Misr J. Ag. Eng., 23(2): 292 - 313 FARM MACHINERY AND POWER

DEVELOPMENT OF A POTATO DIGGER

Samy M. Younis[1] **Mohamed I. Ghonimy** [2] **Tarek H. Mohamed** [3]

ABSTRACT

A potato digger was developed by adding a vibrating device to operate the digging blades and reduce the required drawbar pull and potato tuber bruising. The vibrating device includes beam holder, follower, cam and transmission system. The overall operating parameter (T) was also analyzed. The natural frequencies of the digging blade, potato tuber and disturbed soil were determined. The developed digger was tested at four levels of forward speed (0.9, 1.5, 1.9 and 3.2 km/h), four levels of vibrating amplitude (3, 5, 6 and 10 mm) and five levels of vibrating frequency (400, 600, 800, 1000 and 1200 rpm). The results showed that the drawbar pull of the developed digger was decreased by 25.17, 25.91, 28.43 and 30.47% at forward speeds of 0.9, 1.5, 1.9 and 3.2 km/h, respectively comparing with the original digger records at amplitude of 10 mm frequency of 1200 rpm. On the other hand, the developed digger succeed to operate with lower power tractors thus the harvesting cost was reduced by 28.5 %.

Keywords*:* Potato tuber, development, digger, drawbar pull, bruising

INTRODUCTION

Potato is considered as one of the most important vegetables crops, supplying human with carbohydrate. It is classified as the first alternative of the grain crops to solve the shortage of food in some countries. In last decent; the average world production of potato was 290.34 million tons of different potato varieties. In Egypt, potato production is increasing year by year. On 2005, it was 1.9 million tons meanwhile; the exported quantity was only 228 thousand tons. Although, the recent annual Egyptian potato export is nearly 200-250 thousand tons, Egypt succeeded to export more than 430 thousand tons to U.K and Europe on year 1994 (***C.D.O.A.I 2005***).

Potato harvesting is one of the most important operations has to be performed preciously to have a good potato production. It has a direct effect on the potato bruising. Bruising has an essential effect on potato marketing. The mechanical brushing could be happened when the tractors wheels roll on the potato rows during harvesting.

1 Prof., Agric. Eng. Dept., Fac. of Agric., Cairo Univ.
2 Assoc. Prof., Agric. Eng. Dept., Fac. of Agric., Cairo Univ.
3 Agric. Engineer, Agric. Eng. Unit, Ministry of Agric., Egypt.

Farmers avoid this damage by fitting the tractor with narrow tires but this remedy has a side effect as it requires higher horsepower and causes higher slip percentage.

For this reason during harvesting, farmers fit narrow tires to the tractors axles to avoid crushing the tubers before harvesting and move safely between rows. Although this kind of tires is very helpful to bruising but it decrease tractors drawbar pull. For this reason, using a potato digger with tractor equipped with narrow tires would need a higher horsepower to operate.

The subject of vibrating diggers tools has drawn the attention of many researchers. ***Ibrahim et. al. (1989)*** developed and tested a sugar beet digger to be used under the Egyptian conditions. They studied the effects of tilt angle blade width, and forward speed on the damage occurred due to the developed harvester. The studied parameters levels were as (15, 20, 25°), (17, 20, 23 cm), and (2, 3.5, 5 km/h) for tilt angle, blade width and forward speed ,respectively. Minimum tuber the damage and highest lifting efficiency were realized at 20 cm blade width, 20° tilt angle and 3.5 km/h forward speed. ***Kang and Handelson (1991)*** designed a two-row, three-point-hitch vibrating digger. Each row was composed of a pair of four-bar linkages to which two side plates are attached. A bottom plate for each row was composed of a soil-cutting blade followed by soil-sieving bars. The motion of the bottom plate was also designed to assist with soil flow. The oscillating assemblies were PTO driven by a cam through by roller chain drive. The greatest amount of black spot (24.9%) was observed at highest frequency (1227 rpm) and slowest travel speed [1.7 km/h (1.05 mph)]. Un recovered potatoes significantly increased (7.2-24.0%), as travel speed increased from 1.7-3.3 km/h. Draft force decreased as vibration frequency increased and travel speed decreased. Draft varied from about 7.9-12.2 kN over the range of combinations of frequency and travel speed levels. ***Srivastava et al. (1995)*** mentioned that using the reciprocated blades with plows increases the soil penetration and decreases the drawbar pull of the tractor. So using a web potato digger equipped with reciprocated blades may help to solve many technical and economical problems. ***Yow and Smith. (1976)*** mentioned that one dimensional sinusoidal of vibratory tillage was analyzed theoretically and experimentally. A model was developed in which the instantaneous horizontal force on the tool was equal to a constant plus a linear function of tool velocity. A maximum force reduction of 40% was observed. ***Niyamapa and Salokhe (2000)*** experimented vibrating tillage tools in a sandy loam soil. It was observed that during oscillating operation, initially draft increased slightly with an increase in forward speed but later it decreased. For the non-oscillating operation, draft increased continuously with increase in forward speed. The ratio of draft from oscillating to non-oscillating mode varied from 0.63 to 0.93. The total power required for oscillating operation was 41–45% more than the power required for non-oscillating operation. ***Verma et al. (1977)*** developed and tested an experimental potato digger equipped with an oscillating blade. The machine was tested at frequencies up to 9 cps and amplitude of 30 mm. The results indicated that, with their soil type, a reduction in draft requirement of up to 76% and decrease in percentage of skinned tubers occurred

The objective of this research is considered as an outcome of the interfering alternative of preparing potato digger for efficient performance. Thus, the aim of this study is to develop one of the potato diggers and equip it with vibrating blades to reduce the required drawbar pull and to minimize potato bruising.

MATERIAL AND METHODS

The idea of the modification of the original digger is supplying a new vibrating device to help the original digger to reduce the required drawbar pull and to reach potato tubers bruising to minimal.

1. The Original digger:

The original digger was one of the locally manufactured bulk potato diggers. It was equipped with 9 front fixed blades to make a total width of 1.76 m. Also, it was equipped with 2 webs for separating potato tubers from soil. The digger was attached to the tractor by 3 points hitch CAT. II and take the power from the tractor PTO through the universal joint at speed 540 rpm to move the webs.

2. The developed digger:

The modification was made by adding a new reciprocated device to vibrate the blades. The modification was performed in one of the local workshops in Cairo. Meanwhile, the field experiments were carried out in Nobaria city. The vibrating system consists of three sub-systems; beam holder, vibrating device and vibrating transmission system.

2.1. Beam holder:

The beam holder (Figures 1 and 2) was designed to hold the vibrating ruler and transfer its motion directly to the blades via the beams. Both rear and front beam holders were fitted to shape a rectangular orifice where the vibrating ruler slides between them. The rear beam holder was fitted to the beam with a bolt enter into a threaded hole in the back the beam itself. Meanwhile, the front beam holder was welded to the beam. The mean function of the beam holders was fitted the reciprocating ruler to the blade beams.

