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ANALYSES OF CALCULATED VS MEASURED POWER OF THE LOCALLY FABRICATED COMPOST TURNER

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ABSTRACT

Compost turning machine was designed, modified and fabricated locally. The large number of field testes run by the turner proved that the machine was well designed and reliable. According the machine was field operated for several seasons in turning compost windrows at variable conditions with recognized turning efficiency. The experiments showed that the double drums machine turned the compost windrows pile with high quality. The local fabricated compost turner is powered by two power systems. The first is mechanical system and the second is hydraulic system. The mechanical system is to rotate the double augers drums path by tractor PTO. The hydraulic system is to push the machine and tractor to the front path by hydraulic power from the tractor. The objectives of this study were to optimize power source requirement for the locally fabricated compost turner as follow:-

- Analysis of theoretical parameters that affect both rotating augers and forward motion of turner.
- Determine actual vs measured power required for the turner.

The power requirements were calculated to defy the actual power consumption to be suitable and safe for machine. The maximum power was 45 kW and the minimum was 8 Kw. From design calculation of the main parts in the hydraulic system, the maximum diameter of the cylinder piston is 90 mm to raise and lower the turner box with double drums. Anther side of the two cylinder piston to lower and raise the machine during transportation is 90 mm also. Theoretical hydraulic motor power which is satisfactory to drive the system (turner and tractor) was calculated and the proper motor size was selected.

INTRODUCTION

The total crop residuals are 32 million tons/year (Ministry of agriculture, 2007) for this reason became very important to use this material to produce useful production to increase the nationality income, stop the damages which is happened from storage to reduce the pollution. Composting is the production which produced from piling the crop residues after cutting with animal production residues (manure).

Macmillan (2002) induce that the performance of a tractor depends to a significant degree on its weight and, in particular, on the weight on the driving wheels. It is therefore useful to define a non-dimensional drawbar pull - weight ratio termed: Tractive coefficient, y = Drawbar pull/Weight on driving wheels The tractive coefficient is a number which characterizes the interaction between the wheel and the surface in an analogous way to which coefficient of (sliding) friction characterizes the interaction between one body sliding on another. Where a different wheel and surface may be considered similar to those for which the tractive coefficient is known, then for the same wheel slip

Drawbar pull = Tractive coefficient x weight on wheel

Where: a four-wheel tractor is considered, and with other tractors also, the weight used may be the total weight on all wheels. In quoting values of tractive coefficient, it is therefore necessary to state which weight has been used.

Virginia Nelson (2002) reported that the horsepower requirements of the turners tested were measured while turning the windrows. For PTO operated machines the torque and speed of the PTO were measured. For machines operated hydraulically, the hydraulic pressure, flow, and temperature were measured. The maximum and average horsepower requirements for the Aeromaster 70 and 37 hp and Earthsaver85 and50 hp and Wildcat 55 and32hp respectively Generally, the first turning requires the most horsepower; as the active composting stage progresses and materials break down the horsepower required reduces. The turners that turn the entire width of the windrow in one pass (Aeromaster, and Earthsaver) required more horsepower than those that turn half the width (Wildcat).

Baiomy (2007) reported that the pilling is not homogonous and it is formed as triangle or deltoid form. The delta conventionality is 3 m and the height is 1.7 m. Turner machine is one from many types which used to turn and mix the piling. These piles are generally turned to improve porosity, oxygen content, mix, control moisture, and redistribute cooler and hotter portions of the pile

The main objectives in this study is to estimated and the power required ,energy consumption of the machine operation and to get products with height quality by using dimensional analysis ,theoretical approach of turner machine and Design calculation for the hydraulic system and the mechanical system.

MATERIALS AND METHODS

Composting is methods to facilitate making to profit from crop residues. Local turner was built at a private sector company (Baiomy , 2007). The machine was operated and field evaluated by the senior author. An article was published in which the details drawings and dimensions was presented and discussed. The current article discuss the optimization of mechanical power and hydraulic power required for the turner. The present study include the computation of the hydraulic system to the turner machine (pull type) and calculate the power requirement to operate the mechanical system of turning drums was composted and measured. A torque meter and data aquization system was installed in the systrm to measure the actual power consumed by the turning mechanism.

1- Local turner machine: It consists of Main frame, turning unite, pushing unit, Balance unit) and Hydraulic system. The specification of the local compost turner is as shown in Table (1).

2 Components of power train of the turner: It is comprehended the mechanical drive system and hydraulic power system as shown the diagram in Fig. (1)

Table (1): Turner specification.

No	Main parts	Specification.
1	Main frame: Total Length.	594 cm.
	Total Width.	565 cm.
2	Turner unit: turner box:	
	Width.	295 cm.
	Hight.	175 cm.
	Depth.	200 cm.
	Double drums:	
	Outer diameter.	45 cm
	Length of drum.	300 cm.
3	Source of power:	Tractor 65 Hp

A- Mechanical drive system: It consists of as shown in Fig. (2 **):-**

- **i- Source of power:** Tractor PTO shaft 540 rpm.
- **ii- Universal joint:** Two universal joints were serial conducted to transport power from PTO to the gear box of turner**.**
- **iii- Gear box:** Decrease speed ratio from 1.5 to 1 and connect power augers of the double drum turning mechanism.
- **vi - Double auger drums:** the mechanism that performs the main function of the machine.

