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This is the 186th issue since its maiden issue in the Spring of 1971

EDITORIAL

Happy New Year to AMA readers around the world. The 20^{th} century was over and after the beginning of the 21st century, 20 years has already passed. During the past century, the world suffered various wars, handled severe conflicts and struggled with food shortages, but the world's population has been steadily increasing toward 10 billion people. However, the extend of agricultural land in the world has not increased. So the agricultural land per capita in the world continues to decrease. The divide between rich and poor in the world continues to be widen. In such condition, stress is mounting high and tougher competition for resources leads to major conflicts.

Agriculture produces food that supports the lives of humankind, but the farmers in the world who carry agriculture are never rich. The reason being both unfavorable and unfair trade conditions for agricultural practitioners, as compared to the non-agricultural side. Employees at factories and companies have ways to show their dissatisfaction and to demand fair compensation, if their salary is low, but due to lack of formalized benchmarking system, the farmers may not even do so. They continue to produce food silently and honestly for the people in the world.

It can be said that Japanese rice has been maintained at a fairly high price by government policy. It costs you about 200 yen per kg of brown rice. However If you buy two 500cc PET bottles at a convenience store, you have to pay 200 yen or more. Although it is claimed that Japanese rice is the most expensive in the world, but it is still cheaper than water. This is the market situation for the farmers. Of course, there are some farmers who run very large-scale farm and make big sales and profit, but there are a lot of small-scale farmers in the world. Japan is a mountainous country, and nearly 60% of agricultural land exists in the mountainous areas, and the operational land for farming are very small and difficult for mechanization with higher labor productivity.

There are many such farmlands all over the world. Similarly in China, about one-third of the total farmland area is in the mountains. In North America with very large operational section where one section is about 250 hectares, they can increase the labor productivity of agriculture by using big and powerful machines and expanding work width. That's not the case with mountain or small scale farming. There is the need for new mechanization approach that can increase the labor productivity of agriculture considering such many separated small plots.

We are entering new era where it is possible. In other words, instead of using a 1000 horsepower from tractors, new system could use 200 small agricultural robots with 5 horsepower each. I believe that the future agricultural mechanization of the world will be the agricultural mechanization of the brain using AI.

AMA started publishing in 1971. AMA aims to promote agricultural mechanization for developing countries in order to reduce the gap between the farmers in developing countries and the cities in developed countries. For realizing the peaceful world, we must raise the income of farmers and they can live happily by promoting new mechanization. AMA celebrated its 50^{th} anniversary last year, and I would like to continue to do my best for the agricultural mechanization of the new world, aiming for the next 100th anniversary.

Thank you for your cooperation.

Yoshisuke Kishida
Chief Editor
January, 2021

CONTENTS

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

Vol.52, No.1, Winter 2021

Yoshisuke Kishida	5	Editorial
V. V. Aware A. K. Mehta	7	Development of Pedal Operated Arecanut Dehusker Based on Ergonomical and Mechanical Considerations
S. Zaree J. Khodaei	13	Determination of Essential Indexes in Assessing the Status of Agricultural Mechanization in Kurdistan, Iran
Guillen Sánchez Juan Santos G. CamposMagaña Carlos Sánchez López Oscar M. González-Brambila Gabriela Ramírez-Fuentes	17	Evaluation Parameters Effeccting the Performance of Vibrating Vertical Tillage Equipment, First Stage
Shrinivas Deshpande G. Senthil Kumaran A. Carolin Rathinakumari	24	Development of a Watermelon (*Citrullus lanatus*) Seed Extractor
N. R. Nwakuba	30	Optimization of Energy Consumption of Okra Slices in a Solar-assisted Electric Crop Dryer
R. Pandiselvam, M. R. Manikantan A. C. Mathew, Shameena Beegum K. B. Hebbar	39	Design, Development and Evaluation of Minimal Processing Machine for Tender Coconut (*Cocos nucifera*)
Nawaf Abu-Khalaf Yasser A. R. Natour	44	Agricultural Machinery Manufacturing Sector in Palestine - Reality and Challenges
Ajay Kumar Verma Samir Santiya	48	Modification of Rotary Unit of Power Tiller for Biasi (Interculture Operation) Rice Cultivation in Eastern India
K. P. Singh, Dilip Jat Avinash Kumar Gautam M. P. S. Chouhan	55	Design of Rotary Assisted Broad Bed Former-cum-Seeder for Vertisols
Qizhi Yang, Jing Cai, Xin Zhou Ibrar Ahamd, Jianping Hu, Jun Gu	61	The Optimization of Topological Mechanism and Dimension Design of Parallel Transplanting Machine in Greenhouse
Said Elshahat Abdallah Wael Mohamed Elmessery Ahmed Elseify	67	Implementation of Image Processing and Fuzzy Logic Discriminator of Hatching Eggs Fertility
A. M. Mousa E. A. Darwish	74	Performance Evaluation of a Multi-crop Shelling/Cracking Machine for Shelling of Peanut Pods
Engin ERGÜL Bülent ÇAKMAK	81	Power Requirement and Fuel Consumption Reduction of Forage Harvester Chopper Blades by Thermal Coating

Retirement Message (R. L. Kushwaha)..............23
New Consultant Editor (P. Soni).........................54
Co-operating Editors 91
Subscription Information 98

Development of Pedal Operated Arecanut Dehusker Based on Ergonomical and Mechanical Considerations

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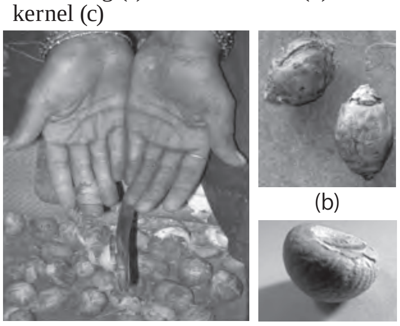
Abstract

The dimensions of pedal operated arecanut dehusker that is hopper height, handle diameter and length, saddle height, etc. were optimised based on anthropometric data of male agricultural workers. The diameter and length of the dehusking cylinder were 350 mm and 500 mm, respectively. The cylinder operational parameters viz, peripheral speed and feed rate were 7 m/s and 15 kg/h, respectively. The flywheel weighing 12.5 kg with rim diameter as 350 mm was opted for reducing cylinder speed fluctuations. The developed pedal operated arecanut dehusker was ergonomically and mechanically evaluated to insure its performance. The mean working heart rate, oxygen consumption rate and energy expenditure rate were 126.5 (± 8.4) beats/min, 1.30 L/min and 27.23 (± 2.79) kJ/min, respectively. The average dehusking capacity, efficiency and kernel breakage were 15.1 kg/h, 77.9% and 4.3%, respectively using single concave. Dehusking efficiency and kernel breakage were 96.6% and 6.7%, respectively when both concaves were used one after another.

Plate 1 Injury prone traditional arecanut dehusking (a), Arecanut fruit (b) and kernel (c)


(a) (b) (c)

Introduction

Arecanut is a kernel obtained from the fruit of arecanut palm (*Areca catechu Linn*.). Arecanut is a tropical crop belonging to the family Palmae. During 2006-07, total-world area under arecanut was about 703 thousand ha with production of 854 thousand tonnes (Anon., 2008) and productivity 1,215 kg/ha. In India, arecanut was grown on 414 thousand ha with production of 535 thousand MT and productivity of 1,292 kg/ha during the year 2011-12 (Anon., 2015).

The moisture content of freshly harvested arecanut fruit is about 68 to 70% (w.b.) (Anonymous, 2012). Arecanuts are harvested when those reaches physiological maturity. After harvesting, fruits are dried to reduce moisture content up to 8 to 10% (w.b.). Arecanut fruits are dried in sun spreading evenly in single layer on the ground or on the roof top (concrete or tin sheet as per availability). The fruits are turned over at regular intervals to facilitate uniform drying. Moisture content of fruits reaches up to 8 to 10% within 35-40 days. After drying, arecanuts are dehusked manually using country kitchen tool direct at farmer's house or at small scale processing units. The kitchen tool being used for the purpose has a wooden plank and curved MS blade. The whole kernel is separated by taking two to three cuts to the dried fruit. Each individual fruit is handled separately which makes the dehusking process time consuming, labour intensive, uneconomical and above all unsafe for fingers and palms. It is observed that dehusking operation constitutes about 35 to 40% of the total cost of processing (Varghese and Jacob, 1998).

Traditional arecanut dehusking process with arecanut fruit and kernel are shown in **Plate 1**. In order to increase productivity few scientists have developed motorized arecanut dehusking machines (Balasubramanian (1985), Varghese and Jacob (1998) and Bundit J. et al. (2009). Also, these machines were developed for arecanut of particular varieties available at the respective region. When used for other varieties, these machines give poor dehusking efficiency and high kernel breakage. Hence, it was necessary to have arecanut dehsuker suitable for the varieties available in Konkan region. Effort was made by Baboo (1981) for manual dehusking machine of arecanut but much success was not achieved because of low capacity of machine. In view of this, a manual arecanut dehusking machine suitable for small and marginal farmers was needed. Human energy has generally been utilised through arms, hands and back. The capacity for arm exercise is dependent upon the amounts of muscle mass engaged (Shephard, 1967). A person can generate more power (about four times) by pedalling than by hand cranking (Wilson, 1986). The use of pedal power seems to be potentially advantages in agriculture sector, where electrical or internal combustion engine power is unavailable (Tiwari et. al., 2011). Hence considering all these points, the pedal operated arecanut dehusker was developed based on ergonomical and mechanical parameters.

Materials and Methods

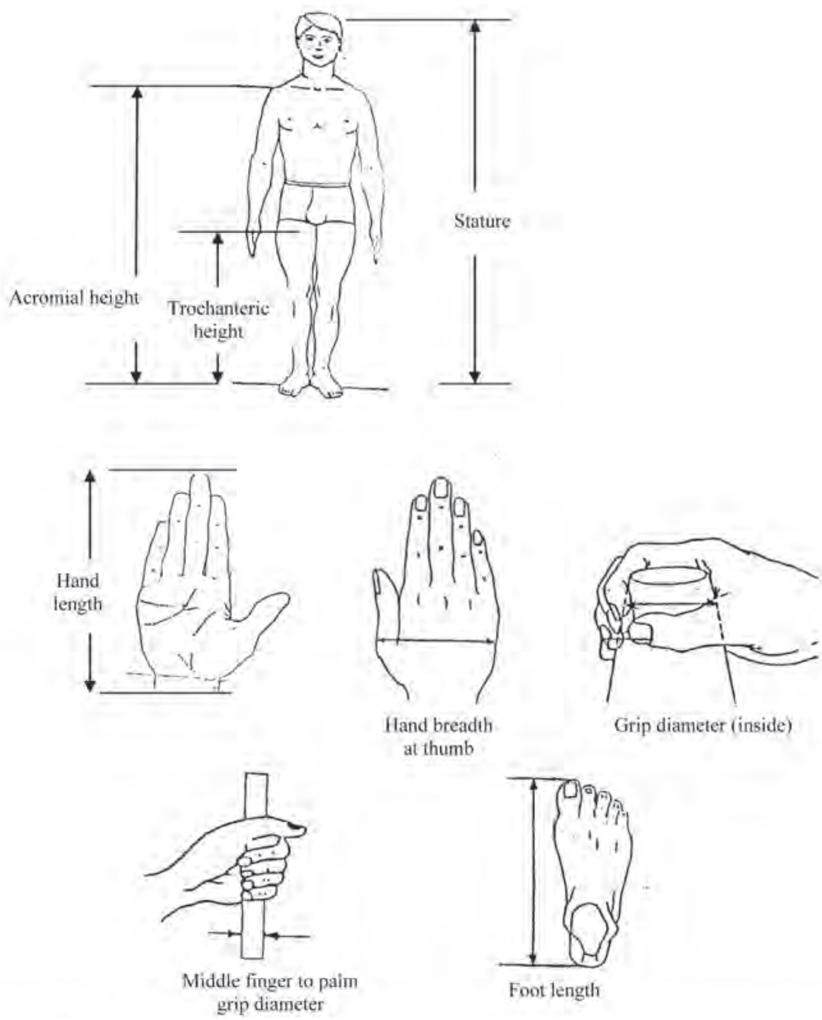
Power Output and Pedalling Rate

Shah (2005) reported that pedal powered electricity generator unit could generate 80 W by pedalling for an hour at the pedalling rate of 40 to 50 rpm. Chen et al. (1999) reported that minimum oxygen consumption rate was at 60 rpm and increased at both higher and lower pedalling rates. Wilson (1986) reported that the reasonable workload for continuous power generation for the western population would be about 75 W for a young and healthy person. He also reported that pedalling at a load of about 90 W could be sustained for around 60 minutes. In view of these facts, the power outcome from a farm worker was considered as 75 W for one hour duration. The present arecanut dehusker was developed to be operated with 75 W power and 60 rpm pedalling rate.

Arecanut Dehusking

Power required for the dehuking of arecanut fruit in different modes of force applications was studied. It was observed that power required for dehusking in compression, impact and shear loading were 18.3, 25.2 and 107.2 W, respectively (Aware, 2012). Hence, the mechanism comprising of combined effect of compression, impact and shear in descending order i.e., maximum compression, medium impact and least shear would be suitable for arecanut dehusking. Hence, a mechanism comprising dehusking cylinder and concave with openings which would impart the combined action of compression, impact and shear, was adopted. The diameter and length of cylinder were 300 mm and 500 mm, respectively. There were four beating elements fabricated with MS flat having width and thickness 40 mm and 10 mm, respectively. Rubber pads having width equal to width of MS flat and thickness as 15 mm, were pasted on the MS flat and fixed with flat headed counter sunk bolts with washers. Hence, the diameter of cylinder at rubber beater top surface was 350 mm. Two concaves were developed; one was having aperture size varying from 16 to 23.5 mm, with average as 19.8 mm. The criteria for selecting the range of aperture size of 16-23.5 mm were the spatial dimensions of fruit and kernel. (Aware et al., 2012). The lower opening of 16 mm was based on the

Fig. 1 The relevant anthropometric dimensions for designing pedal operated arecanut dehusker



5th percentile kernel minor diameter (15.4 mm) while the upper opening, 23.5 mm was based on 95th percentile of the same dimension (23.7 mm) and 15th percentile arecanut fruit minor diameter (23.5 mm). The rationale behind this was to have the provision for passing all kernels and less than 15 per cent fruits which could be dehusked using the second concave. The second concave was with aperture size varying from 14.5 to 20 mm with average 17.2 mm. The range of aperture size was selected on the basis that about all kernels (as 5th percentile kernel minor diameter was 15.4 mm) and less than 5% arecanut fruits (as 5th percentile fruit minor diameter was 20.4 mm) could pass through it.

Optimization of Dimensions of Dehusker Based on Anthropometric Data

The dimensions for the pedal operated arecanut dehusker were optimised based on the design guidelines as suggested by Gite et al. (2009) with anthropometric data of male workers of Maharashtra. The relevant anthropometric dimensions considered are shown in **Fig. 1**.

1. Height of hopper: It should be about acromial height. Limiting user was 5th percentile male. The 5th percentile acromial height for male in Maharashtra was 1,257 mm. The height of hopper was kept as 1,275 mm (less than 1.5 excess over acromial).
2. Saddle height: The utmost saddle height should be based on the 95th percentile trochanteric height (vertical distance from standing surface to the trochanterion) and should be taken as 96 percent of it. The 5th and 95th percentile trochanteric heights of agricultural workers of Maharashtra are 792 and 982 mm, respectively. Therefore, the lowest and highest saddle height came to be 760 and 943 mm, respectively. The saddle height was made adjustable varying from 900 to 950 mm.
3. Crank length: The standard crank being used on adult bicycle was of 178 mm. The same crank was used.
4. Diameter of handle: The diameter of handle grip should be such that while holding the handle, the operator's longest finger should not touch the palm. At the same time the grip should not exceed the internal grip diameter. As the equipment was to be operated by male workers, 95th percentile middle finger palm grip diameter was the lowest limit and 5th percentile grip diameter (inside) was considered as the upper limit. As per anthropometric data of Maharashtra, these values were 35 mm and 42 mm, respectively. Hence, commercially available bike grip having outer diameter as 36 mm was used.
5. Length of handle: The minimum handle length should be 95th percentile of hand breadth across thumb of male workers, which was 104 mm. Considering the different positions in which a person holds the handle, the handle grip length should be up to 5th percentile of hand length, which was 161 mm. Commercially available bike grip having length as 120 mm was used.
6. Height of pedal at its lowest position: The height of pedal from the ground surface when the pedal is at its lowest position should be such that the toes should not touch the ground even when the foot is in fully flexed. This clearance should be based on 95th percentile foot length. The rear half of the foot must be on the pedal. Hence, the clearance of pedal from the ground surface should be about 50% of foot length. The 95th percentile foot length of agricultural workers of Maharashtra was 265 mm; therefore the clearance was 130 mm.

Plate 2 Pedal operated arecanut dehusker with feeding hopper

Mechanical Parameters in Development of Dehusker

The diameter up to top surface of rubber beater was 350 mm. The length of cylinder was taken as 1.5 times the diameter. Considering optimum peripheral velocity as 7 m/s, the cylinder speed was calculated as 382 rpm. The speed ratio between pedal shaft and cylinder shaft was 382/60 = 6.36. The speed ratio was achieved in two steps using chain and sprocket drive. The number of teeth on pedal shaft sprocket and that of on the intermediate shaft sprocket were 44 and 18, respectively in the first step. Similarly, in the second step, the numbers of teeth were 44 and 17, respectively. Standard pitch roller chains having pitch equal to 12.7 mm were used to transmit power from pedal shaft to the intermediate shaft and thereafter to the drum shaft.

During operation, the rotary speed of cylinder fluctuated due to variation in load. Hence, the flywheel was incorporated to smoothen the motion of the cylinder. Assuming maximum fluctuation in pedalling rate as 10 percent, the maximum and minimum pedalling rates were 55 and 65 rpm, which resulted the cylinder speed variations from 411 to 348 rpm (maximum and minimum), with average speed 380 rpm. The flywheel was designed adopting the standard procedure. Accordingly, the weight of flywheel was calculated as 12.5 kg. The outer diameter, width and thickness of rim were 350 mm, 40 mm and 20 mm, respectively, resulting rim thickness to width ratio as 0.5.

The developed prototype of pedal

operated arecanut dehusker is shown in **Plate 2**. The details about dehuskling cylinder and concave are given in **Fig. 2** and **Fig. 3**.

Ergonomic evaluation of arecanut dehusker was carried out with ten male subjects that were selected from workers engaged in various agricultural activities. It was ensured that all selected workers were free from any physical abnormalities and were under sound health condition, at the time of experiments. Basic physical characteristics of the subjects namely age, weight and stature were measured in the laboratory. Before the actual ergonomic evaluation of dehusker, the subjects were calibrated for heart rate and oxygen consumption rate relationship. The bicycle ergometer (MONARK 839E) was used as loading device and mobile breath by breath metabolic system (model $K4b^2$) was used for measuring heart rate (HR) and oxygen consumption rate (OCR) of individual subject. The average dry bulb temperature and relative humidity during the test were 26ºC and 84%, respectively. Correlations between heart rate and oxygen consumption rate for each subject were developed. While conducting ergonomical evaluation of dehusker, only HR was recorded and OCR was noted form the calibration chart and energy expenditure rates (EER) were computed as follows.

EER (kJ/min) = OCR (lit/min) × calorific value of oxygen (20.88 kJ/L) (1)

The dehusking capacity, dehusking efficiency and breakage percentage were calculated using standard equations.

Results and Discussion

The pedal operated arecanut dehsuker was developed considering power output form male workers as 75 W (sustained for one hour duration) with pedalling rate of 60 rpm.

Optimization of Dimensions Based on Anthropometric Dimensions

The dimensions of the machine components viz., height of hopper, saddle height, diameter and length of handle grip, height of pedal from ground at its lowest position etc. were optimized as previously explained using the anthropometric data of Maharashtra (Gite et al., 2009). The details are furnished in following **Table 1**.

The developed pedal operated arecanut dehusker was evaluated ergonomically as well as mechanically with ten subjects who participated in calibration experiment.

Fig. 2 Dehusking cylinder (All dimension are in mm)

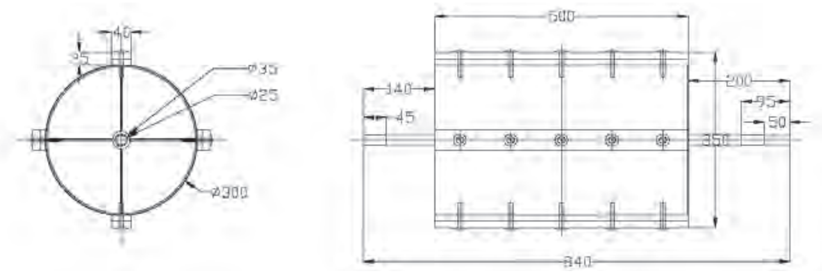


Fig. 3 Concave 1 and 2 (All dimension are in mm)

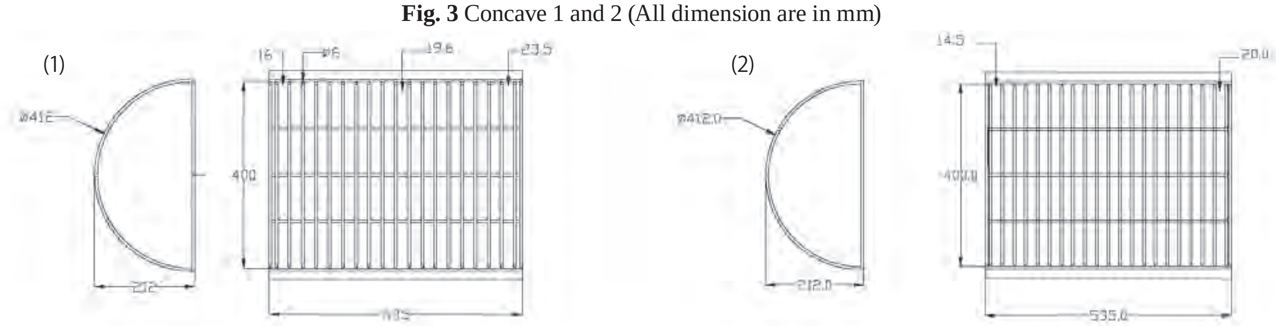
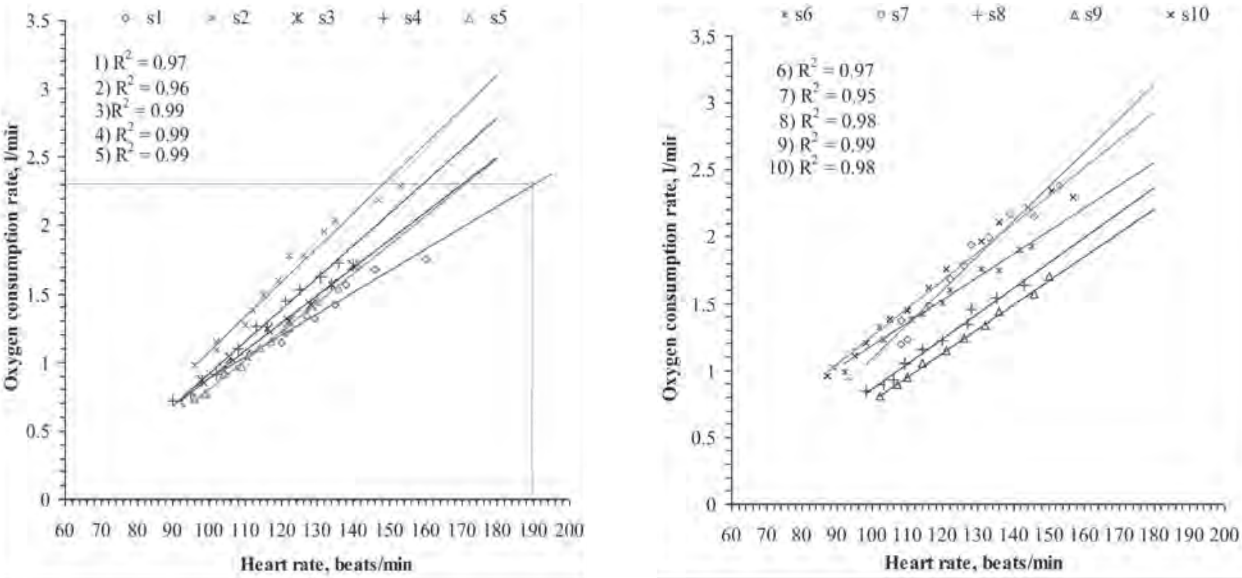


Table 1 The optimised dimensions of arecanut dehusker based on anthropometric data

S. No.	Machine component	Anthropometric dimension considered for male workers	Calculated value, mm	Used value, mm
1	Height of hopper	5^{th} percentile acromial height	1,257	1,275 (keeping practical limitation in mind)
2	Saddle height	5^{th} & 95^{th} percentile trochanteric height	792 and 982	900 to 950
3	Diameter of handle grip	95^{th} percentile middle finger palm grip diameter & 5^{th} percentile grip diameter (inside)	35 and 42	36
4	Length of handle grip	95^{th} hand breadth across thumb & 5^{th} percentile hand length	104 and 161	120
5	Height of pedal at its lowest position	50 percent of 95th percentile foot length	265	130

Fig. 4 Relationship between HR and OCR for subjects (Calibration charts)



Calibration of Subjects

The details about subjects, that are age, weight and stature are given in **Table 1**. The relationship between heart rate and oxygen consumption for the ten subjects is given at **Fig. 4**.

The mean values of age, weight and stature of 10 subjects were 32.8 ± 5.5 years, 59.2 ± 7.3 kg and 165 ± 5.1 cm, respectively.

Ergonomical Evaluation

The heart rate values recorded with the heart rate monitor during the operation were down loaded to the computer and mean HR values were calculated. The corresponding values of oxygen consumption rate (VO_2) of the subjects were taken from calibration chart (**Fig. 4**).

The mean values of heart rate and the calculated values of oxygen consumption for all the subjects are given in **Table 2**. The HR values varied from 116.3 beats/min to 141.6 beats/min. The mean value of HR for ten subjects was 126.5 (± 8.4) beats/min. The OCR varied from 1.14 L/min to 1.54 L/min, with mean of 1.30 (± 0.1) L/min. The EER were calculated using Equation 1. The EER varied in the range of 23.80 to 32.80 kJ/min, with mean of 27.23 (± 2.79) kJ/min. The classification of work load was decided on the basis of Christensen (1953) classification, and accordingly, the operation was graded as “moderate heavy”.

Table 2 Physiological responses of subjects for operation with pedal operated arecanut dehusker

Subject	Heart rate, beats/min	VO_2, l/min	Energy expenditure, kJ/min
I	140.2	1.52	31.74
II	124.2	1.27	26.52
III	141.6	1.54	32.16
IV	121.8	1.23	25.68
V	128.0	1.33	27.77
VI	118.8	1.18	24.64
VII	116.3	1.14	23.80
VIII	128.5	1.34	27.98
IX	121.9	1.23	25.68
X	123.9	1.26	26.31
Mean	126.5	1.30	27.23
SD	8.4	0.10	2.79

Mechanical Performance

The mechanical performance of arecanut dehusker as dehusking capacity, dehusking efficiency and kernel breakage percent are given in the **Table 3**.

The dehusking efficiency varied in the range 77.3 to 78.5% with kernel breakage from 3.7 to 4.9 per cent. The dehusking capacity was in the range of 14.7 to 15.6 kg/h. The mean dehusking efficiency, kernel breakage and dehusking capacity were 77.9%, 4.3% and 15.1 kg/h, respectively.

As the dehusking capacity of the developed arecanut dehusker was 77.9 ≈ 78%, about 22% arecanut fruits were remained unhusked being under size. The second concave with aperture size varying from 14.5 to 20 mm (average aperture size 17.2 mm) as mentioned above was put in use for further dehusking of remaining 22%. The dehusking efficiency and kernel breakage for second concave were 84.5% and 17.6%. The details about efficiency and breakage are given in the **Table 4**.

The overall dehusking efficiency and kernel breakage were as 96.6% and 6.7%, respectively employing two concaves one after another.

Conclusions

A pedal operated arecanut dehusker was developed considering 75 W as the sustained power output from male worker for one hour with 60 rpm pedalling rate. The workers could operate it comfortably with a mean working heart rate of 127 beats/min. The values for dehusking capacity, dehusking efficiency and kernel breakage were 15.1 kg/h, 77.9% and 4.3%, respectively using single concave of 19.8 mm mean aperture size. The dehusking efficiency and breakage values were 96.6% and 6.7%, respectively with two concaves (having mean aperture sizes as 19.8 mm and 17.2 mm) were put one after another and used for dehusking. Therefore, it is recommended to dehusk arecanuts using two concaves of size 19.8 mm and 17.2 mm put one after another to achieve maximum dehusking efficiency.

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■■

Table 3 Mechanical performance of arecanut dehusker

Replications	Dehusking capacity, kg/h	Dehusking efficiency, %	Kernel breakage, %
1	15.6	78.0	3.9
2	15.0	78.2	3.7
3	14.7	78.0	4.1
4	15.1	78.0	4.8
5	15.0	78.2	4.9
6	14.8	77.5	3.8
7	14.9	77.7	4.7
8	15.3	77.3	4.6
9	15.6	78.5	4.5
10	15.0	77.7	4.0
Mean	15.1	77.9	4.3

Table 4 Dehusking efficiency and kernel breakage

Concave	Dehusking efficiency, %	Kernel breakage, %
Concave 1	77.9	4.3
Concave 2	84.5	17.6
Using both concaves one after another	96.6	6.7

Determination of Essential Indexes in Assessing the Status of Agricultural Mechanization in Kurdistan, Iran

by
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Abstract

In order to develop a proper plan for agricultural management in a region, adequate understanding of the current situation in the field of agricultural mechanization is a necessity. In this study, a comprehensive assessment was conducted on developmental situation of agricultural mechanization in Kurdistan, Iran. For this purpose, some factors such as the degree of mechanization, level of mechanization, coefficient of productivity and operational power were calculated. The degree of mechanization in major crops was generally calculated for tillage, planting, growing and harvesting throughout the province, separately. The ratio of this index was highest for tillage operations (nearly 100%) and lowest for the growing operations (less than 50%). The average index of the tractor mechanization level was estimated to be 1.67 (hp/ha) which is higher than the national average (1.5 hp/ha). The coefficient of productivity and machinery usage in the region was estimated to be 71%.

Introduction

Tools, implements, and powered machinery are essential and major inputs to agriculture. The term "mechanization" is generally considered as the overall application of these inputs (Clarke, 2000). Mechanization, defined as the use of machines and mechanical tools in agriculture for overcoming technical and climate limitations as well as time constraints, has facilitated the possibility to increase the area under cultivation and production in the agricultural sector. Adequate understanding of current situation of mechanization is necessary for proper planning in regional agriculture, such that a suitable solution can be offered for the development and institutionalization of mechanization. Basic indices should be used for any type of comparison to provide an accurate analysis. Several indices such as degree of mechanization, mechanization level and coefficient of productivity are used to investigate the agricultural mechanization status in any region.

Singh and De (1999) reviewed the methodologies adopted by several authors to express a mechanization indicator. Mechanization planning requires a quantitative assessment of mechanization indexes (Singh, 2006). Rasouli Sharabiani and Ranjbar (2008) calculated agricultural mechanization indices in Sarab, Iran. They estimated regional mechanization level to be about 0.83 (hp/ha). Literatures have showed that the degree of mechanization for most agricultural operations has been very low. It has been also estimated that 775 tractors are required to compensate the deficit in the region to achieve the mechanization level of 1.5 hp/ha. Olyaoe and Rotimi (2010) calculated the mechanization index in two states in southwestern Nigeria and analyzed the efficiency in both regions which were 31.3% and 28.6, while total efficiency was in the range of 0.0115 to 0.0951 (ha/kW.h). Yield of corn was in the range of 1.2 to 1.7 (tons/ha).The mentioned conditions showed an undesirable situation in terms of agricultural mechanization. In a study, conducted by Fortune and Tawanda (2013) on indices of agricultural mechanization related to the production of tobacco in Zimbabwe, the mechanization level was equal to 0.42 (hp/ha) that was too far from the desirable value (1.5 to 2 hp/ha). Pishbin (2013) calculated the mechanization indices for crops and horticulture products in Fars, Iran. The results of their study showed that mechanization capacity, regardless of the product was 1,711.3 (hp.h/ha) and considering the 50% of orchards, it was 1,410.7 (hp.h/ha). The mechanization level index was calculated to be 1.89 (hp/ha), indicating that the mechanization level was higher than the national average. The role of mechanization in the production efficiency of potato has been studied by Fouladi et al. (2013) in Ardabil, Iran. Their results showed that 98% of plowing, 35% of planting and 57% of growing operations were mechanized which can be improved by compensating the regional power deficits.

Agricultural mechanization status in Bangladesh has been studied by Ziaudin and Zia (2014). According to their results, the use of mechanical farm power has had a rapid increase over the last 2 decades. Bello et al. (2015) investigated the index of mechanization and other productivity functions as indicators of assessing the impact of mechanization on agricultural production in North of Nigeria.

So far, a comprehensive assessment of the status of agricultural mechanization has not been carried out in the province of Kurdistan, Iran. The results of this study can provide an integrated insight into the identification of problems related to agricultural mechanization in terms of accurate planning and improving of the present situation.

Materials and Methods

Climate and Geography of Kurdistan

[Kurdistan is a province located in the West of Iran, between 34º, 44′ to 36º and 30′ of the north latitude and 45º degrees 36 and 31 minutes to 48 degrees 16 minutes of the east longitude of the Greenwich meridian.] Kurdistan is a mountainous region with high plains and wide valleys spread across the region. According to the latest division of the country in 2011, it has 10 cities. Arable lands of the province are 1100 thousand hectares of which about 700 thousand hectares are allocated to crops and 400 thousand hectares are allocated to horticulture products. The main crops of the province are wheat, barley, corn, beans, forage plants, potatoes as well as horticulture crops including grapes, strawberries, walnuts, etc. specific to cold regions. Annual production of crops is about 1.75 million tons and horticulture crops is about 0.25 million tons.

Statistical and Research Methodology

Statistical data and information on the operational status of land and machine, the main products and farming calendar of the region were collected through direct observation, discussion with farmers and the ministry of agriculture statistics. The aim of this study was a comprehensive review of the status of agricultural mechanization development in Kurdistan province using indices such as the degree of mechanization, mechanization level, operational power and coefficient of productivity.

Degree of Mechanization

The degree of mechanization is an index that considers the quantity of issues associated with mechanization and it is defined as the area under mechanized operations divided by the total area under cultivation. The value of this index for main products of Kurdistan province, which has a higher cultivation area, is generally calculated separately for tillage, planting, growing and harvesting operations all over the province.

Mechanization Level

Mechanization level index investigates the quality of mechanization. This index is the ratio of total available tractor power to the total area under cultivation (hp/ha) (Lak and Almassi, 2011). To calculate this index, the number of active tractors used in the agricultural sector is estimated. The actual power was calculated by multiplying the total nominal power by 0.75. The cultivation area is related to all products, while fallow area is not calculated.

Operational Power

Operational power is an index to determine the area that can be covered by machine services during the time which is calculated in terms of area. To calculate this index, the table of cultivation area of a cropping year is provided and machine operation calendar is then checked for months of the year in major crops. According to this calendar, machine operation peak is considered. Usually land preparation and planting operations need more tractors than other operations, i.e. if the farm manager provides the machines required for the preparation and cultivation of wheat and barley there won't be any machine problem, in other months. This index is calculated using the following equation:

$$\text{Operational power} = (n \times t_a) / t_r \quad (1)$$

Where, n is the number of tractors, t_a available working time, and t_r required time for cultivation per hectare.

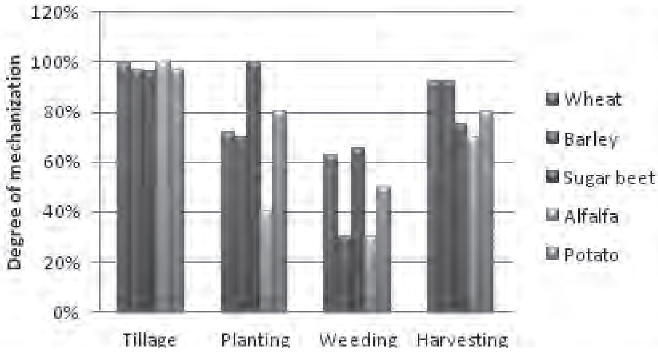
Coefficient of Productivity

The coefficient of productivity or area coefficient of productivity is the ratio of area under cultivation to operational power of machine in a region. It represents the efficiency of operational power according to the power required.

Results and Discussion

The degree of mechanization index for tillage, planting, growing and harvesting operations was calculated separately for the province. This index was highest for tillage operations and lowest for growing operations. **Fig. 1** demonstrates that in the mechanization of tillage operations, the province is in a reasonably good condition. In the section of planting operation, the degree of mechanization was lower than that of tillage operation; such that even the available resources to supply

Fig. 1 Degree of mechanization for important crops in Kurdistan province



the requirements of planting operations were not properly used and in some areas the seeds are manually sprinkled. The number of seeder machines is low; these are not properly managed and are not available to all farmers. The degree of mechanization in growing operations was more related to spraying and other operations such as thinning and weeding were not usually fully mechanized. The degree of mechanization in the harvest of wheat and barley was almost 100% and this number could be considered real and reasonable by eliminating the exceptions of manual harvesting or obsolete traditional methods that is because in this part, the combines with proper power corresponding to harvest requirements (as observed and declaration of regional experts) are used and in the border areas, migrant combines are also used during peak harvest times. Of course, this state was only for the harvest of wheat and barley, and there was a lack of mechanized harvesting for other crops such as alfalfa.

The average of mechanization level index of tractor in the province was estimated to be 1.67 (hp ha^{-1}). The amount of this index in the cities of the Kurdistan province is presented in **Table 1**. Now, the average agricultural mechanization level in Iran is equal to 1.5 (hp/ha). Since this index in Kurdistan is higher, this can prove that the strategic planning for agricultural mechanization has been successful. On the other hand, despite the higher mechanization level index in Kurdistan compared to the national average, the degree of mechanization for some operations such as planting and growing are still low due to a large number of old tractors used in the region, the use of tractors in works rather than farming operations as well as the lack of management in the proper use of agricultural machines. The results also showed that city of Marivan with 5.44 hp/ha and the city of Kamyaran with 1.24 hp/ha have the highest and the lowest levels of mechanization index in the province, respectively.

Current operational power of the province was determined according to collected crop information. The number of effective working days in this region is calculated based on the 60 days and the number of rainy days which is equal to 10 days during the 60 days. The number of rainy days will be doubled and subtracted from the 60 days, working out to be 40 days as effective working days if 8 hours per day is considered as working time during a day. On the other hand, the time required for preparing the ground for one hectare of wheat and barley with conventional operations was calculated to be 7 hours. According to the existence of 23,623 tractors in the entire province and 320 proper hours in the field operations in the region, the operational power was calculated by substituting these numbers in the related equation that would be equal to 1,079,908.57 per hectare.

The coefficient of productivity for tractors according to the value obtained for operational power of the cultivation area was estimated to be 71%; which means that at this time, about one third of the operational power of tractors has been wasted. It is also found that at the peak of the working season which requires more machines, there are problems with providing machines for operations. Therefore, the operations cannot be performed on time. On the other hand, the power of tractors in the region is not fully utilized due to a lack of proper management in terms of distribution and application. Even though, the number of worn-out tractors in the region is high, resulting in relatively high level of mechanization, the incorrect use of machines and their improper maintenance can cause a low ef-

Table 1 Tractor power, area under cultivation and mechanization level in the cities of Kurdistan

City	Total tractor power (hp)	Area under cultivation (ha)	Mechanization level (hp/ha)
Dehgolan	190,938	106,808	1.34
Sarvabad	17,960	7,372	2.11
Kamyaran	90,900	54,673	1.24
Divandarreh	376,650	110,353	2.56
Marivan	77,523	10,685	5.44
Qorveh	237,683	103,052	1.73
Sanandaj	102,903	41,994	1.83
Saqqez	272,046	117,603	1.73
Bijar	321,150	178,525	1.35
Baneh	44,553	12,434	2.68
Total	1,732,306	775,717	1.67

ficiency and quality of operational operations in the farms.

Conclusions and Suggestions

In this study, a comprehensive assessment was conducted on developmental situation of agricultural mechanization in Kurdistan, Iran, and some factors were calculated such as the degree of mechanization, mechanization levels, coefficient of productivity and operational power. The indexes of mechanization degree in dominant products were generally calculated for tillage, planting, growing and harvesting operations throughout the province, separately. This index for tillage operations was the highest (nearly 100%) and for growing operations was the least (less than 50%). The average index of mechanization level for tractor in the province was estimated to be 1.67 (hp/ ha) which was higher than the national average (1.5 hp/ha) and the average calculated value for some parts of the country (Pishbin, 2013). Operational power in this region was calculated in 1,079,908.57 hectares. According to the calculated operational power and the total cultivation area in the province, the coefficient of productivity and the use of tractors and machines in the region were estimated to be 71%. The most important actions that can be effective in improving mechanization are as follows:

1. Promotion of mechanized operations and training farmers and machine owners;
2. Providing tractors and farm machines required for the region;
3. Allocation of loans to replace old tractors and machines as well as equipment repair shops;
4. Education of users and repairmen to fix, adjust, operate and maintain the machines;
5. Control and regulation of manufacturers and dealers of agricultural machines for proper provision of after-sale-services.

