FARM MACHINERY AND POWER

DESIGN CONSIDERATIONS OF A COMBINATION MACHINE FOR CHOPPING AND MILLING *ARUNDO DONAX* (PART I)

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ABSTRACT

The aim of present work was to carry out theoretical analysis to design to design a combination machine to be suitable for chopping and milling Arundo donax plants and then to use the design considerations afterward to manufacture and evaluate the performance of the combination machine to reach up the optimum operating parameters. The design considerations included estimation of the following parameters: number of knives on the cutter-head, cutter-head speed, feed rolls speed and Arundo donax density in the feeding throat. The design considerations were evaluated taking into consideration the following indicators: theoretical cut length, theoretical capacity and required total power; furthermore determination of cutter-head shaft diameter as well as hammer mill shaft diameter. The design results reveal that the highest values of theoretical cut length and theoretical capacity were (29.14 mm and 756 kg/h) respectively, while the lowest values of chopping and milling units power were (9.54 and 20.17 kW), respectively and the values of cutter-head shaft diameter and hammer mill shaft diameter were (50 and 60 mm), respectively under the following conditions: number of knives on the cutter-head of 2 knife, cutter-head speed of maximum 1400 rpm and Arundo donax density in the feeding throat of 6.18 kg/m^3 with adjusting feed rolls speed of 1.36 m/s.

1. INTRODUCTION

Arundo donax (giant reed, giant cane) is one of the largest grass species. A plant that grows in dense stands, it is found in many subtropical and warm-temperate areas of the world. It is thought to be native to eastern Asia, but the precise extent of its native distribution is unknown. *Arundo* has been introduced around the world as an ornamental/crop species, for erosion control, and for the production of reeds (musical instruments, construction, paper and pulp). It has become invasive in many places throughout the world, primarily in riparian habitat.

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Where *Arundo* invades, it often forms dense stands, resulting in a wide range of impacts to natural ecological systems as well as human created infrastructure (**Lowe *et al.* 2000**). *Arundo donax* is a tall, erect, perennial cane-or reed-like grass, 2 to 8 meters high. It is one of the largest of the herbaceous grasses. The fleshy, almost bulbous, creeping root stocks form compact masses from which arise tough, fibrous roots that penetrate deeply into the soil. The culms reach a diameter of 1 to 4 cm and commonly branch during the second year of growth. These culms are hollow, with walls 2 to 7 mm thick and divided by partitions at the nodes. The nodes vary in length from 12 to 30 cm. The leaves are conspicuously two-ranked, 5 to 8 cm broad at the base and tapering to a fine point. The bases of the leaves are cordate and more or less hairy-tufted, persisting long after the blades have fallen (**Hoshovsky, 2003**). Giant reed is one of the most promising crop for energy production for the Mediterranean climate of Europe and Africa, where it has showed advantages as indigenous crop (already adapted to the environment), durable yields, and resistance to long drought period. Several field studies have highlighted the beneficial effect of giant reed crop on the environment due to its minimal soil tillage, fertilizer and pesticide. Furthermore it offers protection against soil erosion (**Heaton 2004**). *Arundo donax* is strong candidate for use as a renewable bio-fuel source because of its fast growth rate, ability to grow in different soil types and climatic conditions. *A. donax* will produce an average of three kilograms of biomass per square meter (25 Mg per acre) once established. The energy density of the biomass produced is 17 MJ/kg regardless of fertilizer usage (**Angelini *et al.* 2005**).

In USA, a great effort (money and time) has been spent in developing effective procedures for *Arundo* removal. Management techniques such as manual and mechanical removal and fire, implemented have proven sub-optimal. Manual or mechanical removal often results in dispersal of plant propagules, including rhizome fragments and stem nodes. Root masses can be missed and/or left behind, removal of the biomass can spread the invasion, and equipment used for physical extraction can spread the plant through fragments being caught in the machinery. In addition to these downfalls, manual eradication and removal of biomass is expensive. The price tag for only the above ground biomass cutting (not including removal from the site) is $5000USD per acre (**Bell 1993**). The larger machines can be

used to remove above ground vegetation and dig up underground rhizomes. Vegetation can be chopped or trimmed back with a tractor. This non-selective method works well in monotypic stands of undesirable plants where there are no native plants nearby. This is often done in preparation of removal of underground rhizomes or prior to chemical treatment (**Forest *et al.* 2003**). Crushing of crop residues is on the increase with the global quest for sourcing of renewable energy through pre-processing of bio-masses. Physical and mechanical properties of biomasses species and varieties are very important when considering the energy requirements for particle size reduction of agricultural residues. Of the various types of grinding equipment available, hammer mills are the best known equipment used for the shredding/grinding, in which the material fragment are subjected to complex forces and then the resulted particles are used in the following operations from the pellet obtaining technology (**Moiceanu *et al.* 2012**). The performance and productivity of size reduction equipments for fibrous materials were analyzed and recommended the hammer mill which accepts the whole stalks without the need for manual chopping. Various size reduction equipments are available in the market based on the classification of the size reduction equipment. They added an extended layout of this classification and suggested hammer mill, knife mill and disc mill as the proper equipment for biomass pulverization. Due to high size reduction ratio, good control of particle size range with relatively good cubic shape of particles, hammer mills are wildly used for grinding of different materials are available. Knife mills (or choppers) work successfully for shredding forages under various crops and machine conditions (**Hoque *et al.* 2007**). So, such care had to be taken to design a combination machine to be suitable for chopping and milling the *Arundo donax* plant to minimize operational cost and improve product quality. Thus, the objectives of this research are to carry out theoretical analysis to design a chopping and milling units for crushing *Arundo donax* plant. Addition to, determine the suitable power source for operating the combination machine.

2. MATERIALS AND METHOD

The design considerations were conducted through the year of 2014 at Department of Agricultural Engineering, Faculty of Agriculture, Zagazig University to design a combination machine then to use the design

considerations afterward to manufacture a machine for chopping and milling *Arundo donax* to reach to optimum operating efficiency.

2.1. MATERIALS:

2.1.1. The used plant:

Arundo donax (Giant reed) was used in this study.

2.1.2. The designed combination machine:

Theoretical analysis of a combination machine suitable for chopping and milling *Arundo donax* was carried out to determine optimum design parameters. It consists of the main following parts as shown in Fig. (1):

A- Chopping unit:

The chopping unit consists of the following parts: feed inlet, feeding rolls, chopping drum and outlet.

B- Milling unit:

The milling unit consists of the following parts: feed hopper, milling chamber, milling drum, cover and concave.

C- Power transmission:

The combination machine was powered by a small tractor type (Kubota-M5400) with engine power of 42.65 kW (58hp) at rated speed of 3000 rpm. Also a flywheel was attached to the chopping shaft to stop the reduction of the rotating speed noticed whenever much stalk materials were added to the chopping chamber. Fig.(2).