B-Hydrualic power system: It component as shown in Fig. (3):-

- **i- Reservoir:** The container in which equate volume of oil stored.
- **ii- Hydraulic pump of the tractor:** It is 44 L/min discharge and 250 psi.
- **iii- Control Valve: Directional control.**
- **vi- Hydraulic motor power.**
- **v- Load check:** A device which prevents a load from dropping when a valve is shifted, until ample pressure and flow is available to hold or move the load.

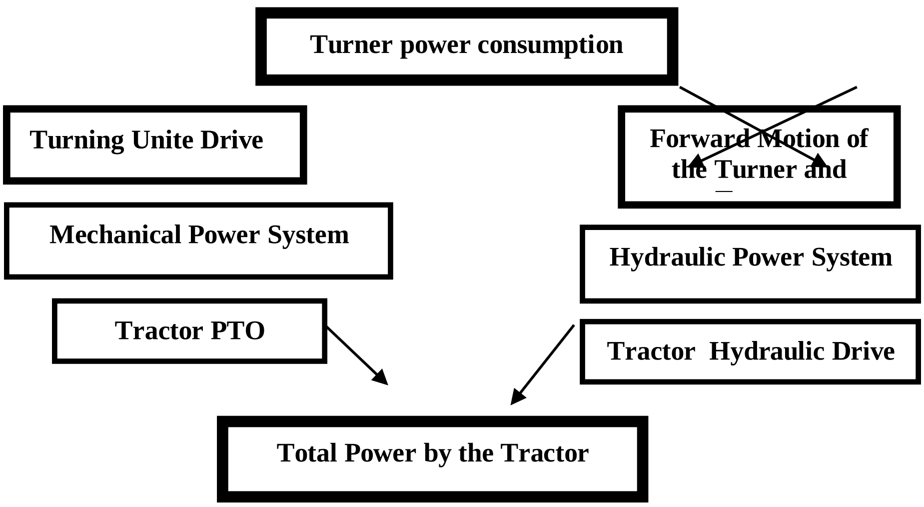

Fig. (1): Diagram for the component of the local turner machine.

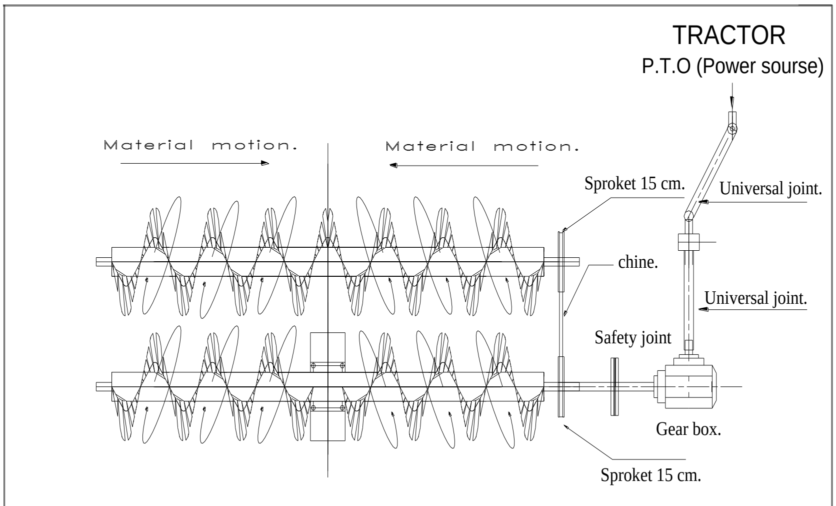

Fig. (2) Mechanical systum that rotat turning mechanism.

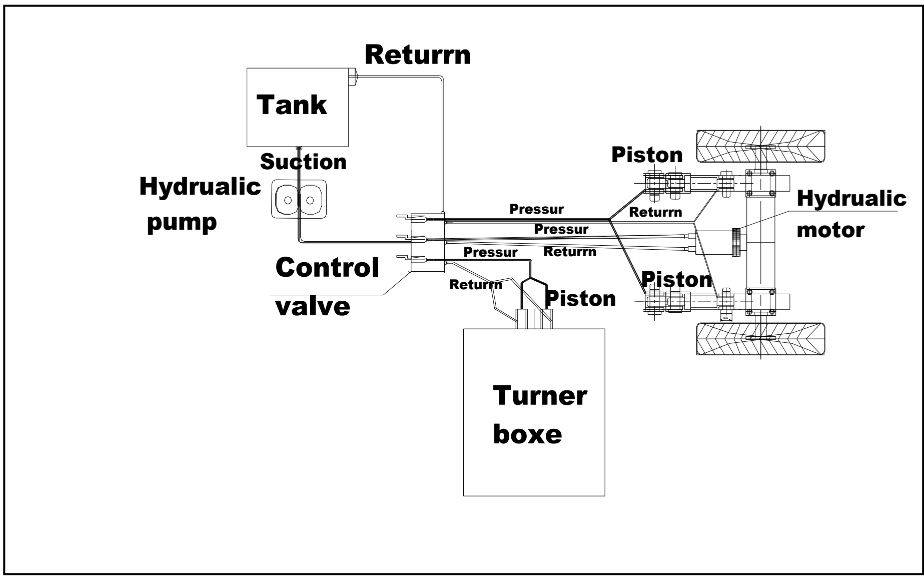

Fig. (3): Hydraulic system for power the forward motion of tractor and the turner.

iv- Hydraulic pistons: There are four hydraulic pistons in the system, two pistons to raise the turner box up. The vertical position is to transmit turner from place to anther. Anther two pistons to raise two hydraulic power wheels during turner travel and lower wheels to ditch with the soil while the hydraulic motor push the turner system front during the machine turning the pile

