_2 =5 cm; ž d_3=7.5 cm; ŏ d_4=10 cm

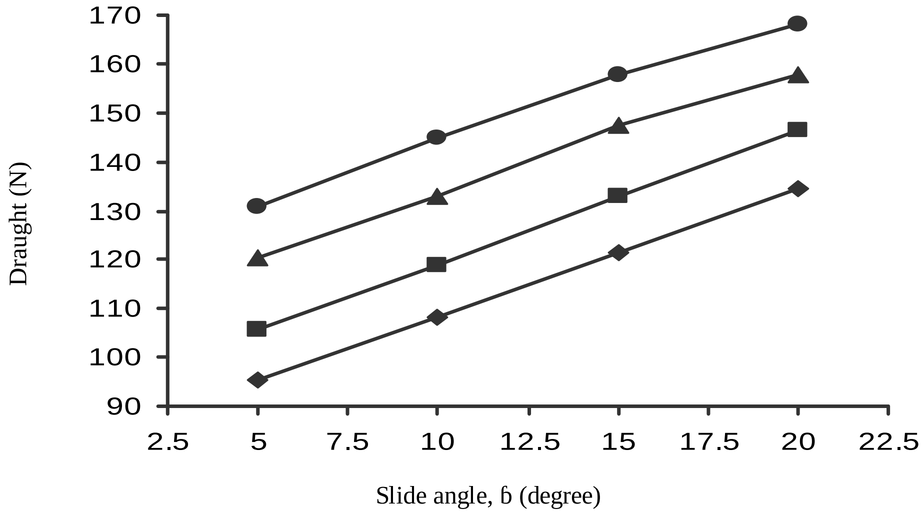

Fig. 3: Influence of slide angle and depth on draught at a speed of 5 cm/s: d=depth; Dž d_4= 2 cm; ƴ d_2 =5 cm; ƶ d_3=7.5 cm; ǒ d_4=10 cm

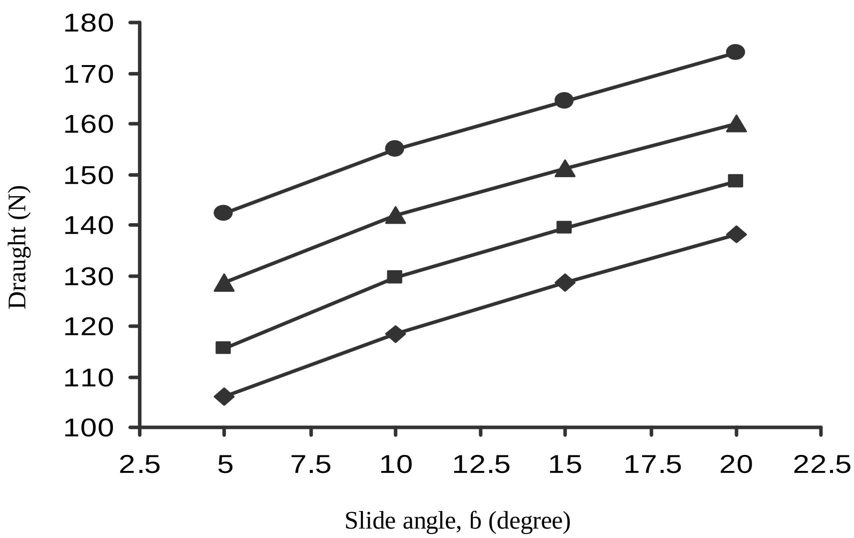

Fig. 4: Influence of slide angle and depth on draught at a speed of 10 cm/s: d=depth; Dž d_4= 2 cm; ƴ d_2 =5 cm; ƶ d_3=7.5 cm; ǒ d_4=10 cm

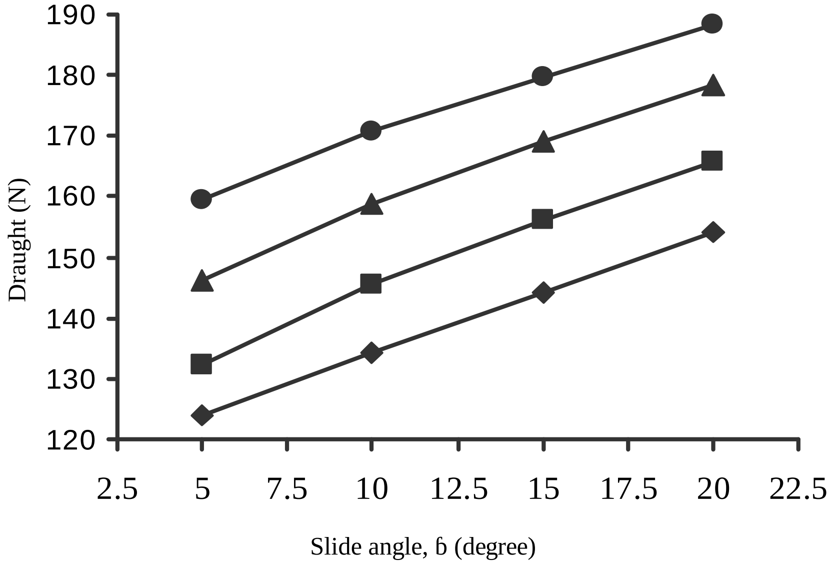

Fig. 5: Influence of slide angle and depth on draught at a speed of 15 cm/s: d=depth; Džd$_1$= 2 cm; ÿ d_2 =5 cm; ž d_3=7.5 cm; ǒ d_4=10 cm

Figures 6-9 show the effects of nose angle, tillage depth and tool travel speed on the draught of the model tools. For each level of tool travel speed, draught increased with increases in nose angle and tillage depth. For all nose angles and tillage depths, draught increased with increases in the levels of tool travel speed. At a tillage depth of 2.5 cm and tool speed of 2 cm/s, mean draught was 76.30 N at a nose angle of 10º. For the same speed of tool movement (2 cm/s) and tillage depth of 10 cm, draught increased to 120.94 N at 10º nose angle and 145.05 N at 30º nose angle. For a tool travel speed of 15 cm/s and tillage depth of 2.5 cm, draught recorded was 124.33 N and for the same speed, draught increased to 184.23 N at a soil depth of 10 cm. These mean values of draught obtained from the interactions of depth and speed on slide and nose angles showed that draughts were significantly ($P< 0.05$) affected by both the tillage depth and tool speed.

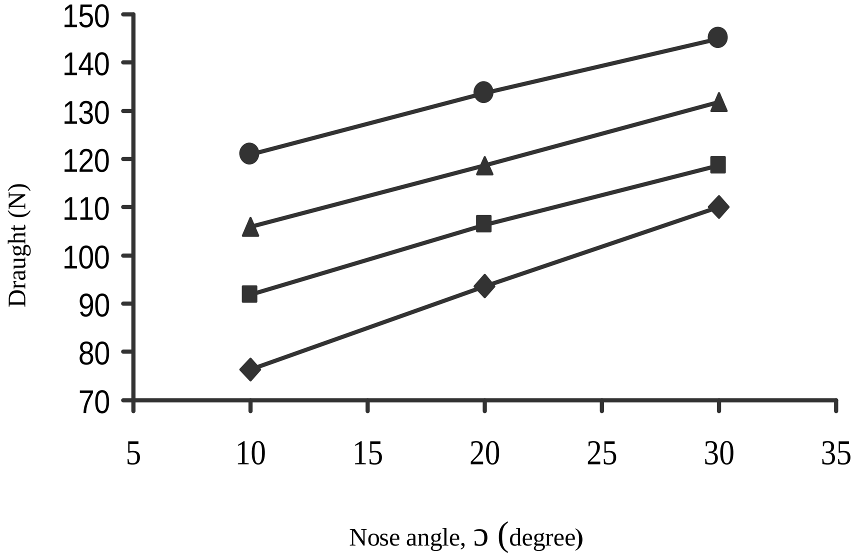

Fig. 6: Influence of nose angle and depth on draught at a speed of 2 cm/s: d=depth; Dž d_1= 2 cm;
ý d_2 =5 cm; ž d_3=7.5 cm; ǒ d_4=10 cm

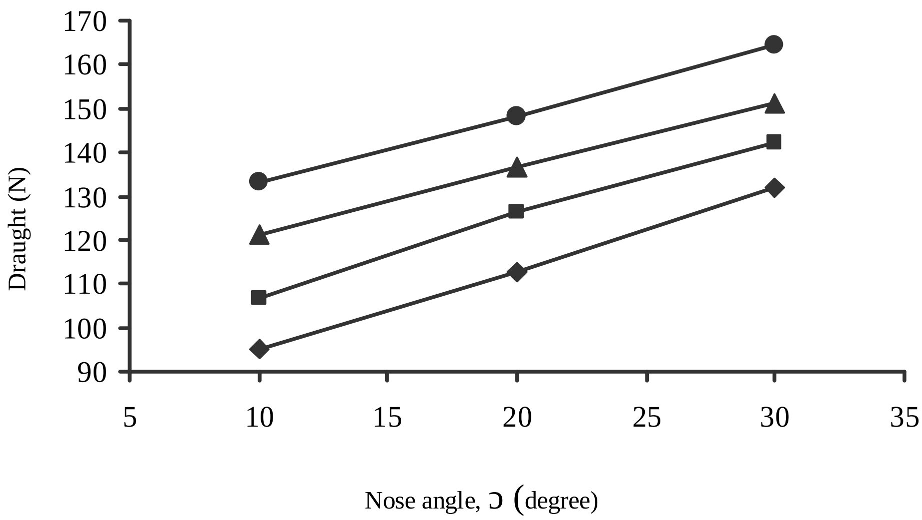

Fig. 7: Influence of nose angle and depth on draught at a speed of 5 cm/s: d=depth; Dž d_1= 2 cm;
ý d_2 =5 cm; ž d_3=7.5 cm; ǒ d_4=10 cm

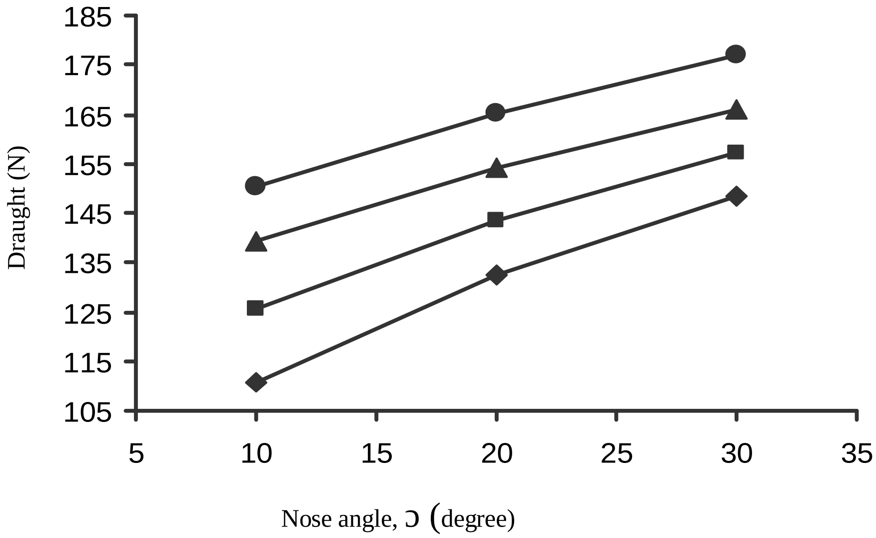

Fig. 8: Influence of nose angle and depth on draught at a speed of 10 cm/s: d=depth; Dž d_1= 2 cm; ӯ d_2 =5 cm; ʑ d_3=7.5 cm; ǒ d_4=10 cm

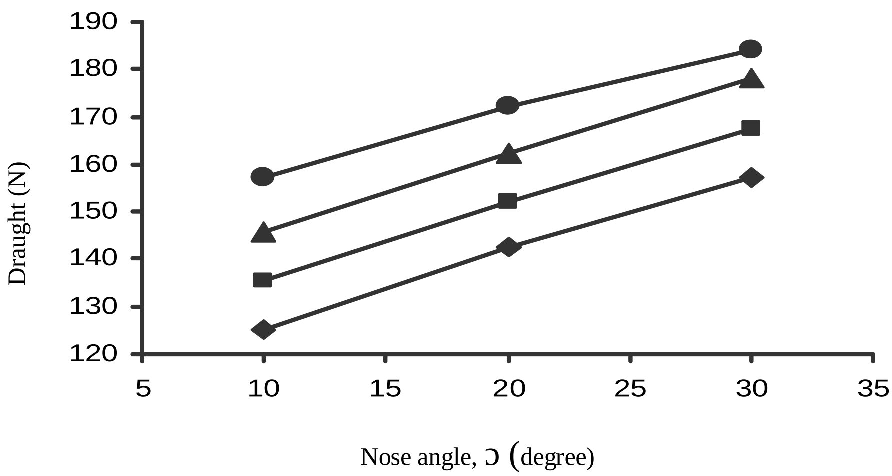

Fig. 9: Influence of nose angle and depth on draught at a speed of 15 cm/s: d=depth; Dž d_1= 2 cm; ӯ d_2 =5 cm; ʑ d_3=7.5 cm; ǒ d_4=10 cm

Figures 10-13 present the influence of tillage depth on the interactions of slide and nose angles on draught of the model tools. For all the figures the draught increased with increases in depth, slide and nose angles. At 2.5 cm depth and 5º slide angle, mean draught was 87.63 N at 10º nose angle. For the same depth (2.5 cm) and slide angle (5º) the draught increased to 104.84 N and 114.91 N at 20º and 30º nose angles, respectively. At 2.5 cm depth and 20º slide angle, the draught increased from115.63 N at 10º nose angle to149.46 N at 30º nose angle. At the highest depth (10 cm) of tillage and 5º slide

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angle, draught increased from 123.48 N at 10º nose angle to 149.93 N at 30º nose angle. For the same depth (10 cm) and 20º slide angle, draught increased from 151.39 N at 10º nose angle to 186.57 N at 30º nose angle. Figures 10 and 11 show that mean values of draught at 10º nose angles were significantly ($P < 0.05$) less than at 20º nose angles. The differences in mean values of draught between 20ºand 30º nose angles were however small. In Figures 12 and 13, the differences in mean values of draught were not significantly different among the three levels of nose angle considered.

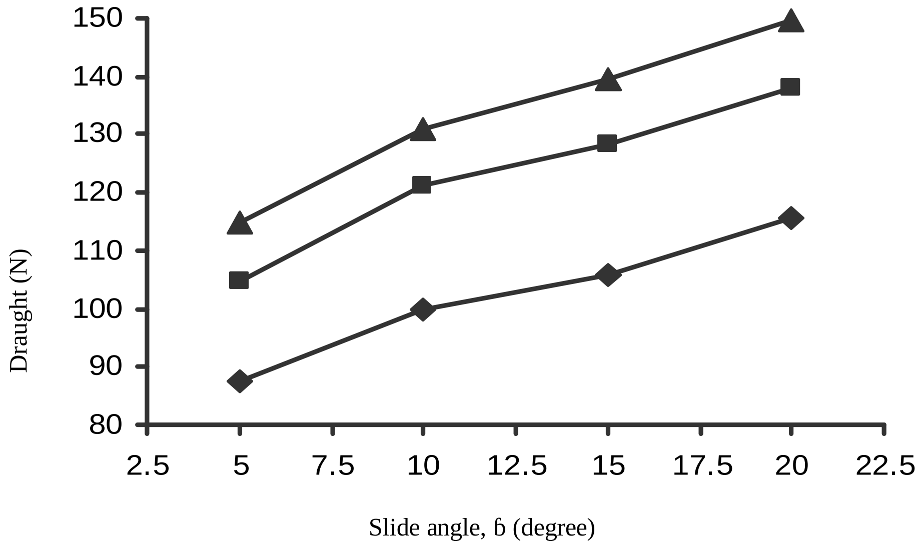

Fig. 10: Effect of the interaction of slide and nose angles on draught at a depth of 2 .5 cm: ɔ=nose angle; Dž ɔ1= 10º; ÿ ɔ2 =20º; ᴢ ɔ3= 30º.

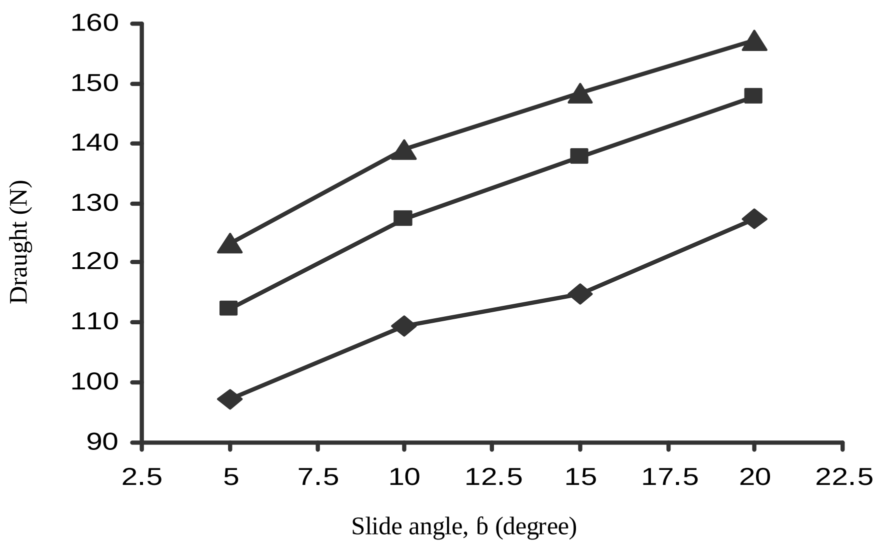

Fig.11: Effect of the interaction of slide and nose angles on draught at a depth of 5 cm: ɔ=nose angle; Dž ɔ1= 10º; ÿ ɔ2 =20º; ᴢ ɔ3= 30º.

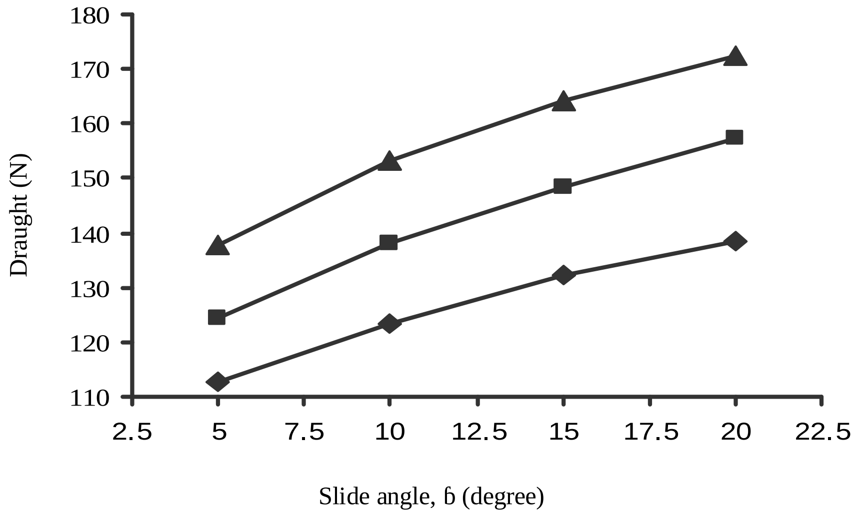

Fig. 12: Effect of the interaction of slide and nose angles on draught at a depth of 7.5 cm: ɔ=nose angle; Džɔ$_1$= 10º; ƴ ɔ$_2$ =20º; ƶ ɔ$_3$= 30º.

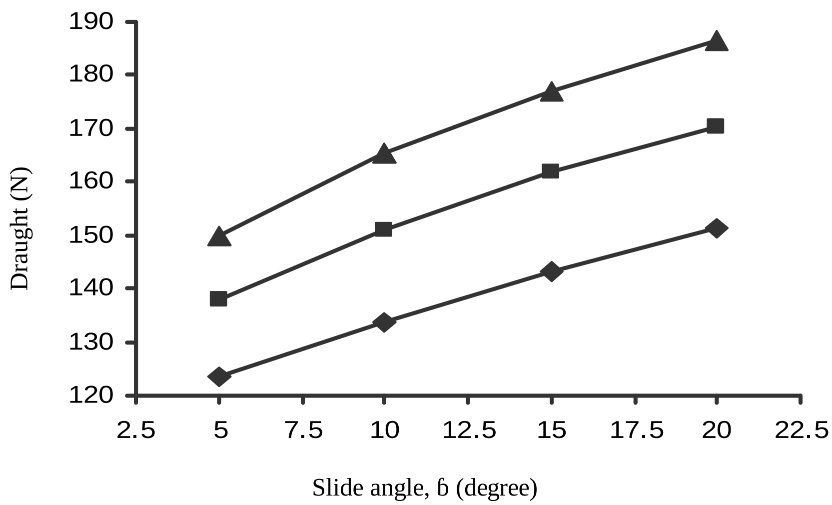

Fig.13: Effect of the interaction of slide and nose angles on draught at a depth of 10 cm: ɔ=nose angle; Džɔ$_1$= 10º; ƴ ɔ$_2$ =20º; ƶ ɔ$_3$= 30º.

Figures 14-17 give the relationships of the influence of speed on the interactions of slide and nose angles on draught of the model tools. At a speed of 2 cm/s, 5º slide angle and 10º nose angle, a mean draught of 78.7 N was recorded while the same speed and slide angle gave a mean draught of 120.43 N at 30º nose angle. For the same speed (2 cm/s) and nose angle (10º), an increase in slide angle from 5º to 20º resulted in an increase in draught from 78.7 N to 116.22N, respectively. Also, at a speed of 2 cm/s and 30º nose angle, draught increased from 120.43 N at 5º slide angle to 145.94 N at

20º slide angle. When the speed was increased to 15 cm/s, a draught of 124.42 N was recorded at 5º slide and 10º nose angles. A draught of 149.25 N was obtained when nose angle was increased to 30º at same speed (15 cm/s) and slide angle (5º). At 20º slide angle and 15 cm/s tool speed, a mean draught of 155.85 N was recorded at 10º nose angle and this increased to 179.37 N at 30º nose angle. Figures 14-17 show that mean values of draught at 10º nose angles were significantly ($P < 0.05$) less than at 20º nose angles. However, the differences in mean values of draught between 20º and 30º nose angles were small.

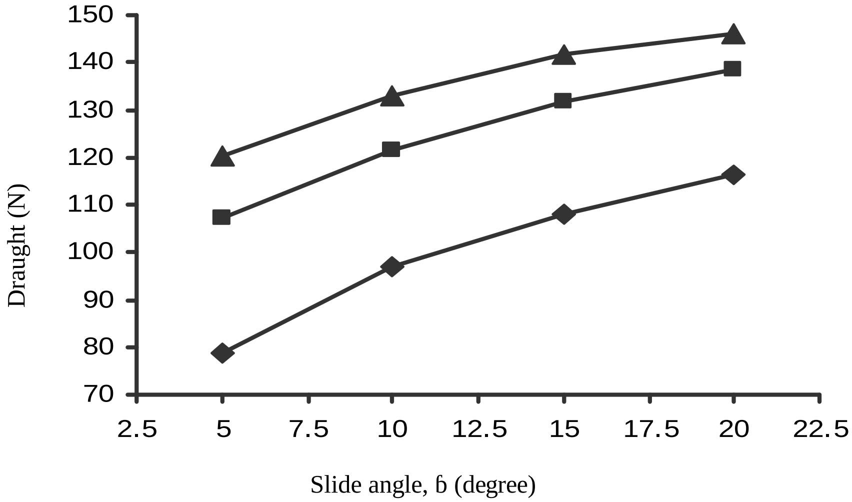

Fig.14: Effect of the interaction of slide and nose angles on draught at a speed of 2 cm/s: ɔ=nose angle; Dž$_1$= 10º; ỵ ɔ$_2$ =20º; ẓ ɔ$_3$= 30º.

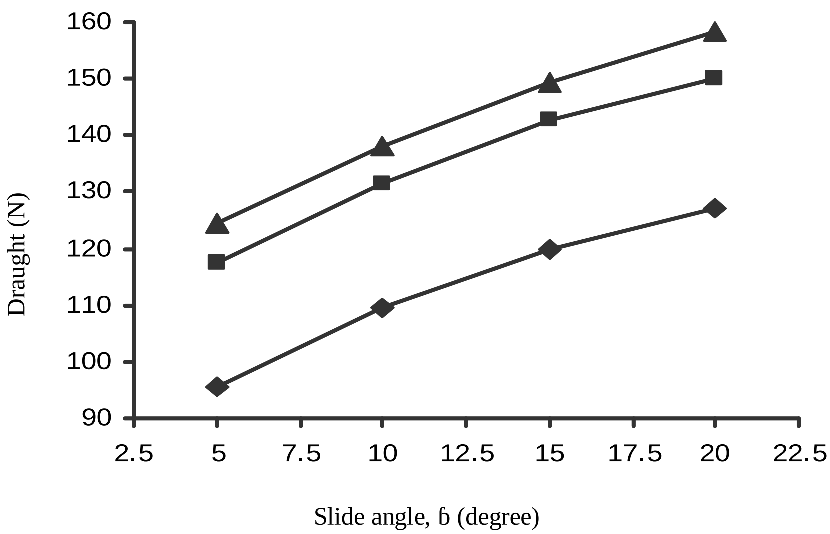

Fig.15: Effect of the interaction of slide and nose angles on draught at a speed of 5 cm/s: ɔ=nose angle; Dž$_1$= 10º; ỵ ɔ$_2$ =20º; ẓ ɔ$_3$= 30º.

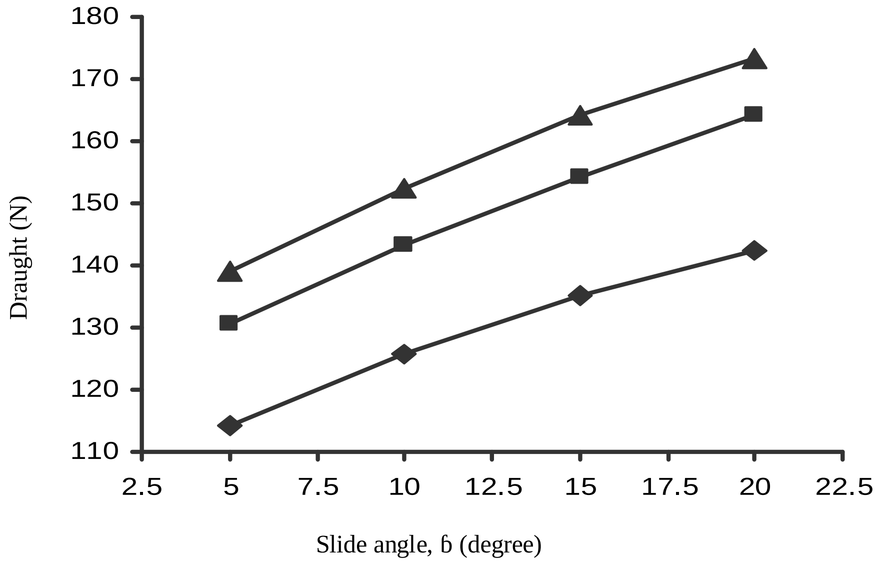

Fig.16: Effect of the interaction of slide and nose angles on draught at a speed of 10 cm/s: ɔ=nose angle; Džɔ$_1$= 10º; ƴ ɔ$_2$ =20º; ƶ ɔ$_3$= 30º.

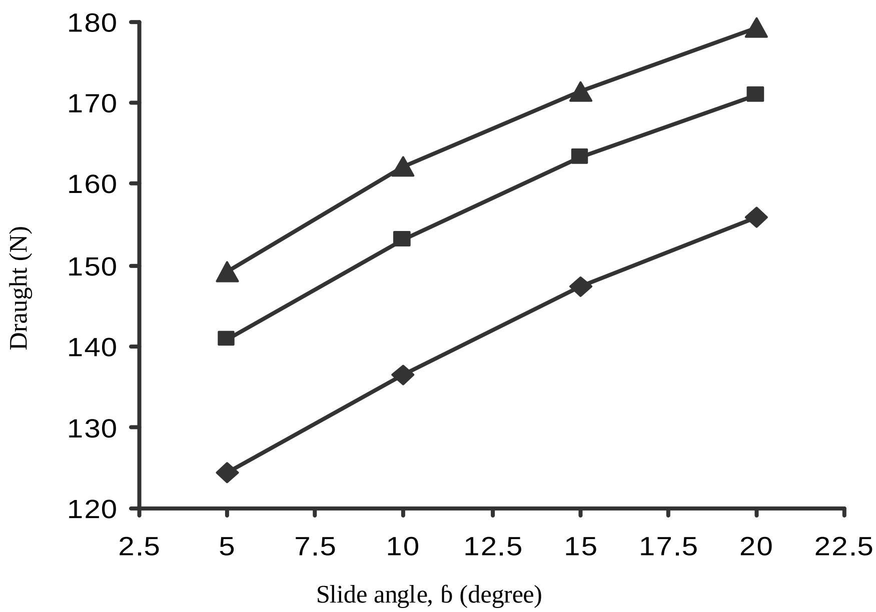

Fig.17: Effect of the interaction of slide and nose angles on draught at a speed of 15 cm/s: ɔ=nose angle; Džɔ$_1$= 10º; ƴ ɔ$_2$ =20º; ƶ ɔ$_3$= 30º.

4. CONCLUSION

Model chisel furrowers were designed and fabricated for creating furrows suitable for seedling transplanting. The performances of the model tools were examined in a vertisol composed into an artificial soil in a laboratory soil bin. The artificial soil composed had properties similar to the vertisol in the field. Tool design parameters considered were slide and nose angles and operating parameters were tillage depth and tool travel speed. The effects of the interactions of depth and speed on slide and nose angles were examined. The influences of depth and speed on the interactions of slide and nose angles were also evaluated. The combined effects of the different levels of depth and speed on the various levels of slide and nose angles increased the draught of the model tools significantly. The interactions of depth and speed on slide angle gave draught values that were a little higher than those of nose angle. The effect of depth and speed on the interactions of slide and nose angles also led to increases in draught with increase in the levels of these design and operating parameters. The effect of depth on the interactions of slide and nose angles gave draught values that were slightly higher than those obtained with effect of speed. These findings show that the tool design parameters were a good combination and with proper selection of depth and speed, the tool could consume less energy in creating furrows.

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DEVELOPMENT AND PRELIMINARY TESTING OF METERING EQUIPMENT FOR MECHANIZED YAM SETT PLANTING

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ABSTRACT

This paper reports the design and laboratory testing of equipment for metering yam setts, as a precursory step in the development of a mechanical yam sett planter. The metering equipment was designed to discharge a yam sett at every 1000 mm of within-row spacing using a ground-wheel drive system. Yam setts flow by gravity from a hopper into a metering drum having an internal helical screw and lift flaps, which discharge the yam setts for planting. Laboratory tests of the prototype gave a metering efficiency of 82.2% with percentages of singles, multiples and skips being 80.5%, 0.2% and 17.7%, respectively. The theoretical field capacity is 0.2 to 0.4 ha h^{-1} at planting speeds of 2.7 to 5.0 km h^{-1}. The work provides a basis for the future development of a mechanical yam sett planter.

KEYWORDS: Yam sett, metering, planting, equipment, design, laboratory testing.

1.1 INTRODUCTION

Yam (*Dioscorea spp*.), a tuber crop, is a major staple food in some parts of the tropics and is also known to have a long-standing economic and socio-cultural significance in the lives of some communities (Onwueme, 1982; Hahn et. al., 1987; Degras, 1993; Otoo et. al., 2001). Yams are grown principally for the carbohydrate they provide. In Nigeria, yam constitutes over 20% of daily calorific intake (Iwueke, et al., 1983) and the country accounts for over half the total world yam production (Hahn et. al., 1987; Ihekoronye and Ngoddy, 1985). The total world production of food yam is estimated at 20 ï 25 million tonnes per year.

A large acreage of land is cultivated annually for yam production but its planting and harvesting operations are largely manual. Manual planting operation is quite inefficient and very cumbersome, requiring between 350 and 600 man-days of labour per hectare (Onwueme, 1982; Nwuba and Kaul, 1987). Traditionally, the farmers cut large tubers into setts of about 100 ï 150 g, or use small whole tubers, called seed yams, as planting materials. Seed yams, which are planted, weigh between 100 to 1500 g, are 45 ï 80 mm in diameter and 150 ï 250 mm long (Odigboh and Akubuo, 1989). Unfortunately, since seed yams are whole tubers, they are often consumed as food, resulting in a scarcity of planting material.

The shortage and the associated relatively high cost of seed yams informed the development of the minisett technique (Okoli et al., 1982). Whole yam tubers are cut into small pieces (25 g or less) called minisetts and are planted to produce seed yams. However, this technique, which is only a means to produce planting material, involves specialised cultivation activities beyond the normal scope of peasant farmers. In contrast, yam setts are culturally accepted, readily available and constitute the main planting material used by peasant farmers.

Consequently, the development of mechanical yam planters, which are compatible, in particular, with the cultural practice of yam sett planting would be worthwhile. In respect of yam sett planting, Onwueme (1982) and Odigboh and Akubuo (1989) reported that the orientation of yam setts does not affect their emergence, since any part of the sett will sprout under favourable conditions. The recommended planting depth for yam setts, reported in literature, is between 80 -100 mm below the top of hand-made mounds or mechanized ridges, with a spacing of 1000 mm between rows and 1000 mm within rows (Irvine, 1969; Onwueme 1982). Traditionally, wider or narrower spacings may be used in ill-defined rows depending on the size of the planting material (Onwueme, 1982).

In general, very little is reported in the literature on the mechanization of yam production. Vandevenne (1973) reported on stanchion ridgers for mechanizing the planting of yams. Odigboh and Akubuo (1989, 1991) reported work on the design and development of automatic planters for planting minisetts and seed yams, respectively. However, in addition to the limitations of using seed yams and minisetts as planting materials, Aluko and Makanjuola (2002) pointed out the importance of developing a mechanical yam sett planter, which will satisfy traditional agronomic practices and will be more culturally acceptable to peasant farmers.

In separate investigations into the mechanization of yam planting, different researchers have reported satisfactory results using planting units that are similar in design (Aluko, 1983; Odigboh and Akubuo, 1991; Aluko and Makanjuola, 2002). In contrast, choking and/or blockage at the hopper throat on account of the size, shape and frictional characteristics of yam planting materials, particularly yam setts, have plagued earlier efforts to develop metering systems for mechanized yam planting (Akubuo et al., 1987; Akinlade and Solaja, 1997; Tomori, 2002).

Indeed, it has been pointed out (Odigboh and Akubuo, 1991) that, in the development of a yam planter, the metering mechanism represents the major work of innovation. This report describes the design of new equipment for metering yam setts from the hopper into the delivery manifold of a mechanical yam sett planter.

The objectives of this work were: to design and fabricate equipment for metering yam setts, test the metering equipment in order to evaluate its performance and predict machine theoretical field capacity.

2. MATERIALS AND METHODS

2.1 Design Principles and Considerations

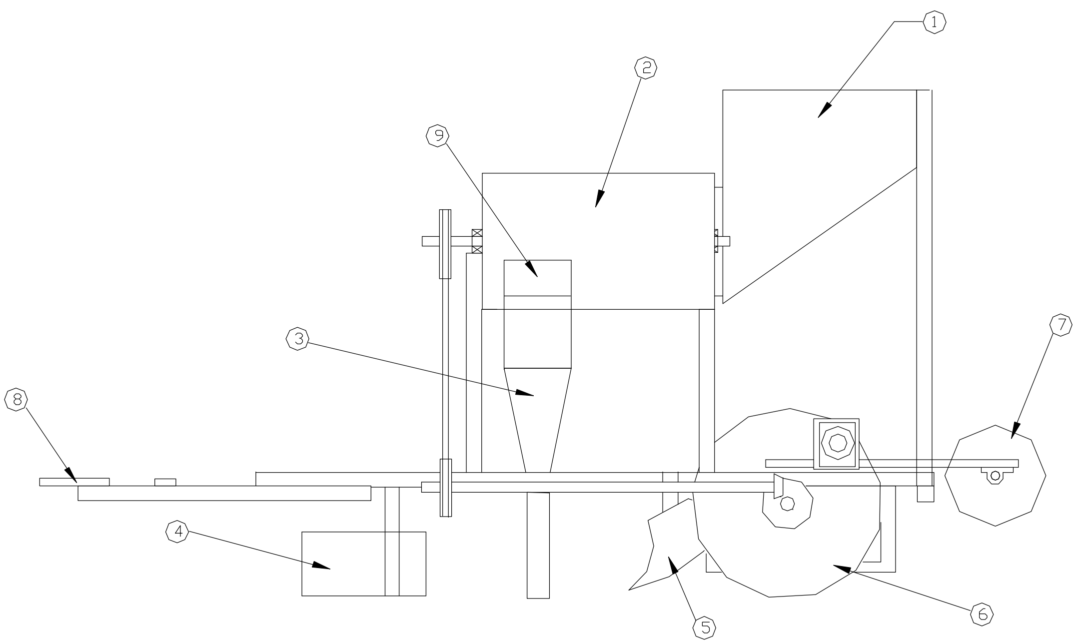
The metering equipment was conceived as a simple device for transferring yam setts from a hopper into the delivery tube of a tractor-mounted yam planter, without bruising the *periderm* of the setts. Also, the equipment should be easy to fabricate and usable by anybody, even without any previous technical training.