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■■

Evaluation Parameters Effeccting the Performance of Vibrating Vertical Tillage Equipment, First Stage

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Abstract

The aim of this work was to evaluate an alternative to reduce energy consumption applied to primary soil tillage. To do this an experimental apparatus was developed to evaluate the operation of vibration induced tillage parameters. The components that integrated the system for field valuation were: frame tool carrier that includes a system of three-point hitch and depth control mechanism; three sensors were included to measure tillage force, torque and the frequency of the subsoiler oscillation. Oscillatory impact force was applied through a rod and crank mechanism through a subsoiler tine whose movement is provided by a hydraulic motor. The apparatus evaluation was complemented by a system of data acquisition and signal conditioner that allows the registration of the variables of interests such as draft power, penetration force, applied torque and speed of oscillation of the system. The results of the calibration of the sensors showed a system correlation higher than 95%. The results obtained during field system verification at different speed, amplitude and depth of tillage showed a reduction of the draft force up to 50% using the oscillated induced tillage compared to the non-vibrating tine condition. The study shows a significant increase in the magnitude of the draft force and torque applied by 33% when the working depth increased from 0.30 to 0.40 m and an increase of 21% of draft force was produced when the amplitude of oscillation was reduced from 0.070 to 0.060 m. No significant difference was found when the tractor speed was increased from 1.5 to 2.5 km h^{-1}. For future work the developed apparatus will allow to determine how the amplitude and the working depth and oscillation frequency of the tine could affect the draft force in tillage work.

Keywords: oscillatory subsoiler, draft force, energy consumption, amplitude of oscillations.

Introduction

Vertical conventional tillage implements have soil-working tools, which do not move relative to the implement frame. Their draft requirements are comparatively high as they are always in contact with the soil and the power for operating these implements is transmitted through traction with tractive efficiency varying between 40 to 75% (Witney, 1995). To improve the tractive efficiency, the tractor drive wheels are ballasted with additional weights. However, this practice causes soil compaction, which is detrimental crop efficiency. Several researchers have reported that oscillating soil-working tools have lower draft requirements and break up the soil better than do their non-oscillating tine implements when working under identical conditions. Therefore, an oscillating soil-working tool may reduce the number of operations to prepare an acceptable seedbed and minimize soil compaction, thus providing a better physical environment for plant growth. In addition, oscillatory tillage uses more efficient tractor power-take-off by 90 to 95% to mechanically oscillate tines (Hendrick, 1980). Lower draft requirement of oscillatory tillage reduces the reliance on less efficient drawbar power; leading to a lower overall demand on engine power may occur (Slattery and Desbiolles, 2003), also is generally recognized that the draft force of a vibratory tillage implement can be reduced by 30% to 80%.

Bandalan et al. (1999) performed an experimental study on the tillage performance of an oscillatory subsoiler, determining the optimum combination of operating parameters of the subsoiler such as frequency of oscillation, amplitude and working speed the pull-force and the total power consumed were measured.

Sahay et al. (2009) developed an equipment induced vibration tillage that change the transmission frequency of vibration from 9 to 13 Hz and from 15 to 35 Hz. Experiments showed that the real depth of the oscillatory system work was 0.153 m while the same equipment without vibration was 0.074 m.

Shahgoli et al. (2009) in their study reported the evaluation of the effect of the angle of swing of the tine in the performance of the subsoiler. The objective of this study was to quantify the optimal angle of oscillation for the reduction of the strength of pull, low requirement of power, low specific draft-force, maximum disturbed area and minimum vibration transmitted to the operator's seat. In terms of performance, -22.5° was the optimum angle of oscillation for the reduction of the draft force and power.

Shahgoli et al. (2010a, 2010b) claimed in his study on the optimization of the oscillatory frequency of vibratory tillage, the tines were oscillated with an amplitude of ± 69 mm at an angle of 27° using a working speed of 3 km h^{-1}. The frequency of oscillation was changed from 1.9 to 8.8 Hz. There was an optimal frequency near 3.3 Hz (1.5 speed ratio), which minimized the total engine power required to operate a subsoiler. It was estimated a decrement of engine power demand above 26% compared to a rigid tillage equipment.

Studies of oscillatory tillage cited by (Shahgoli et al., 2009 and 2010a) reported that the induced oscillation of a longitudinal or vertical tillage in directions tool can significantly reduce by 50% the draft-force requirements. They also mentioned that the oscillatory frequency, amplitude of swing, swing angle, speed of the tractor, tool design and soil properties those are important factors affecting the functioning of a vibrating tool.

Campos et al. (2015a) modified a vertical tillage implement call multicultivador and conducting a series of investigations by applying the theory of critical depth in tillage equipment. The modifications that were made by the researchers team, basically consist of place shallow tines on the front; and greater length at the rear tines. In the tines of the rear, wings of different sizes were evaluated to increase the disturbed cross sectional area. Field tests were conducted with this arrangement and experimentally was demonstrated that there is a decrease in the specific soil resistance by more than 20% in comparison with primary tillage using tines at the same working depth.

The purpose of the present research work was to develop an apparatus that allows determining the effect that has the variation of the most important parameters in vibration induced tillage such as amplitude and frequency of oscillation, as well as the effect of this vibration on the reduction of the draft force on the vertical tillage.

Materials and Methods

The experimental apparatus was design to be mounted to a tractor category II with three points hitch system, and the oscillatory system was powered by the hydraulic system of the tractor. This device allow adjustment of the frequency of oscillation from 3.3 to 4.9 Hz and amplitudes from 0.060 and 0.070 m. **Figs. 1** and **2** show the main components of the device designed and built by the Tecnomec Agricola S. A. de C.V. Company. The apparatus is integrated by an oscillation tine (1), an oscillatory mechanism connecting rod crank (2), an extended octagonal transducer (3), structure or chassis (4), a hydraulic motor (5), a torque transducer (6) and an oscillation frequency sensor (7).

The chassis corresponds to the structure of the apparatus in which all the components including the three-point hitch system are mounted. A crank connecting rod mechanism provides the swing of the tine as shown in **Figs. 1** and **2**. This mechanism is operated by means of a hydraulic motor for high torque and low speed of rotation. The frequency of rotation is controlled by a flow regulator valve. A torque

Fig. 1 Side view of the vibrating tillage equipment. 1. Oscillating tine, 2. Oscillatory mechanism, 3. Extended octagonal ring transducer.

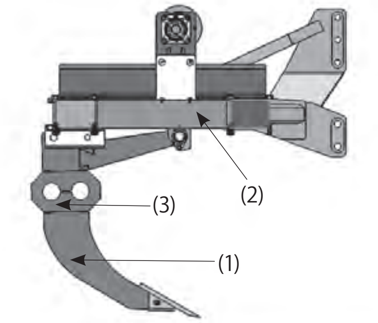
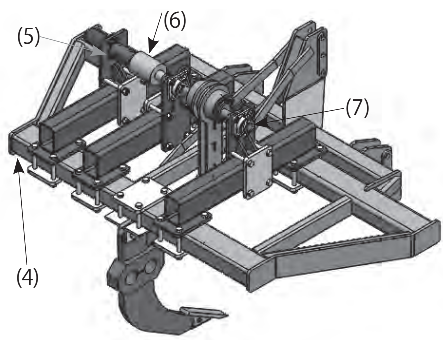


Fig. 2 Isometric view of the vibrating tillage equipment. 4. Chassis, 5. Hydraulic motor, 6. Torque transducer, 7. Frequency sensor.



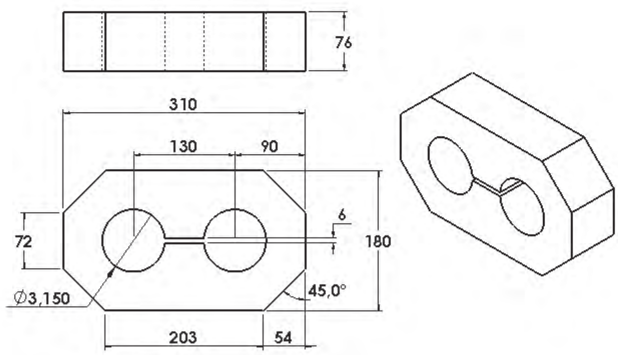


Fig. 3 Dimensions and characteristics in the extended octagonal ring transducer in mm

transducer with a capacity of 25 kW at a frequency of 9 Hz was coupled between the hydraulic motor and oscillating mechanism to measure the torque that is generated at the time of soil tillage.

To measure horizontal and vertical forces applied to the soil an extended octagonal ring transducer (OAE) with capacity of 40 k N was built which dimensions are shown in **Fig. 3**.

An electronic card was developed to determine the frequency of oscillation in real time of the vibrating apparatus, using an infrared optoelectronic sensor H21A1 (Fairchild Semiconductors, USA) and a microcontroller PIC 16f84 (Microchip company, USA) to eliminate rebounds produced by the effect of magnetic noise.

The construction of the apparatus and mounting of sensors were built at Tecnomec Agricola S. A. de C. V., Aguascalientes State, Mexico, during the period from February to December 2014. Laboratory research and field test was carried out from January to October 2015 in the Agricultural Engineering Department and the experimental Station "El Bajío" at the Universidad Autónoma Agraria Antonio Narro. Located at the State of Coahuila, Mexico, at 25° 21.52" N, 101° 50" W and at altitude of 1,740.5 m over the sea level in a clay soil with resistance to penetration of 2.45 k Pa, and average soil moisture of 18%.

A John Deere tractor JD6403, single traction with 105 HP at the engine, was used for the field test. Instrumented tractor include the following equipment: analog to digital converter Logbook360 (Iotech Company, USA), calibrated at a frequency of 20 Hz sampling rate and signal conditioner model DBK43A from the same made, calibrated to a gain of 2500 micro strain (μδ).

In order to measure forces in the horizontal and vertical direction (**Figs. 4** and **5**), it was necessary to make a static calibration of the transducer (OAE) using known loads of (470.8, 470.8, 716.13, 343.35, 343.35, 294.30, and 294.30 N). To run the calibration; a loading and unloading of the weights were carried out and each operation was carried out using 5 replicates.

In order to measure the torque applied to the tine from the hydraulic system of the tractor was necessary a torque transducer calibration, the same Logbook360 data acquisition system was used. Calibration was performed with two-arm lever (0.65 and 0.85 m) and four loading and unloading weights: 294, 294, 343 and 343.5 N, each position was replicated 5 times (**Fig. 6**).

Data analysis was performed with the method of spectral analysis described by Campos and Wills, (1995) using Matlab V2010a and Fast Fourier Transform algorithm to obtain the mean values, curves and calibration constants, using a linear regression of the data analysis. Analysis of variance was performed using Minitab V15 statistical analysis software. The field Treatments performed for two depths of 0.30 and 0.40 m, two working speed at


Fig. 4 Calibration of transducer OAE for horizontal force


Fig. 5 Calibration of the transducer OAE to determine vertical force constant

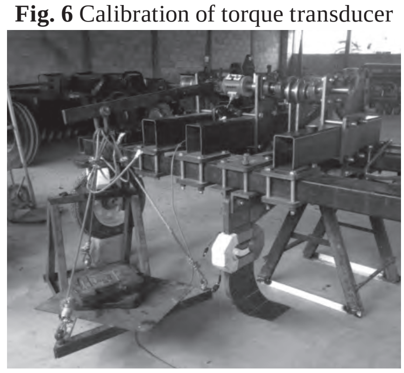
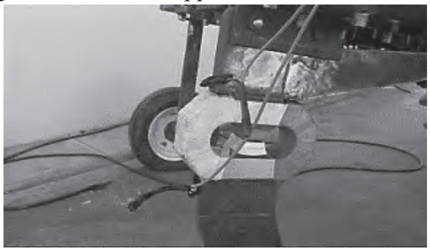


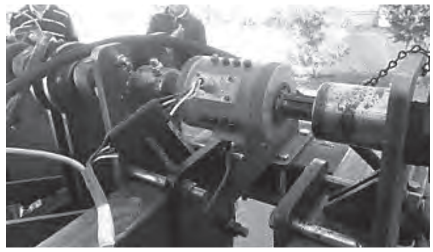
Fig. 6 Calibration of torque transducer

Fig. 7 Instrumented apparatus for field evaluation

(a) Apparatus for vibratory tillage



(b) Transducer OAE



(c) Torque transducer

(d) Sensor optoelectronic

(e) Data acquisition system

1.5 and 2.5 km h^{-1} and two amplitudes of 0.060 and 0.070 m for a single oscillatory speed of 3 Hz. A control test with not vibrating tine at two working depths of 0.30 and 0.40 m at a speed of 1.5 km h^{-1} was also performed. A total of 10 treatments were conducted and four replications were carried out by each arrangement; each trial was carried out on plots of 50 m long by 2.5 m wide.

Results and Discussion

Construction of the Apparatus

Fig. 7 shows instrumented apparatus (a) for vibrating tillage, a transducer OAE (b) that (c) was employed to measure the torque required for impact the soil, an infrared optoelectronic sensor (d) to measure the frequency of the oscillating mechanism and a data acquisition system (e) for collecting all data information from the transducers in real-time to be subsequently analysed.

To obtain the same reliability of measurement in the two transducers, it was necessary to standardize the coefficient of constant by means of the calibration adjustment on 3 channels of the amplifier that correspond to each sensor and adjust the input gain, the compensator and the excitation voltage. **Fig. 8** shows an example of a step graph of OAE corresponding to the vertical force transducer calibration. **Fig. 9** shows the equation of regression with a constant of 75.00 mV N^{-1}, with a correlation coefficient of 99.8%.

Fig. 10 shows an example Graph corresponding to the horizontal force of OAE transducer calibration and **Fig. 11** the linear calibration of horizontal force. In addition, shows the equation of regression with a constant of 47.74 mV N^{-1} with a correlation coefficient of 99.7%.

Fig. 12 shows an example of the graph of the deformation in mV generated from the ascent and descent of four loads with a lever arm of 0.85 m and **Fig. 13** shows the equation of regression with constant 0.4538 mV N m^{-1} with a correlation coefficient of the 98.8%. Similar results were obtained by (Campos et al. 2015b) for the calibration of sensors of the type of OAE.

Tests Under Field Conditions

Fig. 14 shows the forces obtained in field for the vertical force by com-

Fig. 8 Cycles of loading and unloading of the OAE transducer during calibration with different weights for the vertical force

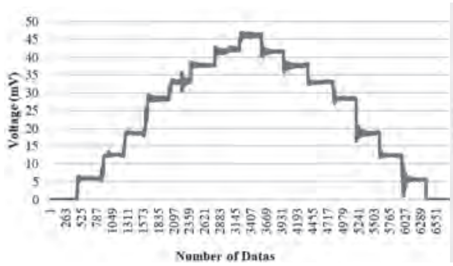


Fig. 9 Graph showing the linearity of calibration of OAE transducer for vertical force. The obtained equation was: Force (N) = 8.073 + 75.00* Voltage (mV)

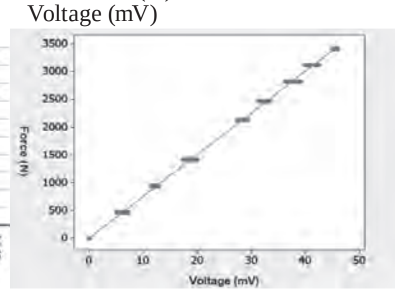


Fig. 10 Cycles of loading and unloading of the OAE transducer during calibration with different weights for the horizontal force

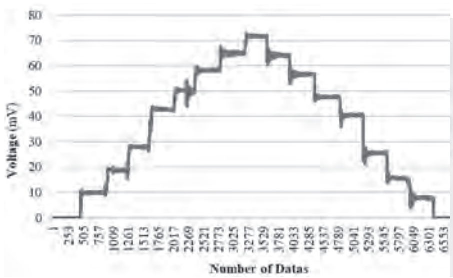
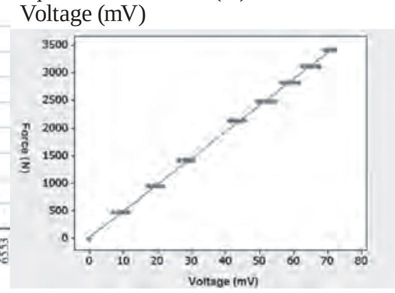


Fig. 11 Shows the Linearity of calibration of horizontal force. The obtained equation was: Force (N) = 21.97 + 47.74 Voltage (mV)



paring the amplitudes of 0.060 and 0.070 m while working at a depth of 0.40 m and a speed of 1.5 km h^{-1}. It can be seen that the applied forces for higher oscillation amplitude (7040A1) are charged towards the positive part of the graph, while in the lower oscillation amplitude forces (6040A1) are distributed both in the positive as negative zone.

Fig. 15 shows an example of the behaviour of horizontal force generated by the rigid tine subsoiler, where it can be seen the difference between two depths of tillage (0.30 and 0.40 m) at a speed of work of 1.5 km h^{-1}.

Fig. 16 compares the draft-force obtained in field with the rigid and vibration tine for a forward speed (1.5 km h^{-1}), at the same depth of 0.40 m, which shows a lower draft-force demand when applying an induced vibrations to 0.070 m amplitude.

From **Fig. 17** can be appreciated the magnitude of the torque obtained in field. Where shows that exist two directions of registered torque magnitude applied to the soil one for the forward working condition into compacted soil and the another magnitudes for the reverse when the tine is acting in loose soil. The results of this analysis are shown in **Table 1**.

In **Table 1**, the variance analysis of the effect of depth of tillage of 0.30 to 0.40 m shows a highly significant difference on the magnitude of the draft force, as well as the force applied for tillage with oscillation chisel to three different magnitudes of swing (0.01, 0.06 to 0.07 m)

For the oscillatory tine, the mean values showed a decrease of 50.1% in the force of pulling, compared with the rigid tine. Similar results were reported by Shahgoli et al. (2010b). The significant reduction in the magnitude of the draft force by change of depth of tillage, from 0.40 to 0.30 m, was also reported by (Spoor and Godwin, 1978; Campos et al., 2015a).

An analysis of variance was performed as shown in **Table 2**, whereas only the magnitudes of the draft force to the two assessed oscil-

Table 1 Analysis of variance of the effect depth (0.30 and 0.40 m) and swing (0.00, 0.06 to 0.07 m) on the magnitude of the pulling force

FV	GL	SC	CM	FC	F_{05}	F_{01}
Blocks	3	4.313	1.438	0.725		
Oscillation factor (O)	2	167.267	84.633	42.670**	5.14	10.92
Oscillation inaccuracy	6	11.901	1.983			
Depth factor (P)	1	2.8380	2.8380	221.21**	5.12	10.56
O × P	2	25.065	12.532	9.768**	4.26	8.02
Depth inaccuracy	9	11.546	1.283			
Total	23	505.892				

(FV) Factors. (GL) degree of freedom. (CM) Mean square. (FC) F factor. (F_{05}) F for ($P < 0.05$). (F_{01}) F for ($P < 0.01$) (**) exist a significant difference between variable levels

Fig. 12 An example graph of cycles of loading and unloading during calibration the torque transducer with different loads and arm lever

Fig. 13 Graph showing the linearity and constant of calibration for torque transducer. Torque (Nm) = 29.34 + 0.4538 Voltage (mV)

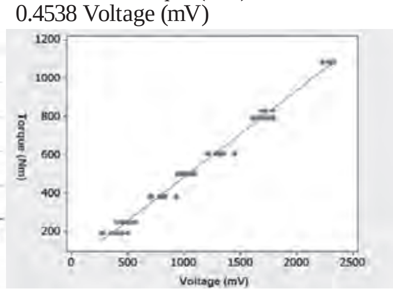


Fig. 14 Comparison of vertical forces to different amplitudes for a depth of 0.40 m. The graph (7040A1) is for an amplitude of 70 mm and first replicate. The graph (6040A1) is for an amplitude of 760 mm and first replicate

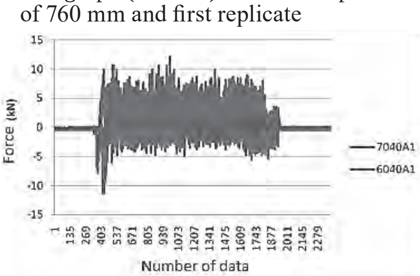


Fig. 15 Comparison of the horizontal force at rigid tine at two depths. Graph (40A1) shows the tine working at 0.40 m depth for the first replicate. Graph (30A1) shows the tine working at 0.30 m depth for the first replicate

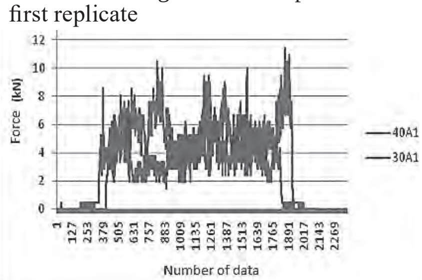


Fig. 16 Comparative force demand at rigid tine and vibration (amplitude of 0.070 m) to the same depth of 0.40 m and speed of 1.5 km h^{-1}. Graph (7040A1) shows the draft force for vibrating tine working at 0.40 m depth. Graph (40A1) shows the draft force for rigid tine working at 0.40 m depth

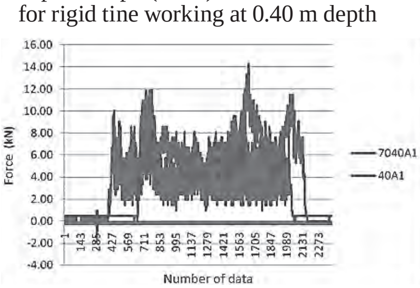
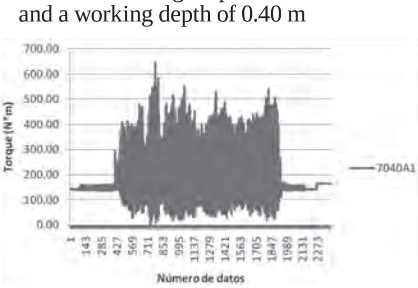


Fig. 17 Graph showing the applied torque for the oscillating amplitude of 0.070 m and a working depth of 0.40 m



lation amplitudes of 0.060 and 0.070 m finding significant difference. Variance analysis shows the effect is also at two-tillage speed, 1.5 and 2.5 km h^{-1} on the draft force, finding not significant difference in force for the effect of change of speed. In this analysis was corroborated the significant effect that produces the depth of tillage on draft-force (Campos et al., 2015c).

In comparison to the change of amplitude of oscillation, **Table 3** shows a significant difference in the measurement of the draft force in the order of 21.0% when increases the oscillation amplitude from 0.060 to 0.070 m.Similar results were obtained by (Shahgoli et al, 2010a).

From **Table 4**, the oscillation of the tine increases the magnitude of the vertical force to a 47.0% between the vibrating and the rigid tine when tillage speed increases from 1.5 to 2.5 km h^{-1}. Likewise, the magnitude of the torque is increased up to 33.0% when tillage depth is increased from 0.30 m to 0.40 m unless this increase has to do with the increased speed of 1.5to 2.5 km h^{-1}. Similar magnitudes were reported by (Xin et al., 2013).

Conclusions

The device designed and built by TecnomecAgricola S. A. de C. V. allowed mounting of sensors for the on-field measurement of horizontal and vertical forces acting on the tine tool as well as frequency of oscillation and torque of impact of the tine in a satisfactory manner.

In the field-testing for verification of the system functions, data recorded at different amplitudes of swing revealed that the apparatus could reduce the draft force up to 50% using the oscillated induced tillage compared to the non-vibrating condition. There was a significant increase in the magnitude of the draft force and torque applied by 33% when the working depth increased from 0.30 to 0.40 m. Besides, an increase of 21% draft force was produced when the amplitude of oscillation was reduced from 0.070 to 0.060 m.

No significant difference was found when the tractor speed was increased from 1.5 to 2.5 km h^{-1}. Then, based on the above results, it was confirmed that there is a significant reduction in the draft-force between a rigid tines versus one vibrating at lower oscillation amplitude.

Table 2 The variance analysis of the effect of working speed and depth, oscillating frequency and amplitude over the magnitude of the draft force

FV	GL	SC	CM	FC	F_{05}	F_{01}
Blocks	3	1.598	0.533	1.251		
Speed factor (V)	1	0.001	0.001	0.002	10.13	
Speed error	3	1.51	0.350			
Oscillation factor (O)	1	14.005	14.005	44.691**	5.99	13.75
V × O	1	1.129	1.129	3.602	5.99	
Oscillation error	6	1.88	0.313			
Depth factor (P)	1	118.157	118.157	186.240**	4.26	7.82
V × P	1	24.939	1.129	39.310**		
O × P	1	2.571	24.939	4.052		
V × O × P	1	3.194	2.571	5.035*		
Total error	24	7.613	3.194			
Total	31	176.138	0.634			

(FV) Factors. (GL) degree of freedom. (CM) Mean square. (FC) F factor. (F_{05}) F for ($P < 0.05$). (F_{01}) F for ($P < 0.01$) (**) exist a significant difference between variable levels

Table 3 Comparison of means values of the magnitude of the draft forces at different frequencies of oscillation and depths of the tine work

Oscillation amplitude (mm)	Average force (kN)	Depth (m)	Average force (kN)
060	7.929a	030	5.153b
070	6.236b	0.40	8.96a

Means in the same column followed by the same letter do not differ significantly ($P < 0.05$) by DUNCAN test.

Table 4 Comparisons of means values of the vertical forces and torque for the evaluated treatments

Treatment	Average force (kN)	Applied torque (Nm)
60 mm swing tine + 0.40 m Depth + 2.5 km h^{-1}	8.93a	304.3a
70 mm swing tine + 0.40 m Depth + 2.5 km h^{-1}	7.85a	303.1a
Rigid tine to 0.40 m Depth + 1.5 km h^{-1}	4.13bc	---
70 mm swing tine + 0.40 m Depth + 1.5 km h^{-1}	4.04bc	318.1a
60 mm swing tine + 0.40 m Depth + 1.5 km h^{-1}	3.96bc	310.2a
60 mm swing tine + 0.30 m Depth + 2.5 km h^{-1}	3.28cd	186.5b
60 mm swing tine + 0.30 m Depth + 1.5 km h^{-1}	3.62cd	203.8b
70 mm swing tine + 0.30 m Depth + 2.5 km h^{-1}	2.39de	169.7bc
70 mm swing tine + 0.30 m Depth + 1.5 km h^{-1}	2.14e	108.78c
Rigid tine to 0.30 m Depth + 1.5 km h^{-1}	1.84e	---

Means in the same column followed by the same letter do not differ significantly ($P < 0.05$) by DUNCAN test.

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■■

Retirement Message

Sincere congratulation to the Farm Machinery Industrial Research Corporation in cooperation with the Shin-Norinsha Co., Ltd and the International Farm Mechanization Research Service, Tokyo on the 50th anniversary of publishing the AMA (Agricultural Mechanization in Asia, Africa and Latin America). It has seen great transformations in its 50-year history in agricultural mechanization thus wishing the AMA a Very Happy Golden Anniversary and anticipates its continuation for many more years to come!

The AMA has nearly one hundred Contributing Editors and Co-operators worldwide. I have been associated with AMA publication for the last seven years as Consulting Editor reading/correcting every submission. I have witnessed progress in every sector food production including crops, vegetables, fruits and fibre production in developing and developed nations. These papers addressed several aspects of mechanization of agriculture such as use of robots and artificial intelligence and even going to organic farming to meet the food and fibre demands of growing world population. It has been an excellent experience to witness this kind of progress in agricultural mechanization and food production in various areas of the world.

My sincere thanks are extended to Yoshisuke Kishida for providing this opportunity to be part of AMA Journal. I also thank various members of staff related to AMA publication who had been corresponding regularly and keeping track of various papers.

Wishing everyone all the best!

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CANADA

Development of a Watermelon *(Citrullus lanatus)* Seed Extractor

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Abstract

Engineering properties such as physical and textural properties of the matured watermelon (Citrullus lanatus) fruit and its seeds were measured in order to design and develop the watermelon seed extractor. The developed seed extractor consisted of following systems for (i) watermelon cutting, (ii) watermelon seed extraction, (iii) seed separation and (iv) power transmission. The extractor was evaluated with three types of scrapers namely stainless steel, wooden and nylon; two cutting planes that of transverse and longitudinal planes and three rotational speeds of 50, 100 and 150 rpm. The best combination was observed to be for transverse cutting with nylon scraper at a speed of 100 rpm. The best performance for capacity, extraction efficiency, seed loss, seed damage, germination percentage and vigour index were found to be 1.98 kg of seeds/h, 99.49%, 0.19%, 0.26%, 97.04% and 3,284, respectively.

Keywords: Watermelon, watermelon seed, seed extractor

Introduction

Watermelon (*Citrullus lanatus*) is one of the important vegetable crop which belongs to the family Cucurbitaceae is a vine-like flowering plant originally from Southern Africa. It is a special kind of fruit referred by botanists as a "pepo", a berry which has a thick rind (exocarp) and fleshy center (mesocarp and endocarp). The skin is smooth, with dark green rind or sometimes pale green stripes that turn yellowish green when ripe. Watermelon is an important summer season crop whose peak season of harvest falls on the hot summer days and is highly relished due to its cool and thirst quenching property. The pulp is juicy and sweet, with an attractive red colour that attracts consumers (Shankara et al., 2012).

Coming to the seed extraction process, most of the government and private agencies are extracting the watermelon seeds manually. This manual method of extracting melon seeds involves manual cracking of the fruits with wooden clubs or cutting off the head or tail then boring the fruits with a knife. The fruit so treated is left about 1-2 days to decompose, and then the seeds are removed by washing in clean water. Thus the manual process is both time and effort-consuming process (Amir, 2004). Once the seed extraction is performed, the whole fruit is being wasting or it is only used as animal feed.

In India, watermelon is cultivated in an area of 74,640 hectare with a total production of 1,810 million tonne with a productivity of 24.2 t/ha. Similarly in the state of Karnataka the watermelon cultivated area is about 9,500 ha with the total production of 317 million tonne and productivity of 33.4 t/ha during the year 2013-14 (Anonymous, 2014).

In India, the recommended seed rate for cultivation of watermelon is 0.3 kg/ha and one kg/ha for hybrids and varieties respectively. Thus the estimated seed requirement to cultivate 9,500 and 74,640 ha of land is about 9 and 72 tonnes, respectively for Karnataka and India. This traditional method requires a lot of time, labour and tedious. By traditional method of extraction process the demand of seeds is difficult to meet economically due to time and energy constraints.

By considering all these points, there is a need to develop a machine to extract good quality or high vigour watermelon seeds from the fruit to meet out seed requirement as well as to extract good quality watermelon juice for further value addition.

Materials and Methods

Physical and Textural Properties of Matured Watermelon Fruit and Seeds

Physical properties such as, linear dimensions (i.e. length, breadth and thickness), arithmetic mean diameter, geometric mean diameter and sphericity of fruit and seeds were measured (Mohsenin, 1970) including rind thickness in order to design the different components of the extractor.

The different textural properties such hardness of the watermelon rind, cutting strength of watermelon rind, cutting strength of watermelon seed and puncture test of watermelon seed were studied with the help of texture analyzer (Make: Stable Microsystems Ltd, UK; Model - HDi) using suitable probes and operation settings to design the fruit cutting and extraction system.

Development of a Watermelon Seed Extractor

The machine was developed for the extraction of seeds as well as quality juice from the whole watermelon fruit. The watermelon seed extractor was designed, developed and fabricated in the Section of Agricultural Engineering, Indian Institute of Horticultural Research, Hessaraghatta, Bengaluru. The various physical and textural properties of matured watermelon fruit as well as its seeds were measured in order to design various systems of the seed extractor and were as follows (**Fig. 1**);

i. Watermelon cutting system
ii. Watermelon seed extracting system
iii. Seed separation system
iv. Power transmission system

Watermelon Cutting System

The watermelon cutting system consisted of a fruit holding trolley, circular cutting blade, a hopper and main frame. A stainless steel circular serrated blade of 400 mm diam-

Fig. 1 Performance evaluation of developed watermelon seed extractor

eter and 2 mm thick was mounted above the frame with necessary shaft, bush and fasteners.

The top surface of trolley frame was covered with 1 mm thickstainless steel sheet. A stainless steel flat of 25 × 3 mm was bent into semicircular form and fitted on the frame to hold the fruit while cutting. Four wheels were provided at four corners of the frame so that the frame could be moved forward and backward. Handle was provided to move the fruit holding trolley in forward and backward movement while cutting the fruit.

The hopper of the cutting system has 620 mm length, 450 mm width with the hopper bottom of 50 mm diameter, which was fabricated using 1 mm stainless steel sheet in order to collect the juice and seeds while cutting.

Watermelon Seed Extracting System

The watermelon seed extracting system consisted of a scraper fitted to stainless steel shaft, and power transmission system. A scraper shaft of 725 mm length, 20 mm diameter made up of stainless steel was mounted on a main frame with the help of pillow bearing blocks. The scrapers with different materials were fabricated such as stainless steel, wooden and nylon. The scraper was fixed on the scraper shaft with the help of nut and bolt. The scraper shaft was driven by chain and sprocket arrangement with necessary speed reduction ratio.

Seed Separation System

The seed separation system consisted of (i) concentric cylindrical drums viz. perforated cylindrical drum and solid cylindrical drum, (ii) pulp beaters and (iii) drive transmission system. The cylindrical drums were fabricated out of stainless steel (SS 304 grade) of 1 mm thickness stainless steel material had diameters of 220 mm and 300 mm respectively for perforated and solid drum, respectively. Two types of perforated drums having 3 mm and 5 mm diameter perforations were fabricated to conduct the preliminary studies on juice separation system.

Beater holders which were fabricated out of 85 mm, 60 mm and 5 mm stainless steel flats. The beaters were fitted to the beater holder and fabricated in 'T' shape. These beaters were mounted on a shaft of 25 mm diameter at an interval of 60 mm at spiral arrangement in order to push and move forward material. The shaft was driven by necessary belt and pulley from the main power source.

Power Transmission System

The extraction unit was fitted with a three phase 50 rpm, one hp geared motor. Provision was made at the base point to move the motor to and fro to provide required belt tension. The cutting system is driven by the geared motor with the help of 'V' belt and pulley. Further the power at the end of the cutting system shaft was transmitted to the load shaft with the help of bevel gears. From this load shaft, scraper shaft got the power through chain drive and crushing shaft got the power from the load shaft through crossed 'V' belt and pulley to rotate the beater shaft in clock wise direction.

Performance Evaluation of the Watermelon Seed Extractor

The developed seed extractor was tested as per the combinations of various treatments as given in **Table 1**. The watermelon fruit was fed through fruit cutting unit where it cut the fruit into two halves. After the cutting one half of fruit was fed to scraper manually. The pulp, juice along with seeds was scraped out from fruit and collected in the seed separation unit. In seed separation unit, juice was collected through juice outlet and the uncrushed pulp along with the seeds was collected from the seed outlet. The collected pulp along with the seeds from seed outlet was kept for fermentation for a duration of 24 h. Once the fermentation period was completed, the seeds were separated by washing the fermented pulp with the help of clean water and were dried under shade. The same procedures were followed for all the trials and were carried out with 3 replications to overcome the experimental errors. After recording all the observations various dependent parameters like Capacity, Seed extraction efficiency, Percentage seed damage, Percentage Seed loss and Seed vigour index were measured by using the following formulae where as germination study was carried out in the walk in germinator chamber which was maintaining 20 ± 1 °C temperature and 90% relative humidity.

Extraction capacity (%) = Weight of seeds extraced (kg) / Time taken (h)

Extraction efficiency (%) = (Total weight of the seed collected at seed outlet (g) / Total weight of the seed collected at all outlets (g)) × 100

Percentage seed loses = (M_o / M_t) × 100

Percentage of damage seeds = (M_d / M_t) × 100

Table 1 Experimental design

Independent parameters	Levels	Dependent parameters	Replications
Scraper type	Sc_1 (Stainless steel) Sc_2 (Wooden) Sc_3 (Nylon)	1. Capacity of the extractor 2. Seed extraction efficiency 3. Percentage seed damage 4. Seed loss 5. Germination percentage 6. Seed vigour index	3
Cutting plane	Cp_1 (Transverse) Cp_2 (Longitudinal)		
Peripheral speed	S_1 (50 rpm) S_2 (100 rpm) S_3 (150 rpm)		

Seed vigour index (VI) = [Whole seedling length (mm) × Germination (%)]

Where,

M_o = Total weight of seeds expelled out of the machine with peel and washed water (g)

M_t = Total weight of the seeds contained in the fruit sample (g)

M_d = Total weight of damaged seeds collected from all outlets (g)

Statistical Analysis

The results of the machine performance for different treatments of watermelon seed extraction were analyzed using Factorial Completely Randomized Design (FCRD) with three replications by using AGRES software (Fisher and Yates, 1963).

Experimental Results

Physical and Textural Properties of Matured Watermelon Fruit and Seeds

The results of different physical and textural of matured watermelon fruit and seeds were tabulated in **Table 2** and **Table 3**. respectively. Based on these properties, the different components of the extractor were designed in order to extract the seeds with negligible damage.

Performance Evaluation of Watermelon Seed Extractor

The prototype watermelon seed extractor was tested for its performance with three different scrapers, two cutting planes and three operating speed with three replications. For each trial one fruit was fed and various observations were recorded. The performance parameters namely capacity of the seed extractor, seed extraction efficiency, percentage seed loss, percentage seed damage, germination percentage and seed vigour index for three scrapers, two cutting planes and three operating speed were tested and results were tabulated in **Table 4**.

Capacity of the Seed Extractor

The results of the extractor capacity revealed that, the capacity of developed seed extractor was increased with increase in the speed for stainless steel and wooden scraper whereas the capacity of the extractor was decreased with increase in speed for nylon scraper. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the capacity of the extractor. The nylon scraper with transverse cutting plane at 50 rpm operating speed showed the highest capacity of 1.98 kg seeds/h. whereas lowest capacity was found forwooden scraper (0.40 kg seeds/h) with transverse cut at an operating speed of 50 rpm. Because, in this combination enough time was available to the seeds to get separated from the fruit as well as sufficient area of contact between the scraper and fruit. The similar findings were obtained for extraction watermelon seeds machine by Eliwa and Elfatih (2012).

Seed Extraction Efficiency

The results of the extraction efficiency indicate that, the extraction efficiency of developed seed extractor was increased with increase in the speed for all scrapers in both cutting planes. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the extraction efficiency of the extractor. The nylon scraper with transverse cutting plane operated at a higher speed of 150 rpm showed the highest extraction efficiency of 99.49% compared to all combinations. Because, of the shape and material factor, serrations provided on the nylon scraper and sharp edge provided at the front which improves the penetration into the fruit. The extractor had shown the highest extraction efficiency at higher speed of 150 rpm because at higher speed almost all the pulp and seed were removed by the scraper irrespective of juice conversion. This resulted in higher extraction efficiency at higher speed. The results of Eliwa and Elfatih (2012) showed that higher extraction efficiency was found at higher speed with transverse cutting plane.

Percentage Seed Loss

The results of the percentage seed loss showed that, the seed loss of developed seed extractor was decreased with increase in the speed

Table 2 Physical properties of matured watermelon fruit and seed

Property	Watermelon fruit		Watermelon seed	
	Mean	Standard deviation	Mean	Standard deviation
Linear dimensions				
a. Length, mm	253.66	17.88	10.64	1.11
b. Breadth, mm	152.50	12.20	5.42	0.78
c. Thickness, mm	143.16	9.30	2.16	0.66
a. Arithmetic mean diameter, mm	183.11	11.58	6.07	0.62
b. Geometric mean diameter, mm	176.82	11.23	4.93	0.74
c. Sphericity	69.78	2.60	46.64	6.90

Table 3 Textural properties of matured watermelon fruit and seed

Sl. No	Property	Mean	Standard deviation
1	Hardness of watermelon rind, N	11.05	1.60
2	Cutting strength of watermelon rind, N	244.09	0.98
3	Cutting strength of watermelon seed, N	15.62	1.95
4	Puncture force for seeds, N	8.45	2.08

for stainless steel and nylon scraper, whereas increase in seed loss with increase in speed was observed for wooden scraper in both cutting planes. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the percentage seed loss of the developed extractor. The minimum percentage seed loss was observed for both nylon and stainless steel scraper (0.19%) with a transverse cutting operated at 150 and 100 rpm speed for nylon and stainless steel, respectively followed by the nylon scraper (0.26%) with longitudinal cutting operating at a speed of 100 rpm. The lesser seed loss is due to the shape and material factor as well as serrations provided on the nylon scraper. Due to this reason, the scooping action performed by scraper led to minimum seed loss compared to wooden scraper.

According to Eliwa and Elfatih (2012) it was observed that increasing cutting speed tends to increase seed losses due to increasing vibration and instability of the fruit over pulp scraping unit. But in case of developed seed extractor decrease in seed losses were observed with increase in speed. Because at higher speed, there was less resident time was given to pulp and seeds and it reached the outlet in lesser time compared to lower speed.

Percentage Seed Damage

The results of the percentage seed damage revealed that, the seed damage of developed seed extractor was decreased with increase in the speed for all the scrapers in both cutting planes. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the percentage seed damage of the developed extractor. The nylon scraper with longitudinal cut operating at a speed of 100 rpm showed the minimum (0.26 %) percentage of seed damage followed by nylonscraper (0.32%) with longitudinal cut at an operating speed of 150 rpm. The maximumpercentage of seed damage (6.87%) was observed in the wooden scraper with transversecut operating at a speed of 50 rpm.

According to Eliwa and Elfatih (2012) most of seeds were damaged during cutting the fruits into two halves and during separation of pulp and seeds from its rind. Due to the impact and shearing force seeds were damaged in any of the section as explained. It was observed that increase in cutting speed tends to decreased seed damage because of the less resident time took in the seed separation system as well as extraction system.