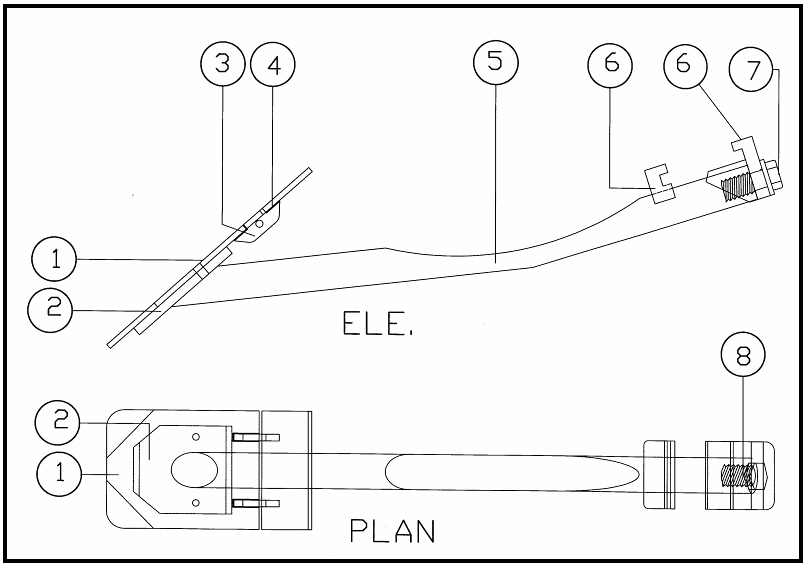

Fig. (1): The beam holders
1- Digging blade; 2- Blade adaptor; 3- Articulation; 4- Blade support ;5- Beam ;6- Beam holder; 7-Bolt ;8- Threaded hole.

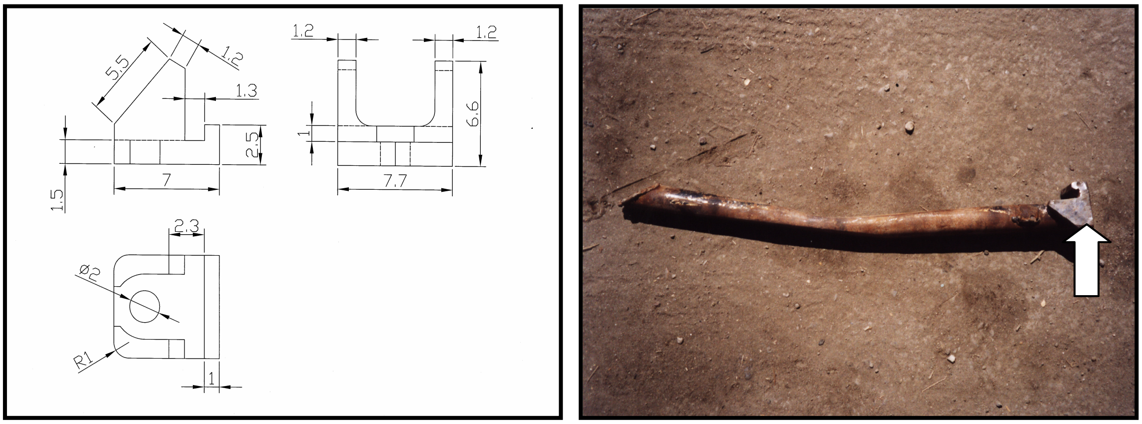

Fig. (2): (Left) The beam holder dimensions. (Right) the rear beam holder fixed to the beam

2.2. The vibrating device:

The vibrating device (**Fig. 3**) consists of three parts; reciprocating ruler, cam and driving arm. The cam transmits the motion from PTO to reciprocating ruler by a driving arm (follower) that was designed to produce the proper amplitudes.

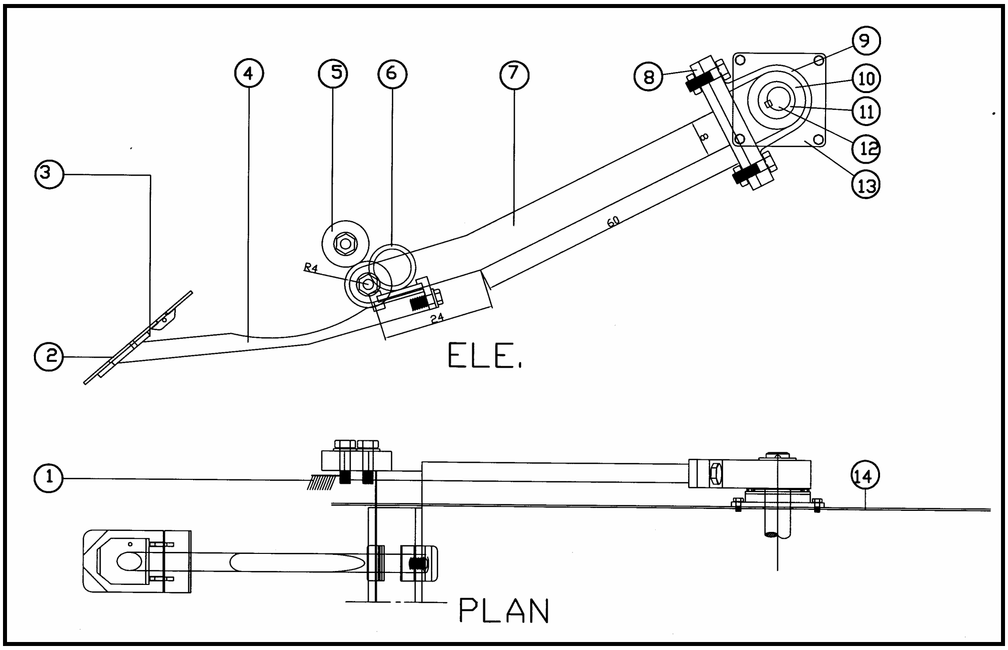

Fig. (3): The reciprocated device parts.

(1) Fixing point; (2) Digging blade; (3)Blade support; (4) Shank; (5) Arm supporting device; (6)Reciprocating ruler; (7) Arm; (8)Arm fitting; (9)Cam bearing housing ;(10)Cam bearing; (11)Cam; (12)Cam shaft; (13)Cam shaft bearing support; (14) Side chasses.

To select the suitable frequency of the vibrating device, the natural frequencies of the digging blade, potato tubers and soil were determined from equation 1 (***Inman 1996***).

$$N = \sqrt{\frac{k}{m}} \tag{1}$$

Where:

- N = Blade, potato tuber and soil natural frequencies, rad/s;
- K = Blade, potato tuber and soil stiffness, N/m;
- m = Blade, potato tuber and soil weight, kg.

