DESIGN AND CONSTRUCTION OF A GINGER SLICER

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CERTIFICATION

This is to certify that this project was carried out by NNAJIOFOR D. KENNETH in the Department of Agricultural Engineering, Federal University of Technology, Minna.

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Again, I say thanks to **ALMIGHTY GOD in JESUS NAME.**

ABSTRACT

The work reported here represents the design and construction of a ginger slicer.

Ginger slicer was designed to combat the problems associated with manual slicing of ginger. The design of ginger slicer involves a research on past literature works on slicing machine, selection of materials for the component parts of the machine, analysis of forces acting on the component parts and testing of the machine inorder to determine its efficiency. From the performance evaluation the effeciency was obtained as 58%.

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INTRODUCTION

In food processing, certain important factors such as time, mass, uniformity, and damage are considered before design and fabrication of processing machine.

Usually, the idea of slicing ginger is to ensure the effective drying of the ginger for further processing.

The major limitation of mechanically operated machines, is that they destroy the surface of some of the materials under action. Hence the mechanism and materials used in the design and fabrication of the ginger slicer would ensure minimal damage to the ginger. Time and Cost is saved, in order to ensure that the machine is economical. The safety of operation is as relevant as the production output of the machine.

1.1 ORIGIN OF GINGER CROP

Ginger can be classified as a perfume and flavoring crop. It belongs to the Zingibraceae family and is a close relation of turmeric. It is generally cultivated in Africa, but extensively cultivated in East and West Africa, which Uganda is the major producer in East Africa, Nigeria and Sierra Leone are the major producers in the West African sub-region.

In the past, ginger was believed to have medicinal properties and was therefore a highly esteemed specie in the Southern American and Red Indian regions. It command a high price in those societies and markets.

In Nigeria, cultivation of ginger is mainly in the Northern part of the country, mainly in Kafanchan, Zaria, Kaduna, Kano etc.

1.2 PROPERTIES OF GINGER

The properties of ginger can be classified into two major area i.e the physical and the chemical properties. Though, it is the physical properties that is vital for the design and construction of a ginger slicer, yet the chemical properties cannot be overlooked.

1.2.1 PHYSICAL PROPERTIES OF GINGER

There is no question to the relevance of the physical feature of the ginger to the design and construction of ginger slicer. Below are some of the physical features of ginger.

i. Ginger has a sandy-brown color.

ii. The ginger plant (herb) has a long, narrow leaf, which die down each year.

iii. The flowers are purple and have a complicated structure.

iv. Ginger has an irregular shape i.e sprouts towards different directions.

v. Ginger has some root hairs, which grow on it.

v. The texture of ginger is slightly coarse i.e it is scaly.

1.2.2 CHEMICAL PROPERTIES OF GINGER.

I. The root of ginger contains an oleo resin (oily resin)

II. Zingrone which is an aromatic ketone is present in the rhizome

III. An essential oil, which is not pungent, can also be extracted from the rhizome of ginger. It consists mainly of the terpene zinigebene and it is used as perfumery and flavoring essence.

I. The percentage composition by mass of the essential oil is 1-3%.

1.3 ECONOMIC IMPORTANCE OF GINGER.

Ginger may not be a popular crop, but non-the-less, it assumes a prominent position in food industries, beverages and brewery industries.

This can be attributed to its use in the manufacture of so many products.

The dried ginger rhizomes are used as a flavoring for cakes, biscuits, pickles, curry powder etc. The mentioned ingredients and confectionery are consumed in very large quantity mainly in homes, offices, schools etc. The economic such importance of industries to the Nigerian Economy cannot be underestimated.

Also, ginger is used as a flavoring in brewery and soft drink industries i.e in ginger beer, ginger ale, wine etc.

Some multinational companies like COCA-COLA consumes large quantity of ginger as raw materials.

It is obvious that the consumption of ginger related beverages, beers, food drink etc. run into very large sum of money.

Those companies that consume ginger as raw material for their products, contributes to a large extent to the revenue generated by the Local, States and Federal Governments.

Also in the improvement of the standard of living of the staff in such private enterprises.

Though, ginger has not been fully or effectively utilized in Nigeria, in some parts of the world i.e. South America, ginger is used in the production of medicine, hence it has a strong market force in that region.