The metering equipment was designed using basic engineering principles. Details of materials specifications, selection and design calculations for the machine components are presented in Appendix A.

Table 1 shows the dimensions of the major components of the equipment. The envisaged planter, equipped with the metering equipment, will be hitched to a tractor as shown in Fig. 1. The metering mechanism is ground-wheel driven so that, the within-row spacing is independent of the tractor forward speed.

Table 1. Specification of major components of the metering equipment

Hopper		
Upper aperture (trapezoidal)		
parallel sides,	mm	450, 280
height,	mm	500
Base Panel (rectangular),	mm	610 x 280
Inclination of base panel,	deg.	35
Metering Drum		
Outer drum		
diameter,	mm	350
length,	mm	600
Inner drum		
diameter,	mm	280
length,	mm	450
Worm (laid on inside of inner drum)		
pitch,	mm	100
height	mm	100
Flap	mm	90 x 30



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2.2 Description of the Yam Sett Metering Equipment

The orthographic views of the equipment are shown in Fig. 2. The hopper, made from gauge 16 mild-steel sheet, has trapezoidal cross-section and its base was inclined at 35^o to the horizontal to discharge by gravity into the metering drum. Its exit is a circular chute with a semi-circular lip to guide the setts into the metering drum. The hopper was designed to hold 200 pieces of yam setts with average diameter of 80 mm and so, containing enough planting materials for 0.2 ha of farmland at a time. The hopper was bolted to the supporting frame and can be easily detached for cleaning when necessary.

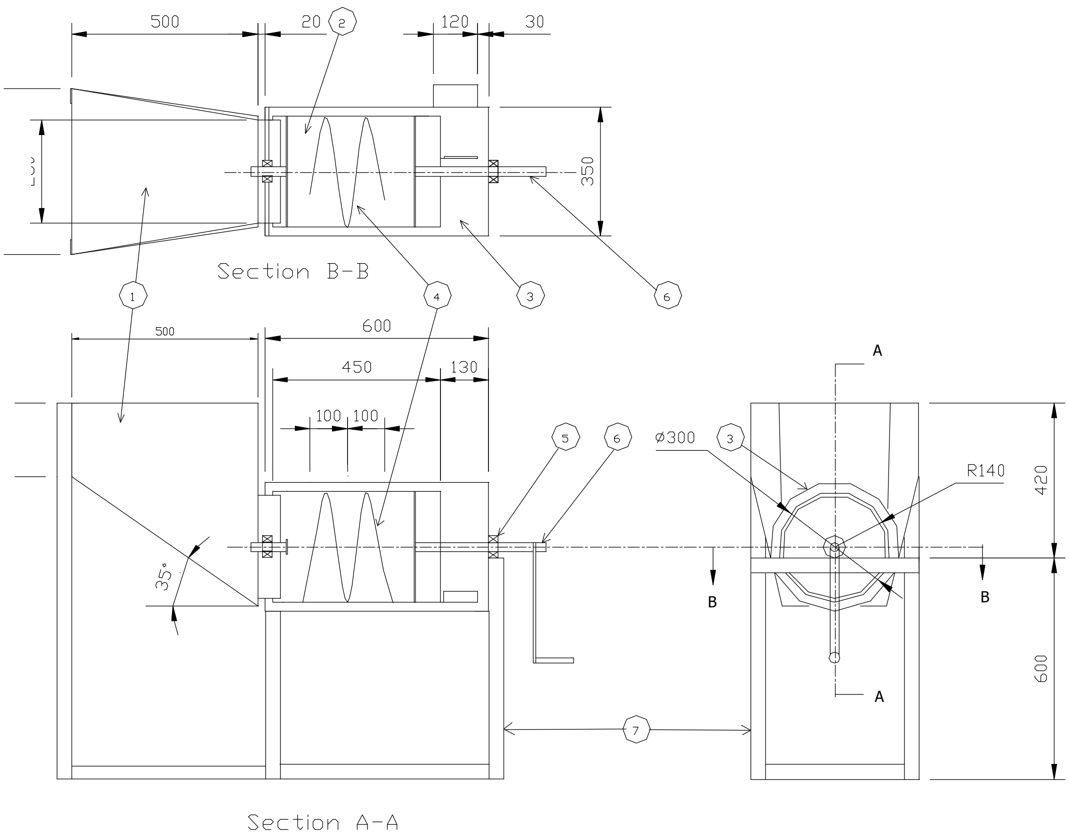

Fig. 2 Orthographic views of the yam sett metering equipment: 1, Hopper; 2, Inner metering drum; 3, Outer drum; 4, Worm; 5, Bearing; 6, Shaft; 7, Frame

The metering drum consists of two co-axial cylinders, an outer drum (ϕ350 x 600 mm long) and an inner drum (ϕ280 x 450 mm long). The outer drum was welded to the supporting frame whereas the inner drum was welded to a 25 mm shaft, which was supported on two bearings, one on a support across the exit of the hopper and the other on the supporting frame. The inner drum was equipped with internal helical screw (100 mm pitch and 100 mm high) running through its length. The shortfall in the length of inner drum compared with the outer drum provided for clearance against the hopper's lip at one end, and the other end provided room for a metal flap, attached to the inner drum. The flap sweeps metered setts, picking one sett at a time until it drops out, through a slot in the outer drum, into the delivery manifold of the planter.

The frame was constructed from 40 x 40 x 4 mm angle iron. Its overall dimension was about 1100 mm long, 450 mm wide and 820 mm high. A handle was fixed on the shaft, to drive the equipment during laboratory tests. A photograph of the prototype showing its two major parts, namely the hopper and the metering drum, is presented in Fig. 3.

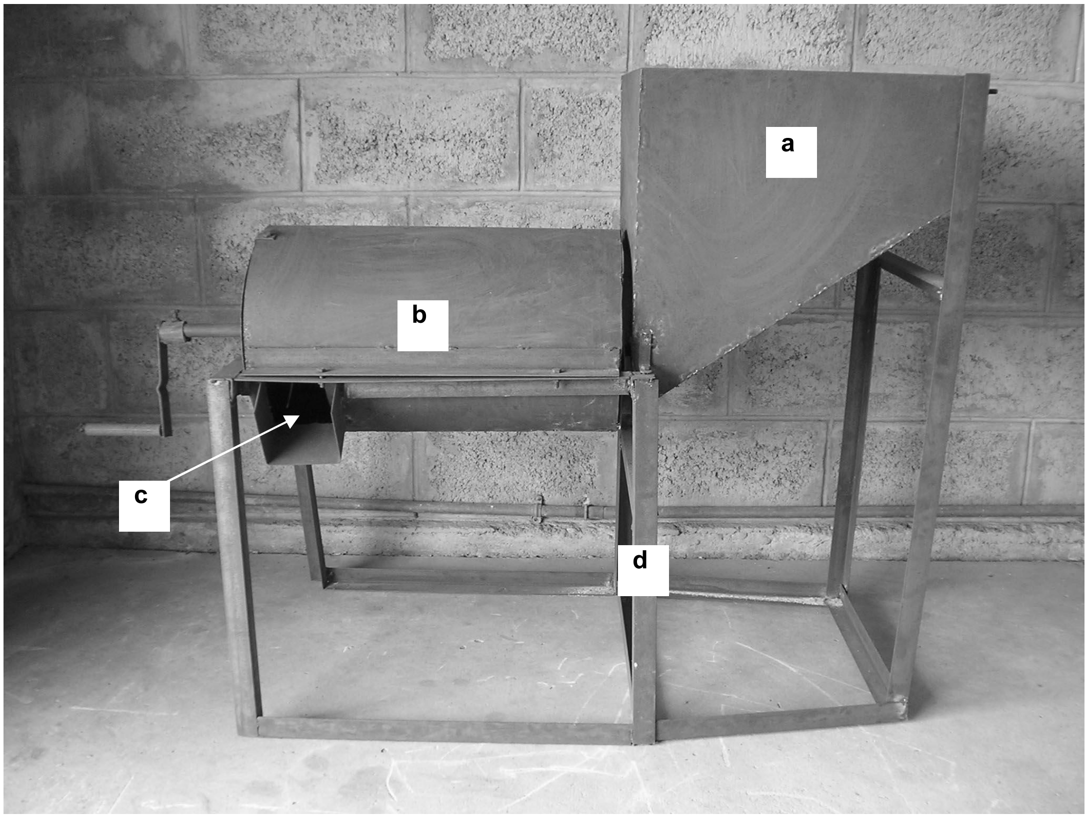

Fig. 3. Photograph of the yam sett metering equipment: a, hopper; b, metering drum; c, sett ejection slot; d, frame

2.3 Experimental Procedure for Laboratory Testing

Yam tubers were obtained from local farmers in Ile-Ife, Nigeria, and were cut into setts with average diameter of about 80 mm. The setts were then stored in a dry, well-aerated environment for two days, as is the practice in traditional yam planting (Onwuneme, 1982). This also helped in reducing the possibility of the setts adhering to the panels of the hopper (Ajav, 1998; Aluko and Koya, in press).

The hopper was filled with yam setts and the metering drum was driven at approximately 45 rev/min. For each experimental run, which consisted of eight revolutions of the metering drum, the number of setts discharged was noted. The actual number of setts delivered was counted and compared with the number expected based on design.

Equipment performance was then quantified in terms of its metering efficiency (ε_m) and the corresponding percentages of skips, singles and multiples (Kepner et. al., 1987). The metering efficiency is an indication of how many stands on the field, will actually receive setts for planting, and was defined as,

$$\varepsilon_m = \frac{N_d}{N_T} x100\% \qquad --- \qquad --- \qquad (1)$$

where, N_d is the number of times setts (single or multiple) were delivered; and N_T is the number of times sett delivery was expected, for the same rotations.

Metering efficiency was considered to be a better performance index than the percentage cell-fill, often specified for grain drills (Kepner et. al., 1987). In percentage cell-fill, empty cells are offset by extra seeds in multiple fills and, not necessarily that every cell contains a single seed. Percentage of skips is the ratio of times with no sett delivered to the number of times sett delivery was expected. Percentages of singles and multiples are the number of times single sett and multiple sett delivery, respectively, were observed, compared with the number of times setts should be delivered. Furthermore, the discharged setts were examined physically to see if the *periderm* was bruised.

The theoretical field capacity of the prototype was estimated, for a two-row planter equipped with the metering equipment at recommended speeds of 2.7 to 5.0 km h^{-1} (Odigboh and Akubuo, 1989; Aluko and Makanjuola, 2002). The estimate was adjusted incorporating the metering efficiency as follows:

$$C = \frac{sw}{10} x \varepsilon_m \qquad \text{---} \qquad \text{---} \qquad (2)$$

where, C is the theoretical capacity in ha h^{-1}; S is the speed of travel in km h^{-1}; and W is the width of implement in m.

3. RESULTS AND DISCUSSION

The results of the tests carried out to evaluate the performance of the metering equipment are shown in Table 2. The prototype metering equipment has a hopper holding capacity of 37.5 kg. A metering efficiency of 82.2% was obtained and the percentages of singles, multiples and skips were 80.5%, 0.2% and 17.7%, respectively. As pointed out earlier, the size, shape and frictional characteristics of yam planting materials pose major challenges in the development of metering equipment. Odigboh and Akubuo (1991) reported a metering efficiency of about 98% for a minisett yam planter. Compared to yam setts (100 ï 150 g) and seed yams (100 ï 1500 g), however, minisetts (about 25 g or less) constitute much smaller planting material. The metering efficiency of the seed yam planter earlier developed by Odigboh and Akubuo (1989) was not specified, thus precluding a comparison with the present results.

Table 2. Performance of the prototype metering equipment

Performance indices		Value ± standard error
Hopper, capacity,	kg	37.5
Metering efficiency,	%	82.2 ± 4.5
Percentage of singles,	%	80.5 ± 4.1
Percentage of multiples,	%	0.2 ± 0.8
Percentage of skips,	%	17.7 ± 4.1

The percentage of multiples though low, implies that in some instances, two, rather than one sett, were dropped but planting (sett delivery) actually took place. For subsequent field tests of the envisaged planter, this anomaly can be alleviated by incorporating a stationary brush to knock out the excess planting material when the flap lifts more than one sett. During the tests, the sharp leading edge of the worm occasionally scratched a few of the setts. It is therefore necessary to shield the sharp leading edge of the worm with a softer material like rubber to reduce the occurrence of bruising the setts.

The percentage of skips obtained in the present results (17.7%) needs to be reduced/eliminated to justify the use of a mechanical planter to replace traditional manual planting. The present results suggest that there is a need to increase the pitch of the worm and the clearance between the inner and the outer drums, to facilitate the worm picking setts without any failure. The required adjustments indicate that yam setts for planting would need to be sorted into size groups prior to loading them into the hopper. This modification will also enhance a satisfactory performance of the metering equipment at different tractor field speeds.

For a within-row spacing of 1000 mm and a planting speed range of 2.7 to 5.0 km h^{-1}, the estimated field capacity ranged from 0.2 to 0.4 ha h^{-1}.

4. CONCLUSION

Equipment for metering traditional yam setts for mechanical planting has been developed. The preliminary tests carried out in the laboratory on the prototype indicate a fairly satisfactory performance. It should be possible to improve the performance of the equipment for field applications, especially with respect to hopper capacity and reducing the percentage of skips, with further modifications and testing.

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NOTATION

ε_m	-	metering efficiency
N_d	-	number of times setts (single or multiple) were delivered
N_T	-	number of times setts delivery were expected based on design
C	-	theoretical field capacity, ha h^{-1}
S	-	speed of travel, km h^{-1}
W	-	width of implement or, between row spacing for a two-row planter, m

APPENDIX A
Design of Major Components of the Yam Set Metering Equipment

A. 1 The hopper

The hopper was made from guage 16 mild steel sheet, formed and welded to hold 200 pieces of yam setts, 80 mm average diameter. The following limiting values of properties of fresh yam setts, reported by Aluko and Koya (in press) were used:

Yam bulk density, $\rho = 1120$ kg m^{-3}
Maximum coefficient of friction, $\mu = 0.5$

Hence, for free flow, the following relationship between the angle of inclination θ and the coefficient of friction prevails:

$$\tan\theta \geq \mu \qquad \text{(A.1)}$$

Therefore, $\theta \geq \tan^{-1}(0.5)$ or 27^o

The design angle was chosen to be 35^o, about twenty - percent above the calculated value.

A. 2 The Internal Helical Screw

The internal helical screw (worm), welded inside the inner drum, was made from gauge 18 mild steel sheet. This thickness was selected to make fabrication easy. The diameter *D* to form a helix of certain diameter d and pitch length *p* can be determined from:
a

$$(\pi D)^2 = (\pi d)^2 + p^2 \qquad \text{(A.2)}$$

This expression was used to calculate the internal and external diameters of circular plates used in forming the helices.

A. 3 The Power Transmission Elements

The basic transmission elements in the design are the shaft and the supporting bearings, and the turning handle. The shaft was designed based on the following assumptions: shaft is solid, and made from low-carbon steel (AISI 1020); shaft length, from bearing to bearing, is 650 mm; density of shaft material is 7860 kg m^{-3}; yield strength of shaft material is 295 MPa; factor of safety, n_s is 2.5.

Shaft was designed on account of rigidity and strength, based on distortion energy theory (DET), which predicts the smallest diameter where failure will first start to occur. According to Hamrock et al. (1999),

$$d = \left[\frac{32 n_s}{\pi s_y} \sqrt{M^2 + \frac{3}{4} T^2} \right]^{1/3} \qquad - \quad - \quad - \quad (A.3)$$

Based on calculations and available material, a conservative 25.4-mm diameter shaft was chosen. Appropriate SKF deep groove ball bearings (6005) suitable for the shaft were selected.

The handle was made from a 20 x 3 mm flat bar, L ï shaped, with a hollow pipe at the free end to engage the shaft, the other bent end provided handle for turning.

AN EXPERIMENTAL MECHANICAL CASSAVA TUBER PEELING MACHINE

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ABSTRACT

Previous cassava peeling research efforts were directed mainly towards developing abrasive type batch peelers. In peeling with most of these peelers, however, big roots remain largely unpeeled while small ones may be completely grated. In this work, an experimental mechanical cassava peeling machine that works on the principle of peel-flesh separation by compression and peel removal with knives was designed and fabricated. It comprised of a horizontal belt conveyor unit and a spring-loaded knivesô unit. In operation, the belt conveyor unit conveys slices of cassava tubers which are peeled under an array of spring-loaded knives. Preliminary tests of the machine were carried out on root slices of different sizes with the conveyor belt driven at a linear speed of 1.31 m/s. For root slices having mid-section diameters ranging between 45 and 55 mm, the peel removal efficiency ranged between 0.920 and 0.973 and had an average value of 0.939. For slices having mid-section diameters ranging between 55 and 65 mm, the peel removal efficiency ranged between 0.953 and 1.000 and had an average value of 0.988. For tubers in the above size ranges, no breakage of roots occurred during peeling. For root slices with relatively larger diameters, however, the peel removal efficiency ranged between 0.152 and 0.820 and root breakage was observed to occur in some instances. In all cases, however, there was no loss of useful tuber flesh.

KEYWORDS: Design; cassava root tuber; peeling machine; preliminary tests; peeling efficiency

1. INTRODUCTION

Cassava (*Manihot esculenta* Crantz) is one of the most common economic and productive root crop in the tropics. Its root tuber is the main economically useful part of the plant. Cassava possesses vast food and industrial potentials. It is arguably the largest single crop of the tropics, particularly of that part of Africa lying south of the Sahara and north of the Zambezi, where it has become the most important crop in terms of both the total land area devoted to its production and the proportion it contributes to the human diet (Odigboh, 1983).

Before the cassava tuber is processed into any of its food and some of its non-food products, however, it must be peeled. Peeling is simply the removal of the cassava peel (made up of the outer corky periderm and cortex) from the rest of the tuber. Ideally and especially in the food industry, the peel needs to be completely removed without removing the useful tuber flesh.

The problem encountered in peeling cassava root tubers arises from the fact that cassava roots exhibit appreciable differences in weight, size, shape, peel thickness, peel texture and strength of adhesion of peel to the root flesh. Furthermore, these properties vary with the month of the year during which the root is harvested and the time that elapses after harvesting, before peeling is done. The ease of peeling varies with these properties. The greatest challenge in the design of a cassava peeler is that of making the machine capable of efficiently peeling roots from all sources all the year round despite the wide differences in these properties. As a result, the development of a technically or economically acceptable cassava tuber peeling machine has continued to pose a challenging engineering problem. Existing machines tend to reduce the cassava tuber to a regular cylinder, thus wasting an intolerably high proportion of the useful cassava flesh before all the peel is removed.

Researchers have experimented with various types of abrasive drums for batch peeling of cassava roots. Suleman (1980) described the vertical drum type; Akinrele et al (1971) described the concrete mixer shape type; Ezekwe (1979) and Nwokedi (1984) in separate investigations reported the development of horizontally-mounted drum peelers that are similar in principle; Odigboh (1988) reported the eccentrically mounted drum type. Though these machines are simple to fabricate and operate, it is generally difficult for the abrasive surfaces of the drums to enter into the crevices and depressions on tuber surfaces. A tuber may be reduced to a cylinder of uniform radius with much wastage of useful flesh before satisfactory peeling is done. Otherwise, hand trimming is required to finish incompletely peeled roots. Although Ezekwe (1979) and Odigboh (1988) reported some improvement by using a technique in which abrasive balls are mixed up with the cassava roots before being loaded into the drums, the balls easily become blunt and ineffective after a few peeling batches.

Odigboh (1976) reported a cylindrical knife cassava root peeler, which basically consists of a cylindrical knife assembly mounted parallel to a solid cylinder with roughened surface. Both were driven clockwise at different speeds. The cassava roots to be peeled were cut into 10cm lengths and fed into the space between the knife assembly and the solid cylinder. Although a peel removal efficiency of 75% and average throughput of 165 kg/h were reported, hand trimming was still required to complete the peeling.

Ohwovoriole et al. (1988) reported a simple test rig for peeling cassava tuber slices manually (Figs. 1(a) and (b)). The rig consists of evenly spaced cutting edges mounted on two

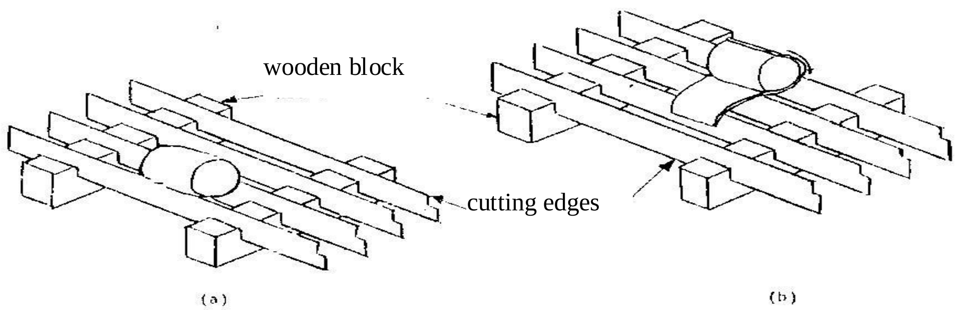

Fig. 1. Test rig for peeling cassava tuber slices manually (after Ohwovoriole et al., 1988).

wooden blocks. The cassava tuber slice resting on it is peeled manually by pushing down on it with the hand; the cutting edges penetrate through the peel into the cambium and, as the slice is rolled forward while the downward pressure is maintained, the peel simply unwinds from the flesh as shown in Fig. 1(b). Based on the principle illustrated by this rig, Ohwovoriole et al. (1988) developed a manual cassava tuber peel lacerating machine. The machine consists essentially of a fixed outer frustum of a cone, which was made up of rigid longitudinal cutting edges, and a rotating wooden inner cylinder. Although a peel removal efficiency of 92% was reported after a limited test run, the machine did not provide for the automatic adjustment of the size of the space between the inner cylinder and the outer cutting edges, which is required to accommodate varying sizes of root slices. A root slice would therefore be crushed, rather than peeled, if it gets into a space where the distance of separation between the outer cutting edges and the wooden inner cylinder is much less than the diameter of the root slice. The principle employed is however quite promising for there was reportedly no loss of useful tuber flesh during peeling.

The objective of this work was to design and construct an experimental cassava root tuber peeling rig on which experiments could be run to determine the optimum design and operating characteristics of a machine that would be developed using the principle reported by Ohwovoriole et al. (1988).

2. MATERIALS AND METHODS

2.1 Design Principles and Considerations

The concept used to develop the present mechanical peeler is illustrated in Fig. 2 and is based on the peeling principle employed by the manual test rig shown in Figs. 1(a) and (b). In Fig. 2, the human hand is replaced by a pressure application platform for applying pressure to, and rolling the tuber over an array of cutting knife edges. The questions that arise in the attempt to employ this peeling concept and which the experimental peeler described in this work has been designed to address are:

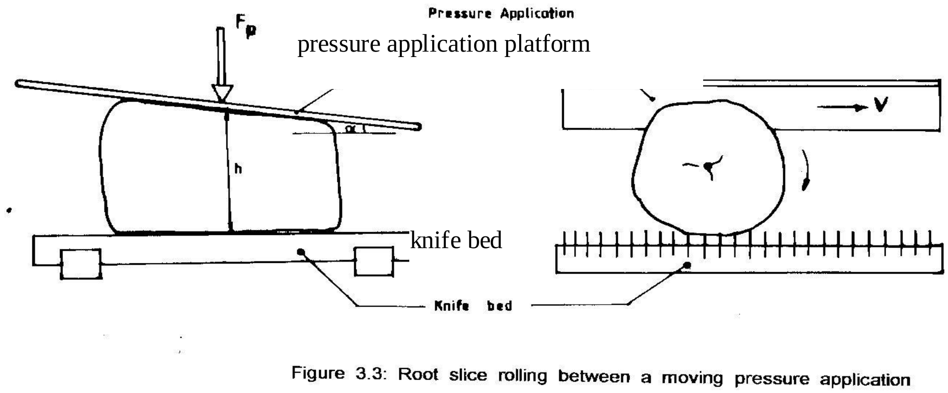

Fig. 2. Root slice rolling between a moving pressure application platform and a bed of knives.

(i) what must be the average distance between the pressure application platform and the cutting edges for a given diameter of roots such that good peeling would be effected?

(ii) at what speed must the pressure application platform move relative to the cutting edges as it rolls the tuber along without slippage?

(iii) what magnitude of the force needs to be applied to the root slice to effect peeling and yet avoid crushing the slice?

2.2 Description of the Experimental Peeling Machine

A photograph of the experimental peeling machine is shown in Fig. 3. Three orthographic views of the machine are also shown in Fig. 4. The machine comprised two units: an upper spring-loaded knivesôunit and a lower horizontal belt conveyor unit.

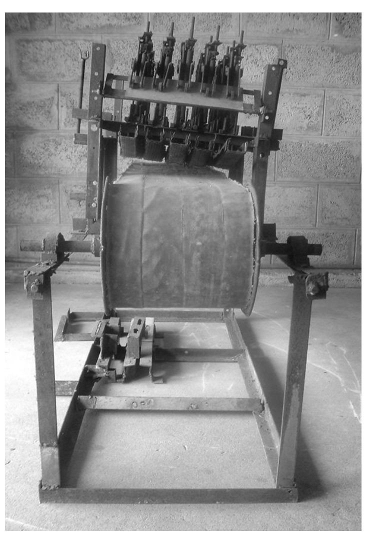

Fig. 3: The experimental peeling machine

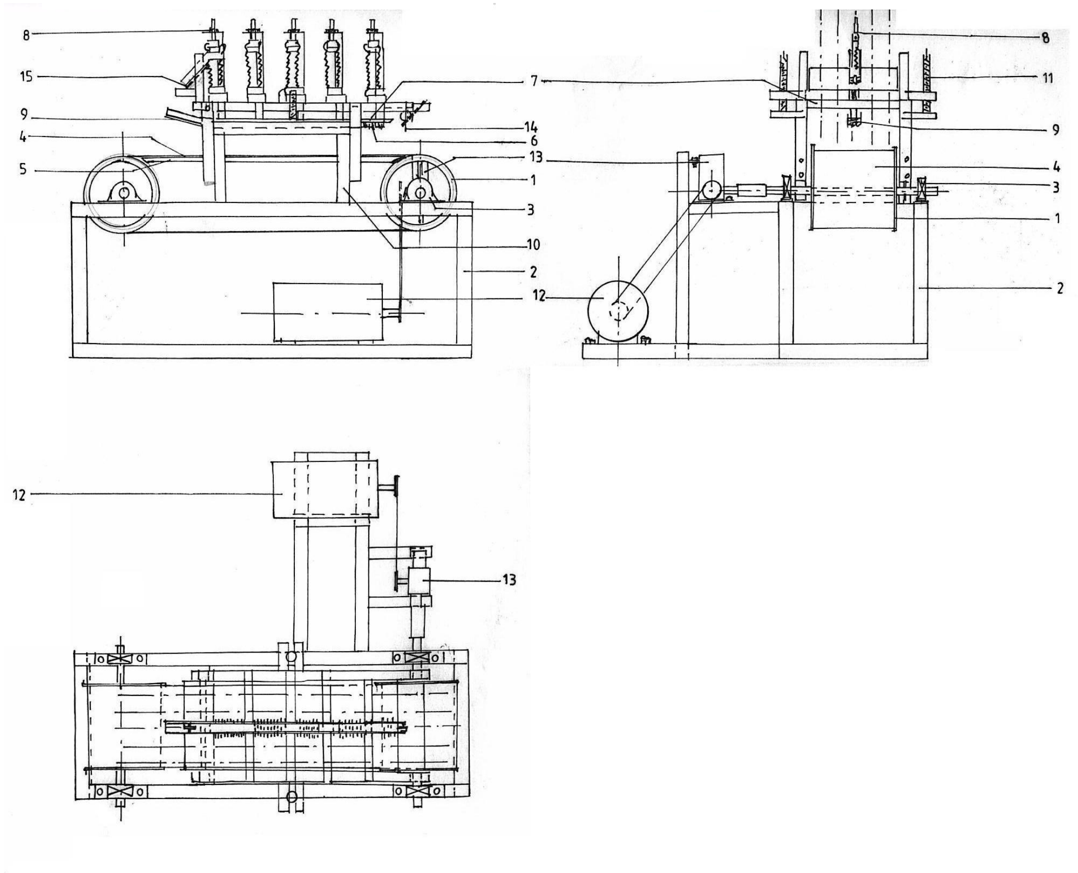

Fig. 4. Orthographic views of the experimental cassava peeling machine in first angle: 1, Conveyor drum; 2, Main frame; 3, Bearing; 4, Conveyor belt; 5, Conveyor belt platform; 6, Peeling blades; 7, Knife assembly bearing frame; 8, Spring load adjuster nut; 9, Root pick-up; 10, Main frame link arm; 11, Knife bed height adjustment screw; 12, Variable speed electric motor; 13, Speed reduction gear box; 14, Rear roller motion guide plane; 15, Front roller motion guide plane.

In the belt conveyor unit, the drums (1) are mounted on a rectangular main frame of angle iron (2) via bearings (3). The centres of the drums are spaced 1050 mm apart and each of the drums measures 250 mm in diameter and 300 mm in length. The belt (4) of the conveyor system is 300 mm wide. This belt was designed to move over a flat aluminum sheet platform (5) lying on top of a frame of angle iron. The platform was rigidly fixed to the frame of angle iron using bolts and nuts. The belt conveyor was driven by a 1.5 kW variable speed electric motor (12) through a speed reduction gear box (13) and a chain drive.

The knife assembly bearing frame of the upper spring-loaded knivesô unit is 800 mm long and 400 mm wide. The unit comprises of five assemblies of peel penetrating knives, arranged across the central 290 mm of the 400 mm width of the rectangular frame. Each assembly is 600 mm long and 50mm wide. The knife assemblies are spaced 10mm apart across their bearing frame. A root pick-up (9) inclined at 15^0 to the surface of the conveyor belt is fixed to the frontal end of each knife assembly. It is wrapped around with a sheet of expanded metal to enhance its grip on the root.

Five spring-loading points, equally spaced at 140 mm intervals, were located along the length of each knife assembly. A set of two 140 mm long helical tension springs, each of which has a spring rate of 2.193 N/mm, was located at each spring-loading point. Thus, there were a total of 25 such pairs of springs in the spring-loaded knives unit. This design of the spring-loading system was based on an earlier work by Adetan et al. (2003) in which the peel penetration force per unit length of knife edge was determined. The tensile load in these springs can be adjusted by turning the spring load adjuster nuts (8). It is this tensile load in the springs that determines the downward force on the knife assembly and, therefore, the force with which the knives attempt to penetrate into the peel of the cassava roots being conveyed under them. Each knife assembly carries 120 knives that are spaced 5 mm apart. Each knife is about 1.5 mm thick, 50 mm long and 15 mm wide. It is made from galvanized iron sheet and one of its 50mm edges is sharpened to form the lower peel penetrating edge.

It is expected that a knife assembly will not only move vertically in response to the root size but it will also tend to move horizontally in the direction of motion of the belt in response to the rolling resistance force in play between the knives and the root. Thus, front and rear rollers are provided on the knife assembly to enhance a frictionless motion. The rollers roll on roller motion guide planes (14 and 15) inclined at 45° to the horizontal and bolted to the knife assembly bearing frame. By this arrangement, the motion of the knife assembly is constrained and made frictionless.

Using bolts and nuts, the knife assembly bearing frame (7) is connected to the main frame link arms (10) (Figs. 4) through the slots in the four legs of the frame. The height of the knife bed above the conveyor belt can be adjusted by loosening these bolts and nuts, turning the square-threaded height adjustment screws (11) on either side of the knife assembly bearing frame until the desired height is reached and then tightening the bolts and nuts again. In this same manner, the lateral angle of inclination of the knives to the conveyor belt surface (across the belt width) can also be adjusted to suit the taper of the root surface from its proximal to its distal end.

2.3 Experimental Procedure for Testing

The root tubers used to test-run the experimental peeler were obtained from cassava plants aged between 18 and 20 months. Fresh roots randomly chosen from two IITA cultivars (TMS 30572 and TMS 30555) of the bitter cassava type commonly grown in Ile ï Ife, Nigeria, were used. The roots were run in the machine within 12 hours of harvesting. They were prepared for the tests by cutting them into 100mm long near cylindrical tubers. The diameters of a tuber at its proximal end, middle section and its distal end were measured using a pair of vernier calipers and recorded as d_1, d_2, and d_3 respectively. The average of d_1, d_2, and d_3 was subsequently computed and denoted d. The roots were then sorted into middle section diameter size groups of 45ï 55 mm, 55ï 65 mm, 65ï 75 mm, 75ï 85 mm and 85ï 95 mm. The angle of inclination of the knife bed to the surface of the conveyor belt across its width, α, was set at about 5.3°. This is the average degree of taper of the typical root surface from its proximal to its distal end (Adetan et al, 2003).

To run root tubers in the diameter (d) size group 45ï 55 mm, the average height (h) of the knife bed above the belt was set at 40 mm; to run root slices in the size group 55ï 65 mm, h was set at 50 mm; for d between 65 and 75 mm, h was set at 60 mm; for d between 75 and 85mm, h was set at 70 mm; and lastly, for d between 85 and 95 mm, h was set at 80 mm.

The drum speed for the conveyor belt drive was monitored using a phototachometer. In this preliminary experimentation, a drum speed (S) of 100 rpm was used. This is the approximate design speed that will ensure the optimal utilization of the power delivered to the conveyor system by the 2-hp variable speed drive motor. This speed, in turn, gave a linear belt speed (V) of 1.31 m/s. The tensile load, F_s, in each of the 25 sets of springs at each spring-loading point was varied between 44 and 110 N in steps of 22 N. The selection of these loads was informed by the results of an earlier work by Adetan et al (2003) which determined the cassava root peel penetration force per unit length of knife edge.

To acquire data on a root tuber at a given F_s, the experimental peeler was started up. The unpeeled cassava root tuber was fed manually into the machine and onto the top of the conveyor belt at the root pick-up end of the knife assembly. It rolled on its cylindrical surface under the spring-loaded knives on one side and over the conveyor belt on the other side until it was discharged at the other end of the belt peeled. The platform of aluminum sheet (5) over which the belt moved prevented it from sagging

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as it conveyed the cassava roots to be peeled under the array of knives set over it. In this way, the knives were kept in close penetration contact with the roots. Using an electronic balance, the mass of each tuber was measured before peeling, after machine peeling and after hand-trimming the machine-peeled tuber. These were recorded as m_1, m_2 and m_3 respectively and subsequently used to calculate the efficiency of peel removal, which is given by Eqn. (1) below.

Adetan et al. (2003) obtained the average percentage by weight of peel in a typical root, p, as 15.1 %. If we define C as the proportion by weight of peel that is removed by the peeler, and R as the proportion of the useful tuber flesh that is recovered after peeling, it can be shown (Adetan, 2002) that

$$C = 1 - \frac{100\,(m_2 - m_3)}{m_1 p} \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

and

$$R = \frac{100\, m_3}{m_1 (100 - p)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

where m_1 is the mass of the unpeeled root tuber, m_2 is the mass of the machine-peeled root tuber and m_3 is the mass of the machine-peeled and hand-trimmed root tuber.

An overall peeling quality index, Q, may also be defined as

$$Q = RC \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

3. RESULTS AND DISCUSSION

The results of the preliminary tests of the developed experimental peeler are summarized in Table 1. In these preliminary experiments, no useful root flesh was lost. For the different sizes of root slices used in these tests, the useful flesh recovery index obtained in each test was 1. Applying Eqn. (3), the overall peeling quality index, Q, is therefore equal to the peel removal efficiency C in all cases.