D- The main frame:

The main frame of the combination machine for chopping and milling *Arundo donax* stalks will be made from iron steel sheet. It includes three hitching points to be drawn with a small tractor and take its rotating motion from (pto) using pulleys and V-Belts.
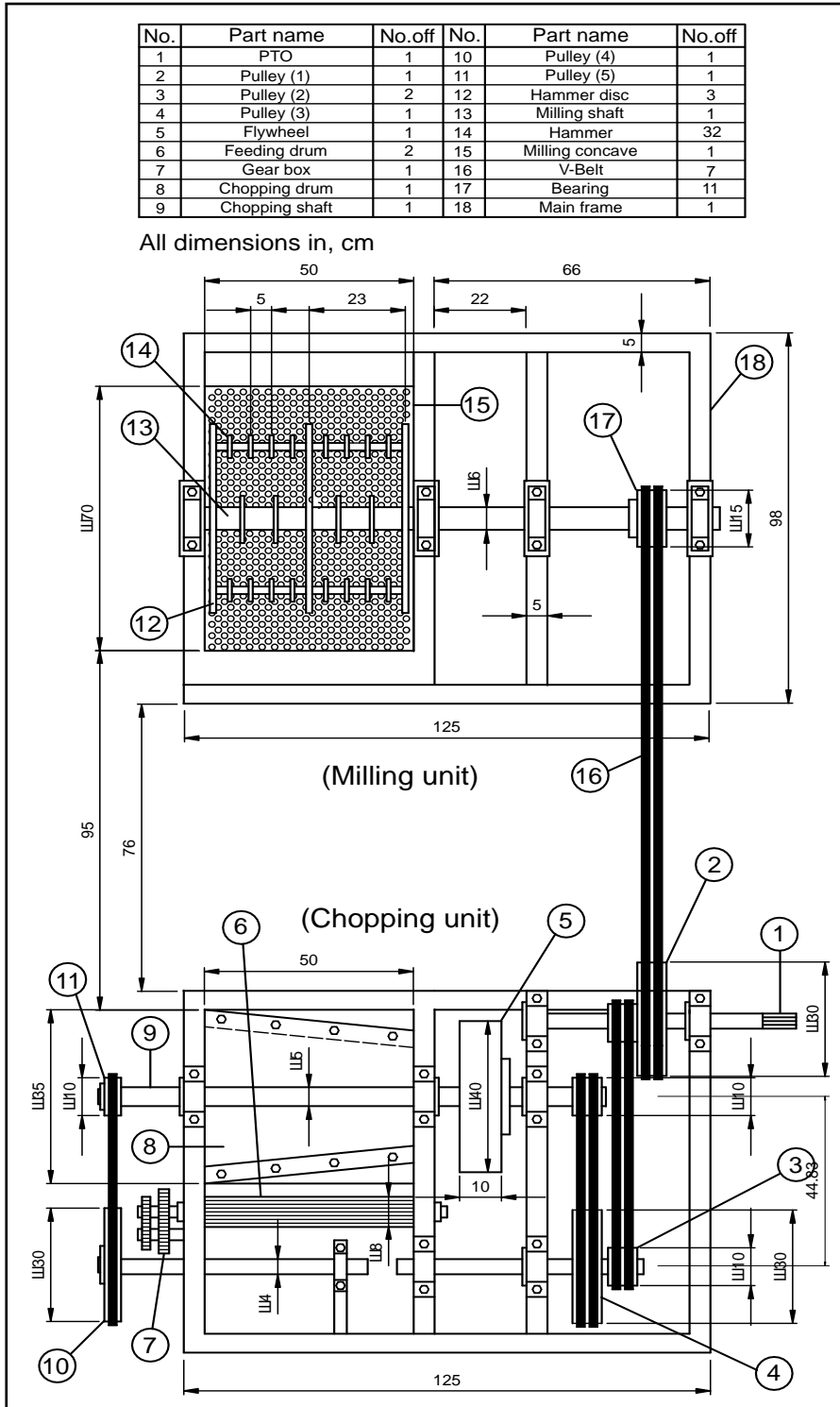
2.2. METHOD:

2.2.1. Physical and engineering properties of *Arundo donax* plant:

Physical and engineering properties *of Arundo donax* plant under study have to be considered to study their effect on the performance of the combination machine.

A random sample of about 100 stalks was taken to determine some physical properties such as: length of plant, diameter of stalk, stalk thickness, particles density and mass of 10 stalks; and some engineering properties such as: coefficient of friction and repose angle.

Fig.1: An elevation and side view of the chopping and milling machine.

No.	Part name	No.off	No.	Part name	No.off
1	Draw bar	1	13	Pulley (5)	1
2	PTO	1	14	Chopping inlet	1
3	Pulley (1)	1	15	Ground wheel (A)	2
4	V-Belt	7	16	Feeding drum	2
5	Pulley (2)	1	17	Chopping chamber	1
6	Chopping shaft	1	18	Chopping outlet	1
7	Milling shaft	1	19	Tunnel	1
8	Pulley (3)	1	20	Hammer	32
9	Main frame	1	21	Sieve	1
10	Flywheel	1	22	Milling pan	1
11	Gear box	1	23	Ground wheel (B)	4
12	Pulley (4)	1	24	Product outlet	1

All dimensions in, cm

ELEVATION

SIDE VIEW

No.	Part name	No.off	No.	Part name	No.off
1	PTO	1	10	Pulley (4)	1
2	Pulley (1)	1	11	Pulley (5)	1
3	Pulley (2)	2	12	Hammer disc	3
4	Pulley (3)	1	13	Milling shaft	1
5	Flywheel	1	14	Hammer	32
6	Feeding drum	2	15	Milling concave	1
7	Gear box	1	16	V-Belt	7
8	Chopping drum	1	17	Bearing	11
9	Chopping shaft	1	18	Main frame	1

All dimensions in, cm

Fig.2: Schematic diagram showing power transmission from (pto) to chopping and milling units.

Dimensions were measured using the digital vernire caliper of an accuracy of 0.01 mm. The weights of samples were conducted using two electrical digital balances. Coefficient of friction of *Arundo donax* plant was measured using an inclined plane apparatus on surface of steel sheet metal. The measurements were repeated ten times for each sample. Some physical properties of the used *Arundo donax* plant were illustrated in Table (1).

Table 1: Some physical and engineering properties of *Arundo donax*.

Item	Value
Length of plant, mm	2400
Diameter of stalk, mm	22.00
Thickness of stalk, mm	4.00
Particles density, kg/m^3	160.65
Coefficient of friction	0.50
Repose angle, degree	30

2.2.2. Design considerations:

Machinery designers and operators require more information concerning the required power for chopping *Arundo* stalks, which affects to a great extent the performance of chopping and milling machines. If this information is available, it is possible to design more developed machines and improve the performance of the existing machines. Chopping *Arundo* stalks can be performed using two cutting blades; the first blade is, responsible for cutting operation and the other is used as a counter-shear (i.e. to provide resistance against the cutting force).

The design of the cutting knife and the stalk type are considered the most affecting factors on performance of chopping machinery. There is a variation in the shear stress of *Arundo* stalks due to the stalk diameter and the thickness. However, the *Arundo* stalks differ in their physical and mechanical properties. The moisture content level and the fiber and lignin content are considered as a limited factor to the stalk physical and mechanical properties.