The stiffness (k) of each of blade and potato tuber was calculated from equation (2)

$$k = \frac{EA}{L} \tag{2}$$

Where:

- E = Modules of Elasticity for steel blades and potato tuber, Pa;
- A = Blade and potato tuber projected area, m^2;
- L = Blade and potato tuber lengths, m;

The potato stiffness for the tubers (k) was determined by using firmness tester to measure the firmness of a freshly harvested 4 different varieties of potato tubers differ in their natural mechanical properties. The average value of firmness for Alpha, Spunta, and Valod varieties were 0.942, 0.721, 0.871 and 0.883 kg/cm^2 receptively. Also, the average tuber weights for the same varieties were 161, 210, 129 and 165 gram /tuber. From the results above, it is realized that the Spunta variety is very sensitive to bruising where it's firmness was 7.21 N/cm^2. From this point, Spunta variety mechanical and natural properties were taken as the min. parameters effect on design.

To find the tubers stiffness (k) it was necessary to find the tuber projected area in three dimensions. Using AutoCAD2000 program and a scanner, the projected area and diameter in 3 dimensions were determined by cutting different sizes of the tubers in 3 dimensions and draw them on paper divided in cm^2 then the paper is scanned on the scanner. **Fig (4)** shows a sample of the projected area and diameters in 3 dimensions in a potato tuber. Using the AutoCAD2000 program the papers pasted with scale (1:1) on drawing worksheet. By drawing an oval on the outlines of the images it was easy to get the projected area and diameters directly. The maximum frequency for 1000 tubers, were indicate that the average projected area for Spunta variety A_1, A_2 & A_3 were 59,03 cm^2,47.77 cm^2 and 25.85 cm^2 respectively and the average tuber diameters in 3 dimensions a, b, c were 11.13 cm ,6.44 cm and 5.03 cm respectively . Thus, the minimum stiffness was 5348.16 N/m while the maximum stiffness was 16745.59 N/m.

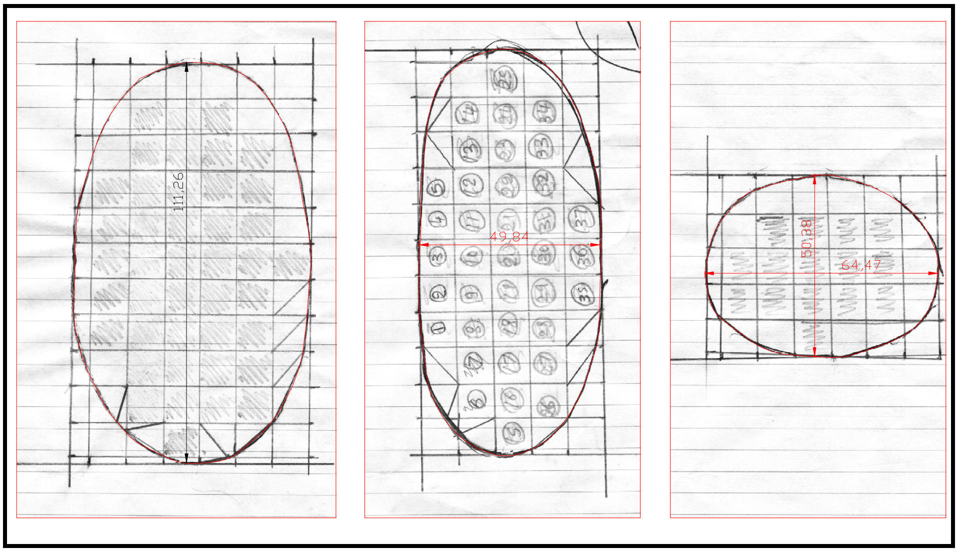

Fig (4): The projected area and diameters in three dimensions

The soil stiffness (k) was calculated from equation (3) as follows:

$$k = \frac{U \times A_c}{s} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Where:

k= Soil stiffness, N/m;

U =Unit draft of soil, N/cm^2;

= (12 N/cm^2 for heavy clay soil that was taken as a sample of the severest working condition during potato harvesting).

The values of *E, A, L, k, m* and *N* for digging blade, potato tuber and disturbed soil are shown in table (1).

Table (1): The values of *E, A, L, k, m* and *N* for digging blade, potato tuber and disturbed soil

Item	**Blade**	**Potato tuber**	**disturbed soil**
E, Pa	200E09	7.21E04	12E04
A, m^2	0.014	59 E-04, 47.77 E-04 and 25.85 E-04	0.528
L, m	0.08	0.1113, 0.644 and 0.503	0.003
k,N/m	3.5E09	Min. 5348.16	2112
m, kg	2.51	0.2	1.32
N, rad/s	4436	Min. = 21.6	6.36

Thus, the suitable frequency ranged between 6.36 and 21.6 rad/s (381.6 to 1296 rpm)

2.2.1. Reciprocating ruler:

The reciprocating ruler has two jobs. First job was transferring the vibration from the follower (arm) and the second job was holding the beam through the beam holders. It makes the two jobs in the same time so the loads are very high on it. The reciprocating ruler consists of 2 parts, a rectangular plate dimensions of (190 cm x 10 cm x 10 mm) (L x W x T) and a pipe diameter of 10 cm and thickness of 10 mm welded along the plate horizontally to be able to fitful the loads. **Fig. (5)** shows the reciprocating ruler and its parts.

The beam holders were fitted on it through the bolts. And each end was welded in the reciprocating arm.

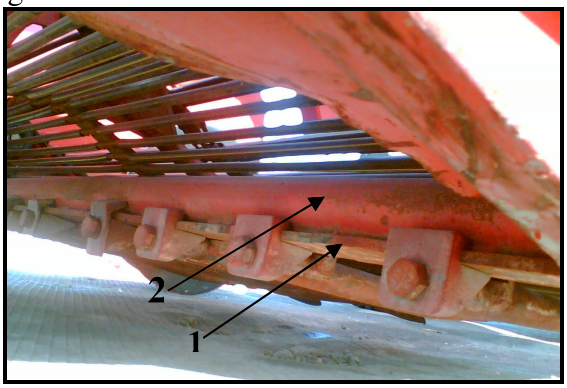

Fig. (5): The reciprocating ruler

(1) The plate (2) The pipe

To design the reciprocating ruler, the dimension of the reciprocating ruler was selected according to the available space. The maximum deflection of the reciprocating ruler was calculated from equation (4):-

$$P = \frac{F_C}{L_R} \tag{4}$$

Where:

P = Distributed uniform load, N/m;

L_R = Beam length, m;= 1.914

F_c = The pulling force at (cutting angle of 40°) = 23084 N == the maximum drawbar pull measured for the original digger.