Table 1: Results of preliminary machine testing at a linear belt speed of 1.31 m/s

D_1 (mm)	D_2 (mm)	d_3 (mm)	d (mm)	h (mm)	F_s (N)	C	R	Q	C_{av}	State of root after peeling
65	52	45	54	40	44	0.939	1.000	0.939	0.939	Whole
53	53	40	49		66	0.920	1.000	0.920		Whole
59	55	39	51		88	0.973	1.000	0.973		Whole
60	47	21	43		110	0.825	1.000	0.925		Whole
60	64	55	60	50	44	1.000	1.000	1.000	0.988	Whole
67	60	51	59		66	0.953	1.000	0.953		Whole
58	59	59	59		88	1.000	1.000	1.000		Whole
41	65	63	56		110	1.000	1.000	1.000		Whole
62	66	54	61	60.	44	0.323	1.000	0.323	0.638	Whole
85	73	53	70		66	0.679	1.000	0.679		Whole
82	69	42	64		88	0.729	1.000	0.729		Broken
80	72	60	71		110	0.820	1.000	0.820		Whole
73	81	72	75	70	44	0.657	1.000	0.657	0.594	Broken
65	80	63	69		66	0.664	1.000	0.664		Whole
59	80	82	74		88	0.664	1.000	0.664		Whole
72	76	60	69		110	0.392	1.000	0.392		Whole
94	95	79	89	80	44	0.630	1.000	0.630	0.344	Broken
76	87	80	81		66	0.220	1.000	0.220		Whole
69	86	56	70		88	0.152	1.000	0.152		Whole
73	94	55	74		110	0.375	1.000	0.375		Whole

In general, two peeling patterns were observed during the preliminary experiments using the developed peeler. For some of the root slices run in the peeler, peels were finely lacerated by the knives and neatly brushed off the tuber surface. For other roots, the penetrated peels adhered to the knives and unwound from the tuber surface as the tuber rolled on.

Although the preliminary tests of the peeling principle were carried out on slices of root tubers, the results obtained indicate that the developed peeler shows good promise of effectively peeling whole roots of different sizes. The whole roots peeling system may explore the use of several knife assemblies (more than the five available on the present experimental peeler), that are independently spring-loaded and arranged across the knife bed such that the width of the bed will be greater than the possible length of any whole root tuber. Further experimentation and modelling studies on the experimental peeler are expected to yield the optimum conditions for the design of a high efficiency whole roots peeling system, which may be able to eliminate the occurrence of root breakage.

4. CONCLUSIONS

The preponderance of previous cassava peeling research efforts have concentrated on the use of abrasion to remove the peel of cassava roots in a batch peeling system. Success has, however, been limited because small roots become completely grated while larger roots remain largely unpeeled. An experimental peeling machine, which utilises the cassava peeling concept of peel-flesh separation by compression and peel removal by knives, was successfully developed and tested. The following conclusions were drawn from the results of preliminary tests using the peeler:

(i) The developed peeling machine achieves a high efficiency and excellent quality of peeling with virtually no loss of useful tuber flesh.

(ii) For relatively small roots, with diameters less than 65 mm, a peel removal efficiency C of approximately 100% was consistently achieved and the peeled roots were never broken. This is a very important improvement over the performance of previous peeling machines.

(iii) The present peeling machine shows promise of overcoming seemingly intractable cassava peeling problems and the limitations of previous abrasive type peelers. Furthermore, based on the peeling concept employed, the present results provide a scientific basis for the development of an efficient whole root peeling system.

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SOME PHYSICAL PROPERTIES OF GROUNDNUTS (*Arachis hypogea L*)

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ABSTRACT

In order to design and fabricate processing and handling equipment for groundnuts (*Arachis Hypogea L.*) some physical properties of shelled and unshelled groundnuts were determined. The mean values of major, minor and intermediate diameter (mutually perpendicular dimensions) were found to be 12.47, 7.21 and 7.78 mm for shelled seeds and 31.19, 10.74 and 10.80 mm for unshelled pods containing two to three seeds. The average sphericity and roundness were found to be 69.61 % and 54.78 % for shelled seeds and 50.17 % and 42 % for unshelled pods. The mean bulk density, porosity, kernel density, volume and angle of repose were found to be 0.60 g/cm^3, 0.46, 1.11 g/cm^3, 0.36 cm^3 and 24.5 degrees respectively for shelled seeds and 0.21 g/cm^3, 0.57 g/cm^3, 0.49 g/cm^3, 1.60 cm^3 and 23.3 degrees for unshelled pods.

KEYWORDS: Physical property, groundnut, shelled pods, unshelled pods, decorticate.

1. INTRODUCTION

Groundnut (*Arachis hypogea* L.), also known as peanut or earthnut, is a member of the *Papilionaical*, the largest and most important of the three divisions of *leguminasae*. Groundnut was cultivated in Peru as far back as about 3000 B. C. from where it was believed to have originated. It was brought to West Africa from Brazil in the 16th century. It comprises of two main types, the first being the bunch or erect type in which the main stem and branches grow upright so that there is little spreading; the nuts are found closed together at the base of the plant. This type is suited to mechanical harvesting. The second is the creeping type with branches trailing along the ground and the plant therefore has spreading habit. Most of the ones grown in West Africa and Nigeria are of the creeping type. Usually after harvesting the crop, they are shelled, decorticated, cleaned before usage either as cake or extraction of the oil for cooking purposes.

To design equipment for shelling and decorticating the groundnuts, their physical properties must be known. Oje and Ugbor (1991), Oje (1993), Koya and Adekoya (1994), Oje et al. (1997), Alonge and Adigun (1999), Adigun and Alonge (2000), Oje et al. (2001), Alonge (2003)) all carried out studies on physical and mechanical properties of some agricultural product. But little is known of the physical properties of groundnut. The purpose of this work was to determine, with sufficient accuracy for design purposes such physical properties as dimensions, weight, angle of repose, coefficient of friction on possible structural surfaces, bulk density, porosity and sphericity of groundnut relevant to its decortication

2. MATERIALS AND METHODS

Ten kilograms of groundnut (shelled and unshelled) used for the experiment were obtained from the Institute of Agricultural Research and Training, Ibadan. The variety used is local, commonly called ñEpa pupaò, an early-maturing type grown in South Western Nigeria. The physical properties determined and the experimental methods are given below:

2.1 Size and Shape

In determining grain size, one hundred each of shelled and unshelled groundnut were randomly selected. Measurements of the three mutually perpendicular axes were made; namely major, intermediate and minor diameters with a micrometer screw gauge

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2.2 Weight

This was determined using an electronic balance for 100 seeds.

2.3 Volume and Solids Density

The volume of each seed for one hundred samples was determined by water displacement method as described by Oje and Ugbor (1991). Individual seeds were immersed in water inside a thin tube-measuring cylinder showing a notable rise in water level. The differences in the final and initial readings were recorded as the volume of seed or pod. The density was obtained by dividing weight of seed by its volume. A sinker was used to screen the pod or unshelled groundnut.

2.4 Frontal Area, Sphericity and Roundness

One hundred seeds were each placed in their natural resting position on a sheet of graph paper. A sharp thin pencil was used to carefully trace the edges of the seed. The frontal area was determined by counting the squares within the traced marks. The projected area and the diameters of circle inscribing and circumscribing the projected areas were measured in order to determine the sphericity and roundness based on the following equations:

Roundness $= A_P/A_c$ é é é é é é é é é é é é é (1)

Sphericity $= d_i/d_c$ é é é é é é é é é é é é (2)

where d_c and A_c are the diameter and area of the smallest circumscribing circle respectively while d_i denotes diameter of largest inscribing circle and represents the projected area of the seed.

2.5 Coefficient of Friction and Angle of Repose

The static coefficients of friction of groundnut were determined on three materials plywood, glass and mild steel. A topless and bottomless material box of dimensions 150 mm x 150 mm x 40 mm was filled with the seeds and placed on adjustable tilting table onto which the material to be tested has previously been fastened. The box was placed on one end of the surface and raised slightly so it was not touching the material. A screw device slowly tilted to the table until the friction force between the seeds and the material was overcome by gravity and movement down the slope began. The angle of inclination was read from a graduated protractor attached to the tilting table. Five replications were carried out for shelled seeds and unshelled pods.

According to Mohsenin (1978) and Chakraverty (1988), the tangent of the angle of inclination is the static coefficient of friction of the grain on the material. Oje and Ugbor (1991) determined static coefficient of friction for different surfaces such as galvanized steel, plywood and glass. They also described the determination of angle of repose by using a specially constructed box with the crop with a removable front panel. The box is filled with the crop, then the front panel is quickly removed. This allows the grain to flow to its natural slope. This slope is a measure of angle of repose. This method was used for the experiment.

2.6 Bulk Density

The bulk density was determined by weighing a 100cm^3 cylinder empty. The cylinder was then filled with the seeds and reweighed.

$$D_b = \frac{M_2 - M_1}{V} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where D_b is bulk density, M_1 is the mass of empty cylinder, M_2 is mass of cylinder plus seeds and V is the volume of cylinder. Five trials were carried out for each set of seed and their means were recorded.

2.7 Porosity

Based on the relationship for porosity by Mohsehnin (1978) the porosity was calculated thus:
where P_f is the porosity of grain, D_b is the bulk density and D_s is the solid or kernel density of grain.

$$P_f = 1 - \frac{D_b}{D_s} \quad (4)$$

3. RESULTS AND DISCUSSION

The summary of the results for all the properties measured is shown in Tables 1 and 2 for shelled seeds and unshelled pods respectively. The frequency distribution of some of the properties is shown in Figures 1 and 2 for shelled seeds and unshelled groundnut pods. For shelled seeds, sixty percent of the major axes fall between 12 and 13 mm, 94% of the intermediate axes falls between 6.7 mm and 9 mm and 94 % of the minor axes falls between 6.5 mm and 8 mm. For the unshelled pods 82 % of major diameter falls between 20 mm and 35 mm, 78 % of intermediate diameter falls between 9.5 mm to 11.5 mm while 80 % of minor diameter falls between 9.5 mm to 10 mm. The frequency distribution curve for major and intermediate axes skewed towards the right while for the minor axes it is near normal. These properties are needed in designing diameter of sieves for sieve cleaner.

The density of the shelled groundnut falls between 1.02 and 1.27 g/cm^3. This implies that the seeds will sink in water, which makes it possible to separate them from other materials that are less dense than water. The frequency distribution for density of the unshelled groundnut is close to normal. It varies between 0.40 g/cm^3 and 0.58 g/cm^3, the mean being 0.49 g/cm^3. Separation from unwanted material denser than water is possible for example stones. The mean porosity obtained for the shelled groundnut was 0.46 and 0.57 for the unshelled. This shows that a pack of unshelled groundnut pods is more porous than the shelled one. This property is required in air and heat flow in an agricultural material. This shows how easily a stream of heated air for drying will pass through a pack of material and thus affect the rate of drying of the materials. From this result, unshelled groundnut pods might dry faster than shelled seeds during air drying process.

With regards to the static coefficient of friction for shelled groundnut determined on the three structural surfaces it was found that, glass surface had the highest mean value of 0.535 while the lowest value was by mild steel was 0.455. For the unshelled pods the plywood has the highest mean value of 0.52 while the lowest value was on glass with a mean value of 0.495. This property is needed in the design of agricultural machine hopper and other conveying equipment. It determines how a pack of grain or seed will flow in these systems.

The angle of repose for shelled seeds was found to be between 22^0 and 27^0, while for the unshelled pods it was between 27^0 and 30^0. This property determines the minimum slope of flow in self-emptying bin or minimum slope of flow in a hopper. The mean frontal or projected area for the shelled is 68.8 mm^2, for the unshelled pod it is 309.2 mm^2. This property affects the velocity of air stream that can be used in order to separate the seed from unwanted material in pneumatic separator or to convey seed in pneumatic conveying.

The highest roundness value obtained for unshelled pods was 58.93 % while the highest sphericity value obtained was 74.78 % for unshelled groundnut shown in Table 2. This implies that the tendency for the material to roll is fairly high. For the shelled seed the highest roundness value obtained was 64.79 % while the highest sphericity value obtained was 85.49. This shows that the material will always tend to roll when it is on a particular orientation. This property is always considered when designing hopper and dehulling equipment for seeds.

Table 1: Some physical properties of shelled groundnut

	No. of observations	Mini Value	Mean Value	Maxi Value	Standard Deviation
Major Diameter (cm)	100	6.71	12.27	15.71	1.92
Intermediate Diameter (cm)	100	6.79	7.78	8.92	0.51
Minor Diameter (cm)	100	5.54	7.21	9.30	0.68
Weight (g)	100	0.29	0.40	0.54	0.06
Geometric Mean diameter (cm)	100	6.96	8.84	10.37	0.67
Roundness	100	49.12	54.73	64.79	6.12
Sphericity (%)	100	61.74	69.61	85.49	4.78
Volume (cm^3)	100	0.25	0.36	0.45	0.06
Solid Density (g/cm^3)	100	1.02	1.11	1.27	0.06
Bulk Density (g/cm^3)	5	0.59	0.60	0.61	0.01
Surface Area (mm^2)	100	600.27	628.91	650.77	19.25
Porosity	5	0.42	0.46	0.52	0.02
Frontal Area (mm^2)	100	58.00	68.80	82.00	9.24
Repose Angle (degrees)	5	22	24.50	27.00	2.07
Coefficient of friction on					
(i) Plywood	11	0.47	0.525	0.58	0.05
(ii) Mild steel	11	0.42	0.455	0.49	0.04
(iii) Glass	11	0.45	0.535	0.62	0.07

Table 2: Some physical properties of unshelled groundnut pods

	No. of observations	Mini Value	Mean Value	Maxi Value	Standard Deviation
Major Diameter (cm)	100	16.29	31.19	41.94	5.16
Intermediate diameter (cm)	100	8.59	10.80	12.23	0.81
Minor Diameter (cm)	100	9.23	10.74	12.51	0.75
Weight (g)	100	0.40	1.25	1.94	0.38
Geometric Mean diameter (cm)	100	12.00	15.28	17.70	1.23
Roundness	100	19.12	42.2	58.93	15.43
Sphericity (%)	100	40.40	50.17	74.78	6.66
Volume (cm^3)	100	1.50	1.60	3.50	0.82
Density (g/cm^3)	100	0.40	0.49	0.58	0.07
Bulk Density (g/cm^3)	5	0.19	0.21	0.23	0.01
Surface Area (mm^2)	100	40.38	46.27	52.16	16.13
Porosity	5	0.53	0.57	0.60	0.02
Frontal Area (mm^2)	100	248	273	298	82.56
Repose Angle (degrees)	5	27.0	28.5	30.0	1.20
Coefficient of friction on					
(i) Plywood	11	0.47	0.520	0.57	0.03
(ii) Glass	11	0.45	0.470	0.49	0.01
(iii) Mild steel	11	0.48	0.495	0.51	0.02

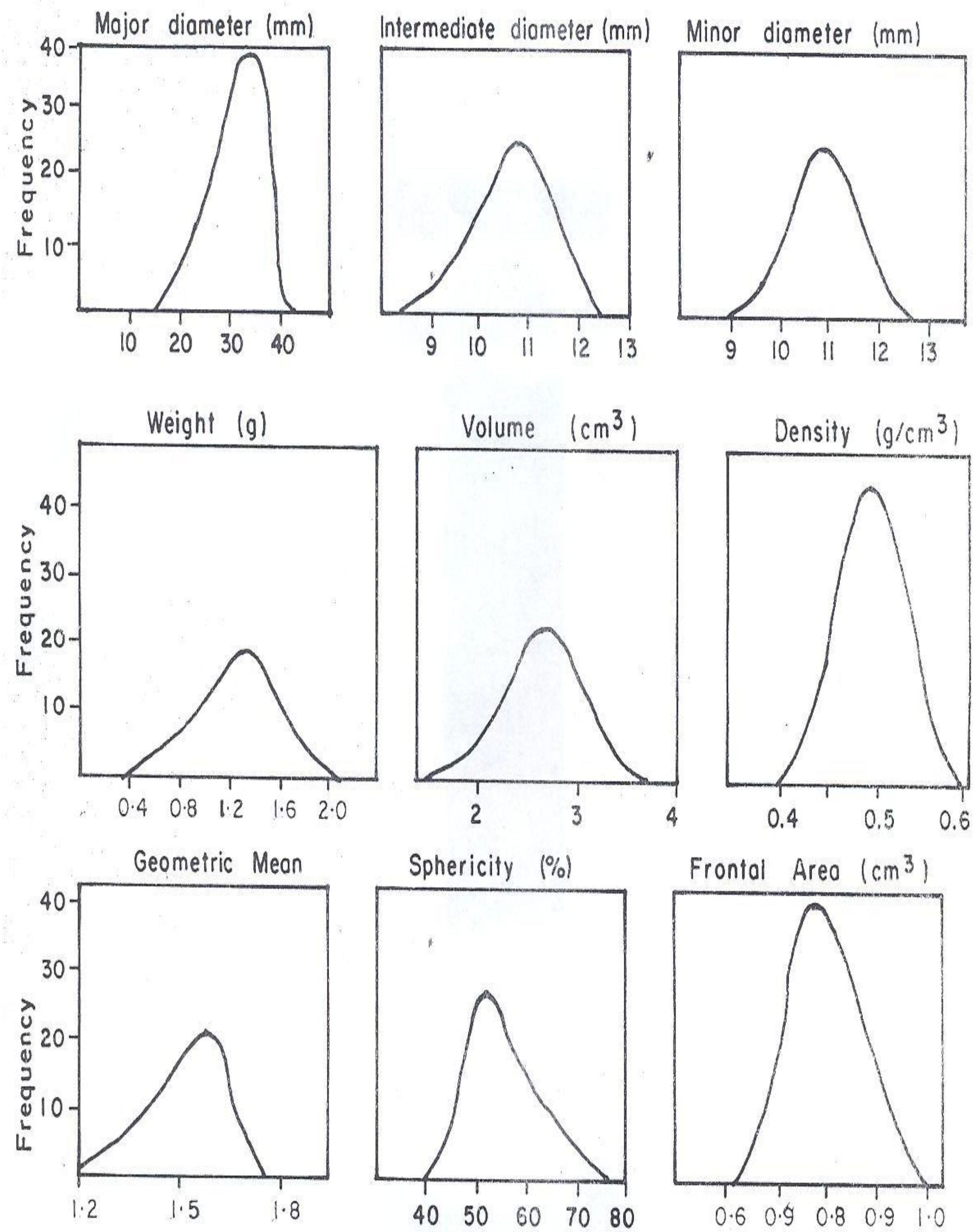

Fig. 1: Frequency distribution of some physical properties of unshelled groundnut pods.

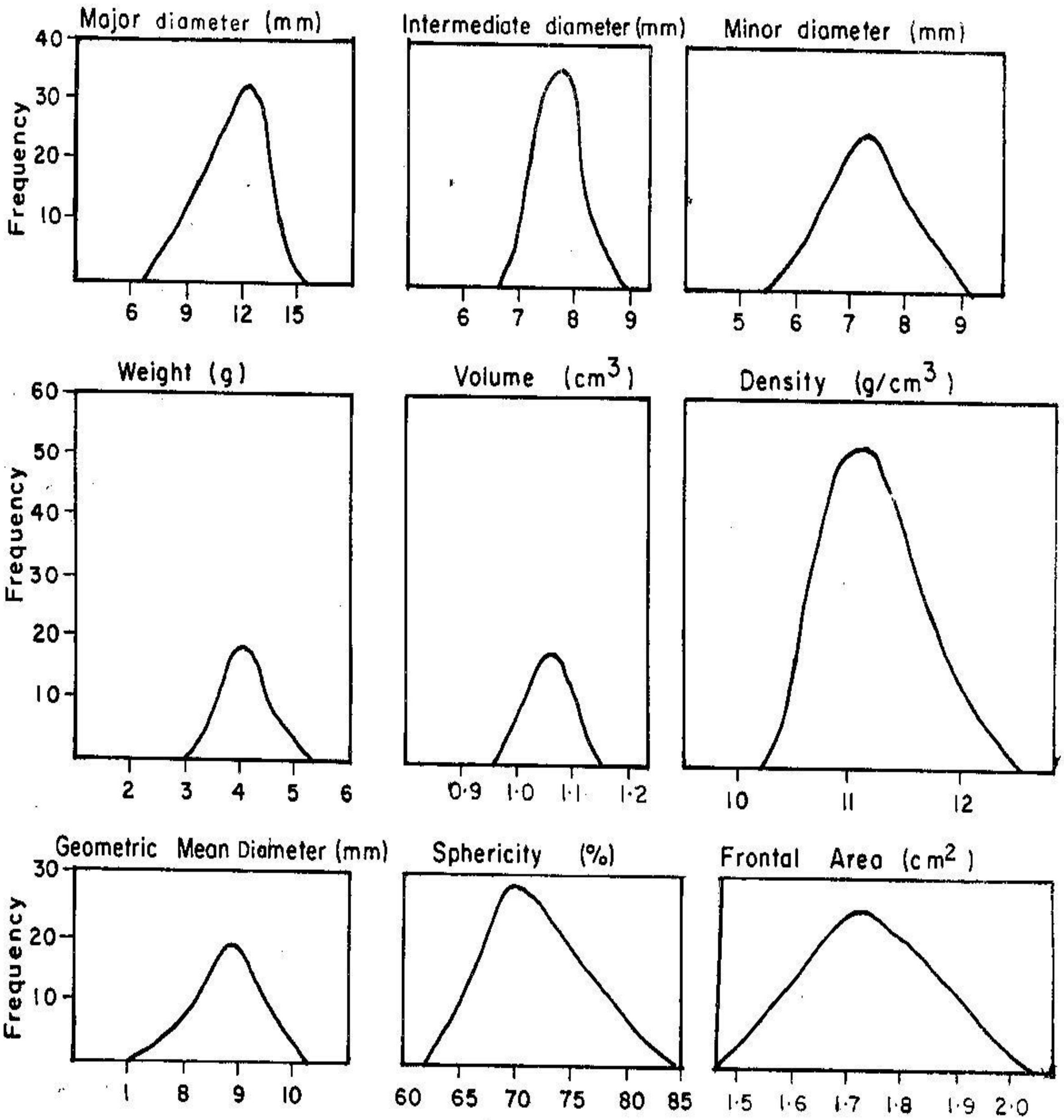

Fig. 2: Frequency distribution of some physical properties of shelled groundnut pods

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4. CONCLUSIONS

The physical properties of groundnut (shelled and unshelled) were measured. The following conclusions are drawn:

(a) For shelled seeds - The estimated mean major, minor and intermediate diameters were found to be 12.49 mm. 7.21 mm and 7.28 mm respectively. The roundness was found to be 69.61 % while the sphericity was 54.76 %. The mean weight was found to be 0.40g.

(b) For unshelled pods ï The estimated mean diameter for major, minor and intermediate axes were found to be 312.19 mm, 10.80 mm and 10.74 mm respectively. The roundness was 50.17 % while sphericity was 42.2 %. The average weight was found to be 1.25g.

(c) The results for the coefficient of friction for both shelled and unshelled groundnut showed that hopper and unloading device need not be built very steeply.

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DEVELOPMENT AND PERFORMANCE EVALUATION OF A POULTRY FEED MIXER

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ABSTRACT

A poultry feed mixer was designed, constructed and evaluated for its performance. The effectiveness of mixing was based on the value of standard deviation and coefficient of variation of the sample collected from the tested mixed materials at the operating speeds. The performance of the mixer at the tested operating speeds of 100rpm, 150rpm 250rpm were: standard deviation, (S.D) 7.37 and coefficient of variation (CV) 19.33%; S.D. 1.29 and CV 3.07% and SD of 10.84 and CV of 46.98% respectively. The effectiveness of mixing of the mixer was high at operating speed of 150rpm, followed by operating speed of 100rpm but the effectiveness of mixing of the mixer was lowest at operating speed of 250rpm.

KEYWORDS: Poultry, feed mixer, standard deviation, coefficient of variation, livestock feeds.

1. INTRODUCTION

Livestock feed is the general term for food given to farm animals. The regular supply of food to farm animals is very essential to a healthy and productive life. Feed is an essential requirement in poultry production as it is in all other livestock keeping. Oluyemi and Robert (1978) stated that once the poultry man has selected a good bird with long life ability, high genetic capacity to grow or lay eggs effectively and has prepared the housing and the management essential for the successful operation, the next thing is to produce the most efficient nutritionally complete diet to suit a particular environmental condition.

Livestock feed as a prominent factor in poultry production influences significantly the overall quality and quantity of birds and feeds. Preparation involves grinding and mixing different ingredients to form homogeneous mix (Davis and Hall, 1979).

Mixing as one of the processes involved in feed preparation must be attended to with care as improper mixing of feed ingredients could produce unbalance ration that undernourishes the birds. Smith and Wilkes, (1984) and Beldin et. al (1968) reported the need to use mixer when attempting to mix two or more ground feeds of known feeding value to obtain a balance ration as it ensures mixes of uniform consistency. However, most of the available mixers currently in the market are imported, sophisticated in design and very expensive for rural farmers to afford and maintain. There is a need for a small sized, simple and inexpensive mixer that would solve the problem of providing an alternative feed mix for rural livestock farmers.

The aim of this paper is to develop and evaluate the performance of such a poultry feed mixer.

2. DEVELOPMENT OF THE POULTRY FEED MIXER

2.1 Background

Mixers are classified according to their method of mixing. Solid particles are mixed by the different methods as reported by Lacey (1984). Convective mixing, occurs by the transportation of group of particles from one location of the mixer to another location. Diffuse mixing, involves the distribution of particles over a freshly developed surface of the materials. Shear mixing involves the setting up of slip planes within the mixture.

Beldin et. al (1968) classified mixing into different categories. These include layer mixing, which is the mixing of ingredients in layers in a feeding wagon. It occurs as different layers move through the beaters and fall into the cross conveyors. Stream mixing is a form of layer mixing on a more refined scale in which accurately metered amount of feed are deposited onto a moving conveyor with each cross section of feed containing exactly the required amounts of each of the different ingredients. Batch mixing involves mixing which take place in the tank of the mixer in batch quantities by means of re-circulation process. The action of the mixing mechanism is such that feed elements are recirculated at different speed than the others.

Feed mixers designers and farmers have use different methods as reported by Beldin et. al (1968) to determine the quality of feed mix. Firstly, visual observation and comparison where farmers closely observe the feed while it is being unloaded in bunks and semi-feeders, and note the feed for uniform mix. Secondly, noting uniform condition of the livestock, where farmers watch the condition of the herds as a measure of mixing ability of the mixer. The lesser the trouble with the cattle the better the mixer. Thirdly, addition of an easily distinguishable tracer and visually observing the overall uniformity distribution is a method where coloured tracers are added to the mix and the uniformity of the tracers distribution is noted. The fourth method is laboratory analysis of samples where mixed feeds are sent to the public or commercial laboratories for overall ration analysis.

The factors affecting the throughput of the auger conveyor according to Stevens (1962) include operating speed, elevation of the auger conveyor and the pitch of the auger. The peak throughput based on the experiment performed on an auger conveyor was attained when the operating speed was in the region of 100rpm, but commercial auger in common use run at speed range between 1000rpm and 2000rpm for a 3inch diameter (76.2mm)-auger and 600 to 750 rpm for a 6inch (152.4mm) auger. As the elevation of the conveyor increases, the throughput decreases except the auger with the smallest value of ratio of pitch to the diameter. Most commercial augers have a value ratio of pitch to diameter of approximately 1, and high capacity conveyor of free flowing materials has a value ratio of pitch to diameter of approximately 1.5.

2.2 Design Consideration

In designing the mixer, the following factors were considered:

-Material Selection; minimum cost, strength, durability, appearance and availability of local materials.
-Stability of the Machine; geometrical shape of the mixer as well as the strength of the materials that would withstand vibrations under different operating conditions.
-Power Requirements; estimation of the total power considers the power to operate the mixer empty and the power to operate it at a desired load and desired elevation.
-Machine Capacity; economic utilization of the feed based on the population of the birds and cost of the feed materials.

2.3 Description of Poultry Feed Mixer

A vertical mixer was designed and constructed. The mixer comprises of the following components as shown in Figures 1 and 2. The major components include: the main body of total volume of 0.1852m^3 and was constructed of a sheet metal of gauge 20. The hopper is right angle triangle in shape and has a volume of 0.01125m^3. The auger conveyor is a hollow shaft with external diameter of 50mm and internal diameter of 44mm and pitch of 160mm. The capacity of the auger conveyor was 1.56tonnes/hr determined from the expression of Stevens (1962). The bearing was rolling contact bearing (single row, deep groove ball bearing). The bearing was chosen according to ASAE standard as given by Hall, et al, (1988). The power requirements consist of the power to run the mixer empty and the power to run the mixer on load. The total power required is 3.80kW or 5hp. The belt chosen is V-belt, rubber impregnated with leather of length 840mm.

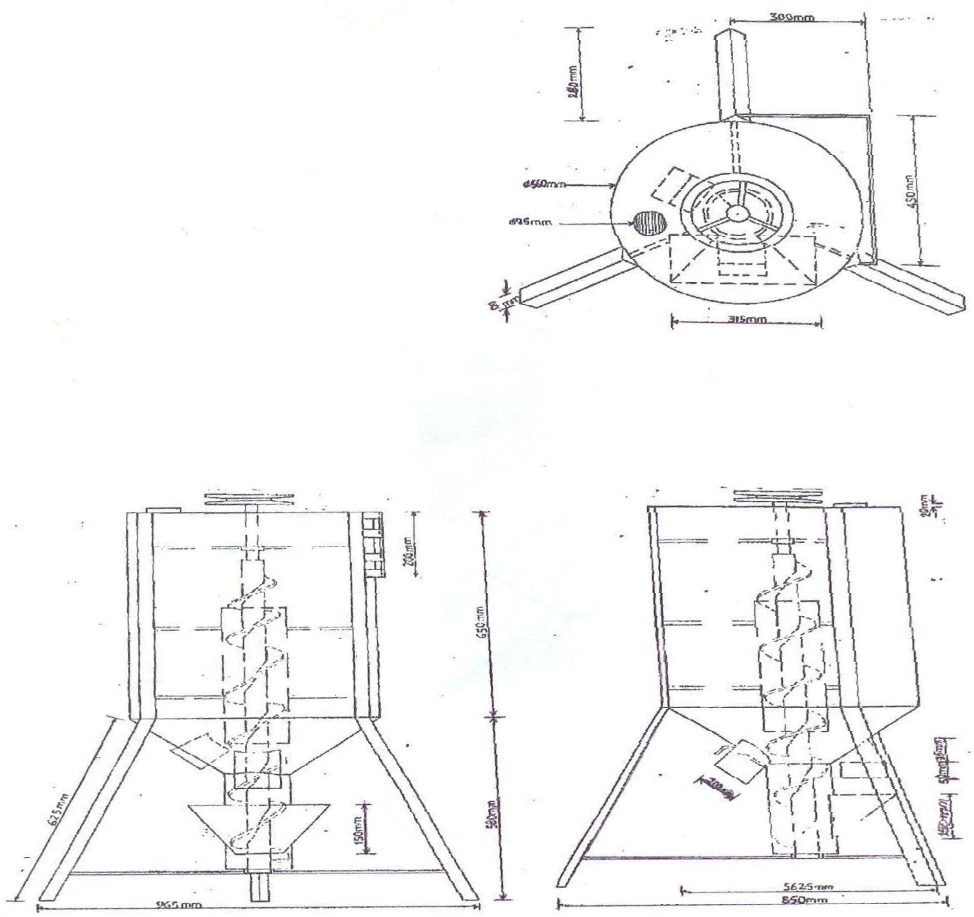

Fig. 1. Details of the poultry mixer

Fig. 2. Picture of a poultry mixer

The machine was developed and constructed as a vertical mixer. The mixer works by the principle of convectional mixing. The mixer is fed from the hopper and can be operated by an electric motor of 5hp. The mixing starts with the charging of a desired proportions of feed ingredients via the hopper into the mixer which is simultaneously followed by addition of additives. The mixer is allowed to run for a period of time to effect proper mixing of the feed ingredients. The mechanism of mixing involves the conveyance of feed materials from the lower part to the upper part of mixer by the auger conveyor working at the desired operating speed.

The materials on getting to the upper part sprinkles on the mixer parts and get down by gravity. The process repeats until the desired mixing is obtained by viewing the degree of mixing in the testing spout of the mixer.

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2.4 Performance Evaluation

The test procedure followed that of Beldin, et al. (1968). Three operating speeds, of 100rpm, 150rpm 250rpm were used for the performance test. As the mixer is working, 4kg of ground corn mixed with 1kg of soyabean seeds serving as local tracer were fed into the mixer via the hopper. The mixer was operated for five minutes for each of the three operating speeds, and the time was measured using stop-watch. After mixing and from the mixed feed, an equally spaced 13 samples measuring 100gm per sample were randomly taken. The soyabean seeds contained in every samples were sorted and counted. The effectiveness of mixing of the mixer was calculated as coefficient of variation from the relationship:

$$CV = \frac{SD}{m} * 100\% \qquad \text{-------------------- (1)}$$

Where, SD = standard Deviation of mixed feeds and is mathematically expressed as

$$SD = \frac{\Sigma (x-m)^2}{n} \qquad \text{-------------------- (2)}$$

m = mean of soyabeans seeds distribution in the samples; and is mathematically expressed as,

$$m = \frac{\Sigma x}{n} \qquad \text{--------------------- (3)}$$

Where: x = the number of soyabeans seeds in the sample, n = number of samples collected.

3. RESULTS AND DISCUSSION

The result of the performance evaluation of the poultry feed mixer are as shown in Table 1 and Table 2. It is seen from Table 2 that the standard deviation (SD) and the coefficient of variation (CV) are high at operating speed of 100rpm and higher at operating speed of 250rpm but low at operating speed of 150rpm. The low value of SD and CV is an indication of uniform distribution of the soyabeans seeds (local tracer) in the tested mixed feed, and the high value of SD and CV is an indication of non-uniform distribution and the higher value of SD and CV is an indication of poor distribution of the local tracer on the tested samples (Beldin, et. al, 1968). The effectiveness of mixing of the mixer could be considered high at the operating speed of 150rpm, followed by the operating speed of 100rpm but considered low at operating speed of 250rpm.

Table 1. Performance test result of the poultry feed mixer

Sample	100rpm		150rpm		250rpm	
No.	% of Soyabeans seeds in the sample (x)	$(x-m)^2$	% of Soyabeans seeds in the sample (x)	$(x-m)^2$	% of Soyabeans seeds in the sample (x)	$(x-m)^2$
1	40	3.42	42	0	20	9.42
2	36	4.41	40	4	40	286.62
3	30	66.42	43	1	10	170.82
4	32	37.82	40	4	25	3.72
5	42	14.82	42	0	15	65.12
6	50	140.42	43	1	5	326.52
7	34	17.22	42	0	10	170.82
8	48	97.02	42	0	20	9.42
9	46	61.62	43	1	25	3.72
10	24	200.22	44	4	30	48.02
11	36	4.62	43	1	25	3.72
12	38	0.023	42	0	40	170.82
13	40	3.423	40	4	35	142.32
Total	416	651.456	546	20	300	1411.06

Table 2. Summary of the performance test results of the mixer

Speed of operation (PM)	100	150	250
No. Sample Collected	13	13	13
Mean Value of the Samples	38.15	42	23.07
SD	7.37	1.29	10.98
CV (%)	19.30	3.07	46.98

4. CONCLUSION

The poultry feed mixer was designed, constructed and evaluated. It was concluded that the effectiveness of mixing is high at moderate operating speed. The mixer performance was high at operating speed of 150rpm with standard deviation (SD) and the coefficient of variation (CV) of 1.29 and 3.07% respectively.

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MODELING THE CHANGES IN SIZE OF AIR-BORNE LIQUID SPRAY DROPLETS IN WINDY AND STILL AIR CONDITIONS

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ABSTRACT

Mathematical models considering the aerodynamic and physical characteristics of liquid spray droplets as well as the transport properties of air were developed to simulate changes in the size of air-borne liquid spray droplets under varying field conditions. The models were programmed for solutions using the fourth order Runge Kuta numerical integration techniques. A wind tunnel was constructed to provide an experimental base for the validation of the mathematical models.

The model predicted a percentage size decrease of 11.6, 10.78, 9.09 and 7.758 % at still air and 29.52, 19.86, 14.99 and 12.04 % at the wind speed of 1.0 m/s for droplet sizes of 100, 150, 200 and 250 microns respectively after 10 seconds of being air-borne. As the size reduction rate in air increases, the chances of off-target deposition of the spray droplets increase. The results of the numerical computations compare fairly well with validation results confirming that the rate of change in the size of an air-borne liquid spray droplet is influenced by the droplet size, droplet retention time in air, wind velocity and intensity.

KEYWORDS: Modeling, change in size, air-borne, liquid, spray, droplet.