3- Instruments.

i-Torque meter: A slip ring strain gage torque meter equipped with a magnetic pickup sensor was used to collect both the torque and the revolution signals in SI unit. The torque transducer is consists of two shafts. The right shaft can be connected to the tractor PTO through the PTO adaptor, while the left shaft can be connected to the implement's joint (universal joint). This torque transducer was constructed by Elgwadi (2005).

ii- Data acquisition system:

A daytronic data PAC model 10k4, capable to convert the voltage signal to the desired SI units, and Hewlett Packard model 110 laptop computer were used. The Data PAC unit conditioned the output signal into proper engineering units. The computer was used to store incoming data and communicate with the data PAC

4-Field Experimental:

The field experiments were run at El-Gmeza research station, Garbia governorate. Universal (UTB) tractor 75 HP was used to pull the turner. That is hatched behind the tractor. The torque meter (right side) was installed between the P.T.O of the tractor and universal joint on the turner.

The data obtained in this study was taken by a daytronic data PAC model 10k4, and by Avow meter version 1.08 interfaced with laptop computer The variable measured were implement torque of PTO revolution as shown in Fig. (6). Compost conditions were characterized by moisture content (60%), bulk density (0.8 kg/m^3). Compost height was manually determined (1.5 m). The device was recalibrated at the beginning of the field experiment. The calibration was done by the designer of the device .the calibration procedures as determined and followed by the device designer are as follows.

Torque measurement (Torque cell):

Torque cell consists of a mechanical element (usually a shaft with a circular section and a sensor (usually electrical resistance strain gages). A circular shaft with four strain gage mounted as shown in Fig. (4) strain (1,3) sense the positive strain while strain (1,4) sense the negative strain, the four strain gage define the principle stress and strain directions for the circular shaft subjected to pure torsion. The shearing stress τ in the circular shaft is related to the applied torque T by the equation;

$$\tau_{xz} = \frac{TD}{2J} = \frac{16T}{\pi D^3} \quad \text{---1}$$

Where

D = is the diameter of the shaft.

J = is the polar moment of inertia of circular cross section.

Wiley *et al*. (1984) calculated the torque by measuring the Wheatstone bridge output voltage Eo and using material property as follows.

Since the normal stresses σx = σy = σz = 0 for a circular shaft subjected to pure torsion, Mohr's circle indicates that;

$$\sigma_1 = -\sigma_2 = \tau_{xy} = \frac{16T}{\pi D^3} \text{ --2}$$

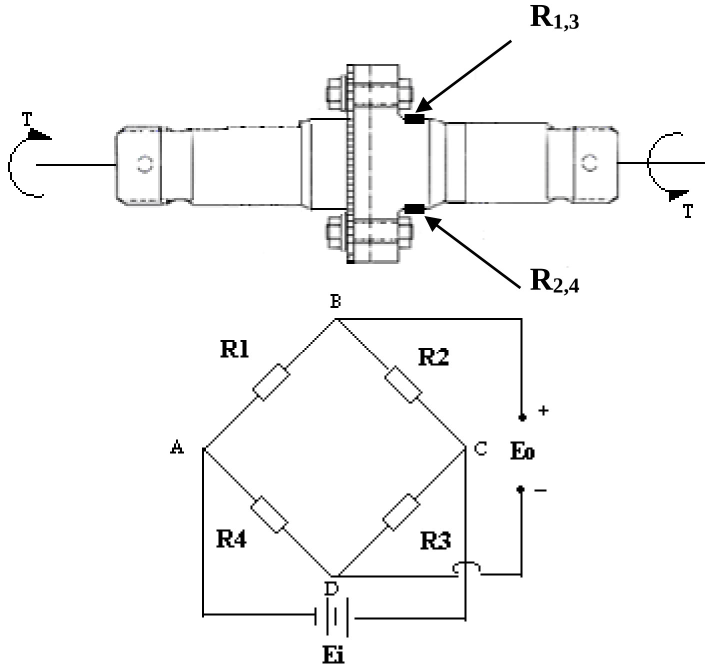

Fig (4): Torque cell with strain gage sensor positioned in the Wheatstone bridge

Principal strains ε1 and ε2 are obtained by using equation 2 and Hooke's law for the plane state of stress. Thus;

$$\varepsilon_1 = \frac{1}{E}(\sigma_1 - \nu\sigma_2) = \frac{16T}{\pi D^3}\left(\frac{1+\nu}{E}\right) \text{ ---------------------------3}$$

$$\varepsilon_2 = \frac{1}{E}(\sigma_2 - \nu\sigma_1) = -\frac{16T}{\pi D^3}\left(\frac{1+\nu}{E}\right) \text{ ------------------------4}$$

Where v is Poisson's ratio of the material

E is the modulus of elasticity of the material

A strain gage exhibits a resistance change ΔR/R that is related to the strain ε in the direction of the grid lines by the expression;

$$\frac{\Delta R}{R} = S_g \varepsilon \text{ ---5}$$

Where: ΔR is the resistance change of the gage due to the gage an applied strain ε.

R is the resistance of the gage.