Germination Percentage

The results of the germination study revealed that, the germination percentage of the watermelon seeds extracted from the developed seed extractor was increased with increase in the speed for all the scrapers in both cutting planes. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the germination percentage of watermelon seeds extracted from the developed seed extractor. The nylon scraper with longitudinal cut operatingat a speed of 100 rpm gave the maximum germination percentage of 97.04%. The least germination percentage was observed inwooden and nylon scraper with transverse cutting operating at a speed of 100 rpm. According to Kushwaha et al. (2005) at higher peripheral speed there was increased deformation of seed and may be the possible cause of the lower germination and constant speed in crushing chamber. But in developed seed extractor, germination percentage was increasing as the speed increased because; same speed was maintained in all system. Thus at 100 rpm speed, higher germination percentage was observed.

Seed Vigour Index

The results of the Seed vigour

Table 4 Performance evaluation of seed extractor

Scraper	Cutting Plane	Speed	Capacity (kg of seeds/h)	Efficiency, %	Seed loss, %	Seed damage, %	Germination, %	Vigour index
Sc_1	Cp_1	S_1	0.61	93.95	0.77	5.28	72.00	1134
		S_2	1.06	97.93	0.19	1.87	80.00	1374
		S_3	0.95	95.01	0.83	4.16	82.00	1540
	Cp_2	S_1	0.49	94.04	0.72	5.24	86.57	2069
		S_2	0.66	94.87	0.37	4.76	84.00	1554
		S_3	1.21	95.92	0.58	3.51	88.00	2544
Sc_2	Cp_1	S_1	0.40	89.46	3.67	6.87	84.00	1571
		S_2	1.27	95.54	1.83	2.63	70.00	1044
		S_3	0.66	92.35	2.49	5.16	72.00	1158
	Cp_2	S_1	0.46	90.24	3.12	6.64	68.00	1078
		S_2	1.13	94.83	2.20	2.97	94.00	2645
		S_3	0.46	91.55	3.44	5.00	92.61	2472
Sc_3	Cp_1	S_1	1.98	98.88	0.38	0.74	92.00	2933
		S_2	0.89	97.39	1.15	1.46	70.00	1325
		S_3	1.39	99.49	0.19	0.32	86.00	2039
	Cp_2	S_1	1.00	97.65	0.95	1.41	72.48	1238
		S_2	1.53	99.48	0.26	0.26	97.04	3284
		S_3	1.23	98.92	0.53	0.55	96.64	2904

Where,
Sc_1 - Stainless steel scraper, Sc_2 - Wooden scraper, Sc_3 - Nylon scraper; Cp_1 - Transverse plane, Cp_2 - Longitudinal plane; S_1 - 50 rpm, S_2 - 100 rpm, S_3 - 150 rpm

index indicated that, the vigour of the watermelon seeds extracted from the developed seed extractor was increased with increase in the speed for all the scrapers in both cutting planes. The data presented in the **Table 4** observed that different scrapers, cutting planes and operating speeds had higher significant effect over the vigour index of watermelon seeds extracted from the developed seed extractor. The highest seedvigour index (3284) was observed in nylon scraper with longitudinal cut operating at aspeed of 100 rpm whereas the least seed vigour index (1044) was observedin wooden scraper with transverse cut operating at a speed of 100 rpm. According to Kushwaha et al. (2005) at higher peripheral speed there was increased deformation of seed and may be the possible cause of the lower germination. This lesser germination percentage results in lower vigour values. Thus at 100 rpm speed higher Seed vigour index was observed with longitudinal cutting plane.

Conclusions

By considering the overall machine performance, the nylon scraper with transverse cutting plane at 100 rpm operating speed was recommended as the best combination for watermelon seed extractor based to extract high vigour seeds as well as good quality juice for further value addition.

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Optimization of Energy Consumption of Okra Slices in a Solar-assisted Electric Crop Dryer

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Abstract

Optimization of energy consumption of okra slices in a solar-assisted electric dryer was presented. Response surface methodology (RSM) was adopted to optimize the energy consumption of the drying process as well as the drying parameters: air velocity (1.0, 1.5 and 2.0 ms^{-1}), slice thickness (10, 15 and 20mm) and temperature (50, 60 and 70 °C). The central composite rotatable design of the Design Expert 7.0 statistical package was used for the three drying parameters. The energy consumed for drying each 1000 g batch of okra slices at varying parameters was measured and recorded by an arduino microprocessor. A second order polynomial model was found suitable for energy consumption response with high R^2 value of 0.9984. Statistical analyses were conducted to verify the suitability of the model. A predictive energy consumption model was developed and validated with the experimental data. The process parameters had significant effects on the energy consumption as well as its interaction effects. Results obtained showed that the maximum and minimum energy consumed in drying 1000 g batch of sliced okra were 33.94 and 1.02 kW-h, respectively. The mean energy consumption was found to decrease with temperature at constant air velocity and vice versa, as well as increasing with air velocity at constant temperature. The maximum energy consumption values were obtained at: 50 °C and 1.0 ms^{-1}; 20 mm and 50 °C; and 1.0 ms^{-1} and 20 mm; whereas the minimum values were obtained at: 70 °C and 2.0 ms^{-1}; 10 mm and 70 °C; and 10 mm and 2 ms^{-1}. The optimum drying conditions of 1.54 ms^{-1} air velocity, 15.45 mm slice thickness and 54.8 °C temperature were found at a predicted energy consumption value of 34.27 kW-h for okra slices in a hybrid solar-electric dryer. A recommendation for further work was stated.

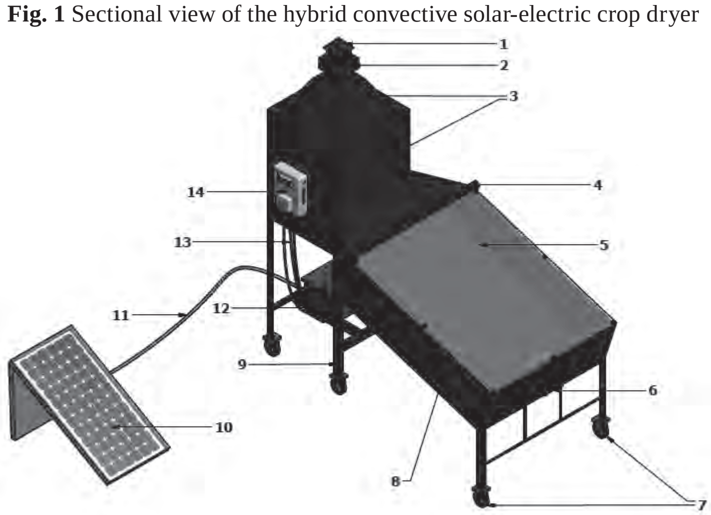
Keywords: Energy consumption, optimization, crop dryer, okra slices, response surface method

Introduction

Okra (*Hisbiscus esculentus or Abelmoschus esculentus* L.) is an economically important vegetable crop grown in tropical and sub-tropical parts of the world. It is widely cultivated as a garden crop as well as on large commercial farms for its fibrous fruit or pods (containing round, white seeds) which is used as a vegetable both in its green and dried state. It is a good source of vitamin C, vitamin A, dietary fiber, calcium and is low in saturated fat (Doymaz, 2005). Each 100 g of edible portion of okra contains about 89.6 g of water, 1.9 g of protein, 0.2 g of fat, 6.4 g of carbohydrate, 0.7 g of minerals and 1.2 g of fibre (Kumar et al., 2011). FAOSTAT (2008) reports that okra is grown commercially in India, Turkey, Iran, Western Africa, Yugoslavia, Bangladesh, Afghanistan, Pakistan, Burma, Japan, Malaysia, Brazil, Ghana, Ethiopian, Cyprus and the Southern United States. India ranks first in the world production of okra with 3.5 million tonnes accounting for 70% of the total world production which is obtained from over 0.35 million ha of land. Okra is known by many local names in different parts of the world. It is called lady's finger in England, gumbo in the United States of America, guino-gombo in Spanish, guibeiro in Portuguese and bhindi in India. It is quite popular in India because of ease of cultivation, dependable yield and adaptability to varying moisture conditions (Schalau, 2002). In the Eastern States of Nigeria, many good local varieties exist such as perkin's long pod, new lady's finger, nwaidu etc. (Anyawu et al., 1986).

It is usually available in large quantities between April and December in the South eastern part of Nigeria. Two distinct seasons exist for its production in Nigeria: the peak and the lean seasons. During the lean season (October to March), the product is produced in small quantities and the product becomes very scarce and expensive. Large quantities are produced in the peak season (April to September) in excess of the consumption capacity of the local populace and thereby it requires some level of preservation. Owolarafe and Shotonde (2004) reported that the traditional method of preserving okra in Nigeria and most African countries involves spreading it in the open in cold weather or refrigeration. This can only preserve okra for a limited period of time. Due to its perishable nature, the fruit deteriorates easily. This causes appreciable economic loss in the annual revenue of the farmer. Preservation of the fruit is commonly done by slicing the pod with a sharp knife, sun-drying and storing in calabash gourds or in a jute bag under normal room temperature. Drying has proved to be one of the most effective ways of preserving vegetable crops. The success of any drying operation depends on removing enough moisture from the food material to achieve a water activity too low to inhibit microbiological growth to take place. This in turn means that there must be sufficient transfer of heat to provide the latent heat of vaporization needed. As a result of the relatively high moisture content (88-90% w.b) of okra at harvest (Doymaz, 2005; Kumar et al., 2011; Tiwari, 2012), considerable amounts of energy is consumed during its drying process to a desired safe moisture level (Nwakuba et al., 2016). Drying is a highly energy-consuming process which has significant effects on the dried product quality such as its nutritional values, colour, shrinkage and other organoleptic properties (Darvishiet al., 2013). Therefore, the energy consumption of crop drying is important for the following major reasons: (i) in order to estimate the optimum quantity of temperature, air flow, and drying time most suitable for a particular sliced crop so as to avoid over drying which is consequent to undermining the nutritional value of the dried product; (ii) applied he design of appropriate cost effective drying system which would consume minimum amount of energy to convey the required sensible and latent heat given the crop's physical and biological characteristics; (iii) for simulation of drying systems. Therefore, with increasing pressures to reduce environmental degradation, both from the public and governments, it is necessary to improve drying processes by optimizing its process parameters to reduce its high energy consumption and greenhouse gas (GHG) emissions, while still providing a high quality product with minimal increase in economic input. This, in all would reduce the cost of drying operation, enhance the crop shelf-life, product diversity, substantial volume reduction, availability and acceptance of dried food products in the market at a reasonable price (Maskan, 2001; El-Mesery and Mwithiga, 2012).

Fig. 1 Sectional view of the hybrid convective solar-electric crop dryer



1. Chimney, 2. Outlet/exhaust fan housing, 3. Drying chamber hood, 4. Heating chamber, 5. Plain glass, 6. Inlet sensor fan, 7. Rollers, 8. Solar collector, 9. Angle iron support, 10. Solar panel, 11. Solar panel cable, 12. Inverter/battery unit, 13. Control unit cable, 14. Control unit

Materials and Methods

Dryer Description

The hybrid solar-electric convective dryer consists of two major integral components namely: the air heater units (solar collector and resistance wire) and the drying chamber having two layer of drying racks made of wire mesh where the sliced crop samples are placed in for drying as shown in **Fig. 1**.

Other components of the dryer include: solar panel (80 W, 12 V), DC battery (75 Amps, 12 V), control unit, liquid crystal display (LCD), inverter system, weighing balance, pyranometer (Apogee model) inlet and exhaust fans, plain glass of 4 mm thick, temperature and relative humidity sensors (LM-35 transducers), weighing balances, weight sensors, resistance wire (1,500 W), frame support (angle iron), and rollers. The air heater unit (solar collector) is in the shape of a wooden box. The box was made with a special type of plywood (NBF laminated board) and was lagged with fiberglass. A mild steel plate was formed in the shape of an open box of approximately the same size as the solar box and inserted in the solar box to act as an absorber plate. The whole unit was painted dull black (inside and outside) to enhance absorptivity of solar radiation. A 15 cm diameter hole is located in front of the solar collector with a suction fan to draw ambient air into the solar collector and subsequently into the drying chamber. A hood was formed at the outlet end of the collector to direct the incoming solar heated air into the drying chamber and also serves as the heating unit of the electric heat source: a 1500 W resistance wire was mounted across both ends of the plenum perpendicular to the direction of flow of the incoming air.

A plain glass of 4 mm thickness with the same cross-sectional area with the solar collector was used to place over the solar box. The glass therefore, serves as a glazing material that allows radiation from the sun to pass through it before it is absorbed by the absorber plate. However, the heart of the hybrid dryer comprises an Arduino microprocessor which controls the overall operation of the system and automates tasks such as temperature and humidity control, sample weight loss, and electrical energy consumption (from AC and DC sources). The system also contains a main heating element powered by alternating current (AC) from the Public Power Supply or an electricity generating set. Transducers (for recording both temperature and relative humidity) are placed at five strategic points on the hybrid dryer namely: chimney, two drying racks, solar collector and inlet fan, where measurements are taken automatically by the microprocessor unit and displayed on the LCD as shown in legend S/No. 1 of **Fig. 2**.

Different drying temperatures and air flow velocities can be selected using a 4 × 4 matrix keypad panel and LCD for displaying the current state of the system. In the drying chamber, the drying racks are rigidly suspended on a weighing balance that records the sample weight loss through the use of a weight sensor attached to it with the help of a flat iron bar. A 1,500 W resistance wire supplies electrical heat to the drying chamber at fixed temperatures.

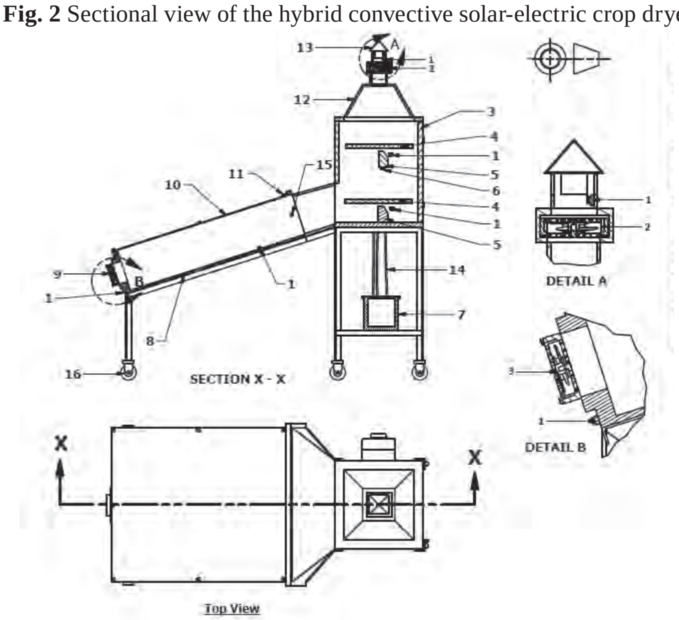
The control unit and its accessories as well as other instrumentations are powered by a 75 Amps, 12 V accumulator which is simultaneously charged by an 80 W solar panel; whereas the resistance wire (heater) is powered by a public power supply or an electric generator. The energy consumption from the accumulator and AC are measured and recorded by the control unit. When the control unit is connected to the computer through the use of a universal serial board (USB), a specialized software known as SCADA (Supervisory Control and Data Acquisition) is used to log the readings at 30 minutes interval and the results, stored in a database for immediate or future analyses, thus the system is fully automated.

Experimental Procedure

A local variety of fresh okra samples (*Nwaidu Spp.*) were procured from relief market in Owerri, Imo State of Nigeria. The samples were sorted according to relative size, washed and sliced into three thickness sizes of 10, 15 and 20 mm using a sharp stainless steel knife and a Vernier caliper with the direction of cutting perpendicular to the vertical axis of the tomato samples. The dryer heating units were switched on; the required air flow and temperature were initiated by the 4 × 4 matrix keypad of the control unit. The drying chamber was allowed to maintain steady-state condition. The dryer was designed to operate at three different fixed air temperature thresholds of 50, 60 and 70 °C, and at three varying air velocities (1.0, 1.5 and 2.0 ms^{-1}).

The initial mass of the sliced samples was measured by a digital weighing balance (0.01 g, Camry instruments, China) and recorded, and placed on the drying racks in such a way that the drying air flows axially into the sample matrix (for faster drying). The initial moisture content of the samples was measured by drying 20 g of representative sliced sample in an oven dryer set at 105 °C for 24 hours (Koyuncu et al., 2007; Darvishi et al., 2013). At every 30 minutes time interval, the programmed arduino microprocessor measures, records and displays on the screen, the weight loss, temperatures and relative humidity of the two drying racks, chimney, solar collector, and ambient environment. The difference between the initial dried mass and the 30-minutes in-

Fig. 2 Sectional view of the hybrid convective solar-electric crop dryer



1. Temp./humidity sensors, 2. Exhaust fan, 3. Door frame, 4. Drying racks, 5. Weighing balance, 6. Iron bar (support), 7. Battery/inverter assembly, 8. Solar collector, 9. Inlet fan, 10. Plain glass, 11. Screw bolt, 12. Drying chamber hood, 13. Chimney, 14. Cable, 15. Heating element, 16. Roller

terval measured masses were used to calculate the percentage weight loss for each dried batch. The weight losses were recorded by the use of a weight sensor attached to the weighing balance (with a precision of 0.01 g) in the drying chamber.

The amount of electrical energy used by the heating element, drying rate and time for each drying batch (at various slice thicknesses, temperatures and air velocities) were measured and recorded by the control unit. The experiment was repeated for varying sample thicknesses, temperatures and air velocities for a batch of 1,000g. Drying was stopped when the sample attained a weight corresponding to a desired final moisture content of 8% w.b (Eke, 2014; Ekpunobi et al., 2014; Tiwari, 2012).

Statistical Methodology

The experiment was designed using Design Expert version 7.0 and randomized in order to minimize the effects of unexplainable variability in the observed responses. Air velocity at levels 1.0, 1.5 and 2.0 m/s; slice thickness at 10, 15 and 19 mm; and temperature at levels 50, 60 and 70 °C were the experimental treatments or factors involved with three replications. The mean values used for the analysis is presented in **Table 1**. Response surface method (RSM) was used to determine the relative contributions of each of the experimental variable (treatments: A, S, and T) to the response factor, energy consumption. RSM is a statistical procedure that applies quantitative data from appropriate experimental design to determine optimal conditions (Kusuma and Mahfud, 2016). It is used in determining the relationship between one or more measured responses and multiple input variables, as well as providing statistically acceptable results with fewer numbers of experiments (Kaur et al., 2009; Kumar et al., 2011).

A second-order polynomial function was used to analyze the relationship between the response variable (energy consumption) and the experimental data as expressed in Equation (1):

$$E = \beta_{no} + \sum_{i=1}^{k} \beta_{ni} \cdot A_i + \sum_{i=1}^{k} \beta_{ii} \cdot A_i^2 + \sum_{i \neq j=1}^{n} \beta_{nij} \cdot A_i \cdot A_j + \epsilon_r \quad (1)$$

where, E = response variable (energy consumption, kW-h); β = regression coefficient; n = number of variables studied in the experiment (A, S and T); β_i = regression coefficient for linear effect terms; β_{ij} = interaction effects; β_{ii} = quadratic effects; A is = coded process variables (-1, 0 and 1); ϵ_r = random error (which measures the experimental error).

A statistical package, Design Expert, version 7.0 was used for analyzing the data as well as to perform RSM to obtain three dimensional response graphs. The fitted polynomial function was expressed as 3-D surface plots in order to visualize the relationship between the response and the experimental levels of each variable and to deduce the optimal conditions. The regression analysis and analysis of variance (ANOVA) were conducted to fit the model and to find the statistical significance of the model terms. From the ANOVA, the effect and regression coefficients of each linear, quadratic and interaction terms were determined. Statistical calculations were made using the regression coefficients in order to generate 3-dimensional and contour maps from the regression models. P-values less than 0.05 were considered to be statistically significant. A 3-D graphical optimization of process variables based on the single and overall responses of the A, S and T variables was performed using Design Expert 7.0 software, which seeks to find the variables which can result in maximum and minimum energy when the dryer is optimally operated. This was done by keeping one variable constant at the centre point and varying the other two variables within the experimental range.

Table 1 Levels of treatment variables and energy consumption for drying of okra slices

Std.	Run	Air velocity, A (m/s)	Slice thickness, S (mm)	Temeprature, T (°C)	Energy consumption (kW-h)
1	17	1.00	10.00	50.00	20.36
2	12	2.00	20.00	70.00	9.10
3	10	2.00	15.00	50.00	22.84
4	5	1.00	20.00	60.00	40.11
5	16	1.50	10.00	70.00	3.21
6	8	2.00	10.00	60.00	4.36
7	4	1.50	20.00	50.00	50.47
8	14	1.00	15.00	70.00	13.83
9	18	1.50	15.00	60.00	21.91
10	9	2.00	10.00	50.00	8.99
11	6	1.00	20.00	50.00	60.02
12	2	2.00	20.00	50.00	40.89
13	13	1.00	10.00	70.00	4.46
14	3	2.00	10.00	70.00	2.03
15	7	2.00	20.00	70.00	9.10
16	1	1.00	10.00	50.00	20.36
17	11	2.00	10.00	50.00	8.99
18	15	1.50	15.00	60.00	21.91

Energy Consumption

The energy consumption for drying of agricultural products is expressed as:

$$E_R = AV\rho_a C_{pa} \Delta TDt \quad (2)$$

where, E_R = energy required for each drying phase (kW-h); A = sample plate area (m^2), V = air velocity (m/s), ρ_a = air density (kg/m^3), Dt = total drying time of each sample (h), ΔT = temperature difference between ambient and hot air (°C), and C_{pa} = specific heat of air (kJ/kg°C).

This energy consumption for drying a given batch of okra slices was measured by an Arduino microprocessor and compared with the values obtained from Equation (2) and was found to be less by an average of 9.78%. This difference could probably be as a result of the constant air density and air velocity used in the calculated values (Equation 2), whereas in the measured values, different air velocities were considered by the Arduino platform and was considered negligible. The calculated energy values were used to calibrate the Arduino system as well as validating the Arduino-measured values.

Results and Discussion

The mean energy consumption for drying a batch of okra slices increased with increase in the slice thickness and decreased with temperature at constant air velocity as shown in **Fig. 3**.

This is so because with increasing slice thickness, sample capillary distance increases, thereby consuming more time to diffuse and evaporate its internal and surface moisture, which increased the mean energy consumption but decreased as air velocity was increased. At increased air velocity, more convective air passes across the product surface increasing surface moisture evaporation as well as reducing the drying time, thus decreasing the energy consumption. Increasing the air velocity at increasing temperature reduced the mean energy consumption as a result of more convective air entering the drying chamber to increase the rate of surface moisture evaporation as well as increasing the sample kinetic energy of internal moisture for rapid diffusion and reduced resistance to capillary transport. For each slice thickness and air velocity, mean energy decreased as temperature increased as a result of quicker drying rate due to increased thermal gradient between the drying air and the sample product. Less drying time was obtained at higher temperatures, thus less power consumed by the heating unit (resistance wire), and less energy was required to dry the sliced samples. Less energy was required to dry the fresh samples to a corresponding final moisture level of 8% wet basis (w.b) at increasing air velocity and constant slice thickness and temperature. This is because at low air velocity, drying time increases due to low convective flow of heated air to evaporate moisture from the product surface hence, more time is taken for the electric power to be consumed, amounting to more energy. This was also observed by Koyuncu et al. (2007) in drying of cornelian cherry fruits; Sarsavadia (2007) in drying of onion slices; Tripathy et al. (2009) in drying of potato slices; Yahya et al. (2011) in drying of green tea leaves; Afolabi et al. (2014) in drying of ginger slices; Minaei et al. (2014) in drying of St. John's Wort leaves (*Hypericum Perforatum*). The maximum and minimum energy consumed in drying 1000 g batch size of sliced okra were 33.94 and 1.02 kW-h, respectively.

Effects of Drying Parameters on Energy Consumption

The drying variables affecting energy consumption for drying as considered in this study are: air velocity, slice thickness and temperature. The following combined effects of these variables (at varying treatment levels) on the energy consumption of 1 kg batch of okra slices are considered.

Air Velocity and Temperature

Fig. 4 illustrates the effect of drying air velocity and temperature on the energy consumption for drying of okra samples in a hybrid crop dryer. The energy consumption decreased as temperature increased at constant air velocity; with increasing temperature, the drying time was reduced due to increased thermal gradients inside the material that consequently increased the product drying rate. Also, energy consumption decreased with increasing air velocity and drying temperature: little quantity of energy is required to dry a given product sample at constant slice thickness and at increasing temperature and air velocity due to greater heat transfer (high kinetic energy of internal moisture) and water vapour

Fig. 3 Energy consumption of 1 kg batch of okra slices in a hybrid dryer

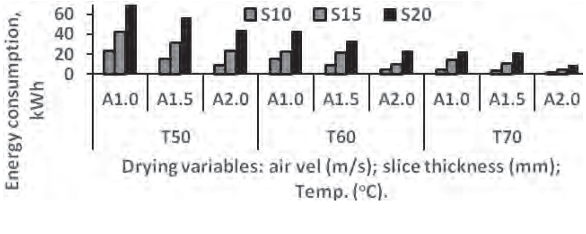


Fig. 4 Effects of temperature and air velocity on the energy consumption for drying of okra samples in a hybrid crop dryer

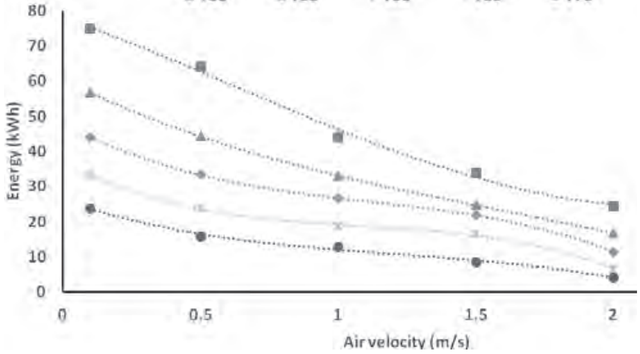


Table 2 Response surface central composite design (CCD) and energy consumption of okra slices

Run	Actual variables			Energy consumption (kWh)		
	A (m/s)	S (mm)	T (°C)	Experimental	Predicted	Residue
1	1.00	10.00	50.00	20.36	20.37	-0.014
2	2.00	20.00	70.00	9.10	9.35	-0.25
3	2.00	15.00	50.00	22.84	24.44	-1.60
4	1.00	20.00	60.00	40.11	40.78	-0.67
5	1.50	10.00	70.00	3.21	3.38	-0.17
6	2.00	10.00	60.00	4.36	5.12	-0.76
7	1.50	20.00	50.00	50.47	50.26	0.21
8	1.00	15.00	70.00	13.83	13.22	0.61
9	1.50	15.00	60.00	21.91	20.86	1.05
10	2.00	10.00	50.00	8.99	8.65	0.34
11	1.00	20.00	50.00	60.02	60.29	-0.27
12	2.00	20.00	50.00	40.89	40.23	0.66
13	1.00	10.00	70.00	4.46	5.17	-0.71
14	2.00	10.00	70.00	2.03	1.59	0.44
15	2.00	20.00	70.00	9.10	9.35	-0.25
16	1.00	10.00	50.00	20.36	20.37	-0.014
17	2.00	10.00	50.00	8.99	8.65	0.34
18	1.50	15.00	60.00	21.91	20.86	1.05

pressure deficit that occur at higher drying temperatures and air velocities, thereby product moisture faces less resistance to evaporation which gives rise to greater heat and mass diffusion (to and from the product) in a shorter time, thus reducing the amount of energy required for drying process. This phenomenon was observed in the works of Akanbi et al., 2006; Billiris et al., 2011; Motevali et al., 2012; Minaei et al., 2014; Sepehrimehr and Kohan, 2015.

Temperature and Slice Thickness

The effects of drying temperatures and slice thickness on the energy requirements are presented in **Fig. 5**. Increase in sample thickness increases the energy consumption at constant drying temperature: the thicker the sample material, the higher the energy as a result of longer heat transfer distance (from the food matrix to the surface), hence longer drying time (as energy consumption is a function of time).Increase in drying temperature generally reduces the energy consumption. Adding more heat to the drying medium (at constant slice thickness) increases the temperature gradient between the drying air and the product sample which increases the rate of intra-particle moisture diffusion and evaporation from the material at a minimum time due to increased kinetic energy of the internal water molecules of the sample products, as well as reducing the relative humidity of the air which increases its moisture-carrying capacity. Therefore, drying a 10 mm sample layer at 70 °C air temperature consumes a little amount of energy.

Slice Thickness and Air Velocity

Increase in air velocity at constant slice thickness reduces the energy requirement as shown in **Fig. 6**. Drying rate increases at higher air flow which lessens the drying time, hence reduced energy consumption. It is clear that energy consumption of sliced okra varies directly and inversely with the sample thickness and air velocity respectively. More energy is consumed at increased slice thickness as well as at reduced air velocity due to increased capillary distance of moisture diffusion and reduced mass transfer rate.

Statistical Analysis and Model Fitting

The results of RSM analysis conducted are presented in **Table 2**. The coefficients of the response surface equation (Equation 1) were determined using the experimental values of the energy consumption. The regression coefficients of the linear (A, S and T), quadratic (A^2, S^2 and T^2) and interaction terms (AS, AT and ST) were obtained using the least square method (LSD). The energy consumption of fresh sliced okra samples at varying air velocities, slice thicknesses and temperatures was obtained by a response surface

Fig. 5 Effects of temperature and slice thickness on the energy consumption for drying of okra samples in a hybrid dryer

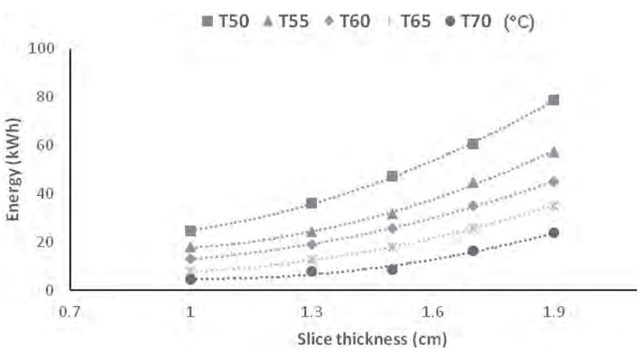
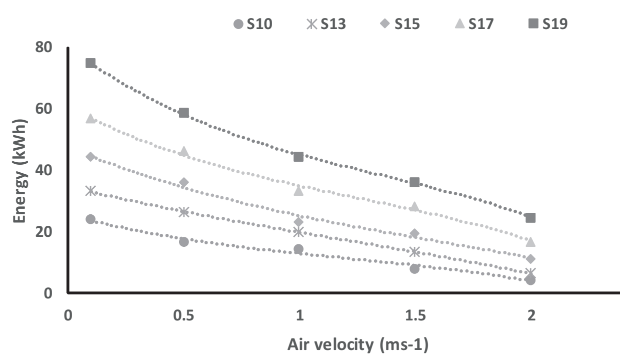


Fig. 6 Effects of slice thickness and air velocity on the energy consumption for drying of okra samples in a hybrid dryer



quadratic model expressed as:

$$E_C = 0.23A^2 + 0.32S^2 + 0.0086T^2 - 0.58AS + 0.2AT + 0.0018ST - 13.62A - 5.93S - 1.95T + 135.16 \quad [R^2 = 0.9984] \quad (3)$$

where, E_C = Energy consumption (kW-h); A, S and T = air velocity (m/s), slice thickness (mm) and temperature (°C), respectively.

The high value of coefficient of determination ($R^2 = 0.9984$) of the quadratic energy model is an indication of close correlation of the energy variables to the model equation. The model predicted the energy consumption so well with minimal residue. The analysis of variance (ANOVA) of the fitted model and the experimental results are shown in **Table 3**. The R^2 and adjusted R^2 values were 0.9984 and 0.9976, respectively. The high R^2 indicates that the variation could be accounted for by the experimental data satisfactorily fitting the energy model. And can be defined as a ratio of the explained variation to the total variation (Kumar et al., 2011; Kusuma and Mahfud, 2016).

The value of coefficient of variation (CV) is less than 10%, which indicates that the model for energy consumption was adequate gave better reproducibility (Mason et al., 1989; Giri and Prasad, 2007; Kumar et al., 2011; Kusuma and Mahfud, 2016). The predicted residual sum of squares (PRESS), which is an index of how a model fits each point in the design was 0.853 (Kusuma and Mahfud, 2016). The F-value of the model (28.61) implies that the model was significant. The CV was 4.17, which is < 10. Myers and Montgomery (1995) suggest that a ratio greater than 4 is adequate to precisely measure the signal-to-noise ratio. The energy model had a value, 15.275 which is a very good signal-to-noise ratio (Kumar et al., 2011; Kusuma and Mahfud, 2016). All these parameters indicate the reliability of the energy consumption model.

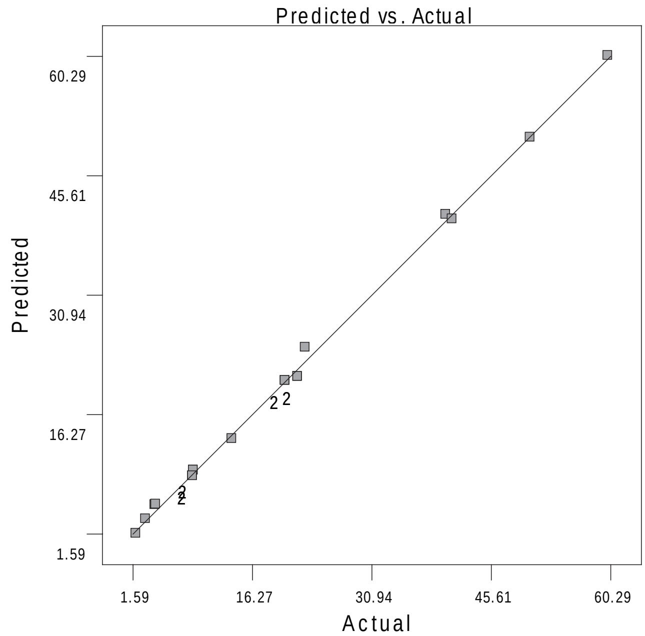
The significance of each of the coefficients in the model was checked using the P-values; which in turn indicate the pattern of the interactions between the model variables. The smaller the p-values, the more significant were the corresponding coefficients. It is evident from **Table 3** that the linear coefficients (A, S and T) and the S-quadratic coefficients were significant with very small p-values ($p < 0.05$). The other terms were not significant ($p > 0.05$) but significant at $p < 0.1$. The predicted values calculated from Equation 1 (**Table 2**) were in good agreement with the experimental values as shown in **Fig. 7** with R^2 approaching unity. Hence the quadratic model is well suited for the experimental

Table 3 ANOVA results for the fitted model

Variables	Sum of Squares	DF	Mean Square	F-value	P-value
Model	4,772.15	9	530.24	28.61	< 0.0001*
A - Air velocity	1,073.59	1	1,073.59	37.68	0.0001*
S - Slice thickness	2,371.02	1	2,371.02	83.22	< 0.0001*
T - Temperature	628.55	1	628.55	22.06	0.0008*
A^2	0.59	1	0.59	0.021	0.8880**
S^2	616.15	1	616.15	21.63	0.0009*
T^2	10.63	1	10.63	0.37	0.5549**
AS	49.05	1	49.05	1.72	0.2188**
AT	29.92	1	29.92	1.05	0.3297**
ST	0.050	1	0.050	1.741E-003	0.9675**
Residual	7.78				
R-Squared	0.9984				
Adj. R-Squared	0.9976				
PRESS	0.853				
CV	4.17				
Adequate precision	15.275				
Lack of Fit	5.08				
Pure Error	5.000				
Cor. Total	4,995.02				

*Significant at $p < 0.05$, **Significant at $p < 0.1$.

Fig. 7 Plot of predicted and experimental energy consumption under optimum conditions



set up and described the energy consumed by okra slices in a hot air convective solar-electric dryer.

Only air velocity, slice thickness and temperature had significant effects on the energy consumption ($p \leq 0.05$); whereas the cross product terms had significant effects ($p \leq 0.1$) on the energy consumption of sliced okra samples. The effects of the drying parameters on the energy consumption are shown in **Fig. 8**.

From **Fig. 8(a-c),** increasing drying temperature at constant air velocity, decreased mean energy consumption and vice-versa. Increasing the air velocity at constant temperature increases the evaporation rate of the surface moisture, thus increasing the drying rate and reducing the energy consumption. Maximum energy consumption is obtained at the lowest temperature and air velocity (50 °C and 1.0 ms^{-1} respectively); thickest sample slice and lowest temperature (20 mm and 50 °C respectively); lowest air velocity and thickest sample slice (1.0 ms^{-1} and 20 mm respectively). Whereas the minimum energy occurred at the highest temperature and air velocity (70 °C and 2.0 ms^{-1}); and the lowest slice thickness and highest temperature (10 mm and 70 °C respectively); lowest slice thickness and highest air velocity (10 mm and 2 ms^{-1} respectively). The optimum values of the process variables, obtained from desirability function using Design Expert 7.0 statistical package were: 1.54 m/s air velocity, 15.45 mm slice thickness and 54.8 °C temperature. Energy consumption of about 34.27 kW-h was predicted for drying of sliced okra in the hybrid solar-electric dryer under AST conditions.

Conclusions

Optimization of energy consumption for drying of sliced okra in a hybrid solar-electric dryer using response surface method has effectively estimated its interaction effects with the three independent variables: air velocity, slice thickness and temperature, as well as predicting the optimal operational conditions. A batch of 1000 g of okra slices (*Nwaidu spp.*) was dried from initial moisture content of 87.7% w.b to desired final moisture content of 8% w.b using a hybrid convective hot air dryer. Drying experiments were performed varying the process parameters: A (1.0, 1.5 and 2 ms^{-1}), S (10, 15 and 20 mm) and T (50, 60 and 70 °C) using a central composite rotatable design. The drying parameters were optimized based on the energy consumption of the drying process. The process parameters had significant effects on the energy consumption as well as interaction effects. The mean energy consumption decreases with increasing drying temperature at constant air velocity and vice versa, as well as increasing with the air velocity at constant temperature. Maximum energy consumption values were obtained at: 50oC and 1.0 ms^{-1}; 20 mm and 50 °C; and 1.0 ms^{-1} and 20 mm; whereas the minimum values were obtained at: 70 °C and 2.0 ms^{-1}; 10 mm and 70 °C; and 10 mm and 2 ms^{-1}. Based on the analysis of variance and the agreement of the experimental and predicted results, the generated energy model consumption model was suitable for the simulation of solar-assisted electric drying of okra slices under the stated treatment combination. The drying conditions of 1.54 ms^{-1} air velocity, 15.45 mm slice thickness and 54.8 °C temperature were found optimum for solar-assisted electric drying of okra slices.

Optimization of energy consumption of solar-electric crop dryers with heat recovery units for different sliced fruit vegetables is recommended for further studies.

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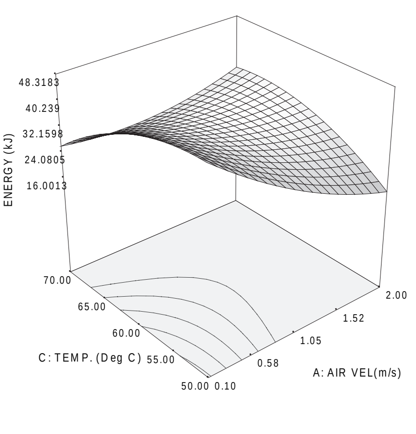

Fig. 8a Response surface plot of the interaction effect between temperature and air velocity on the energy consumption for drying of sliced okra samples in a hybrid crop dryer

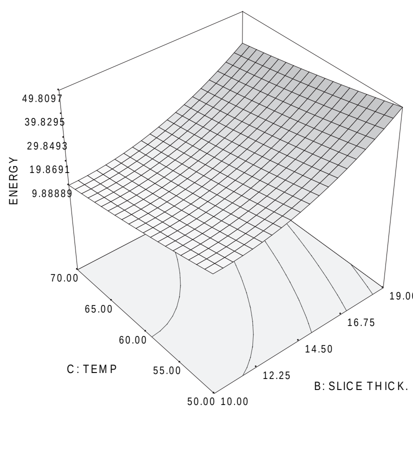

Fig. 8b Response surface plot of the interaction effect between temperature and slice thickness on the energy consumption for drying of sliced okra samples in a hybrid crop dryer

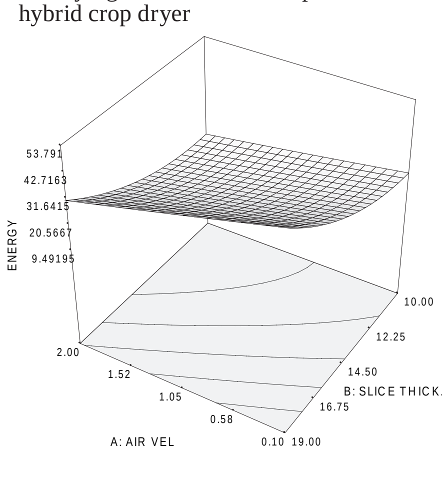

Fig. 8c Response surface plot of the interaction effect between air velocity and slice thickness on the energy consumption for drying of sliced okra samples in a hybrid crop dryer

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Design, Development and Evaluation of Minimal Processing Machine for Tender Coconut (*Cocos nucifera*)

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Abstract

This research involved designing and developing a prototype for minimal processing of tender coconut that could trim the outer husk to make a pentagonal shape coconut that could be cut open easily. The prototype consisted of a trimming unit and a power transmission system. The trimming unit consisted of a tender nut holder, shoulder-trimming knife, and bottom cutting knife. Shoulder trimming knife was set to the angle of 60° giving rise to minimum fibrous surface and short processing time. During operation, the tender coconut was placed vertically at the holder and clamped before the shoulder trimming took place. When the fruit rotated, the operator manually adjusted the trimming blades to trim the shoulder part of the fruit. The results show that the coconut trimming machine could trim 160 tender nuts. h^{-1} with a trimmed husk of 52.56 kg/h when the rotational speed was set at 428 rpm.