1. INTRODUCTION

In agriculture, agro-chemicals are used to protect crops against insects, weeds, and diseases. One popular method of doing this is by spraying. Whenever a liquid spray droplet is exposed to any environment of less than 100% relative humidity, the droplet will undergo a reduction in size by the net mass transfer of the water molecules in vapor state (Law and Bowen, 1975). The size reduction process, induced by such factors as the environment, aerodynamic and physical characteristics of the spray droplet, transport properties of the transport medium (air) and droplet fluid (water), influence the degree of off-target or on-target deposition of the spray droplets. Off- target deposition of agrochemicals causes not only wastage of costly chemicals, but also environmental pollution of nearby crops, as well as air, soil and water resources around the farm.

There is the need therefore to understand the interaction of the various factors affecting the size reduction processes of liquid spray droplets in dynamic conditions. One of the ways of achieving this understanding is by mathematical modeling and computer based simulation.

Previous attempts at developing mathematical models for simulating the reduction in size of spray droplets have been made (Law and Bowen, 1975, Goering et al, 1972, Williamson and Thread gill 1974 and Miller and Hadfield 1989). From review of the literature, it was seen that no single model has been able to incorporate most of the parameters influencing the reduction in size of liquid droplets in air. For instance while some of the researchers view spray droplet evaporation as being closely related only to convection of heat from bodies of the same shape; others see it as solely a molecular diffusion problem. These different understandings of the problem led to the development of different models some of which vary very significantly (Miller and Hadfield 1989).

The objective of this study was to develop and validate a simulation model which will consider all the various factors of the environment, physical, aerodynamic and external factors affecting the size reduction and deposition of an air-borne liquid droplet. Such a model will provide an insight into the dynamics and deposition of an air-borne spray droplet thereby supplying the required environmental, physical and aerodynamic data for maximizing efficient spray droplet deposition and minimizing drift under varying spraying conditions.

2. MATERIALS AND METHODS

2.1 Model Development

In developing the model, the basic principles from the earlier works were applied (Marshall, 1955; Ranz and Marshall 1952; Williamson and Thread gill 1974; Goering et al, 1972 and Marchant,1977.

Earlier studies on air-borne evaporation of spray droplets resulting to size reduction (Ranz and Marshall 1952), reported that the rate of change of mass of an air-borne spray droplet was strongly related to the surface area of the droplet in air, the mass transfer coefficient of the droplet, the flow regime and the vapor pressure difference between the ambient air and the evaporating droplet surface. They mathematically expressed these relationships as follows:

$\partial m/ \partial t = K_g.A_s.\Delta P.M_p$ é 1

Where $\partial m/ \partial t$ = mass transfer rate (gs^{-1})
K_g = Mass transfer coefficient (moles $s^{-1}m^{-2}$)
M_p = Molecular weight of the droplet liquid (g $mole^{-1}$)
A_s =Area of mass transfer = total surface area of droplet (m^2)
ΔP = Vapor pressure difference (gm^{-1} s^{-2})

Assuming that the droplets are spherical in shape (Smith et al., 1982; Braig 1986; Goering et al, 1972), then we obtain,

$A_s = \pi D_p/ 4$ and $V_s = 4\pi r^3 / 3$

A_s = Area of the spherical spray droplet (m^2)
V_s =volume of sphere (m^3)
D_p=diameter of sphere (m)
r = radius of the sphere (m)

From the relationship: density = mass/ volume, we have
Mass = density x volume
= ρ_p x volume of sphere.
$M = \rho_p \pi D_p^3/6$ é é é é é é é é é é é é é é é é é é é .2

Where M= mass of droplet liquid (g)
ρ_p=density of liquid (gm^{-3})
Differentiating equation 2 with respect to D_p, we have,

$\partial m/ \partial t = \rho_p \pi D_p^2/2 * \partial D_p/ \partial t$é é é é é é é é é é é é ..3

Comparing equations 1 and 3, we have,

$= \rho_p \pi D_p^2/2 * \partial D_p/ \partial t = K_g * A_s * \Delta P * M$é é é é é é é é é é .4

The density of fluid (air), droplet diameter, diffusivity and the flow regime of the droplet in air influence the mass transfer coefficient, K_g. Hence K_g could be expressed as

$K_g = \rho_a * D_v * N_N/ M_a * D_p * P_a$ é é é é é é é é é é é é é é 5

Where ρ_a = density of air (kgm^{-3})
D_v = diffusivity of the droplet vapor in air (m^2s^{-1})
N_N= Nusselt Number of heat and mass transfer (dimensionless)
M_a = Molecular weight of the fluid (air) (g $mole^{-1}$)
D_p= droplet or particle diameter (m)
P_a = Partial pressure of air (g $m^{-1}s^{-2}$)

$N_N = 2.0 + 0.6\,(RN)^{1/2}\,(SN)^{1/3}$ é é é é é é é é é é é é é é 6

Therefore arranging equations 5 and 6 gives

$K_g = (\rho_a D_v / M_a D_p P_a)\,[2.0 + 0.6\,(RN)^{1/2}\,(SN)^{1/3}]$ é é é é .7

Substituting equation 7 into 4, we obtain,

$\rho_p \pi D_p^2/2 * \partial D_p / \partial t = (\rho_a D_v / M_a D_p P_a)\,[2.0 + 0.6\,(RN)^{1/2}\,(SN)^{1/3}]\,(\pi D_p^2 / 4 * \Delta P * M_p)$é .8

Simplifying equation 8 gives

$\partial D_p / \partial t = -(\rho_a D_v . M_p \Delta P / 2 * M_a . D_p . P_a . \rho_p)\,[2.0 + 0.6\,(RN)^{1/2}\,(SN)^{1/3}]$é é é ..9

The negative sign indicates that the air borne droplet is reducing in size.

From previous work (Marshall, 1955)

$\partial Dp / \partial t = 8 \Delta TK / \rho_p . \delta$ é 10

Where ΔT = temperature difference between air and droplet surface (^oC)
K = average thermal conductivity of gas film surrounding the droplet (KW m^{-1} K^{-1})
ρ_p =density of the droplet liquid (g m^{-1})
δ = latent heat of vaporization (KW kg^{-1})

From equations 9 and 10, we have,

$\partial D_p / \partial t = 8 \Delta TK / \rho_p . \delta = (\rho_a D_v . M_p \Delta P / 2 . M_a . D_p . P_a . \rho_p)\,[2.0 + 0.6 (RN)^{1/2} (SN)^{1/3}$é ¨11

Simplifying and rearranging equation 11, gives

$\partial D_p / \partial t = (\rho_a . D_v . M_p \Delta P . \delta / 8 * M_a . D_p . P_a . \Delta T . K)\,[1 + 0.3\,(RN)^{1/2}\,(SN)^{1/3}]$é é é .12

Where RN = Reynolds Number (dimensionless)
SN = Schmidt Number (dimensionless)
Other parameters as earlier defined.

Equation 12 gives the complete mathematical model for describing the rate of change of size of any air borne water based spray droplet in any environment where the relative humidity is less than 100%.

2.2 Wind Effect

During, unstable atmosphere like that of windy environment often experienced during field spraying operations, an air borne spray droplet tends to be drifted horizontally or vertically down to the target or off-target.

In order to predict what happens to the droplet along the horizontal and vertical directions of motion of the droplet, it is assumed that the droplet moves with an initial velocity V_o, direction ϕ , relative to the horizontal and time, t. Wind with velocity, V_a, blows on the droplet at an angle α , relative to the horizontal as shown in Fig. 1.

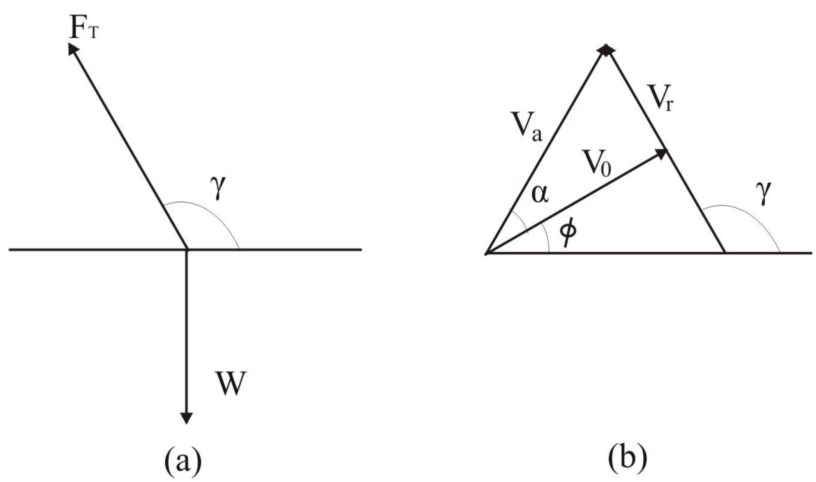

Figure 1.0: (a) Forces acting on the droplet. (b) Velocity diagram

Analysis of figure 1.0 results to equations 13 and 14

$\sum$ (RH) = V_a cosŬ+ V_o cos ϕ ……………………………………..13

$\sum$ (RV) = V_a sinŬ+ V_o sinϕ é é é é é é é é é é é é é é é ..14

Where RH = component of the air and droplet velocities along the horizontal direction.
RV = component of the air and droplet velocities along the vertical direction.

2.3 The Model

The evaporation rate of an air-borne spray droplet is directly proportional to the velocities of the droplet liquid and its medium (air) (Sutton, 1953).
Therefore,

λ Ð ($V_a + V_o$)

$\lambda = k (V_a + V_o)$é ..15

Where λ = evaporation rate for water based droplet, K = factors of evaporation, V_a= velocity of air. V_o = initial velocity of the droplet.

Comparing and arranging equations 12, 13 and 14 with 15 gives

$\partial D_p (x)/\partial t = -1/8 (\rho_a.D_v.M_p.\Delta P.\delta \ /*M_a.D_p.P_a. \ \Delta T.K) (1 + 0.3 (RN)^{1/2} (SN)^{1/3})$ RHé 16

$$\partial D_p (y)/\partial t = -1/8 (\rho_a.D_v.M_p \Delta P.\delta / *M_a.D_p.P_a.\Delta T.K) (1 + 0.3 (RN)^{1/2} (SN)^{1/3}) RVé \quad 17$$

Where $\partial D_p (x)/\partial t$ = droplet evaporation rate in the horizontal direction. $\partial D_p (y)/\partial t$ = droplet evaporation rate in the vertical direction.

Equations 16 and 17 were used to simulate spray droplet evaporation along the horizontal and vertical directions of wind motion, respectively.

2.4 Numerical Solution

The set of equations 1-17 represents the set of mathematical models used to predict the evaporation of an air-borne spray water-based droplet. The equations were programmed for solutions using 4th order Runge Kuta numerical integration technique.

Fourth Order Runge Kutta numerical method was used in solving the set of equations. The equations were first reduced to a system of first-order equations and then solved simultaneously. The advantage of this method is that as the X, Y locations of the droplet are changing with time, so also is the droplet diameter which is decreasing through evaporation. Thus, at every discrete step, the droplet diameter is updated to get the new location.

The fourth-order Runge-kutta algorithm for the one-step integration of a single first-order equation with one appropriate initial condition was written for a system of n-first order equations with n-initial conditions as presented below.

For a system of n equations:

$ÖΠ_1/Ö = f_1(t, Y_1, Y_2,.., Y_n)$
$ÖY_2/Ö=f_2 (t, Y_1, Y_2,..,Y_n)$
$ÖY_n/Ö=f_3 (t,Y_1,Y_2,é ,Y_n)$
The one-step integration across the ith interval is:
$Y_{j,i}+_1=Y_{ji}+hØ_j$
$=y_{ji}+h (k_{ji}+2k_{j2}+2k_{j3}+k_{j4})/6$
Where:

$K_{ji}=f_j (t_i, y_{1i}, y_{2i},é , y_{ni})$
$k_{j2}=f_j (t_i+½h, y_{1i}, y_{2i},.é \underline{y_{ni}})$
$k_{j3}=f_j (x_i + ½ h, y_{1i}, y_{2i},\overline{é}*, y \underline{_{ni,}} *)$
$k_{j4}=f_j (x_i + h, y_{1i}, \overline{y_{2i}},é \; y_{ni},)$
* * *

2.5 Model Validation

Results of the numerical computations were compared with experimental results obtained during a series of model validation tests under the controlled environmental conditions of a wind tunnel facility.

The wind tunnel was constructed to provide an experimental base for the validation of the mathematical model. The tests were carried out inside the test chamber of the wind tunnel at different heights, wind velocity and time intervals. The spray droplets were generated using a knapsack sprayer and were collected at different equidistant positions along the vertical and horizontal trajectories of the spray droplets.

After the spray droplet stains were collected on the chromatography papers, the papers each measuring 50cm x 40cm were cut into 5cm x 4cm units giving a total of 100 units. The stain sizes were placed under a microscope fitted with eyepiece micrometer for size or diameter measurement.

The chromatography paper has at its back water sensors connected to an alarm system. The alarm system indicates the time at which the water droplets touched the chromatography papers. For each cloud of spray droplets, 30 droplets were sized. At the end of the experiments, measurements such as the time-distance or flights-time, droplet initial diameter, droplet velocity, wind speed, wet and dry bulb temperatures, barometric pressures, droplet final diameter, etc were made. The experiments were carried out at two air conditions of still air and varying air velocity of 1.0 m/s.

3. RESULTS AND DISCUSSION

The results of the numerical computations and experimental validation tests are presented in Figures 1-4 and Tables 1-2. Figure 1 presents the comparison of the results of the simulation and validation experiments at still air condition. It indicates that 10 seconds after leaving the sprayer nozzle, the simulation predicted a size reduction range of 100 to 88.40 microns, while the experiment recorded a size reduction range of 100 to 92.00 microns. The trend is noticeable in the other droplet sizes of 150, 200 and 250 microns.

Figure 2 illustrates the comparison of the droplet size reductions as obtained from simulation and experiment at the wind velocity of 1.0 m/s. It is observed that droplet size reduction was generally higher at the wind velocity of 1.0 m/s than at the still air condition. Also, droplet size decreases as the air-borne time increases thereby increasing the rate of change of the spray droplet sizes. The observation is true for all the spray droplet sizes used for the validation experiments and was in line with similar studies (Miller and Hadfield 1989; Law and Bowen, 1975).

The two Figures (1&2) compare the droplet size reductions at still air and windy field conditions. The comparison indicates that droplet size reduction increased significantly as wind velocity increased from 0 m/s to 1.0 m/s. This could be explained using the kinetic theory of molecular diffusion. It is well known that liquids contain molecules which are in constant random motion. Each of these molecules moves at different random velocities. In a non-convective evaporation process, the diffusion of liquid molecules away from the droplet surface controls the the mass transfer rate of the droplets. At the surface of the liquid droplets, some molecules with sufficient kinetic energy may break through the liquid surface and encounter some molecules of air or vapor and as such may be forced to fall back to the droplet surface after the collision. When wind blows, the molecules of the air or vapor surrounding the droplet surface will be removed, clearing the way for the liquid molecules breaking through the surface of the droplet liquids. The faster the wind blows, the faster will be the rate of movement of the liquid molecules out of the droplet surface. Hence the faster will be the rate of evaporation of the liquid droplet sizes.

In the case of still air, the diffusion and random motion of the liquid molecules control the rate of loss of weight of the droplet. The absence of wind reduces the rate of removal of the air or vapor molecules surrounding the droplet surface. Therefore, the liquid molecules at the droplet surface will continue to collide with the molecules of air or vapor and hence forced to fall back to the liquid droplet surface with random velocities. The random collision of the molecules and their different random velocities caused the non- linear pattern of the percentage reduction in the sizes of the droplets at still air as shown in Figures 3 and 4.

Figures 3 and 4 compare the percentage size reductions of the simulated and experimental results at still air and wind velocity of 1.0 m/s respectively. Figure 3 indicates that at sill air, a 100 microns droplet size recorded a size reduction of 8% for the experimental results and 11.60 % for the simulated results. It is observed from Figure 3 that percentage size reduction increased as the droplet size decreased. For instance, the 100 microns recorded 8 % size reduction, while the 250 microns recorded 4.32 % reduction during the experimental validation tests. The result is in agreement with similar studies by Williamson and Threadgill (1974) Goering et al, (1972). The same trend holds for the simulated results, with a size decrease of 11.6 % for 100 microns and 7.78 % for 250 microns droplet sizes.

The reason for the observed increase in percentage size reduction as the droplet size decreases is based on the understanding that droplet size reduction due to evaporation is essentially a surface phenomenon, which depends on the ratio of surface area to volume. This ratio is low for large droplet sizes and high for small droplet sizes. The higher the value of the ratio, the higher the increase in percent size reduction and the lower the value of the ratio, the lower the percent size reduction.

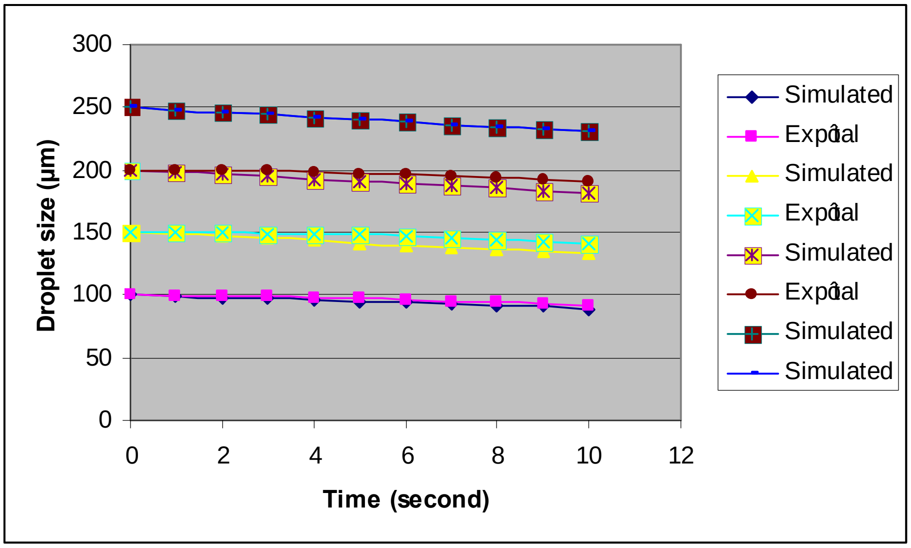

Fig.1: Comparison of droplet size reduction of simulated and experimental results at still air.

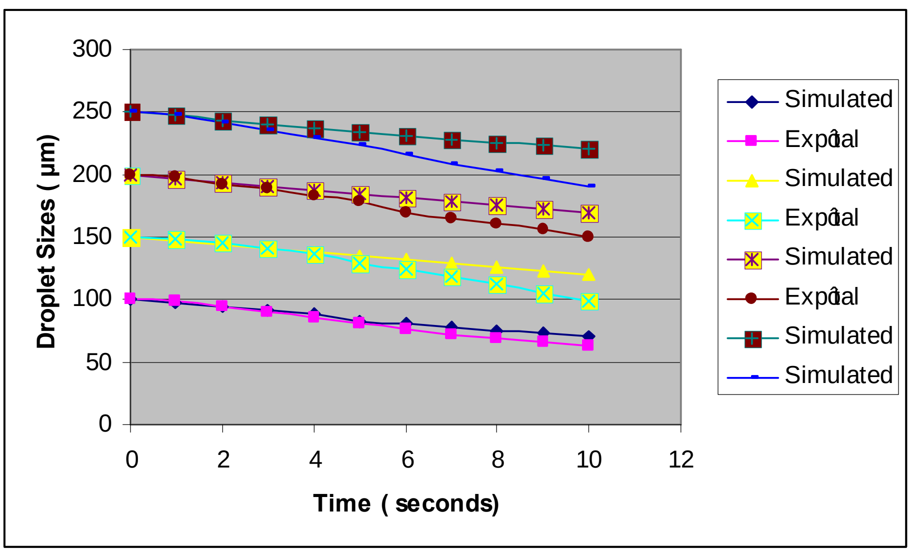

Fig.2: Comparison of droplet size reduction of simulated and experimental results at the wind velocity of 1.0m/s.

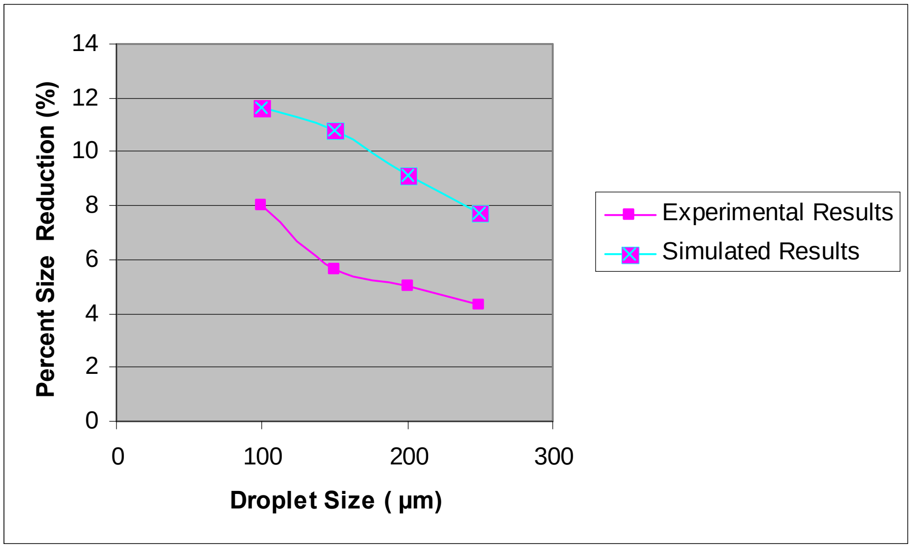

Fig.3: Percentage droplet size reduction of the simulated and experimental results at still air

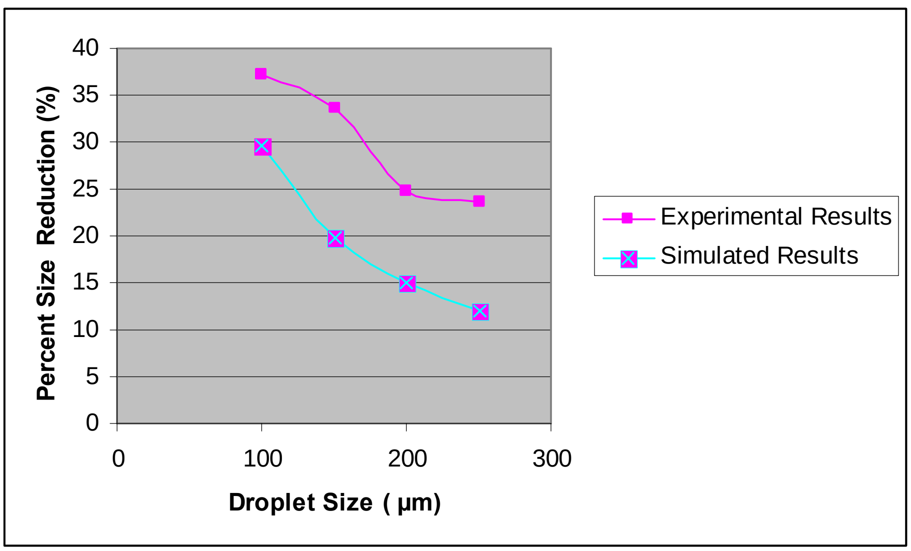

Fig. 4: Percentage droplet size reduction of the simulated and experimental results at wind velocity of 1.0 m/s.

Table 1: Range of droplet size reduction and percentage deviation of the simulated results from the experimental results at still air.

Original droplet size (microns)	Size reduction range for Experimental results(microns)	Size reduction range for Simulated results(microns)	Deviation of simulated results from Expŧal results (%)
100	100-92.00	100-88.40	3.60
150	150-141.50	150-133.83	5.12
200	200-190.00	200-181.81	4.09
250	250-239.20	250-230.55	3.46

Table 2: Range of droplet size reduction and percentage deviation of simulated results from the experimental results at the wind velocity of 1.0 m/s.

Original droplet size (microns)	Size reduction range for Experimental results (microns)	Size reduction range for Simulated results(microns)	Deviation of simulated results from Expŧal results (%)
100	100-62.80	100-70.48	7.68
150	150-99.48	150-120.20	13.82
200	200-150.25	200-170.01	9.88
250	250-191.20	250-219.88	11.48

With the percentage deviation range of 3.60-5.12 % for 100-250 microns respectively (Table 1) at still air and 7.68-11.48 for 100-250 microns respectively (Table 2) at the wind velocity of 1.0m/s, the simulated and experimental results compare fairly well. The comparison of the percentage droplet size reductions of the simulation and experimental results at the wind velocity of 1.0 m/s(Figure 4) reveals that droplet size reductions were higher for the experimental results than the simulated results. The experimental results recorded a 37.2 % size reduction for 100 microns and 23.52 % size reduction for 250 microns. In the case of the simulated results, 29.525 and 12.04 % size reductions were predicted for the 100 microns and 250 microns respectively.

The difference between the values of the experimental and simulation results is attributed to the intrinsic properties of air-vapor mixtures whereby the water molecules present in any air-vapor mixtures exert pressures on the surroundings and do not exactly follow the perfect gas laws (Singh and Heldman, 2005). However, the models were developed to follow the gas laws. Generally, when Figure 4 and 2 are compared with Figures 1 and 3, there is the indication that the model actually described and predicted size reduction of spray droplets at still air condition and at the wind velocity of 1.0 m/s. The good agreement of the simulated and experimental results justifies the inclusion of the latent heat of vaporization of the liquid droplets and the thermal conductivity of the transport medium (air) as important parameters in predicting air-borne size reduction phenomenon in liquid spray droplets. This should be so especially in a tropical environment where the ambient air temperature could be as high as 40 ^{o}C at certain periods of the year.

4. CONCLUSIONS

From the simulated and experimental results, the following conclusions were deduced:

At still air, the developed model predicted a percentage size decrease of 11.6, 10.78, 9.09 and 7.758 % for droplet sizes of 100, 150, 200 and 250 microns respectively after 10 seconds of being air-borne. The percent deviations of simulated results from validation results were 3.6, 5.12, 4.09 and 3.46 % for the droplet sizes of 100, 150, 200, 250 microns respectively.

In the case of windy environment at the wind speed of 1.0 m/s, the model predicted a percentage size reduction of 29.52, 19.86, 14.99 and 12.04 % for the droplet sizes of 100, 150, 200, and 250 microns respectively after 10 seconds of leaving the sprayer nozzle.

The percent deviation of simulated results from validation results indicates 7.68, 13.82, 9.88, and 11.48 % for the droplet sizes of 100, 150, 200, and 250 microns respectively

Under still air condition, small liquid droplets have longer air-borne time or retention time in air than the large ones. The longer the retention time, the more the evaporation rate which leads to decrease in droplet size and hence, the more chances of off-target deposition.

The size reduction of an air-borne spray droplet increases as the droplet retention time in air increase. This means that the distance between the spray nozzle and the target should be as small as possible to reduce the air-borne time. Wind velocity affects the evaporation of air-borne liquid droplets. The higher the wind velocity, the higher the rate at which droplet sizes decrease, implying that spraying at windy environment should be discouraged.

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TECHNOLOGIES FOR EXTRACTION OF OIL FROM OIL-BEARING AGRICULTURAL PRODUCTS: A REVIEW

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ABSTRACT

A critical appraisal of technologies for oil extraction from oil-bearing agricultural products is presented. Different types of oil- bearing agricultural products are discussed. The products include; groundnut, coconut, sheanut, castor, sunflower, sesame, oil-palm, etc. This work has also discussed the pre-processing conditions including the removal of hulls and shells, pre-processing conditioning such as size reduction, moisture content adjustment, heat treatment and pressure application, as well as the methods employed in the extraction, namely; traditional and modern (improved) methods. The improved method include; oil expeller, screw press, and solvent (chemical extraction). Problems (technical, socio-economic and institutional) associated with each method and the need for more research for the improvement of the methods are analysed. It has been shown that for any developing country to effectively adopt modern methods in the production of edible vegetable oils, improvement on the existing traditional methods, environmental factors, government policies, socio-economic and cultural considerations of the users need to be studied. This can be achieved through more research in the recommended area of need.

KEYWORDS: Vegetable oil extraction, oil bearing agricultural products, oil expression.

1. INTRODUCTION

Oil extraction is the process of recovering oil from oil-bearing agricultural products through manual, mechanical, or chemical extraction. The agricultural products are classified into oil-seeds (cotton, castor, sunflower, etc), nuts (coconut, groundnut, sheanut, etc) and mesocarps or fruits (oil palm). Plants bearing these agricultural products have greatly contributed to the economic development of many countries especially the development of West African countries where the products are grown for commercial purposes. Of the many types of oil that can be obtained from these products, only few are very significant in terms of world production and traded as major commodities (Robbelen *et al,* 1989). However, the oil extracted from these products have diverse domestic and industrial uses. The oil serves as a major source of vegetable oil that constitutes a good percentage of meal in the diets of common people. The oil as well as the by ïproducts are also very useful as food and non-food materials for the production of snacks, cake, margarine, biscuit, cosmetics, detergent, plastics, etc. Oil production is important not only among small-to-medium scale industrialists, but also to rural populace, employing quite a substantial workforce serving as a source of income to many communities engaging in the exercise (Abalu, 1978; UNIFEM, 1987).

Extraction of oil from oil-bearing products could be done in two major ways; traditional and improved methods. The traditional method is usually a manual process and involves preliminary processing and hand pressing. The improved method consists of chemical extraction and mechanical expression. The chemical extraction method requires the use of organic solvents to recover the oil from the products. Mechanical method involves the application of pressure to already pre-treated oil-bearing products. It employs the use of devices like screw and hydraulic presses as a means of applying the pressure (Gunstone and Norris, 1983). Other mechanical devices include oil expellers and improved ghanis which are used for seeds and nuts because of the high pressure required to express the oil (UNIFEM, 1993). Whichever method is employed, researchers (Norris, 1964; Ward, 1976; Khan and Hanna,

1983; Adekola, 1991) reported that the yields and quality of the oil extracted depend on the content adjustment, heating time, pressure application, operating temperature, etc.
Although processing of oil-seeds, nuts, and fruits for oil production is achieved by both traditional and improved techniques, oil extraction techniques have not changed significantly over the years as the bulk of this trade is still in the hands of rural women employing traditional systems only. Apart from discouraging many oil producers from continuing with the trade especially at old age, this difficult task also limits the capacity and oil yields of those determined to continue with the trade (IAR, 1992). Accessibility to modern equipment and spare parts, degree of complexity of the equipment, maintenance, and the availability of power source constitute other major setbacks.

In most of the developing countries, there has been a steady rise in the demand of edible oil both for domestic and industrial uses. Therefore, continuous review of existing methods of oil expression or extraction from oil-bearing agricultural products will no doubt continue to reveal the current state of the art especially on aspects that require further improvement. This in turn will sensitize engineers towards developing better machines and techniques to increase both the quality and quantity of oil yield to meet the increasing demand. It is to this end that this work has been conceived. Therefore, the specific objectives of this paper are: to review the techniques of oil extraction from oil-bearing agricultural products, and to analyse the constraints to and prospects for oil extraction techniques.

2. OIL- BEARING AGRICULTURAL PRODUCTS

2.1 Oil-Seeds and Nuts

Oil-bearing seeds and nuts are found in the roots, stems, fruits and leaves of some tropical and sub-tropical plants. They are mostly grown as annual crops and constitute the major source of vegetable oil for domestic and industrial uses. Some of the most common oil seeds and nuts cultivated in the tropics, subtropics and temperate regions include; groundnut, coconut, sheanut, castor, sunflower, sesame, oil palm, etc.

2.1.1 Groundnut (Arachis hypogaea L.)

Groundnut is the most common oil nut grown as an annual crop on about 19 million hectares in tropical and subtropical regions and in the warmer areas of temperate regions of the world, principally for its edible oil and protein rich kernels or seeds. It thrives best on sandy loam soil (Oke *et al*, 1975). It is considered to be the second most important source of vegetable oil in the world. In 1985, the recovered output of groundnut worldwide contained about 14.4 million tons of groundnut kernels, corresponding to the production of about 21 million tons in their shell (at shelling percentage of 70%) with about 7.2 million tons of oil (FAO, 1986).

The yield of kernels generally ranges from 0.5 to 4.0 tons per hectare. In developing countries where 80% of the crop is produced, the average yield is about 1 ton per hectare. The major world producers of groundnut are India, China, USA, Brazil, Senegal and Nigeria. The yield of about 900 kg/ha in India and 907 kg/ha in Nigeria have been reported (Irvine, 1969; Miller, 1983; Adeeko and Ajibola, 1990). According to the reports of TPI (1971) and Miller (1988), groundnut contains about 25% protein and 38 ï 50% oil in its seed.

2.1.2 Coconut (Cocos nucifer L.)

Coconut is cultivated all over the world in the humid tropics, mostly close to the seashore. The world leading producers of coconut are Philippines, Indonesia, India, Malaysia and Thailand. In tropical Africa, the crop is mostly produced in Mozambique, Cote dôIvore, Tanzania and Nigeria (de Neve de Roden et al., 2001).

The matured coconut plant produces fruits after pollination. The fruit is a drupe consisting of seed (coconut) covered by thick fibrous envelope or husk. The husk (mesocarp) is covered by thin leathery epidermis (exocarp). The coconut has hard thick shell (endocarp), which contains thick layer of firm, white, oily endosperm or albumen, called copra and a central cavity partly filled with sugary coconut water (Salunke and Desai, 1986).

World production of copra (dried coconut meat) has been estimated at 3.3 millions per year. This corresponds to 2.3 million tons of coconut oil (Essiamah, 1985). Green copra consist of 40% oil, 43% water and 17% non-oil products (de Neve de Roden *et al,* 2001). Studies have shown that oil contents of white meat coconut can be increased from 40% to 71% when dried to copra (Khan and Hanna, 1983; Adekola, 1992). The oil contains about 91% saturated fatty acids (44.1-51.3% lauric acid, and 5.4-9.5% coprylic acid, 4.5-9.7% capric acid 13.1-18.5% myristic acid, 7.5-10.5% palmitic acid and 1.0-3.7% stearic acid). The unsaturated fatty acid (9%) constitute about 5.0-8.2% oleic acid and 1.0-2.6% linoleic acid, while the copra cake contains 20% protein (de Neve de Roden *et al.*, 2001). The oil is used in margarine, baking, biscuit production and cooking. In addition, it is also used for making soap, detergent and candle. The by-product of the copra, the coconut cake, is sold as a valuable animal feed (TPI,1971; Adekola, 1992).

2.1.3 Sheanut (Butyrospermum paradoxum (gaertn.F.) Hepper)

In Africa, sheanut grows in the wild. The largest sheanut populations are found on the dry soil of Mali, Burkina Faso, Northern Togo, Ghana, Niger, Nigeria, Cote dô Ivoir and Benin. Burkina Faso could produce about 460,000 tons of sheanuts per year, while the annual production of Cote dô Ivoir and Mali range from 17,000 to 20,000 tons (Michael and Ban Koffi, 2001).

Vegetable fat (oil) called butter are extracted from the kernels. The butter (oil) contents range from 34 to 55% in a kernel with high content of saturated (palmitic and stearic) fatty acids in comparison with unsaturated oleic and linoleic) fatty acids. The average saturated / unsaturated fatty acids ratio is 60:40 (TPI, 1971; Michael and Ban Koffi, 2001). Sheanut butter is used in manufacturing chocolate, cosmetics and as a medicinal substance for skin treatment.

2.1.4 Castor (Ricinus Communis L.)

Castor is grown as an annual crop in Tropical and Mediterranean climate zones. The major producers of castor seed and oil are India , Brazil, China, USA and Thailand with small contribution from African countries such as Ethopia, Tanzania, South Africa, Kenya, Nigeria, etc. (Gobin *et al*., 2001).

At maturity castor plant produce fruits which are globular with at least 3- lobed capsules. Each capsule usually contains three shiny seeds. The seed consists of brittle testa, a copious endosperm and a small embryo with papery cotyledons.

The primary products of castor seed are oil and seed cake. The seed contains about 40 to 55% oil and 14 to 21% protein (TPI, 1971). The oil constitutes about 90% recinoleic acid of the entire fatty acid. The proteins (pomace) contain the toxic substances ricin and allergen (Gobin *et al*., 2001). The oil is used as a purgative in medicine and as an illuminant. In industries, castor oil is used for making cosmetics, plastics, resins, dyes, paints and lubricants. The pomace is widely used as organic fertilizer (Gobin *et al*., 2001).