2.2.3. Design of chopping and milling units:

The dimensions of the chopping and milling units were selected according to the physical properties of the investigated plant such as; dimensions of feeding throat, shape of feeding rolls, shape of knives and diameter of concave openings that is executed efficiency optimum. While the remaining dimensions such as; hammer length, hammer width and hammer weight

were designed according to the affecting force (shear force or impact force) at the end of the hammer to mill the chopped *Arundo donax*.

♦ *Calculation of the theoretical cut length:*

The length of cut is controlled by the peripheral speed of the feed rolls relative to the speed of the cutter-head. A smooth feed roll is placed near the shear bar to maintain the grip on the *Arundo* as close to the shear bar. In determining peripheral speeds of feed rolls, the pitch (effective) diameter is slightly less than the outside diameter. The theoretical length of cut can be calculated using the following equation (**Srivastava** *et al.* **1993**):

$$L_{th} = \frac{60000 \times v_f}{\lambda_k . n_c}$$ (1)

Where: L_{th} = theoretical length of cut, mm.

v_f = feed velocity, m/s = peripheral speed of feed rolls.

λ_k = number of knives on the cutter-head.

n_c = rotational speed of cutter-head, rev/min.

Some particles will be longer than the theoretical length when stems are not oriented parallel to the direction of feed. Others will be shorter than the theoretical length when the arrival of the ends of stems does not coincide with the arrival of a cutter-head knife. The actual length of cut is usually close to the theoretical length because the stalks are oriented nearly perpendicular to the shear bar. For direct-cut *Arundo,* actual average lengths of cut are generally about 50% longer than the theoretical length. When *Arundo* stalks are being chopped, the average actual length of cut is much longer than the theoretical length due to the random stalks alignment. Large increases in theoretical length of cut are made by removing knives from cutter-head. Smaller changes in length of cut are made by adjusting the feed velocity. The feed rolls will separate further to accommodate the reduced feed velocity until the maximum separation distance is reached; further reductions in feed velocity will then reduce the capacity of the machine.

♦ *Calculation of the theoretical capacity:*

The theoretical capacity of a precision-cut *Arundo* chopper can be calculated using the following equation (**Srivastava** *et al.* **1993**):

$$\dot{m}_f = \frac{\rho_a \, A_t \, L_{th} \, \lambda_k \, n_c}{6 \times 10^8}$$ (2)

Where: $\dot{m}_f$ = theoretical capacity or feed rate, kg/s.

ρ_a = density of *Arundo* in the throat, kg/m^3.

A_t = throat area, cm^2.

Before using equation (2) to calculate the maximum allowable feed rate, the maximum throat area must be calculated. As follows:

$$A_t = \text{cutter-head width} \times \text{throat height} \tag{3}$$

After *Arundo* was chopped by the cutter-head, centrifugal force holds it against the housing as the cutter-head moves it toward the exit; the housing terminates at the bottom or rear of the cutter-head to allow the chopped material to escape. In a cut-and-throw machine, the cutter-head imparts sufficient energy to throw the chopped material to milling unit.

♦ ***Determination of the total power for combination machine:***

The required power of a combination machine is so large that chopping and milling capacities can be limited by available power. Power is consumed in feeding, conveying and compressing the material to be cut and mill. There after the final product can be packaged in sacks. The unexploited power losses in a combination machine include bearing friction, friction of the cut material on the cutter-head housing, impact between the hammers and materials, and pumping of air at the cutter-head and blower. Within the cutter-head and the mill hammers, energy is required for compressing shearing and impacting, for acceleration and air movement, and for overcoming friction at the housing. Material parameters such as shear strength, moisture content and friction coefficient, and machine parameters such as sharpness of the knife and length of cut will all affect the distribution of power in the combination machine.

♦ ***Calculation of the required power for chopping unit:***

Chopping usually requires the greatest energy at the cutter-head, while friction generates the greatest energy requirement at the blower. The power requirement for chopping can be calculated using the following equation: (**Srivastava *et al.* 1993**):

$$P_c = \frac{1000 \, C_f \, F_{smax} \, \dot{m}_f}{\rho_a \, L_{th}} \tag{4}$$

Where: P_c = chopping power, kW

C_f = ratio of average to maximum specific cutting force.

F_{smax} = maximum specific cutting force, N/mm of counter shear length.

$\dot{m}_f$ = feed rate, kg/s.

ρ_a = density of material in the throat, kg/m^3.

L$_{th}$ = theoretical length of cut, mm.

Thus, as with straight cutting, C$_f$ is about equal to 0.64 for typical oblique cutting. Chopping power varies with feed rate and length of cut; the specific cutting energy provides a better index for comparing machines of differing design. Specific cutting energy is defined as follows:

$$E_{sc} = \frac{1000\ C_f\ F_{smax}}{\rho_a} \qquad (5)$$

Where: E$_{sc}$ = specific cutting energy, J.m/kg.

By measuring power consumption while changing feed velocity and removing knives to change the theoretical length of cut. Once a value of E$_{sc}$ is known, the following equation can be used to calculate specific cutting energy:

$$E_{sc} = \frac{P_c\ L_{th}}{\dot{m}_f} \qquad (6)$$

Notice from equation (4) that E$_{sc}$ is proportional to the maximum specific cutting force. Thus, maintaining sharp knives and shear bar/knife clearance is very important to reduce the power requirement for chopping *Arundo*. Power to overcome friction between the cut stalks and the cutter-head or blower housing can be calculated by using the following equation:

$$P_{fc} = \frac{\beta\ \mu\ \dot{m}_f\ V_{pc}^2}{1000} \qquad (7)$$

Where: P$_{fc}$ = power absorbed by rubbing friction, kW.

β = average arc of housing by chopped material, radians.

μ = coefficient of friction between *Arundo* and steel housing.

$\dot{m}_f$ = feed rate, kg/s.

v$_{pc}$ = peripheral velocity of cutter-head or blower, m/s.

Required power to accelerate the chopped material at the blower is derived by assuming that the *Arundo* leaves the blades at about the peripheral speed of the blades:

$$P_{ac} = \frac{\dot{m}_f\ V_{pc}^2}{2000} \qquad (8)$$

Where: P$_{ac}$ = power to accelerate the chopped material, kW.

Both the cutter-head and the blower move air, although the latter at a greater rate. According to well-established fan laws, fan power varies with the cube of the peripheral speed; the following equation was derived for the approximate air power:

$$P_{air} = \frac{V_{pc}^3}{16600} \tag{9}$$

Where: P_{air} = power to move air, kW.

By combining equations 4, 7, 8 and 9 (chopping power, power absorbed by rubbing friction, power to accelerate the chopped material and power to move air), the following equation can be used to calculate the chopping unit power:

$$P_{co} = P_c + P_{fc} + P_{ac} + P_{air} \tag{10}$$

♦ **Calculation of the required power for milling** *unit:*

Milling usually requires the greatest power at the hammer mill, while friction or accelerate the milled material generates the lowest energy requirement at the concave. When the maximum shear force of the investigated equal to the kinetic force at the end of hammer, kinetic force of hammer equal 1108 N, the following equation can be used to be calculated or assumed for some parameters (**Srivastava** *et al.* **1993**):

$$F_s = \frac{m \, v^2}{2 \, r_h} \tag{11}$$

Where: F_s = maximum shear force on stalks = kinetic force of hammer, N.

m = mass of hammer, kg.

v = peripheral velocity of hammer mill, m/s.

r_h = radius of hammer, m.