Thus, the value of distributed uniform distribution load (P) is 12060.61 N/m

The maximum bending moment on the beam was calculated from equation (5) according to (***Hall et al, 1980***).

$$M_{b\max.} = \frac{PL_R^2}{8} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

$$M_{b\max.} = \frac{12060.61 \times 1.914^2}{8} = 5522.84 \;\; N.m$$

The maximum deflection of the reciprocating ruler can be calculated from equation (6).

$$\delta_b = \frac{M_b y}{I} \le S_{allow} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Where:

δ_b = The maximum deflection of the reciprocating ruler ;

I = Moment of inertia, m^4;

$$I = \frac{1}{12}(b_{beam})^3 T_{beam} + \frac{\pi(D^4 - d^4)}{64}$$

b_{beam} = Beam width = 0.1 m

T_{beam} = Beam thickness = 0.010 m

D = Outer diameter = 0.10 m

d = Inner diameter = 0.08 m

Thus, the value of maximum deflection of the reciprocating ruler was 49793211 N/m^2

$$\delta_b \le S_{allow}$$

$$S_{allow.} = S_y / (F.S)$$

$$(F.S) = S_{allow} / S_y \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

Where:

S_y was taken from the steel tables= 120000 Psi for steel SAE1095 Annealed.

$F.S$ =Factor of safety = 8.68

2.2.2. Cam:

The cam (**Fig. 6**) was supported to the cam rotating shaft by a key. The rotating cam shaft transfers the selected rpm to make vibration. The outer diameter of the cam was surrounded by a taper bearing that selected according to the loads effecting on it. The bearing inner diameter was 5 cm meanwhile the outer diameter was 8 cm. The bearing was put in a bearing housing with inner diameter of 5 cm equipped with greaser.

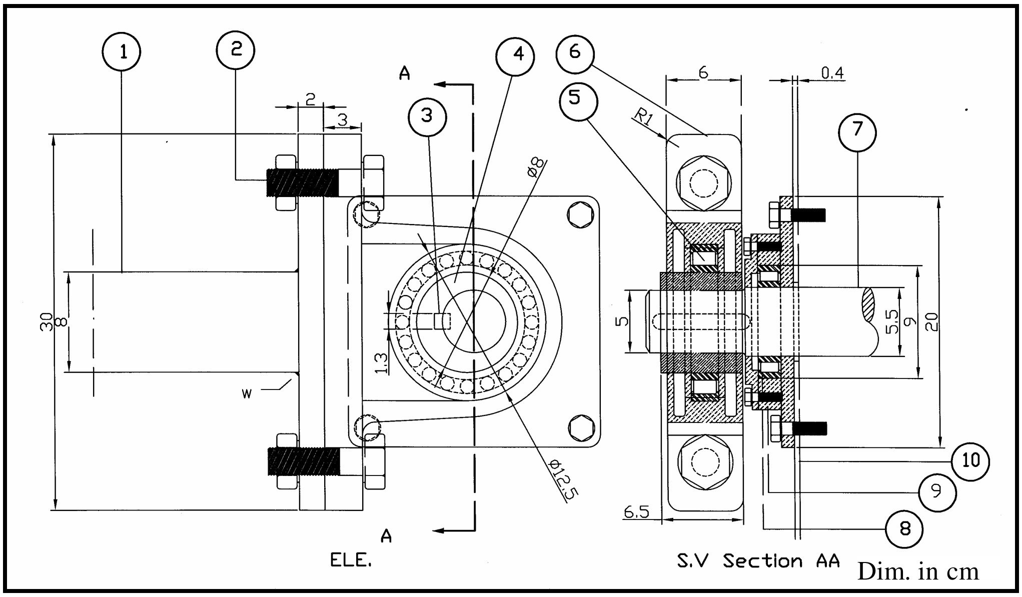

Fig. (6): The cam position in the reciprocating device.

(1)The arm; (2) Arm bolts; (3) Cam key; (4) Cam; (5) Cam bearing; (6) Cam bearing housing; (7) Cam shaft ; (8) Cam shaft supporting bearing ; (9) Supporting housing; (10) Side chasses.

The cam itself was made of steel SAE1050 Annealed. Four sets of cams were manufactured with displacement (e) equal 3, 5, 6 and 10 mm according to ***Black (1986).*** It was necessary to test the digger to find the optimum amplitude that reduces the drawbar pull and does not affect the tubers.

2.2.3. Arm (Follower)

The main job of the arm is converting the rotating motion of the cam to reciprocating motion. The reciprocating ruler was welded to the underneath of the arm. On the rear of the arm, the cam housing was fitted to the arm with 2 bolts M22, meanwhile, the front of the arm was hanging freely with 2 supporters.

The cross sectional area of the arm (A) was calculated from equation (8) according to (***Hall et al, 1980***).

$$\sigma_c = = \frac{F_{follower}}{A} \leq S_{allow} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (8)$$

where:

σ_c = Compression stress,N/m^2;

$F_{follower}$ = The max. load on the arm $= \frac{\tau}{e}$, N;

τ = Max. torque, Nm; $\cong M_{b\max.}$ (The max. bending moment on the beam) = 2885.50 Nm from equation (5)

Where there were two arms on the digger

τ on each arm $= \frac{2885.50}{2}$ =1442.75 Nm

e = Maximum distance between cam center and maximum stroke point; m, =0.01 m

$$F_{follower} = \frac{1442.75}{0.01} = 144275 \text{ N}$$

A = Follower (Arm) cross section area,m^2. = 0.08×0.04 =0.0032 m^2

$$\sigma_c = = \frac{144275}{0.0032} = 45085737.5 \text{ N/m}^2$$

$$S_{allow} = \frac{S_e}{F.S} \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

S_e =Endurance limit (fatigue strength) of the follower, MPa;

$$S_e = k_a \times k_b \times k_c \times k_d \times k_e \times S_e^{/} \ldots\ldots\ldots\ldots\ldots\ldots (10)$$

Where:

k_a = Surface finish factor =0.9 from tables

k_b = Size factor =1

k_c =Reliability factor =1 for R>90%

k_d =Temperature factor =1 for T <450 C^o

k_e =Stress concentration factor =0.7

$S_e^{/}$ = Endurance limit of a standard rotary spectrum ,MPa;

$S_e^{/}$ = ½ S_{UT}

S_{UT} = Ultimate tensile strength of the steel

S_{UT} for steel (SAE1050) $\rightarrow S_{UT}$ =95000 psi

$$S_e^{/} = 0.9 \times 1 \times 1 \times 1 \times 0.7 \times \left(\frac{95000}{2} \right) = 29925 \ psi$$

$$S_{UT} = 2104.43 \times 10^4 \times 10 = 210443038 \ N/m^2$$

$$F.S = \frac{210443038}{45085737.5} = 4.66$$

2.2.4. Shaft:

The diameter of the transmission shaft was calculated according to ASME equation (***Hall et. al, 1980***). The bending moment, axial load, and the torque acting on shaft were calculated.

Using the **ASME** code equation, the shaft diameter was 49.9 mm at max. cam stroke, max. digging angle and max. drawbar pull and max. forward speed.