2.1.5 Sunflower (Helianthus annuus L.)

Sunflower is an annual crop that grows well in many regions of the world. The largest producers of sunflower seeds are ex-USSR, Argentina, France, Spain, China, India and USA. While in tropical Africa, the major producers are Zimbabwe, Sudan, Tanzania, Angola and Malawi (Ravagnan, 2001).

After pollination, the sunflower plant produces fruit consisting of a seed (kernel) and an adhering pericarp (hull). The average seed yield of traditional open-pollinated cultivars on small scale farms is

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about 500kg/ha. Well managed crops of improved varieties have yield of 1.5 tons/ha with an oil yield of 700kg/ha (Ravagnan, 2001). The seed consists of seed coat, endosperm and embryo.
The seed contains about 25 to 40% oil, 40 to 60% protein and 10% to 14% fibre (TPI, 1971; Ravagnan, 2001). The oil contains about 90% unsaturated fatty acids and 10%saturated fatty acids of which are oleic and linoleic acids. Sunflower oil is used in the production of margarine, salad oil, cooking oil, as well as non semi-drying oil in paints (Ravagnan, 2001).

2.1.6 Sesame (sesamu indicum L)

Sesame is an annual plant grown in tropical and sub ï tropical regions. It is probably the most ancient oil seed known and used by mankind. It is stated that sesame has its origin in Africa and spread through west Asia, India, China and Japan (Kafiriti and Deckers, 2001).

Sesame is grown as an intercrop system or pure stand. Upon maturity, the crop produces fruits which are erect capsules. Each capsule has about 70 seeds. Under good management, seed yield can be as high as 3000kg/ha. A yield of 2000kg/ha is considered necessary for profitable commercial production (Kafiriti and Deckers, 2001).

Sesame seed contains about 45 to 55% edible oil, 19 to 25% protein and 5% water (TPI,1971; Kasfiriti and Deckers, 2001). The oil is used in industries for manufacture of margarine, cooking fats, soaps, paints, lubricants, illumnants and insecticides, as well as for medicinal drugs (Kafiriti and Deckers, 2001).

2.1.7 Other Oil –Seeds and Nuts

Other oil-seeds and nuts include cotton, melon, breadfruit, neem, coconut, rubber, mustard, lin, cashewnut, palm kernel nut, etc., each contributing about 3.5 to 5.1 million tones of oil world-wide in a year, depending upon the amount of the products being produced (FAO, 1986). They also have oil content in the range of 15 to 30% in their respective seeds and nuts and are mostly glycerides of unsaturated fatty acids oleic acids with linoleic acids, which are useful for industrial and domestic purposes (TPI, 1971; Fasina and Ajibola, 1989). Table 1 shows the percentage of oil content in some oil- seeds and nuts and their uses.

Table 1: Percentage of oil content in some oil –seeds and nuts and their uses

Agricultural Product	Oil content (%)	Uses
Oil Seeds		
Castor	35 ï 55	Paints, lubricants,
Cotton	15 ï 25	Cooking oil, soap making
Linseed	35 ï 44	Paints, varnishes
Niger	38 ï 50	Cooking oil, soap, paint
Neem Kernel	45	Soap making
Rape / Mustard	40 ï 45	Cooking oil
Sesame	35 ï 50	Cooking oil
Sunflower	25 ï 40	Cooking oil, soap making
Nuts		
Coconuts: Dried copra Fresh nut	 64 35	Cooking oil, body / hair cream, soap
Groundnuts	35 ï 50	Cooking oil, soap making
Palm kernel nuts	46 ï 57	Cooking oil, body / hair cream, soap making.
Sheanut	34 ï 44	Cooking oil, soap making.

Source: TPI (1971)

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2.2 Oil - Mesocarps

These are oil-bearing fruits mostly grown in tropical regions of Africa. The most common oil fruit grown is oil palm (Elaesis guineensis) which is found growing in forest and swampy areas. The main producing countries are India, Malaysia, Indonesia, Nigeria, Zaire, Sierra Leone, Cameroon, Angola, Ivory Coast and Congo.

The oil palm is a single-stemmed plant growing to a height of about 8 m or more at maturity. The leaves or fronds are arranged spirally on the trunk and it has an extensive root system. However, the ovaries of the female inflorences of the oil palm develop into large bunches and each bunch may contain about 800 to 1000 fruits. Each fruit (Fig.7) is a drupe. The drupe pericarp is made of three layers an outer exocarp, a middle fibrous mesocarp and an inner endocarp (shell). The kernel comprises of a testa (skin) a solid endosperm and embryo.

Palm oil is the major product obtained from the oil palm fruit and it constitutes one of the main sources of edible oil with vitamin A precursors (ECA, 1983). It is also used for industrial purposes such as in the manufacture of soap, margarine, candles, etc.
FAO (1986) also reported that in 1985, the recorded worldwide production of oil from oil palm fruits ranged from 3.5-5.1 million tons. The percentage of oil content in the oil palm fruit range from 42-56% (TPI, 1971; Khan and Hana, 1983; Adekola, 1989).

2.3 Composition of Oil in Some Oil-Bearing Agricultural Products

Groundnut and oil palm are the two major oil crops grown in Nigeria and most other African countries. A fully matured harvested groundnut seed and oil palm fruits consist of oil, protein, carbohydrate, fatty acid and water in different composition (Tables 2 and 3).

Table 2 Composition per 100g edible portion of dried groundnut seed

Content	Percentage/weight
Oil	38-50%
Water	7.3%
Protein	23.4g
Fat	45.3g
Carbohydrate	21.6g
Ash	2.4g
Calcium	5.8g
Iron	2.2mg
Thiamin	1.0mg
Riboflavin	0.13mg
Nicotinic acid	16.mg

Source: FAO, 1982.

Table 3 Composition of palm kernel fruit

Content	Percentage (%)
Oil	47-50
Protein	7.5-9.0
Extractable non- Nitrogenous	23-24
Cellulose	5.0
Ash	2.0
Water	7.5-9.0

Source: Hartley, 1971.

It is clearly indicated that the percentages of oil content in the groundnut seed and palm kernel fruit as shown in Tables 2 and 3 respectively, is higher than the other components which implies that these products contain oils in large quantities. Therefore, proper choice and selection of an efficient expression methods and /or devices will be required to express-oils from these products in order to obtain high oil yields. The oils being expressed from these products contain some percentages of acid concentrations and other constituents in various compositions as shown in Tables 4 and 5 respectively. Thus, due to the presence of high concentration of acids and other constituents in the oils, the oils being expressed need to undergo clarification and refining processes.

Table 4: Composition of Groundnut Oil

Oil content	Percentage (%)
Percent Saturation	17.7
Percent Oleic	5.65
Percent Linoleic	25.8

Source: Hartley, 1971

Table 5: Composition of Palm Kernel Oil

Content	Percentage (%)
Free fatty acid	3.0 ï 4.0
Volatile matter include water	0.15 ï 0.20
Impurities	0.05 ï 0.10
Peroxide (Inch Equivalent, kg)	2.0 ï 4.0
Saponification value	242.0 ï 222.0

Source: Hartley, 1971

3. OIL EXTRACTION TECHNOLOGIES

3.1 Pre-Extraction Process

Oil-bearing agricultural products need to undergo certain preliminary processing to get the seeds or nuts ready for oil extraction. In some cases, oil expression devices are assembled as a complete unit with pre-processing equipment included. This preliminary processing aids is easy and efficient oil extraction, and involves removal of hulls and shells and pre-treatment conditioning such as size reduction, optimum moisture content adjustment, heat treatment and pressure application.

3.1.1 Removal of Hulls and Shells

This involves cleaning (removal of foreign matter) and decorticating (removal of hulls and shells). The cleaning process consists of the removal of sand, stalks, plant debris, and any other foreign matter in the seeds or nuts. The cleaning can be done by rotary or table sieves, usually with air separators by fans, and cyclones for dust removal from the air. According to Galloway (1976), mechanical system that consists of sieves and/ or shakers, both with perforated metal or screen are employed. In these machines, the material flows over the drum or tray, thus cleaning the seeds or nuts.

Decorticating or shelling is the process of freeing oil-seeds and nuts from the shells or pods by cracking the shell using a device known as decorticator. The decorticators are of two major types, hand-operated and power decorticators, both work with the same principle. They consist of the following main units; loading, beating and separating units. The seeds and nuts are loaded in the loading unit and passed to the beating unit where they are cut and cracked or broken by rotary action of the beater bars. The decorticating operation is then followed by separation of the cracked seeds by means of sieves and air separators.

3.1.2 Pre-treatment Conditioning

Pre-treatment conditioning is a preliminary processing activity that involves size reduction, moisture content adjustment, heat treatment and pressure application. These activities depend on the nature of the oil-bearing material and the methods and devices to be adopted in the oil extraction.

The purpose of size reduction is to expose a greater area of oil-bearing cells to the moisture and heat during cooking. However, excessive size reduction is not desirable as it reduces extraction efficiency. Higher oil yields were obtained from coarsely ground groundnuts compared to the finely round samples (Ward, 1976; Adeeko and Ajibola 1994, and Hanna, 1984). Finely ground melon seeds on the other hand, gave higher oil yield than the coarsely ground samples (Ajibola et al., 1990).

The moisture content of seeds is an important factor that affects the yield and quality of the oil extracted. Cloudy oil is obtained from seeds with high moisture level; therefore, moisture adjustment of the seed is necessary before pressing. For example, a moisture content of 6% (wet basis) was found to be optimum for extraction of oil from peanuts and sunflower seeds (Bongiwar et al., 1977; Singh et al., 1984).

Oil-seeds are subjected to heat treatment in order to lower the viscosity of the oil to be extracted, coagulate the protein in the meal, and also adjust the moisture level of the meal to the optimum level of extraction. Sivakurarah et al. (1985) reported that heating temperature, heating time and moisture content were interactive factors that influenced the yield of oil extracted. Adeeko and Ajibola (1990) also observed that prolonged heating time beyond 25 minutes for groundnut samples heated at temperature above 90^{o}C did not improve oil yield. Pressure has also been shown to have significant effect on oil yield (Dedio and Dorrell, 1977; Adeeko and Ajibola, 1990).

3.2 Traditional Oil-Extraction Methods

Before attempts are made to introduce improved methods of oil expression/ extraction, effort should be made to understand the traditional methods employed. Improved technologies, which are not based on an understanding of traditional processing, tend to have low acceptance rate. The various steps involved in traditional methods of processing, differ somewhat from place to place, thus it will not be feasible to record all the minor variations. Therefore, examples are given of fairly standard processing methods which can serve as a basis for comparison with the system used in any particular area. the procedure for the extraction is divided according to raw materials/agricultural products being used.

3.2.1 Oil-Seeds

Oil-seeds (cotton, castor, sunflower, etc.) in most cases, are ground to a paste without removing the husk or outer covering. In some instances sunflower seeds are husked. The seeds are ground manually and the paste is heated alone at first and then with boiling water. The mixture is stirred and brought to boil. After boiling, the mixture is allowed to cool and the oil settles at the top and is scooped off. With this method of processing, the extraction efficiency is about 40%, that is percentage of oil extracted based on the total theoretical content. (NRI, Unpublished information).

3.2.2 Oil-Nuts

The processing methods of oil-nuts vary because of the variation in the procedures. The most common oil-nuts grown in most countries are groundnuts and coconuts. Groundnuts are shelled, cleaned and roasted lightly. The roasted nuts are skinned by placing them on a mat and rolling a wooden block over them, and winnowing them to separate the skin from the nuts. The skinned nuts may be pounded with a mortar and pestle or ground using grinding stones to a smooth paste. The paste is kneaded and pressed by hand to remove the oil-water mixture. Then the oil-water mixture is fried to remove most of the water.

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3.3.1 Oil-Expellers

Expellers use a horizontally rotating metal screw, which feeds oil-bearing products into a barrel shaped outer casing with perforated walls. The products are continuously fed to the expeller, which grinds, crushed and presses the oil out as it passes through the machine. The pressure ruptures the oil cells in the product and oil flows through the perforations in the casing and is collected in a trough underneath (Gate, 1979). The residue of the material from which oil has been expressed exits from the unit, and is known as the cake. With some types of expellers takes place. This allows for greater oil expression and reduces wear and tear on the machine.

Most expellers are power-driven, and are able to process between 8 and 45 kg per hour of product depending upon the type of expeller used. Bigger units processing greater quantities of oil are available for use in larger mills. The percentage of oil expressed by expellers is as high as 90% depending upon the type and kind of products as well as the expeller being employed (Gate, 1979).

The friction created by the products being expressed wears down the worm shaft and other internal parts. With small machines this occurs often after expressing as little as 50 tons, after which parts must be replaced or repaired through resurfacing by welding. Maintenance of an oil expeller, therefore, calls for machinery and equipment rarely found in small repair shops and local manufacture of expellers would be most unlikely at the village /small town level.

3.3.2 Oil-Plate Presses

Plate presses are normally used for expressing oil from mesocarp (fruits) like oil palm, but depending upon the amount of pressure applied, oil-seeds and nuts can also be expressed. With oil plate presses, the expression efficiency is about 90%, depending upon the nature and the amount of product being expressed (Gate, 1979). Generally, oil-plate presses are of two types; screw presses and hydraulic presses.

In screw press that is manually operated, the substance from which the oil is to be expressed is pressed slowly and with maximum pressure by plunger (round steel plate), forced down by a screw, and into a cylinder with a large number of small holes (Gate, 1979). Capacities of the screw press depend upon the size of the cage which holds the products, an average being about 15 kg per batch. The screw press consists of worm shaft, cage chain drive speed, bearings and other accessories.

In hydraulic press that can be manually or power operated, pressure is exerted by hydraulic device such as a lorry jack. They require a heavy, rigid frame structure. Because of the weight of such a structure the press must be stationery and cannot be moved as easily as a screw press. Hydraulic presses can process mesocarp (fruits), oil-seeds and nuts as they generate greater pressure than a screw essential to ensure that hydraulic fluid, which may be toxic, does not come into contact with the food stuff (Gate, 1979).

In most cases, oil presses can be manufactured locally in rural areas with the exception of the screw which needs a special lathe. It is generally recommended that the nut (through which the screw operates) should be of softer metal so that it will be subject to wear and tear rather than the screw, which is expensive to replace or repair. Hydraulic press can be manufactured locally if lorry jacks are available. The screw presses also exist in different types and makes some of which are TCC press and Kit spindle press.

3.3.3 Improved Ghanis

Ghanis originated in India where they are primarily used to express oil from mustard and sesame seeds, although in some cases they can be used for coconut and groundnut processing. Traditionally ghanis are operated by animals and can be manufactured locally. They consist of a wooden mortar and wood or stone pestle. The mortar is fixed to the ground while the pestle, driven by one or a pair of

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bullocks or draught animals is located in the mortar where the seeds are crushed by friction and pressure. Oil runs through a hole at the bottom of the mortar while the residue or cake is scoop out. Depending on the size of mortar and type of seeds, an animal ï powered ghani can express about 10 kg of seeds every two hours.

Mechanized versions of the traditional animal-powered ghanis are common. In these power-driven ghanis the pestle and mortar units are usually arranged in pairs with either the pestle or mortar held stationary while the other is rotated. Power ghanis have a greater capacity than the traditional ghanis and can process about 1000 kg of seed per day (Srikanta Rao, 1978). Power ghanis yield an oil with a lower pungency.

3.4 Solvent Extraction

Solvent extraction method involves the use of organic solvents such as straight chain hydrocarbons, chlorinated hydrocarbons, alcohols and ketones to recover the oil from the sources. The process for solvent extraction of nut (groundnut) is similar to that of seeds (soyabean, cotton, etc). Generally, nuts or seeds are shelled and winnowed to remove fibre-rich shells, and whitened by removing the tannin-containing the skins. Next, the nuts or seeds are cracked into piece and conditioned 10-11 percent moisture at 70^{o}C or more, an then flaked by passing through rolls. Sometimes the nuts or seeds flakes are cooked before they are converged to the extractor. In the extractor, the oil is removed by means of a solvent. The solvent ladden flakes are then passed through a desolventizer, which recovers the solvents. The defatted and desolventized cake may undergo further treatment before it is used as feed. The crude oil may be clarified by passing it through a filter press.

Solvent extraction is capable of removing nearly all the available oil from oil-seeds or nuts. About 98% of the oil is being extracted by solvent method. (Ngoddy and Iherokoronye, 1985; Cecoco, 1988). In addition to the high yield of oil, the method produces oil with better qualities, and a higher protein meal (Khan and Hanna, 1983). The method generally requires more capital expenditure, and refining the oil before use. There is also possibilities of toxicity from the solvent used and danger of fire explosion from the use of volatile organic solvent.

3.5 Clarification and Refining of Oil Extracted

The extracted oil contains a suspension of finely divided seed debris, which needs to be cleared. Clear oil can be obtained by allowing the oil to stand for a few hours or days in a clean container. The finely seed debris then settle at the bottom of the container leaving an upper layer of clear oil which can be decanted. Alternatively, the oil should be filtered through a filter press, or locally by using a plastic funnel fitted with a cloth. The refining process is basically the same for most vegetable oil. The procedure can also be applied to crude oil expressed using a hydraulic or screw press. Refining consists of alkali-refined, to neutralize the free fatty acids, using sodium hydroxide, bleaching, to improve flavour stability and odour, using steam distillation under vacuum.

4. PROBLEMS AND RESEARCH NEEDS

4.1 Technical Problems

In the traditional methods of oil extraction from oil-bearing agricultural products, almost all the stages involved are manual and labour intensive. The processing stages are therefore, strenuous, arduous, tiring, time consuming and particularly inefficient. Also, some of the operations have been reported to be hazardous to human health. The crude nature of the processing procedures, no doubt affects both the quantity and quality of oil recovered at the end. This is as a result of the inefficiency of the traditional methods and techniques. Often times a certain percentage of the oil extracted is found containing some quantity of free fatty acid, water, etc, resulting from the chemical breakdown of oil giving it unpalatable flavour.

Although the improved methods provide greater efficiency in oil extracted (high oil yield), they have a lot of problems. Technologies (oil-expeller and plate presses) have normally been introduced with the intention of improving traditional methods. However, many have had limited success because they have tended to ignore important considerations such as friction created by the products being expressed which wears downs the worm shaft and other parts, the degree of complexity of the equipment and its maintenance, accessibility of the spare parts, and availability of the power source inputs (fuels, diesel, water and electricity) for powering the mills. The oil being produced also contains some debris, high percentage of free fatty acid and sometimes mixed with the hydraulic fluid or fuel being used.

4.2 Socio-Economic Problems

It has been discovered that both those who use traditional processing methods as well as those who use modern technology often run into the problem of selling their oil at lower (uneconomic) prices in order to meet up with urgent needs. This occurrence has been noted to be as a result of the level of abject poverty of most of the farmers who produce oil. No doubt, the effect of this is multiplied. The farmer incurs losses, is not able to pay for the next production, equipment repairs and maintenance and eventually runs at a lost. All these will result in cumulative low quantity and poor quality of oil produced. In addition, certain technological innovations do not go down well with people in some cultural and environmental settings, either due to their inherent beliefs, superstitions or the local environment, which are not conducive or very difficult to handle.

Furthermore, another major problem, which need to be considered, is that the technology should be such that the local producers (mostly women) should be able to afford it or have access to credit, otherwise they will be reluctant to accept it particularly if it is capital intensive. The technology should be able to pay for itself in a given time through sales of its products and even create a secondary enterprise by making use of the by-products. Moreover, the improved technologies have an adverse effect on local producers because some of them produce and sell at loss, since the returns from oil, produced do not justify the cost of production.

4.3 Institutional Problems

Local farmers who are into agricultural oil production have also suffered serious setbacks in that the support services which they should get from institutions such as Agricultural Development Programmes (ADP), Ministries of Agriculture, government and private functionaries are either not there at all or denied of them by some unscrupulous officers. Also credit facilities which should aid the farmers in improving their production are either not in existence or the farmers may lack the collaterals required to benefit from such credit facilities.

Furthermore, there is an inadequate provision or maintenance of the existing research centres, either from the universities or other related institutions that are actively in production and processing of oil-bearing agricultural products. Lack of careful and strategic planning such as the provision of training through the extension workers in the use and maintenance of equipment, in order to make new or improved technologies understood and accessible to the rural farmers added to the problems. Government policies have not also addressed specifically the problems of these local producers of agricultural oils.

4.4 Research Needs

In view of the considerable range of technologies and devices for traditional and improved methods of oil expression / extraction and processing, selection of the most appropriate method for a given environment requires a research into traditional methods employed, in comparison to the improved methods under a given condition. The research study under the traditional method should cover areas that will improve the equipment and techniques being used, pre-processing conditions, socio-economic problems and the existing government policies.

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In introducing new technologies, it is important to make comparisons between the proposed improved technology and traditional methods existing. Introduction of any new improved methods to rural areas requires a thorough understanding of the socio-economic and cultural relations of the users. Research into the socio-economic needs should be conducted. A careful study and examination of the above factors can help to determine which of the various stages of oil extraction require improvement as well as offering maximum benefits to the intended beneficiaries. Therefore, continuous research effort and efficient use of human resources and technical inputs are essential pre-requisites for improving the yield and quality of the oil being extracted from oil-bearing agricultural products.

Besides this, there is a need to establish an institutional framework of network of all those involved either in agricultural oil research or production (such as NIFOR, Risonpalm, Sunseed, etc.) to enhance net working and easy access to useful research findings or developments in this area. Such a network will no doubt help in reducing or possibly eliminating the problems of slow rate of development and repetition of the same mistakes, which others have made and improved upon.

5. CONCLUSION

The extraction of oil from oil-bearing products has been reviewed. It is obvious that the oil yield from these products depends on two major factors, that is pre-processing conditions and the methods or techniques being adopted. It has been observed that higher oil yields could be obtained from oil bearing agricultural products that are well pre-processed such as removal of hulls and shells, heating, moisture content adjustments, etc. Considering all other factors, the quantity and quality of the oils being expressed / extracted by improved methods are better compared to the traditional methods.

However, the adoption of simple machines by small oil producers needs more attention before Nigerian and other developing African community-based Agro-allied activities can be viable and sustainable. The adoption of these simple machines will no doubt develop the capacity of local artisans in the oil extraction business who are quite talented but hardly diversify due to lack of new and adaptable technologies that they can copy and add to their cost of productions.

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COMPONENT DESIGN OF A 187kg/hr LOW-COST EXTRUSION COOKER FOR FULL FAT SOYMEAL

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ABSTRACT

The design of a motorized low-cost extrusion cooker for full fat soy-meal production from soybean is presented. The machine was designed based on the high shear single screw extrusion systems, with segmented screws, which has three major zones: feeding zone, compression zone, and melting zone. It has a capacity of 187kg/hr (1.5 tons/day), power requirement of 20hp, screw rotation of 450rpm and a cost of seven hundred thousand Naira (₦700,000.00) as against two million Naira (₦2,000,000.00) for imported unit. The Chemical and Microbial analysis of the full fat soy meal satisfies the standard (BIS).

KEY WORDS: Extrusion cooker, design, gelatinization, extrudate, full fat, shear, puff, soy-meal.

1. INTRODUCTION

An extruder is a machine, which shapes materials by the process of extrusion. Extrusion cooking combines the heating of food product with the act of extrusion to create a cooked and shaped food product. Cooking is accomplished through the application of heat, either directly by steam injection or indirectly through jacket and by dissipation of the mechanical energy through shearing occurring within the dough (Harper, 1979). The results of cooking the feed ingredient during the extrusion process are the gelatinization of starch, the denaturation of protein, the inactivation of many raw food enzymes which can cause food deterioration during storage, the destruction of naturally occurring toxic substance such as trypsin inhibitors in soybeans, and the diminishing of microbial counts in the final product (Harper, 1979, Osborne and Mendel, 1917). Extrusion cooking tends to maximize the beneficial effect of heating foods, improved digestibility and precooking, while minimizing the detrimental effect (browning, inactivation of vitamins and essential amino acids, production of off-flavors, etc).

Proper extrusion cooking can only be achieved with proper design of the various extruder components. Extrusion cooker consists of a flighted screw(s) or worm(s) rotating within a sleeve or barrel. The action of the flights on the screw pushes the plasticized material forward. As the moist hot material moves through the extruder, the pressure within the barrel increases due to restriction at the discharge end of the barrel. At this elevated pressures, boiling and flashing of moisture does not occur within the confines of barrel because the pressure exceeds the vapor pressure of the water at the extrusion temperature. Once the food emerges from the die, the pressure is released causing the product to expand with the flashing of moisture. The loss of moisture from the product results in adiabatic cooling of the food materials, which solidifies and sets, often retaining its expanded shape.

There is an ever-growing demand for soybean products in the market, especially full fat soy meal. This product formed using extrusion cooker is used in animal production, and could be consumed by man. Imported extrusion cookers are very expensive with attendant operational complexities, hence the need to develop a low cost extrusion cooker with minimal operational requirement.

The objective of this work was to develop a low cost extrusion cooker locally. The machine was designed based on the high shear single screw extrusion systems, with segmented screws, which has three major zones, Feeding zone, Compression zone, and Melting zone.

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2. MATERIALS AND METHODS

2.1 Description of the Extrusion Cooker

The extrusion cooker consists of a flighted Archimedes single screw, which rotates in a tightly fitting cylindrical barrel by the aid of an electric motor. The raw soybean is introduced into the feed end of the extrusion screw from the hopper by gravity discharge. The action of the flights on the screw push the products forward and in so doing, work and mix the constituents into viscous dough like mass. Heat is generated for cooking the dough by viscous dissipation of mechanical energy. As the moist hot material moves through the extruder, the pressure within the barrel increases due to restriction along the process line (steam lock) and at the discharge end of the barrel. Once the food emerges from the die, the pressure is released, causing the product to expand with flashing of moisture. This loss of moisture from the product results in adiabatic cooling of the product, which solidifies and sets, often retaining its expanded shape.

Because the flights are usually full, the food product is subjected to high shear rates as it is conveyed and flows by the action of the screw. This results in the extruded foodôs unique texture. The porous, rigid structured product is then milled, sieved and bagged or used directly.

The major components of the extrusion cooker are; the extruder drive seed assembly, extrusion unit and extruder discharge.

The extruder drive comprises of electric motor, power transmission shaft, pulley, and transmission belt. The feed assembly comprises of a hopper, and feed control. The extrusion unit comprises of power screws, steam locks and barrel with grooved wall. The extruder discharge comprises of the die, nose cap and nose cone.

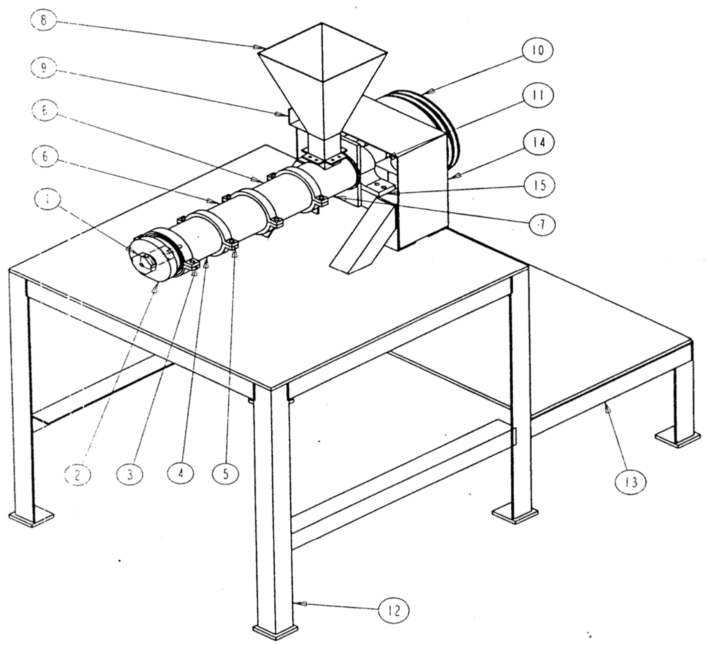
The machine offers many basic design advantages that enable it to be used for minimizing energy and process costs, like versatility, high productivity, low cost, ability to produce irregular shapes, high product quality, production of new foods and non-production of effluents etc. Figure 1 shows the low cost extrusion cooker. Description of the components follow.

2.2 The Hopper

The hopper houses the soybean kernels to be extruded and from here the kernels are fed into the extrusion unit at a regulated rate. The hopper as shown in Figure 2 has a shape, which facilitates loading, maximum volume utilization and reliable and complete gravity discharge through its outlet.

Volume of Hopper = $^{1}/_{3}$ HWL + hL = 19.0×10^{-3} m^3. é é é .(1)
Assuming a soybean density of 786.86kg/m^3. The weight of soybean it can hold
= 14.0kg.
Weight of hopper = Volume of hopper material × density of steel (Galvanized steel)
.é é ..(2)
= 1.18kg.

3



1 ï NOSE CONE
2 ï NOSE CAP
3 ï BRACKETS
4 ï METERING BARREL
5 - LOCK NUT
6 ï COMPRESSION BARREL
7 ï FEEDING BARREL
8 ï HOPPER
9 ï FEED CONTROL
10 - 4-WAY PULLEY
11 - THRUST BEARING
12 - MACHINE STAND
13 - MOTOR STAND
14 - BEARING HOUSING
15 - BEARING SUPPORT

Figure 1. The low-cost extrusion cooker

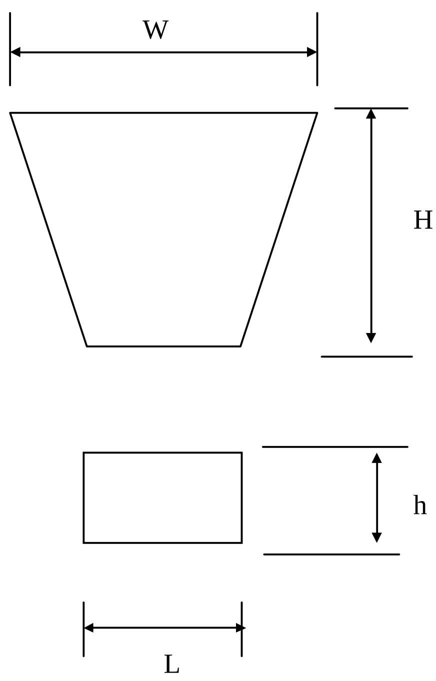


Figure 2 Side view of the hopper

2.3 The Screw

This is the central portion of the food extruder. It accepts the feed ingredient at the feed port, conveys, works, and forces them through the die restriction at the discharge. The screw is divided into

three sections whose names correspond to the function each section performs as shown in figures 1 and 3.
The screw as shown in Figure 4 has the following parameters.

Actual screw diameter, D_s = D ï 2δ = 98cm. é é é (3)

Where δ is the clearance between the barrel and screw.
Flight height, H_s = H - δ = 0.7cm é é é (4)
Root diameter, Dr = D ï 2H = Ds ï 2Hs = 8.4cm é é ..(5)

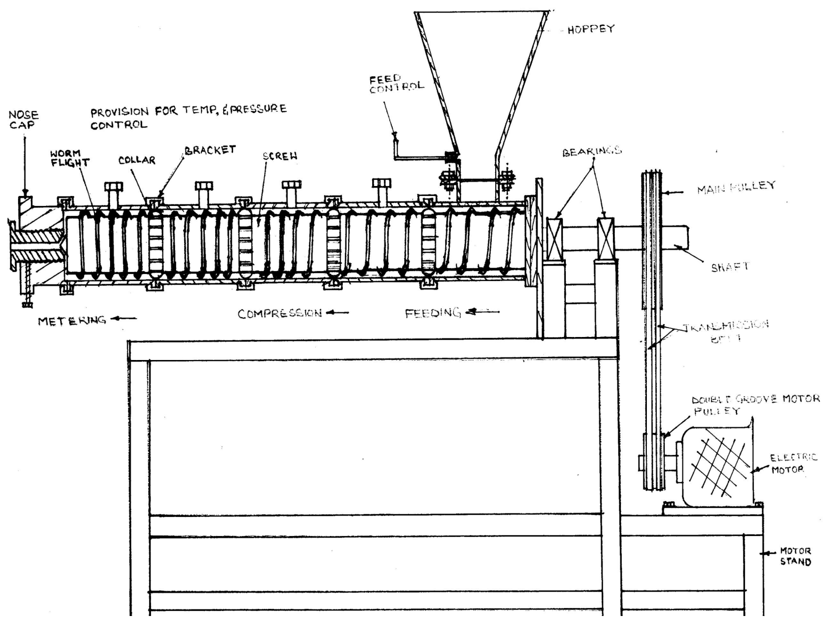


Figure 3. Sectional view of the extruder.

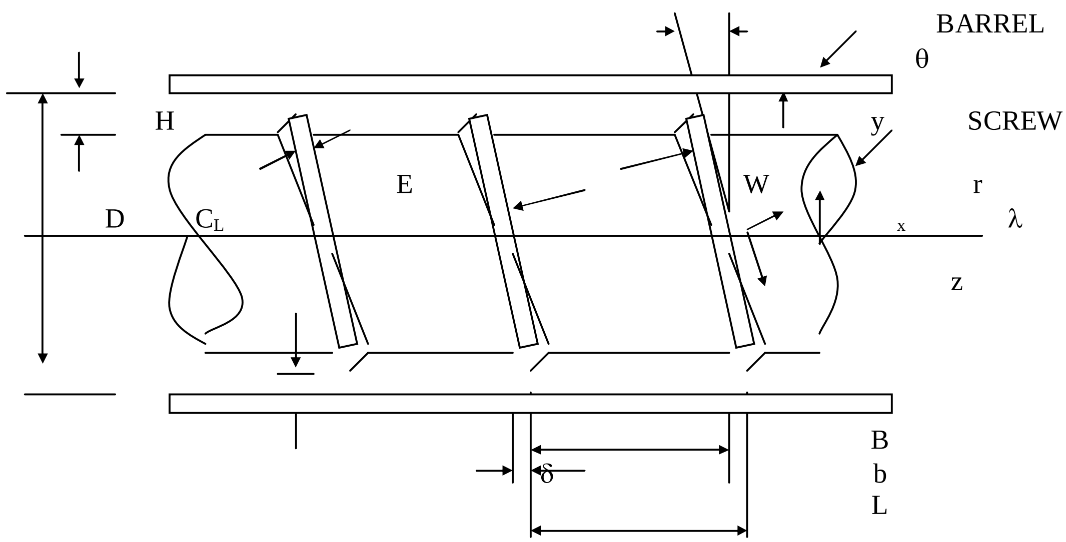


Figure 4 Geometry of the metering section of an extrusion screw

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Lead, the axial distance from the leading edge of a flight at its outside diameter to the leading edge of the same flight in front of it, L = 1.94cm

Helix angle, $\theta = \tan^{-1}(L/\pi D_s)$ = 3.6° é é .(6)

$\theta = \tan^{-1}(L/\pi D_s) \cong \tan^{-1}(L/\pi D)$, for δ less than 0.1mm.

Axial channel width, B = 1.5cm

Channel width, $W = B/\cos\theta$ = 1.503cm é é .. (7)

Axial flight width, b = L ï B = 0.44cm é é .. (8)

Flight width, $e = b\cos\theta$ = 0.439cm. é é .. (9)

Peripheral speed or tip velocity, $V = \pi D_s N \cong \pi DN$ = 2.3 m/s é é . (10)

Height to diameter ratio, H/D = 0.07

Volume of screw = 1.75 × 10 ï3 m^3

Weight of one screw = ρ x v =13.65kg. é .é (11)

2.4 Extruder Barrel

This is the cylindrical member which fits tightly around the rotating extruder screw as shown in Figure 5. The barrel is segmented. This makes it easy to alter the interior configuration of the barrel and to replace the discharge section, which wears the most rapidly. Harper, (1980) and Patil,, (1999), reported that the internal diameter, D, ranges from 5cm to 25.4cm.