Therefore, the required milling power can be calculated using the following equation:

$$P_m = F_s \, \omega \, r_s \tag{12}$$

Where: P_m = milling power, kW.

ω = Angular velocity of hammer shaft, rad/s

r_s = radius of hammer, m.

$$\omega = \frac{2\pi \times N}{60} \tag{13}$$

Where: N = rotation speed of hammer shaft, rpm.

Power to overcome friction between the milling materials and housing of the milling hammers can be calculated by using the following equation:

$$P_{fm} = \frac{\beta \, \mu \, \dot{m}_f \, V_{pm}^2}{1000} \tag{14}$$

Where: P_{fm} = power absorbed by rubbing friction, kW.

β = average arc of housing by milled material, radians.

μ = coefficient of friction between *Arundo* and steel housing.

$\dot{m}_f$ = feed rate, kg/s.

v_{pm} = peripheral velocity of milling hammers, m/s.

Required power to accelerate the milled material at milling hammers can be calculated by using the following equation:

$$P_{am} = \frac{\dot{m}_f \, V_{pm}^2}{2000} \tag{15}$$

Where: P_{am} = power to accelerate the milled material, kW.

By combining equations 12, 14 and 15 (milling power, power absorbed by rubbing friction and power to accelerate the milled material), the following equation can be used to calculate the milling unit power:

$$P_{mo} = P_m + P_{fm} + P_{am} \tag{16}$$

By combining equations 10 and 16, an equation for the total consumption power, (chopping unit power and milling unit power) the total power of a combination machine; the following equation can be used to calculate the required total power:

$$P_t = P_{co} + P_{mo} \tag{17}$$

♦ *Design of machine shafts:*

It is a great importance to design both cutter-head shaft and hammer mill shaft so as to save them from over loads and high stresses. Both shafts are subjected to combine torsion and bending stresses. So, they are calculated on the combined stress. The diameters of both shafts can be calculated according to the maximum shear stress theory as following (**Khurmi and Gupta, 2005**):

$$Z_{max} = \frac{1}{2}\sqrt{\delta^2 + Z^2} \qquad\qquad Z_{max} = \frac{1}{2}\sqrt{\left(\frac{32\,M}{\pi\,d^3}\right)^2 + 4\left(\frac{16\,T}{\pi\,d^3}\right)^2}$$

$$Z_{max} = \frac{16}{\pi\,d^3}\sqrt{M^2 + T^2} \qquad\qquad Z_{max} = \frac{16}{\pi\,d^3}\sqrt{K_m^{\,2}\,M^2 + K_t^{\,2}\,T^2} \tag{18}$$

Where:

Z_{max} = Maximum shear stress, Z_{max} = 500 kg/cm^2

δ = Bending stress, kg/cm^2 Z = Torsion stress, kg/cm^2

T = Maximum torque, kg.cm d = Diameter of cutter-head, cm

M = Maximum bending moment, kg.cm

K_m = Shock factor for bending, K_m = 2.0

K_t = Shock factor for tension, K_t = 1.5

♦ **The theoretical analysis and design considerations were carried out as a function of changing in the following parameters:**

- Four different number of knife on the cutter-head of (1, 2, 3 and 4 knife).
- Four different cutter-head speeds of (1000, 1200, 1300, and 1400 rpm) or (18.32, 21.98, 23.81 and 25.64 m/s).
- Four different feed rolls speed (0.89, 1.05, 1.23, and 1.36 m/s).
- Four different *Arundo donax* densities in the feeding throat (5.00, 5.40, 5.74, and 6.18 kg/m^3).

The design considerations were studied taking into consideration the following indicators: theoretical cut length, theoretical capacity and required total power; furthermore determine both cutter-head shaft and hammer mill shaft diameters.

3. RESULTS AND DISCUSSION

The obtained results will be discussed under the following items:

3.1. Influence of cutter-head speed and number of cutter-head knives on theoretical cut length:

The effect of cutter-head speed on theoretical cut length is shown in Fig.3. Results show that increasing cutter-head speed from 1000 to 1400 rpm measured at different feed rolls speed of about 0.89, 1.05, 1.23 and 1.36 m/s, decreased theoretical cut length from 26.70 to 19.07, from 31.50 to 22.50, from 36.90 to 26.36 and from 40.80 to 29.14 mm, respectively at constant number of knives on the cutter-head of 2 knives.

The effect of number of the cutter-head knives on theoretical cut length is given in Fig.3. Results show that increasing number of the cutter-head knives from 1 to 4 knives measured at the same feed rolls speed, decreased theoretical cut length from 38.14 to 9.54, from 45.00 to 11.25, from 52.71 to 13.18 and from 58.29 to 14.57 mm, respectively at constant cutter-head speed of 1400 rpm.
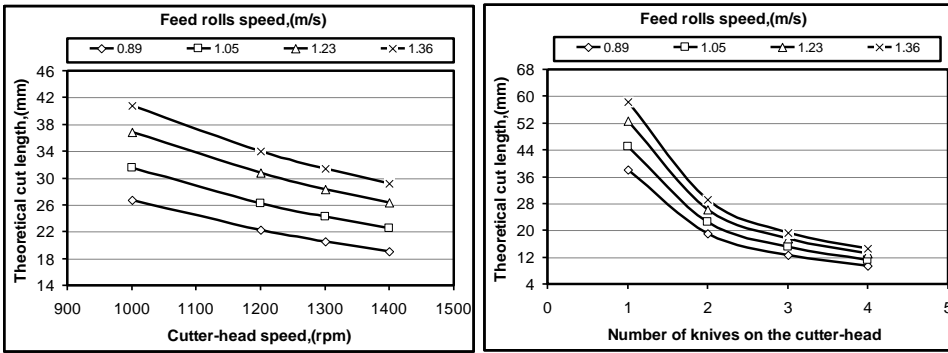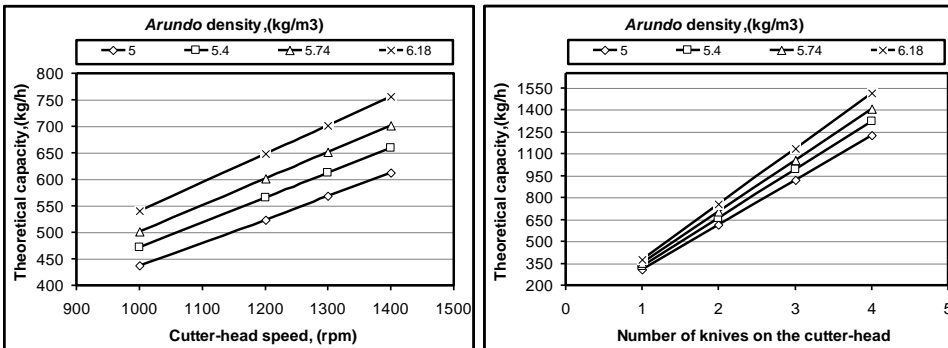
Fig.3. Effect of cutter-head speed and number of cutter-head knives on theoretical cut length.