1.4 DIFFICULTIES ASSOCIATED WITH LOCAL SLICING OF GINGER

By local slicing in this project it means cutting by hand using a knife. There are so many limitations in local slicing of ginger, some of which includes:

Waste of time, due to the slow pace of slicing by hand using a knife. And this is not encouraging at all in processing.

Injury is Common when using knife to slice. This is so, basically because there is little or no safety measures while cutting by hand using a knife i.e the fingers and the cutting edges of the knife are exposed and in steady contact.

The product cannot attain an accurate weight, this is because there is no known size control system employed by the slicer.

1.5 PROJECT JUSTIFICATION

The essential requirement for today's high-speed processing factories is the size reduction machine, which not only offers regular size of cut, but also operates at a large capacity.

Without modern food processing equipment it will be impossible to produce the uniform and regular size essential to the attainment of ideal processing condition.

The design of the ginger slicer is simple, hence the operator will not need acquisition of special skill to operate it.

The mechanism is not complicated, hence it is easy to service the machine i.e the operator could service the machine all by himself/herself.

1.6 OBJECTIVE OF THE PROJECT.

The Ginger Slicer medicine is designed to combat all the difficulties highlighted earlier in manual Cutting. Some of the objective of designing the Ginger Slicer are stated below:-

i. To attain a suitable surface area of product that would ensure effective drying of the ginger.

ii. To ensure the time of processing (cutting) is effectively utilized.

iii. To ensure simplicity of operation, while carrying out a large-scale job.

iv. To reduce the rate of accident during the slicing of ginger.

CHAPTER TWO

2.0 LITERATURE REVIEW

Determination of the physical properties of agricultural products is problematic because of their variability in sizes, irregularity in shape and porosity of their tissue.

The size and shape of agricultural products have been determined using roundness, sphericity, charted strollers, resemblance to already known geometric shape. Mohsenin (1970), Frechette and Zadradnik (1966) have calculated the surface area of biomaterial through the peeling process and by projecting the material on a planimetre.

2.1.0 GINGER PROCESSING

In general ginger is processed for the market in two major forms: the fresh green ginger used for the preparation of sweetened ginger in the orient or ginger beer in Nigeria and the dried ginger employed in the spice trade for the preparation of extracts/oleresin or essential oils.

The dried ginger is sold commercially either as peeled ginger or unpeeled ginger. In order to allow fast and thorough drying this could be halved (split) longitudinally and sold as dried split ginger commercially.

2.2.0 CONVENTIONAL PROCESSING METHOD.

The processing of ginger in Nigeria has not been standardized. Most farmers simply do what they feel is the right thing thereby making it difficult for Nigerian processed ginger to compete in the international market. The steps generally used in the processing of ginger in Nigeria include sorting, washing, peeling, slicing, drying and sometimes grinding. A schematic diagram represents the steps as follows:

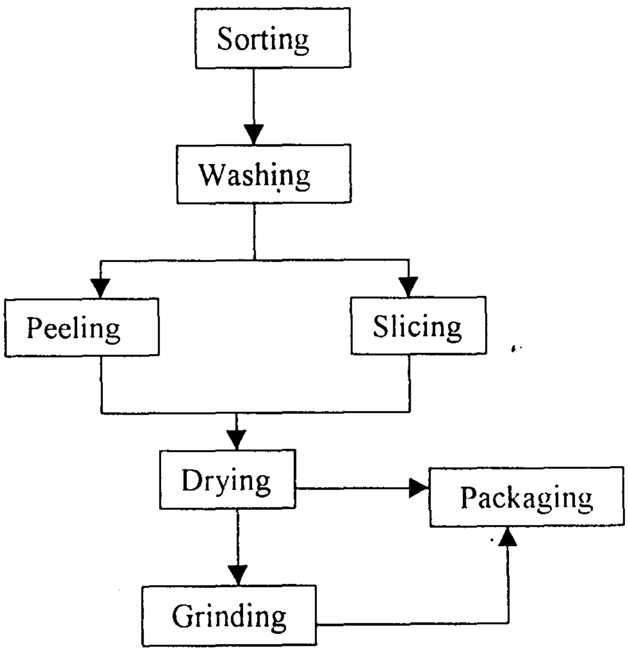


Fig. 2.1 flow chart of ginger processing.