2.3. Transmission system :(Fig. 7)

The machine takes its motion from the tractor PTO at speed of 540 rpm by a universal joint to a gearbox equipped with 2 bevel gears gives a reduction ratio of (2.92:1). The motion transfer to the right side to the main idler with 30 teeth (z =30) that transmits the motion to the main sprocket with 30 teeth (z =30) by a chain ¾ inch.

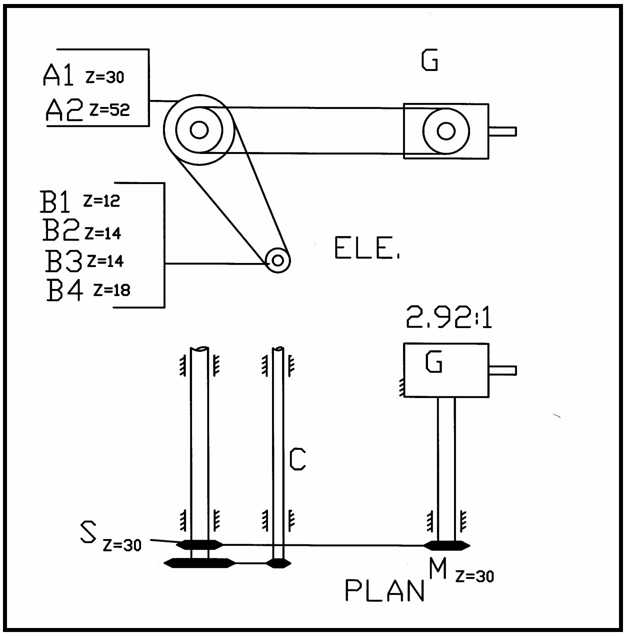

Fig. (7): Transmission to the cam shaft.

(G) Gear box (M) Main idler (S) Main sprocket (A1&A2) Cam idlers (B1,B2, B3&B4) Cam sprockets (C)Cam shaft

It was necessary to modify the transmission system to transfer the motion to the cam shaft. Two new idler (z=30) & (z=52) were manufactured to rotate the sprocket of the cam shaft by a chain ¾ inches.
The two new idlers were fixed on the same shaft of the main sprocket.
Also, four sprockets were manufactured to give 400, 600, 800, 1000 and 1200 rpm to determine the suitable frequency that doesn't affect the tubers but cause soil failure. The calculation of the number of teeth was according to ***Black (1986).***
Fig. (8) Shows the digger after the modification

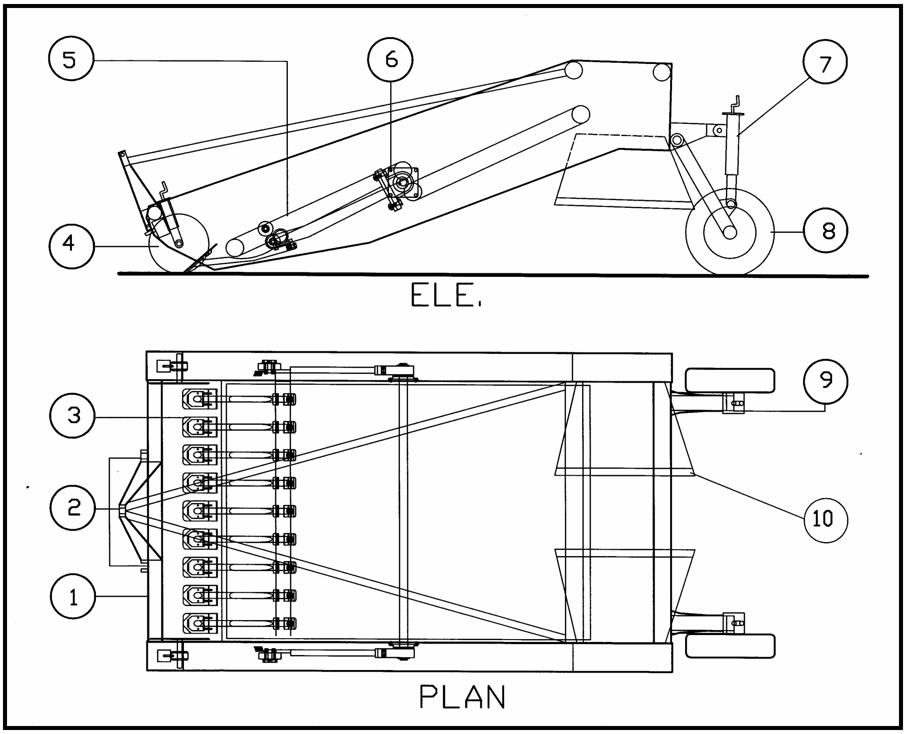

Fig. (8): The developed digger

(1) Main frame; (2)Three- point hitch; (3) digging blade; (4) side disc; (5) primary web; (6) reciprocating device; (7) frame adjustment; (8) rear wheel ; (9) wheel axel; (10) rear harvested tuber reception.

Tractors:

A Kubota tractor model M1-110, 110 (SAE) hp engine, was used to operate the machine. The tractor was equipped with 440 kg front weight. Three forward speeds were selected during the experiments. Another two tractors, first a UTB Romanian made, 75 hp, was used during the drawbar tests and another tractor New Holland TM150 was used with the original digger. All tractors were equipped with narrow tires during experiments.

Potato crops : Sponta variety.

Treatments:

1. Digging forward speed:

The tested values of digging forward speed were 0.9, 1.5, 1.9 and 3.2 km/h

2. Amplitude:

The tested values of amplitude were 3, 5, 6 and 10 mm (according to ***Black 1986***).

3. Frequency (ω):

The tested values of frequency (*ω)* were 400, 600, 800, 1000 and 1200 rpm.

Measurements and calculations

1. Potato tuber dimensions

The dimensions of potato tuber were measured by using vernier caliper (accuracy = ± 0.1 mm).

2. Drawbar pull (F)

The required drawbar pull (***F***) from the tractor was measured by hydraulic dynamometer. The draft was recorded in the measured distance of (20 m) as well as the time taken to traverse it. The difference between records in the same speed gives the draft of the implement. ***Smith et.al (1994).***

3. Frequency *(ω)*

The frequency was determined by using digital tachometer (accuracy 1:6 rpm).

4. Potato tuber bruising

The potato tuber bruising was calculated from equation (10) according to ***Bishop and Maunder (1980)***.

Bruising = Scuffed x 1 Peeler x 3 Severe x7 ………..(10)

5. Digger field capacity (*FC*)

The potato digger field capacity was calculated from the following equation:

$$FC = A / t \tag{11}$$

Where:

FC = Digger field capacity; fed./h;

A = Digging area, fed;

t = Total consumed time ,h.

V_t = Digger forward speed, m/sec.

The k ratio was used to evaluate the performance of vibrating digger blade.