S_g is strain gage sensitivity

The response of the strain gages is obtained by substituting equation 4 into equation (5)

$$\frac{\Delta R_1}{R_1} = -\frac{\Delta R_2}{R_2} = \frac{\Delta R_3}{R_3} = -\frac{\Delta R4}{R_4} = \frac{16T}{\pi D^3}\left(\frac{1+\nu}{E}\right)S_g \quad \text{--------6}$$

With fixed-value in the circuit, the open-circuit output voltage Eo can be expressed as:

$$E_{o\,AB} = \frac{R_1}{R_1+R_2}E_i = \frac{1}{1+r}E_i \quad \text{--------7}$$

$$E_{o\,AD} = \frac{R_4}{R_3+R_4}E_i = \frac{1}{1+r}E_i \quad \text{--------8}$$

An output voltage Δ Eo develops when resistances R1, R2, R3, R4 are varied by amounts ΔR1, ΔR2, ΔR3, Δ R4 respectively, with these new values of resistance of the bridge is;

$$\Delta E_{o\,AD} = \frac{r}{(1+r)^2}\left(\frac{\Delta R_1}{R_1} - \frac{\Delta R_2}{R_2} + \frac{\Delta R_3}{R3} - \frac{\Delta R_4}{R_4}\right)E_i \quad \text{--------9}$$

Where E_i is the input voltage.
r is the resistance ratio R_2/R_1.

If the gages are connected into a Wheatstone bridge, as illustrated in Fig. (4), the relationship between output voltage Eo of the bridge and torque T is obtained by substituting Eq. 6 into Eq. 9. The result is:

$$E_o = \frac{16T}{\pi D^3}\left(\frac{1+\nu}{E}\right)S_g E_i \quad \text{--------10}$$

Or

$$T = \left(\frac{\pi D^3 E}{16(1+\nu)S_g E_i}\right)E_o \quad \text{--------11}$$

Or

$$T = C * E_o$$

Where D is the diameter of the shaft, (0.02m);
E is the modulus of elasticity, equals, (1931 X 10^8 N/m^2);
v is Poisson's ratio of the material (0.3);
S_g is sensitivity factor, equals (2.1 for constantan strain gage);
E_i is voltage input (9 volts);
E_o is voltage output (to be measured by avometer in millivolts); and
C is constant

Calibration of the strain gage torque transducer:

Two channels of the Daytronic Data PAC were calibrated to their transducer. Channel number1 was calibrated to measure torque, while channel2 was calibrated to measure number of PTO revolution Gwadi (2005). During calibration, a slope (EMM) and an intercept (BEE) were stored in the Data PAC for later use during field test. The slope and intercept values defined a straight-line relationship between the torque measured from the manufactured transducer and the value of the load applied in SI units Fig. (5). During Calibration Torques was also calculated using the above equation Wiley et al, (1984), by measuring Eo in millivolts which was recorded using Avometer version 1.08 to receive calibration data measurement in millivolts (mV), this was interfaced with laptop computer for data storage. A digital strain meter model (P-3500) for Eo signal enlargement, was used. Torque calculated was compared with torque measured using, a daytronic data PAC model 10k4. Fig (5) and Table (2) define differences between torque calculated using Wiley *et al*, equation, and torque measured using a daytronic data PAC model 10k4 which did not exceed 11%.

Table (2): Applied torque with torque calculated and torque measured

Torque			Volts	Diffiation %
Applied	Measured (dytronic)	Calculated (Avometer)	Measured	
N.m	N.m	N.m	mV	
50	48.5	46.1	3.8	0.05
180	175.3	163.0	13.5	0.07
196	190.9	175.6	14.6	0.08
200	220.0	213.4	17.7	0.03
250	243.5	219.2	18.2	0.10
300	292.3	274.8	22.8	0.06
355	345.9	321.7	26.7	0.07
410	399.5	379.6	31.5	0.05
425	450.0	405.0	33.6	0.10
425	414.2	395.5	32.8	0.05
480	467.8	452.4	37.5	0.03
510	497.0	469.0	40.6	0.06
590	575.0	536.5	44.5	0.07
640	633.5	563.9	46.8	0.11
670	653.0	587.7	48.7	0.10
710	692.0	631.1	52.3	0.09

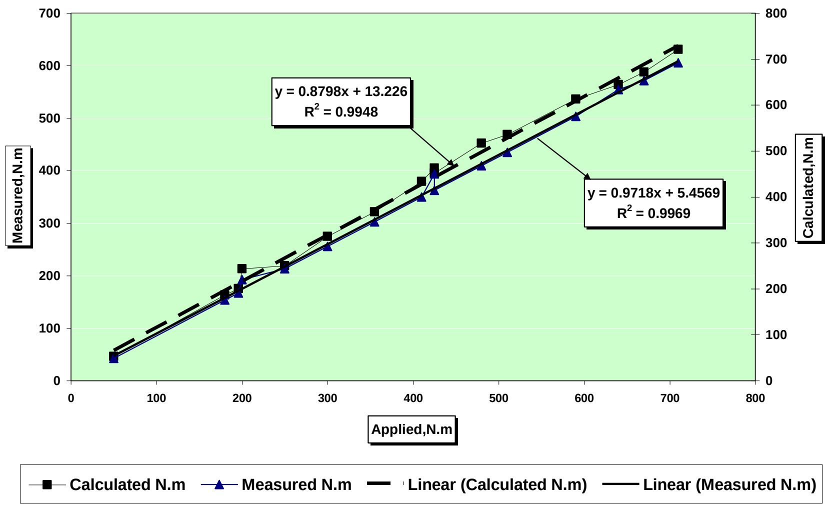

Fig (5): Applied torque with torque calculated and torque measured in SI units.