2.2.1 PEELING

Peeling is a Manual process, which is mostly done by women. Most of the losses encountered during ginger processing is at this stage, where the peeler may not exercise the necessary care required. The flavour components of ginger are concentrated just below the peel , (Ashurt et al, 1973), hence losses are encountered in the peeling process. Hand peeling of ginger is not the best in terms of preserving the flavour and reduction of wastage. Losses of ginger material of 15-20% by untrained hands have been observed (Akomas, 1988, Unpublished data)

2.2.2 SLICING

Ginger is sometimes sold as split ginger. However, they must be dried and would be further processed by some farmer to ginger powder. To enhance drying, slices 2.5 to 4.0mm thickness are often made. Some flavour losses do occur at every stage of processing of ginger. Hence it is important that slicing be done in such a way that will minimise flavour losses. Very sharp slicer be used as to minimise tissue damage. (Akomas and Oti, 1988).

2.3 HISTORICAL BACKGROUND OF CUTTING PROCESS.

The initial development was from slicing or cutting by hand, using a knife, to a set of equally spaced, straight knives (Urschel, 1980)

This system required a great deal of pressure to push product through a large number of knives. Its effect is worsened by the product been squeezed through adjacent knives, the Development resulted in the production of grid of knives, achieved by placing two frames of straight knives at right angles to each other. Product slices were then pushed through this grid by a similar ram, the great force developed as each dice compressed into two different directions, resulting in serious crushing and cell damage. With fragile product, considerable breakage could occur. This process was also very slow, as the ram had to be returned to be loaded. An updated method of this cutting unit is still manufactured, it uses slicing knife directly offer the grid to control the length of slices being cut. The majority of the applications are in the meat industry but the original design constraint still offer the same problem both for application and product finish (Urschel, 1980)

2.4 DEVELOPMENT OF CUTTING MACHINE

The greatest break through in three dimensional cutting was made possible by the invention of a high speed slicing unit, by Urschel,.

This new technique subjected the product to centrifugal force, by rotating in a cylinder. A suitable open ended impeller driving the product produces the certrifugal force which hold the product securely in its place as it rotates. With controlled feeding of production, the slices will normally generate from the largest face, the prevention of random product movement permits slice thickness of great regularity to be secured.

Another useful feature of centrifugal placement of the largest product face, relative to the slicing knife, is of particular significance where the longest strips are required from product which is not normally round. Now it was possible to produce slices singly which moved away from the cutting area at high speed, with great capacity. This permitted the addition of linked cutting units, enabling production of three dimensional cuts in many size permutations.

Today, most foods are diced in one machine, where slices are first generated, then cut into strips and finally cut into individual dices. Immediate exchange and installation of cutting assemblies, without the need for further adjustment, will permit the dicer to be run just for slicing or strip cutting. Many cutting assembly permutation are readily installed or removed. (Macrae, 1993).

2.5 TRADITIONAL METHOD OF CUTTING.

Here, the cutting operation is usually of low scale, and the conventional way of using hand with knife is employed.

This method of cutting takes place in the family, peasant's farm etc. The material (product) being cut are usually food products for immediate or latter consumption.

2.5.1 MECHANICAL METHOD OF CUTTING

This involves the use of high speed processing machines. The size reduction machine not only offer regular sizes of cut, but also process at large capacity. Priority attention is paid to the cutting assemblies on the shaft.

Some examples are dicers, slicers, choppers etc.

2.6 SLICING MACHINE

Many dicing machines will provide slices, strip cuts and dices depending on the cutting assemblies used and the type of product being processed. Where unusually shaped products or difficult, possibly fibrous materials require to be sliced a specific choice of machine may be required.

For an elongated product, such as carrot or celery requiring slices to be made transversely and at high capacity, there is an obvious choice, the model OV SLICER. Regularly fed products entering the machine feed hopper, drops into two high speed conveyor belts, arranged to form a "V" trough Cross-Section. A third, non-powered but moving belt, above 'V' Conveyor, completes the product enclosure, ensuring positive feed to the slicing wheel. Knives that are under several thousand Nm tension serve as spokes and supports the rim of the slicing wheel. The knife mountings are angled, to make a uniform pitch from the hub to the rim. Both the pitch and number of knives determines the thickness of the slice and maintain the feeding speed of the product, whilst being sliced. Knives pass through the product at $29m5^1$. Using a similiar principle slices are available to produce bias cuts of either 30^0 to the long axis, or an alternate 45^0. Slice thickness ranging as lowas 0.8mm and a maximum of 54mm are available. Several options of plain knives are available, plus further alternatives of two patterns of crinkle knife. (Dickson, 1993)

Many meat and fibrous vegetable products cannot be effectively cut using a stationery knife edge. A good option in this case is a rotating knife, linked to the simplicity of the centrifugal slicing principle, model S-A SLICER.