Introduction

Tender coconut water is a pure and healthiest natural beverage with many medicinal properties. The husk and hard shell of coconut is an exceptional package for the inside water which contains essential minerals, sugars, complex carbohydrates, ascorbic acid and vitamin-B complex, amino acids, phytohormones, electrolytes and cytokine (Harach and Jarimopas, 1995). It is an alternative to sports drinks and aerated beverages and low in calories. According to Sports Science Institute (USA), tender coconut water suitable for the budding sports drink market. Due to high perishable nature of tender coconut, it gets lost natural freshness within 24 to 36 h even under cold conditions unless treated scientifically (Geetha et al., 2016). Maturity and development of the tender coconut also strongly affects the fruit quality (Jarimopasa and Kuson, 2007).

The tender coconut is a bulky commodity due to husk that poses lot of transportation, packaging and handling problems. The average tender coconut weight is 1.5 kg of which the husk constitute of two-third of the volume of tender coconut or 65% of the total weight (Raju et al., 2002; Haseena et al., 2010). Hence, handling of tender coconuts will be easy if a major part of the husk is removed. But, the partial removal of husk resulted in rapid spoilage of nuts due to biochemical and enzymatic reactions are constrains in marketing of trimmed tender coconut in areas where coconut not grown (Haseena et al., 2010). To retain the wholesomeness, freshly harvested tender coconuts are transported to the processing house for trimming of individual nuts. The tender coconuts should be minimum six months old and the trimming process should allow the husk at least 1 cm thickness over the soft eye for minimal processing of tender coconut (Mohpraman and Siriphanich, 2012). The tender coconut is usually trimmed like pentagonal shape (conical shaped top with tapered cylindrical body and a flat base) to remove 50-60%

of the outer husk, and then shrink wrapped before being packed in corrugated boxes. The technology for preservation of trimmed tender coconut developed by Raju et al. (2002) involves dipping partially dehusked tender coconut in a citric acid (0.50%) and potassium metabisulphate (0.50%) solution for three minutes. They found that dipped tender coconut can be stored up to 24 days in refrigerated condition (5-7 °C). By using this process, tender coconut can be exported to distant place and served like other soft drinks.

Minimal processing of tender coconut reduces transportation cost and facilitates ease of packaging. Conventional trimming process requires skilled labour to shear the husk of tender coconut with sharp knife (Jarimopas and Ruttanadat, 2007). Hence, it is requires significant physical strength and extremely dangerous procedure. Other troubles related with conventional trimming process are the shortage of skilled labour, product is not uniform size, high production cost, and the time consumption (Yahya and Mohd Zainal, 2014). Consequently, several food engineers and coconut growers have attempted to develop tender coconut trimming machine (Harach and Jarimopas, 1995; Jarimopas and Pechsamai, 2002; Jarimopas and Ruttanadat, 2007; Jarimopas et al., 2009). Unfortunately, their machines were unacceptable by farmers, processors, exporters and traders due to low trimming capacity and complicate procedure involved during trimming process. Jarimopas and Ruttanadat (2007) developed a machine to trim tender coconut by horizontal lathe cutting mechanism. However, most of the operating time was wasted to reposition of the tender nut and readjustment of trimming knife for each trimming operation (Jarimopas et al., 2009).

In fact, many coconut growers, traders, exporters and entrepreneurs have expressed the interest to have machinery that is able to trim the tender coconut into a pentagonal shape (flat bottom, round body and pyramid top) as fast as can currently be done by labor. Therefore, there is an urgent need for minimal processing machine to trim the tender nuts in simple and safer way. The main aim of the present study was development and performance evaluation of minimal processing machine for tender coconut with vertical holding and trimming mechanism.

Materials and Methods

Raw Materials

Freshly harvested tender coconuts were obtained at Farm Office, Central Plantation Crops Research Institute (CPCRI), Kasaragod. Knowledge on the physical properties of the coconut was substantial in designing the tender coconut trimming machine. Hence, 50 of intact tender coconuts from west cost tall (WCT) variety were selected and measured to determine the diameter, height and husk thickness of each tender coconut using the vernier caliper. It is important to ensure that the tender coconuts were free from physical damage beforehand. Each tender coconut was dehusked manually to determine the husk thickness.

Machine Design

The design concept was to trim the vertically rotating tender coconut by applying an inclined knife in translation motion in order to get the desired shape of the tender coconut. The tender coconut trimming machine consisted of a main frame, spring arm, sliding slot, tender coconut holder, a shoulder trimming knife, a bottom cutting knife and power transmission unit. **Fig. 1** illustrates the engineering diagram of the machine and its important components.

The horse power of motor required to rotate tender coconut holder was calculated by using the following formula (Khurmi and Gupta, 2006),

Hp required to operate tender coconut minimal processing machine = $2\pi NT / 4500$

Where,

N= rpm of tender-nut holder,

T= Torque, kg-m,

Self weight of the spring arm = 4.5 kg

Average tender coconut weight = 1.2 kg

Force required to trim the tender coconut (0.4 cm) = 19.8 kg

Diameter of tender coconut holder = 0.09 m

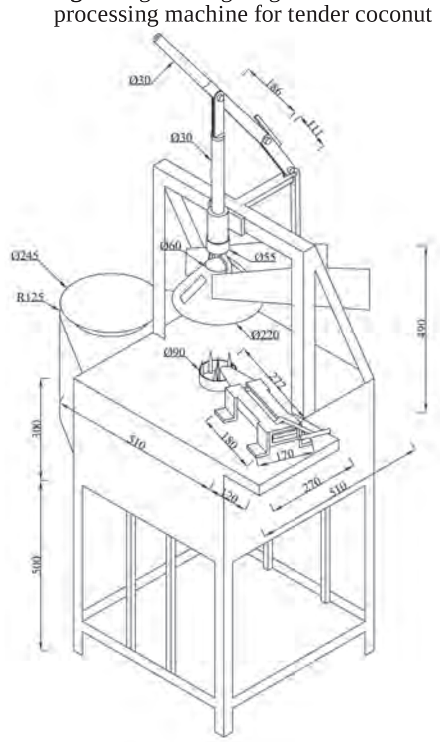
Speed of tender coconut holder = 450 rpm

Torque required = load × distance = 25.5 × 0.09 = 2.295 kg m

Hp required to operate tender coconut trimming machine = $(2\pi NT \times 2.295 \times 450) / 4500 = 1.44$ hp ≈ 1.5 hp

The machine was driven by a single phase electric motor. An AC motor of 1.5 hp with maximum speed of 1,425 rpm was used. However, around 400 to 450 rpm was normally applied for trimming the

Fig. 1 Engineering diagram of minimal processing machine for tender coconut



tender coconuts (Yahya and Mohd Zainal, 2014).

Calculation of diameter of pulley required to rotate tender coconut holder

$D_1 N_1 = D_2 N_2$

$N_1 / N_2 = D_2 / D_1$

Where,

D_1 = Driving pulley diameter (mm)

N_1 = Revolutions of driving pulley (rpm)

D_2 = Driven pulley diameter (mm)

N_2 = Revolutions of driven pulley (rpm)

$1425 / 450 = D_2 / 75$

Diameter of driven pulley = 237.5 mm

The standard size of pulley available in market is 250 mm. Hence, the bottom holder rotates at 428 rpm.

The bottom holder consisted of a circular platform in the middle and was surrounded by six peg teeth. The top holder comprised a handle, a spring arm, sliding slot and a cone. The spring arm with sliding slot was used to clamp the top portion of tender coconut based on the height of tender coconut. The function of cone was to tighten the top portion of tender coconut. A shoulder trimming knife attached on the cone to perform the trimming operation. According to Yahya and Mohd Zainal (2014), high carbon stainless steel blade was best to undertake the trimming process. Hence, we used high carbon stainless steel blade to trim the shoulder part and to cut the bottom portion of tender coconut. The most suitable angle of the shoulder trimming knive (60°) were determined according to the short trimming time, appearance and defect rate of the final trimmed tender coconut. The AC motor (1.5 hp, single phase, 1,425 rpm, 50 Hz, 220/230 V) was attached through V belt to drive the bottom holder. About 250 mm diameter pulley was used to reduce the rpm of 1,425 to 428.

Fig. 2 Performance evaluation of developed tender coconut trimming machine

Performance Evaluation

To evaluate the performance of the machine, WCT was chosen as the test variety. A total of 100 freshly harvested WCT coconuts were obtained from Farm Office, Central Plantation Crops Research Institute, Kasaragod. To evaluate the machine performance, 50 intact tender coconuts, which have uniform shape and size, were selected. The tender coconuts were trimmed continuously using the prototype minimal processing machine and the performance of the machine was evaluated. The production rate was calculated based on the finished produce capacity per hour whereas product quality was evaluated in terms of appearance of the final trimmed coconut.

A single operator was required to perform the trimming operation. Once the tender coconut was fitted with the holder, the operator starts the trimming process (**Fig. 2**). During the trimming process, the shoulder part of coconut trimmed first and then the high carbon stainless steel knife was used to penetrate the husk of the rotating tender coconut thus resulting in a cutting of bottom portion of the tender coconut. At the end of the trimming process, machine capacity was measured and percentage of defects was calculated using fallowing equations (Jarimopas and Ruttanadat, 2007).

Machine capacity = Number of trimmed tender coconuts / Time taken (1)

Percentage of defects = Number of broken product + Unsuccessful trim product / Total number of trimmed tender coconut (2)

Results and Discussion

Physical Properties of Tender Coconuts

The average height and diameter of intact tender coconut were 16.10 ± 0.87 cm and 12.51 ± 0.89 cm respectively (**Table 1**). The physical properties were the substantial data which determined the design of the tender coconut holder and trimming knife position and angle settings (Jarimopas et al., 2009). Generally, the upper husk thickness was higher than the bottom husk thickness of all samples. However, left and right sides husk thickness of the tender coconut vary and predominantly depended on the shape of the tender coconuts. Even wide variation

Table 1 Important physical properties of tender coconut (Variety - WCT)

Property	Mean ± SD	Maximum	Minimum
Tender coconut height	16.10 ± 0.87	17.2	14.2
Tender coconut circumference	39.29 ± 2.81	43.50	33.50
Tender coconut diameter	12.51 ± 0.89	13.85	10.67
Thickness of husk at top (stem to shell)	5.05 ± 0.36	5.8	4.4
Thickness of husk at right side	2.88 ± 0.23	3.2	2.4
Thickness of husk at left side	1.94 ± 0.30	2.6	1.6
Thickness of husk at bottom (shell to base)	3.20 ± 0.21	3.6	2.9

in tender coconuts shape, size and husk thickness exist within types and populations (Chan and Elevitch, 2006).

Performance Evaluation

The distance between top and bottom holder of tender coconut was designed according to the average height of WCT variety (160 mm). In order to hold the different variety of tender coconuts including Chowghat Orange Dwarf (COD), Chowghat Green Dwarf (CGD), Malayan Orange Dwarf (MOD) and Malayan Yellow Dwarf (MYD) an additional of 80 mm height was provided. Hence, the total clearance provided between top and bottom holder was 240 mm. Jarimopas et al., (2009) provided shoulder knife height of 180 mm for trimming of ‘‘Namhom’’ variety.

The average holding time taken by the operator to set the tender coconut was 11 s nut^{-1}. This was considered as time consuming as the operator had to manually positions the tender coconut. A total of 2% of the coconuts were damaged during the trimming process due to broken shell as a result of excessive trimming. The optimum operating conditions of the prototype machine were as follows: a) 428 rpm rotational speed; b) the trimming knives need to be changed or sharpened after 80 tender coconuts had been trimmed; c) average shoulder trimming time (machine) was about 9 s/nut; d) average bottom-up cutting time (manual) was about 3 s/nut. The total processing rate of this prototype machine was 160 nuts per hour including the holding, trimming and bottom cutting, which was almost four times faster than conventional manual trimming method. Moreover, the trimmed tender coconut produced by unskilled labors during manual trimming was not presentable due to uneven contour (Yahya and Mohd Zainal, 2014), whereas the developed machine managed to trim the tender coconut uniformly. A shoulder knife angle of 60o gave the best result in terms of final product quality. Jarimopas and Ruttanadat (2007) also found that shoulder knife angle between the knife and the XY-plane (61°) gave the best result in lathe cutting machine mechanism.

As shown in **Fig. 3**, the output of tender coconut using prototype machine had less protruding fibre and there were no marked differences in terms of appearance compared to manual trimming. In comparison with the previous work done by Jarimopas and Ruttanadat (2007) and Yahya and Mohd Zainal (2014), their prototype machine could trim 21 and 49 nuts per hour at 300 and 400 rpm rotation, respectively (**Table 2**).

Fig. 3 Trimmed tender coconut

Conclusion

A prototype of semi automatic tender coconut trimming machine was developed and evaluated. The mechanism used features a sharp inclined knife which operates in translation motion in a vertical plane to trim the fruit, which is clamped tightly and rotates about a vertical axis. Machine components include a main frame, a shoulder-trimming station, a base-cutting station, rotary base holder, and a power drive. Optimal settings included 428 rpm rotation of the trimmed fruit, and a shoulder knife angle of 60°. The cutting edge angles of the shoulder knives have proven to generate smooth trimming with less protruding fibres on the final product. The machine performance test indicated a trimming (shoulder) and

Table 2 Comparison of tender coconut trimming machines

Method	Mechanism	Capacity (nuts. h^{-1})	Product Quality	Operation	Reference
Manual trimming method using knife	Shearing and cutting	35	Shape is depends on skills of labour	Risky operation and time consuming process	Jarimopas et al. (2009)
Young coconut fruit trimming machine	Lathe cutting machine mechanism	21	Uniform shape and size	Safe and time consuming operation (knife angles 61° produced the least amount of fibers)	Jarimopas and Ruttanadat (2007)
Young coconut fruit opening machine	Shearing the rotating trimmed young coconut by a stationary knife	120	Uniform shape and size	Safe operation (An angle of 50° was considered the optimum)	Jarimopas and Kuson (2007)
Automatic trimming machine	Vertical trimming mechanism with 3 fruit holders	86	Uniform shape and size	Safe operation and knife changes were necessary after 50 trimming operations.	Jarimopas et al. (2009)
Young coconut shaping machine	lathe trimming mechanism	49	Uniform shape and size	Safe operation but the trimming knives need to be changed after 30 trimming operations	Yahya and Mohd Zainal (2014)

cutting of bottom portion of tender coconut with total holding and trimming process of about 160 nuts per hour. About 328.5 g of husk can be removed from each tender coconut by trimming operation. The trimming knives need to be sharpened or changed after 80 fruits had been trimmed. Grading and sorting of the tender coconuts are not necessary to perform trimming operation in the developed machine, because of the large clearance (240 mm) and sliding slot between holders. The developed machine could trim the tender coconuts at higher level of efficiency compared to the manual method.

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Agricultural Machinery Manufacturing Sector in Palestine - Reality and Challenges

by

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Abstract

This paper presents briefly the current status of agricultural machinery manufacturing sector in Palestine. The current agricultural mechanization status, workshops names, locations and activities are mentioned. In addition, companies dealing with agricultural mechanization are listed. The motivations for improving agricultural machinery manufacturing sector are also discussed. Some technical recommendations for improving this manufacturing capacity are provided.

Introduction

Agricultural plays a fundamental role in national economy and food security. There is diversity in Palestinian agricultural products due to climate condition, which is similar to the Mediterranean region, i.e. warm in summer and mild in winter. Nevertheless, it is characterized by geographical and terrain diversity, featuring the coastal plains (high fertility) and the mountains with varying altitudes ranging between 150-1,050 m that span from the north to the south of the country. There are also the eastern valley regions with suitable temperatures in winter that is a unique environment to produce vegetables and some of crops, e.g. grapes and citrus. Moreover, there are the internal plains, which are an important source of food for the production chain and self-sufficiency in Palestine, where a range of cereals and summer vegetables are cultivated. The diversity of the geology and climate in the country has given a unique historical advantage, a great botanical biodiversity, as well as a wide range of animal husbandry. The mountainous highlands have different soil depths and thickness, and they are generally rocky. Moreover, they have different rainfall rates, slopes and

Fig. 1 Mounted moldboard plow

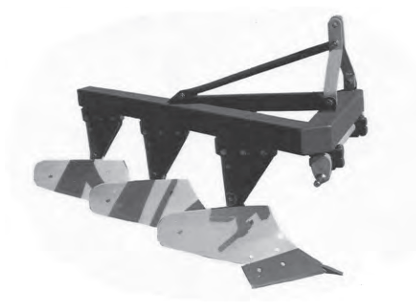

Fig. 2 Mounted chisel plow with nine blades

Fig. 3 Two-ton trailer that has a braking system and rear lights

Fig. 4 Nylon rolls spreading machine

Fig. 5 Mounted chisel plow with seven blades

Table 1 Percentages of cultivated land area of total area by governorate (Source: Palestinian Central Bureau of Statistics, 2015)

Region/Governorate	Cultivated land area (km^2)	Area (km^2)	Percentage of cultivated land area from total area (%)
Palestine	**931.5**	**15.5**	**6,020**
West Bank	**843.5**	**14.9**	**5,655**
Jenin	180.9	31.0	583
Tubas	52.1	13.0	402
Tulkarm	77.4	31.5	246
Nablus	130.4	21.6	605
Qalqiliya	48.9	29.5	166
Salfit	68.6	33.6	204
Ramallah & Al-Bireh	77.0	9.0	855
Jericho & Al-Aghwar	20.6	3.5	593
Jerusalem	16.0	4.6	345
Bethlehem	31.8	4.8	659
Hebron	139.8	14.0	997
Gaza Strip	**88.0**	**24.1**	**365**
North Gaza	16.6	27.2	61
Gaza	17.1	23.1	74
Dier Al-Balah	14.1	24.3	58
Khan Yunis	24.0	22.2	108
Rafah	16.2	25.3	64

heights.

According to the Palestinian Central Bureau of Statistics (PCBS), the percentages of cultivated land area of the total area by governorate are shown in **Table 1**.

Agricultural machinery plays a crucial role in supporting a sustainable agriculture system. The situation of agricultural machinery in Palestine was discussed in different studies (Abu-Khalaf, 1998; Abu-Khalaf and Natour, 2014). However, there is scarcity of information about the agricultural machinery industrial sector in Palestine. The types of agricultural machinery available usually affect farmers' choice for cultivated crops. Farmers usually avoid crops, e.g. lentils and sesame, due to the lack of harvesting machines, despite that these crops have economic potential. The available agricultural machines are generally old and have low efficiency. Moreover, availability of agricultural areas is limited, and there is a lack of integration in agricultural machinery and some necessary machines are often limited and insufficient.

The aim of this article is to discuss the situation of agricultural machinery manufacturing, i.e. current situation, products and future vision.

The Current Situation of the Agricultural Machinery Industry

This industry is taking place in modest workshops and factories with skilled workers. The goal of this business is to get a profit and to

Fig. 6 Mounted chisel plow with seven blades for deep plowing

Fig. 7 Two-ton trailer transport trailer (equipped with braking system, lighting and dumper mechanism)

Fig. 8 Four-ton trailer transport trailer (equipped with braking system, lighting and dumper mechanism)

Fig. 9 Trailed threshing machine powered by PTO

Fig. 10 Tailed waste water (sewage) suction tank

Fig. 11 Trailed water transporting tank

provide the needed machines for the agricultural market in the current circumstances. Some agricultural equipment locally produced are shown in **Figs. 1-11**.

The owners and the workers in workshops or factories are often qualified with modest local industrial experience. Most of their experience is inherited from parents and grandparents, who have an agricultural background. In addition, some workers obtained their experience by working and practicing in workshops and factories, or by attending agricultural and industrial vocational education.

There are several factories and workshops developing agricultural equipment in Palestine. **Table 2** shows some of these factories. Some of these workshops and factories have ISO certifications (regardless their levels), and they are able to cover the minimum requirements of the local mechanization market.

However, most of the workshops lack equipment for welding, turning, bending, cutting and metalworking, and this limits the ability to produce certain needed parts. The type of agricultural machinery produced depends on the quality and extent of the integration of equipment and processes in workshops or factories. Moreover, these manufacturers and technical workers still lack international expertise, exchange of information, technical skills, ideas and technology transfer.

There are some advanced agricultural machines that are still missing in the agricultural machinery market. For example, there is a lack of potatoes cultivation and seeds sowing machines for field crops. Also row crop planter and precision planting machines to grow some crops such as spinach, onions and carrots are also still missing, as well as harvesters, straw pressing and balers.

This problem can be overcome by importing. Nevertheless, the cost is relatively high even from some of the nearby countries such as Turkey, Egypt and Italy. **Table 3** shows a list of several dealers, who import some agricultural equipment brands.

Some of the workshops are located close to the countryside and have direct contact with farmers. While others (large factories), which have advanced manufacturing equipment and tools, are located inside or near cities, because they need high electric power loads to operate their machines (Farmers and owners of workshops and machine factories, 2018). This can be considered as an obstacle in the communication between farmers and manufacturers.

The local workshops in countryside carry out maintenance works (e.g. welding, fixing and adjusting) for many agricultural machines. Some of the workers in the field come up with innovation solutions to fit local models. These solutions are inspired by imported models. They also make some modifications to local machines, such as seeders and potatoes farming machines, and sometimes agricultural machinery used in agricultural research or different farming systems, for example minimum tillage or conservation agriculture, which is carried out with high expertise.

Future vision

The challenges for developing agricultural manufacturing sector can be summarized in: limited agricultural area in general, increasing of urbanization, growing demand for agricultural crops and food (food security) and lack of specialized engineering expertise in manufacturing of agricultural machinery.

In order to overcome these challenges, it is necessary to develop the agricultural machinery manufacturing sector in Palestine. There is a need to build a large scale of this sector, to cover all agricultural operations from preparing the land to the final product. This will contribute to food security needed to meet the demand of Palestinian people.

It can be suggested to take actions in these issues:

1. Optimal use of the available agricultural land: this will help in achieving the best utilization and introducing of modern agricultural machines and to obtain the highest quality and quantity production.

Table 2 Some of workshops and factories producing agricultural equipment in Palestine (Source: Farmers and owners of workshops and machine factories. 2018)

No.	Factory/workshop name	Area	Type of work
1	Haddad Factory	Jenin Industrial Zone - Jenin	Manufacturing of heavy agricultural machinery
2	Waleed Al - Ardaa Factory	Sanur - Jenin	Manufacturing of heavy agricultural machinery
3	Haj Raafat Factory	Arrabeh - Jenin	Manufacturing of heavy agricultural machinery
4	Al-Jalbouni Factory	Jenin Industrial Zone - Jenin	Manufacturing of heavy agricultural machinery
5	Fattair Workshop	Askar - Nablus	Manufacturing of agricultural machinery
6	Awni Al-Yamoni Workshop	Alyamoon - Jenin	Manufacturing and modification of agricultural machinery
7	Abu Ammar Al - Saadi (Fayek Al - Saadi)	Faqua - Jenin	Manufacturing and modification of agricultural machinery
8	Sulieman Al-Khateeb Workshop	Beit Qad - Jenin	Manufacturing and modification of agricultural machinery
9	Noor El Din Zakarneh Workshop	Deir Ghazala - Jenin	Manufacturing and modification of agricultural machinery
10	Abdel Baset M. Ameen Workshop	Faqua - Jenin	Manufacturing and modification of agricultural machinery
11	Muayad Hamdan Workshop	Arraba - Jenin	Manufacturing and modification of agricultural machinery

2. Applying Ministry of Agriculture's law that protects agricultural land with high agricultural value (fertility, availability of water, availability of large areas and suitability for agricultural mechanization) against urban expansion.
3. Introducing a specific law for large agricultural areas belonging to several owners, i.e. family inheritance, that enables treating large areas with high productivity as one unit in all agricultural operations, and then distributes the revenue between land owners. It is expected that this suggested law will attract the attention of legislators in the country. It is proposed that these large areas can be rented as one unit, to increase the quantity of agricultural production through mechanization, making it feasible to use agricultural machinery with high technology and performance in order to increase food production.
4. The development or importing small (compact) agricultural machines suitable for using in mountains area. This will contributes in encouraging owners of narrow areas to use them.
5. Carrying out more research and development (R&D) in agricultural machinery: the responsibility of investment in scientific research lies upon the Ministry of Agriculture and local universities to create research groups specialized in agricultural mechanization, and to lead this type of research that provide solutions for the problems facing agricultural mechanization in Palestine. It should be noted there is not any university offering agricultural engineering (i.e. mechanization) degree until now in Palestine (Kishida, 2017).
6. Development and supporting of existing factories and workshops: by expanding and increasing their production lines, to achieve greater diversity of products, meet the needs of local agricultural market and allow the exporting to neighboring countries, some of which have similar agricultural conditions to Palestine (e.g. Jordan, Yemen, Iraq, etc.). Also, providing technical support for working in this area to enhance their ability.

Conclusions

Agricultural mechanization sector is an important economical and agricultural factor that contributing significantly to improving food security. There are several skilled workshops who are working in this area and many locally made agricultural machines are produced in Palestine. However, this sector is facing several cumbersome factors limiting its expanding and technical development. There is a potential for improving this sector in several methods, e.g. providing technical training for workers and forming research groups in agricultural mechanization.

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Table 3 Authorized dealers and importers companies dealing (or used to deal) in agricultural machinery (Source: Importers of agricultural machinery, 2018)

No.	Company name / Brand	Area
1	Palestinian Tractors Co.	Industrial Area - Ramallah
2	Mr. Munther Salah / Thomson Authorized dealer	
3	Arab Investment Company / Al-Salam	Arrabeh - Jenin
4	New Holland Dealer	Industrial Zone - Jerusalem Road - Ramallah
5	Al-Junidi Company	Deir Sharaf Junction (Importer)
6	Al-Shafi' Company	Qabatiya - Jenin (Importer)

Modification of Rotary Unit of Power Tiller for *Biasi* (Interculture Operation) Rice Cultivation in Eastern India

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Abstract

Biasi (Beushening) is a traditional method of rice cultivation in rainfed risk prone areas of Eastern India. It is performed by animal drawn plough. The productivity of rice drastically reduces due to delayed *Biasi* operation by the use of animal drawn implement. Therefore, effort was made to develop a tiller-based rotary unit *Biasi* implement. Design considerations included soil condition during *Biasi* operation for Inceptisols and Vertisols soils [soil moisture content 60-65% (db) flooded, specific weight 14-18 kN/m^3 & Cone Index 48-54 kPa]. Accordingly, the cage wheel parameters (73 cm wheel diameter, 30 cm wheel width, 450 lug angle and 20 cm lug pitch) for *Biasi* operation were optimized. The rotary unit of a 13 hp power tiller was modified for *Biasi* operation. The rotary unit with 8 tines, 152 rpm rotor speed for 70 cm working width, blade distance 18 cm, length of soil slice 7-9 cm at 0.4 m/s forward speed was found suitable. The study revealed that the modified rotary unit of power tiller was found suitable for *Biasi* operation under rainfed rice cultivation because of their higher field capacity, better quality of work and higher benefit cost ratio as compare to traditional method.

Introduction

Biasi or Beushening method of rice cultivation is commonly used under highly variable climates and a poor resource base. In this system, rice seeds are broadcasted in dry or wet soils after normal field preparation. The rice plants as well as weeds grow simultaneously up to 30-40 days and thereafter shallow ploughing (single or cross) is done in presence of 5-10 cm of water in standing rice crop by animal drawn indigenous plough followed by manual weeding and gap filling. This operation is called *Biasi*. Under rainfed situation, *Biasi* is necessary because it helps to maintain the plant population, better plant growth and development of roots. In India, more than 1.5 million ha area of rice is under *Biasi* system of rice cultivation. Indian farmers, particularly the small and marginal, have been dependent on human and animal power for performing various farm operations. Animals meet the power requirements of small and marginal farms with associated limitations. The availability of draught animals power has come down from 0.133 kW/ha in 1971-72 to 0.094 kW/ha in 2012-13 (Mehta et al., 2014), whereas the share of tractors and power tillers has increased during the same period. Since the traditional method of *Biasi* gives low output resulting in delayed operation with high cost, mechanization of this particular operation through mechanical means plays a major role to enhance the field efficiency with minimum loss of time, energy and materials. Chandravanshi (1989) informed that direct seeded rainfed *Biasi* cultivation of rice occupies 87% of the rice area in Chhattisgarh. Fujisaka (1991), emphasized that *Biasi* (Beusani) is a common crop establishment practice in rainfed lowland rice. *Biasi* is used in dry seeded lowland fields to control weeds. *Biasi* reduces not only the weed population, but also the rice plants population. Kawade (2001) reported that farmers adopted *Biasi* method to minimize the labour requirement in weeding operation, and to enhance the plant growth. Mishra (2010) developed an animal drawn *Biasi* implement.

The field capacity (0.80 ha/day) was significantly high; which was more than four times than that of local *Biasi* plough, (0.18 ha/day). It was due to proper shape and arrangement of curved tines in the developed *Biasi* implement, which facilitated self-unclogging and minimized the time lost in removal of clogging. The draught requirement of developed *Biasi* implement was within the draught capacity of local bullocks (55-65 kgf).

As the traditional method of *Biasi* gives low output resulting in delayed operation with high cost, mechanization of this particular operation plays a major role to enhance the field efficiency with minimum loss of time, energy and materials. But its mechanization is not up to the level of farmers' expectations till date. So if an inter-cultural power operated implement can be introduced, it will reduce drudgery of farmers and also increase the yield due to timeliness of field operation. In order to bring down the area work pressure on animate power and to assess the possibility of mechanization of the *Biasi* operation, the power-operated *Biasi* implement was designed, developed and tested.

Material and Methods

To develop a rotary *Biasi* implement, various research findings on crop and soil parameters for *Biasi* operation available in literature were taken in to consideration, and are given in **Table 1**.

Table 1 Crop and soil parameters for *Biasi* operation

Particulars.	Value	Reference
Soil type	Loam, silt clay loam, clay loam	Verma et. al. (2006)
Field preparation	Two cross ploughing by tractor with cultivator depth of ploughing 8-12 cm, bulk density 1.3-1.4 Mg/m^3	Verma et. al. (2006)
Seed rate	80-100 kg/ha	Lakpale and Shrivastava (2012)
Time of *Biasi*	30-40 days after sowing	
Depth of water in the field at the time of *Biasi*	5-10 cm	
Distance of *Biasi* (distance between furrow)	20 ± 3 cm	Kawade (2001)
Depth of *Biasi*	6-8 cm	
Gap -filling process	Within 3 days of Biasi operation	Lakpale and Shrivastava (2012)
Plant population	125 plants/m^2	

Table 2 Technical specifications of power tiller

Specification	Value
Engine Type	Vertical Diesel Engine
Number of cylinder	1
Engine maximum power at 2400 rpm	13 hp
Engine maximum torque at 1900 rpm	4.2 kNm
Tilling Width	800 mm
Tilling Depth	160-220 mm
No. of Tynes	24
Gears	6 forward & 2 reverse
Rotational speed PTO shaft	540 rpm
Total weight	120 kg

Features of Power Tillers That Outfit the *Biasi* Operation

The reasons for taking power tiller to perform *Biasi* operation was its compact construction and good trafficability. The lightness and low centre of gravity facilitate easy operation, low sinkage on submerged soils and rare casualties. The narrow wheel tread enables them to go through the narrow patch in the country side. Due to absence of two front wheels and narrow wheel tread, power tillers have short turning radius. Thus less land was left untilled during field operations. Rotary type power tiller have been selected for *Biasi* operation because the reduction in traction demanded of power tiller driving wheels due to the ability of the soil working blades to provide some forward thrust (Benny et al., 1970).

Modification in Rotary Power Tiller Unit for *Biasi* Operation

Modification work in rotary power tiller units for *Biasi* rice cultivation was carried out during the year 2013 to 2015. During this period the cage wheel parameters such as type and number of blades, and their combinations were optimized through field experiments for *Biasi* operation. The technical specifications of the power tiller taken for development of rotary *Biasi* are given in **Table 2**.

Development of Rotary Unit for *Biasi* Operation

Various research findings supported to choose C-shaped rotary blades for *Biasi* operation because of hooking between the rotary blades and plants (low plant mortality), less torque and power consumption (unclogging in submerged rice field). Salokhe et al. (1993) studied the power requirement and puddling of a rotavator in wet clay soil. It was observed that rotor of C-shaped blades consumed less power than the rotor of L-shaped blades. Chertkiattipol et.al. (2007) explained that it was the phenomenon of reaction forces between soil and tillage tool that related to soil failure pattern. The torque acting on the rotary shaft of L- shaped blades was slightly higher than for the Japanese C-shaped blade. Design of rotary tiller blades depends on soil types, number of blades and working condition. The specific work carried out by a rotary tiller at each rotation of tillage blades was considered as the volume of tilt soil. Specific work can be calculated by following equation (Bernacki et al., 1972).

$A = A_O + A_B$ kg-m/dm^3 (1)

Where: A is specific work of rotor, A_O is static specific work, A_B is dynamic specific work. A_O and A_B were obtained by using following relationship:

$A_O = 0.1C_o\, k_o$ kg-m/dm^3 (2)

$A_B = 0.001a_u\, u^2$ kg-m/dm^3 (3)

$A_B = 0.001a_v\, v^2$ kg-m/dm^3 (4)

Where, C_0 is coefficient relative to the soil type (for clay loam $C_o = 2.5$) k_o is specific strength of soil (60 kg/dm^3), u is the tangential speed of the blades (m/s), v is tractor forward speed. au and av are dynamic coefficient and it can be calculated by

$a_v = a_u\,(u / v)^2$ (5)

$\lambda = u / v$

The static specific work A_o, which is related to the cutting soil slice must be greater than the specific work of rotary tine and of other passive tools, because each rotary unit is bound to cut a considerably greater part of the soil surface than the tine. The larger the slices, the lower the specific work, and other portion represents the dynamic specific work resulting from striking the soil by the blades while penetrating and from acceleration of the soil slices during the *Biasi* operation. Dynamic work determines the magnitude of power consumed by the rotary units.

Calculation of Performable Work of the Power Tiller for *Biasi* Operation (Ac)

The maximum performable work that can produced by the power tiller, was calculated by the following equation:

$A_C = (7.5\, N_c\, \eta_c\, \eta_z) / (V.a.b)$ (6)

Where: N_c is the power tiller (hp); η_c is the efficiency of power tiller for forward rotation of the rotary tiller shaft which is equal to 0.9, η_z is the coefficient of reservation of power tiller power (0.7-0.8); V is the forward speed of power tiller (m/s); a is rotary tiller working depth (dm); b is tiller working width (dm). The values of different parameters used in designing of rotor of *Biasi* implement and values of specific work of rotary tiller and maximum power tiller work for different values of V & λ are given in **Table 3** and **Table 4**.

Power Requirement of Rotary Power Unit

Drawbar power requirement of rotary tiller was calculated by the following equation:

$P = (V \times D_f) / 3.6$ (7)

Where; P is the drawbar power (kW); V is the forward speed (km/h); D_f drawbar force (kN).

$D_f = (D_r \times a \times b) / 1000$ (8)

$D_f = (25 \times 9.0 \times 70) / 1000$

$D_f = 15.6$ kN

$P = (2.16 \times 16.6) / 3.6$

$P = 9.36$ kW

Where; D_r is average rotary tiller specific resistance (kN) (21 N/cm^2 loam & silt-clay loam), 25 kN/cm^2 for clay soil. a & b are working depth and working width of *Biasi* operation respectively in cm.

Design of Rotary Tiller Blades

In this design, C-type rotary blades were considered. Design of rotary tiller blades depends on the parameter including: soil types, number of blades and working condition.

The number of rotary tiller flange (f) was calculated by the following equation:

$f = b / b_i = 70 / 18 = 3.88$ (9)

Where, b is the working width (cm) and b_i is the distance between the flange on the rotor (cm). Two blades were considered on each flange (z = 2). So total number of blades was obtained by following relationship:

$N = f \times z = 4 \times 2 = 8$. Arrangements of blades on rotory unit is shown in **Fig. 1**.

Diameter of Rotor

The factors that affect the rotor diameter include torque and bend-

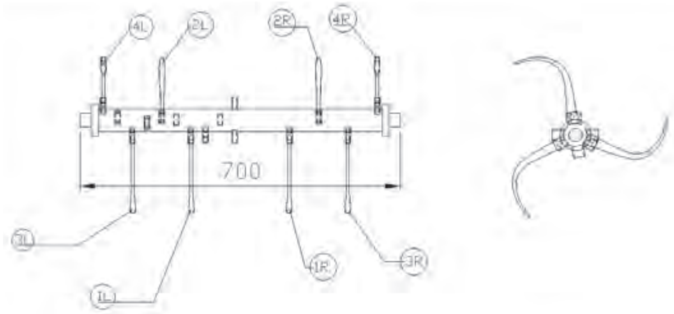
Table 3 Values of different parameters used in designing of rotor of *Biasi* implement

Particulars	Values
C_o, coefficient relative to the soil type	2.5
K_o, specific strength of soil (kg/dm^3)	60
A_u, dynamic coefficient	400
N_c, the power tiller (hp)	13
η_c, traction efficiency	0.9
η_z, coefficient of reservation of power tiller	0.8
a, rotary tiller working depth (dm)	0.982
S_y, Yield stress MPa	520
K, Coefficient of stress concentration	0.75
C_s, Reliability factor non rocky soil.	2
FS, factor of safety	2
V, m/s	0.2-0.6
R, cm	25
U_{min}, m/s	1.19

Table 4 Specific work of rotary tiller and maximum power tiller work for different values of V & λ

Velocity (m/s)	λ	$a_v = a_u(u/v)^2$	$A_B = 0.001a_v v^2$	$A_o = 0.1C_o k_o$	$A = A_o + A_B$ kg/dm^3	A_c kg. m/dm^3
0.2	22	193,600	7.74	15	22.74	51.06
0.3	18	129,600	11.66	15	26.66	34.04
0.4	10	40,000	6.4	15	21.40	25.53
0.5	4	6,400	1.6	15	16.60	20.42
0.6	2	1,600	0.576	15	15.57	17.02

Fig. 1 Arrangements of blades on rotory unit



ing moment. Torque is an important factor that was affected by the dimension of rotor diameter. For safe design of rotor diameter, careful selection of rotor axle and torque was important. Diameter was calculated by the following relationship:

$$d = \sqrt[3]{[(16\ M_s) / \tau\pi]} \quad (10)$$

Where; d is the diameter of the rotor (cm); M_s is maximum torque at rotor axle in N.cm; τ is allowable shear stress at the rotor axle N/cm^2.

$$\tau = (0.577\ K\ S_y) / FS \quad (11)$$

$$\tau = (0.577 \times 0.75 \times 520) / 2$$

$$\tau = 112.5\ N/cm^2$$

Where; K is coefficient of stress concentration, S_y is yield stress (MPa) based on material of rotor; FS is factor of safety.

$$M_s = K_s \times R \quad (12)$$

$$M_s = 294.95\ kN.m$$

Where, R is the rotor radius (mm)

$$K_s = (75\ C_s\ N_c\ \eta_c\ \eta_z) / u_{min} \quad (13)$$

$$K_s = 1179.83\ kg$$

Where, K_s is the maximum tangential force at the rotor axle (kg); u is minimum linear velocity of rotor (m/s); C_s is reliability factor taken as 1.5 for non-rocky soil and 2 for rocky soil. By using the equations (11-13) the optimal diameter of rotor was calculated as 5.03 cm.

Development of Rotary Unit of Power Tiller for *Biasi* Operation

In the rotary unit, power was transferred to the tiller from the engine via the power-take-off drive. A shaft containing blades was located at 90° to the line of travel and rotated in the same direction as the forward travel of the power tiller. Since the shaft turns at a rate that was considerably faster than the corresponding power tiller speed, soil churning was accomplished. The proper selection of forward speed was dependent on the tangential speed of the blades (that is a function of rotational speed of rotor) and the length of sliced soil. The tangential speed of the blades (u), the rotational speed of the rotor (N), and the length of sliced soil (L) could be obtained by the following equations:

$$U = (2\ \pi nR) / 6000 \quad (14)$$

$$n = 6000\lambda v / 2\pi R \quad (15)$$

$$L = 2\pi R / \lambda Z \quad (16)$$

Where,

R = rotor radius (cm),

v = forward speed (m/s),

Z = number of blades on each side of the rotor flanges.