Mechanical calibration

To perform this, a calibration setup with lever arm and weights was used to create a torque, Fig. (6). Setup for calibration:

i- The rated torque was applied to the transducer, and then the load was removed.

ii- The zero point was precisely adjusted to the transducer.
iii- A known torque was applied to the transducer.
iv- Progressively greater torques were applied to the transducer up to the full rated torque. Gradually the torque was removed in the same way. At each step, last at least 30 second for the torque reading to stabilize, and then record.
v- Output voltage E_o was recorded at each step.
vi- A calibration curve was plotted Fig. (5), torque applied was drawn versus the torque measured and torque was calculated using the torque transducer. Torques was recorded and was saved on a laptop computer.

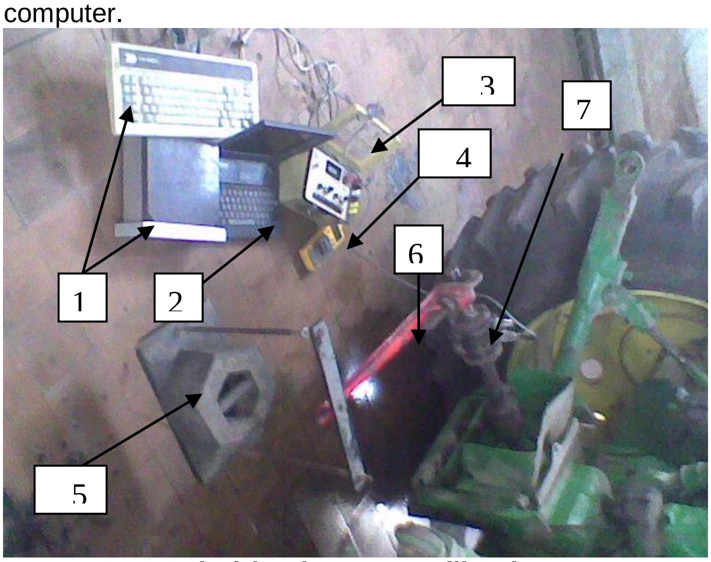

Fig (6) Laboratory calibration

1. Daytronic Data PAC
2. Laptop computer
3. Strain meter
4. Digital Avometer
5. weight
6. Lever arm
7. Torque meter

Table (3): Variables of turner and compost material for dimension analyses.

Variable	Definition.	Dimensions
σ^2	coefficient of variation	--------------
D	Drum diameter	L
d	Particle diameter of materials.	L
V	Volumetric charge of drum	L^3
g	Gravitational acceleration	L/T^2
ρ	Density of materials.	M/L^3
N	Rotational speed of drum	1/T
t	Time of turning.	T
L	Length of compost turner drum.	L
M	Mass of charge.	M
P	Power consumption	ML/T
μ	Coefficient of friction	------------

DIMENSION ANALYSES AND THEORETICAL APPROACH

1- Dimension analyses and Power requirement for turning drums.

Materials are consists of cutting crop residues and animal residues (manure). Consider the turning of two materials or more in tumbling drums. Assume that the two materials are identical in all aspects. A relationship is sought for the turning of these ingredients as a function of the variable that describes the system. First define the quality of turning by a coefficient of variation which is dimensionless. The coefficient of variation is calculating as developed by **(Kenneth et al, 1991)** induce the dimension analyses and following equations:-

$$\sigma^2 = f\left(\frac{N^2D}{g}, \frac{V}{D^3}, \frac{L}{D}, Nt, \frac{d}{D}\right) \text{-------------------------------12}$$

replace mass with volume and density would have been included in the relationship as follow: $V = \frac{M}{\rho}$

$$\sigma^2 = f\left(\frac{N^2D}{g}, \frac{M}{\rho D^3}, \frac{L}{D}, Nt, \frac{d}{D}\right) \text{------------------------13}$$

The diameter of the turner drum is very large than particles size of composting materials. The small value of L/D and d/D were neglected.

$$\sigma^2 = f\left(\frac{N^2D}{g}, Nt\right) \text{-------------------------------14}$$

Turning time calculate from the flowing equation

$$Nt\,turning = f\left(\frac{N^2D}{g}\right)$$

$$N \alpha 1/D^{0.5} \text{--- 15}$$

$$t \alpha D0.5 \text{---16}$$

Power consumption of the turner machine.

$$\frac{P}{\rho N^3 D^5} = f\left(\frac{N^2D}{g}, \mu, \frac{m}{\rho D^3}\right) \text{--------- ----------------------17}$$

With the fixed discharge of the drum (fixed forward speed and drum rpm) the result would be:-

$$\frac{P}{\rho N^3 D^5} = f\left(\frac{N^2D}{g}, \mu\right) \text{----------- ----------------------18}$$

$$\frac{P}{\rho N^3 D^5} \alpha \frac{g}{N^2 D}, \frac{M}{\rho D^3} \text{---------------------------------19}$$

4-2- Calculation of the power requirement of turner drums.

Power required in turning operation is consuming to overcome motion resistance which is involving two different types of resistance in turning operation namely:-

Dynamic resistance to auger drums and blade knives motion.