6. Overall parameter (*T*)

The overall parameter (*T*) was calculated from equation (12) according to ***Kang and Handelson (1991)***

$$T = \lambda . k \tag{12}$$

Where:

λ = The ratio between vibrating and forward speeds;= $\frac{\omega A}{V_t}$

ω = Angular speed, rad. /sec;

A = The amplitude of the vibration, m;

V_t = Digger forward speed, m/sec.

k = The ratio of blade acceleration to gravitational acceleration;

= $\frac{\omega^2 A}{g}$

g = Gravitational acceleration, m/sec^2 .

7. Cost analysis and economic evaluation:

The cost analysis (***Oida, 1997***) was performed in two steps. The first step was to calculate the cost of the materials and the fabrication. The second step was to calculate the developed digger operating cost.

In order to evaluate the financial viability of the developed digger, three parameters were computed and were analyzed. These parameters are the developed system operating cost, the internal rate of return (IRR) and the pay-back-period (PBP).

RESULTS AND DISCUSSION

1. The performance of the original digger:

The tests of the original digger were carried out in EL-Behera Governorate – El- Nobaria city in a sandy soil, with 147 hp tractor with Spunta variety plants at 4 different forward speeds.

1.1. The original digger field capacity

The average values of the field capacity for the original digger at four levels of forward speed and three levels of digging angle in a sandy soil with moisture content of 24 % are shown in **Fig. (9)**

From **Fig. (9)** it is clear that the digger felid capacity increased by increasing the forward speed and decreasing the blade digging angle. The maximum value of the digger field capacity was 0.68 fed./h at forward speed of 3.29 km/h at digging angle of 30 ^{o} while, the minimum value was 0.38 fed./h at forward speed of 0.90 km/h at blade digging angle of 40 ^{o}.

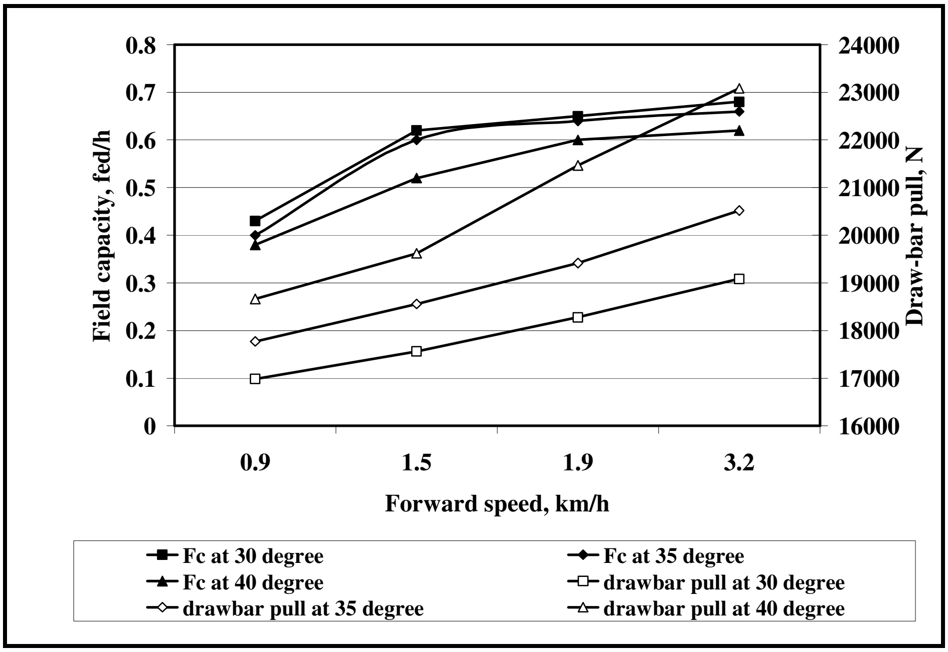

Fig. (9): The original digger field capacity and drawbar pull at different forward speeds and different digging angles.

1.2. The required drawbar pull

The average values of the drawbar pull of the original digger are shown in fig. (9). It is clear that the drawbar pull was increased by increasing the forward speed and blade digging angle. The minimum value of the drawbar pull was 16987 N at 0.90 km/h forward speed and 30^{o} blade digging angle.

1.3. Potato tuber bruising and harvesting losses

The results of inspecting samples for tuber bruising after harvesting are shown in **Fig. (10**). From **Fig. (10)** It is clear that both bruising and un-harvested potato tubers percentage increased by decreasing the blade digging angle and increasing the forward speeds. Also it's realized the poison of the blade near the web is effect on the bruising ratio.

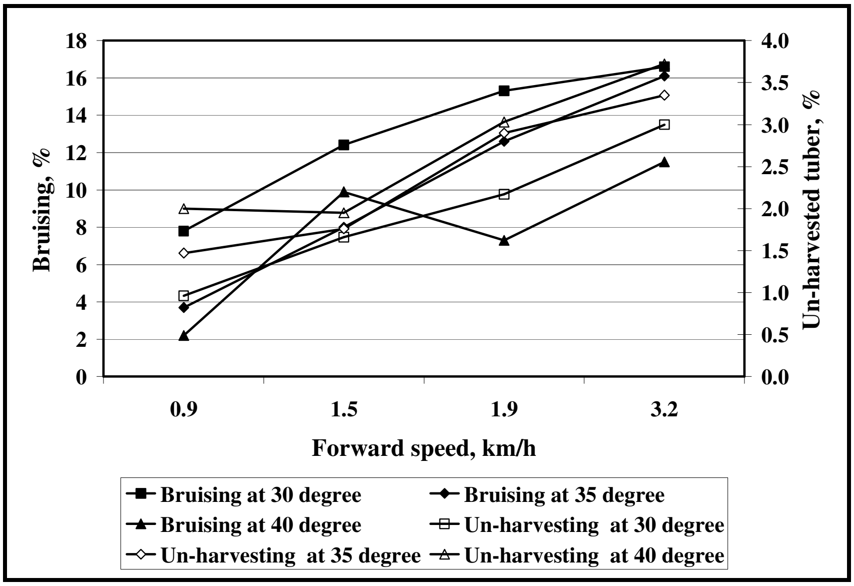

Fig. (10): The effect of forward speed on the bruising and un-harvested potato tubers

On the three digging angles of 30°, 35° and 40° it realized that the optimum digging angle is 35°. Although, the digging angle of 30° gives high field's capacities but it gives high bruising. On the other hand, the digging angle of 40° gives low bruising, but it needs high drawbar pull and give low felid capacities.

2. The performance of the developed digger

2.1. The required drawbar pull (*F*)

A relationship between the overall parameter (*T*) and the drawbar pull was drawn. The average values of the drawbar pull (*F*) of the developed digger at different performance parameter (*T*) are shown in **Table (2).** It is clear that the F values increased by decreasing the T values at the same frequency and amplitude. Also, the maximum value of the drawbar pull is 20220 N was found at vibrating amplitudes of 3 mm at forward speed of 3.2 km/h and blade frequency (ω) 400 rpm, while, the minimum value of the drawbar pull was 13300 N at vibrating amplitudes of 10 mm at the forward speed of 0.90 km/h at blade frequency (ω) 1200 rpm.