In this design, two blades were considered on each side of the flanges (Z = 2). Lakpale and Shrivastava (2012) reported that for better rice yield, the *Biasi* operation can be performed at the working width of 20 ± 3 cm and 8-10 cm depth. Therefore, the design parameters considered for the development of rotary *Biasi* unit were a definite line of spacing and 10 cm depth of operation. Matyashin (1968) suggested that the radius of rotor for rotary tillers selected should be greater than the working depth. Considering these explanations, a 50 cm diameter was determined to be appropriate for the rotary tiller rotor. By incorporating the selected values for the rotor diameter in the equations (15) and (16) become:

$$n = 6000\lambda v / 2\pi R = 38.21\lambda v \quad (17)$$

$$L = (2\pi \times 25) / (\lambda \times 2) = 25\pi / \lambda \quad (18)$$

The possible selections for the rotary tiller working width (b), forward speed (v) and rotational speed of rotor (n) are presented in **Table 5** and **Table 6**; which were obtained through equations (17) and (18). For

Table 5 Possible selections for the rotary tiller working width, forward speed and rotational speed of rotor

S. No.	Working width (cm)	Forward speed (m/s)	Rotor speed (rpm)	λ	Length of sliced soil (L) (cm)
1	60	0.2	168.08	22	3.57
2		0.3	206.28	18	4.36
3		0.4	152.80	10	7.85
4		0.5	76.40	4	19.63
5		0.6	45.84	2	39.27
6	70	0.2	168.08	22	3.57
7		0.3	183.36	16	4.91
8		0.4	122.24	8	9.82
9		0.5	57.30	3	26.18

Table 6 Modification in rotary unit of Power Tiller for *Biasi* operation

Particulars	General Operation	Biasi operation
Blades (tines) numbers	20 numbers	6/8 numbers
Blades (tines) Distance	2-4 cm	18 cm
Gear	I Forward heavy	I Forward heavy
RPM	2000-1800	2000-1800
Blade specification	C-shaped blade (4.5-cm tilling blade width), Length 28 cm, Heat -Treatment process	15° lengthwise slice angle, 2.5-cm tilling blade width Length 32 cm, Heat -Treatment process

each of the selected working width, the closest values of the rotary tiller specific work of the power tiller were determined at each of the forward speed. Then, the corresponding values of λ for each forward speed were determined to calculate the rotor speed and the length of soil sliced. By selecting the rotary tiller specific work and the performable work of the power tiller close together at each of the forward speeds, an appropriate conformity will be continued between the rotary tiller and power tiller. Considering the suitable domain obtained for the rotor speed, the length of sliced soil and the forward speed, at the working width of 70 cm, this width was selected as a proper working width for the power tiller (**Table 5**).

Considering the results presented in **Tables 1** and **5**, it becomes evident that the selected power tiller for this design only at the gear one can supply a rotary tiller with the working width of 70 cm at a working depth of 9.82 cm. After specifying the appropriate working width for the power tiller, the length of sliced soil, the rotational speed of the rotor and the tangential speed of the blades should be calculated at the selected gear (the forward speed of 0.40 m/s). Before performing the mentioned calculations, the appropriate value of λ, proportional to the selected forward speed for the power tiller should be obtained. For this purpose, the specific work of the rotary tiller and the performable work of the power tiller should be equal together. Therefore, gear no. 1 (forward heavy) was selected for *Biasi* operation.

Gear No.: 1 Forward heavy → $v = 0.4$ m/s, $\lambda = 8$

By obtained value for λ at the equations (14), (15) and (16) we will have, $L = 9.82$ cm, $n = 122$ rpm. On the basis of these calculations the following modification were made in the rotary unit of tiller **Table 6** and **Figs. 2** and **3**.

Results and Discussion

Table 7 Impact of different implements on various parameters affecting rice production (Date of *Biasi*, DAS-35)

Parameters	*Biasi* implement		
	Animal drawn		Power tiller with modified rotary tines
	Local Deshi plough	Biasi plough	
Seed rate kg/ha Variety-Mahamaya	100	100	100
Distance of *Biasi*	20 ± 3 cm	20 ± 3 cm	18 ± 3 cm
Plant mortality, %	22-37	10 -18	21-29
Plant population/ m^2	114	129	118
Weeding efficiency, %	47	62	60
Effective tillers, No./m^2	222-298	324-419	301-412
Field capacity, ha/h	0.034	0.127	0.215

Comparative Performance of *Biasi* Implement

The comparative performance of the improved implement over that of traditional plough (**Fig. 4**) was assessed in terms of operational indicators as well as yield indicators.

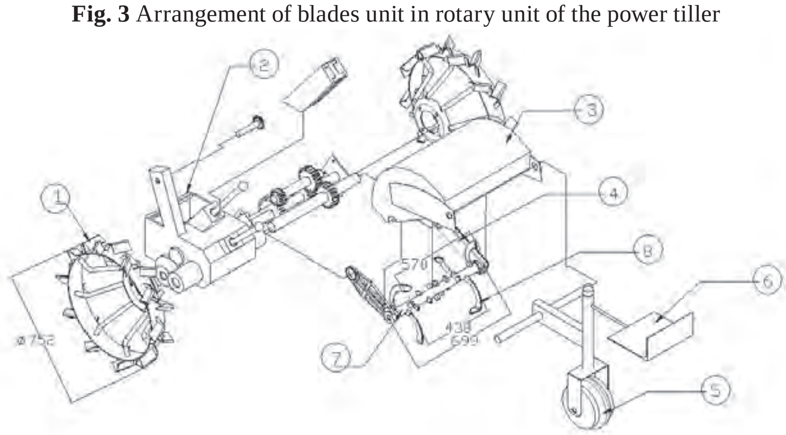
Operational Indicators

The average performance of different *Biasi* implement ploughs in terms of operational indicators such as field capacity, distance of *Biasi*, plant mortality and weeding efficiency is shown in **Table 7**. Study indicates that the improved *Biasi* implement offered a higher field capacity as compared to traditional method. The quality of work indicators observed were: plant mortality (%), weeding efficiency %, plant population/ m^2, and effective tillers No./m^2.

The developed *Biasi* implement has shown significant increase in

Fig. 2 Modification of number of blades, size and spacing on rotary unit

Fig. 3 Arrangement of blades unit in rotary unit of the power tiller



1. Cage wheel, 2. Gear Box, 3. Mud cover, 4. Power transmission chain, 5. Depth control cum *Biasi* wheel, 6. Operator seat, 7. Shaft, 8. C- type tines

Table 8 Effect of different *Biasi* implements/ploughs on rice yield (q/ha)

Biasi Plough	Yield, qha^{-1}	% yield increase*	B:C ratio
Trifal Biasi plough	39.2	21^{**}	2.11^{**}
Power tiller operated modified rotary Biasi implement	38.2	18^{**}	2.34^{**}
Traditional (Control)	32.4	-	1.65

* As compared to using traditional Biasi plough, ** - significant

grain yield due to minimized plant mortality and better root stimulation that led to better crop growth and development of yield attributes. The benefit-to-cost (B:C) ratio was highest for power tiller operated modified rotary tines (2.34) and the lowest for traditional method (1.65). The study clearly indicates that by mechanization, timeliness of operation can be achieved and the cost of production is reduced, and eventually significant increase in yield is observed.

Yield Indicators

The comparative performance of different *Biasi* implements in terms of rice yield is shown in **Table 8**. All the implements used for *Biasi* operation performed better with regards to grain yield of rice as compared to that of the traditional plough used for *Biasi*. The highest grain yield was recorded in Trifal *Biasi* plough followed by power operated 5 tines plough and power tiller operated modified rotary *Biasi* implement. The lowest yield was recorded in traditional plough which served as control. The rice grain yield was recorded as 39.2 qha^{-1} (*Trifal*), 36.2 qha^{-1} (power tiller operated modified rotary *Biasi* implement), and 32.4 qha^{-1} (traditional). Significantly highest rice yield over traditional plough was obtained in Trifal *Biasi* plough (21%), followed by power tiller operated modified rotary *Biasi* implement power tiller operated modified rotary *Biasi* implement (17.9%).

Conclusion

The study revealed that the mechanized *Biasi* implement could be adopted under rainfed *Biasi* rice cultivation. The improved *Biasi* implement was found suitable because of its higher field capacity, better quality of work and better benefit-cost ratio.

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Fig. 4 Operation with different *Biasi* implements

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Design of Rotary Assisted Broad Bed Former-cum-Seeder for Vertisols

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Abstract

A tractor operated rotary assisted broad bed former-cum-seeder was designed and developed at ICAR-Central Institute of Agricultural Engineering, Bhopal, India. The basic aim of this development was to performed rotary tillage followed by broad bed formation, sowing and reshaping of bed by using dumbbell shape re-shaper. Performance of developed machine was evaluated and compared with flatbed sowing for soybean and wheat crops at ten randomly selected farmer's field in vertisols. The field capacity and fuel consumption of developed machine were found as 0.32 ha/h and 15 L/ha respectively. The grain yield under broad bed seeded soybean was found 0.96 t/ha, however, conventionally sown soybean yielded 0.71 t/ha. Thus, the yield gain in soybean crop under broad bed technology was found 35.95%. The yield of broad bed and conventionally sown wheat, were found 4.99 and 4.15 t/ha, respectively. Therefore, grain yield increased by 20% in wheat crop due to the sowing with developed machine under broad bed condition. This machine saved 17.3% cost of operation as compared to the traditional flatbed sowing.

Keywords: Broad bed former, Vertisols, Rolling type bed shaper and Seeder.

Introduction

Vertisols are potentially productive soils within the dry semi-arid regions of central India. These soils have high water holding capacity, low water infiltration, high incidence of inundation, accelerated runoff and soil erosion during high rainfall year and drought stress during the low rainfall year. Consequently, crop yields on vertisols using traditional systems of management are low (Lal, 1995). Vertisols of the central part of the country have a fairly high potential for crop production when improved soil and water conservation practices are adopted. The improved technique with engineering interventions to drain excess water can help to increase the productivity of vertisols and allow farmers to harness the maximum crop productivity.

Broad bed and furrow (BBF) crop production system has been developed as a measure for water conservation in vertisol, to deal with waterlogging and improving soil structure. BBF technique requires less quantity of water by crops. It drains out the excess amount of water automatically which is not possible on flat beds. Ghani et al. (2007) has found 36% water saving for broad beds, about 10 % for narrow beds and grain yield increase of about 6% for wheat crop, 33% of maize crop. Thus, constructing broad beds and furrows has been a new idea which draws soil from the furrows on either side of the bed and thrown on top of the bed. It involves making beds of height 150 to 300 mm (Gupta and Undadi, 1994) and width of 1.7 to 2.0 m alternatively to allow drainage of excess water (Astatke et al., 2002).

Wheat has traditionally been planted on flat beds either by drilling closed spaced rows 100-300 mm apart or broadcasting and then incorporating it by means of shallow tillage operation (Sayre and Moreno-Ramos, 1997). Sowing on broad beds has improved yield, increased fertilizer and irrigation use efficiency, reduced weed incidence, facilitate better field management by providing passage to mobility in cropped field and save seed, fertilizer and irrigation water (Mandal et al., 2013; Rao et al., 2015; Shrivastava et al., 2017). Also, broad bed seeded crops cope up with excess and continuous rainfall conditions. The objective of the present study was to investigate the sowing of soybean and wheat crops on broad bed and compared to the conventional method of sowing on flat surface at farmers field. Thus in view of climate worthiness of the bed seeding system, rotary assisted broad bed former-cum-seeder has been designed and developed by ICAR-CIAE, Bhopal. Rolling type bed shaping system enables reshaper to make better quality beds even in poor tilled soil. The paper presents details of the developed machine and its performance results compared to the commercial bed planter and flatbed method of sowing for soybean and wheat crops.

Materials and Methods

Design Consideration

Design of the rotary assisted bed former cum seeder mainly consist of design of seed box, design of fertilizer box, design of ridger bottom, design of frame and design of bed shaper cum ground wheel. Technical specifications of the developed rotary assisted broad bed former-cum-seeder are shown in **Table 1**.

Drilling Unit: Five row drilling unit was used for fertilizer and seed placement on broad bed. Main components of the drilling unit were seed box, fertilizer box, fluted roller metering mechanism, frame, rate controller, furrow opener and seed and fertilizer tubes. The frame was made of mild steel square box of 50 × 50 × 5 mm. Length and width of the frame were 2000 and 700 mm. The frame was fabricated having three members i.e. front, middle and rear. Rotary unit was mounted on front member. The middle member of frame was used to stagger the three tynes and two ridgers. While the rest of the tynes and bed shaper unit were attached to the rear member. The frame of seed drill is subjected to both torsion and bending moment due to the horizontal and vertical force acting on the tynes. Tynes were bolted to the frame with the help of clamp to provide easy adjustment of tynes to meet the suitable spacing. Seed and fertilizer box was mounted on the main frame. They were made of mild steel sheet. It was designed for the seed rate of 100 kg/ha. Two seed rate controllers were fitted on the box for seed and fertilizer, separately.

Design of Seed Box: Among the seed used for owing by the seed drill, Wheat has the maximum seed rate 70-100 kg/ha. Therefore the seed drill may be designed for the seed application rate of 100 kg per ha. The effective field capacity of machine was calculated by following formula:

Effective field capacity of drill = $(W \times S) / 10 \times \eta$

Where, W is the working width of the machine (m), S is the operating speed of the machine (km/h) and η is the field efficiency (%).

The effective field capacity of seed drill was calculated as 0.42 ha/h considering the speed of operation of 4 km/h and field efficiency of 70%. The design of the seed box with capacity was calculated on the basis of refilling of seed requirement after 1 hour.

Therefore, the weight of seed to be used in 2 h = seed rate (kg/ha) × area covered /h × time = 100 × 0.42 × 1 = 42 kg

Now, volume of seed box = Weight of seed / Bulk density of seed = 42 / 800 = 0.052 m^3

Consider spillage losses of 10%. Therefore, total volume of seed drill is calculated as:

Volume of seed box (Vs) = 0.052

Table 1 Technical specification of rotary assisted bed former cum seeder

Particulars	Description
Dimension of machine	1760 × 2600 × 1160 mm (Length × Width × Height)
Number of furrow opener	5
Type of furrow opener	Shovel type
Seed metering mechanism	Sliding fluted roller type
Weight of machine	550 kg
Number of ridger	2
Width of furrow	80 mm
Dimension of bed	Top width: 1200 mm and bed height: 200 mm
Row to Row spacing	Adjustable
Plant to Plant spacing	Adjustable
Fertilizer attachment	Sliding fluted roller type with metering mechanism
Metering unit drive	Drive to the metering unit is provided from rear mounted two rolling dump bell with spring loaded chain
Ground Wheel	Two 517 mm diameter spiked roller with spring to maintain contact with ground.
Hitching	3 point linkage is provided
Shaper	Rolling dump bell is provided for seed covering
Power Transmission	Chain and sprockets
Required tractor power (hp)	With rotavator: 60-70 hp (44.74-52.2 kW) Without rotavator: 40 hp (29.83 kW)

$+ 0.0052 = 0.058\ m^3$

According to geometry of seed box, let the seed box is of trapezoidal section, total seed box was divided into two sections. The volume of seed box (Vs) is given by:

$(V_s) = [(a + b) / 2] \times h \times l_b$

Where, Vs is the volume of seed box having trapezoidal section, a is the bottom width of seed box, b is the top width of seed box, h is the height of seed box and lb is the length of seed box

Then, $(V_s)_1 = [(0.186 + 0.190) / 2] \times 0.106 \times 1.596 = 0.031\ m^3$

$(V_s)_2 = [(0.190 + 0.160) / 2] \times 0.100 \times 1.596 = 0.027\ m^3$

Total volume of seed box $= 0.031 + 0.027 = 0.058\ m^3$

The thickness of seed box (t_s) is given by:

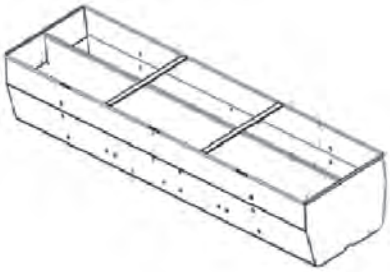
$t_s = \sqrt[3]{(3 \times \rho \times a^2 \times h^2) / (4 \times a \times b_s)}$

Where, t_s is thickness of seed box (cm), ρ is the bulk density (kg/cm^3), a is the bottom width of seed box (cm), h is the height of seed box (cm), bs is the bending stress in kg/cm^2 (1000 kg/cm^2), Therefore, $t_s = \sqrt[3]{(3 \times 0.008 \times 16^2 \times 10^2) / (4 \times 16 \times 1000)} = 0.132$ cm

The thickness of seed box was calculated as 1.32 mm (say 1.5 mm).

Design of Fertilizer Box: Design of fertilizer box was same as the seed box. All the designing parameter considered for fertilizer box was similar to the design of seed box. The volume of fertilizer box is 0.058 m^3. Therefore, the total volume of seed and fertilizer box was calculated as 0.11 m^3. The drawing of the seed and fertilizer box is shown in **Fig. 1**.

Fig. 1 Drawing of seed and fertilizer box



Ridger Unit: The ridger unit consisted of mainly shank, shovel, and adjustable wings (**Fig. 2**). It was designed for handling the volume of 1200 mm width of soil. The shank was made of MS flat. The dimension of shank was 650 × 50 × 25 mm. The shovel was attached to the shank so the overall length of tyne from tip of shovel to the upper end of shank was 675 mm. The lower portion of the shank was curved with radius of curvature of 150 mm. It consisted of two holes of 12 mm diameter at its bottom for mounting shovel. Commercially available reversible shovel of 250 × 50 mm was used. It was made from medium carbon steel. Width and length of shovel were 50 mm and 250 mm, respectively. The leading edge of the opener was a sharp-pointed triangle. The boot wedge and rake angle were 45° and 40°, respectively. The shovel was attached to shank with nut and bolt for easy replacement. Two ridgers were used in the developed machine. Wings of the ridger have provision to change the width of furrow according to bed size and shape in the range of 200-400 mm.

Design of Ridger Bottom: the ridger bottom was made by joining the two wings to the shank. Calculations for designing the ridger bottom ares given below. The draft on ridger bottom shank (D_f) calculated by following formula:

$D_f = K \times w \times d$

Where, D_f is draft on ridger bottom (kN), k is specific soil resistance (kPa), w is the width of furrow opener (mm), d is the depth of sowing (mm). The furrow slice cut by the furrow bottom was in trapezoidal shape. Assuming the specific soil resistance of 39.2 kPa for vertisol soil, the draft of ridger bottom was calculated as:

$D_f = 39.2 \times (600 + 300) / 2 \times 150 = 2.7$ kN

Now, for mild steel tynes factor of safety can be taken as 1. So, the design draft of furrow opener would be 2.70 kN.

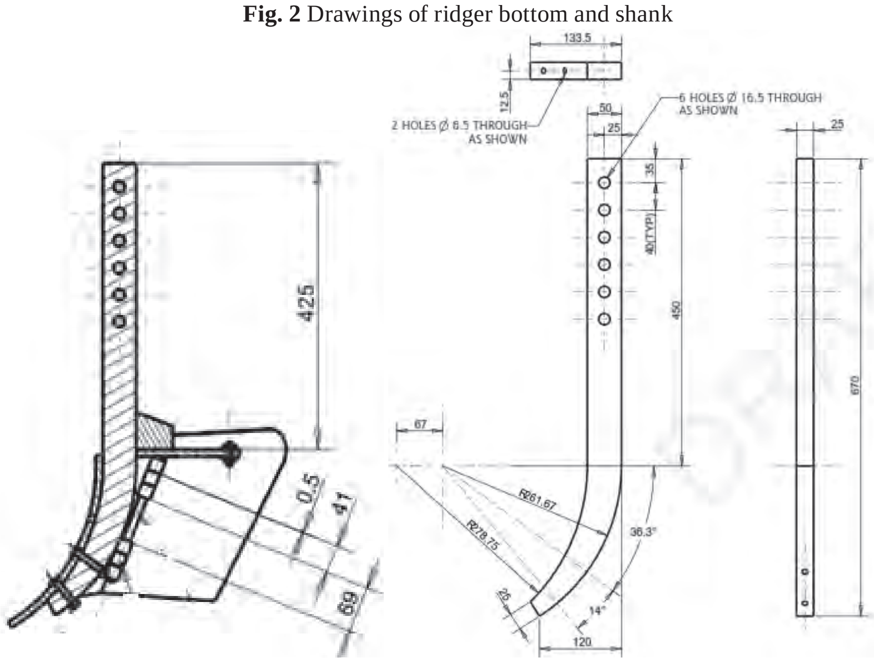
Considering the ridger bottom tyne as a cantilever beam of 500 mm size fixed to the frame at one end the maximum bending moment in the tyne is given by

M = Design Draft (kN) × Beam span (m)

Then, M = 2.7 kN × 0.50 m = 1.35 kN-m = 1350000 N-mm

Now, section modulus of tyne Z is calculated as:

Fig. 2 Drawings of ridger bottom and shank



$\sigma_b = MC / I = M / Z$ (as $Z = I / C$)

Where, σ_b is bending stress in tyne, N/mm^2 (for mild steel 100 N / mm^2), M is bending moment in tyne (N-mm), C is distance from neutral axis to the point at which stress is calculated (mm), I is polar moment of inertia of rectangular section (mm^4) and Z is section modules of tyne (m^3).

Also, for rectangular section

$Z = t \times b^2 / 6$

The ratio between the thickness and width (t:b) is 1:2.

Therefore, $Z = t \times (2t)^2 / 6$

Also, $Z = M/ \sigma_b = 1350000/100$

$13500 = 4t^3 / 6$

$t^3 = 20250$

t = 27 mm (say 25 mm)

b = 25 × 2 = 50 mm

Therefore, cross section of the tyne was 25 × 50 mm. Therefore, the mild steel flat of 25 × 50 mm was selected for the tyne of ridger seeder. The drawing of the ridger bottom is shown in **Fig. 2**.

Design of Frame: The frame was fabricated having five members to stagger the five furrow openers on seed drill. The frame of the seed drill was considered as simply supported beam with point load. The maximum banding moment was acting at the centre of the beam at point. Maximum bending moment (M) is given by:

$M = (800 \times 3p) - (1000 \times p) - (600 \times p) - (200 \times p)$, Where, p is the maximum load per tyne.

Therefore, $M = (2400 - 1000 - 600 - 200) \times 60 \times 9.8 = 352800$ N.mm

Again,

Torque produced on toolbar (T) = maximum bending moment × ground clearance × number of furrow openers

Then, $T = 60 \times 9.8 \times 350 \times 6 = 1234800$ N.mm

The equivalent torque (Te) is given by:

$T_e = \sqrt{(M^2 + T^2)}$

So, $T_e = \sqrt{352800^2 + 1234800^2} = 1284211$ N.mm

Again equivalent torque is given by:

$S_s / Y = T_e / I$

Or $I / Y = T_e / S_s$

And $I / Y = d^3 / 6$

Where, S_s is shear stress, Te is equivalent torque, I is moment of inertia, y is the distance from the neutral axis to the point at which stress is determined and d is size of square rod.

So, $d^3 = 1284211 / 60 \times 6$

d = 50.4 or say 50 mm

Therefore, the square box of 50 mm was used for the fabrication of frame of the machine. The drawing of frame of seed drill is shown in **Fig. 3**.

Rolling type Bed Shaper: Broad bed seeder consists of a rolling type bed shaping system which creates an intact and smooth bed with a proper height. The design of bed shaper was done for achieving the bed top width, bottom width and height of 1200, 1500 and 200 mm, respectively. The rolling type bed shaping system consisted of shaft, dumbbell shape re-shaper, levelling pipe and pegs. The shaft was made of MS rod having length and diameter of 2000 and 25 mm, respectively. One end of the shaft was connected to the sprocket of 14 teeth. It is rolling type bed shaping system which has also been used as power wheel which is a special feature of this seeder. Dumbbell shape re-shaper is also used as power source for metering device. The distance between levelling pipe and the edge of the dumbbell was kept 200 mm for making 200 mm bed height. Twelve numbers of triangular shape mild steel pegs of 25 mm width were kept for providing smooth rotation to drilling unit. Due to rolling action of bed shaper, the clods get burrowed in the bed and achieve the desired geometry of broad bed.

Design of Bed Shaper-cum-Ground Wheel: The design of bed shaper was done according to size of bed and as per tractor wheel base. The beds were in size of 1200 mm of top width, 1500 mm of bottom width and 200 mm height. The distance between bed to leveling pipe and edge of the shaper was kept 200 mm for making 200 mm bed height and pegs were kept 25 mm for providing smooth rotation to metering device. Due to rolling action of bed shaper the clods get burrowed in the bed which is desirable for intact shape of bed and better look. The drawing of bed shaper-cum-ground wheel is given in **Fig. 4**.

Field Testing of Developed Machine

Performance evaluation of developed machine was carried out at ten randomly selected farmer's field from *Kachhiberkheda* village in Bhopal district of Madhya Pradesh, India. The soil type was vertisols (12.6% sand, 32.7% silt and 54.7% clay) and bulk density varying from

Fig. 3 Drawing of frame of machine

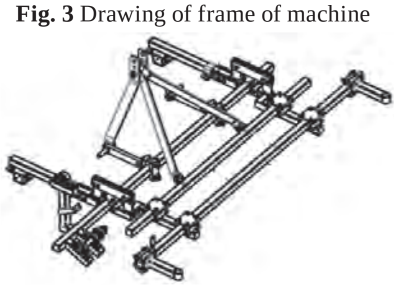
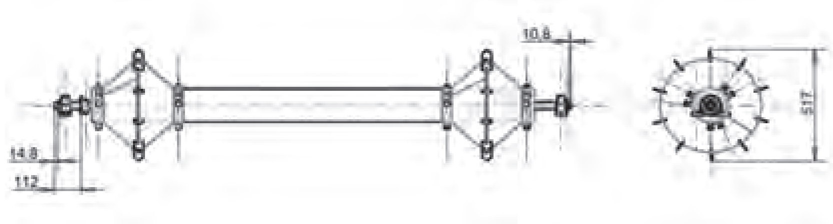


Fig. 4 Drawing of bed shaper-cum-ground wheel



1.21 to 1.25 g/cc. The experiment was designed in two blocks i.e. rotary assisted bed former-cum-seeder (M1) and flatbed sowing (M2) with ten replications. Soybean crop was sown in last week of June and harvested in the first week of October and wheat crop was sown in Mid-November and harvested in the first week of April. Soybean (80 kg/ha) and wheat (100 kg/ha) crop were sown with the developed machine on broad bed at row to row spacing of 300 and 200 mm, respectively. The shape of the bed was trapezoidal having top width and height 1200 and 150 mm, respectively. The yield of wheat was estimated at different random locations from 1 m^2 area. The difference in the grain yield, cost of cultivation, net income and benefit-cost (B:C) ratio were analyzed using random block design (RBD) at 5% level of significance ($P < 0.05$). **Fig. 5** shows the operation of developed rotary assisted bed former cum seeder in field.

Results and Discussion

Field Performance

The performance parameter of rotary assisted bed former-cum-seeder and flatbed sowing method is shown in **Table 2**. Total time required for seed bed preparation and sowing operation with rotary assisted broad bed former-cum-seeder was 2.85 h/ha. The field capacity of rotary assisted bed former-cum-seeder and flatbed sowing machine was 0.35and 0.36 ha/h at speed of operation of 3 km/h, respectively. The rotary assisted bed former-cum-seeder consumed 15 L/ha diesel for bed making and simultaneously sowing operation. This machine saved 17.3% cost of operation compared to the flat bed sowing. Rolling type bed shaping system as a power wheel is a special feature of this seeder. Due to rolling action of bed shaper, bigger size clods do not get exposed and a smooth shape of bed was formed.

Effect of treatments on grain yield (q/ha), cost of cultivation ($/ha), net income ($/ha) and B:C ratio is presented in **Table 3**. The difference in grain yield between both the treatments was found to be significant ($P < 0.01$). Higher soybean yield was obtained in treatment M1 (0.96 t/ha) as compared to M2 (0.71 t/ha). However, broad bed shown soybean crop yielded more due to well-aerated root zone in broad bed condition. A similar

Table 2 Performance parameter of rotary assisted bed former-cum-seeder and flatbed sowing

Particulars	Rotary assisted broad bed former-cum-seeder	Flatbed sowing
Type of soil	Vertisol (12.6% sand, 32.7 silt and 54.7% clay)	Vertisol (12.6% sand, 32.7 silt and 54.7% clay)
Crop	Soybean (JS-9560) and wheat (HI-1544)	Soybean (JS-9560) and wheat (HI-1544)
Row spacing, mm	300 (for soybean) and 200 (for wheat)	300 (for soybean) and 225 (for wheat)
Seed rate, kg/ha	60 (for soybean) 80-100 (for soybean)	80-100 (for soybean) 100-120 (for wheat)
Forward speed, km/h	3	3
Width of coverage, m	1.6	1.6
No. of rows	4 (for soybean) 5 (for wheat)	5 (for soybean) 6 (for wheat)
Depth of seed placement, mm	40-60	50-70
Field capacity, ha/h	0.35	0.36
Inter row variation in field for seed, %	5.0 ± 3.5	6.5 ± 4.5
Inter row variation in field for fertilizer, %	7.4 ± 4.6	6.5 ± 5.5
Fuel consumption, L/ha	15	12.6

Fig. 5 Developed rotary assisted bed former cum seeder

Table 3 Economic analysis of soybean and wheat crop at farmer's field

Treatment	Soybean				Wheat			
	Grain yield (t/ha)	Cost of cultivation ($/ha)	Net income ($/ha)	B:C ratio	Grain yield (t/ha)	Cost of cultivation ($/ha)	Net income ($/ha)	B:C ratio
Rotary assisted bed former-cum-seeder (M1)	0.96^A	221.81^B	235.65^A	1.05^A	4.86^A	342.53^B	762.48^A	2.23^A
Flatbed Seeding (M2)	0.71^B	242.83^A	98.13^B	0.40^B	2.74^B	388.4^A	194.28^B	0.51^B
Mean	0.84	232.33	166.89	0.72	3.80	365.46	478.39	1.37
p-Value	<.0001	<.0001	<.0001	<.0001	<.0001	0.001	<.0001	<.0001
CV (%)	5.49	2.05	12.05	10.13	8.82	6.08	20.83	21.58

trend was found in wheat crop also. Higher net returns in treatment M1 for soybean crop (235.65 $/ha) was observed as compared to M2 (98.13 $/ha). The cost of cultivation in M2 was 242.83$/ha and it was higher than M1 (221.81$/ha). The B:C ratio in M1 for soybean was found higher (1.05) as compared to M2 (0.40). Lower B:C ratio in M2 was due to the high cost of cultivation and low net income per hectare. Similar trends for grain yield, cost of cultivation, net income and B:C ratio were observed for wheat cultivation also. The results of increase in yield in wheat and soybean crops under broad bed conditionin vertisol are in agreement with the study reported by Shrivastava et al., 2017 and Jat et al., 2017. This suggests that rotary assisted bed former-cum-seeder helped to increase economic benefit of the farmers adopting broad bed seeding technology.

Conclusions

The rotary assisted broad bed former-cum-seeder is useful for seeding of soybean and wheat crops on broad beds. Provision has been made for attachment of a rotavator in order to form fresh bed or perform sowing operation with reshaping of bed by using bed shaper only. The field capacity of rotary assisted bed former-cum-seeder was 0.32 ha/h at speed of 3 km/h. This machine consumed 15 L/ha fuel during operation. Field demonstration of this technology showed higher grain yield for soybean and wheat crops as compared to flatbed sowing. The higher net return was obtained with the use of rotary assisted broad bed former-cum-seeder technology. The rotary assisted broad bed former-cum-seeder helped to increase economic benefit of the farmers by suggesting them the management practices suitable for sowing under broad beds conditions.

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The Optimization of Topological Mechanism and Dimension Design of Parallel Transplanting Machine in Greenhouse

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Abstract

The transplanting machine is designed based on the parallel mechanism in view of recent facility agriculture which is expensive and with huge size. And the layout of convey line and the demands are put forward before designing the main body of transplanting machine by analyzing the demands for transplanting and operation. Then the position and posture relationship of workspace and accessible space are analyzed by selecting the optimal parallel mechanism from 5 kinds of parallel mechanisms. The difficulty of size optimization through the special point on a workspace is reduced to achieve the optimal size by Ergodic search method.

Introduction

It is the key work that replanting seedlings from higher density plug to lower during the growth process of vegetables and flowers in greenhouse. Transplanting machines which are imported from other countries couldn't be used in China efficiently, because the scale of greenhouse in China is smaller than other countries, even the distribution isn't concentrated (X. Wang et al., 2010). An automatic seedling transplanting machine with 6 picking seedling claws has been developed by Visser and the transplanting rate of this automatic transplanting machine is 15-20 times higher than the transplanting rate of conventional transplanting machine (Netherland VISSER).

Vegetable transplanting robot with 4 fingers has been developed by Dong, the capacity of this transplanting robot is 2,800 pots/hour and the rate of success is 99% in 2012 (Dong, 2012). In the present age, the use of vegetable transplanting machines in greenhouse is developing rapidly (ZU, 2015; Y. Nagasaka et al., 2013). X. Y. Yan has designed a high speed transplanting machine to achieve picking and planting seedlings in the two dimensional plane (Yan et al., 2012; J. Ma et al., 2013). L. Han has designed a simple automatic transplanting machine with effective outcomes (L. Han et al., 2016).

Recently, plug seedlings in greenhouse is associated with the preset trajectory along the straight line path during operation, and picking seedling claw can only pick seedlings from pre-set position because the program of transplanting is in row, so it couldn't remove the residual and illness seedlings from the plugs during the operation effectively. Tandem transplanting machine has a high cumulative error and bigger volume, while transplanting machine with parallel mechanism has the advantages of high transplanting speed, high efficiency, large rigidity and better dynamic performance, it is effectively suitable for automatic greenhouse. A transplanting machine with parallel mechanism is designed to remove illness and residual seedlings effectively. The convey line layout is designed according to the growth of seedlings, characteristics of seedling plugs and target containers by taking the transplanting seedlings as object (Gasparetto, 2010). The suitable parallel mechanism is selected as the main body and the workspace is analyzed, while the minimum size is selected as the optimal target to achieve the optimal size of the transplanting machine.

Demand Analysis of Transplanting Machine in Greenhouse

The general purpose of transplanting machine in greenhouse is that transplanting the seedlings into lower density plugs, pipes and bowls. The process of transplanting consists of picking up seedlings from trays, delivering seedlings and feeding the seedlings in three dimensional spaces. Actions of picking and feeding seedlings demand a special matching picking seedlings machine for effective performance, even the movement of seedling plug should be matched with the movement of three dimensional space while delivering the seedlings into three dimensional space, it demands us to design a specific transplanting machine with parallel mechanism and three translating movements. The planning system of transplanting machine includes main body of transplanting machine with parallel mechanism, container conveying platform, control system, and the arrangement of container conveying platform. The workspace of this parallel mechanism includes all the transplanting trajectory. It is clear that the workspace is a cube area, and the size of workspace could be according to the actual size of plug plate with 128 holes and the height of the young seedlings.

Optimization of Topological Mechanism with Three Translational Parallel Mechanism

I. Five Kinds of Three Translational Parallel Mechanism

Five kinds of mechanisms with certain characteristics are selected in this paper shown in **Table 1**.

II. Optimization of Mechanism

All the mechanisms fulfil the requirements of transplanting machine because of the driven pairs could be three rotation pairs on the static platform from **Table 1**. There is no difference among them according to the amount of calculations on coupling degree. The main difference among them is equal to the number of over restraint and the input-output decoupling. The stiffness of mechanism A isn't very high as the addition of the ball pair. Ball couple can't be guaranteed when introduced a heavy load and angle. The principal differences among B, C, D and E are the power properties of every chain when moving and the input-output decoupling, which are all composed of two kinds of chains. B mechanism can only achieve input-output decoupling with a translation direction. C mechanism can achieve decoupling with 3 translation directions when the driven pairs are arranged vertically, but the workspace is large and easy to interfere. D and E mechanism can achieve input-output decoupling with 2 translation directions when the driven pairs are arranged vertically, while the force of chain is lower than others; finally choosing E mechanism as the main body of transplanting machine and the arrangement method is shown in **Fig. 1**. The driven axis of chain A is parallel to the driven axis of chain C; and the driven axis of chain A and B are vertical to the driven axis of chain B, the axis of A, B and C

Table 1 Topology characteristics of parallel mechanism

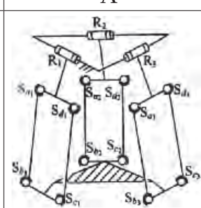
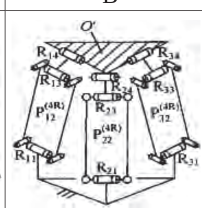
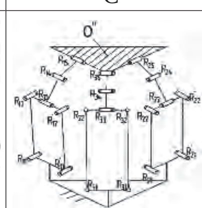
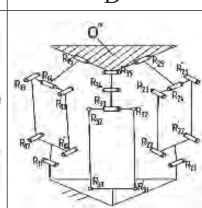
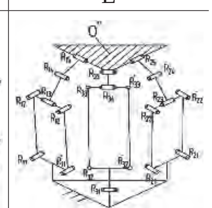
	A	B	C	D	E
Diagram					
Movement pair number	15	21	21	21	21
Active components number	10	16	16	16	16
Over restraint number	0	12	12	12	12
Position of drive pair	Driving pair could be the 3 rotation pair in static platform				
Coupling degree of BKC	κ = 1				
Input-output decoupling	No decoupling	1 direction decoupling	3 direction decoupling	2 direction decoupling	2 direction decoupling

Fig. 1 Topology style of parallel mechanism E

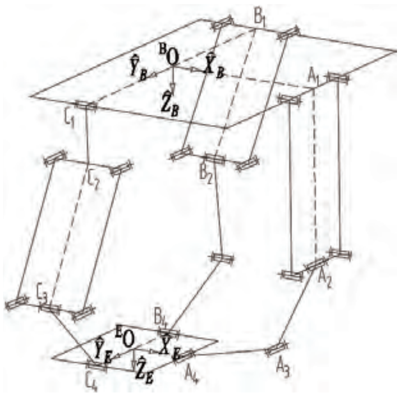


Fig. 2 Lowest accessible boundary

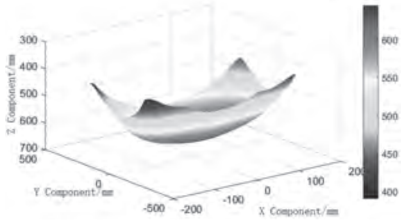
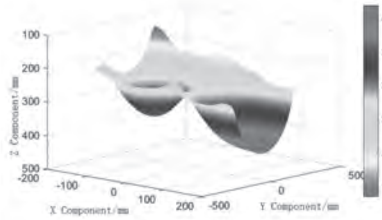


Fig. 3 Highest accessible boundary



are located in the same plane. It can improve the force of the chain and get partial input-output decoupling.

Analysis of Workspace

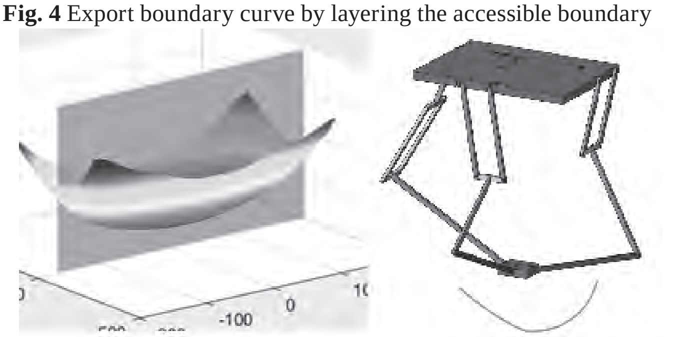
I. Accessible Space of the Parallel Mechanism

Following restrictions are proposed according to the singularity position and combined with the restriction of driven pressure angles on the parallel mechanism.

$$\text{Chain A: } \begin{cases} -90° < \theta < 90° \\ 40° \le \theta'' \le 140° \end{cases}$$
$$\text{Chain B: } \begin{cases} -90° < \beta < 90° \\ 40° \le \beta'' \le 140° \end{cases} \quad (1)$$
$$\text{Chain C: } \varphi'' < 0$$

MATLAB is used to descend dimension to search the accessible workspace of parallel mechanism according to the restriction of the angle. There is no significant, when the drive angles are closed to ± 90°, it makes little contribution to accessible space and workspace during actual analysis. Therefore, the restriction of the driven angle of chain A and B are limited to ± 80° and the driven angle of chain C is limited to ± 90°. Then the lowest accessible boundary and the highest accessible boundary are achieved by MATLAB shown in **Fig. 2** and **Fig. 3**.

Fig. 4 Export boundary curve by layering the accessible boundary



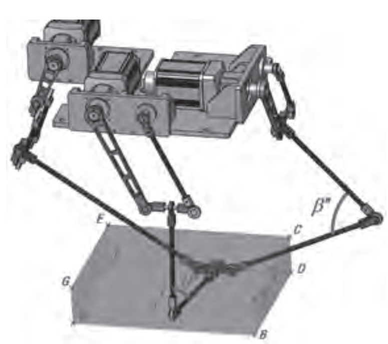
II. Relationship of Position and Posture

The map of boundary is dealt with by stratifying and delaminating the boundary line, the driven angles on chain B are -80°, -60°, -30°, 0°, 30°, 60°, 80° and vertical to the XOY platform stratified shown in **Fig. 4**, the boundary which is the shadow of left figure is selected, and the boundary curve of driven angle is achieved which is the solid line in the below of right **Fig. 4**.

Transforming the curve into three-dimensional coordinate and then importing into PROE to synthesize a series of spatial spline in the space (the solid line in **Fig. 4**). It is possible to observe the position relationship of the accessible boundary surface and the parallel mechanism shown in **Fig. 5**, the position between the upper and lower space can fill the accessible space of the parallel mechanism. It is necessary to contain the accessible space in the workspace. The cube shadow is the expected workspace, then minimizing size of the general parallel mechanism.

Fig. 5 Position and configuration relationship between work space and accessible space

Fig. 6 Special points in workspace



Optimization of Dimension

I. Special Points of the Workspace

The 8 vertices and 2 intermediate points of the workspace boundary are representative by further analyzing the relationship between the position and posture of the parallel mechanism and workspace. It is possible to obtain the whole space when the parallel mechanism platform reaches at point B, D, F, H and J from **Fig. 6**.

The key angles θ" and β" will change when the moving platform moves to the 12 edges of the workspace, the statistical rules of their changes are shown in **Table 2**. The second column is movement line, such as A→B means the movement line is from point A to B; the third and fourth columns are the changes of the angle θ" and β", they show the trend of changes, the trend ↑ means the angle increases, and ↓means the angle decreases, L means the maximum angle, and K means the minimum angle, W isn't the non-limited angle. For example, the angle increases from a non-limited value to the maximum angle and then to another non-limited value when W↑L↓W appears in the eighth row and the third column.