Fraction at boundaries t of materials clouds created about turning blade.

i- Dynamic resistance to auger drum motion.

$$D_f = D_p * A_b$$ -----------------------------------20

Where:-

D_p = dynamic pressure kg/ m^2
D_f = dynamic force kg
A_b = projected area of blade knives m^2

$$A_b = Lb$$ ---21

Where:

L = length of drum , m
b = blade knives projection width , m

$$D_p = \frac{\rho V^2}{g}$$

$$WD_f = \frac{D_f V}{102} \text{ kg}$$ ---------------------------------22

Where:

ρ = density of turning materials, kg/m^3
v = tangential velocity . m/sec
g = acceleration of gravity , m/sec^2
102 = 75x 1.36 (conversion factor)
WD_f = Power of dynamic resistance (kW)

ii- Second resistance to motion of(blade knives)

Tangential velocity $V = \omega r$ m/sec -------------------------------23

The force d Df acting on an element of the blade knives dr

$$dD_f = D_p bdr$$

$$dD_f = \rho V^2 bdr / g$$

$$dD_f = \rho \omega^2 r^2 bdr / g$$

The partial power d Wr required overcoming the friction resistances estimated as follows

$$d W_r = V d D_f$$

$$= \int \omega r \rho \omega^2 r^2 bdr / g$$

$$= 1/g \int \rho \omega^3 r^3 bdr$$

$$W_r = \rho \omega^3 r^4 b / 4g * 102$$ ---------------------------24

$$W_{rt} = W_r Z$$

Where:

r = Radius of turning drum
W_r = power requirement for one blade to overcome blade motion

W_{rt} = total power requirement for all blade to overcome blade motion
Z = number of blade knives on the turning drum

2- Friction between materials circulating around blade knives

W_f = D_f / C * AI * ω/102 --25

Where:

W_f = friction power -----kw
D_f =dynamic force kg
C =center distance between blade knives ------m
AI = area of friction between material , m^2
AI = L x C m^2

Total mechanical power required $WTM = WD_f + Wtr + W_f$ ----- 26

The total mechanical power required were calculated at different rotational speed of the turner drum (200, 250, 300, 350 and 400 rpm).

3- Hydraulic power calculation

The weight of the tractor 3500kg and the turner weight 6000kg. The total weight 9.5 ton. The coefficient of rolling resistance were .0.02 to 0.4 depending on the soil type.

rolling resistance = coefficient of rolling resistance x total weight

horse power $h_p = \frac{475*200}{3600*75} = 0.35 \quad h_p$

assume the factor of safety equal = 3

Drawbar pull horse power required = $1.05 h_p$

hp power motor = psi x gpm /1714X(pump efficiency) x (power train efficiency)

$$1 = \frac{250x(Gpm)}{1714x0.85x0.96}$$

Flow rat to the hydraulic motor (Gpm) = 5.6 Gpm = 21.28 L / min

Horsepower drive pump for tractor 75 hp its discharge 11.62 Gpm (44 litter per mint) was calculated from equation:

$$h_p\,power\ pump = psix\text{Gpm}\ x0.0007 = 2.1 h_p = 1.55kW$$

Cylinder calculation to lift the total weight of the turnenr box 2000kg (4500 pounds)

$$A = F/P = \frac{4500}{250} = 18 \quad in^2 = 116.12\ cm^2$$

Where

A is Area (in square inches)
F is Force (in Pounds)
P is Pressure of the tractor hydraulic pump (in PSI) =0.068 bar
diameter = 12 cm Factor of safety equal = 1.5
diameter = 18 cm = 2 Cylinder 9cm diameter
gpm of Flow Needed for Cylinder Speed calculated from equation:

$$gpm = \left(\frac{\text{Cylinder Area } x \text{Stroke Length}}{231}\right) x \frac{60}{Time(for-one-stroke)}$$

$$gpm = (\frac{18x36}{231})x\frac{60}{45} = 3.77 \quad gpm = 14.271L/\min$$

Cylinder Area =18 in^2 (116.12 cm^2)
Stroke Length = 36 inches (90 cm)
Time = 45 seconds for one stroke

Cylinder calculation to lift the rear weight of the turnenr 2750kg (6063 pounds)

Area square inches= Force (in Pounds)/ Pressure (in PSI)

$$A = \frac{6063}{250} = 24.25 \quad in^2\left(156.4cm^2\right)$$

Diameter = 14 cm Factor of safety equal = 1.5
= 2 Cylinder 10 cm diameter

GPM of Flow Needed for Cylinder Speed calculated from equation
=(Cylinder Area X Stroke Length in Inches / 231) X (60 / Time in seconds for one stroke)

$$= (\frac{24.25*24}{231})*\frac{60}{30} = 5.038 \quad gpm\left(19.07L/\min\right)$$

The total flow rate = The flow rate for pushing system + The flow rate to left turner box piston

$$= 5.038 + 3.77 = 8.808 \quad gpm\left(33.6L/\min\right)$$

So that the system was safety becouse system delver 11.62 gpm and the hydrolaic system of turner consume 8.808 gpm and the hydrolaic Power required.

RESULTS AND DISCATION

Experiments were run at El-Gmeza research station, Garbia governorate. To measure the power composition of compost turner and verification the theoretical calculation data to optimize the power requirement. the theoretical calculation was included mechanical and hydraulic power requirement .

1-Mechanical power required to drive turning augers:

The results of the measure mechanical power and computed power were recorded and analyzed as shown in Table (4) and Fig. (7). From the results the minimum power requirement was 5 kW as calculated and the maximum 42 kW. The minimum power requirement was 8 kW as measured and the maximum 45 kW. The difference between the calculated and the measured value As the average 12%.