From fig. (9) and table (2) it's clear that the minimum value of drawbar pull for the developed digger (13300 N) decreased by 21.7 % than the original digger (16987 N).

The using of vibrating blades cause a failure in the soil, thus the required drawbar pull decreased.

Table (2): The effect of the performance parameter (T)on the drawbar pull (F)

Frequency, rpm	Forward speed, km/h	Amplitude, mm							
		3		5		6		10	
		T	F	T	F	T	F	T	F
400	0.90	0.27	17651	0.75	17125	1.08	16628	3.00	15829
400	1.50	0.16	18432	0.45	18111	0.65	17832	1.80	16889
400	1.90	0.13	19211	0.35	18950	0.51	18210	1.42	17726
400	3.20	0.08	20220	0.21	19637	0.30	19345	0.84	18493
600	0.90	0.91	16725	2.53	15990	3.64	15679	10.11	14888
600	1.50	0.55	18320	1.52	16352	2.18	16073	6.07	15286
600	1.90	0.43	19105	1.20	18811	1.72	17798	4.79	16790
600	3.20	0.26	19507	0.71	19125	1.02	18291	2.84	17291
800	0.90	2.16	16182	5.99	15292	8.63	15032	23.97	14321
800	1.50	1.29	17931	3.60	15682	5.18	15420	14.38	14630
800	1.90	1.02	18926	2.84	15862	4.09	15588	11.36	14803
800	3.20	0.61	19105	1.69	18133	2.43	17998	6.74	16766
1000	0.90	4.21	15560	11.71	14792	16.86	14460	46.82	13722
1000	1.50	2.53	15904	7.02	15183	10.11	14892	28.09	14099
1000	1.90	2.00	16122	5.55	15359	7.98	15090	22.18	14207
1000	3.20	1.19	16450	3.29	15766	4.74	15470	13.17	14687
1200	0.90	7.28	15120	20.23	14352	29.13	14077	80.91	13300
1200	1.50	4.37	15533	12.14	14745	17.48	14488	48.55	13690
1200	1.90	3.45	15730	9.58	14939	13.80	14652	38.33	13895
1200	3.20	2.05	16124	5.69	15333	8.19	15055	22.76	14268

2.2. Tuber bruising (*B*):

The results of the total bruising against overall parameter (T) are shown in **Table (3)**. It is clear that the maximum value of bruising was 23.24 % at forward speeds of 3.2 km/h and blade frequency (ω) 1200 rpm. On the other hand, bruising of the original digger at the same speed was 16.62%.

This means that the amplitude of 10 mm is affecting bruising with extra percentage of 6.62%. At amplitude 5 mm, the minimum value of bruising was 2.96 % at forward speed 0.9 km/h at ω 800 rpm. On the other hand, bruising of the original digger at the same speed was 3.66 %. The same trend was found at 3 mm amplitude.

To determine the optimum (T) parameter, it necessary to select the one that requires lowest drawbar pull with minimal tuber bruising. From table (3) at amplitude 6 mm and frequency 800rpm, it is clear that the values of bruising at the select amplitude were 3.24, 7.88, 9.87 and 16.82 % at forward speeds of 0.9, 1.5, 1.9 and 3.2 km/h respectively. On the other hand, bruising of the original digger at the same speeds were 3.66, 7.98, 12.64 and 16.07 % respectively.

Table (3): The effect of the performance parameters (*T*) on the Tuber bruising (*B*)

Frequency, rpm	Forward speed, km/h	Amplitude, mm							
		3		5		6		10	
		T	*B*	*T*	*B*	*T*	*B*	*T*	*B*
400	0.90	0.27	4.08	0.75	4.43	1.08	4.16	3.00	8.06
400	1.50	0.16	8.88	0.45	9.18	0.65	9.67	1.80	11.74
400	1.90	0.13	12.87	0.35	13.39	0.51	14.8	1.42	13.86
400	3.20	0.08	16.77	0.21	17.86	0.30	18.88	0.84	17.35
600	0.90	0.91	3.94	2.53	3.11	3.64	3.07%	10.11	5.84
600	1.50	0.55	8.64	1.52	8.61	2.18	8.95	6.07	13.52
600	1.90	0.43	12.30	1.20	13.06	1.72	10.1	4.79	16.41
600	3.20	0.26	16.88	0.71	18.22	1.02	15.80	2.84	23.10
800	0.90	2.16	3.22	5.99	2.96	8.63	3.24	23.97	5.79
800	1.50	1.29	8.99	3.60	8.26	5.18	7.88	14.38	13.43
800	1.90	1.02	8.80	2.84	13.00	4.09	9.87	11.36	14.21
800	3.20	0.61	17.50	1.69	18.49	2.43	16.82	6.74	20.56
1000	0.90	4.21	3.47	11.71	6.55	16.86	3.73	46.82	6.26
1000	1.50	2.53	8.27	7.02	11.87	10.11	9.52	28.09	14.00
1000	1.90	2.00	7.17	5.55	17.27	7.98	10.8	22.18	15.32
1000	3.20	1.19	17.05	3.29	17.14	4.74	20.75	13.17	21.13
1200	0.90	7.28	7.12	20.23	8.18	29.13	7.81	80.91	9.08
1200	1.50	4.37	10.10	12.14	12.63	17.48	14.5	48.55	14.50
1200	1.90	3.45	11.86	9.58	22.54	13.80	21.0	38.33	15.40
1200	3.20	2.05	17.00	5.69	19.62	8.19	21.99	22.76	23.24

At the same conditions, the drawbar pull records of the developed digger were 15032, 15420, 15588 and 17998 N respectively. These values are less than the original digger records by 15.42, 16.93, 19.7 and 12.29 % respectively.

2.3. Field capacity (*Fc*):

The field capacity (*Fc*) of the developed digger was calculated at the optimum performance parameters. Due to the soil failure caused by the reciprocating blades, the forward speed surly increased and the drawbar of the developed digger is decreased. The developed digger field capacity is shown in **Fig. (11)**. From **Fig. (11)**, it is clear that the field capacity of the developed digger forward speed was increased compared with the original digger field capacity. The *Fc* was increased 8.3% at the optimum forward speed of 1.55 km/h.