One special slicing requirement is the production of very thin slices for the manufacture of potato crisps (known as potato chips in Nigeria). The specification is for accurate slice thickness to restrict under or over cooking, and slices generated with the minimum of starch cell damage, for this reason very thin sharp knives, similar to a safety rezor blade, which are inexpensive throw-away type are used. Correctly set machines with sharp knives, using average potatoes, will offer 80% of the slices varying less than 0-1mm. Machine capacity in this case is achieved by having eight cutting stations equally spaced around the periphery, model CCSLICER, (Dickson, 1993)

2.7 DICING MACHINE

The model GK DICER will cut uniform slices as strips, three dimensional cuts, with Crinkle and two plain cuts, or all plain cuts. They will handle a range of soft ripe fruits and brittle vegetables, producing well-defined cuts. Slices are as a result of centrifugal principle, but the Crossed spindle progresses the slices forward by knives edge at a constant 90^0 . This progressive cutting action of clean, square cut to be produced with the least product loading.

Model RA DICER is three (3) dimensional dicing machine. One limit in the design is that it is impractical to use a circular knife spacing that is less than 3mm. The different cutting action developed by a cross cut knife assembly will permit a small dimension of 1.6mm to be achieved. One special feature of this type of dicer is the ability to by-pass the slicing knife station. This is particularly useful in the reduction of dried fruit, or any suitable material where thickness is already predetermined, where small products are fed into the machine and these require the minimum of cutting. It is possible to use the impeller as a feeding unit, provided that the slicing knife has been removed (Dickson, 1993).

CHAPTER THREE

3.0 DESIGN CALCULATION

3.1 DETERMINATION OF PHYSICAL PROPERTIES OF GINGER

Nwandikom and Njoku (1988) from the experiment they carried out at the National Root Crops Research Institute, Umudike, they established formula for determining the mass, surface area and volume of ginger crop. I.e

$V = 0.503ab\,L_o + 0.55\ (cm^3)$ 3.1

Where

V = Volume of the ginger

L_o = Vertical Length of the ginger (cm)

a = Major diametre (cm)

b = Minor diametre (cm)

$M = 0.500\,ab\,L_o + 0.54$ 3.2

Where M = Mass of ginger (g)

And

$S = 0.91(a+b)(L_1+L_2) - 3.12\ (cm^2)$......................................3.3

Where

S = Surface area of the ginger finger (cm^2)

L_1, L_2 = Slanting length of the finger (cm)