The underline and bold fonts are the limited values in **Table 2**. It is clear that the maximum angle only appears in two situations, that is the maximum value of would appear in the middle of A→C, and the maximum value of would appear in the middle of G→A from the table, even those two points are the point I and point J. The minimum value appears many times on the table, while only three points appear with removing the repeated points, the three points are the vertex in the workspace which the minimum value appears in point D and point F, the point D and point F are symmetrical for chain A. The minimum value of β" appears at point F and point H, even point F and point H are symmetrical for chain B so that point F is the same minimum point of θ" and β". For this parallel mechanism, if point F can meet the minimum limit of θ" and β", point I can meet the minimum limit of θ", and even point J can meet the maximum limit of β", so that it would meet the angle limit conditions of θ" and β". The limited condition doesn't need to be considered during the process of analysis because the chain C is at any point of the workspace.

$$\min_{sH} sH = F(l_1, l_2, l_3, m_1, m_2, m_3, b_1, b_2, e_1, e_2, \theta, \beta)$$

$$\text{s.t.} \begin{cases} -90° < \theta < 90° \\ -40° \le \theta'' \le 140° \\ -90° < \beta < 90° \\ -40° \le \beta'' \le 140° \\ 0 < l_2 < l_{2s};\ 0 < l_3 < l_{3s} \\ 0 < m_2 < m_{2s};\ 0 < m_3 < m_{3s} \\ l_1 = 350;\ m_1 = 280;\ b_1 = 350;\ b_2 = 408;\ e_1 = 45;\ e_2 = 56 \end{cases} \quad (2)$$

II. Process of Search and Results

It is not clear to set the minimum size as the target during the process that integrating the size of the parallel mechanism, while the fact is that small size of length and height the better in the actual mechanism. The size of length and height would be determined when assuring three dimensions on the rack, and the swing angle and workspace of the driven arm are all determined; so the key optimiza-

Fig. 8 Search flow chart of optimal size

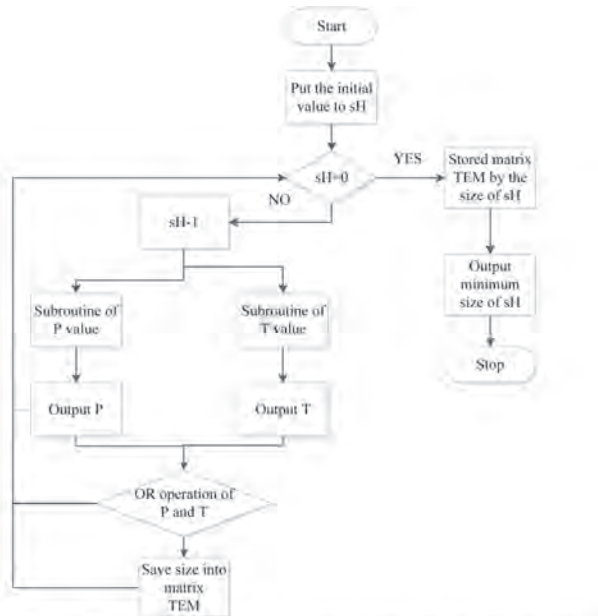


Table 2 Changes when moving around the sideline of workspace

No.	Movement Line	θ"	β"
1	A→B	W↓W	W↓W
2	C→D	W↓**<u>K</u>**	W↓W
3	E→F	W↓**<u>K</u>**	W↓**<u>K</u>**
4	G→H	W↓W	W↓**<u>K</u>**
5	A→C	W↓W	W↑**<u>L</u>**↓W
6	C→E	W↑W↓W	W↓W
7	E→G	W↑W	W↑W↓W
8	G→A	W↑**<u>L</u>**↓W	W↑W
9	B→D	W↓K	W↑W↓W
10	D→F	**<u>K</u>**↑W↓**<u>K</u>**	W↓**<u>K</u>**
11	F→H	**<u>K</u>**↑W	**<u>K</u>**↑W↓**<u>K</u>**
12	H→B	W↑W↓W	**<u>K</u>**↑W

tion target is the height parameter. The optimization goals are presented in the following Eq. 2.

While sH represent the vertical distance from the plane of the upper surface of the workspace shown in **Fig. 7**. The length and height of the workspace are only related to chain A and B because of the decoupling in X axis and Y axis according the

Fig. 9 Subroutine flowchart of P

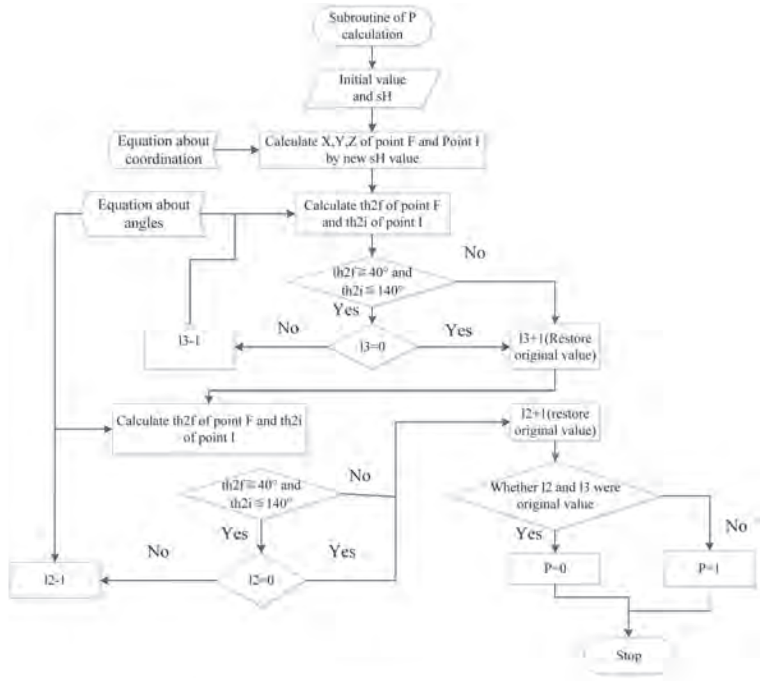
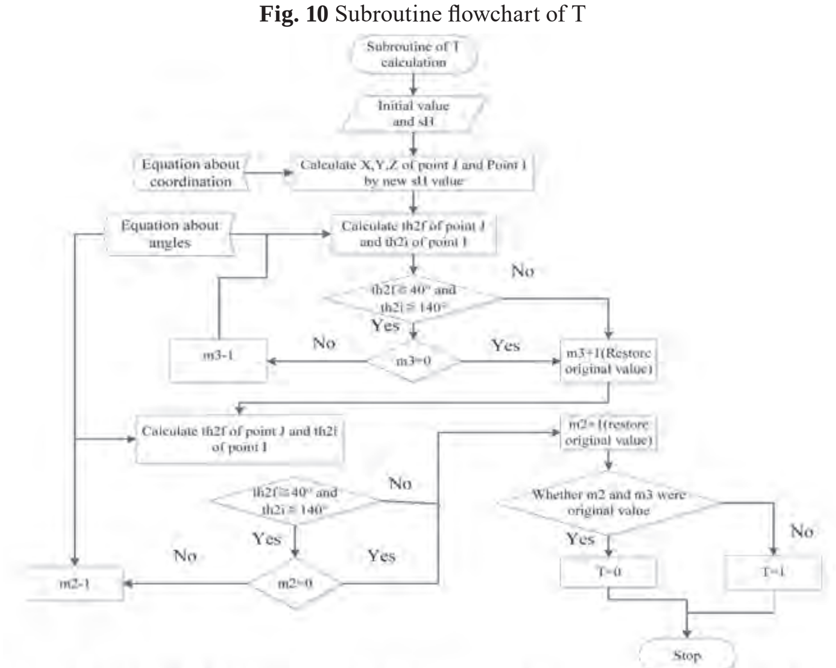


Fig. 10 Subroutine flowchart of T



$$\begin{cases} b_1 = 350 \\ b_2 = 408, \\ b_3 = 300 \end{cases} \begin{cases} e_1 = 45 \\ e_2 = 56, \\ e_3 = 56 \end{cases} \begin{cases} l_1 = 350 \\ l_2 = 356, \\ l_3 = 496 \end{cases} \begin{cases} m_1 = 280 \\ m_2 = 436, \\ m_3 = 526 \end{cases} \begin{cases} n_1 = 144 \\ n_2 = 320, sH = 661\,(mm) \\ n_3 = 676 \end{cases} \quad (3)$$

above analysis, chain C is responsible for the amount of translation in the Z axis direction by changing the movement platform. It could meet the appropriate size of chain C if achieving the condition of chain A and B during the process of size optimization, and it isn't necessary to use any parameters on chain C when calculating θ" and β". There is no relationship between the size of sH and chain C during the size synthesis, that means it is only necessary to meet the condition of driven rod l_2, l_3, m_2, m_3 which are on chain A and B, so that reducing the variable to 4 and greatly simplified the calculation and improve the speed of calculation. Although it has a large amount of calculation within the acceptable scope, Ergodic search method could achieve the highest accuracy. Ergodic search method is used to search the optimal solution in this paper, the specific search process is shown in **Fig. 8**. Briefly speaking, firstly selecting a large sH and the size of every bar which could meet the conditions, then reducing the height of sH within the speed of 1 mm; secondly checking whether existing the angular restrictions that could meet the condition of angle in every height; continuing reducing and repeating the process of search if existing until sH is 0; and arranging all possible sizes as sH, then outputting the minimum sH.

The calculation program of P and T respectively are calculated the limited angle on the chain A and B, they all have the same process except the calculation equations and initial values, the flow chart are shown in **Fig. 9**.

In this **Fig. 9** and **Fig. 10**, th2f means the θ" of point F and th2i means θ" of point I, beta2j means β" of point J and beta2i means β" of point I for the unity with MATLAB. The optimization results are shown as follows, and n1, n_2, n_3 are the results which are made up after obtaining the optimization value of other parameters, they must need to meet the requirements of the constraints on chain C (Eq. 3).

Conclusions

In this paper, it is put forward the layout of transplanting machine convey line and design requirements of the main body on transplanting machine. Special parameters of topological mechanism are calculated according to the five selected series of parallel mechanisms, then the 2-HSOC{-R//R//R-P4R-}1-HSOC{-R//R(-P4R)//R-} parallel mechanism is determined to be the main body mechanism of transplanting machine which contained partial decoupling with high over-constrained and rigidity.

The accessible workspace of parallel mechanism based on the singular position and posture has been achieved and gotten the accessible position and posture of workspace. By reducing the difficulty of the size optimization of special points on workspace and taking the minimum size as the target, a set of optimization assemble is achieved finally. In this case, it will contribute to the development of transplanting robot in greenhouse.

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■■

Implementation of Image Processing and Fuzzy Logic Discriminator of Hatching Eggs Fertility

by

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Abstract

The capability to automatically identify fertile eggs for incubation could achieve high profits of hatcheries. Infertile eggs manually detection and removal cost a lot and can lead a pathogen contamination of chicks. Many systems engineering software are developed in last few decades for applications that require test, measurement and control with rapid access to hardware and data insights. LabView™ is one of these systems engineering software which offers a graphical programming approach. Fertility detection techniques are expensive to be applied widely, hence an investigation of the possibility of using a low-cost method. Egg fertility visualization using machine vision are expensive, but the using LabView™ software package can provide a suitable low-cost method for egg fertility by visualization using image processing and fuzzy-logic system control kits, a suitable candling unit designed for illuminating the eggs which potentially enables an camera with 13 Mega Pixels in fifth day of incubation. Captured images by camera mobile phone (Huawei Y6 Pro), and results extract four rules from fuzzy logic and Pixels value histograms results showed a significant difference between the mean pixels values of fertile and infertile eggs at 95% confidence level.

Introduction

With the huge development in machine vision systems and image processing, the usage of such methodologies became a focus of attention by those who are interested in poultry reproduction. Hence non-destructive methods for detecting fertility using machine vision and using different types of imaging were investigated before (Klein et al., 2002; Jones et al., 2005; Bain et al., 2007; Zhu and Ma, 2011; and Liu and Ngadi, 2013) because invasive and semi-invasive methods cause damages to the egg (Cain et al., 1967; Howe et al., 1995; Akiyama et al., 1999; Tazawa et al., 1999; Moriya et al., 1999, 2000; Kato et al., 2002; and Liang et al., 2011). If poultry hatching facilities needs to achieve hatch success rate of 86-95% according to USDA (2006), they will need a proper, non-invasive and quick detection of infertile eggs because they could cause a contamination and taking space and time (Smith et al., 2008; Zhu and Ma, 2011; Liu and Ngadi, 2013; Hai-ling et al., 2016; and Önler et al., 2017). Image processing is one of methods that are non-invasive and quick for detecting fertility of hatching eggs, so different methods like color pattern matching, pixel value histograms and color spectrum were investigated to determine the feasibility of using these methods in sorting infertile eggs. LabView™ programming software package is used in many engineering and scientific applications (Hahn and Elmessery, 2011; Elmessery, 2011; Elmessery and Abdallah, 2014). Candling is a followed procedure for identifying fertility of hatching eggs in Egypt (FAO, 2009). The relationship between Region of Interest (ROI) area and numbers of correct matches was investigated and if there is any significant difference between mean pixels values of fertile and infertile eggs. Designing and investigating systems that are depending on fuzzy logic such as used in (Abdallah and Elmessery, 2014) to learn and predict the fertility of hatching eggs was considered to evaluate the discrimination systems depending on fuzzy logic instead of simple Boolean logic. Also control surfaces and rules generated by this system could be used by anyone to predict fertility of hatching eggs. Therefore the present investigation aims to study the following objectives:

1. Investigate the feasibility of using color pattern matching in fertility detecting and the effect of ROI area on number of correct matches;
2. Find the significance difference between the mean pixels values of fertile and infertile eggs; and
3. According to the results obtained from the two previous objectives a fuzzy logic control system was designed to learn and predict fertility.

Materials and Methods

Egg Samples Evolution

Chicken eggs samples chosen for the present investigation were from Hubbard breed. Samples images were taken at Tarek Diab Hatchery, Nashil, Qotour, Gharbia Governorate, Egypt during the year of 2018. The freshly laid eggs were incubated in a Smart™ incubator (Pas Reform Hatchery Technologies, Zeddam, the Netherlands) at 99.7 °F and 54% relative humidity, and were turned every hour. On the fourth day of incubation, eggs were taken out from the incubator to be candled through transferring the whole tray of incubated eggs from the incubator to the candling unit. Candling process did not take more than 1 min to return eggs tray into the incubator immediately for the safety of chick embryos. Sample picture were captured at the investigated age to be processed to investigate the feasibility of using image processing techniques to detect the fertility of hatching eggs.

Experimental Setup

To get an image of candled eggs to be processed using an image processing software to investigate its ability to discriminate between fertile and infertile eggs, it needed an appropriate candling unit allows an easy and fast transferring of trays. A local-made candler is consisted of 70 × 40 × 30 cm container was manufactured from 3 mm mild steel sheets and furnished to be like a trolley for easy movement. The candler was manufactured in a workshop located at Gharbia Governorate, **Fig. 1**. After removing eggs tray at the ages of five days old from the incubator it is installed on the candler to capture an image for the tray. Light source used for candling was lamp of 40 W. Twelve lamps

Fig. 2 Flowchart of LabView™ implementation for image processing

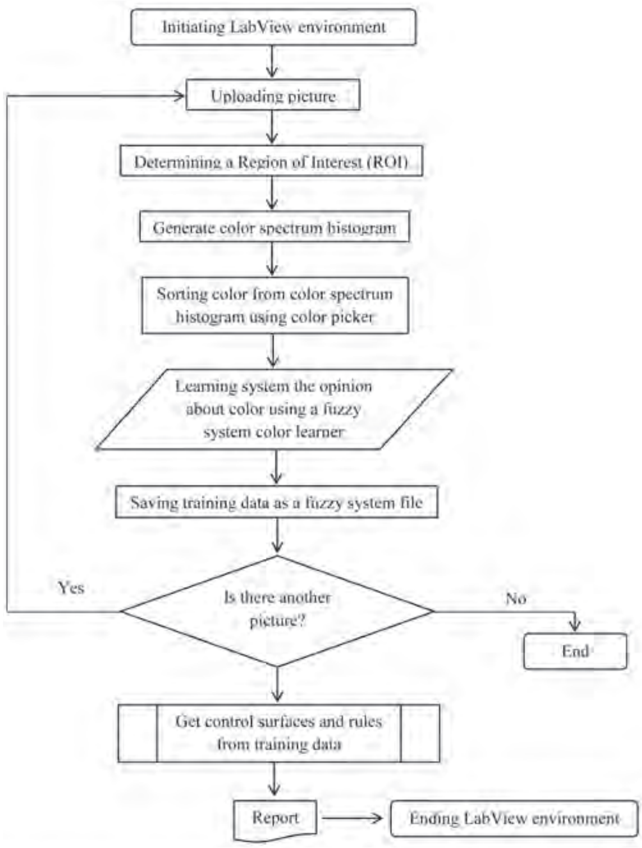
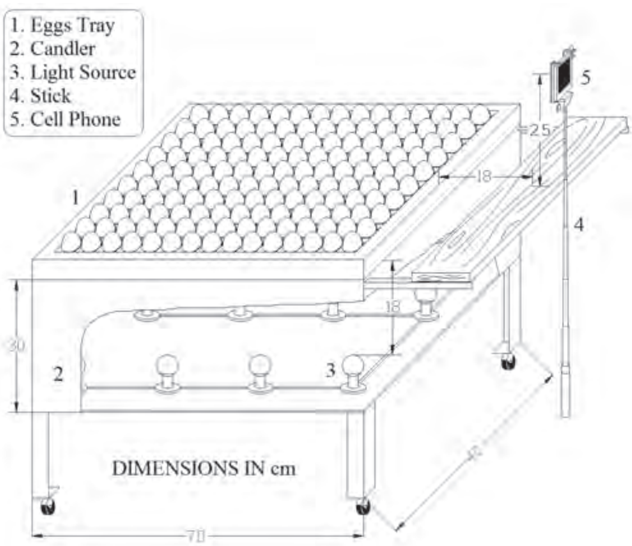


Fig. 1 Schematic drawing of candling unit and dimensions at which the image was captured to be processed



were distributed in the bottom of the container and were coordinated to be in three rows transversely and four columns longitudinally as depicted in **Fig. 1**. Distance between lamps tops and eggs trays was 18 cm. several trials were conducted to get the best image from above and sides and the best chosen position was as depicted in **Fig. 1**.

Image Processing

Image processing is a method to perform some operations on an image, in order to get an enhanced image or to extract some useful information from it. Image processing basically includes three steps: 1. Importing the image via image acquisition tools, 2. Analysis and manipulating the image and 3. Output of which result is the report. The image processing steps for egg fertility discrimination by LabView™ program are shown in **Fig. 2**.

IMAQ Match Color Pattern

Color pattern matching was used to determine the behavior of the software if it is able to determine the infertile eggs by drawing a ROI on an infertile egg. The “IMAQ Match Color Pattern” function also was essential to perform this test; the software could read the image using “IMAQ Read File” function. If there are a number of matches that is replicated on the same egg or matches on the layout of the egg or outside of it was considered as wrong match, three different ROI areas were investigated to determine the number of correct and incorrect matches, the ROI area was 14 × 14, 18 × 18 and 22 × 22 pixels.

IMAQ Histogram Function

IMAQ Histogram function is used to analyze the overall grayscale distribution in the image. Use the histogram of the image to analyze two important criteria that define the quality of an image saturation and contrast. If the captured image is underexposed because it was acquired in an environment without sufficient light, the majority of image pixels will have low intensity values, which appear as a concentration of peaks on the left side of the histogram. On the other hand: If the captured image is overexposed because it was acquired in an environment with too much light, the majority of the pixels will have high intensity values, which appear as a concentration of peaks on the right side of histogram. If the captured image has an appropriate amount of contrast, your histogram will have distinct regions of pixel concentrations. Use the histogram information to decide if the image quality is sufficient enough to separate objects of interest from the background. One of issues that was a matter of interest is using image processing techniques to study its feasibility for images that had been captured for the egg tray on the candler and for this purpose LabView™ Vision Development Module 2017 toolkit and its accompanying utilities (National Instruments Corporation, 2009a) were used to investigate their feasibility in detecting fertility of hatching egg. An image was captured from 25 cm height above the tray and from a distance of 18 cm. One of the previously mentioned toolkit functions was “IMAQ Histogram” to set forth histogram information about the portion of the image in a Region of Interest (ROI), **Fig. 3**. The pixel value and number of pixels of each image was analyzed statistically by ANOVA to study if there are any significant differences between the two samples without replications between mean pixel value for both fertile and infertile eggs, the ROI area was a square of 22 × 22 pixels and it is supposed

Fig. 3 Application of IMAQ Histogram function in detecting fertility of hatching egg

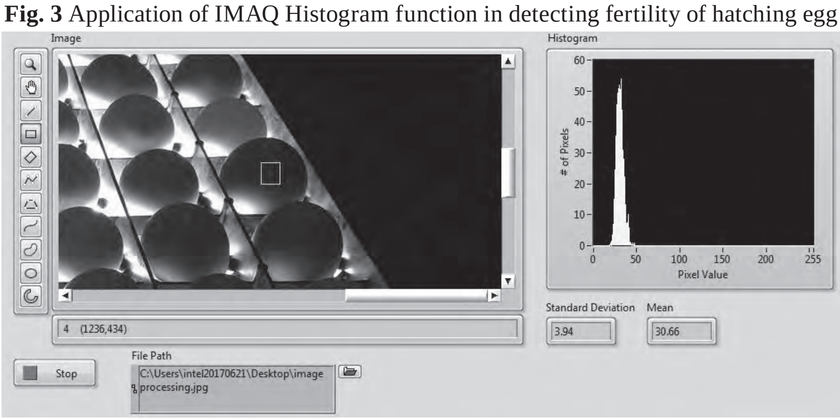
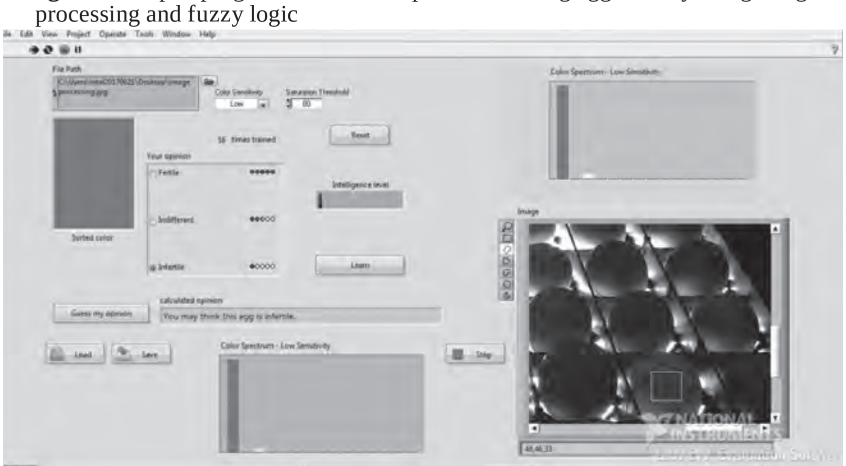


Fig. 4 Developed program to learn and predict hatching egg fertility using image processing and fuzzy logic



that the pixel value of infertile eggs is greater than the fertile one.

Developing a Fuzzy Logic System to Learn and Predict Fertility

From the conclusion of the results obtained by the first investigation of egg fertility discrimination (Abdallah et al., 2018), the Boolean egg fertility discrimination (the value of 1 is completely true and 0 is completely false) that method used by (Abdallah et al., 2018) could be fortified by applying fuzzy logic system. In this paper a fuzzy logic system is developed to handle the partial truth which is not completely true (fertile) or false (infertile) according the data achieved from processed images of eggs. Fuzzy logic is a method of rule-based decision making used for expert systems and process control (National Instruments Corporation, 2009b). To develop a system that is based on fuzzy logic; it could learn and predict eggs fertility it is needed to gather between "Fuzzy Logic VIs" and "NI Vision" functions. The captured egg image is processed to color spectrum histogram and then the color picker is applied to choose among obtained colors on the histogram then the fuzzy color learner is going to guess if the egg is fertile or not after making numbers of trains. Color spectrum histogram that was generated using "IMAQ Color Learner" function was essential because using the color picker directly on the egg tray image are going to generate large number of colors and that is difficult to be sorted by the fuzzy system but color spectrum histogram generate an indicated number of colors, **Fig. 4**. In this fuzzy system at the beginning if there is no training there are no rules after training the rules will be created programmatically. The "RGB to binary numbering" RGB numeric representation function converts a red, green and blue value from 0 to 255 to the corresponding color of red, green and blue. After twenty-three times trained (ten infertile, ten fertile and two indifferent eggs) the rules was generated and acquired and also the three dimensional control surfaces graphs. It was noticed from color spectrum that the common color between infertile eggs was the red so it was chosen to learn the system that it is infertile egg and for fertile one it was the black, indifferent egg which is known as infertile but the red color does not appear while the brown color appears so it was chosen to learn the system that it is indifferent egg. Data of training was saved as a fuzzy system file and loaded again to calculate the opinion at any time as shown in **Fig. 4**.

The fuzzification method was Min-Max method, there were three input variable (Red, Green and Blue) each input variable membership function can describe a measurable form that the system can take (Eyabi, 1999), Each input variable has a range of [0, 255] while the output variable has a range of [0, 5], **Fig. 5**. The defuzzification method was used was center-of-Area method.

Table 1 Data summary of pixels values for both fertile and infertile egg

Summary	Count	Sum	Average	Variance
Fertile	10	363.06	36.306	137.168
Infertile	10	670.51	67.051	337.379

Table 2 ANOVA of pixels values for both fertile and infertile egg

Source of Variation	*SS*	*df*	*MS*	*F*	*P-value*	*F crit*
ROI	1,897.734	9	210.859	0.800	0.628	3.179
Fertile or Infertile	4,726.275	1	4,726.275	17.924	0.002	5.117
Error	2,373.185	9	263.687			
Total	8,997.194	19				

Results and Discussion

Pixel Value Histograms

For infertile eggs mean pixel values taken from ten samples ranged

Fig. 5 Input and output membership functions: (A) sigmoid shape for the three input variables Red, Green and Blue; (B) Triangle shape for output variable

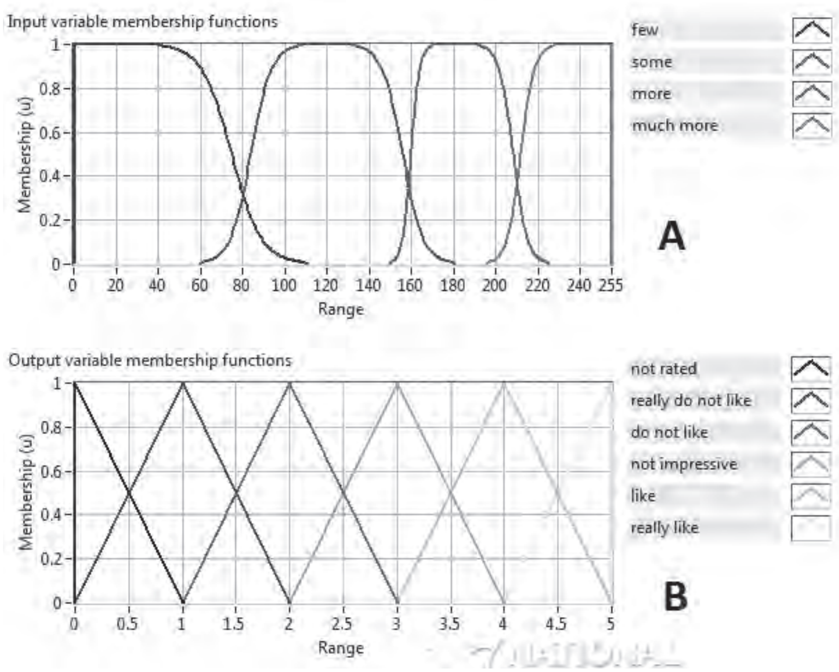
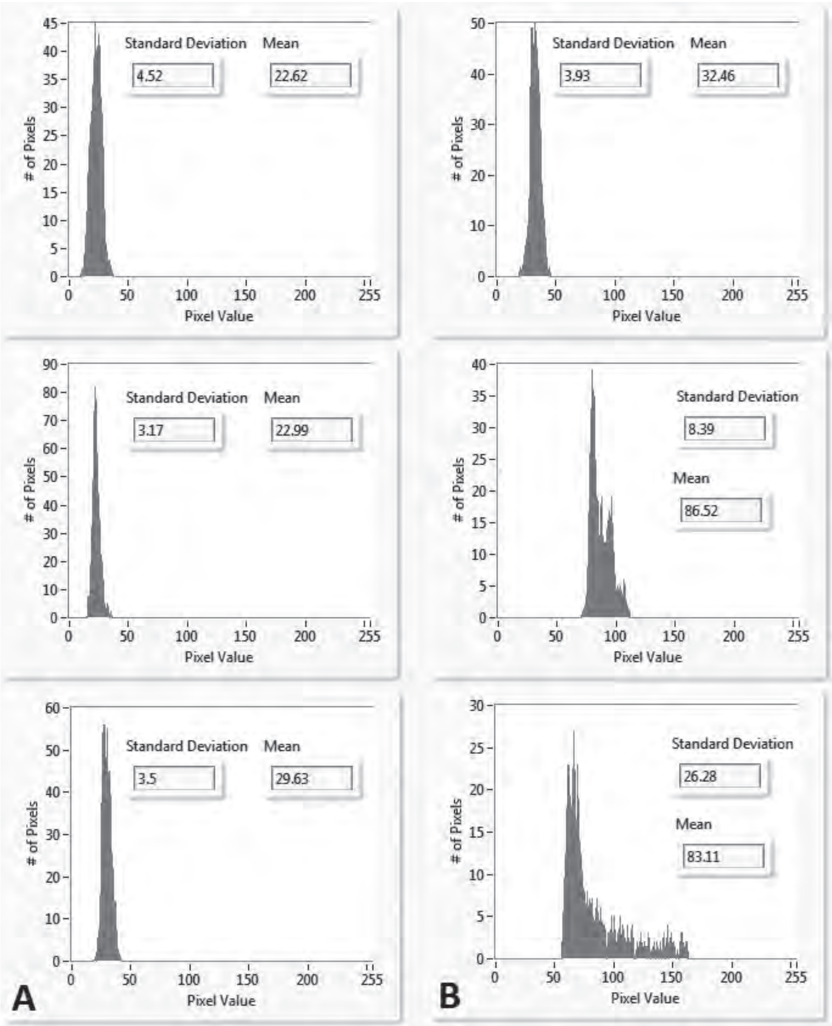


Table 3 The number of correct and incorrect matches at three different ROI areas

ROI Area, pixels	Trial No.	No. of matches at max No. of matches = 26	
		Correct	Incorrect
14 × 14	1	4	1
	2	4	2
	3	15	11
	4	10	12
18 × 18	1	1	0
	2	0	0
	3	9	13
	4	13	12
22 × 22	1	3	4
	2	6	4
	3	9	7
	4	11	4

from 32.46 ± 3.93 to 88.09 ± 6.67 and for the other ten samples also of fertile eggs ranged from 22.62 ± 4.52 to 56.19 ± 19.5. As shown in **Fig. 6** the mean value of depicted samples from both fertile and infertile eggs, pixel value is higher in the infertile one, **Table 1**. The ANOVA results shows that P-value is less than 0.05, indicating that they are

Fig. 6 Samples of pixel values of ROI taken from (A) fertile and (B) infertile eggs



significantly different from 0 at 95% confidence level, **Table 2**.

Color Matching Patterns

As listed in **Table 3** the number of correct match increased with the decrease in ROI area. Sorting efficiency was calculated by dividing correct matches number on the total matches and for the average sorting efficiency for each ROI area was calculated and it was 62, 48 and 57% for 14 × 14, 18 × 18 and 22 × 22-pixels ROI area.

Fuzzy System Results

Four rules were generated after training the system as set forth in **Table 4**. As shown in **Fig. 7** the relationship between every two input variables and the opinion was obtained by the control surface graphs. Control surface with four rules demonstrating a linear characteristic and it means that it is less sensitive to small changes about the set point, i.e., when the input variables are zero (flatness) (Eyabi, 1999). As shown in **Fig. 7-D** and to understand the control surface if the input variables values for Red, Green and Blue are 113.182, 99.545 and 0, respectively the output value of the opinion would be 2.0167 and this invokes the fourth rule in **Table 4** because backing to **Fig. 5-B** this value is ranging in ‘do not like’ opinion.

Conclusions

1. Color matching patterns results shows an increase in number of correct matches with the decrease in Region Of Interest ROI area although numbers of incorrect matches increased also, so this methods is not recommended;
2. Pixels value histograms results showed a significant difference between the mean pixels values of fertile and infertile eggs at 95% confidence level; and
3. Control surfaces were less sensi-

tive because of their linear characteristic and they invoked four rules to predict fertility of hatching eggs.

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Fig. 7 Three dimensional control surface of the fuzzy system rule for the three input variables; (A) Red and Green; (B) Green and Blue; (C) Red and Blue (D) working principle of control surface to get a measurable value of the opinion at two input variables

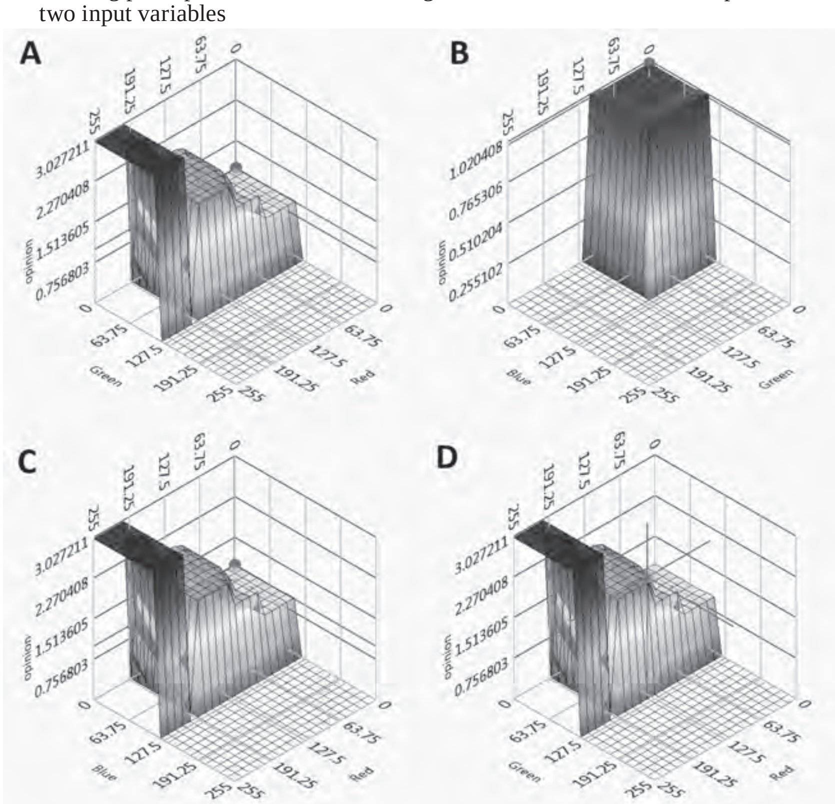

Table 4 Created rules after training the fuzzy system

No.	Rules
1	IF ‘Red’ IS ‘few’ AND ‘Green’ IS ‘few’ AND ‘Blue’ IS ‘few’ THEN ‘opinion’ IS ‘really do not like’
2	IF ‘Red’ IS ‘much more’ AND ‘Green’ IS ‘few’ AND ‘Blue’ IS ‘few’ THEN ‘opinion’ IS ‘not impressive’
3	IF ‘Red’ IS ‘more’ AND ‘Green’ IS ‘more’ AND ‘Blue’ IS ‘more’ THEN ‘opinion’ IS ‘not impressive’
4	IF ‘Red’ IS ‘some’ AND ‘Green’ IS ‘few’ AND ‘Blue’ IS ‘few’ THEN ‘opinion’ IS ‘do not like’

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Performance Evaluation of a Multi-crop Shelling/ Cracking Machine for Shelling of Peanut Pods

by

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Abstract

This study aims to evaluate performance of a developed shelling/ cracking machine for shelling of peanut pods (PPs) American variety cultivated in Egypt. All shelling experiments of PPs were carried out at average moisture content of 7.44, 5.89 and 12.48% d.b. for pods, seeds and shells, resp. Some physical properties of PPs were studied such as; %-ages of peanut pods parts, axial dimensions, mass and bulk density for pods and seeds of peanut. The results showed that; the used machine achieved high efficiency in shelling of PPs. Under the tested speed range [100 to 400 rpm] and the tested clearance range [9 to 12 mm], the optimum operation conditions for shelling of PPs was 200 rpm for speed and 10 mm for clearance. Under the tested sample mass range [200 to 1000 g], the mean values of shelling efficiency were not significantly with average of 97.54%. Also; the mean values of %-age of unshelled pods, broken seeds, split seeds, intact seeds and shell with dust were not significantly with average of 2.46, 4.95, 14.52, 52.20 and 25.63%, resp. The results revealed that the highest value of machine productivity was 48.29 kg/h, whereas the mean value of power consumption was 689.59 W.

Keywords: Peanut pods, shelling/ cracking machine, shelling efficiency.

Introduction

Peanut/Ground nut (*Arachis hypogaea* L. is scientific name under family of Fabaceae/ Leguminosae) is an important food crop and good source of edible oil. Peanut seeds contain of 36-54% for oil, 16-36% for protein and 10-20% for carbohydrates, (Knauft and Ozias-Akins, 1995). The total cultivated area of peanut pods (PPs) in Egypt about 147619.05 Feddan and producing about 199000 Mg according to, (FAO, 2017).Peanut seeds are important nutritional and economical crop, used for human feeding and different industrial aspects such as sweets, peanut butter, paint, insecticides, nitroglycerin etc. Peanut shells are used in the manufacture of plastic, wallboard, fuel and cellulose, (Mady, 2017).Werby and Mousa (2016) mentioned that the knowledge of the physical, mechanical and aerodynamic properties of agricultural products are necessary and important in design of the different component of machines and equipment of processing, handling, cleaning, transporting and storage. Physical and mechanical properties of several varieties of peanut pods and seeds were studied by (Ghanem et al., 2009; El-Sayed et al., 2001; Aydin, 2006; Dilmac & Ebubekir, 2012 and Kurt & Arioglu, 2018).

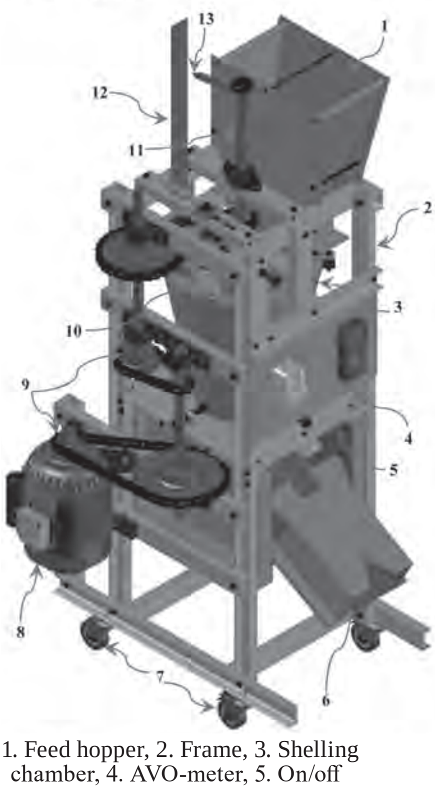
Gitau et al. (2013) tested a two manually operated decorticators/ sheller; the results showed that for wooden beater sheller, the shelling efficiency increased with decreasing the moisture content for all the groundnut varieties. The highest values of shelling efficiency were 55.3% for (ICGV 99568), 39.2% for (ICGV 90704) and 29% for (ICGV 12991) at feeding rate of 30 kg/h, 22.6 mm clearance and moisture content of 5.92% w.b.; while, for rod beater sheller, the highest values of shelling efficiency were of 58.3% for (ICGV 99568), 42.7% for (ICGV 90704) and 35% for (ICGV 12991) at moisture content of 7% w.b.

Helmy et al. (2013) evaluated performance of a reciprocating peanut sheller before and after modification by supplying the sheller with feeding mechanism (conveyor belt), increasing the friction area of shelling box, and using rubber for enhancing shelling process. The results showed that, the performance of a reciprocating peanut sheller after modification is better than that before modification. The shelling efficiency and productivity after modification were 98.85% and 155.98 × 10^{-3} Mg/h, reps., at feed rate of 160 kg/h, box speed of 1.4 m/s, moisture content about 17.12% w.b. and air velocity of 8.37 m/s. But before modification, the shelling efficiency and productivity after modification were 95.32% and 89.20 × 10^{-3} Mg/h, reps., at feed rate of 100 kg/h and the other studied operating conditions.

Mady (2017) manufactured and evaluated a peanut sheller under the following operational conditions; drum rotary speeds of 150, 200, 250 and 300 rpm, feeding rates of 170, 210 and 250 kg/h and air speeds of 4.9, 6.8 and 8.8 m/s. The results showed that the highest shelling efficiency was 96.23% at drum speed of 150 rpm and feeding rate of 170 kg/h. The highest productivity of machine was 250 kg/h at drum speed of 300 rpm and feeding rate of 250 kg/h, whereas the highest cleaning ratio was 98% was at drum speed of 150 rpm and air speed of 8.8 m/s.