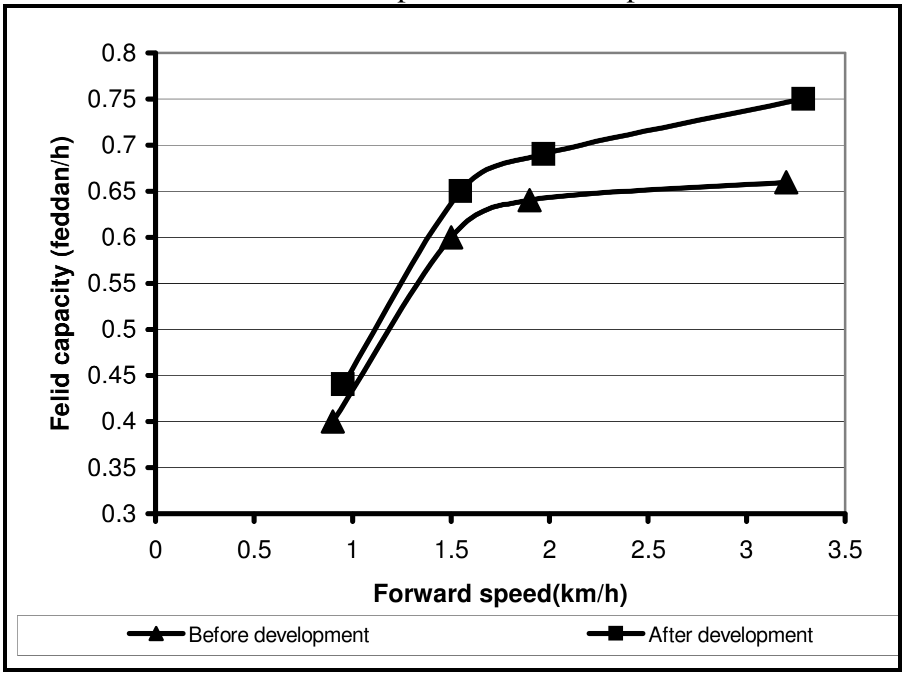

Fig. (12): A comparison between the field capacity of the original digger and developed digger

2.4. The developed digger operation cost:

The calculation of the operating costs included fixed and variable costs were made for two kinds of tractors and the developed digger. The developed digger can operate without load on tractor 110 hp meanwhile; it needs a tractor of 140-150 hp if the reciprocating device is disengaged. The total operating costs for a 140 hp tractor and original digger were 77.14 LE/h and 27.04 LE/h respectively. On the other hand, the total operating costs for a 110 hp tractor and developed digger were 45.99 LE/h and 28.51 LE/h respectively with represents a decrease of about 30 LE/h (28.5 %) due to using the developed digger.

Economical studies

The total fabrication cost of the modification in the developed digger including workshop cost was 2500 LE at 2006 price level. The total operating costs for a 140 hp tractor and original digger were 77.14 LE/h and 27.04 LE/h respectively. On the other hand, the total operating costs for A 110 hp tractor and developed digger were 45.99 LE/h and 28.51 LE/h respectively.

The developed digger achieved an internal rate of return (IRR) of 72 %. The developed digger IRR shows that the investment is worthy. The developed digger indicated NPV of 4587 LE at 11 % interest rate. The developed digger pay back period was about two years

CONCLUSION

From this investigation the following conclusions can be done:

1. Comparing with original digger records at the same forward speeds, the drawbar pull of the developed digger less by 25.17, 25.91, 28.43 and 30.47 % at forward speeds 0.9,1.5,1.9 and 3.2 km/h respectively at amplitude10 mm and frequency 1200 rpm.

2.The suitable amplitude was 6 mm with frequency of 800 rpm that give lowest drawbar pull with minimum bruising ratio.

3.The developed digger can be operated with 110 hp tractor instead of 140 hp tractor or more thus the harvesting costs are less by 28.5 %.

4.The modification success to increase the machine field capacity by 8.3%
5.The IRR of the developed digger is 72 % and NPV of 4579 LE. The pay back period is 2.1 years.

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المراجع العربيه

الاداره المركزيه للارشاد الزراعي .وزاره الزراعه . نشره رقم ٨١٣.زراعه وانتاج البطاطس،

الملخص العربي

"تطوير الة لتقليع البطاطس"

سامي محمد يونس[١]　　محمد إبراهيم غنيمى[٢]　　طارق حسين على محمد[٣]

يهدف هذا البحث الي تطوير احد الالآت المحليه التي تقوم بتقليع محصول البطاطس حيث تم أستعمال نظريه الاسلحه المترردده في قطع التربه لتقليل قوه الشد دون ضرر علي الدرنات. و تم تحقيق هذا الهدف من خلال المراحل التالية:

١- دراسه بعض الخصائص الطبيعيه لبعض اصناف البطاطس ومن هذه الخصائص ابعاد الدرنات والوزن ومساحه السطح والحجم والقابليه للاختراق.

٢- اختبار اداء الة التقليع الاصلية التى سيتم تطويرها .

٣- أشتمل النموذج النهائي لالة التقليع المطورة علي الوحدات الوظيفية التالية:

١- ماسك القصبات. ٢- وحده توليد التردد. ٣- نظام الحركة المعدل.

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١ أستاذ بقسم الهندسة الزراعية – كلية الزراعة – جامعة القاهرة
٢ أستاذ مساعد بقسم الهندسة الزراعية – كلية الزراعة – جامعة القاهرة.
٣ مهندس زراعى – وحدة الهندسة الزراعية – وزارة الزراعة

تم أختبار اله التقليع المطوره عن طريق دراسة تأثير أربعة سرعات أمامية ٠,٩، ١,٥، ١,٩، ٣,٢ كم/ساعة و أربعة مشاوير٣، ٥، ٦، ١٠ مم و خمسة ترددات مختلفة ٤٠٠، ٦٠٠، ٨٠٠، ١٠٠٠، ١٢٠٠ لفة / دقيقة علي كل من قوه الشد المستهلكة و نسبه التجريح للدرنات ومعدل الاداء الحقلي.

وقد بينت الدراسة ما يلي:

١. نجحت اله التقليع المطوره في تقليل قوه الشد بنسبه ٢٥,١٧ و٢٥,٩١ و٢٨,٤٣ و٣٠,٤٧ % عند سرعات ٠,٩ ، ١,٥ ، ١,٩ ، ٣,٢ كم/ساعة عند تردد للاسلحه ١٢٠٠ لفه/ د ومشوار ١٠ مم علي الترتيب عن القياسات المأخوذه قبل التطوير.
٢. كان المشوار المثالي للاسلحه ٠,٦ مم مع تردد٨٠٠ لفه / د حيث انخفضت قوه الشد و كذلك نسبه التجريح.
٣. يمكن تشغيل اله التقليع المطوره بواسطة جرار ١١٠ حصان بدلا من ١٤٠ حصان او اكثر مما يقلل من تكلفة الحصاد للساعة بنسبه ٢٨,٥. %.
٤. نجح التعديل في زياده السعه الحقلية للاله المطوره بنسبه ٨,٣ % .
٥. اوضحت الدراسه الماليه ان معدل العائد الداخلي لاله التقليع المطوره ٧٢ % وقيمه صافي القيمة الحالية ٤٥٧٩ جنيه كما وصلت فتره استرداد رأس المال الي ٢,١ سنه .