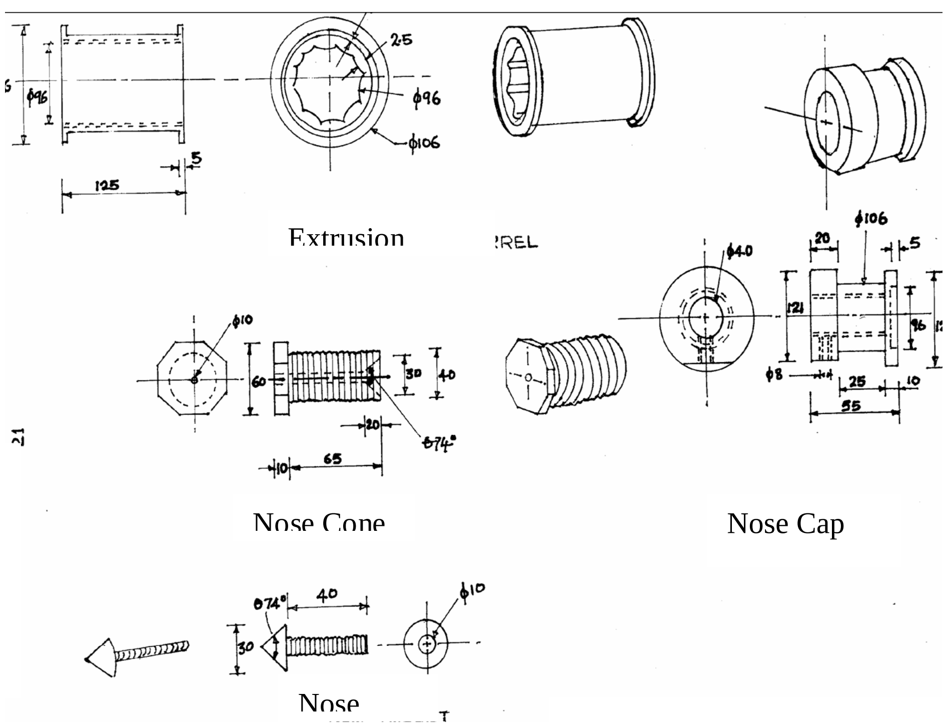

Figure 5. Other components of the extruder

Length to diameter ratio, L/D: Food extruders typically have L/D ranging from 1:1 to 20:1. The designed extruder has L/D = 5.56 : 1. The barrel was designed with grooved walls to prevent slippage of the material against back pressures.

The designed Width of groove is 0.5cm, while the Height is 0.2cm. Approximate volumetric capacity of barrel available for feed = 5.66 x $10^{-4}m^3$

Thickness of barrel wall = 0.5cm.
Length of barrel = 54.5cm
Diameter of barrel = 9.8cm
Weight of one barrel = 3.42kg (calculated)
Total weight of barrels = 13.69kg.

By symmetry, the three principal stresses in the barrel shell will be the circumferential or hoop stress, the longitudinal stress, and the radial stress. According to Ryder, (1982) and Nash, (1982), since the ratio of its thickness to internal diameter is less than about 1/20, it was assumed with reasonable accuracy that the hoop and the longitudinal stresses are constant over the thickness and that the radial stress is small and can be neglected, as shown in Figure 6.

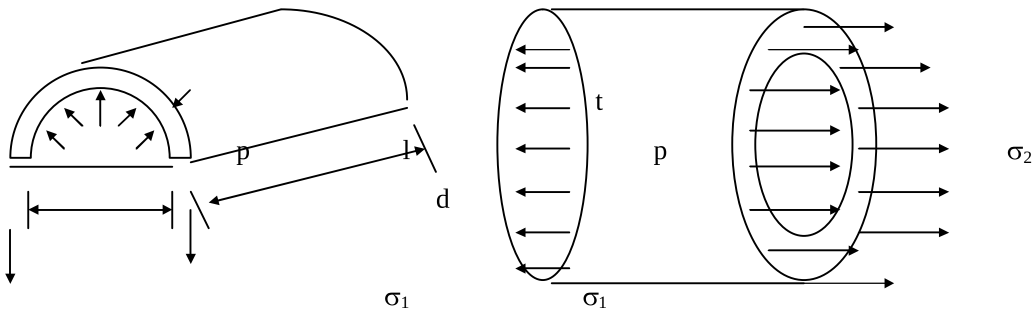

Figure 6. Hoop and longitudinal stresses in extrusion barrel

Hoop Stress, $\sigma_{1,} = pd/2t$ é é .(12)

=78.7 kN/m^2. This is safe, because it is less than the allowable shear stress.

Longitudinal Stress,σ_2 = pd/4t é é (13)

In the case of a long barrel, or barrels held by brackets, the longitudinal stress becomes too small and so is neglected (Nash, 1982).

2.5 Nose Cap

This is the component that houses the die (nose cone). It was designed to withstand the discharge pressure as well as help in pressure builds up. The bore at the center has a conical shape that tapers towards the discharge aperture as shown in Figure 5, this helps to raise the pressure and temperature of the extrudate, so enhancing discharge. High pressure will mean more force available to push out material, while higher temperature will melt the oil in the material and so helps flow. The Conical angle of the bore is 51.9^o

Actual volume of nose cap = 3.7x $10^{-4}m^3$
Weight of nose cap = volume of nose cap x density of mild steel = 2.89kg

2.6 Nose Cone

This is the last component through which the hot and plasticized extrudate is discharged. It plays similar role as the nose cap. Figure 7 is the sectional view of the cone.

Volume of cone = 8.4 x10^{-5} m3

Weight of cone = 6.58 x 10^{-1}kg (658g)

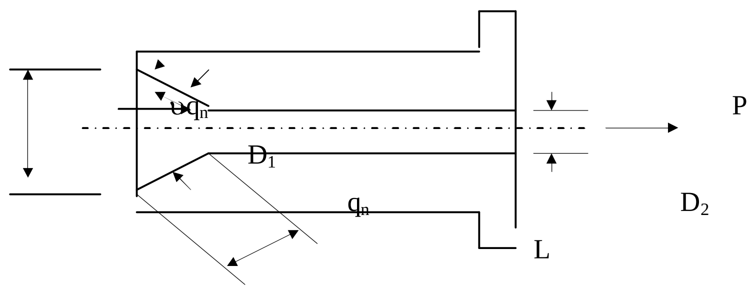

Fig 7 Sectional view of the Nose cone

The force, q_n, exerted by the nose cone on the extrudate is given by the following expression (Jensen, 1957):

$$P - [(A_1 - A_2)/ \sin\alpha](q_n \sin\alpha + \upsilon q_n \cos\alpha) = 0 \quad \ldots\ldots (14)$$

Where υ is the coefficient of friction between cone and material and is given as 0.21 as indicated by Rossen and Miller, (1973):

A = Area, P = Pressure and α = cone angle

$q_n = 3.4$ kN

The splitting force between the two halves of the cone, F, is given as Jansen, 1957,

F = P/A = 2.4 7kN

2.7 Extrusion Drive

The transmission belt transfers drive from the electric motor to the power transmission shaft with the aid of a pulley. The size and speed of the pulley including forces acting on the belt (Figure 8) needs to be carefully analyzed for proper belt and pulley selection.

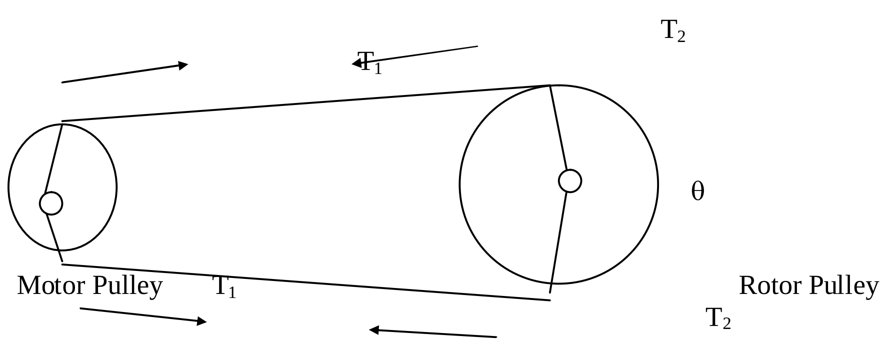

Figure 8 Belts – Pulley Arrangement

$$D_r = (N_m \times D_m) / N_r \quad \ldots\ldots (15)$$

Where N_m = Motor pulley speed (1250 rpm), D_m = Diameter of motor pulley; N_r = Rotor speed (450 rpm), D_r = Diameter of rotor pulley from equation 15, D_r = 280 mm

Minimum center distance C_m is given as (Nash, 1982; Jansen, 1957; Quayle, 1986):
$C_m = (D_m + D_r)/2 + D_m = 290mm$ é é (16)

Theoretical length of belt is given as:
$L = 2C + 1.57 (D_r + D_m)/2 + (D_r ï D_m)^2/4C$ é é . (17)
But $C < C_m$, (Shirgley, 1980)
With center distance C, as 350mm then L = 950.11 mm
Tension in the Belt:

Arc of contact made by the belt on the pulley θ_p, is given as;
$\cos \beta = (R_r ï R_m) / C$ é é (18)
Where β = wrap angle, R_r = radius of rotor pulley, R_m = radius of motor pulley.
So, $\beta = 14^o 9^1$ (0.25rad)

Angle of contact on belt on motor pulley (θ_m is given as);
$\theta_m = 180^o ï 2\beta$ é é (19)
$= 151.70^o$ (2.65rad)

Angle of contact on rotor pulley θr, is given as:
$\theta r = 180^o + 2\beta$ é é . (20)
$= 208.3^o$ (3.64rad)

Power transmitted is given as:
$P = (T_1 ï T_2) wr$ é é .. (21)

Where T_1 = tension at the tight side, N; T_2= tension at the slack side, N; W = angular velocity of motor pulley, 130.9 rad / sec; r = Radius of the motor pulley (50mm)

Evaluating the above yields the following;
T_1 = 8,202.28 N; T_2 = 3,280.91 N

Power transmission shaft:
Shaft design consists primarily of the determination of the correct diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading conditions and so the shaft was designed from the stand point of strength and rigidity.

This shaft is subjected to both bending and torsional loads. The bending load is due to weight and moments of the screw shaft.

Bending moment M, $= {}^{-1}/_2 W/ Lx^2 = -64.09Nm$ é é ..(22)

According to Harper (1980), Shirgley (1980), Hannah and Stephens (1980) the required diameter of shaft from the ASME load equation is given as:
$D^3 = 16PS_s [(K_bM_b)^2 + (K_tM_t)^2]^{1/2}$ é é (23)

Where S_s = Allowable combined shear stress for bending and torsion for steel shaft. K_b = Combined shock and fatigue applied to bending moment = 1.5 to 2.0 for minor shock; K_t = Combined shock and fatigue factor applied to torsional moment = 1.5 to 2.0 for minor shock; M_b = Bending moment; M_t = Torsional moment. Hence D = 3.6cm, approximately 4cm.

The calculated diameter, d = 4cm is less than the diameter originally selected (d = 5cm). So the selection is in order because strength criterion is satisfied.

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In order to design the required rigidity, it was necessary to determine the bending and torsional deflections (Harper, 1980). For steel shaft, the permissible deflection is 0.83mm/m. Also, the slope of a steel shaft subjected to cantilever force application must not exceed 1^o, Quayle (1986), and Shirgley (1980).

But deflection at the free end, δ, = $-1/24\ (W/EIL)(x^4 ï\ wL^3x + 3L^4)$ é é (24)

$= -0.15 mm/m$

Slope, θ, = $1/6\ WL^2/EI$ ..é ..(25)

$= 0.02^o$

This is less than the permissible deflection of 0.83mm/m and slope of 1^o. So bending deflection criterion is satisfied.

Torsional deflection is based on permissible angle of twist (3deg/m) as shown by Quayle (1986) and Shirgley (1980). But torsional deflection (δ) is calculated to be 0.24^o/m. This is less than the permissible value. Hence torsional deflection criterion is satisfied and so rigidity criterion is met.

2.8 The Bearing Unit

The machine has a pair of thrust bearings located at the inlet end of the screw. Static and dynamic conditions as well as the life requirements were considered in selecting these bearings. For combined radial and axial load,

$P_o = X_oF_r + Y_oF_a$ é é (26)

Where P_o = Equipment load, X_o = Radial load factor, F_r = Radial load on shaft Y_o = Axial load factor, F_a = Axial load
And P_o is found to be = 4,197.612N. Since the static load, C_o is unknown, employing trial and error, (Shirgley 1980), the right bearing was selected.
C = 6,070Ib (26,999.36N); C_o = 4,540Ib (20,193.92N)
Rating life, $R_r = (^C/_{Po})^3 \times 10^6$ revs $= 2.66 \times 10^8$ revs é é ..(27)

Designing for short operating conditions (i.e. 4,000hr); Desired life, $D_L = 1.44 \times 10^8$ revolutions. Since the desired life $D_L < R_L$, the selection is in order.

2.9 The Frame

The weight of the screw shaft (W_a), the barrel (W_b), the hopper (W_h), the bearing (W_{bb}), cone and cap (W_{cc}), and the material, soybean (W_{sb}) are considered in designing the frame.

A 50 (50mm angle iron having a Young Modulus of $20GN/M^2$ was chosen. The compressive stress on each frame is given as: $D_c = F/(4A)$ é é .(28)
Where, F = force (N); A = cross-sectional area of frame leg ($2.5 \times 10^{-3}m^2$); so, Dc = 39.94 kN/m^2.

The calculated compressive stress is much less than the Young Modulus ($20GN/m^2$) of the material. Therefore, there will be no bending of the frame. The standard minimum ratio of the frame lengths is given as $L_1/L_2 = 0.5$, (Shirgley 1980, Hannah and Stephens 1980). The designed ratio is 0.7, therefore the above condition is satisfied.

2.10 Performance Evaluation

The machine as shown if Figure 9, was designed to operate at a moisture range of 15.0% to 17.5%. Preliminary evaluation was performed to establish the best moisture level at which soybean seeds will be well extruded using the extrusion cooker. The machine extruded well at a moisture content of 16.55%, hence the actual evaluation was done at a moisture content of 16.55%.

The soybean was weighed and conditioned to the required moisture content. This was fed into the machine through the hopper. The temperature was recorded at 5minuites interval. This was used in establishing the temperature gradient at the four critical points on the machine; the feed section, compression section, metering section and the die.

Fig. 9. The extrusion cooker

3. PERFORMANCE EVALUATION RESULTS

The results of the performance evaluation are shown in Fig.10.

The low temperature gradient in the feed zone is due to the fact that no heat is generated in the zone. Since all it does is to convey the whole soybean. Hence little or no internal shear of the food takes place in the solids as contrasted to shear flow in the metering section.

The higher temperature gradient in the compression zone is attributed to the fact that the material is heated and worked into a continuous dough mass during passage through this section. Here the character of the feed material changes from granular or particulate state to amorphous or plasticized dough. This change in physical character is associated with a set of chemical reactions.

The relatively higher temperature gradient in the metering section is as a result of the shallow flights.

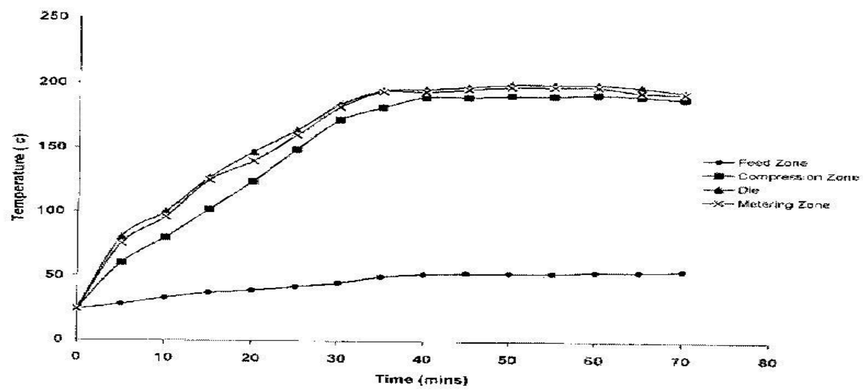

Fig. 10. Temperature gradient across the Barrels

The shallow flights increase the shear rate in the channel to the maximum level within the screw. The viscous dissipation of mechanical energy is typically large in the metering zone so that the temperature increases rapidly.

The relatively lower temperature gradient at the die is due to the rapid decrease in pressure and expansion with some loss in moisture due to flashing, and adiabatic cooling.

The full fat soy meal was subjected to chemical and microbiological analysis. The results shown in Table 1 satisfies the standard BIS specification (IS: 7837-1975) for full fat soy flour.

Table 1. Chemical and microbiological analysis of the full fat soy meal

CHEMICAL ANALYSIS		
% Moisture		6.16
% Crude Protein		38.87
% oil		19.01
Total Ash		5.94
PH (10 % Solution)		6.70
% Crude Fibre		5.54
Urease Activity	Nil	
% Carbohydrate	32.62	
MICROBIOLOGICAL ANALYSIS		
Total Plate Count		1×10^3
Yeast and Mould count	100	
Coliform count	Nil	

4. CONCLUSION

The development of any equipment requires a critical design of its components based on a operating loads and conditions. This will enable the designer eliminate or reduce to the barest minimal possible causes of failure of the components or equipment. With such detailed design, a reliable operating manual will be produced. The component design of a 187kg/hr soybean extrusion cooker was successfully carried out, and the machine fabricated. The low cost extrusion cooker for soybean processing was evaluated to establish its performance with respect to the temperature gradient across the barrels, the chemical and microbiological status of the full fat soybean flour. The result of the evaluation shows that the machine satisfies its objectives.

It is believed that with this equipment, processing of soybean into animal feed and possibly human food will be enhanced. This will encourage to a great extent, the development of livestock industry, which has great potentials in the country, and also the reduction in the level of calorie and protein deficiency in such meals.

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DESIGN, CONSTRUCTION AND PERFORMANCE EVALUATION OF A COWPEA THRESHER

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ABSTRACT

The effective threshing of cowpea with minimum grain loss, improved threshing capacity and efficiency was achieved with a dynamically stable thresher designed and fabricated with a power rating of 2.9kW, belt speed of 12m/s and cylinder speed of 5.03m/s. A horizontal centrifugal fan was used with straight blades. The spike tooth (rubber beaters) were arranged spirally to serve as conveyor. The machine has an efficiency rate of 96.58% and the threshing capacity of 27.58kg/hr for cowpea; at an average moisture content of 13.16 %. (dry basis), and concave-beater clearance of 9mm ± 0. 5. Separation losses were minimal.

KEYWORDS: Threshing capacity, cracking efficiency, separation losses, cowpea.

1. INTRODUCTION

Cowpea threshing (which involves the detachment of grain kernels from the panicles) is one of the most critical post-harvest operations. Grain losses are experienced during threshing. Cowpea is the most susceptible leguminous crop to impact of loading, due to the di-cot nature of its kernels and is most affected in threshing with iron beaters. Threshing of cowpea is achieved mechanically or traditionally (manually). Manual threshing is mostly applied using cocoa bags, or spreading large clean cloth or tarpaulin on the floor, laying a bundle of cowpea on the cloth and beating with heavy sticks and clubs. Alternatively, animals (horses and bullocks) are allowed to trample on them (Igbeka and Oluleye, 1986). Mechanical threshing of cowpea employs various thresher mechanisms such as spike tooth, rasp bar and angle-bar mechanisms. (Claude Culpin 1987).

The performance of a machine is determined (to some extent) by the properties of the crop it is designed to handle. (Ige, 1978). Most designs in existence use a cylindrical drum, while others use drums with square cross section. It was felt that the cylindrical shape gave a little or no fanning effect to the discharge of the particles, also there are lots of carry over effect in such designs and this may encourage damage and high power consumption. (Ige, 1978). Large, portable threshing machines are suitable mainly for contracting and for large commercial farms. It is uneconomical for a small farm to have such a machine unless it produces between 250-300 hectares each season. Built-in threshers were designed for small farms with power requirement of 4HP and 28HP for larger machines.

The axial flow threshing machine can effectively thresh rice, soybean, etc over a wide range of grain moisture levels with low grain losses. A 7-10hp air-cooled gasoline engine can power the thresher and power is transmitted through a series of v-belts to the major components. The institute of Agriculture Research and Training in 1985 developed a thresher that employed the combine actions of beating and rubbing. The use of star beater threshing drum has also been investigated (Bolufawi 1989).

It has been widely reported that cylinder speed and concave clearance are major factors that determine the efficiency of a thresher (Ahaneku et al 2001). For efficient threshing, there are peripheral speeds and concave clearances specified for different crops along with the types and details of thresher cylinder and operating moisture level (Joshi 1981) In order to improve on the performance of the existing cowpea threshers, this paper reports a machine that employed the actions of impact, rubbing and transport of the threshed product along the threshing drum with adjustable rubber beaters to vary the concave clearance.

2. MATERIALS AND METHODS

The materials of construction were locally sourced, the bearings (ball type) and the belts were the bought-out materials. The threshing chamber, fan assembly and hopper were made of mild steel (18 & 20 gauge). The frame structure and the engine seating were made of angle iron (11/2ò x11/2ò x1/4ò). The beaters were made of rubber. The Construction of the machine was carried out by marking out the plate and sizing using scriber and share cutters. Shafts were turned on lathe machine while seams and various components were welded with gauge 10 electrodes; Assembly of parts was done with fasteners (bolt & nuts).

Two varieties of cowpea were used for the tests at two different moisture contents of 13.16% and 15.43%. The varieties are Ife Brown and TVX3236 (with red eye). Two clearances were also used; 11mm and 9mm

2.1 Machine Description and Operation

The machine consists of the following units: the hopper, threshing chamber, threshing unit, the delivery chutes, the fan assembly and frame (Figure 1 and 2). In operation, the material is feed into the threshing chamber through the hopper made of mild steel metal plate (gauge 18). Threshing and pre clearing (Grain/chaff separation) takes place within the threshing chamber. The threshing unit consists of the threshing drum, 76.2mm in diameter, the rubber beaters arranged in spiral form around the drum (to form a screw conveyor for transport of chaff and stalk) the lower concave screen and the side plate covers. The drum is 380mm long and 80mm in diameter. The beaters were space 40mm apart.

The delivery chute houses the lower concave (screen) and also serves as support for the fan assembly. The fan is centrifugal type and has three straight blades, (279.4mmx80mm) arranged at 120^0 to each other around the shaft. Threshed grains falls through the screen, while the chaffs were conveyed to the axial end of the threshing chamber where they were thrown out from the chaff outlet. Clean grains were collected through the outlet while the lighter particles were blown off from the fan assembly. The entire threshing components were mounted on a frame network made of angle iron, 844.6mm long. The overall height of machine is 869.6mm and the base area (153.0 x 460.0) mm^2.

2.2 Design Considerations

The following factors were considered in the design of the machine;

a. Properties of the materials to be threshed dependent on type, variety, moisture content, addition of green matter etc.
b. Technical conditions dependent on drum selection, peripheral speed of drum, number of beaters etc.
c. Delivery of material to the drum dependent on feed rate, positioning, point of contact on delivery with the circumference etc.

The design focused on reduction of power consumption and peripheral speed for threshing. A rubber beater with spiral arrangement round the drum (serving as conveyor) was selected because of the low resistance of cowpea to loading impact. A beater ï screen clearance suitable for threshing was considered. The beaters have provision for adjustments to suit variety of crops (grain sizes) to be threshed. The clearance can be adjusted between 25.4mm and 9.5mm at a belt speed of 12m/s.

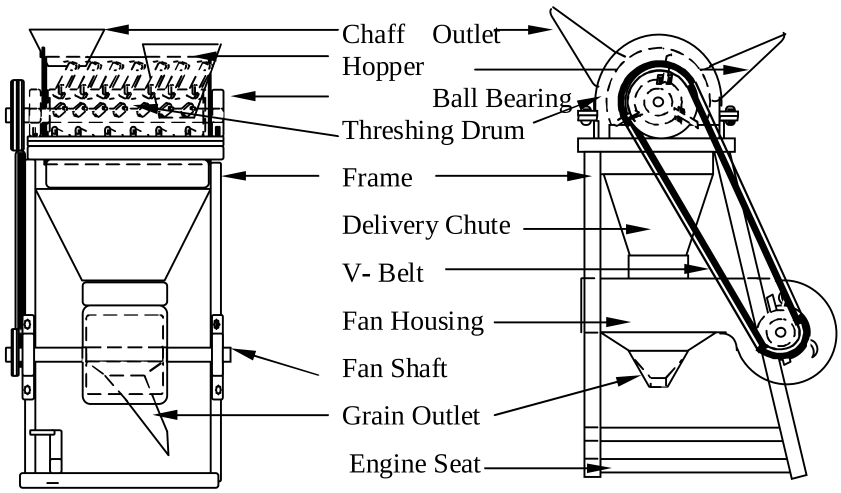

Fig. 1. Details of the thresher

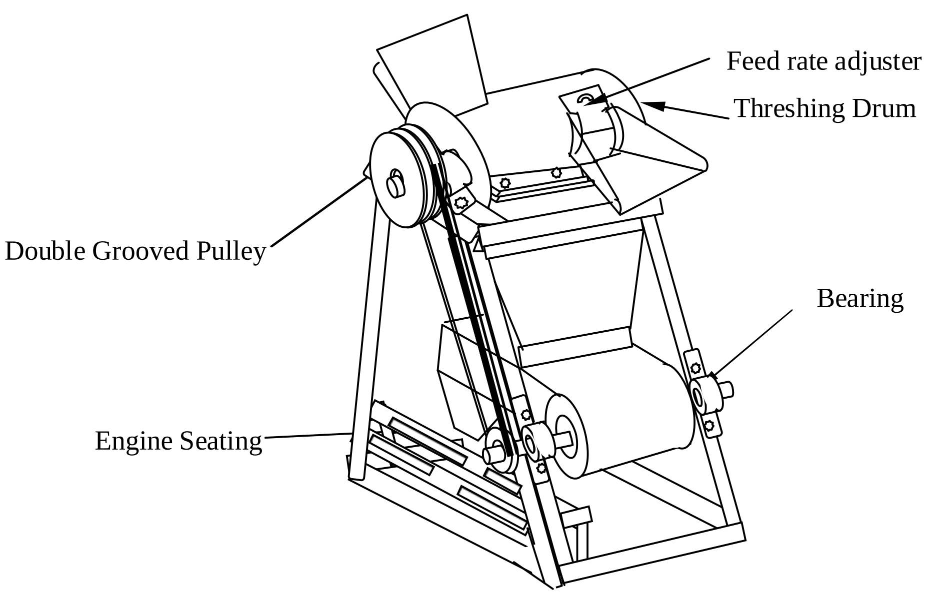

Figure 2: Isometric drawing of thresher

2.3 Design Calculations

Machine components were designed according to the procedures outlined in Design Data compiled by the Faculty of Mechanical Engineering P.S.G College of Technology, Coimbatore 641004 India (1982).

2.3.1 Power Requirement

Power required to drive the drum is given by:

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P_d =F.v = mv^3/R é é é é (1)

= 4.63 x 5.03^3 /0.04 = 2.9kw. (3.9HP)

Where P_d = power required to drive the threshing drum (kW), F = mv^2/R = centrifugal force, m = Total mass of the threshing drum = 4.6kg (45.38 N), R = Radius of shaft 0.04m. V= Velocity 5.03m/s g = acceleration due to gravity (9.81m/s.) From available motor standard sizes, a motor of 4.5HP was selected for the design.

2.3.2 Belt Design

Angle of contact for pulley belts is given by equation:

$\theta = \pi \pm 2 \sin^{-1}$ (D-d / 2c) radian é ... é é é é 2

d= pitch motor diameter (m) selected from table for design purpose, diameter D for larger pulley is calculated from D = 60v/ πn, n = Design rotational speed (rpm) for drum. c =centre distance between drum and motor shafts, π (rad) = 180^0 . The conventional negative and positive signs indicating the contacts in the smaller (motor pulley) and larger pulleys (threshing head pulley) respectively (Design Data, 1982)

Centre distance is given by

c = b+ã (b^2 ï 32(D- $d/16)^2$) é é é é é .. 3

Where

b = ¼ (4L^1 ï 6.28 (D+d)) é é é é é é 4

Where L^1 = pitch length of the belt selected from data table (Design Data, 1982),

Load Carrying capacity C is determined for both pulleys and the lowest value is taken to govern the design.

C = $e^{(\varepsilon\theta/\sin Ŭ2)}$ é é é é é .5

For V- belts where C = Load carrying capacity (3.47), ε = Coefficient of friction (friction factor, 0.4) for rubber belts. θ = Contact angle (2.8rad.) for smaller angle Ŭ= 40^0 (groove angle) degrees.
Power transmitted by belt is given by

P = (T_1 ï T_2) V é é é é é é é 6

Where V = belt speed (m/s)
T_1 = Tension in tight side (347.43N) and T_2 = Tension in slack side (98.68N) is calculated from:

$T_1 / T_2 = e^{\varepsilon\theta / (\sin Ŭ2)}$ é é é é é é . 7 (Design Data, 1982),

Belt pull factor for V-belt is between 0.7 and 0.9 (above that, the belt will be unstable and wears at a faster rate). The belt pull factor calculated is 0.73

Stress in belt: Various portions of the belt were subjected to tension or stresses such as tensile stresses due to initial tension, tangential and centrifugal forces, bending stresses etc.

2.3.3 Shaft Design

The shaft is subjected to two types of directional loading: vertical loading and horizontal loading.
Vertical loads resulted from: Loads due to weight of pulley acting downward, Torque or radial force, load due to weight of drum and reactions at the supports (bearings)
Horizontal loading resulted from; Load due to tangential force, reactions at the support due to the tangential force.
These forces were determined as:

Weight of pulley = 4.9N

Radial force M_t	=	23.28N
Weight of drum	=	17.26N
Tangential force	=	243.75N

These forces were resolved into resultant forces by the following equations. (Design Data, 1982),

$$R_A = \sqrt{(R^2_{av} + R^2_{ah})}, \quad R_B = \sqrt{(R^2_{bv} + R^2_{bh})}, \quad T_p = \sqrt{(T_t^2 + M_t^2)} \quad \ldots \ldots 8$$

Where:
R_{av} and R_{bv} are reactions at the bearings due to vertical loading
R_{ah} and R_{bh} are reactions at the bearings due to horizontal loading

Torsional moment in threshing shaft is given by: (Ademosun and Olukunle 2003)

$$T_m = P/2\pi n = [9550 \times P\ (Kw) / N\ (rpm)]\ (Nm) \quad \ldots 9$$
$$= 2.93 \times 9550/1200 = 23.28Nm$$

Maximum bending moment was obtained was obtained from a bending moment diagram for the loadings as $m_{B\ max}$ = 25.85Nm with a factor of safety K_t = 1.5. The shaft diameter that will withstand the loads is calculated from the maximum shear stress theory and the combine stress equation is used thus

$$\pi d^3_s /16 = \sqrt{[(T^2_{max} + M^2_{max})]} / S_{all} \quad \ldots 10$$

Where: T_{max} = maximum torsional (twisting) moment (23.28Nm), M_{Bmax} = Maximum bending moment (25.85Nm) S_{all} = Allowable design shear stress = (34.5mpa) 3.7921 x 10^7 N/m^2 , D_s = shaft diameter to be determined, the value of D_s calculated is 0.017 m. If a safety factor of 1.6 is assumed for the design the shaft diameter is 2.9cm. (Dobrovolski et al, 1974).

Assume that the torque is constant within limits of each shaft step the angle of twist as a result of torsion is given by the expression below.

$$\theta_t = \sum_{i=1}^{n} m_t\, x l_i / J_{ti} \times G \quad \ldots .11$$

Where:
M_t = critical torque applied to the shaft,
$J_{ti} = \pi d_s^4/32$ Polar moment of inertia of shaft (cm^4)
G= shear modulus or modulus of rigidity (8.1 x 10^{10} Nm^2 for steel), l_i = shaft length, θ_t = calculated is 2.33 x 10^{-3} rad (0.162^0). This is within the acceptable value range of 0^0 – 5^0.

2.4 Performance Evaluation

The machine was evaluated using the following indicators:

Threshing Efficiency was measured by the ratio of threshed grains to expected weight of clean grains in %
Threshing Capacity was based on the ability of the machine to remove good and matured grains from the pods, measured by weight per unit time.

Cracking efficiency was measured by the ratio of number of cracked grains to the total number of grains loaded.
Unthreshed losses are the good grains not detached from the pod after passing through the thresher
Loses are associated with the wastage recorded during threshing operation. *Grain loss/ unit time* is the weight of grains in the reject per unit time spent in threshing.

3. RESULTS AND DISCUSSION

The results of performance evaluation indicators are listed below the test results. Tables 1 and 2 indicate the performance of the machine under the moisture content and beater ï concave clearance variables. From the tables, the most suitable conditions for threshing cowpea are with 13.16% moisture content at 9mm beater ï concave clearance and 11mm beater ï concave clearance respectively. The threshing efficiency and threshing capacity reduced with increase in beater clearance and moisture content. The threshing efficiency and threshing capacity for Ife Brown reduces from 97.98% and 30.91kg/hr to 82.05% and 22.42kg/hr respectively at 13.16% and 15.43% moisture contents at 9mm and 11mm beater clearances.

Grain loss in separation increased with increase in clearance and moisture content while unthreshed losses reduced with decrease in clearance and moisture content. A high percentage of grain is lost as unthreshed losses through the chaff outlet. Threshing capacity as a function of time increases as a result of a reduction in clearance hence, the capacity of the machine is a function of time of threshing, moisture content and beater- concave clearance. At higher moisture content, 15.43% performance efficiency of the machine decreases (Table 2) with an increase in grain loses irrespective of concave clearance. The di-cot nature and susceptibility to loading impact of the crop will make cracking efficiency significant (30%) above 15.43% moisture content and cracking efficiency less significant (6%) at 13.16% dry basis.

Table 1. Performance evaluation of thresher at 9mm concave clearance with 13.16% and 15.43% moisture content dry basis

	13.16% M.C		15.43% M.C	
Performance indicators	IFE BROWN	TVX 3236	IFE BROWN	TVX 3236
Threshing efficiency %	97.78	95.38	78.89	81.06
Threshing Capacity (kg/hr).	30.91	24.25	19.26	20.00
Grain Loss/Unit time (kg/hr)	0.22	0.52	21.19	0.33
Cracking Efficiency %	6.00	3.33	30.00	18.33
Unthreshed losses (kg/hr)	3.21	5.15	0.93	0.97
Grain loss %	2.30	2.85	11.12	14.51

Table 2. Performance evaluation of thresher at 11mm concave clearance with 13.16% and 15.43% moisture content dry basis

	13.16% M.C		15.43% M.C	
Performance indicators	IFE BROWN	TVX 3236	IFE BROWN	TVX 3236
Threshing efficiency %	82.05	87.35	81.30	79.23
Threshing Capacity kg/hr.	22.42	0.36	18.75	17.89
Grain Loss/Unit time kg/hr	0.19	0.10	0.85	1.49
Cracking Efficiency %	1.67	1.67	14.00	6.67
Unthreshed losses kg/hr	48.52	16.55	1.60	8.16
Grain loss %	12.00	17.01	11.68	24.21

4. CONCLUSION

The machine is dynamically stable and able to withstand vibration. The materials under test behaved in the same way under test conditions (parameters) but with slight variations due to size and some other parameters that often affect its mechanical properties such as resistance to impact loading etc. The Ife brown with bigger sizes suffered a higher percentage of grain loss in terms of cracking the grains. At relatively high moisture content above 15% more grains were lost, at lower clearance, threshing efficiency increased but threshing capacity was reduced for TVX3236 due to the small nature of the grains hence increase in time of operation.

Threshing capacity, cracking efficiency and unthreshed losses are directly affected by the concave ï beater clearance and the grain moisture content. The best operating conditions for the thresher was when the clearance was set at between 0.0085m and 0.009m, and moisture content maintained at 13.16%.The machine capacity was dependent on the threshing efficiency and time of operation, thus the machine is rated at 27.58kg/hr. This is an improvement over existing cowpea thresher with rasp bar beater rated at 17 kg/hr capacity.

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ON THE VALUE OF SHORT-TERM WEATHER FORECASTS IN SCHEDULING SUPPLEMENTAL IRRIGATION OF POTATOES IN THE UNITED KINGDOM

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ABSTRACT

Many agricultural activities can be adversely affected by the weather. Effects may impact on farm-level production economics and/or the wider environment. Developments in weather forecasting have led to improvements in spatial and temporal resolution, but does the information meet the needs of UK farmers in making day-to-day operational decisions? In this study, a farm-level production economics model was used to investigate the value of short term weather forecasts in scheduling irrigation of potato crop. Irrigation scheduling was simulated over a range of irrigation costs for potatoes under conditions representing wet, dry and average years. The results demonstrate that the use of weather forecast in scheduling can deliver increased economic benefit when compared to the strategies not using weather forecasts. In the wet year the value of weather forecast (i.e. the increased economic benefit compared to alternative strategies) derives from increased profits due to cost savings from unnecessary irrigation. In the average and dry years the value of weather forecast would be in terms of greater profit per unit of water application and would derive from improved efficiency in the use of limited or expensive water supply.

Results from simulations using perfect knowledge of short-term weather to estimate the value of potential improvement in weather forecast suggest that profit would increase with improvements in quality and lead-time of weather forecast. The increase in profit would be due to further cost savings from unnecessary irrigations in the wet year and more rational use of limited or expensive water supply in the dry and average years.

KEYWORDS: Weather forecast; irrigation scheduling; value of information; potato.