Table (4): The relationship between the rotational speed and power requirement (measured Vs calculated)

Rotational PTO rpm	TOURQE N.m	Power ,kW		Difference %
		Measure	calculation	
150	505	7.9	5	36.7
200	519	10.9	8	26.6
250	661	17.3	15	13.3
300	765	24	22	8.3
350	920	33.7	31	8
400	1070	44.8	42	6.3

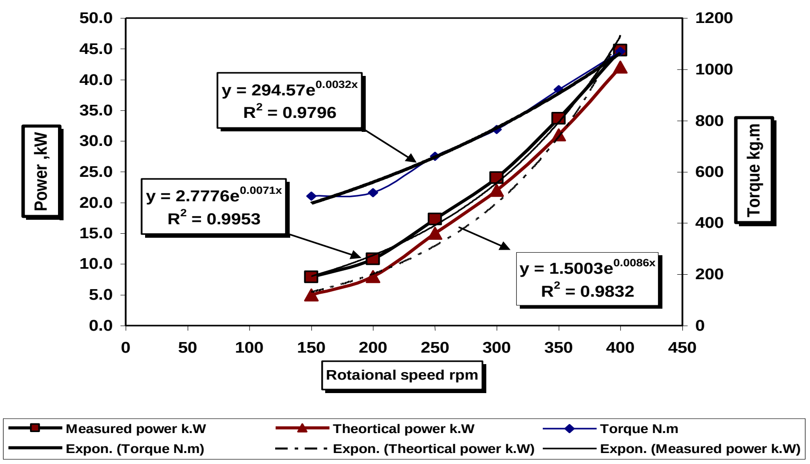

Fig. (7): The relationship between the rotational speed and power requirement (measured Vs calculated)

The compost turning augers powered by the PTO of the tractor. The strain gage and dytronic system was used to measure the torque applied to drive the turning system.

The power consumed was determined according to the measured torque which considered the actual power consumed to drive the turning mechanism. The results were compared to the power computed according to the theoretical analysis.

Power required increased as the rotation speed increase. Minimum power to drive the mechanism was 7.9 kW, when recorded at the lower PTO (rotation speed of 150 rpm). The measured power consumption increased as the PTO rpm increase to reach about 44 kW, at PTO rotation of 400 rpm. The computed theoretical power started at 5 kW, corresponding to the 150 PTO rpm. As substituting higher PTO rpm, the theoretical power computed increase to reach 42 kW, at 400 rpm. The table show that the percent difference between the measured and computed power still positive. It is understood that the computed power that drive the turning augers did not include friction due to transmission from PTO to turning augers. The percent difference between measured and theoretical power was as large as 36.7% for the lower speed decreased to be less than 6.3 % at higher speed of 400 rpm. The lower values of computed theoretical power in respect to measured power may refer to the fact that the theoretical calculated power was only to compute power required to overcome the resistance of compost windrow to be turned and did not include power required drive the transmission system from the tractor to the augers

2-Hydraulic power requirement

From the results the hydraulic motor was one horsepower to push compost turner. The maximum flow to drive hydraulic motor 21.28 litter per mint and the pressure 250 psi. The specification of the piston to lift and lower the turning box as flow: Cylinder piston 90mm and the length of the piston rod 90cm .the flow to lift the and lower the turning box was14.32 litter per mint. The flow for the other two piston to lift and lower the push axel was 19.8 litter per mint. from the previous results the maximum flow for the system 41 litter per mint (hydraulic motor was 21.28 and the flow for pushing axel was 19.8 litter per mint) are safe ,because the delver flow of the tractor pump is 44 litter per mint.

The quality performance of turner operation determined the forward motion of the turner along the windows to be 0.2 km/h. This result was obtained according to several trials, several experiments and field operation of the turner. The compost turning system which consists of the tractor and the turner should move together at the previously mentioned advancing speed. It was difficult to obtain such constant low speed from the tractor. The alternative solution was to leave the tractor on the neutral position and push the system with a suitable hydraulic power obtained from the remate hydraulic power system of the tractor. A hydraulic motor fixed on the turner axle drive a differential system that move the turner wheels at the slow speed mentioned previously. The moving turner pushes the tractor at this slow speed through the hitch point. Calculation considering rolling resistance on the firm soil on which the compost windrow formed show that power required to push the system at this slow speed is less than one horsepower. According a one horsepower hydraulic motor was used and tested for uniform forward motion. The repeated operation of the turning system proved that the one hp motor was satisfying and saving to push the system at this advancing slow speed.

CONCLUSIONS AND RECOMMENDATIONS

1- The minimum power requirement calculated was 5 kW and the maximum 42 kW, at 150 rpm and 400 rpm respectably.
2- The minimum power requirement measured was 8 kW and the maximum 45. kW, at 150 rpm and 400 rpm respectably.
3- The difference between the calculated and the measured value As the average 10%
4- The difference between the calculated and the measured value of torque As the average 10 %
5- The maximum flow to drive hydraulic motor one horse 21.28 litter per mint and the pressure 250 psi
6- The maximum flow for the system 41 litter per mint.
7- The recommended power to drive compost turner 3 m is 75 horsepower .

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التحليل الرياضى والقدرات المقاسة لألة تقليب الكمبوست المصنعة محليا

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تمعظمت مشكلة المخلفات الزراعية والحيوانية فى الأونة الأخيرة حيث وصلت اجمالى النواتج الثانوية للمحاصيل الزراعية ٣٢ طن/سنة والمخلفات الحيوانية (روث خيوانى) ١٢ مليون طن/سنة حسب احصائيات وزارة الزراعة لسنة ٢٠٠٧. أنتاج الأسمدة العضوية أحد الحلول للتخلص من المواتج الثانوية الزراعية والحيوانية وكذلك زيادة الدخل القومى والمساهمة فى انتاج غذاء زراعى أمن. لذلك تم تطوير وتصميم وتصنيع الة تقليب للمبوست محلياً (بيومى ٢٠٠٧).