The theoretical cut length decreased by increasing cutter-head speed due to the increase in cutting force and friction of the *Arundo donax* stalks on the cutter-head. And also, the theoretical cut length decreased by increasing number of the cutter-head knives due to the increase in number of strikes on *Arundo donax* stalks per time unit.

3.2. Influence of cutter-head speed and number of cutter-head knives on theoretical capacity:

The effect of cutter-head speed on theoretical capacity is shown in Fig.4. Results show that increasing cutter-head speed from 1000 to 1400 rpm measured at different *Arundo* density in the throat of about 5.00, 5.40, 5.74 and 6.18 kg/m^3, increased theoretical capacity from 437 to 612, from 472 to 660, from 501 to 702 and from 540 to 756 kg/h, respectively at constant number of knives on the cutter-head of 2 knives.

Fig.4. Effect of cutter-head speed and number of cutter-head knives on theoretical capacity.

The effect of number of the cutter-head knives on theoretical capacity is given in Fig.4. Results show that increasing number of the cutter-head knives from 1 to 4 knives measured at the same *Arundo* density in the throat,

increased theoretical capacity from 306 to 1223, from 330 to 1321, from 351 to 1404 and from 378 to 1512 kg/h, respectively at constant cutter-head speed of 1400 rpm. The theoretical capacity increased by increasing cutter-head speed on account of the increase of chopped *Arundo donax* flow through the feeding opening at the same time unit. With respect to the theoretical capacity increased by increasing number of the cutter-head knives due to the increase in number of cutter-head action at the same time unit.

3.3. Influence of feed rate and theoretical cut length on required total power:

The effect of feed rate on the required total power is shown in Fig.5. Results show that increasing feed rate from 612 to 756 kg/h measured at different *Arundo* density in the throat of about 5.00, 5.40, 5.74 and 6.18 kg/m^3, increased required total power from 29.72 to 31.67, from 29.10 to 30.91, from 28.65 to 30.35 and from 28.13 to 29.71 kW, respectively at constant theoretical cut length of 26.36 mm. The effect of theoretical cut length on the required total power is shown in Fig.5. Results show that increasing theoretical cut length from 19.07 to 29.14 kg/h measured at the same *Arundo* density in the throat decreased required total power from 35.38 to 30.56, from 34.35 to 29.89, from 33.59 to 29.38 and from 32.72 to 28.82 kW, respectively at constant feed rate of 756 kg/h. The required total power increased by increasing feed rate on account of the increase of the excessive load of materials on the machine devices which consumed more power. With respect to the required total power decreased by increasing theoretical cut length due to the decrease of the loaded materials on the machine devices which consumed less power.
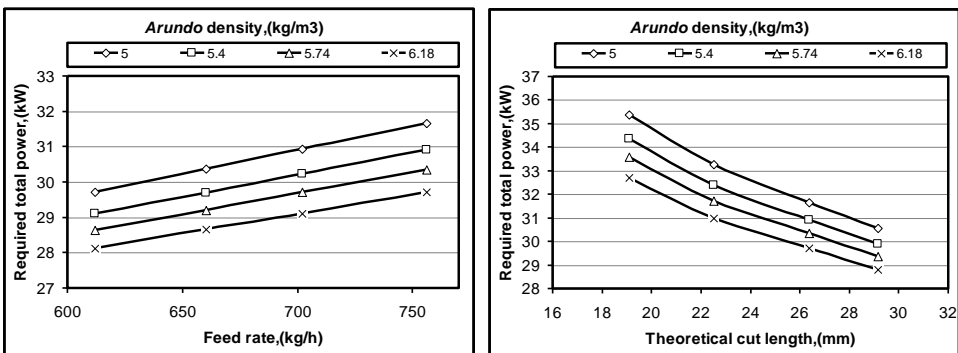
Fig.5. Effect of feed rate and theoretical cut length on the required total power.

3.4. Design the cutter-head shaft taking working power of 10 kW.

a) Vertical loads subjecting on the cutter-head shaft:

♦ Torque transmitted by the cutter-head shaft (T) can be calculated as follows: (**Khurmi and Gupta 2005**).

$$T = \frac{Po \times 60}{2\pi N} = \frac{10.5 \times 10^3 \times 60}{2 \times 3.14 \times 1400} = 71.65\,N.m \cong 72 \times 10^3\,N.mm \qquad (19)$$

Assume the torque at pulley-3 (Point C) and pulley-4 (Point D) is in same value; therefore the tangential force (F_1 and F_2) will be as follows:

$$F_t\,at\,C\,and\,D = \frac{T}{r} = \frac{72 \times 10^3}{53} = 1360\,N \qquad (20)$$

♦ **Determination of F_1 and F_2:**

$$F_1 = T_1 + T_2 \qquad (21)$$

Where: F_1 = Tension force on pulley-3 and pulley-4, N

T_1 = Tension on the tight side of belt, N

T_2 = Tension on the slack side of belt, N

$$T_m = F_{max} \times r \qquad (22)$$

Where: T_m = Torque at pulley shaft, N.mm

F_{max} = Maximum tension force at cutter-head shaft, N

r = Radius of pulley-3 and pulley-4, mm

$T_m = 1360 \times 53 = 72 \times 10^3\,N.mm$

$$T_m = (T_1 - T_2) \times r \quad \therefore\ (T_1 - T_2) = 1360 \qquad (23)$$

$$2.3\,log\,(T_1/T_2) = \mu\phi\,cos\alpha \qquad (24)$$

Where: μ = Coefficient of friction, 0.3

α = Groove angle of pulley-3 and pulley-4, ($30°$-$40°$)

ϕ = Angle of contact, rad.

$$\phi = (180 - 2\alpha).\pi/180 = (180 - 2 \times 35)3.14/180 = 1.92\,rad \qquad (25)$$

$$\because 2.3\log\frac{T_1}{T_2} = 0.3 \times 1.92 \times cos35 \ \therefore\ 2.3\log\frac{T_1}{T_2} = 0.472 \qquad (26)$$

$$\therefore \frac{T_1}{T_2} = e^{0.472 \times 2.3} \quad \therefore\ \frac{T_1}{T_2} = 2.96 \qquad (27)$$

$$\because (T_1 - T_2) = 1440 \quad \therefore 2.96T_2 - T_2 = 1440 \quad \therefore 1.96T_2 = 1440$$

$So, T_2 = 693.8 \approx 694\ N\quad and\quad T_1 = 2.96 \times 694 = 2054.24 \approx 2054\ N$

$Then\quad F_1 = 2748\ N\quad and\ also\quad F_2 = 2748\ N$ (28)

But each pulley transfer tension force at 16° from horizontal. So, the vertical tension force acting on the pulleys (3 and 4) equal:

$$T_V = T.sin16° = 2748 \times 0.276 = 757.45 \cong \boldsymbol{757\ N}$$ (29)