1. INTRODUCTION

In the UK supplementary irrigation has become an essential component of a competitive crop production system on approximately 175,000 ha of arable land. Potatoes (*Solanum Tubersorum*) constitute the main irrigated crop and account on the average, for over 40% of the irrigated area and over 50% of water application. Irrigation water constitutes only about 1% of all surface and ground water abstraction nationally but because of its temporal and spatial distribution, the impact on water resources is far greater than the low percentage would indicate. Demand is concentrated in the Southeast region of England and over a relatively short period of 8 to 12 weeks in the summer months when water resources are scarcest. Irrigation is therefore a very significant consumptive user of water in the catchments at this period (Evans, 1994; University of Newcastle Upon Tyne, 1990). Widespread public concern about water scarcity and its environmental effects (e.g. Royal Society for the Protection of Birds, 1995) and the growing pressure from other competing users are limiting irrigatorsô access to water. A real need therefore exists for irrigators to use water more wisely through proper irrigation scheduling.

A major difficulty in scheduling supplemental irrigation is rainfall uncertainty. The irrigator faces the risk of excessive water stress that could potentially reduce crop yield and quality when irrigation is delayed and the anticipated rainfall fails to occur. On the other hand, if there is an unexpected rainfall soon after irrigation, over watering may occur with a possibility of crop damage, trafficabilty problems, erosion, excessive leaching of pesticides and fertilisers, and the wastage of water, labour and energy.

Water application in the UK is mainly by the overhead (spray) method using mobile hosereel-raingun irrigators. An irrigation cycle typically lasts for several days due to constraint of equipment capacity and/or of labour. This results in a queue of fields or farm sections waiting for irrigation. In the special circumstance of supplemental irrigation an important aspect of improved irrigation scheduling would be the use of weather forecast information, if available, for daily decisions on whether or not to initiate irrigation.

Existing irrigation scheduling models used by several commercial scheduling services in the UK are based on the ñtrigger levelò concept of available water. They do not consider weather forecasts or attempt to optimise returns from timing of irrigations. Gowing and Ejieji (2000) reviewed the models and others developed elsewhere based on dynamic programming frame work (e.g. Wilks and Wolf, 1998). They noted the constraints to the practical application of the dynamic programming approach, in particular the large size of the state space which occurs i.e. the so called curse of dimensionality and the associated computational cost. Adapting the free-form simulation approach of Swaney et al. (1983) therefore, Gowing and Ejieji (2000) proposed a model which utilises short-term weather forecasts and a decision model in scheduling supplemental irrigation. They applied the method to potato irrigation. The short-term weather forecasts utilised in the model are produced by the UK Meteorological Office for farmers through a commercial service. The weather forecast is provided for several regions of the country on daily basis for the current day and five days ahead. Information is provided on several weather variables including rainfall amount, rainfall probability, sunshine hours and, maximum and minimum temperatures.

The decision on whether or not to utilise the weather forecast information is however ultimately at the discretion of individual irrigators who may have their own perceptions regarding the value of this information.

The objective of this paper therefore is to present an objective assessment of the value of weather forecasts in relation to two irrigation scheduling strategies which may reasonably be considered as reflecting sensible alternatives to dealing with the scheduling problem without the use of weather forecasts. The first bases irrigation decision only on projected soil water deficit while the second combines the projection of soil water deficit with a simple assumption of short term weather persistence. The value of potential improvement in short-term weather forecasts was also evaluated using simulations assuming perfect knowledge of short-term weather.

2. MATERIALS AND METHODS

2.1 The Simulation Procedure

The simulation model of Gowing and Ejieji (2000) divides the entire season into three time segments all of which are taken into consideration in each irrigation decision. The time segments are termed the past, forecast and future periods. The past period represents the time from the beginning of the season to the day on which irrigation decision occurs and actual weather data are utilised in this period to describe the state of the crop-soil system. The forecast period represents the short-term horizon for the irrigation decision and is set at seven days in the model. The weather data input in this period is mainly from the available weather forecast. The future period refers to the time from the end of the forecast period to the end of the season. Since in a real time situation, information on weather conditions is unavailable beyond the forecast period, a projection of the state of the system to the end of the season is achieved in the forecast period by averaging the results from repeated simulations using several years of historical weather data.

The simulation model considers a farm made up of several fields which are defined as a zone of the farm that can be irrigated in one day. The irrigatorôs equipment and labour constraints are accounted for by specifying a minimum irrigation interval for the fields. For the description of the crop-soil interaction the model incorporates a dynamic physiologically-based crop model capable of simulating the growth of potato from planting to harvest (Ejieji and Gowing, 2000).

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The simulation model updates the state of the crop-soil system on each day of the season by simulating for the current day the soil water balance and the crop growth and development. In order to reduce the possibility of excessive water stress in the last field to receive water in an irrigation cycle, a projection of the soil water deficit in all the fields is made to the end of the forecast period. If the largest or smallest projected soil water deficit is in excess of the allowable value and the minimum irrigation interval of the driest field would be equalled or exceeded by the next day, the irrigation criterion is applied. If irrigation is favoured, the driest field is flagged as the first to receive water in an irrigation cycle which would be initiated the next day. Irrigation decision is therefore essentially taken a day in advance. The decision is revised on daily basis taking advantage of the most recent weather forecast information. An irrigation cycle once started can therefore be terminated on any day without necessarily completing it.

The irrigation decision criterion is stated as follows

$$I_{t+i} = d, \quad i = 1, 2, ., ., ., \eta \qquad (1. a)$$

$$\text{if} \quad E[\varphi(I,t)] > E[\varphi(I^0,t)] \qquad (1. b)$$

else

$$I_{t+i} = 0, \quad i = 1, 2, ., ., ., \eta \qquad (1. c)$$

Equation 1.a represents the decision on day ñtò, to apply an irrigation depth d on each of the next η days i.e. from day t+1 to t+η where η is the number of days required to complete an irrigation cycle. In Equation 1.b $E[\varphi(I,t)]$ represents the expected profit estimated on day t for initiating irrigation on day t+1 , $E[\varphi(I^0,t)]$ is the expected profit of not initiating irrigation, φ denotes profit while I and I^0 denote the cases of irrigation and no irrigation respectively. Profit is computed as return from crop sale less variable cost of irrigation. Fixed cost is not considered because it is assumed that the irrigation system is already in place and the irrigatorôs concern is how best to use it.

The expected profit of irrigation is computed from the expression

$$E[\varphi(I,t)] = \sum_{k=1}^{m} P_{Tk} \frac{1}{n} \sum_{j=1}^{n} \varphi(R_{Tk}, W_j, I) \qquad (2)$$

where P_{Tk} is the probability of experiencing a rainfall event sequence in the first three days of the forecast period, R_{Tk} is the resulting sequence of rainfall amounts and m is the number of all possible rainfall event sequences. W_j is one of the years the historical daily data of which is used to describe the future period and n is the number of the years. Similarly, $E[\varphi(I^0,t)]$ is computed by substituting I with I^0 in Equation (2) i.e. making I_{t+i} , (i = 1, 2, ., ., ., η) identically zero (Equation 1.c). In the first three days of the forecast period there are eight possible rainfall event sequences ranging from three consecutive wet days to three consecutive dry days. There are 8n number of simulations required, from the first day after the forecast period to the end of the season, for the computation of an expected profit. For each of the simulations, a different weather year W_j is utilised.

It should be noted that the estimated event probabilities of the rainfall event sequences are based on the assumption of independence and ignores persistence. The deficiency of the assumption has been considered by Ejieji(1998) who suggested that the deficiency would be less severe in the summer months when the convective storms are common than in the winter period when precipitation in the UK is predominantly due to frontal systems.

Although the short term horizon (i.e. the forecast period) for irrigation decision is 7 days ahead, the model assumes that weather forecast is available for a lead-time of five days. Only the first three days however explicitly influences irrigation decision because the forecast quality deteriorates with lead-time (Golding, 2000). The fourth and fifth days are categorically assumed to be dry or wet depending on whether the forecast rainfall probability for the days were high (≥ 0.8) or low (< 0.80). If the day (i.e. fourth or fifth day) was considered to be wet, the rainfall amount for the day was taken to be the expected rainfall given by the product of the forecast amount and the forecast probability. In order to simulate as closely as practicable a realistic rainfall situation for the last two days (i.e. sixth and seventh days) of the forecast period, the rainfall occurrence is generated stochastically as first order Markov process while the rainfall amount for a wet day is obtained by sampling from a gamma distribution the parameters of which are derived from historical daily rainfall data. The algorithm of the process is similar to that of Geng et al. (1986).

The main features of the simulated strategies are described in Table 1. Irrigation assuming perfect knowledge of short-term weather is not a practical proposition but only indicates the upper

Table 1. Main features of the irrigation strategies simulated

*Strategy	Use of the irrigation decision criterion	Description of the time segments		
		Past Period	Forecast period	Future period
With forecast (IRWF)	Yes	Uses actual weather data	Uses forecast weather data	Average of repeated simulations using several years of historical weather data
Perfect knowledge (IRPK)	Yes	Uses actual weather data	Uses actual weather data	Same as in IRWF
No Forecast (IRNF)	No	Uses actual weather data	Weather forecast not considered. 10-day moving averages of weather variables of the past period used to project soil water deficit	Not considered
Persistence (IRPSS)	No		Same as in IRNF but persistence of short-term weather is also assumed	Not considered

*IRWF = irrigation with weather forecast (The model of Gowing and Ejieji, 2000)

IRPK = irrigation with perfect knowledge of short-term weather

IRNF = irrigation without weather forecast

IRPSS = irrigation assuming persistence of short-term weather

bound of performance that could be expected with improvement in the quality and lead-time of the forecasts. In the case of irrigation without weather forecast, only water deficit projection to the end of the forecast period influenced irrigation decision. It did not consider any possible rainfall during the forecast period. Irrigation assuming short-term weather persistence followed the same procedure as irrigation without weather forecast except that short-term persistence of wet weather was assumed. Thus irrigation decision on a wet day was deferred to the next day. Since irrigation is taken a day in advance, the incidence of rainfall effectively postponed irrigation by at least two days. Irrigation without forecast was considered to represent the extreme attitude of an irrigator who gives no consideration for the uncertain rainfall within the short-term horizon while irrigation assuming persistence of short-term weather was assumed to approximate to a moderate attitude of an irrigator with local knowledge of the long-term average rainfall characteristics.

2.2 Estimation of the Value of Weather Forecast and the Value of Potential Improvement in Weather Forecast

The estimation of the value of short-term followed a conventional approach in which the estimate of the value of information is taken in a relative sense as the difference in benefit obtained when using information optimally and that achieved under some basic information level. The value of short-term weather forecast in relation to each of the evaluated strategy (i.e. with the strategy as the basic level) was therefore estimated as the amount by which the profit was less than that of irrigation with weather forecast. In the case of value of potential improvement in short-term weather forecast however, it was evaluated as the amount by which the profit of irrigation assuming perfect knowledge of short-term weather exceeded that of irrigation using the available short-term weather forecast.

2.3 Input Data and Simulated Conditions

The data used in this study and the simulated conditions are representative of Cambridgeshire, East Anglia. They are the same as those that have been described elsewhere by Gowing and Ejieji (2000). The meteorological data comprised the relevant weather forecast data obtained from the Norwich Weather Centre of the Meteorological Office and twenty-six years (1970 ï 1995) of historical weather data from the UK Meteorological Office. The historical data include rainfall, potential evapotranspiration of grass, soil temperature, maximum and minimum air temperatures and sunshine hours; they are required to drive the crop growth simulation model.

The growing seasons considered were 1992, 1994 and 1995 representing a wet year, an average year and a dry year respectively. The average year has about average seasonal rainfall while the wet and dry years respectively have seasonal rainfalls above and below the average. Twenty years (1970 - 1989) of historical weather data were used to describe the future period. The soil water deficit situations triggering irrigation in the field were as discussed by Gowing and Ejieji(2000). All simulations were carried out at seven values of variable cost of irrigation of £1.7/ha mm, £3.4/ha mm, £5.8/ha mm, £9.2/ha mm, £12.6/ha mm, £14.3/ha mm and £17/ha mm. The crop prices of £58.4/t, £186/t and £150/t representing the actual crop prices for the 1992, 1994 and 1995 seasons respectively were used in the computation of profits for the respective seasons.

3. RESULTS AND DISCUSSION

3.1 Rainfall Characteristics

The growing season (April to September) rainfall showed marked variability in the seasonal amounts. The seasonal rainfall for a 26-year (1970-1995) period averaged 285.4mm with a coefficient of variability of 21.3%. The long-term daily rainfall pattern indicate that for each month of the growing season, the chance of a wet day being followed by another wet day is on the average more than 50% and is more than double that of a wet day following a dry day (Table 2.). The high variability of actual daily values of the transitional probabilities as reflected in the range of daily values (also shown in Table 2) however limit the validity of any generalised assumption regarding wet weather persistence especially for the short-term.

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Table 2. Monthly averages of the daily transitional probabilities of wet day to wet day (P(W/W)) and dry day to wet day (P(W/D)), and monthly proportion of wet days during the growing season.

Month	*Average of daily transitional probabilities		Proportion of rainy days
	P(W/W)	P(W/D)	
April	0.69 (0.88 - 0.36)	0.28 (0.56 ï 0.08)	0.48
May	0.71 (1.00 ï 0.40)	0.25 (0.45 ï 0.00)	0.46
June	0.70 (1.00 ï 0.38)	0.23 (0.38 ï 0.00)	0.43
July	0.59 (1.00 ï 0.25)	0.26 (0.54 ï 0.00)	0.38
August	0.59 (1.00 ï 0.11)	0.28 (0.55 ï 0.07)	0.39
September	0.63 (1.00 ï 0.25)	0.25 (0.50 ï 0.00)	0.39

* The ranges of actual daily values in the respective months are shown in brackets

The amount of rainfall for the 1992, 1994 and 1995 seasons were 377.1mm, 276.9mm and 187.1mm respectively. The intra-seasonal rainfall distribution for the three seasons was characterised by dry spells with varied frequency of occurrence among the seasons. There were 87, 59 and 43 wet days in the 1992, 1994 and 1995 seasons respectively. Dry spells of two days duration or more occurred 16, 19 and 18 times in the 1992, 1994 and 1995 seasons respectively. The average duration of the dry spell was 4 days in 1992, 5 days in 1994 and 9 days in 1995 season while the maximum duration in the respective seasons was 8, 11, and 36 days. The longest duration spells tended to occur between June and mid August (i.e. between Julian days 151 - 230). The daily rainfall amounts ranged from 0.1mm to 26mm with rainfall amounts in the range of 0.1mm - 5mm predominating. The average rainfall depth per precipitation event was 3.4mm, 3.0mm and 2.2mm for the 1992, 1994 and 1995 seasons respectively.

3.2 Reliability of Rainfall Forecasts

The comparison between the forecast rainfall probabilities and the actual relative frequency of rainfall occurrence are shown in Fig.1 representing the pooled result from 1992 to 1995 seasons. To obtain the relative rainfall frequencies, the daily forecast rainfall probabilities for each of the 5 days ahead were assigned to given classes of equal rainfall probabilities. To calculate the relative frequency for each class, the number of rainfall occurrences was divided by the number of days having forecast rainfall probabilities in that class.

Within the range of probability values of 0.4 - 0.70 for the first day ahead and 0.5 - 0.70 for the second day ahead, the forecast probabilities agreed closely with the relative rainfall frequencies with an average absolute deviation of less than 0.03. A similar situation obtained only at the probability value of 0.70 from the third to the fifth days ahead. Forecast rainfall probabilities below the stated value or range of values for the respective days ahead were less than the relative rainfall frequencies implying that rain occurred more frequently than forecast. The reverse was generally the case for forecast rainfall probabilities above the stated value or range of values. The average absolute deviation of the forecast rainfall probabilities from the relative rainfall frequencies were 0.096, 0.126, 0.131, 0.150, and 0.136 for the first, second, third, fourth and fifth days ahead respectively confirming that the quality of forecast deteriorated with lead-time.

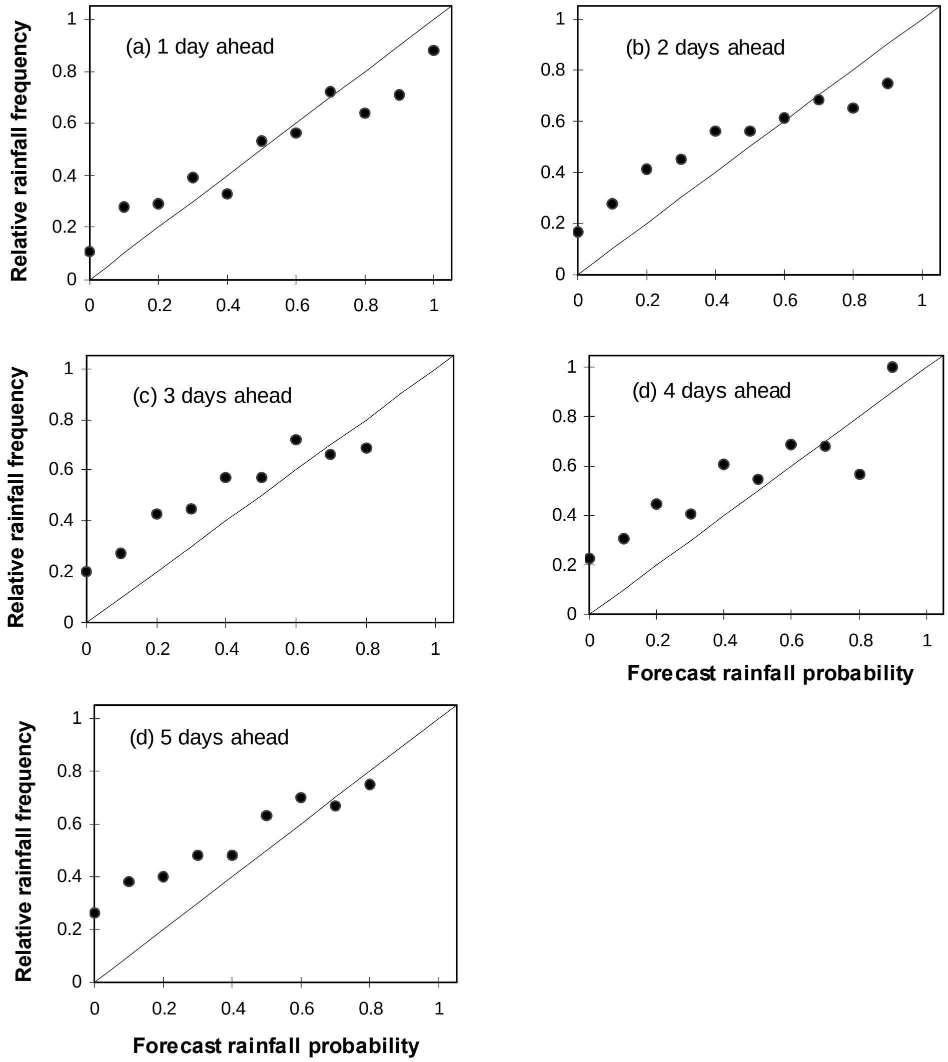

Fig.1. Comparison between forecast rainfall probabilities and relative rainfall frequencies for 1992 to 1995 seasons.

A comparison of the daily forecast and actual rainfall amounts during the growing seasons showed that rainfall amounts of less than 5mm were more frequently forecast and generally were, on the average, more closely predicted (Fig. 2). Actual rainfall amounts higher than this value were generally under predicted. The closest predictions were for rainfall amounts of 2.5mm or less. Golding(2000) has also found that larger events (10mm threshold) are less well predicted than small events(1mm threshold). It should be noted that except in the case of the forecasts of zero amount, actual forecast rainfall amounts are given as a range of values with the lower and upper values of the range provided. For this study the average of the upper and lower values was taken as the forecast amount.

During periods of extended dry spells, there was a tendency to erroneously predict the occurrence of rainfall over a greater part of a dry spell duration thus forecasting non-zero rainfall amounts for the dry days.

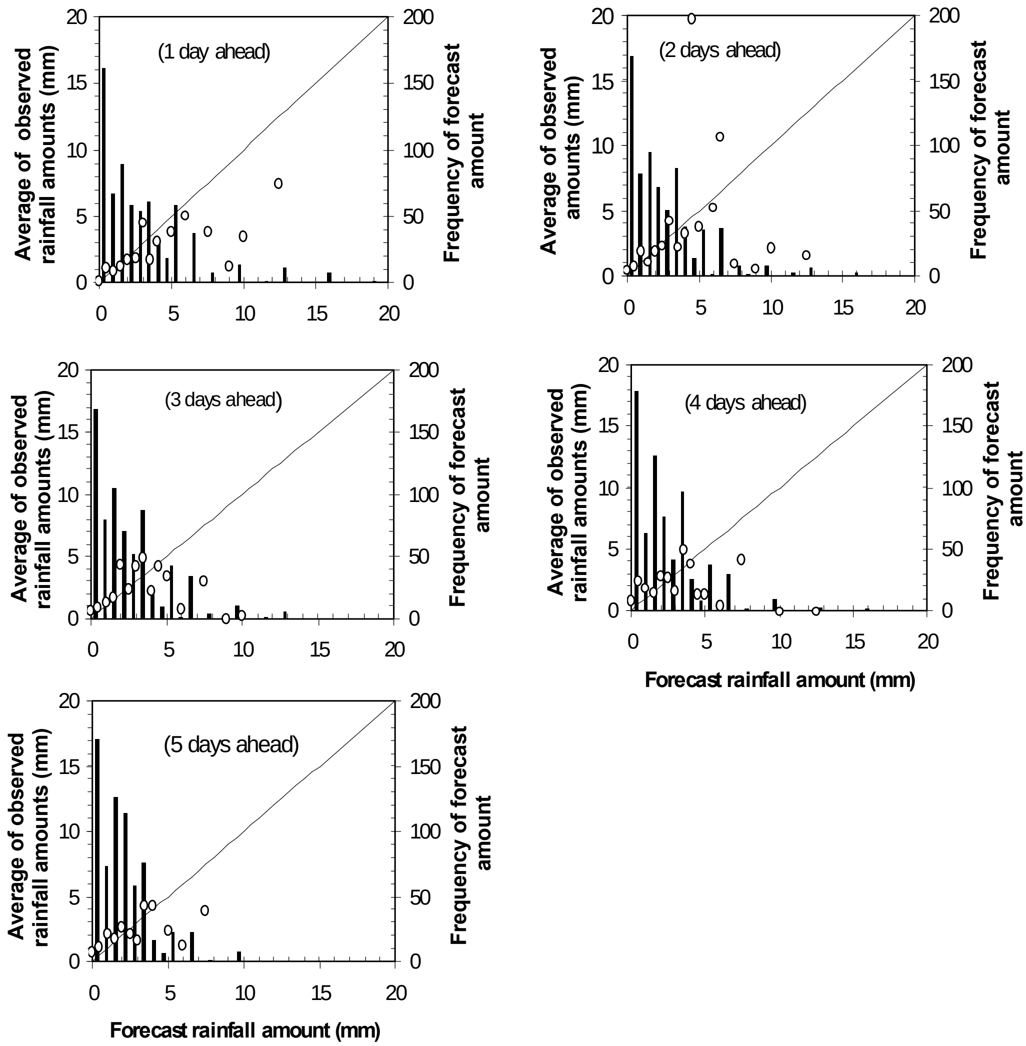

Fig 2. A comparison between forecast rainfall amounts and the average amounts observed (circles) for the respective forecast amounts in the 1992 to 1995 seasons, at March, Cambridgeshire,. Frequencies of the forecast amounts (columns) are also shown. (— = 1:1 line between the forecast and average observed amounts)

3.3 Water Application in Relation to Irrigation Cost and Yield

The average irrigation amounts and the average yields obtained for the simulated strategies are presented in Table 3. Actual number of irrigation events per field varied among the fields at each irrigation cost. The average irrigation amounts and average yield represent respectively the mean seasonal water application and mean crop yield for the seven fields.

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Table 3. Average irrigation amounts and average crop yield obtained for the wet, average and dry years

	*IRNF	*IRPSS	IRPK and IRWF at various irrigation costs (£/ha mm)													
			1.7		3.4		5.8		9.2		12.6		14.3		17	
			IRPK	IRWF	IRPK	IRWF	IRPK	IRWF	IRPK	IRWF	IRPK	IRWF	IRPK	IRWF	IRPK	IRWF
(a) Wet year																
Average irrigation amount (mm/ ha)	207	143	96	114	96	111	75	104	0	86	0	61	0	57	0	57
Average yield (t/ha)	51.4	50.9	50.7	50.7	50.7	50.7	49.7	50.1	38.2	49.6	38.2	47.7	38.3	47.1	38.2	47.1
(b) Average year																
Average irrigation amount (mm/ ha)	296	218	282	214	282	214	282	193	282	182	43	177	29	164	21	161
Average yield(t/ha)	52.1	34.5	51.3	34.7	51.3	34.7	51.3	33.0	51.3	31.9	28.8	31.0	28.3	30.0	27.8	28.9
(c) Dry year																
Average irrigation amount (mm/ ha)	321	275	318	279	318	239	318	207	214	182	50	104	43	96	32	86
Average yield(t/ha)	40.6	36.7	41.1	37.9	41.1	36.1	41.1	35.8	38.0	34.6	31.1	31.2	30.5	31.1	30.0	30.9

* IRNF and IRPSS correspond to the cases of irrigation scheduling without weather forecast and with assumption of short-term weather persistence respectively and are therefore independent of irrigation cost.

For irrigation with weather forecast (IRWF) and irrigation assuming perfect knowledge of short-term weather (IRPK), the total seasonal irrigation application was generally sensitive to irrigation cost decreasing as irrigation cost increased. Stepped sensitivity to irrigation cost was however exhibited by IRWF over some ranges of irrigation cost and has been discussed elsewhere (Gowing and Ejieji, 2000). IRPK equally showed stepped sensitivity over the two lowest irrigation costs of £1.7/ ha mm and £3.4/ ha mm in 1992 (i.e. wet year) and 1995 (i.e. dry year), and over the irrigation cost range of £1.7/ ha mm to £9.2/ ha mm in 1994 (i.e. the average year). The water application was constant over the respective cost ranges. The fixed application depth per irrigation not adjusted to actual field soil water deficits is a contributory factor to the stepped sensitivity. The fixed application adopted in the simulation however reflects the normal UK practice, as it is not practical to adjust application to suit actual deficits.

In the cases of irrigation assuming short-term weather persistence (IRPSS) and irrigation without weather forecast (IRNF) where irrigation was not influenced by economic considerations, the corresponding results (also presented in Table 3) are independent of irrigation cost.

Crop yield generally increased with water application. IRNF applied the highest amount of water in all the years and except in the dry year had the highest yield.

3.4 Value of Weather Forecast and of Potential Improvement in Weather Forecast in terms of End of Season Profit

The values of weather forecast and of potential improvement in weather forecast in terms of end of season profit are presented in Fig. 3 for the three seasons at the various irrigation cost. The value of weather forecast for each strategy at a given irrigation cost is numerically equal to the profit for IRWF minus the profit for that strategy. The value of potential improvement in weather on the other hand is numerically equal to the profit for IRPK minus the profit for IRWF. In the wet year i.e. 1992 season, the value of weather forecast was positive for both IRNF and IRPSS and increased with irrigation cost (Fig. 3). The value of potential improvement in weather forecast also mostly increased with irrigation cost. The value ranged from £22 /ha at the lowest irrigation cost of £1.7/ ha mm to £445 /ha at the highest irrigation cost of £17/ ha mm or 8.0% to 25.6% of the profit of IRWF at respective irrigation costs. IRPSS saved some water from unnecessary irrigations relative to IRNF. The value of weather forecast was therefore higher for IRNF than IRPSS. Their difference in values of weather forecast ranged from £118 /ha at the lowest irrigation cost to £1065/ ha at the highest irrigation cost.

The water application by IRPSS was lower than that of IRNF by 30.9%. The largest water application by IRWF, which was at the lowest irrigation cost, was less by 20.0% and 44.8% than the seasons application by IRPSS and IRNF respectively but was 18.6% more than highest application by IRPK.

The value of weather forecast in the 1994 season i.e. average year was negative at all the irrigation costs for IRNF and at all but the two lowest irrigation costs for IRPSS. The negative values when expressed as absolute values (i.e. without the negative signs) represent the amounts by which the profits for the strategies exceeded those for IRWF. In the case of IRPSS the negative values averaged - £120/ ha or ï 3.1% of the profit for IRWF in the irrigation cost range of £5.8/ ha mm to £17/ ha mm. In that range of irrigation cost, the water application by IRPSS was 13.0% to 35.6% more than the applications by IRWF. For IRNF the negative values increased from - £3111/ ha at the lowest irrigation cost to - £2008 at the highest irrigation cost or ï 51.5% to ï 75.8% of the profit for IRWF at the respective irrigation costs. The water application by IRNF was however more than the applications by IRWF by 38.3% and 84.4% at the lowest and highest irrigation costs respectively.

It should be noted that the mainly negative values of weather forecast for IRWF and IRPSS in the 1994 season is attributable to the exceptional crop price in that year which was the highest crop price in 20 years (1975 ï 1994). Returns from higher yields from more generous applications of water by the strategies were therefore generally more than those achieved through more conservative

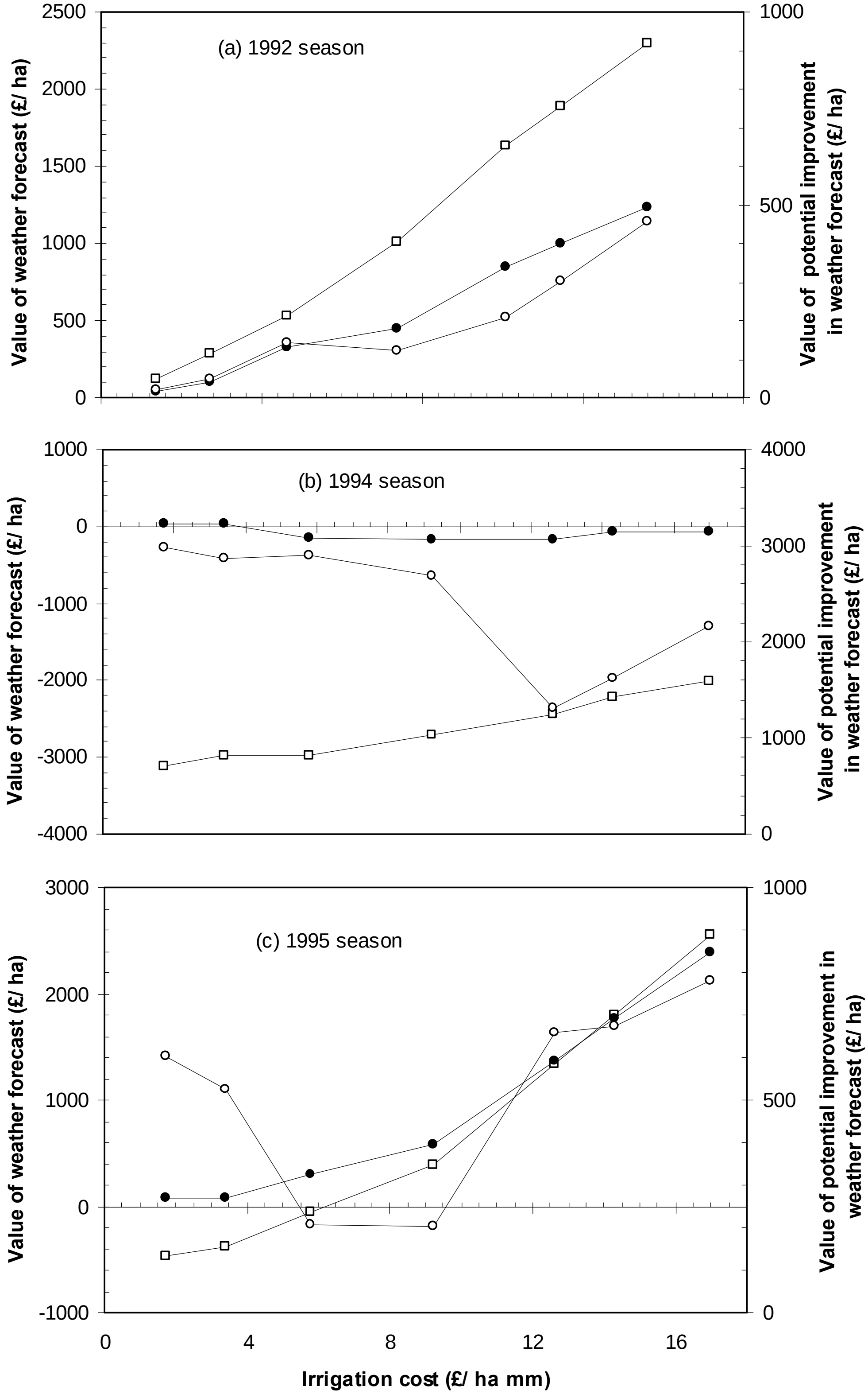

Figure 3. Value of weather forecast in relation to irrigation cost for the (a) wet season, (b) average season and (c) dry season with respect to the strategies of irrigation without weather forecast (□) and irrigation assuming short weather persistence (•). Value of potential improvement in weather forecast (o) in relation to irrigation cost is also shown.

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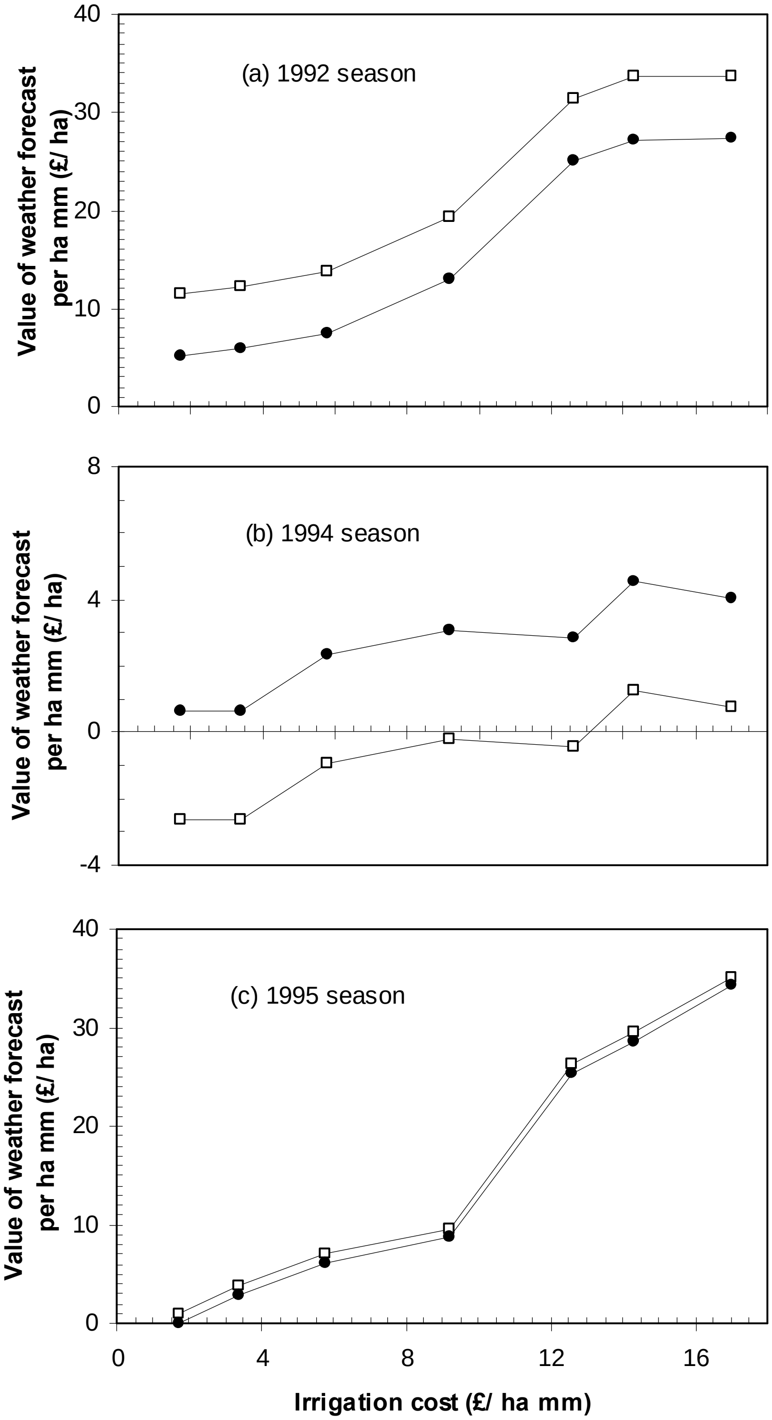

Figure 4. Value of weather forecast per ha mm of irrigation in relation to irrigation cost for the (a) wet season, (b) average season and (c) dry season with respect to the strategies of irrigation without weather forecast (□) and irrigation assuming short weather persistence (●).

irrigation by IRWF even at high irrigation costs. An analysis (not reported) of the simulation result obtained for the year by Ejieji (1998) with a lower projected crop price of £82/t showed that IRWF applied less water at corresponding irrigation costs and that the value of weather forecast was positive and increased with irrigation cost at the lower crop price.