Prem et al. (2017) mentioned that the shelling of pods is very difficult and very time-consuming operation in case of carrying out manually. In addition to unavailable of small shelling machines in Egypt, Mousa (2018) developed a shelling/cracking machine depends on friction rubbing action for some agricultural products and evaluated performance for shelling of *Jatropha curcas* L. fruits (JCFs) cultivated in Egypt. The results showed that the suggested design of the developed machine achieved high efficiency in shelling of JCFs. The speed of 600 rpm is the optimal for the shelling efficiency at clearance of 12 mm. Also; the mass sample of 1000 g is the optimal for shelling efficiency and productivity of machine, the results revealed that the highest value of machine productivity was 226.08 kg/h. Meanwhile; the average values of shelling efficiency, %-age of broken seeds, unshelled fruits and intact seeds were 95.90, 0.43, 4.10 and 57.93%, resp., as well as; he concluded that testing and evaluate performance of the developed shelling/cracking machine for shelling of peanut pods; So; this study was carried out to evaluate performance of the developed shelling/cracking machine for shelling of peanut pods cultivated in Egypt.

Fig. 1 3D drawing of developed shelling/cracking machine



1. Feed hopper, 2. Frame, 3. Shelling chamber, 4. AVO-meter, 5. On/off switch, 6. Discharge outlet, 7. Wheels, 8. Electric motor, 9. Sprocket wheels and chains, 10. Transparent cover, 11. Rotational shaft, 12. Scale of clearance, 13. Pointer

Materials and Methods

The shelling/cracking machine was developed by Mousa, (2018). The shelling experiments of peanut pods “PPs” (American variety) were carried out in the Laboratory of Agric. Mech. and Power Eng. Dept., Fac. of Agric. Eng., Al-Azhar Univ., Cairo, Egypt.

2.1. Materials

2.1.1. Shelling/Cracking Machine

The main components of the shelling/cracking machine included: frame, feed hopper, shelling/cracking chamber, discharge outlet and source of power, as shown in **Fig. 1**.

Frame of the Machine:

The main frame of the machine was constructed from iron channel sections. It is main dimensions were 50 mm depth, 25 mm flange width and 5 mm web thickness, the frame was jointed using helical screws to ease any modification and was fixed on four wheels to facilitate transfer of machine.

Feed Hopper:

The experimental hopper was constructed from galvanized iron sheet and composed of four sides. These sides were joined with helical screws. The two parallel sides of motor axis were fixed, while the other two sides are crossed and connected with joint from the top for ease articulation motion and control in feed slot. In addition; a gate prevents the fruits before the shelling/cracking chamber. The depth of hopper from top to the gate was 36 cm and the top area of hopper was 32.0 cm × 21.5 cm.

Shelling/Cracking Chamber:

The shelling/cracking process in the developed machine depends on friction or rubbing action. The shelling/cracking chamber was composed of; fixed part (external) and rotary part (internal), as shown **Fig. 2**. The fixed part is conical shaped and made from stainless steel (SS”304”) sheet with protrusions (1.6 mm height) to avoid slipping the fruits or seeds during shelling/cracking process. The conical shape was formed using plate rolling machine then was welded with the following dimensions: 35 cm upper internal diameter, 20 cm lower internal diameter and 37.5 cm depth. The conical part was put into a hollow square plate, in addition to four screws and four parts from steel angle sections to adjust the level of the conical part.

The rotary part consists of a conical block in the top and cylindrical block in the bottom. These blocks are made from low carbon steel (mild steel) and turned by lathe machine. The lower block was covered with the same material of fixed conical part. The thickness of sheet without protrusions is 5 mm. Also;

the shelling surface was formed using plate rolling machine, then was welded to take cylindrical shape with external diameter of 20 cm. The dimensions of fixed conical and rotary parts (friction surfaces) were chosen for the following reasons: Each height of 5 mm of the rotary part produces increases for the final clearance of 1 mm, to give clearance between two friction surfaces ranging from 0 to 75 mm and to possibly test of dry agricultural products which need the shelling/cracking. The SS material was used to form the two fraction surfaces for the following reasons: High toughness, acceptable forming and welding, high resistance to corrosion and rusting; in addition to, the SS material is safe for agricultural products. A screw rod with length of 57.66 cm was attached with the rotary part from bottom using two thrust ball bearings and was tied using nut into the rotary part. A handle was fitted in the bottom of the screw rod to adjust the required clearance. The screw rod is passed through a fixed nut in steel channel section under the shelling chamber. In addition, two moving nuts were put in the top and bottom of the fixed nut to lock the screw rod after adjusting the required clearance. A smooth rod with diameter of 25 mm and length of 58.9 cm was installed in the rotary part from the top through two bearings (FL 205). A pointer was installed above the smooth rod to indicate the value of clearance on a vertical scale; this scale was installed inside the frame at the top.

Discharge Outlet:

The discharge outlet was installed under the shelling chamber and composed of three parts: two parts were installed vertical into the frame on the two parallel sides of the motor, whose dimensions were 45.5 cm × 30.7 cm for both of them; whereas the other part has an inclination and was put to receive the shelled/cracked material, the dimensions of the inclined part were 61 cm × 25.6 cm and was installed with two hinges for ease motion and control in the angle of inclination. The discharge outlet was constructed from galvanized iron sheet with thickness of 0.5 mm.

Source of Power:

An electric motor "AC" was used to operate the shelling/cracking machine with power of 1.5 kW and speed of 1400 rpm. The power was transmitted to the rotary shaft using sprocket wheels and chains on three stages to reduce the speed of motor to the required speed.

2.1.2. Measuring Instruments

(1) An electric oven was used to determine the moisture content of pods, seeds and shells. (2) A digital electric balance with accuracy of 0.01 g was used to weigh the mass of samples before and after shelling. (3) A digital stopwatch with accuracy of 0.01 s to measure the time of shelling. (4) A digital Vernier-caliper with accuracy of 0.01 mm was used to measure the dimensions of pods and seeds. (5) A digital tachometer was used to measure the rotational speed "rpm". The specs of tachometer are as follows: Non-contact but; with laser photo, range of the measurement is 2.5 to 99999 rpm and its accuracies are 0.1 rpm through the speed 2.5 to 999.9 and 1 rpm over 1000 rpm. (6) A digital AVO-meter was used to measure the consumed electrical current (Amperes "A") during the shelling process. The specs of device are as follows: accuracy of device is 0.001, range of the measurement AC/DC voltage up to 600V and AC/DC current up to 10 A.

2.1.3. Sample of Peanut Pods

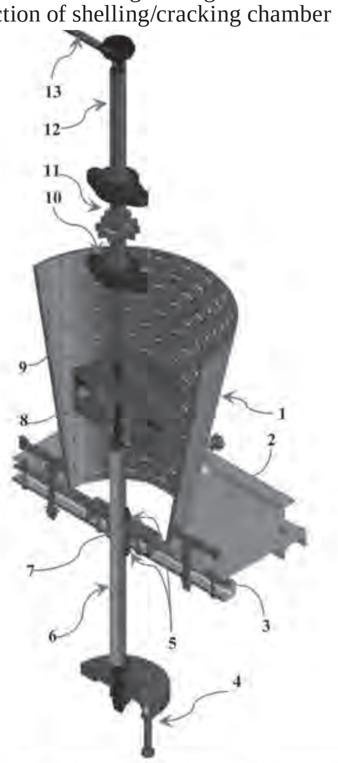
The peanut used in the shelling process is variety of American and cultivated in Egypt. Some physical properties of peanut were carried out such as (%-ages of peanut pods parts "%, by wt.", axial dimensions "mm", mass "g" and bulk density "g/cm3") for pods and seeds of peanut were studied.

2.2. Methods

2.2.1. The Experimental Procedure

The developed machine was evaluated using four clearances (C) were; 9, 10, 11 and 12 mm. For each clearance, four speeds were tested; 100 (1.047), 200 (2.094), 300 (3.142) and 400 rpm (4.189 m/s). These variables were tested using 200 g of PP switch batch system. Then; the optimal clearance and speed were constant at study the variable of sample mass. This variable was studied using five masses with batch system were; 200, 400, 600, 800 and 1000g. The experiments were carried out according the following steps: after adjustment of the required clearance and speed; the known mass of PPs was poured into the hopper above the gate. Then the machine was switched on and stayed for about one minute. Then the gate of hopper was removed quietly to allow the pods go to the

Fig. 2 3D drawing of longitudinal section of shelling/cracking chamber



1. Fixed part, 2. Base of shelling chamber, 3. Screw for balance, 4. Handel, 5. Two nuts to tie screw, 6. Screw rod, 7. Fixed nut, 8. Rotary part, 9. Protrusion, 10. Pillow block bearing, 11. Sprocket wheel, 12. Rotational shaft, 13. Pointer

shelling chamber under the gravity. At the end; the shelling product were carefully collected in collector, then; classified into five categories by manual separation; then weighed. These categories were as follow: unshelled pods (unshelled whole pods and partially shelled), broken seeds, split seeds, intact seeds, and shells with dust. At study of the sample mass variable; the shelling time was measured at the moment when it reaches the shelled pods to half input sample mass of PPs by using the balance under the collector. This value was multiplied by 2 to give the total time required for shelling the same input mass in case of continuous feeding. All experiments were replicated three times.

2.2.2. Performance Evaluation

Moisture Content:

The moisture content (M_C; "%, d.b.") of pods, seeds and shells were determined by drying method in a hot air oven at 130 °C for 7 hours. This test was repeated three times. The moisture content was determined by using the following equation:

$M_C = (m_b - m_a) / m_a \times 100$ (1)

Where; m_b: mass of sample before drying (g) and m_a: mass of sample afterdrying (g).

Performance Evaluation:

The following equations were used for the performance evaluation of shelling process of peanut pods:

$P_U = (M_U / M_t) \times 100$ (2),

$P_{BS} = (M_{BS} / M_t) \times 100$ (3),

$P_{SS} = (M_{SS} / M_t) \times 100$ (4),

$P_{IS} = (M_{IS} / M_t) \times 100$ (5),

$P_{ShD} = (M_{ShD} / M_t) \times 100$ (6), and

$\eta_{Sh} = [1 - (M_U / M_t)] \times 100$ (7).

Where; P_U: is the %-age of unshelled pods (%), M_U: mass of unshelled pods (g), M_t: total mass of input pods (g), P_{BS}: %-age of broken seeds (%), M_{BS}: mass of broken seeds (g), P_{SS}: %-age of split seeds (%), M_{SS}: mass of split seeds (g), P_{IS}: %-age of intact seeds (%), M_{IS}: mass of intact seeds (g), P_{ShD}: %-age of shells with dust (%), M_{ShD}: mass of shells with dust (g) and η_{Sh}: shelling efficiency (%).

Shelling Capacity (Productivity):

The shelling capacity (Q; "kg/h") was calculated according to the following equation:

$Q = (M_t / 1000) / T$ (8).

Where; T: is the time required to shell the sample (h).

Power Consumption:

The power requirement of shelling process (P; "W") was calculated using the following equation:

$P = I \times V \times \cos\varphi \times \eta_m$ (9).

Where; P: is the total power requirement with load (W), I: current consumed with load (Amperes), V: voltage difference (Volts), cosφ: power factor assumed 0.80, φ: phase angle between current and voltage and η_m: mechanical efficiency of motor assumed 85%.

The obtained results were statistically analyzed using spread sheet software program: Microsoft Excel and SPSS; V. "23".

Table 1 Some physical properties for pods and seeds of peanut

S, (rpm)	C,(mm)		Min.	Max.	Mean	SD	CV, (%)
M_C (%, d.b.)	Pods		7.29	7.54	7.44	± 0.13	1.76
	Seeds		5.86	5.96	5.89	± 0.06	0.94
	Shells		12.41	12.61	12.48	± 0.12	0.92
%-ages of PPs parts, (%, by wt.)	Seeds		74.31	75.81	74.91	± 0.79	1.06
	Shells		24.49	25.80	25.25	± 0.68	2.69
Axial dimensions, (mm)	Pods	L	23.70	52.89	39.19	± 5.59	14.26
		W	13.35	20.05	15.74	± 1.50	9.52
		T	11.39	17.42	13.98	± 1.14	8.12
	Seeds	L	13.36	23.30	20.17	± 2.24	11.09
		W	7.20	12.93	10.42	± 1.14	10.90
		T	6.29	10.41	8.78	± 0.77	8.79
Mass, (g)	Pods		1.25	3.46	2.40	± 0.58	24.09
	Seeds		0.30	1.36	0.97	± 0.22	22.27
Bulk density, (g/cm^3)	Pods		0.278	0.300	0.287	± 0.01	2.93
	Seeds		0.598	0.613	0.602	± 0.01	1.05

Results and Discussions

All experiments of shelling for peanut pods (PPs) were carried out under the average moisture content of 7.44, 5.89 and 12.48% d.b. for whole pods, seeds and shells, resp. Some physical properties of peanut were studied such as; %-ages of peanut pods parts (%, by weight), axial dimensions (mm), mass (g) and bulk density (g/cm^3) for whole pods and seeds of peanut as shown in **Table 1**.

Effect of Rotational Speed and Clearance on Performance Indicators

The shelling mixing of PPs output from used machine was separated manually to determine all %-ages of PPs parts as shown in **Fig. 3**. **Table 2** illustrates the average %-ages of peanut pods (PPs) parts after shelling process at tested four speeds [100, 200, 300 and 400 rpm]. For each speed (S) four clearances (C) were: 9, 10, 11 and 12 mm.

For the %-age of unshelled PPs:

Generally; the unshelled PPs decreased with increasing the speed and decreasing the clearance. The %-age of unshelled PPs decreased from 32.49 to 0.36% at speed of 400 rpm, 41.47 to 0.23% at speed of 300 rpm, 46.55 to 1.54% at speed of 200 rpm and 44.86 to 1.08% at speed of 100 rpm when decreasing the clearance from 12 to 9 mm as shown in **Table 2**. The Duncan Multiple-Range Test (DMRT) in **Table 3** showed that the mean effect of speed on the %-age of unshelled PPs. The mean values of unshelled PPs decreased significantly (at 5% level "$P < 0.05$") from 17.60 to 10.07% with increasing the speed from 200 to 400 rpm. Also; **Table 4** shows the result of Duncan's test for the mean effect of clearance on the %-age of unshelled PPs. The mean values of unshelled PPs decreased significantly (at 5% level "P < 0.05") from 41.35 to 3.25% with decreasing the clearance from 12 to 10 mm.

For the %-age of Broken Seeds:

From **Table 2** generally; the broken seeds decreased with increasing the clearance for all tested speeds. Duncan's test in **Table 3** showed that the mean effect of speed on the %-age of broken seeds. The mean values of broken seeds decreased significantly from 6.21 to 4.67% with increasing the speed from 100 to 200 rpm. Also; **Table 4** shows the result of Duncan's test for the mean effect of clearance on the %-age of broken seeds. The mean values of broken seeds decreased significantly from 9.00 to 1.84 with increasing the clearance from 9 to 12 mm.

For the %-age of Split Seeds:

Table 2 showed that the values of split seeds decreased with increasing the clearance and decreasing the speed. Duncan's test as shown in **Table 3** showed that the mean effect of speed on the %-age of split seeds. The mean values of split seeds decreased significantly (at 5% level "P < 0.05") from 23.76 to 9.89% with decreasing the speed from 400 to 200 rpm. Also; **Table 4** shows the result of Duncan's test for the mean effect of clearance on the %-age of split seeds. The mean values of split seeds decreased significantly (at 5% level "P < 0.05") from 24.43 to 5.86% with increasing the clearance from 9 to 12 mm.

For the %-age of Intact Seeds:

From **Table 2** the highest value of intact seeds was 53.51% at speed of 200 rpm and clearance of 10 mm. Duncan's test as shown in **Table 3** showed that the mean effect of speed on the %-age of intact seeds. The mean values of intact seeds were not significantly (at 5% level "P < 0.05") from speed of 100 to 300 rpm.

For the %-age of Shells with Dust:

The results showed that the %-age of shells with dust increased with decreasing the tested clearance as shown in **Table 2**. Duncan's test in **Table 3** showed that the mean effect of speed on the%-age of shells with dust. The mean values of shells with dust increased from 20.47 to 22.64% with increasing the tested speed from 100 to 400 rpm. Also; the mean values of shells with dust increased significantly (at 5% level "P < 0.05") from 14.82 to 25.13% with decreasing the clearance from 12 to 9 mm as shown in **Table 4**.

For Shelling Efficiency:

Fig. 4 shows the relationship between shelling efficiency (%) and clearance [9, 10, 11 and 12 mm] at four speeds [100, 200, 300 and 400 rpm]. Generally; the shelling efficiency of PPS gradually increased with decreasing the tested clearances and increasing the speeds. Duncan's test in **Table 5** showed that the mean values of shelling efficiency increased significantly (at 5% level "P < 0.05") from 58.65 to 96.75% with decreasing the clearance from 12 to 10 mm. Also; The mean values of shelling efficiency increased significantly (at 5% level "P < 0.05") from 82.40 to 89.93% with increasing the speed from 200 to 400 rpm; in addition to not significantly (at 5% level "P < 0.05") in mean values of shelling efficiency at speeds of

Table 2 The percentages of PPs parts after shelling process at tested speeds and clearances (mean value ± standard deviation)

S, (rpm)	C, (mm)	P_U, (%)	P_{BS}, (%)	P_{SS}, (%)	P_{IS}, (%)	P_{ShD}, (%)
100	9	1.08 ± 0.29	10.87 ± 1.63	13.56 ± 3.82	48.75 ± 1.92	24.54 ± 0.48
	10	4.90 ± 1.45	6.97 ± 1.99	9.66 ± 1.35	52.97 ± 3.08	24.06 ± 0.82
	11	24.77 ±4.16	5.23 ± 2.57	5.59 ± 1.01	44.02 ± 2.83	19.32 ± 0.61
	12	44.86 ± 4.20	1.78 ± 1.71	3.72 ± 0.69	29.38 ± 5.55	13.97 ± 0.92
200	9	1.54 ± 0.74	8.71 ± 3.62	16.11 ± 1.70	47.24 ± 2.18	25.70 ± 0.59
	10	4.88 ± 2.78	5.43 ± 1.04	12.15 ± 3.91	53.51 ± 2.58	24.36 ± 0.08
	11	17.43 ± 2.88	2.73 ± 1.47	7.58 ± 2.24	50.79 ± 0.78	20.80 ± 0.81
	12	46.55 ± 8.40	1.80 ± 1.25	3.73 ± 0.55	33.81 ± 5.12	13.33 ± 1.58
300	9	0.23 ± 0.40	7.16 ± 0.81	26.85 ± 3.49	39.49 ± 2.70	24.88 ± 0.70
	10	2.46 ± 1.19	5.30 ± 2.00	17.74 ± 0.55	49.15 ± 3.01	24.58 ± 0.55
	11	14.75 ± 2.82	3.94 ± 1.87	5.63 ± 0.71	52.97 ± 1.81	22.00 ± 0.58
	12	41.47 ± 5.31	1.86 ± 0.36	5.81 ± 0.68	35.02 ± 4.62	14.90 ± 1.56
400	9	0.36 ± 0.62	9.27 ± 1.24	41.21 ± 0.90	22.66 ± 1.22	25.41 ± 0.63
	10	0.76 ± 0.36	4.23 ± 1.93	27.74 ± 4.89	42.42 ± 3.18	24.66 ± 0.50
	11	6.65 ± 3.68	2.99 ± 0.86	15.90 ± 2.19	49.30 ± 2.28	23.40 ± 1.06
	12	32.49 ± 3.25	1.94 ± 0.93	10.18 ± 0.85	37.42 ± 1.45	17.09 ± 0.58

Fig. 3 Parts of PPs sample after mechanical shelling and manual seprating

(A) Unshelled pods "unshelled whole pods and partially shelled" (B) Broken seeds (C) Split seeds (D) Intact seeds (E) Shells with dust

Table 3 Duncan's test for the mean of effect of tested speeds on %-age of peanut pods parts

S, (rpm)	P_U, (%)	P_{BS}, (%)	P_{SS}, (%)	P_{IS}, (%)	P_{ShD}, (%)
100	18.91^{c}	6.21^{a}	8.13^{a}	43.78^{b}	20.47^{a}
200	17.60^{c}	4.67^{b}	$9.89a^{a}$	46.34^{b}	21.05^{ab}
300	14.73^{b}	4.56^{b}	14.01^{b}	44.16^{b}	21.59^{b}
400	10.07^{a}	4.61^{b}	23.76^{c}	37.95^{a}	22.64^{c}

Table 4 Duncan's test for the mean of effect of tested clearances on %-age of peanut pods parts

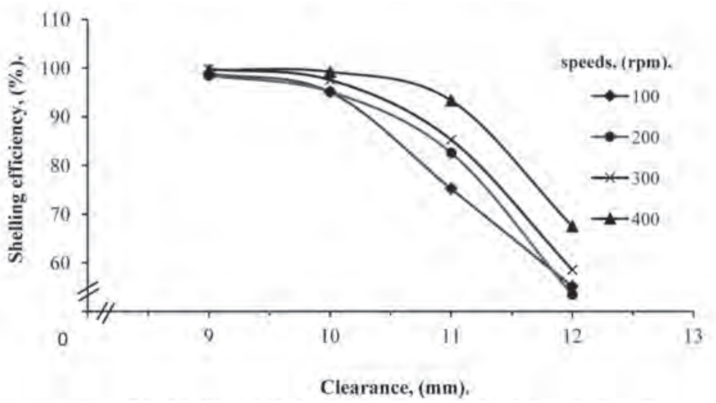
C, (mm)	P_U, (%)	P_{BS}, (%)	P_{SS}, (%)	P_{IS}, (%)	P_{ShD}, (%)
9	0.80^{a}	9.00^{d}	24.43^{d}	39.54^{b}	25.13^{d}
10	3.25^{a}	5.48^{c}	16.82^{c}	49.51^{c}	24.41^{c}
11	15.90^{b}	3.72^{b}	8.68^{b}	49.27^{c}	21.38^{b}
12	41.35^{c}	1.84^{a}	5.86^{a}	33.91^{a}	14.82^{a}

200 and 100 rpm and clearances of 10 and 9 mm. Whereas; the results indicated that the highest %-age of intact seeds was 53.51% at speed of 200 rpm and clearance of 10 mm as shown in **Table 2**. So; the optimal operation conditions are 200 rpm for speed and 10 mm for clearance at shelling of PPs.

Effect of Sample Mass of PPs on Performance Indicators

The variable of sample mass was studied with batch system under optimal operation conditions for speed of 200 rpm and clearance of 10 mm to investigate the effect of sample mass (200, 400, 600, 800 and 1000 g) on shelling efficiency and %-age of shilling mixing parts, productivity and power consumption of machine.

Fig. 4 Effect of the tested clearance on the shelling efficiency at different speeds



For Shelling Efficiency Under Tested Sample Mass Range:

Fig. 5 shows the relationship between shelling efficiency (%) and mass of sample (g); the trend of curve was roughly constant, in addition to Duncan's test in Table 6 showed the mean values of shelling efficiency were not significantly (at 5% level "P > 0.05"). The mean values of shelling efficiency ranged from 96.62 to 98.53% with average of 97.54% for all experiments.

For the %-age of Shilling Mixing Parts:

Generally; Duncan's test in Table 6 showed that the mean values of %-age of unshelled, broken seeds, split seeds, intact seeds and shell with dust were not significantly (at 5% level "P > 0.05") with average for all experiments were 2.46, 4.95, 14.52, 52.20 and 25.63%, resp.

For Productivity of Machine:

Fig. 6 shows the relationship between productivity of machine (kg/h) and mass of sample (g). Generally; the mean values of productivity increased from 21.11 to 48.29 kg/h with increasing mass of sample from 200 to 1000 g as shown in **Table 6**. The relationship between productivity of machine and mass of sample can be expressed as a power function by the following regression equation:

$Q = 1.79\ m^{0.47}$······ ($R^2 = 0.95$)

The highest value of machine productivity was 48.29 ± 1.95 kg/h at tested sample mass of 1000g. Meanwhile; the manual shelling of PPs was carried three times using

Fig. 5 Effect of the tested clearance on the shelling efficiency at different speeds

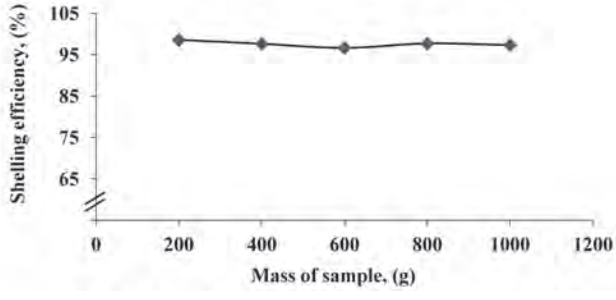
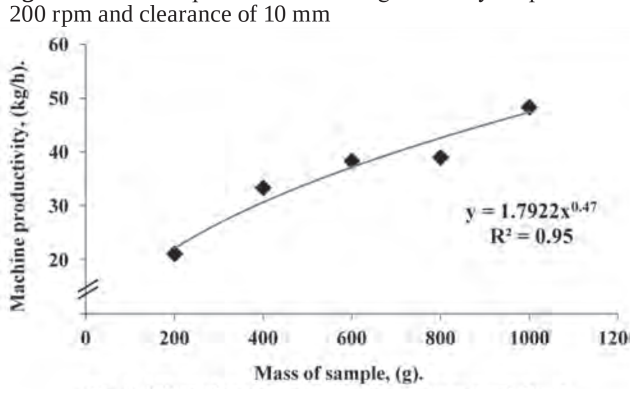


Fig. 6 Effect of sample mass on shelling efficiency at speed of 200 rpm and clearance of 10 mm



sample of 150 g. The results indicated that the productivity of manual shelling was 1.09 ± 0.19 kg/h.

For Power Consumption:

Duncan's test in **Table 6** showed that the mean effect of sample mass on the power consumption (W). The mean values of power consumption are not significantly (at 5% level "P < 0.05") from 200 to 800 g and significantly (at 5% level "P > 0.05") when increasing the sample mass from 800 to 1000 g.

Conclusions and Recommendations

- The used machine achieved high efficiency in shelling of PPs. Under the tested speed range [100 (1.047) to 400 rpm (4.189 m/s)] and the tested clearance range [9 to 12 mm]. The optimum operation conditions for shelling of PPs was 200 rpm for speed and 10 mm for clearance.
- Under the tested sample mass range [200 to 1000 g], the mean values of shelling efficiency were not significantly with average of 97.54%. Also; the mean values of %-age of unshelled, broken seeds, split seeds, intact seeds and shell with dust were not significantly with average of 2.46, 4.95, 14.52, 52.20 and 25.63%, resp.
- The results revealed that the highest value of machine productivity was 48.29 kg/h; whereas the mean value of power consumption was 689.59 W.
- To increase the productivity of machine at shelling of peanut pods preferable add a screw or blades from suitable material on drum of machine (rotary part).

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Table 5 Values of shelling efficiency at tested clearances and speeds (mean value ± standard deviation) and result of Duncan's test

C, (mm)	Shelling efficiency, (%)				
	S, (rpm)				
	100	200	300	400	Mean
9	98.92 ± 0.29	98.46 ± 0.74	99.77 ± 0.40	99.64 ± 0.62	99.20^a
10	95.10 ± 1.45	95.12 ± 2.78	97.54 ± 1.19	99.24 ± 0.36	96.75^a
11	75.23 ± 4.16	82.57 ± 2.88	85.25 ± 2.82	93.35 ± 3.68	84.10^b
12	55.14 ± 4.20	53.45 ± 8.40	58.53 ± 5.31	67.51 ± 3.25	58.65^c
Mean	81.10^c	82.40^c	85.27^b	89.93^a	--------

Table 6 Duncan's test for effect of tested masses on shelling efficiency, %-age of shilling mixing parts, productivity and power consumption

M_S, (g)		200	400	600	800	1000
η_{Sh}, (%)		98.53^a	97.59^a	96.62^a	97.67^a	97.28^a
%-age of shilling mixing parts	P_U	1.47^a	2.41^a	3.38^a	2.33^a	2.72^a
	P_{BS}	5.03^a	4.43^a	4.55^a	5.68^a	5.06^a
	P_{SS}	14.91^a	12.94^a	14.97^a	14.75^a	15.04^a
	P_{IS}	51.89^{ab}	54.69^b	51.81^{ab}	51.74^{ab}	50.88^a
	P_{ShD}	26.52^a	25.15^a	25.24^a	25.27^a	25.93^a
Q, (kg/h)		21.11^a	34.36^b	38.33^b	38.99^b	48.29^c
P, (W)		666.79^a	677.21^a	680.86^a	676.69^a	698.59^b

Power Requirement and Fuel Consumption Reduction of Forage Harvester Chopper Blades by Thermal Coating

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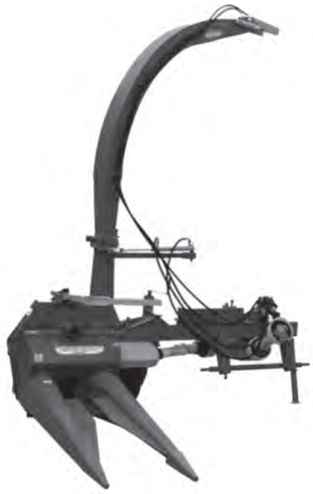
Abstract

The objective of this study wasto scrutinize the improvement of cutting efficiency of chopper blades used in forage harvesters and to determine the change in energy consumption by this means. The chopper blades used conventionally were coated thermally with two type coating powder alloys in the study. The coated blades and conventional ones were used on the same machine during all harvest season. Power requirement and fuel consumption of the harvest machine during harvesting were determined separately by using conventional and coated blades. It was determined that the blades coated with WC-Ni-Cr-Co-B-Si powder alloy (WC+Co) generated high rate of energy saving in comparison to both conventional blades and other coated blades. It was determined that conventional blades have higher fuel consumption than the coated blades. The fuel consumption of WC+Co powder alloy coated blades was approximately 22.5% lower than that of conventional blade. This difference increased as the harvested area increased. The fuel consumption difference between conventional blade and WC+Co powder alloy coated blades reached 3.89 Lh^{-1} (40.65%) when the harvested area reached 3 ha.

Keywords: Fuel cost, HVOF, Powder alloy, power requirement

Fig. 1 Single row forage harvester



Introduction

Silo forage is a gross forage source used for economic, balanced and adequate nourishment of ruminants and made of numerous products (Saner, 1993; Konca et al., 2005; Avcıoğlu et al., 1998; Kılıç, 1986). Particularly corn silage is used frequently for providing forage to ruminants due to its high efficiency and easily digestible property and its taste and storability (Kılıç, 1986, Komleh et al., 2011). More than 70% of silo forage is corn silage in World (Yaylak and Alçiçek, 2003). Harvesting is one of the most critical cost parameters in the production of silage (Barut et al., 2011). The plant must be harvested and chopped in an appropriate length to make silage. Forage harvester harvests forage plants to make silage (**Fig. 1**). Various types of harvesters are used in silage corn harvesting. Even though the machines used for harvesting are different in construction, the work is similar. Single row forage harvesters widely used in Turkey and also in the Middle East countries.

The plant is cut-chopped and conveyed during harvesting. Cutter/chopper blades are active components of forage harvesters for cutting and chopping plants. The blades apply cutting force at high rotation speed to the stem and chopping is made. It is essential that the plant is chopped in an appropriate length for a sought silage quality. The number of the blades, blade material properties and blade geometry are critical for quality of cutting and chopping (Reily et al., 2004). Chopping length, quality and especially needed well-compacting product during silage production is important for obtaining silage in sought characteristics (Komanduri et al., 1998). However, blades lose their cutting effectiveness due to wear like all machine components during cutting porcess. These worn blades with less cutting effect cannot chop neatly and the product chopping length homogeneity is lost. In this case, silage quality reduces and more power is required for cutting and chopping (Komanduri et al., 1998; Chen et al., 2004). Because of this, fuel consumption and carbon emissions increased (Houshyar et al., 2015).

The features of the blades used in forage harvester affect the machine efficiency and chopped product quality directly therefore, improvements of the chopper blades change machine effectiveness and the resulting product quality positively. One of these improvements is to increase wear resistance of the blades. Industrial solutions are applied in different sectors for this purpose. The most general and low cost of these solutions is coating the equipment surface with a harder material.

In this study, improvement of cutting effectiveness of the chopper blades used in forage harvesters was aimed. The coated blades werer used in single row forage harvesters in real harvest conditions.

Single row forage harvesters are mounted on a three point linkage of a tractor. The active component of forage harvesters is the chopper blades and there are 10 or 12 blades generally in a single row forage harvester (**Fig. 2**).

Forage harvester has a drum (cutterhead) or a flywheel with 12 chopper blades fixed. The chopped crop and the silage blows out a discharge spout of the harvester into a forage wagon that is connected either to the harvester or to another vehicle driving alongside. The dimension and features of the product harvested in forage harvester depends on the sharpness of the chopper blades and the sustained time of this sharpness (O'Dogherty, 1982).

Table 1 Values of material load in cutting devices common in agricultural cutting devices

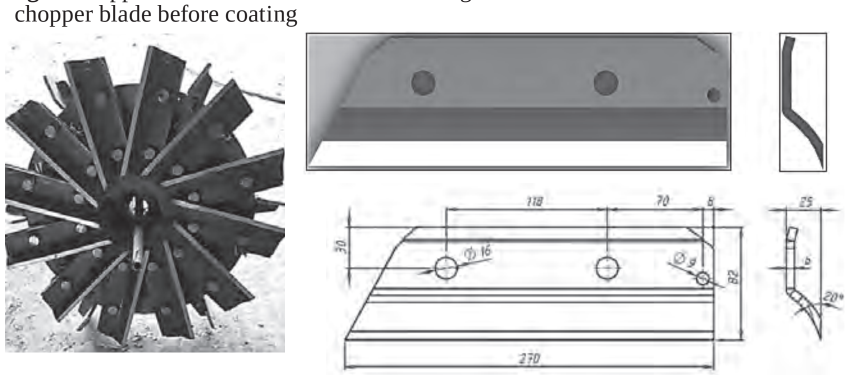
	Material load in cutting device (Mg mm^{-2})	Material solid thickness (mm)	Moisture content, wet basis (%)
Sickle-bar mower, grasses	0.22	0.15	74
Forage harvester (flywheel type), alfalfa	5.30	3.70	70
Forage harvester (cylinder type), alfalfa	3.00	2.00	70
Forage harvester, corn	11.00	7.60	60

Corn cutting force is high when compared with other products used for forage (**Table 1**) (Persson, 1987). The blades quickly blunt while the plant cutting force is high. Higher cutting force will be needed as the cutting blades become blunt (Prasad and Gupta, 1975; Xu and Yao, 2009). Blades are manufactured of medium carbon steel and generally hardened or selected areas (cutting edge) induction hardened to sustain their cutting features for a long time. Effectiveness of chopping blades diminishes after a certain operation time depending on some basic parameters including the variety of the chopped product, product moisture, harvested area, operator's skill, blade material quality and heat treatment of blades

Decreasing the cutting efficiency of the blades during harvest increases the shear force and increases the fuel consumption (Akritidis, 1974; Reily et al., 2004, Kathirvel et al., 2009, Mathanker et al., 2015). On the other hand, crushing and tattering occurs instead of cutting and chopping due to the loss of sharpness (Persson, 1987; Atkins, 2009). In this case, juice loss of the tattered and crushed corns increases relatively. Therefore, blades are grinded by a system equipped with a blade-grinding stone mounted on the machine for improving chopping efficiency of blunt blades. Grinding process is carried out for all blades simultaneously. This process goes on periodically until the cutting edge of the blades is completely worn out. Each grinding work causes the shortening of the cutting edge of the blades superficially than the amount of the lost material annually due to grinding. Furthermore, the effective usage period of the blades after each grinding decreases relatively.

It is critical to minimize sharpness losing of the chopper blades of forage harvesters and to form a controlled abrading mechanism for ensuring usage of the blades for a

Fig. 2 Chopper blade mounted disc used in single row harvester and dimensions of chopper blade before coating



longer time without grinding and for continuity of cutting effectiveness and machine efficiency (Lau et al., 2000). For this purpose, coating of the surface used in other industrial sectors with a harder material and improvement of abrasion resistance is a practical solution. Hence, a section of the surface of conventional chopper blades used in forage harvesters was coated with special powder alloys by using thermal spraying method. The abrading section of blade for improving of abrasion resistance and continuity of blade cutting effectiveness was coated with harder material than blade material.Thus, power requirement and fuel consumption of forage harvesters will goes down. It was reported that increasing in energy efficiency of forage harvester and diminishing greenhouse gas emission has been achieved by this way (Alluvione et al., 2011; Soltani et al., 2013; Khoshnevisan et al., 2014).

Table 2 Technical specifications of single row forage harvester used in study

Lenght, mm	2500 mm
Widht, mm	2300 mm
Height, mm	3500 mm
Shredder blades	12 pieces
PTO	Minimum 540 min^{-1}
Chopping crop lenght	5-20 mm
Attaching to tractor	With 3 point linkage sytem
Working speed	7-9 km h^{-1}
Working capacity	35 Mg h^{-1}

Table 3 Technical specifications of tractor used in harvest trials

Engine	
Model	Perkins 1104 C-44
Type	Diesel, Liqued-cooled
Power	67 HP/ 50 kW
Number of Cylinder	4
Cylinder volume	4400 cm^3
Fuel tank capacity	103 L
Power Take-off	
PTO Shaft	Rear PTO
Rotational speed	540 min^{-1}
Transmitted power	62 HP/ 46.2 kW
Power transmission	
Gear box	12 forward and reverse
Max. speed	40 km h^{-1}
Lift capacity	3692 kgf
Hydraulic pump flow	52.3 L min^{-1}

Materials and Methods

Harvesting studies were carried out in Izmir city located in the western part of Turkey. The harvest area is situated at 38°34' north latitudes and 27°02' east longitudes and its level from the sea level is approximately 10 m.

The operator, tractor and single row forage harvester was not changed during harvesting works for determinig power requirement and fuel consumption of blades. The chopper blades used in the machine were the conventional (uncoated) and the blades coated with a coating having two different powder alloys by using thermal spraying method.

Crop

A KWS6565 FAO 570 variety hybrid corn seed was used for harvest crop production. This variety is a very early variant. It can be grown as main crop or second crop and has no soil selectivity. Average crop yield in all area of study is carried out as 6.5 Mg ha^{-1} ± 0.01. Corn grown as a second crop was harvested by using single row forage harvester.

Single Row Forage Harvester

In the study was used single row forage harvester. Some technical specifications of the machine used in the experiments were given in the **Table 2**.

The power requirement and fuel consumption of the machine during harvesting was separately determined while working both conventional blade and coated blades.

Tractor

Landini Powerfarm 75 model tractor was used in the all harvest trials. The power of the tractor is 67 HP and the maximum torque is 272.6 Nm. The selected tractor is suitable for harvesting according to its technical specifications and is capable of operating with single row harvester (**Table 3**).

Blade Material and Coating Powder Alloys

The quality of the steel used for the blades on forage harvester is AISI 1040 and the hardening is carried out locally in the heat treatment furnance (Hardening at 845 °C and Normalizing at 540 °C). AISI 1040 steel is defined as medium carbon steel because of the middle amount of carbon it contains and commonly used for single row forage harvesters' blade (conventional blade). The chemical, physical and mechanical properties of AISI 1040 quality steel are given in **Table 4** and **Table 5** respectively.

The conventional blades' selected surface coated with a harder material to improve of abrasion resistance. Tungsten carbide (WC) among

Table 4 Chemical composition (content in %) of AISI 1040 steel

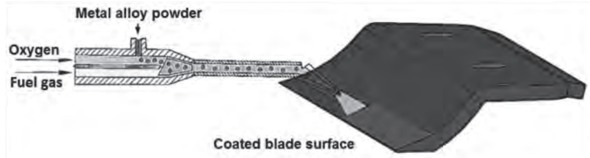
Carbon (C)	Silicon (Si)	Manganese (Mn)	Phosphorus (P) ≤	Sulfur (S) ≤
0.37-0.44	0.07-0.06	0.60-0.90	0.03	0.05

Table 5 Some mechanical and physical properties of AISI 1040 steel

Mechanical Properties		Physical Properties	
Young's modulus	200000 MPa	Thermal expansion	10.10^{-6} K^{-1}
Tensile strength	650-880 MPa	Specific heat	460 J kg^{-1} K^{-1}
Elongation	8-25 %	Melting temperature	1450-1510 °C
Fatigue	275 MPa	Density	7700 kg m^{-3}
Yield strength	350-550 MPa		

coating materials is preferred and used for improvement of abrasion resistance (Nascimento et al., 2001). The chemical composition and some physical properties of two different powder alloys are given in **Table 6**.

Fig. 3 Coating selected face of forage harvester chopper blade by using thermal spraying method



Methods

Coating

Thermal spraying method was selected for coating blade surfaces since its application is rapid and easy and its cost is lower in comparison to other methods (Tafralı, 2006; Pawlowski, 2008). The abrasion resistance of coating is directly related with the hardness of coating (Bozzi and de Mello, 1999). Two different powder alloys prepared with tungsten carbide and were used for improve effectiveness of blade. Coating powder alloys (A and B) were applied as a band form (selected surface) on the back side of the cutting edges of the chopper blades by using thermal spraying method (**Fig. 3**). Twelve blades were coated with each powder alloy and used in the machine.

Both powders were thermally sprayed on to degreased and grit blasted AISI 1040 Carbon Steel substrates to an average thickness of 630 µm. Coating were carried out using the HVOF (High velocity oxygen fuel thermal spray) system. The HVOF spraying method were performed by oxygen and acetylene as the fuel gas and two kinds of commercial powder alloys were used as coating materials (**Table 7**).