♦ Vertical force acting on the chopping shaft due to its weight (Acting as uniformly distributed load): $F_{1V} = W_1 = 100\ N/m$ (30)

♦ Vertical force acting on the pulley-3 due to its weight at (Point C):

$$F_{2V} = W_2 = 5\ N$$ (31)

♦ Vertical force acting on the pulley-4 due to its weight at (Point F):

$$F_{3V} = W_3 = 5\ N$$ (32)

♦ Vertical force acting on the flywheel due to its weight at (Point E):

$$F_{4V} = W_4 = 80\ N$$ (33)

b) Horizontal loads subjecting on the cutter-head shaft:

♦ Horizontal tension force acting on the pulleys (3 and 4) equal:

$$T_H = T.cos16° = 2748 \times 0.961 = 2641.55 \cong \boldsymbol{2642\ N}$$ (34)

♦ Horizontal force acting on the chopping shaft due to the weight of *Arundo donax* feed rate (Acting as uniformly distributed load):

$$F_{1H} = W_4 = 100\ N/m$$ (35)

The vertical and horizontal loading diagram is shown in (Fig.6).

♦ The maximum bending moment for vertical and horizontal loadings could be obtained after calculating the reactions at the bearings A and B. From the loading diagram in Fig.(6), the calculation vertical reactions were:

$$\begin{array}{lcl} R_{AV} = 460\ N & \& & R_{BV} = 1194\ N \\ R_{AH} = 1610\ N & \& & R_{BH} = 3724\ N \end{array}$$ (36)

♦ The maximum bending moment for vertical loading being at (Point B) and equal:

$$M_{BV} = 31590\ N.cm$$ (37)
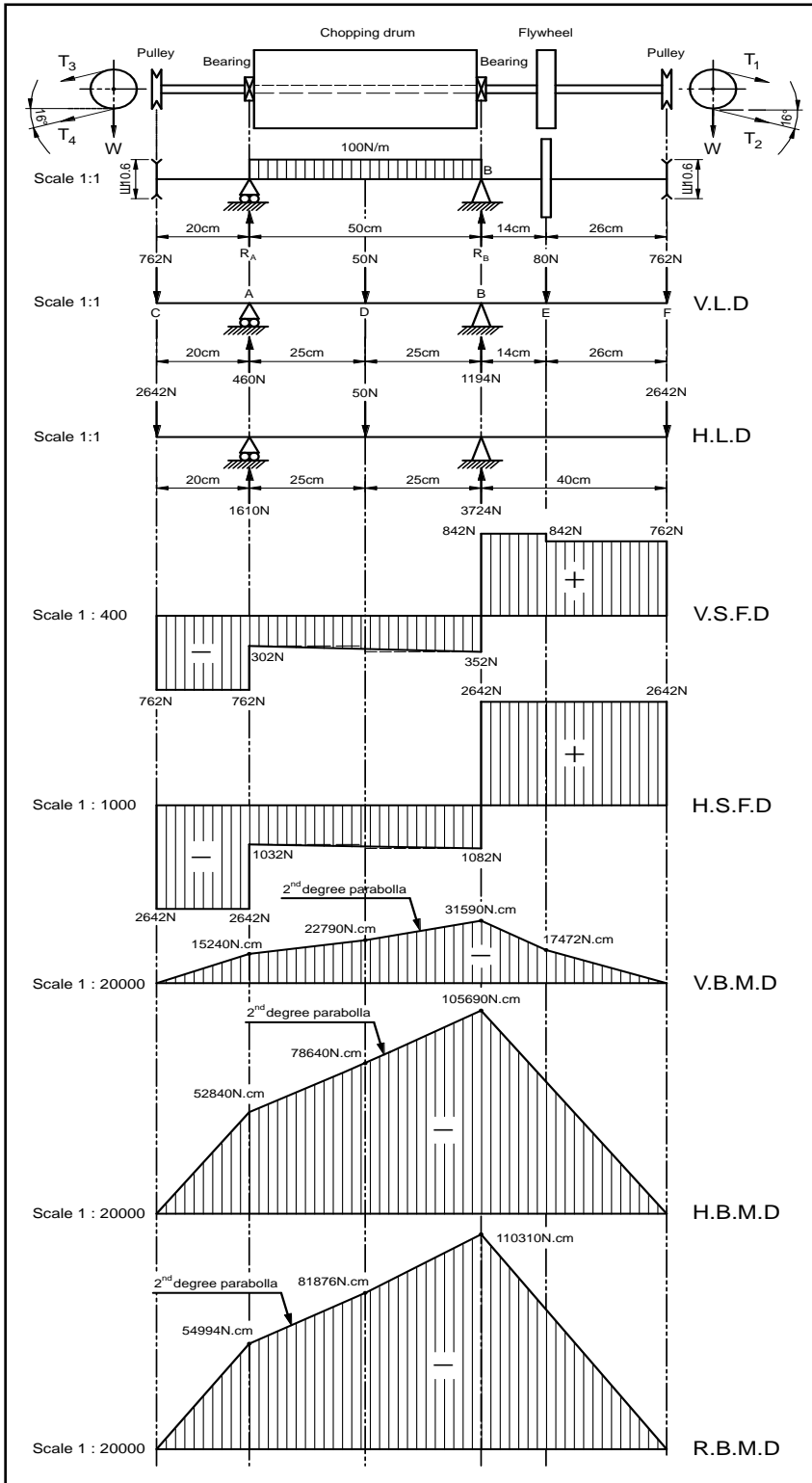
♦ The maximum bending moment for horizontal loading being at (Point B) and equal:

$$M_{BH} = 105690\ N.cm$$ (38)

♦ Resultant bending moment at (Point B) equal: (**Khurmi and Gupta 2005**).

$$M_R = \sqrt{(M_{BV})^2 + (M_{BH})^2} = \sqrt{(31590)^2 + (105690)^2} = 110310\ N.cm$$

$$\therefore M_R = 1103100\ N.mm$$ (39)

**Fig.6: Vertical and horizontal diagrams of loads, shearing
and moments for the designed cutter-head shaft.**

♦ Maximum shear theory at (Point B) equal:

$$Z_{max} = \frac{16}{\pi d^3} \sqrt{K_m^2 M^2 + K_t^2 T^2} \tag{40}$$

$$500 = \frac{16}{\pi . d^3} . \sqrt{2^2 (1103100)^2 + 1.5^2 (72 \times 10^3)^2} = \frac{16}{3.14 d^3} \times \frac{2208842}{9.81 \times 10}$$

$$\therefore d^3 = 114.81 \quad \therefore d = 4.86 \ cm = 48.59 \ mm$$

$$500 = \frac{16}{\pi . d^3} . \sqrt{2^2 (1103100)^2 + 1.5^2 (72 \times 10^3)^2} = \frac{16}{3.14 d^3} \times \frac{2208842}{9.81 \times 10}$$

$$\therefore d^3 = 114.81 \quad \therefore d = 4.86 \ cm = 48.59 \ mm$$

So, the diameter of the cutter-head shaft was designed to be **50 mm.**

3.5. Design the hammer mill shaft taking working power of 21 kW.

a) Vertical loads subjecting on the hammer mill shaft:

♦ Torque transmitted by the hammer mill shaft (T) can be calculated as follows: (**Khurmi and Gupta 2005**).

$$T = \frac{Po \times 60}{2 \pi N} = \frac{21 \times 10^3 \times 60}{2 \times 3.14 \times 2500} = 80.25 \ N.m \cong 80 \times 10^3 \ N.mm \tag{41}$$

Assume the torque at pulley-2 (Point C) = (80×10^3), therefore the tangential force will be as follows:

$$F_t \ at \ C = \frac{T}{r} = \frac{80 \times 10^3}{50} = 1600 \ N \tag{42}$$

$$F_1 = T_1 + T_2 \tag{43}$$

Where: F_1 = Tension force on pulley-2, N

T_1 = Tension on the tight side of belt, N

T_2 = Tension on the slack side of belt, N

$$T_m = (T_1 - T_2) \times r \quad \therefore \ (T_1 - T_2) = 1600 \tag{44}$$

$$80 \times 10^3 = (T_1 - T_2) \times 50 \quad \therefore \ (T_1 - T_2) = 1600$$

$$2.3 \ log \ (T_1 / T_2) = \mu \phi \ cos \alpha \tag{45}$$

$$\phi = (180 - 2\alpha) . \pi / 180 = (180 - 2 \times 35) 3.14 / 180 = 1.92 \ rad \tag{46}$$

$$\because 2.3 \log \frac{T_1}{T_2} = 0.3 \times 1.92 \times \cos 35 \quad \therefore \ 2.3 \log \frac{T_1}{T_2} = 0.472 \tag{47}$$

$$\therefore \frac{T_1}{T_2} = e^{0.472 \times 2.3} \quad \therefore \ \frac{T_1}{T_2} = 2.96 \tag{48}$$

$$\because \ (T_1 - T_2) = 1600 \quad \therefore \ 2.96 T_2 - T_2 = 1600 \quad \therefore \ 1.96 T_2 = 1600$$

$$So, T_2 = 816.32 \approx 816 \ N \quad and \quad T_1 = 2.96 \times 816 = 2415.36 \approx 2415 \ N$$

$$Then \quad F_1 = 3231 \ N \tag{49}$$

But the pulley transfer tension force at 16° from horizontal.

So, the vertical tension force acting on the pulleys-2 equal:

$$T_V = T.sin16° = 3231 \times 0.276 = 892\ N \tag{50}$$

♦ Vertical force acting on the hammer mill shaft due to its weight and chopped *Arundo donax* (Acting as uniformly distributed load):

$$F_{IV} = W_I = 524\ N/m \tag{51}$$

♦ Vertical force acting on the pulley-2 due to its weight at (Point C):

$$F_{2V} = 5\ N \tag{52}$$

b) Horizontal loads subjecting on the hammer mill shaft:

♦ Horizontal tension force acting on the pulleys-2 equal:

$$T_H = T.cos16° = 3231 \times 0.961 = 3105\ N \tag{53}$$

The vertical and horizontal loading diagram is shown in (Fig.7).

♦ The maximum bending moment for vertical and horizontal loadings could be obtained after calculating the reactions at the bearings A and B.

From the loading diagram in Fig.(7), the calculation vertical reactions were:

$$R_{AV} = 838\ N \qquad \& \qquad R_{BV} = 1997\ N$$
$$R_{AH} = 3353\ N \qquad \& \qquad R_{BH} = 6458\ N \tag{54}$$

♦ The maximum bending moment for vertical loading being at (Point B) and equal:

$$M_{BV} = 48450\ N.cm \tag{55}$$

♦ The maximum bending moment for horizontal loading being at (Point B) and equal:

$$M_{BH} = 167650\ N.cm \tag{56}$$

♦ The resultant bending moment at (Point B) equal:

$$M_R = \sqrt{(M_{BV})^2 + (M_{BH})^2} = \sqrt{(48450)^2 + (167650)^2} = 172590\ N.cm$$
$$\therefore M_R = 1725900\ N.mm \tag{57}$$
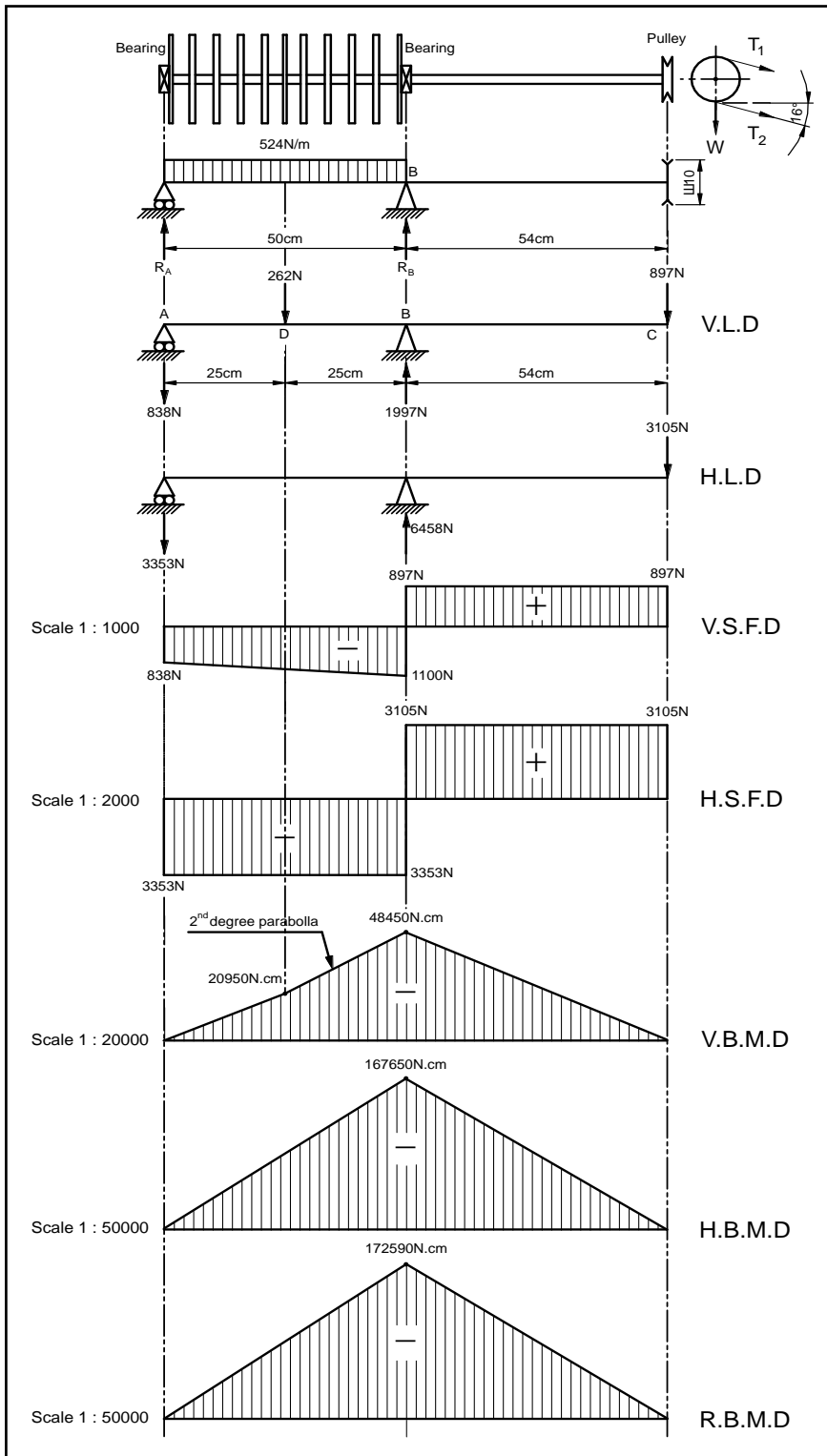
♦ Resultant bending moment at (Point B) equal: (**Khurmi and Gupta 2005**).