The value of potential improvement in weather forecast was positive at all the irrigation costs. The values were between £1304/ ha and £2980/ ha. However the water application by IRPK was higher than by IRWF in the range of irrigation costs of £1.7/ ha mm to £9.2/ ha mm after which the water application reduced drastically below that by IRWF at higher irrigation costs. In the range of irrigation cost of £1.7/ ha mm to £9.2/ ha mm also, the amounts by which the profits for IRNF exceeded those for IRWF were higher than values of potential improvement in weather forecast by only an average of £84/ ha or less than 1% of the profits for IRNF. The water application by IRNF was however 5.1% higher than the applications by IRPK in that range of irrigation cost. This outcome suggests that at the lower irrigation costs in the average year, improvement in weather forecast may lead to increased water application by IRWF. Water savings would however be achieved compared to IRNF with marginal difference in their profits.

In the 1995 season i.e. the dry year, the values of weather forecast for IRNF and IRPSS increased with irrigation cost and were positive at all irrigation costs except in the case of IRNF where the value remained negative up to the irrigation cost of £5.8/ ha mm. The values were higher for IRPSS than IRNF at irrigation costs less than about £13.5/ ha mm. The difference in the values decreased from £545/ ha at the lowest irrigation cost to zero at irrigation cost of £13.5/ ha mm above which the values for IRNF were higher with the difference increasing to £165/ ha at the highest irrigation cost. This implies that the profits for IRNF were higher than for IRPSS at irrigation costs below £13.5/ ha mm. The water application by IRNF was however 16.9% more than that by IRPSS in the season.

At the irrigation costs of £1.7/ ha mm, £3.4/ ha mm and £5.8/ ha mm respectively, the negative values of weather forecast for IRNF were - £470/ ha, - £384/ ha and - £45/ ha or ï 9.3%, -8.35% and ï 10.47% of the profits for IRWF at the respective irrigation costs. However at the irrigation costs IRNF applied 15.4% to 55.2% more water than IRWF.

The value of potential improvement in weather forecast was positive at all the irrigation costs and ranged from £604/ ha at the lowest irrigation cost through a minimum £203/ ha at the irrigation cost of £9.2/ha mm to £781/ ha at the highest irrigation cost or 11.9% through 5.6% to 24.6% of the profit for IRWF at respective irrigation costs. In the irrigation cost range of £1.7/ ha mm to £9.2/ ha mm, the water applications by IRPK were 14.1% to 17.7% more than those by IRWF. The water application was below that of IRWF at higher irrigation costs. The potential value of improved weather forecast was more than the amounts by which the profits for IRNF exceeded those of IRWF at the irrigation costs of £1.7/ ha mm to £5.8/ha mm. The difference was on the average £145/ ha or 3.0% of the profit for IRNF. At the irrigation costs, IRNF water applied 1.1% more water than IRPK. The results suggest that in the dry year at the lower irrigation costs, the benefit of improvement in weather forecast would result in higher profits but through increased water applications by IRWF. The water applications would however, be less than by IRNF with the profits for IRWF also higher than for IRNF.

3.5 Value of Weather Forecast in terms of Efficiency of Irrigation Water Use

The values of weather forecast in terms profit per ha mm of applied water for IRPSS and IRNF in the three seasons are presented in Fig. 4. The values represent the amounts by which the profit per ha mm for IRWF exceeded those of the respective strategies. In the wet and dry years i.e. 1992 and 1995 respectively, the values were positive and increased with irrigation cost for both IRPSS and IRNF. The values for IRNF exceeded those for IRPSS by an average of £6.3/ ha mm and £0.9/ ha mm in the 1992 and 1995 seasons respectively showing that IRNF was less efficient than IRWF and IRPSS in the use of water in the two seasons.

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In the 1994 season i.e. average year, the values were higher for IRPSS than for IRNF with the values being negative for IRNF at irrigation costs lower than £13.2/ ha mm. The difference in their values was £3.3/ ha mm on the average. The negative values obtained for IRNF despite its more generous water application is due to the very high crop price in that year.

3.6 Discussion

The results show that the use of weather forecasts in place of other scheduling strategies can enhance the returns from supplemental irrigation in terms of end of season profits and profit per unit of water application. There is however considerable variation between years as was also found by Fox et al. (1999a) in a similar evaluation of weather forecast information in winter wheat production.

In the 1992 season (i.e. the wet year), values of weather forecast in terms of end of season profit were positive for at all irrigation costs were achieved through greater cost savings from unnecessary irrigation. In contrast, the values of weather forecast were either very small or negative at all the irrigation costs in 1994 (i.e. the average year). In 1995 (i.e. the dry year) the value of weather forecast becomes positive only at relatively high irrigation cost assumptions. Clearly, the use of weather forecast does not always pay, as was also found by Fox et al. (1999b) in the case of alfalfa hay production. The negative values of weather forecast however occurred under significantly higher water applications that may not be feasible in practice due to the resource constraint typical of such years.

Under such resource constraint, the advantage of using weather forecast becomes apparent as shown in the positive values of weather forecast in the dry year when the values are in terms of profit per ha mm of irrigation (Fig. 4). This demonstrates the advantage of the use of weather forecast over IRNF and IRPSS in rational and efficient use of scarce or expensive water supply.

The value of weather forecast in terms of profit per ha mm of irrigation that was mostly negative for IRNF in the average year reflects the effect of the very high crop price for the year. An analysis (not reported) of a simulation result of Ejieji (1998) obtained with a reduced crop price of £82.6 /t for that year showed that the values of weather forecast for IRNF in terms of profit per ha mm followed a trend similar to that in the dry year.

The values of potential improvement in weather forecast obtained in this study suggest that the benefit in terms of higher profit from the use of weather forecast would increase with the improvement in the quality and lead-time of short-term forecasts. The benefits in the wet year would result from more cost savings from unnecessary irrigations. In the average and dry years the benefits would result from more rational use of water with increased water application at low irrigation costs and reduced water application at high irrigation costs. The increased water application would nevertheless still be less than water application by IRNF and, the efficiency of water use better than IRNF and IRPSS.

It should be noted that the analysis in this study did not explicitly consider environmental costs except in so far as they are represented by the use of excess water. Also, the incorporated crop model did not include any yield penalty function for excess irrigation and nutrient leaching. The soil profile was assumed to be freely draining. Inclusion of such cost and penalty factors may likely result in higher values of weather forecast than reported for the strategies especially at the lower irrigation costs in the average and dry years.

The threshold irrigation cost at which irrigation cost at which the end of season profit equalled that of the unirrigated crop for all the strategies (i.e. the threshold irrigation cost for beneficial irrigation) has been reported elsewhere (Ejieji, 1998). It was found to be highest for IRWF compared to IRNF and IRPSS except in the average year where the threshold cost of £16.5/ ha mm for IRNF was the highest. Based on the estimated actual variable cost of irrigation for 1994, the percentage contribution of water charges to the variable cost and the likely scenario for water charge escalation, Gowing and Ejieji (2000) concluded that for such average years, only real water-scarcity or institutionally limited water supply rather than water price signal are likely to force the irrigator to use water more efficiently. Outright banning or restriction of water abstraction is however usual in a typical resource-stressed summer (Upton, 1995; University of Newcastle Upon Tyne, 1990). This study has demonstrated that compared to the alternative strategies of irrigation scheduling, the use of weather forecast under such conditions of resource constraint would increase the benefit per unit of applied water through improved efficiency of water use.

4. CONCLUSION

It has been demonstrated that the use of available weather forecast in irrigation scheduling could potentially yield increased benefit from supplemental irrigation over that from strategies which do not use the weather forecast but rely only on soil water deficit projection or on soil water deficit projection combined with a simple assumption of wet weather persistence. The value of weather forecast expressed as the increase in benefit over that of the alternative strategies would increase with irrigation cost.

The increased benefit of using weather forecast in the wet years would be more profit derived from cost savings from unnecessary irrigations. In the average and dry years the increased benefit would be in terms of higher profit per unit of water application resulting from improved efficiency in the use of limited or expensive water supply.

Improvement in quality and lead-time of forecasts would result in greater profit from the use of weather forecast in the wet year through more cost savings from unnecessary irrigations. In the average and dry years however, water application may increase with the use of weather forecast if the irrigation cost is low or decrease if the irrigation cost is high with increased profit in either case. The level of water in any increased application would nevertheless be lower than for irrigation without weather forecast with potentially higher profits for using weather forecast than for not using weather forecast. Efficiency of water use and the resulting benefit would also be higher for using weather forecast than for the strategies that do not.

It should be emphasised that the values of weather forecast reported for the various strategies were measured against a model that attempts to use the available weather forecast optimally in irrigation decisions. The actual economic value achievable in practice would therefore depend on the effective use of the available weather forecast information. It should be noted that the weather forecasts were taken at their face value in this study without consideration of their performance characteristics. The crop price uncertainty faced by the irrigator was also not considered. This study however demonstrates the advantage of the use of weather forecast in practical decision-making for supplemental irrigation scheduling based on simulated demand.

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HYDRAULIC MODELING OF BEGEMANN GATE FOR EFFECTIVE WATER MANAGEMENT SYSTEM IN HADEJIA VALLEY IRRIGATION PROJECT (HVIP), NIGERIA

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ABSTRACT

A mathematical model of Begemann gate was developed for Hadejia Valley Irrigation Project (HVIP) located in Sudan Savannah of Northern Nigeria. Begemann gate is an automatic check gate installed at the main canal of HVIP system with principal aim of regulating (maintaining) upstream water surface profile at constant depth under variable discharges passing through it. The hydraulic gate is made up of structures: a weir and a check gate which is simple to manufacture, easy to replace when damaged and self operational with the control of the discharge at off-take among other advantages. The principal hydraulic parameters considered in the modeling process include the total discharge (Qt) passing through the gate, discharge under the gate (Qd), sides discharges (Qc), and the pressure force acting on the gate plate.

Two mathematical models were developed based on principle and concept of trajectory of free over-fall; and energy conservation as well as the concept of real and virtual weirs. The models of trajectory of free over-fall given by Davis and Cavailhe were evaluated using a prototype model installed in the irrigation laboratory of ENSAM, Montpellier, France. The predicted flow equations (models) of Qt, Qc and Qd are statistically the same with the measured values at 5% level of significance. The covariance of the Cavailhe model (163.0) is twice that of Davis (82.27) when compared with the measured ones indicating that the latter is a better estimator of the hydraulic behavior of the Begemann gate model under real field situation.
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KEY WORDS: Begemann gate, side discharge, discharge under the gate, off-take, free overfall

1.0 INTRODUCTION

Water flow in irrigation canal of gravity system is usually maintained above certain hydraulic head for easy diversion into distributory canals or field channels. This is achieved by the use of hydraulic structures installed across the canals in forms of check gates, cross regulators, weirs and automatic control gates. They protect the canal against the erosive force of flowing water as it can be seriously damaged by a scouring water flow.

Generally, the flow in the main (or secondary) canal that feeds secondary or tertiary canals vary from 100 percent maximum design flow in the peak of irrigation to about 20% (Brouwer, 1987) when irrigation demand is low. Thus, seasonal or even daily variations occur as upstream turnouts are opened or closed according to the demand of the irrigation plots. While this variation in the canal flow is necessary for economic use of water, it is essential to reduce the variations in upstream water levels caused by variations in canal flow through the use of automatic check gates

Begemann gate is one of these automatic check gates. Its advantages are many: simple to manufacture, simple to adjust to the required/desired upstream water levels, easy to replace when damaged, practically maintenance free and self operational with the control of the discharge at off take. Despite its enormous advantages the gate is relatively unknown in the field of irrigation. Except in Bangladesh, Nigeria and USA, there are few other countries where this type of gate is employed in irrigation schemes (Litrico, 2001). Begamann gate has been a subject of research work to hydraulic Engineers for a long time. Vlugter (1940) studied diverse configurations of the gate and made modification for use in an area with a relatively small hydraulic head (gradient < 1%). Similarly, it has been observed that the gate had become a subject of recent publications (Litrico 2000) with the reported works of De Graff, (1998), Raemy and Hager, (1998), Burt et al (2001) among others. The main research interests of the gate is the hydraulic modeling of the force exerted on the gate and the discharge passing through it. As rightly observed by Litrico (2000), the main problem in modeling the Begemann gate is the computation of the force exerted on the plate by the water. In closed position, the pressure distribution is hydrostatic but this pressure becomes more complicated to compute when the gate opens. The approach followed by Raemy and Hager (1998) and Burt et al (2001) was deriving an expression for determination of the pressure force experimentally. De Graaf (1998) used momentum on conservation of energy between the upstream and down stream to compute the pressure force on the plate. These methods are difficult to use in a real situation on the field where the channel geometry is complicated as in the case of unlined canals. On the modeling of the discharge passing through the gate, it can be observed that the gate function hydraulically different when the gate is slightly open with a small angle of opening as compared to when it opens with maximum angle of opening (Cavailhé, 2001). In a small angle of opening, the gate functions as an orifice while with a maximum angle of opening the gate functions as weir due to minimum interference of the gate on the water profile passing through. Consequently, the modeling of the discharge passing through the gate must take into consideration these different hydraulic behaviors of the gate as it functions. It is therefore imperative that the structures are studied more closely to explore their full potentials for wider applications in the country. Therefore, the objective of the study is to develop hydraulic models that take into account the discharge passing through the gate and thus, exert force on the plate of the gate, which can be used to analyze the performance of the distribution system among other usages.

2.0 MATERIALS AND METHODS

The methodology was divided into two: the first part was the theoretical development of the gate equations and the second part was the experimental evaluation of the developed equations using physical model of the gate

2.1 Theoretical Development of Discharge and Pressure Force Equations of the Gate

The working principle of Begemann gate is simple. It is made of a steel plate rotating around a horizontal axis to a pivot located above the upstream water level. In a closed position, the plate closes along the broad crested weir, where water at the upstream of the gate forces the gate to open while the counterbalance weight tends to keep it in a closed position. At a given water depth (full supply level FSL or a desired hydraulic head) at the up stream of the gate, the moment due to the weight of the counterbalance plus the weight of the gate is in equilibrium with the moment due to the force of the flowing water acting on the plate. A small rise in the water level above FSL pushes the gate further open and increases the flow through the gate while a drop in the water level is checked by the closing of the gate (Othman, 2002). Thus, the FSL at a depth ñhò in the canal is maintained with variable flow rate (Fig. 1 and 2).

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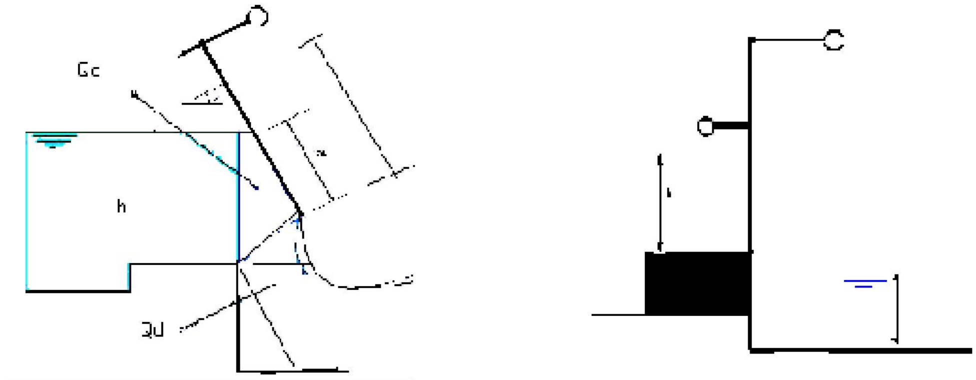

Figure 1 and 2 Schamatic Diagrams of Begemann Gate in opened and closed positions

Hydraulically, the gate is made up of two different hydraulic structures, weir and the gate plate and arms. Each of these structures plays important role for the operation of the gate. As mentioned earlier ,the gate functions as weir structure when it is open at maximum angle, at this position the gate gives a minimum disturbance to the surface water profile passing on the weir. When it functions with a small angle of opening (as orifice) it has tremendous effect on the discharge passing through it. Consequently, the discharge through the gate depends principally on the angle of opening and depth of the gate opening. Similarly, the pressure force that is exerted on the gate plate also depends on the discharge, angle of opening and the design upstream head (FSL).

2.1.1 Discharge equations

There are two different discharges passing through the gate: the discharge directly, under the gate (Q_d) and the discharge through the two sides (Q_c) of the gate (Figures 2 and 3). Q_d is directly proportional to the gate opening while Q_c increases proportionally with the gate opening up to a point where it begins to decrease with the increase of the angle of opening.

Fig. 3 Physical model of Begemann gate

Thus, the total discharges Q_t:

$Q_t = Q_d + 2Q_c$ .... (1)

It is apparent that there are two discharge limits; the lower limit when the gate is closed, the discharge is zero and the upper limit when the gate is completely opened; the discharge is at maximum level Q_{max}.

2.1.2 Side Discharge Equation

Considering figures 1 and 2 which show the surface area where the side flows Qc passes through the gate, it can be said that Qc is a function of some parameters of the gate. Hence:

$$\mathbf{Qc = f(S, C_d, K, g, h)} \quad \text{é é é . (2)}$$

Where S = surface area in which side discharge passes through (m^2)
C_d = Coefficient of discharge (no unit)
K = Multiplier depending on the angle of opening.
g = Acceleration due to gravity m/s^2
h = Upstream head (m)

Experimentally, it can be observed that, as the angle of opening δ starts from zero to maximum value: the value of Qc starts from zero, reaches its maximum at angle between zero and maximum and then decline to zero level at the angle δ maximum. Therefore; a multiplier K was chosen to take care of this phenomenon. Thus, K is given as;

$$K = \left(1 - \frac{\delta}{\delta_{max}}\right) \quad \text{é é é .. (3)}$$

Where
δ = A given angle of opening.
δ_{max} = maximum angle of opening at which the gate just touches the surface water profile passing over the Broad-crested weir of the gate.
In order to determine δ_{max} it is necessary to consider the equation of surface water, profile passing on the weir (trajectory of free over-fall). Two equations of water surface profile were chosen, those given by Davis et al (1999) and Cavailhé (2001). Thus;

$$Y = \frac{2}{3}y_c - x\tan\alpha - \frac{1}{2y_c}\left[\frac{x}{\cos\alpha}\right]^2 \quad \text{............ (4)}$$

$$Y = h_e - x\tan\beta - \frac{g}{2}\left[\frac{x}{V\cos\beta}\right]^2 \quad \text{............ (5)}$$

Where
y_c = critical depth, the upstream depth above the weir
It is given by Cavailhé, (2001) as

$$y_c = \left[\frac{Q^2}{gL_a^2}\right]^{1/3}$$

Q = discharge in m^3/s passing over the weir
g = acceleration due to gravity in m/s^2
L_a= width of the canal in m

$$\alpha = \tan^{-1}\left[\frac{\frac{1}{3}y_c}{L}\right] \qquad \text{Refer to figure 4}$$

L = length of the weir in m
$h_o = y_c$

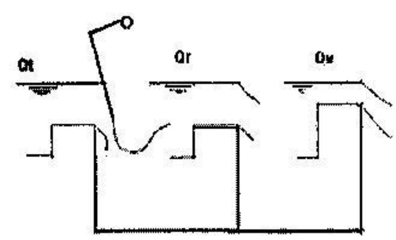

Fig. 4. The concept of real and virtual weirs

$$h_e = \frac{2}{3} y_c = \frac{h_o f^2}{f^2 + \frac{2}{5}}$$

f = Froude Number

The two equations were experimentally evaluated as discussed in 2.2. It can be noted that the two equations are in form of quadratic equation ($y = ax^2 + bx + c$). As the gate swings to open/close, it makes an arc of a cycle with a equation in the form of $R^2 = x^2 + y^2$

When resolved using trigonometry δ_{max} is found as:

$$\delta_{max} = \sin^{-1}\left(\frac{x+p}{R}\right) - \sin^{-1}\left(\frac{p}{R}\right) \qquad \text{..................} \; (6)$$

$R^2 = L^2 + p^2$

Therefore,

$$Y = (ax^2 + bx + c) - (-\sqrt{R^2 - (x-p)^2} + L \qquad \text{é é é} \quad (7)$$

Where the parameters a, b and c are given below:

Case I. Equation of Cavailhe

$$a = -\frac{1}{2 y_c (\cos\alpha)^2} \; ; \qquad b = -\tan\alpha \; ; \qquad c = \frac{2}{3} y_c$$

Case II. Equation of Davis

$$a = -\frac{g}{2V \cos\beta} ; \qquad b = -\tan\beta \; ; \quad c = h_e$$

Equations (6) and (7) are solved using computer software *Mat lab* thereby determining δ_{max} and thus, computing the multiplier K for a given (any) angle of opening

The surface area S is determined considering the geometry of the cross section in which the discharge passes through (figure 2)

The surface area (S) is determined as

$$S = (h - U^*)\left[Uh - (h - U^* \frac{\tan\delta}{2}\right] + \frac{UhU^*}{2} \qquad \text{é é é} \quad (8)$$

Thus, using energy Q_c is determined as (referring to equation 2)

$$Q_c = C_d K S \sqrt{2gh} \qquad \text{é é é} \quad (9)$$

Where C_d is the coefficient of discharge.

Having determined Q_c, what remains is either to determine Q_d or Q_t. Note that Q_t is the sum of Q_d and Q_c. It can also be observed that Q_t is equivalent to the discharge over the weir when the gate opening is at maximum level. In this position, Q_t can be computed using flow equation over weir but at any other position the equation of weir cannot be used to determine Q_t. Using equation (9) and a selected discharge coefficient, side discharges were determined from ñδò equal to zero up to maximum attainable angle of 17^o. Similarly, side discharges were measured for the same angles of openings. Consequently, to determine Q_t, an assumption is made with two hypotheses using the concept of real and virtual weir

Referring to figure (4) above, it can be said that:

$$Q_t = Q_r - Q_v \qquad \text{é é é} \quad (10)$$

Where:

Q_t = Total discharge passing through the gate

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Q_r = Discharge passing over the real weir
Q_v = Discharge passing over the virtual weir

Therefore

Q_t (h, δ) = Q_r (h) ï Q_v (h_{sv}, δ)

Hypotheses:

i) if δ = 0, then Q_t = 0 and Q_r = Q_v gate is closed and there is no flow
ii) if δ = δ_{max} then Q_t = Q_{max} and Q_v = 0 flow at maximum discharge

Using energy equation, Q_v and Q_r can be obtained, thus,

$$Q_v = C_d b_v \sqrt{2g}\left(h - h_{sv}\right)^{\frac{3}{2}} \qquad \text{é é é} \quad (11a)$$

$$Q_r = C_d b_v \sqrt{2g}\left(h\right)^{\frac{3}{2}} \qquad \text{é é é} \quad (11b)$$

Where b_v = width of the virtual weir
b_r = width of the real weir
h = hydraulic head on the real weir
h_{sv} = hydraulic charge on the virtual weir.

Hence, referring to equation 1

$$Q_t = Q_d + 2Q_c = Q_r - Q_v = C_d b_v \sqrt{2g}\left[h^{\frac{3}{2}} - (h - h_{sv})^{\frac{3}{2}}\right] \qquad \text{é é} \quad (12)$$

$$Q_d = C_d b_v \sqrt{2g}\left[h^{\frac{3}{2}} - (h - h_{sv})^{\frac{3}{2}}\right] - 2C_d KS\sqrt{2gh} \qquad \text{é é é} \quad (13)$$

and Qt can be determined using equation 10 at any given angle of opening.

2.1.3 Pressure Force Equation

When the gate is in closed position, the pressure distribution is hydrostatic but in opened position, it is no longer hydrostatic as it changes with the angle of opening. There are fundamentally two opposing forces, the one(s) pushing the gate to open and the other(s) pushing the gate back to remain in a closed position. The equation of the pressure force is determined based on the assumption that at any given angle of opening the two opposing forces must be in equilibrium.

$$F_1 + P_1 = F_2 + F_x \qquad \text{é é é} \quad (14)$$

Where

F_1 = force due to flow velocity at position 1
P_1 = hydrostatic pressure force at position 1
F_2 = force due to flow velocity at position 2
F_x = component force due to weight of the gate

Thus;

$$\ell Q_1 V_1 + \ell g \frac{h_1^2}{2} b = F_2 + F_x$$

F_2 is the sum of pressure force and force due to flow velocity at the down stream level. However, in reality F_2 doesnøt have any effect on the gate as the activities downstream level is quite below the plank of the gate in this case (Begemann gate) not like the situation with Vlugter gate. Consequently, F_2 is safely neglected.

Hence;

$$F_x = \ell Q_1 V_1 + \ell g \frac{h_1^2}{2} b \qquad \text{é é é} \quad (14b)$$

But;

$$V_1 = \frac{Q_1}{bh_1}$$

And

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$$F_v = \frac{F_x}{\cos\delta}$$

Therefore

$$F_v = \frac{\ell}{\cos\delta}\left[\frac{Q_1^2}{bh_1} + \frac{gh_1^2 b}{2}\right] \quad \text{é é é} \quad (14c)$$

At equilibrium the force(s) responsible for opening the gate is always equated to the force F_v trying to close it.

2.2 Laboratory Evaluation of the Developed Equations

A Laboratory experiment was conducted to evaluate the developed equations of discharges passing through the Begemann gate at the irrigation laboratory of ENSAM, Montpellier, France. A physical model was constructed based on dynamic similarity with a Begemann gate in operation at the irrigation scheme of HVIP, Hadejia, Nigeria, using ratio of Froude number as described by Yalin, (1971). The pysical model was installed in the irrigation canal of ENSAM (see fig.5). A regulator (modules à masques) was installed at the upstream, which supplies the canal with water for a desired discharge up to maximum of 65l/s. The gate angle of opening was measured using potential meter, which translates angle of rotations of the gate into resistance in ohms. Two small plastic canals were installed at the two sides of the gate for measuring the side discharges and thin-crested (calibrated) weir was similarly installed at the downstream level for measuring the total flow rate passing through the canal (gate). The water level at upstream of the gate was measured using meter rule installed at upstream position. Two set of data were measured. The first was water profile for different discharges passing over weir without the gate. Small alloy rods installed in a plastic frame following the pattern of the water trajectory was used to measure the water profile. This data was obtained to evaluate the two water profile equations developed by Davis et al. (1999) and Cavailhé (2001). The second set of data were angles of opening δ, side discharges $Q_{c,}$ upstream water level h and total discharge Q_t for different flow rates ranging from 10 l/s to 60 l/s at 5 l/s interval. The experiment was repeated three times for statistical analysis.

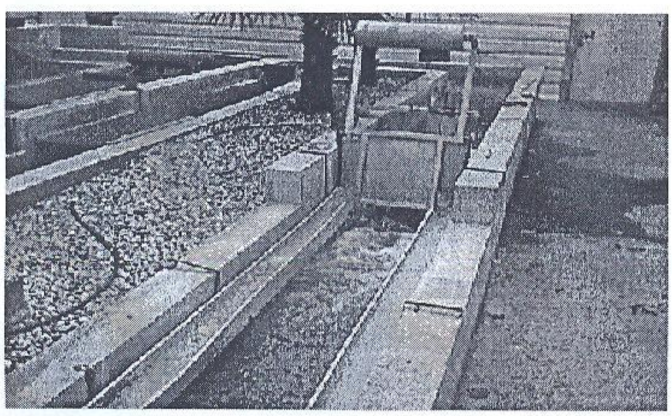

Figure 5 The prototype model installed in Irrigation lab. of ENSAM, Montpellier, France

3. RESULTS AND DISCUSSION

3.1 Comparison of Equations for Water Trajectory Lines

To measure the accuracy of the water profile equations (4) and (5) which describe the type of water profile which has direct relation with equation (1), the measured values were compared with the computed values.

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Table 1 presents the water profile depths over the weir. The data is represented in Figure 6 (where the water profile depths were plotted against the horizontal distance from the edge of the weir). The line of trajectory estimated by Cavailhéôs equation was found to under estimate the profile depths when compared with those directly measured, while the Davis equation gave a relatively better estimation with the measured values in comparison with the equation of Cavailhé. This means that the Davis equation is more sensitive in measuring discharge over weir, taking into account the horizontal distance covered by its trajectory, as compared with Cavailhé equation. From Figure 6, the trajectory line from Cavailhé equation would terminate earlier than that of Davis for the same co-ordinate axis. It is to be noted that the value for maximum angle of gate opening depend heavily on the positions of the surface water profile shooting over the weir. Hence it is important to use an equation which gives a better prediction of the water surface profile. Thus, the equation of Davis was found to predict better values of water profile at different angles of opening compared with one given by Cavailhe as shown in table 1. Statistically, the covariance of Davisôs equation (82.27) is lower than that of Cavailhe (163.01). Consequently, the use of Davis equation is recommended for use of the developed model.

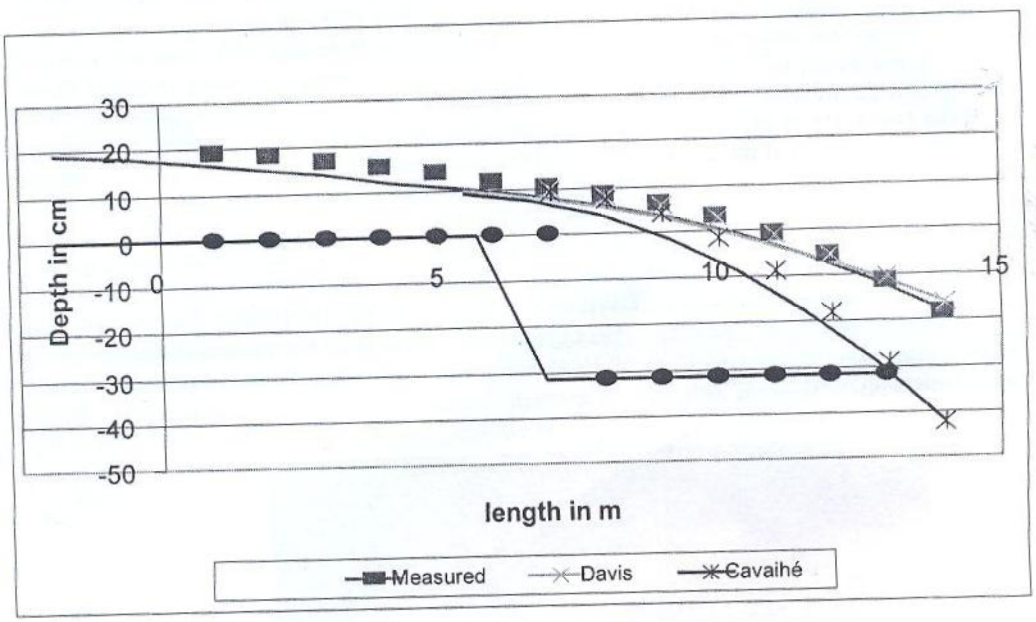

Fig. 6. Surface water profiles for measured and predicted equations of 60l/sec discharge

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Table 1 Water profile depths of measured and predicted values from Davis and Cavailhé equations. The reference point is on top edge of the broad crested weir

Distance From the edge of the weir(cm)	Depth (cm)		
	Measured X	Predicted using Davis Equation Y 1	Predicted using Cavailhé Equation Y2
0	10.20	9.75	9.10
5	8.50	8.08	7.40
10	6.10	5.72	3.83
15	3.20	2.68	-1.61
20	-0.70	-1.04	-8.93
25	-5.50	-5.45	-18.12
30	-11.50	-10.54	-29.18
35	-18.50	-16.32	-42.12
Covariance (X Y1, X Y2)		82.27	163.01

3.2 Sides and Total Discharge Equations

The predicted discharges were graphically compared with the measured sidesô discharges experimentally. Figure (7) shows predicted and measured discharges. It is to be noted that the predicted values reasonably agreed with the measured values at 5% level of significance. The predicted total discharge Q_t passing through the gate (The sum of the discharges passing under the gate Q_d and the two sides of the gate Q_c) and the measured discharge were similarly compared. Table 2 shows the predicted total discharge and the measured discharge, it also shows the discharge under the gate Q_d and sides discharges for the angle of openings used for the experiment. From figure 7, the estimated and measured sidesô discharges were reasonably within same range for the range of angle of opening. The curves for both estimated and measured show similar pattern; that the side discharges were zero at zero angle of opening and continued to increase to an optimal angle of opening therefore continue to decline until it approached zero at the max angle of opening. This pattern describes a normal distribution curve with the optional angle (in this case, figure 7) at 10^o, corresponding to a discharge of 8.4L/S. Table 2, however, presents two values for both predicted using the model and measured discharges which followed similar pattern with R2 value of 0.975. The pattern of these values were found to be different with those estimated for side discharge. They do not describe a normal distribution with the same range of angle of opening as in the case of side discharges. This dissimilarity is attributable to the effect of the discharge directly under the gate (Qa). As described by the relations, the higher the angle of opening the more the total discharge tend to equalize with the direct discharge under the gate ($Q_t \longrightarrow Q_d$) while the side discharges tends to zero value($Q_c \longrightarrow 0$). The optimization of the angle of opening for the gate is applicable to side discharges while a near to linear relation is the dominant feature of the curves for total discharges.

3.3 Optimization Angle of Gate Opening

The data presented in Table 1 and 2 as well as the curves in Figure 7 indicates that the value for the maximum angle of gate opening depend largely on the position of the surface water profile shooting over the weir. Hence, it is important to use an equation which give a better estimation of the water surface profile. Thus, Davisô equation (1999) was found to give higher values of angle of opening compared with one given by Cavailhe. Consequently, the use of Davis equation is recommended for use of the developed model.

Table 2 Predicted and measured discharges

Angle of Opening in Degrees	Total Discharges		Side discharge Qc	discharge Under Qd
	Qt measured l/s	Qt predicted l/s	l/s	l/s
0	0	0	0	0
1,35	15,01	7,41	3,19	4,23
2,84	21,59	15,48	6,59	8,89
4,00	29,61	21,69	9,16	12,53
5,26	34,33	28,26	11,79	16,47
6,47	38,70	33,99	13,73	20,26
8,10	43,31	41,24	15,88	25,36
9,82	48,86	47,91	17,16	30,75
12,99	52,48	56,79	16,13	40,66
15,78	60,08	60,73	11,33	49,40
17,27	61,66	61,53	7,46	54,07
R^2 value 0.975				

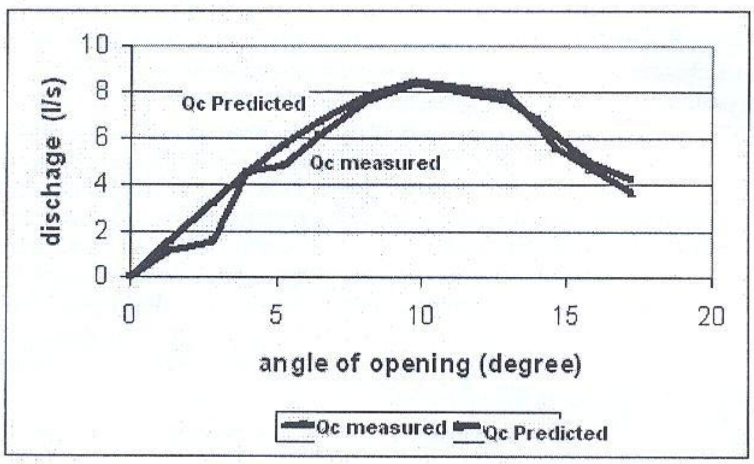

Figure 7 Graph of side discharges predicted and measured

4. CONCLUSION

The hydraulic modeling of Begemann gate for the discharge equation(s) and the pressure force acting on the gate were successfully developed and evaluated for incorporation into a procedure that form a model and the discharge equation which link upstream water level and angle of opening is a significant input in irrigation canal management. Principally, the aim of the Begemann gate is to maintain upstream water surface profile at a constant depth under variable discharges passing through it. In the development of this model the activities at the immediate down stream level of the gate were ignored as these activities practically do not influence the gate functioning. The model can be used in irrigation canal simulation when incorperated with appropriate equations (models) of various hydraulic structures found along irrigation canal network. This is a significant input in simplifying canal management.

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