The chopper blades in forage harvesters were used by coding them in the following way (3 × 12 sets as conventional and coated);

- Blade S; is the conventional blade (with no coating) situated in the machine as mounted in the factory and used without making any changes.
- Blade A; is coated with coating material coded "A" powder alloys by thermal spraying application (12 blades were coated).
- Blade B; is coated with coating material codes "B" powder alloys by thermal spraying application (12 blades were coated).

Hardness, Microstructures and Element Analyses of Blade Material and Coating

The randomly selected cross-sections of coated blade were prepared to measure hardness, determination of microstructures and element contents. The cross-sections of coated blade cold-mounted in polyester resin were grinded using 2500 mesh SiC paper and diamond slurries (up to 0.5 µm) and were polished by colloidal silica slurry. Vickers micro hardness (HV/0.5/20) measurement test was made on cross-sections of blades and coating separately. The measurements were repeated three times for the thermal coating and main material.

The microstructures of coating and blade material (AISI 1040) were studied by scanning electron microscopy (X-ray diffraction, Rigaku-Rint 2200/PC-Ultima 3). Additionally, Energy Dispersive Spectroscopy (EDS) technique (JEOL JSM-6060) was used for elemental analysis of blade and coating materials.

Determination of Fuel Consumption

A total 3.0 ha area was harvested for each blade set during harvest. While the power requirement was determined during harvesting and

Table 6 Chemical composition of coating A and B powder alloys

Designation of powder	Composition of weigh percent %	Element	Weight Percent %	Manufacturing method	Particle size µm	Particle speed m s^{-1}
A powder alloys	60	Carbon	0.7-0.8	Agglomerated and sintered	20-45	700-800
		Boron	3.0-3.4			
		Silicon	3.9-4.9			
		Iron	2.4-4.6			
		Chrome	14.0-16.0			
		Nickel	Balance			
	40	Carbon	3.6-4.3			
		Cobalt	11.0-13.0			
		Tungsten	Balance			
B powder alloys	88	Carbon	0.7-0.8			
		Boron	3.0-3.4			
		Silicon	3.9-4.9			
		Iron	2.4-4.6			
		Chrome	14.0-16.0			
		Nickel	Balance			
	12	Carbon	3.8-4.3			
		Tungsten	Balance			

Table 7 Technological parameters of the high velocity oxygen fuel thermal spray (HVOF) process

Powber Alloys	Fuel flow rate L min^{-1}	Oxygen flow rate L min^{-1}	Powder feeding rate g min^{-1}	Spraying distance mm
A	240	900	90	370
B	210	850	80	370

fuel consumption was also measured simultaneously and the data were recorded.

At first total fuel consumption was determined during running of the tractor at its own operation speed and pulling forage harvester (idle running of harvester) and agricultural trailer. Average fuel consumption values during idle running were subtracted from total fuel consumption data were obtained during harvesting and calculated as Lh^{-1}. The fuel consumption measurements of blade sets A, B and S.were repeated three times for every 0.2 ha (± 0.001 ha). Total 3.0 ha of crop area was harvested during fuel consumption trials.

Flow meters with two ways (input-output) (KRACHT Gear Type VC-Germany) were used for determining fuel consumption of forage harvester powered by tractor during harvest (**Fig. 4**). Flow meters produce an electronic pulse for each volumetric passing (0.025 cm^3) of fuel used by tractor engine. Fuel consumption was calculated by using produced pulse value and constant volumetric flow value (Lh^{-1}). The difference between the fuel coming from the fuel tank and the fuel (unused) returning to the tank during harvesting was determined. Fuel consumption of each blade (S-A-B) data was measured for every 0.2 ha (± 0.001 ha) in total 3.0 ha harvested area. Power requirements and fuel consumption measurements were imlpemented simultaneously during all harvesting trials.

Determination of Power Requirement

Silage corn harvesting was made at approximately 5.3 km h^{-1} ± 0.02 working speeds. All data obtained from from trials were recorded by data logger. The 9.0 ha total crop area was harvested during all tests. Torque and revolution measurements were made in real time on the PTO shaft, which transmits the power to the forage harvester during harvesting, and the power requirements of forage harvester was calculated by using Equation 1. For this purpose, Datum 420 PTO model torquemeter and Grant 2020/2040 model Squirrel data logger were used to collect data during silage corn harvesting studies (**Fig. 5**). Torque and revolution measurements for each blade set (S-A-B) were repeated for every 0.5 ha ± 0.001 in harvested area.

Power (kW) = (Rotational speed (min^{-1}) × Torque (Nm)) / 9549 (Eq. 1)

Silage corn harvester need average of 10 kW power during idle running (running at 540 min^{-1} without any cutting and chopping process) and average net power requirement values of blades were estimated by deducting this determined power value from the value of the machine during harvesting.

Determination of Chopping Length

Chopping size analysis was carried on samples comes from the chopped crop during harvesting. For each trials 2 kg chopped crops taken from the harvesting discharge spout exit during harvesting were sieved in laboratory conditions using a 6-fold sieve system with different hole diameters (**Table 8**). The 6-fold sieve system operated at 70 min^{-1} revolution and rotated by an electric motor.

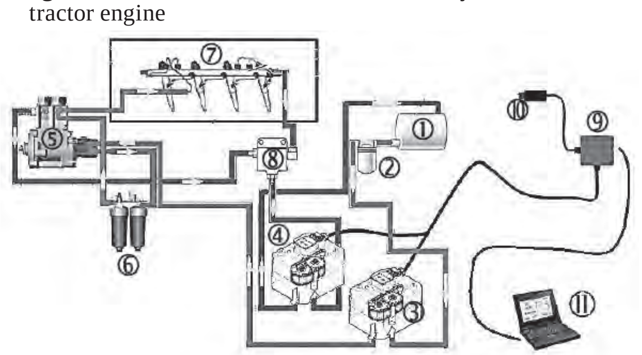
The weighted mean chopping lenght of blades for harvested crop was calculated using Equation 2.

Weighted Mean Chopping Lenght (mm) = [(First fold mean diameter (mm) × First fold sample weight (g)) + (Second fold mean diameter (mm) × Second fold sample weight (g))... + (Sixth fold mean diameter (mm) × Sixth fold sample weight (g))] / Total Sample Weight (g) (Eq. 2)

Statistical Anaylsis

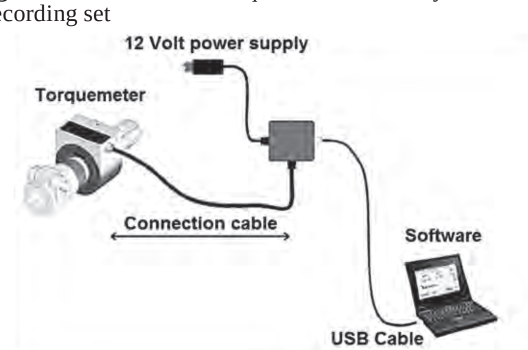
The test was planned according to shape of crop area and blades for trial to be mounted on the machine were selected coincidentially. The arithmetic means and SEM values of data were calculated. ANOVA mode and Duncan's multiple range tests were used for the statistical

Fig. 4 Schematic view of fuel measurement system used on tractor engine



(1) Fuel tank, (2) Primary fuel filter, (3) Measurement of fuel coming from the tank, (4) Measurement of fuel returning to the tank, (5) Fuel pump, (6) Secondary fuel filter, (7) Fuel injectors, (8) Collection pump, (9) Data recording device, (10) Power supply (12V), (11) Computer

Fig. 5 Schematicview of torque measurement system and data recording set



evaluation of results at $p < 0.05$ level. For all statistical analysis MS Excel and SPSS V15 package program was used.

Results

Vickers micro hardness measurement test was carried on the coating A because of it has the best performance. The other coating powder alloys did not show the expected performance, so was not tested. Additionally, the blade material was analyzed during the coating A analysis.

The average material hardness (HV/0.5/20) values for blade and coating A are 290.6 HV and 445.6 HV respectively (**Fig. 6**). The hardness of coating was found higher than the main material as expected. The hardness of the coating A of blades was 53% higher than the hardness of the main material. Thus, due to the structure with two different hardnesses on the sharp edge of the blade its self-grinding was ensured without any need for grinding during operation. No superficial grinding process was applied to the blades including conventional blades along all harvesting studies.

Coated Powder Alloy Characterization

The microstructures of coating A and blade materialwere studied by scanning electron microscopy corresponding to the black rectangle in **Fig. 7**. The results of EDS element analysis was shown that area numbered one and two area contains of all present phases. These areas have more Chromium (Cr), Nickel (Ni) and Boron (B). So, it was indicated that by high amounts of Boron and Nicel could act as a barrier for linking the unreacted alloy particles to the matrix phase (**Table 9**).

Fuel Consumption

Fuel consumption values during harvesting made by single row harvester with blades A, B and S increased based on harvested area for each three blade set (**Table 10**). However average fuel consumption of Blade A lower than that of the other two blades (B-S). Decrease in average fuel consumption started after 1.8 ha of harvest area was reached particularly for blade A. This finding was similar to the decrease seen in power requirement. This can be explained as a proof the sharp edge regained its cutting effectiveness after certain abrasion since the coating on the blade was harder than the main material. Fuel consumption for the other two blades (B-S) increased with an increase in the harvested area. **Table 8** shows net fuel consumption and standard error values estimated based on the average values of the data obtained from blades A, B and S. Fuel consumption for blades A, B and S at the end of 3 ha harvest made by each blade set were 5.68 Lh^{-1}, 8.73 Lh^{-1} and 9.57 Lh^{-1} respectively. Both blade type and area-based change in fuel consumption was found statistically significant at

Table 8 The 6-fold sieve system with different hole diameters

6^{th} fold (top)	5^{th} fold	4^{th} fold	3^{th} fold	2^{nd} fold	1^{st} fold (bottom)
80-40 mm	40-20 mm	20-10 mm	10-5 mm	5-2.5 mm	Pan

Table 9 EDS (Energy Diffraction Spectroscopy) element analysis values performed on the coating A and blade material

Material	Element	Mole Concentiration	Concentiration wt %	Intensity c s^{-1}	Error 2.sig
Coating A	Si	3.349	6.565	28.05	1.059
	Cr	3.027	10.984	24.31	0.986
	Ni	2.800	11.471	12.92	0.719
	Fe	0.645	2.516	4.09	0.404
	Co	0.045	0.184	0.24	0.098
	W	0.022	0.281	0.05	0.044
	B	90.112	68.000	3.47	0.372
		100.000	100.000		
Blade	Si	3.349	6.565	28.05	1.059
	Cr	3.027	10.984	24.31	0.986
	Ni	2.800	11.471	12.92	0.719
	Fe	0.645	2.516	4.09	0.404
	Co	0.045	0.184	0.24	0.098
	W	0.022	0.281	0.05	0.044
	B	90.112	68.000	3.47	0.372
		100.000	100.000		

kV: 20.0, Take of Angle: 35.0°, Elapsed Livetime (μs): 75.2

Fig. 5 HV tracks in the blade material and coating A

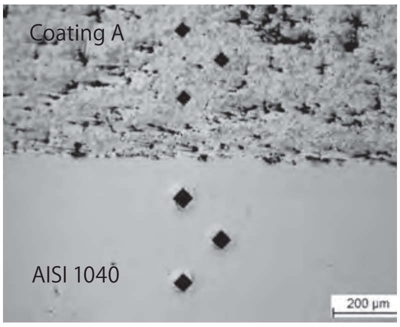
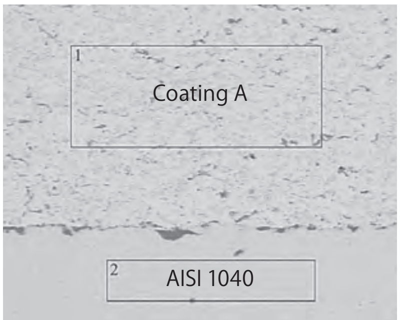


Fig. 7 SEM overviews (back scattered electrons) of the polished cross-sections of the coating A and blade material



p < 0.05 level.

Blade S had higher net fuel consumption in comparison to the other two blades (A and B). The average net fuel consumption of Blade A was 1.65 Lh^{-1} lower than that of blade S and 1.33 Lh^{-1} lower than that of blade B. Especially when net fuel consumption of blade A is compared with that of blade S it is approximately 22.5% lower. When the harvested area is 3.0 ha the average net fuel consumption differences between blades S and A reached 3.89 Lh^{-1}. This means that fuel consumption decreases 40.65% in comparison to the fuel consumption of blade S in a total of 3 ha area. The findings obtained in the study points out that Blade A which has an average net fuel consumption of 5.68 Lh^{-1} at the end of 3.0 ha harvest has a significant superiority to Blade S which is used as conventional (**Table 10**).

Power Requirement

Average net power requirement increased based on harvested area in three blades. However, it was determined that the average power-consumption value of Blade A was different from the change in the values of the other two blades (B-S). As shown in **Table 11**, as the harvested area increased, blade-cutting effectiveness decreased and power need increased based on this. However this increase continued until 2.0 ha-harvesting areas in Blade A and the coated sharp edge completed its self-grinding process after it reached this area and depending on this power requirement decreased. This situation can be assessed as the sharp edge regaining its cutting angle because the coating on the blade is harder than the main material (Balde S). There was an increase in power requirement in the other two blades (B-S) with an increase in the harvested area. **Table 7** shows the average net power requirement values of blades A, B and S during silage corn harvest and standard error values of the average. Power requirements for blades A, B and S at the end of 3 ha harvest made by each blade set were calculated 22.35 kW, 28.04 kW and 28.09 kW, respectively.

Change in power requirement based on both blade type and area was statistically significant ($p < 0.05$). These results show that power requirements of the blades are statistically different. On the other hand, Duncan's multiple range tests were achieved at $p < 0.05$ level for grouping the difference. As shown in **Table 7**, conventional blade (S) had higher power requirement in comparison to blades A and B. The difference between the average power requirements of the blades for blade A in comparison to blade S was 3.41 kW (13.5%) lower and it was 2.72 kW (11.1%) lower for blade B (**Table 11**). This situation becomes even more prominent as the harvested area increases. When the harvested area reached 3.0 ha, the power requirement difference between blade S and blade A reached 5.74 kW. This value shows that the power requrement decreased 20.3% in comparison with conventional

Table 10 Average fuel consumptions and Duncan grouping of blades A, B and S during corn harvest

Area ha	Blade A		Blade B		Blade C	
	Net fuel consumption Lh^{-1}	± SEM	Net fuel consumption Lh^{-1}	± SEM	Net fuel consumption Lh^{-1}	± SEM
0.2	4.36	0.09	5.22	0.08	4.56	0.06
0.4	5.23	0.07	5.94	0.08	4.91	0.04
0.6	5.93	0.11	6.57	0.08	6.33	0.07
0.8	5.73	0.09	6.81	0.07	6.63	0.09
1.0	5.48	0.14	6.43	0.09	6.21	0.08
1.2	4.94	0.09	6.62	0.08	6.45	0.07
1.4	5.52	0.13	7.23	0.06	7.16	0.14
1.6	5.76	0.11	6.74	0.07	7.21	0.11
1.8	6.62	0.12	7.65	0.09	8.26	0.09
2.0	6.34	0.11	6.96	0.06	8.00	0.10
2.2	6.18	0.11	7.85	0.08	8.51	0.10
2.4	6.41	0.12	7.26	0.08	8.99	0.13
2.6	5.79	0.13	7.57	0.09	8.79	0.09
2.8	5.29	0.10	7.56	0.08	8.32	0.07
3.0	5.68	0.08	8.73	0.07	9.57	0.10
	$\bar{x}$: 5.68[a]		$\bar{x}$: 7.01[b]		$\bar{x}$: 7.33[c]	

Means in the same row followed by different uppercase letters are significantly different (probability p < 0.05) according to Duncan's multiple range tests

Table 11 Average power requirement of blades A, B and S during crop harvest

Harvested area ha	Blade A		Blade B		Blade C	
	Net power consumption kW	± SEM	Net power consumption kW	± SEM	Net power consumption kW	± SEM
0.1	17.89	0.56	20.55	0.55	20.35	0.18
0.5	18.89	0.61	22.29	0.50	23.29	0.30
1.0	20.92	0.60	23.73	0.54	24.71	0.32
1.5	23.79	0.39	24.57	1.08	25.66	0.42
2.0	24.51	0.44	25.53	0.73	26.09	0.36
2.5	23.50	0.39	26.14	0.64	27.54	0.30
3.0	22.35	0.31	28.04	1.37	28.09	0.42
	$\bar{x}$: 21.69[a]		$\bar{x}$: 24.41[b]		$\bar{x}$: 25.10[c]	

Means in the same row followed by different uppercase letters are significantly different (probability p < 0.05) according to Duncan's multiple range tests.

blade.

Particle Size Distribution of Harvested Crop

In the crop harvest trials conducted with all three blades, the particle distribution was carried out in predominantly in the range of 0-20 mm chopping length. The weighted mean chooping lenght of the blades were close to each other but blade A showed better performance (91.8%) as aparticle distribution rates (**Table 12**).

Economic Evaluation

Coating cost of the blades is approximately half of the obtaining cost. Coating cost of the prototype study was $10 US (including VAT) per blade. This cost reduces even more in serial production. Twelve blades were coated for each chopping set. Price of fuel (diesel oil) used by agricultural tractors in Turkey is approximately $1.3 US L^{-1}. Total fuel cost of the harvest in 3 ha area made by the chopping blades in the study was $65 US (49.42 L) $100 US (75.95 L) and $109 US (83.26 L) for blades A, B and S respectively (**Table 13**). The fuel cost difference for blades A and S for 3 ha harvest was $44 US.

Duscussion

A portion of the cutting surface of the chopper blade used in single row forage harvester was coated with different powder alloys by using thermal coating method. Chopping action made during silage corn harvest-process is an event causing abrasion of the blades rapidly. In this study the coated surface is harder than the other portion of the blade in order to improve abrasion resistance of the blades during chopping. In the study was predicted that relatively soft blade material will wear off before than the hard coating of cutter edge of blade. This situation ensured self-grinding of the blade cutting edge. In the study, no measurement was made to determine how the blade-cutting angle changed. But the results gave aspect that the cutting angle of the sharp edge was protected for a long time since the hard coating was abraded less in comparison to the blade main material during cutting process. Therefore, there was no need for grinding process for sharpening of blades which were needed to be applied in the conventional method by forming a self-grinding mechanism.

Blade S had higher net fuel consumption in comparison to the

Table 12 Sieve analysis results of crop chopping lenght and the distribution

Blade	Weighted mean chopping lenght mm	Particle size distribution range of 0-20 mm %
S	14.2	87.2
A	13.9	91.8
B	14.3	86.6

Table 13 Total fuel consumptions of blades A, B and S depending on the harvest area

Harvested , ha	Blade A				Blade B				Blade C			
	Fuel consumption, Lh^{-1}	Cumulative harvesting time, h	Cumulative fuel consumption , L	Fuel consumtion, L ha^{-1}	Fuel consumption, Lh^{-1}	Cumulative harvesting time, h	Cumulative fuel consumption , L	Fuel consumtion, L ha^{-1}	Fuel consumption, Lh^{-1}	Cumulative harvesting time, h	Cumulative fuel consumption , L	Fuel consumtion, L ha^{-1}
0.2	4.36	0.58	2.53		5.22	0.58	3.03		4.56	0.58	2.64	
0.4	5.23	1.16	6.07		5.94	1.16	6.89		4.91	1.16	5.70	
0.6	5.93	1.74	10.32		6.57	1.74	11.43		6.33	1.74	11.01	
0.8	5.73	2.32	13.29		6.81	2.32	15.80		6.63	2.32	15.38	
1.0	5.48	2.90	15.89	**15.89**	6.43	2.90	18.65	**18.65**	6.21	2.90	18.01	**18.65**
1.2	4.94	3.48	17.19		6.62	3.48	23.04		6.45	3.48	22.45	
1.4	5.52	4.06	22.41		7.23	4.06	29.35		7.16	4.06	29.07	
1.6	5.76	4.64	26.73		6.74	4.64	31.27		7.21	4.64	33.45	
1.8	6.62	5.22	34.56		7.65	5.22	39.93		8.26	5.22	43.12	
2.0	6.34	5.80	36.77	**20.88**	6.96	5.80	40.37	**21.72**	8.00	5.80	46.40	**28.39**
2.2	6.18	6.38	39.43		7.85	6.38	50.08		8.51	6.38	54.29	
2.4	6.41	6.96	44.61		7.26	6.96	50.53		8.99	6.96	62.57	
2.6	5.79	7.54	43.66		7.57	7.54	57.08		8.79	7.54	66.28	
2.8	5.29	8.12	42.95		7.56	8.12	61.39		8.32	8.12	67.56	
3.0	5.68	8.70	49.42	**12.64**	8.73	8.70	75.95	**35.58**	9.57	8.70	83.26	**36.86**

other two blades (A and B). The average net fuel consumption of Blade A lower than that of blade S and blade B. Especially when net fuel consumption of blade A is compared with that of blade S it is approximately 22.5% lower. When the harvested area reaches 3.0 ha the average net fuel consumption difference between blades S and A reached 3.89 Lh^{-1}. This means that fuel consumption of blade A decreases 40.65% in comparison to the fuel consumption of blade S in a total of 3 ha area. The findings obtained in the study points out that Blade A which has an average net fuel consumption of 5.68 Lh^{-1} at the end of 3.0 ha harvest has a significant superiority to Blade S which is used as conventional. The average net power requirement was 21.69 kW for blade A. It was 24.41 kW for blade B and 25.10 kW for blade S.

Blade A was coated with WC-Ni-Cr-Co-B-Si powder alloy. It was reported that WC within powder alloy used in coating improved anti-adhesion and corrosion resistance characteristic as well (Mo and Zhu, 2011; Bolelli et al., 2014; Liao et al., 2000). The best harvest performance was achieved by blade A in terms of fuel consumption and power requirement according to these data. The abrasion resistance of blade A was improved by coating it with WC-Ni-Cr-Co-B-Si powder alloy. The coating of blade B could not holded on to the main material surface and was separated from the surface. This is the reason why seperating of coating from surface was seen locally during the study due to the lack of Co which has a binding characteristic within the powder alloy used for the coating of blade B and/or inadequate amount of similar alloy elements. Additionally Cobalt eliminates thermal effect during coating (Geng et al., 2015).

While the harvested area increased, the difference between the total fuel consumptions of the chopping blades increased. On the other hand, the grinding system that is used conventionally will not be added to the machine and the machine production cost will decrease since there is no need for a manual grinding process. In addition, controlled abrasion achieved by coating of the chopper blades will eliminate material loss formed by superficial abrasion made in the conventional method. This situation means that usage time of the chopper blades increases relatively. This means that the period of use of the chopper blades has increased relatively. However, in order to consolidate these considerations, work should be continued using different coating materials in larger harvest areas.

Conclusions

The main outcomes of this study can be summarized as continuoing usage of chopper blades without manual grinding, reducing the fuel consumption and power requirements of forage harvester and demonstrate the economic impact of the new implementation.

Tungsten carbide-cobalt which is commonly used cotaing powder alloy in other sectors was used for improving abrasion resistance of chopping blade. Coating cost of the blades is approximately half of the obtaining cost as mentioned before. This cost can be reduced with serial production. In addition coating cost is acceptable when considering the saving considerable fuel consumption. For instance the fuel cost difference for blades A (coated) and S (conventional) for solely 3 ha harvest is $44 US. Beside this, removing the grinding system from the machine design and reducing the amount of material worn by grinding will reduce operatioanl costs. Both of these solvings are serious saving in forage harvest for farmers input costs.

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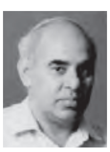
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Back Issues

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

(Vol.49, No.4, Autumn, 2018-)

◇ ◇ ◇

Vol.50, No.2, Spring 2019

Current Situation of Agricultural Mechanization and Conservation Agriculture in Latin America (P. P. Rondón, Y. M. Mesa, H. C. Fernandes, M. V. G. Águila, A. M. Caballero) ... 13
Current Status and Future Prospect of the Agricultural Mechanization in Brazil (E. C. Mantovani, P. E. B. de Oliveira, D. M. de Queiroz, A. L. T. Fernandes, P. E. Cruvinel) 20
The Valorization of Embedded Technology in the Sprayers to Obtain Operating Gains and Pulverization Quality (Marcella Guerreiro de Jesus) ... 29
Situation of Agricultural Mechanization in Argentina - A Perspective (H. A. Cetrangolo).. 33
Present Status and Future Prospects of Agricultural Mechanization in Mexico (H. Ortiz-Laurel, D. Rosas-Calleja, H. Debernardi de la Vequia) ... 40
Current Situation of Agricultural Mechanization in Mexico (E. R. Carbajal, G. H. Cuello, O. G. Mejía) ... 46
Mechanization of Irrigation in Latin America (G. H. Cuello, E. R. Carbajal, J. P. Petitón) · 52
Ecuador: Current Mechanization Status and Issues That Rice Producers Facing Now (A. Utsunomiya) ... 57
Agricultural Mechanization in Ecuador (L. Shkiliova, C. E. I. Coronel, R. X. C. Mera) 72
Mechanization of Cassava Cultivation (Manihot Esculenta L., Cranz) in Venezuela in View of Its Physical-Mechanical Properties (A. G. Pereira, E. P. Motta, A. H. Gómez, J. G. Coronado, M. L. Acosta) ... 78
Abrasive Wear Assessment under Laboratory Conditions in Disks of Tiers Used in Venezuela (A. H. Góez, I. J. M. Ortiz, A. G. Pereira, J. G. Coronado) ... 83
Future Trends in the Chilean Agricultural Machinery Industry (M. A. López, C. Correa, E. J. Hetz) ... 88
Present Status and Prospects of Agricultural Mechanization in Cuba (Y. M. Mesa, P. P. Rondón, L. J. S. Diaz) ... 94

◇ ◇ ◇

Vol.50, No.3, Spring 2019

Effect of Rotary Plough and Precision Land Levelling on Faba Bean Response to Organic Fertilization (O. T. Bahnas, M. Y. Bondok) ... 7
Physico-Mechanical Properties of Cassava Stem as Related to Cutting (Sahapat Chalachai, Peeyush Soni) ... 14
Development of a Watermelon (*Citrlluslanatus*) Seed Extractor (Shrinivas Deshpande, G. Senthil Kumaran, A. Carolin Rathinakumari) ... 23
Predicting Wheat Harvest Time Using Satellite Images and Regression Modelsr (Sepideh Taghizade, Hossain Navid, Yasser Maghsod, Mohammad, Moghadam Vahed, Reza Fellegarii) ... 28
Design, Development, and Evaluation of a Fuzzy-based Automatic Guidance System for JD955 Combine Harvester (Alireza Mahdavian, Saeid Minaei, Ahmad Banakar) ... 34
Seed Drill Discharge Rate Variation Due to Varietal Differences Using an Automated Calibration Test Rig (Daanvir Karan Dhir, Pradeep Rajan, S. R. Verma) ... 43
Optimization of Combine Crop Parameters for Paddy Harvesting by Head Feed Combine (Ingole Omprakash Avdhut, Anil Kumar, Vijaya Rani, S. Mukesh, Nitin Kadwasra, Kanishk Verma) ... 48
Development of a Chopping Apparatus for Cactus Prickly Cladodes (H. M. Duran-Garcia, U. Marquez-Perez, E. J. Gonzalez-Galvan, E. D. Rössel-Kipping, H. Ortiz-Laurel) ... 58
Modification of Rotary Power Tiller units for Biasi (Interculture Operation) Rice Cultivation in Eastern India (Ajay Kumar Verma, Samir Santiya) ... 62
Design and Experiment of Associated Baler for Combine Harvester (Li Baoqian, Wu Gaofeng, Liu Yunpeng, Lv Fengzhao, Shi Dongsheng) ... 69
The Role of Agricultural Mechanization in the Process of Modernization of Agriculture in Vietnam - Contribution of Agricultural Engieering to Production After Years of Conducting Renovation (Pham Van Lang, Nguyen Hay, Đo Thi Tam, Nguyen Tien Han) ... 79

◇ ◇ ◇

Vol.50, No.4, Autumn 2019

Development of an Efficient Fruit Cum Vegetable Grader for Spherical Commodities (S. Mangaraj, R. K. Pajnoo) ... 7
Design and Experiment of a Fertilizer Deep Applicator for Twin-row within One Ridge (Wu XueMei, Guy Fipps, Fugui Zhang, Xu Li, DeLong Fu) ... 13
Current Situation and Perspectives of Education for Agricultural Mechanization in the Republic of Buryatia of the Russian Federation (Mikhail Dorzhiev, Hideo Hasegawa, Tsyden Sandakov, Nadezhda Sandakova, Konstantin Luzbaev) ... 20
Design and Development of Thresher for Onion Umbels (*Allium Cepa Variety Aggregatum L.*) (M. M. Pragalyaashree, R. Kailaapan, Z. John Kennedy) ... 25
Influence of Surface Hardening with Carbon Nanotubes- Hard Chrome Composite on Wear Characteristics of a Simple Tillage Tools (A. M. Zein El-Din, Saad F. Ahmed, M. A. Khattab, R. G. Abdel Hamied) ... 32
Research on a Method to Measure and Calculate Tillage Resistance of Tractor Mounted Plough (Han Jiangyi, Lin Cunhao) ... 38
Automatic Seed Cum Fertilizer Drill: Modification and Performance Evaluation for Intercropping (Ajay Kumar Verma, Mukesh Kumar Pandey) ... 44
Manufacturing and Testing the Performance of Prototype for Grading of Dates (Said Elshahat Abdallah, H. M. Sorour, A. M. Deris, Awad Ali Tayoush) ... 49
Impact of Slice Size on Kinetic Behavior and Drying Time of Fresh-Cut Apple (*Malus domestica*) (Destiani Supeno, Pandu Sandi Pratama, Won-Sik Choi) ... 61
Designing and Testing an Innovative Soybean Seed Grader with Oval-hole Screen Type (I. K. Tastra, Uning Budiharti, N. R. Patriyawaty) ... 65
Yield and Economics Attributed Study of Direct Seeding and Transplanting Method on Beds for Onion (*Allium Cepa L.*) Cropwith Pneumatic Precision Multicrop Planter and Manual Transplanting Method Along With Rotary Tiller Cum Bed Former in Indian Conditions (Surinder Singh Thakur, Manjeet Singh, Rupinder Chandel) ... 76

◇ ◇ ◇

Vol.51, No.1, Winter 2020

Evaluation Parameters Afeccting the Performance of Vibrating Vertical Tillage Equipment – First Stage (Guillen Sánchez Juan, Santos G. Campos Magaña, Carlos Sánchez López, Oscar M. González-Brambila, Gabriela Ramírez-Fuentes) ... 7
Design, Fabrication and Evaluation of a Power Operated Walnut Grader (Syed Zameer Hussain, Umbreen Showkat, Sheikh Idrees, Monica Reshi) ... 14
Single Locking Cotton Feeder for Enhancing Ginning Efficiency of Double Roller Gin (V. G. Arude, S. P. Deshmukh, P. G. Patil, S. K. Shukla) ... 24
Development and Testing of a Coconut Dehusking Machine (P. M. Chukwu, B. A. Adewumi, I. A. Ola, O. D. Akinyemi) ... 29
Maize Ear Threshing – an Experimental Investigation (Yang Liquan, Wang Wanzhang, Zhang Hongmei, Wang Meimei, Hou Mingtao) ... 34
Design, Development and Performance Evaluation of CIAE-Millet Mill (S. Balasubramanian, S. D. Deshpande, I. R. Bothe) .. 42
Development of a Front Mounted Cultivator for Power Tiller (Sourav Srichandan Das, Hifjur Raheman) ... 49
Development of Mathematical Model for Predirecting Peel Mass of Cassava Tubers (John C. Edeh) ... 55
Design Modification and Comparative Analysis of Cassava Attrition Peeling Machine (J. C. Edeh, B. N. Nwankwojike, F. I. Abam) ... 63
Development a Table Top Centrifugal Dehuller for Small Millets (N. A. Nanje Gowda, Satishkumar, Farheen Taj, S. Subramanya, B. Ranganna) ... 72
Design, Development and Evaluation of Manually-Operated Check Row Planter for Dry Sowing of Rice (Ajay Kumar Verma) . 79
Development of Semi Mechanised Tools for Cutting and Splitting of Jack Fruit for Bulb Separation (C. Nickhil, N. A. Nanje Gowda, B. Ranganna, S. Subramanya) ... 84

◇ ◇ ◇

Vol.51, No.2, Spring 2020
Could Conservation Tillage Farming Be the Solution for Agricultural Soils in Iraq? (Ali Mazin Abdul-Munaim, David A. Lightfoot, Dennis G. Watson)······························ 7
Design, Development and Testing of 4-Row Tractor Drawn Gladiolus (*Gladiolus Grandiflorus* L.) Planter for Uniformity in Corm Spacing (T. P. Singh, Vijay Guatam, Padam Singh, Santosh Kumar)···················· 10
Design and Development of Power Operated Walking Type Weeder (Ajay Kumar Verma)
Optimization of Parameters of Axial Flow16 Paddy Thresher (Ritu Dogra, Desai Kishor Waman, Baldev Dogra, Ajeet Kumar)········ 22
Measuring Spray and Spray Deposition on Plant and Unwanted in Field Under Iraqi South Conditions (Majid H. Alheidary, Qusay. Sameer, Abdul Salam G. Maki, Ali. F. Nasir)·· 28
Current Situation and Perspectives for Soybean Production in Amur Region, Russian Federation (Boris Boiarskii, Hideo Hasegawa, Anna Lioude, Elizaveta Kolesnikova, Valentina Sinegovskaia)······························ 33
Development and Evaluation of Rasp Bar Mechanism for the Extraction of Onion (*Allium Cepa* L.) Seeds (R. Pandiselvam, R. Kailappan, Anjineyulu Kothakota, B. Kamalapreetha, G. K. Rajesh)···················· 39
Design and Development of Low Cost Multi-Row Manual Jute Seed Drill (V. B. Shambhu)·· 46
Performance of Milking Machine at Different Vacuum Levels in Crossbred Dairy Cows Milked in Automated Herringbone Parlour (A. Fahim, M. L. Kamboj, A. S. Sirohi)······ 52
Development of Integrated Small Scale Lac Processing Unit (S. C. Sharma, N. Prasad, S. K. Pandey, V. K. Bhargava)···················· 58
Design and Construction of a Farm Scale Evaporative Cooling System (Gürkan Alp Kağan Gürdil, Pavel Kic, Bahadır Demirel, Emel Demirbas Yaylagül)···························· 67
Development and Performance Evaluation of Tractor Drawn Cultivator Cum Spike-Roller (V. R. Vagadia, Rajvir Yadav, D. B. Chavda, Geeta Tomar, D. V. Patel)·············· 72
Development of Mat Nursery Raising and Uprooting Techniques for Paddy (*Oryza Sativa* L.) Crop and Their Field Evaluation with Mechanical Transplanter for South East Asia (Mahesh Kumar Narang, Rupinder Chandel, Baldev Dogra, Gursahib Singh Manes)··· 79

◇ ◇ ◇

Vol.51, No.3, Summer 2020
Specual Issue: VIM 90th Anniversary
The Federal Scientific Agro-engineering Center VIM: History of Foundation and Development (Andrey Yu. Izmailov, Yuliya S. Tsench, Yakov P. Lobachevsky) ········· 7
Technical Support of Vegetable Growing in Countries of the Eurasian Economic Union (Aleksandr G. Aksenov, Aleksei V. Sibirev)12
The Prospect of Using Gas Turbine Power Plants in the Agricultural Sector (Valentin A. Gusarov, Zakhid A. Godzhaev, Elena V. Gusarova)·· 19
The State, Promising Directions and Strategies for the Development of the Energy Base of Agriculture (Andrey Yu. Izmaylov, Yakov P. Lobachevsky, Dmitry A. Tikhomirov, Anatoly V. Tikhomirov)·············· 24
Regional Features of Scientific-technical and Technological Modernization of Agro-industrial Sector of Bashkortostan at the Present Stage (Pavel A. Iofinov, Ildar I. Gabitov, Salavat G. Mudarisov, Denis A. Mironov, Badry H. Akhalaya)···················· 36
Technological Support of Soybean Cultivation (Aleksei S. Dorokhov, Marina E. Belyshkina, Ivan A. Starostin, Narek O. Chilingaryan)··· 42
Strategy of Technical Support of Grain Harvesting Operations in Republic of Kazakhstan (Vladimir L. Astaf'yev, Vladimir A. Golikov, Eduard V. Zhalnin, Sergey A. Pavlov, Igor A. Pekhalskiy)························· 46
The Methodology of Modeling and Optimization of Technologies in Crop Production (Vladimir V. Mikheev, Andrey G. Ponomarev, Pavel A. Eremin, Vladislav S. Mikheev)··· 52
Benefits of Using Liquid Nitrogen Fertilizers for Russian Farm Enterprises (Leonid A. Marchenko, Igor G. Smirnov, Tatiana V. Mochkova, Rashid K. Kurbanov)··············· 58
Trends in the Use of the Microwave Field in the Technological Processes of Drying and Disinfection of Grain (Alexey N. Vasil'yev, Alexey S. Dorokhov, Dmitry A. Budnikov, Alexey A. Vasil'yev) ··································· 63
The Main Stages of Agriculture Mechanization in Russia (Yuliya S. Tsench) 69
The Choice of Combine Harvesters and Their Adapters for the Conditions of Northern Kazakhstan (Mikhail E. Chaplygin, Sergey V. Tronev, Igor A. Pekhalskiy)···················· 74
The Trend of Tillage Equipment Development (Sergey I. Starovoytov, Badri H. Akhalaya, Sidorov S. A., Mironova A. V.)··················· 77
N. A. Borodin's Firsthand Study of the USA Power Farming Experience: the Lessons from History (Roman A. Fando, Maria M. Klavdieva)··· 82
Agricultural Robots in the Internet of Agricultural Things (Vyacheslav K. Abrosimov, Zakhid A. Godzhaev, Alexander V. Prilukov) 87
Improving the Resource, Reliability and Efficiency of Worn-out Machines with New Methods of Their Maintenance (Anatoliy V. Dunayev, Sergey A. Sidorov)················· 93

◇ ◇ ◇

Vol.51, No.4, Autumn 2020
Specual Issue: AMA50th Anniversary
SDGs of Agricultural Machinery Industry in Japan (Masatoshi Kimata) ·························· 31
Smart Agriculture Research in IAM-NARO (Ken Kobayashi)··· 36
As a Successful Contribution for New Agriculture Paradigm (Yasushi Hashimoto)······ 38
Smart Agriculture for 9 Billion People's Food Production and Environmental Conservation Aiming SDGs (Naoshi Kondo)··········· 40
Smart Robots for Production Agriculture for SDGs (Noboru Noguchi) ···························· 42
Digital Farming Strategy toward Agricultural Transformation (Sakae Shibusawa)············ 48
Mechanization, Digitizing and Innovations in Agriculture (Teruaki Nanseki)···················· 53
Need for Mechanization of Agriculture with Environmental Protection in Developing Countries (Kenji Omasa) ···························· 55
Overseas Expansion of Japanese Agricultural Machinery through Cooperation with International Cooperation Agency (Hideo Hasegawa)·· 56
SDGs and Kubota's Vision (Yuichi Kitao)······ 58
Agricultural Mechanization in the United States of America (John K. Schueller) ········ 60
Fifty Years of Progress in Agricultural Mechanization (Brian Sims, Josef Kienzle)········ 64
50 Years of AMA, the SDGs and Agriculture in Germany Karl Th. Renius)···················· 67
Development Trend "Digital Agriculture" from a German Perspective (Peter Pickel)·· 70
Mechanization of Agriculture in Germany (Peter Schulze Lammers)·························· 74
The Role of Life Sciences Universities in Relation and Strategy of Sustainable Development Goals (P. Kic)······································ 77
Serbian Agriculture, Agricultural Engineering – Past and Future (M. Martinov, Dj. Djatkov, S. Bojic, M. Viskovic) ·················· 82
Sustainable Development of Chinese Agriculture and Food Security (Chen Zhi)············· 84
50 Years of Agricultural Mechanization in China (Yang Minli, Li Minzan, Luo Xiwen) ··· 86
Current Status and Prospects of Agricultural Mechanization in China (Zhao Yanshui)···· 93
SDGs and Agricultural Mechanization in India (Indra Mani) ·· 96
Smart Farm Mechanization for Sustainable Indian Agriculture (C. R. Mehta, N. S. Chandel, Y. A. Rajwade)····························· 99
Agricultural Mechanization in Bangladesh: Status and Challenges towards Achieving the Sustainable Development Goals (SDGs) (A. Rahman, Md. R. Ali, Md. S. N. Kabir, Md. M. Rahman, Md. R. A. Mamun, Md. A. Hossen)·· 106
The Regional Network for Agricultural Machinery (Reynaldo M. Lantin)···················· 121
Agricultural Mechanization Today in Indonesia in Relation to the SDGs (Kamaruddn Abdullah) ··· 133
Sustainable Development for Agricultural Products Processing Industry and Agricultural Mechanization in Vietnam (Nguyen Hay) ·· 135
Egyptian Agriculture and Current Situation of Agricultural: Tractors and Equipment in Egypt 2009-2018 (Taher Kadah, Rania Khamis Ibrahim, Hanafi Radwan, Ahmed El Behery) ·· 137
SDGs and Storage Obstacles of Agricultural Production in Egypt (Said Elshahat Abdallah, Wael Mohamed Elmessery)················· 146
SDGs and Agricultural Mechanization Practice in Nigeria (Oyelade, Opeyemi Adeniyi)151
Agricultural Mechanization Today in Nigeria in Relation to Sustainable Development Goals (Akindele Folarin Alonge) ··············· 155
Draft Efforts' Behavior of a Vibratory Tool to Different Forward Speeds (L. O. M. Cabrera, A. M. Rodriguez, A. G. de la Figal Costales, Y. M. Mesa) ································ 157

■■

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