$$M_R = \sqrt{(M_{BV})^2 + (M_{BH})^2} = \sqrt{(48450)^2 + (167650)^2} = 172590\ N.cm$$
$$\therefore M_R = 1725900\ N.mm \tag{58}$$

♦ Maximum shear theory at (Point B) equal:

$$500 = \frac{16}{\pi.d^3} \cdot \sqrt{(1725900)^2 + (80 \times 10^3)^2} = \frac{16}{3.14 d^3} \times \frac{1727753}{9.81 \times 10} \tag{60}$$

$$\therefore d^3 = 179.48 \quad \therefore d = 5.64\ cm = 56.4\ mm$$

So, the diameter of the cutter-head shaft was designed to be **60 mm.**

Fig.7: Vertical and horizontal diagrams of loads, shearing and moments for the designed hammer mill shaft.

4. CONCLUSION

Design consideration of a machine for chopping and milling *Arundo donax* plants were carried out in Agricultural Engineering Department, Faculty of Agriculture, Zagazig University as they are necessary in manufacture process. The obtained design results reveal that the highest values of theoretical cut length, theoretical capacity were 29.14 mm and 756 kg/h, respectively, while the lowest values of chopping and milling unit's power were 9.54 and 20.17 kW, respectively. The values of cutter-head shaft diameter and hammer mill shaft diameter were 50 and 60 mm, respectively under the following conditions: number of knives on the cutter-head of 2 knives, cutter-head speed of about 1400 rpm and *Arundo donax* density in the feeding throat of about 6.18 kg/m^3 with adjusting feed rolls speed on 1.36 m/s.

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<u>الملخـــص العربـــي</u>

الاعتبـــارات التصميــمية لآلـــة مجمعـــة لفـــرم وطحـــن نبـــات الغـــاب (جزء-١)

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تعاني الدولة من نبات الغاب الذي ينتشر بكميات كبيرة على جوانب الترع والمصارف نتيجة ما قد يسببه من تلوث لمياه النيل وخطورة ذلك علي التلوث البيئي في مناطق عديدة بسبب إهدار كمية كبيرة من المياه وإعاقته لعمليات الري في المجاري المائية وعمليات التطهير بسبب غلق المجارى المائية في مناطق عديدة والذي يستغل مساحات واسعة دون أن يكون لها أي مردود اقتصادي.

لذا كان التفكير في كيفية الاستفادة من هذا النبات في العديد من الاستخدامات التصنيعية ذات المردود الاقتصادي. ومن هذا المنطلق كان الهدف الرئيسي لهذا البحث هو تصميم آلة مجمعة لفرم وطحن نبات الغاب كمرحلة أولي ومن ثمّ يمكن استخدام نواتج الفرم في العديد من الصناعات المختلفة كإنتاج الكمبوست العضوي ومحسنات التربة وإنتاج بيئات زراعية وتصنيع أخشاب مضغوطة يمكن الاستفادة منها في عدة مجالات. تمت عملية التصميم بقسم الهندسة الزراعية ـ كلية الزراعة ـ جامعة الزقازيق للوصول إلى الاعتبارات التصميمية التي تحقق الهدف الرئيسي.

وفيما يلي الأهداف الخاصة بهذا البحث:

♦ تصميم كل من وحدة الفرم ووحدة الطحن لنبات الغاب.

♦ الوصول لبعض القيم التشغيلية القصوى نظرياً لكل من: (عدد السكاكين على رأس القطع ـ سرعة درفيل الفرم ـ معدل التلقيم وكثافة السيقان عند فتحة التلقيم نتيجة المحتوى الرطوبي).

♦ تحديد مصدر القدرة الملائم لتشغيل الآلة المجمعة.

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ولتصميم الآلة المجمعة تم أخذ بعض المعاملات التالية في الاعتبار: أربع سرعات مختلفة لدرفيل الفرم (١٠٠٠ ، ١٢٠٠ ، ١٣٠٠ و ١٤٠٠ لفة/د) (١٨,٣٢ ، ٢١,٩٨ ، ٢٣,٨١ و ٢٥,٦٤ م/ث) وأربع قيم مختلفة لعدد السكاكين على رأس القطع (١ ، ٢ ، ٣ و ٤ سكينة) وأربع سرعات مختلفة لاسطوانات التلقيم (٠,٨٩ ، ١,٠٥ ، ١,٢٣ و ١,٣٦ م/س) مع أربع قيم مختلفة لكثافة النباتات عند فتحة التلقيم (٥,٠٠ ، ٥,٤٠ ، ٥,٧٤ و ٦,١٨ كجم/م٣). وللوصول إلى الاعتبارات التصميمية تم أخذ القياسات التالية: (طول القطع النظري، السعة النظرية، القدرة الكلية اللازمة لتشغيل الآلة المجمعة "القدرة اللازمة لتشغيل وحدة الفرم بالإضافة إلى تقدير القدرة المناسبة لتشغيل وحدة الطحن" وحساب قطر كل من عمودي درفيل الفرم وعمود مطارق الطحن).

وقد أظهرت النتائج التصميمية المتحصل عليها ما يلي:

- أعلى قيمة لطول القطع النظري والسعة النظرية كانت (٢٩,١٤ مم، ٧٥٦ كجم/س) على الترتيب عند سرعة للفرم ١٤٠٠ لفة/د وعدد ٢ سكينة على الرأس القاطع.

- قيمة القدرة المطلوبة لعملية الفرم وعملية الطحن كانت (٩,٥٤ كيلوات و ٢٠,١٧ كيلوات)، على الترتيب عند نفس ظروف التشغيل السابقة.

- قطر كل من عمود رأس القطع وعمود مطارق الطحن كان (٥٠ و ٦٠ مم).

ومن ثمّ توصى الدراسة عند إجراء عملية التصنيع للآلة المجمعة لفرم وطحن نبات الغاب باستخدام القيم التالية:

- ضبط سرعة اسطوانات التلقيم عند (١,٣٦ م/س).

- كثافة النباتات عند فتحة التلقيم (٦,١٨ كجم/م٣)

- سرعة درفيل الفرم (١٤٠٠ لفة/د) (٢٥,٦٤ م/ث) مع تركيب عدد ٢ سكينة على رأس القطع.