الهدف من هذه الدراسة تفعيل وتعظيم القدرات المطلوبة للتشغيل الأمثل لألة القليب وذلك عن طريق الحساب النظرى والقياس الفعلى للقدرات الازمة لادارة النظام الميكانيكى للتقليب والنظام الهيدروليكى لدفع الالة والجرار وذلك بأستخدام أجهزة القياس والتحليل البعدى والرياض.

أهم مكونات الالة: الشاسيه - وحدة التقليب – وحدة الاتزان – النظام اليكانيكى – النظام الهيدروليكى.

النظام اليكانيكى ويتكون من مصدر القدرة من عمود الأدارة الخلفى للجرار – وصلتان كردان متصلتان على التوالى لتوصيل الحركة من الجرار الى صندوق تروس – صندوق تروس مخفض للحركة من ١,٥ الى ١ لتوصيل الحركة لدرافيل التقليب بالالة عن طرق تروس وجنزير.

النظام الهيدروليكى ويتكون من طلمبة الهيدروليك بالجرار التى تمد منظم التوزيع بضغط هيدروليكى ٢٥٠ ض ج وتصرف ٤٤ لتر/د – منظم التوزيع: ذو ثلاث أذرع توجيه ١- لمد بساتم الرفع والخفض لصندوق التقليب بالضغط ٢- لمد بساتم الرفع والخفض لعجل دفع الالة والجرار بالضغط ٣- مد المحرك الهيدروليكى بالضغط لدفع الالة والجرار أثناء التقليب.

التجارب المعملية والحقلية:

التجارب المعملية: تم معايرة احهزة قياس العزوم مكانيكيا باستخدام اوزان معلومة وتقدير العزوم عند سرعات دورانية يتم اختبار الالة عندها.

التجارب الحقلية: تم أجراء التجارب بمحطة بحوث الجميزة بالغربية بأستخدام جرار روسى ٧٥ حصان. وتم تعليق جهاز قياس العزوم بين عمود الادارة الخلفى للجرار ووصلة الكردان وتم قياس العزوم اثناء التقليب عند سرعات مختلفة وتم التسجيل للبيانات المقاسة من خلال جهاز تجميع البيانات وحفظها على الكمبيوتر المعملى المستخدم فى عملية القياس.

- التحليلات البعدية والرياضية لالة التقليب المحلية.

١- تم عمل التحليل البعدى لمعادلات حساب القدرات الازمة لعملية التقليب الميكانيكى للمواد المقلبة(مخلفات زراعية وحيوانية) لحساب القدرة الازمة للتقليب.

٢- تم حساب القدرات المطلوبة لدرفيلى التقليب (النظام الميكانيكى) من خلال حساب: المقاومة الديناميكية لحركة الدرافيل - المقاومة الديناميكية لحركة السكاكين على الدرافيل - قوى الاحتكاك الناتجة من الاحتكاك بين سكاكين التقليب والمواد المقلبة.

٣- تم حساب القدرة الكلية للقوى الثلاث السابقة وذلك عند السرعات المختلفة التى تم القياس عندها بجهاز قياس العزوم. وتم مقارنة القدرة المحسوبة والمقاسة.

٤- تم حساب القدرات المطلوبة للنظام الهيدروليكى لرفع وخفض بساتم صندوق التقليب.
٥- تم حساب القدرات المطلوبة للنظام الهيدروليكى لرفع وخفض بساتم الخاصة بعجلات الدفع.
٦- تم حساب القدرات المطلوبة لادارة المحرك الهيدروليكى الدافع لالة التقليب والجرار.
٧- تم حساب الاقطار المناسبة لاسطوانات البساتم المستخدمة لرفع وخفض صندوق التقليب وعجلات الدفع.

أهم النتائج والتوصيات؛-

١-أعلى قدرة مقاسة ٤٥,٢ ك وات واعلى قدرة محسوبة٤٢ ك وات عند سرعة٤٠٠ لفة | د حسب الجدول التالى.

Rotational PTO rpm	TOURQE N.m	Power , kW		Difference%
		Measure	calculation	
١٥٠	٥٠٥	٧,٩	٥	٣٦,٧
٢٠٠	٥١٩	١٠,٩	٨	٢٦,٦
٢٥٠	٦٦١	١٧,٣	١٥	١٣,٣
٣٠٠	٧٦٥	٢٤,٠	٢٢	٨,٣
٣٥٠	٩٢٠	٣٣,٧	٣١	٨,٠
٤٠٠	١٠٧٠	٤٤,٨	٤٢	٦,٣

٢-أقطار اسطوانات البساتم المستخدمة فى النظام الهيدروليكى من الحسابات التصميمية كانت ٩ سم باستخدام اربع بساتم أثنين لرفع صندوق التقليب وأثنين لرفع عجلات الدفع.
٣-القدرة اللازمة لدفع النظام(جرار ولة التقليب) ١ ك وات.
٤-التصرف الازم للمحرك الهيدروليكى ٢١,٢٨ لتر/د
٥- التصرف اللازم لرفع وخفض بساتم صندوق التقليب ١٤,٣٢ لتر/د.
٦- التصرف اللازم لرفع وخفض بساتم عجلات الدفع ١٩,٨ لتر/د