For example A, the following values are obtained

$Lo = 70mm$

$L_1 = 85mm$

$L_2 = 45mm$

$da_1 = 22.0mm$

$da_2 = 32.0mm$

$da_3 = 40.0mm$

$da_4 = 43.0mm$

$db_1 = 10.7mm$

$db_2 = 14.1mm$

$db_3 = 15.4mm$

$db_4 = 14.1mm$

Hence

$V = 0.503abL_o + 0.55cm^3$

Where

A= average of da_1

Where

A=average of da_1

B=average of db_1

$V = 0.503x34.25x13.58x70x0.55 = 1641.5m^3$

$S = 0.91\ (34.25+13.58)(58+45) - 3.12 = 4479.99(4480mm^2$

$M = 0.500x3.425x1.358x7.0x0.54 = 16.2g$

For sample B,

$L_o = 65mm$

$L_1 = 50mm$

$L_2 = 43mm$

$da_1 = 45.0mm$

$L_2 = 43mm$

$d_{a1} = 45.0mm$

$da_2 = 45.0mm$

$da_4 = 25.0mm$

$db_1 = 13.5mm$

$db_2 = 15.3mm$

$db_3 = 15.8mm$

$db_4 = 14.3mm$

Hence

$V = 0.503x39.69x14.73x65x0.55=19115.2mm^3$

$S = 0.91\ (39.69+14.73)(65+50) - 3.12=5691.93mm^2$

$M = 0.500x3.969x1.473x6.5+0.54=19.54g$

For sample

$L_o = 74mm$

$L_1 = 48mm$

$L_2 = 40mm$

$d_{a1} = 20.0mm$

$da_2 = 22.0mm$

$da_3 = 27.0mm$

$da_4 = 20.0mm$

$db_2 = 15.0mm$

$db_3 = 15.0mm$

$db_4 = 14.7mm$

Hence,

$V=0.503x20.7x14.45x74x0.55=11134.21mm^3$

$S=0.91(20.7x14.45)(48x40)-3.12=2811.69mm^2$

$M=0.500x(20.7x14.45)10x74x10-1x0.54=11.6g$

For sample D,

$L_o = 68mm$

$L_1 = 54mm$

$L_2 = 41mm$

$d_{a1} = 25.0mm$

$da_2 = 21.5mm$

$da_4 = 22.0mm$

$da_4 = 12.1mm$

$db_1 = 13.4mm$

$db_2 = 17.4mm$

$db_3 = 18.2mm$

$db_4 = 13.0mm$

Hence,

$V=0.503x20.15x15.5x68x0.55=10737.8mm^3$

$S=0.91(20.15x15.5)(54x41)-3.12=3078.82mm^2$

$M=0.500x2.015x1.55x6.8x0.55=11.17g$

Sample E,

$L_o = 48mm$

$L_1 = 36mm$

$L_2 = 42mm$

$d_{a1} = 30mm$

$da_2 = 25mm$

$da_3 = 15.7mm$

$db_1 = 11.7mm$

$db_2 = 14.2mm$

$db_3 = 12.0mm$

Hence,

$V=0.503x23.6x12.63x46x0.55=7197.1mm^3$

$S=0.91(23.6x12.63)(36x42)-3.12=2568.49mm^2$

$M=0.500x2.36x1.263x4.8)x0.54=7.6g$

Sample F,

$L_o = 83mm$

$L_1 = 64mm$

$L_2 = 55mm$

$d_{a1} = 25.5mm$

$da_2 = 25.0mm$

$da_3 = 35.0mm$

$da_4 = 60.0mm$

$da_5 = 28.8mm$

$db_1 = 14.0mm$

$db_2 = 17.4mm$

db_3= 18.7mm

db_4= 18.0mm

db_5= 16.0mm

Hence,

V=0.503x34.86x15.79x83x0.55=22980.84mm^3

S=0.91(34.86x15.79)(64x55)-3.12=5481.77mm^2

M=0.50x3.486x1.579x8.3)x0.54=25.66g

Sample G,

L_o = 65mm

L_1 = 50mm

L_2 = 43mm

d_{a1} = 33mm

da_2= 49.0mm

da_3= 53.0mm

da_4= 40.5mm

db_1= 13.5mm

db_2= 15.3mm

db_3= 15.8mm

db_4= 14.3mm

Hence,

V=0.503x43.93x14.73x65x0.55=21157.12mm^3

S=0.91(43.93x14.93)(50x43)-3.12=4961.28mm^2

M=0.500x4.393x1.473x6.5)x0.54=21g

3.2 Determination of the thickness of cut/slash

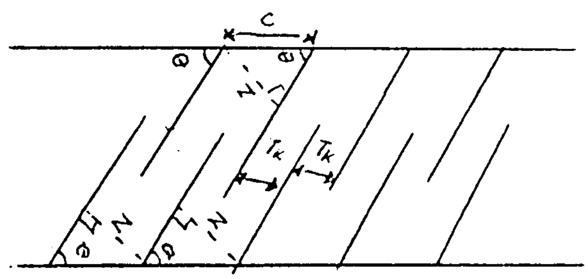


Fig 3.1 Cutting Knives.

The cutting knives (disc) can be inclined at the following angles 300,600 and 900 to the long axis and the clearance adjacent blades is

Sin $\theta = Z/C$

But Z = 2 TK

Where

Z = depth of vertical cut (mm)

C = Clearance between adjacent blades (mm)

Tk = depth of cut (mm)

For 30^0 angle of inclination of blade.

Tk = Sin 30xC/2 = 0.5x20/2 = 5mm

But since blade thickness is 2mm

Tk = 5-2=3mm

For 45^0 angle of inclination of blade

Tk=sin 45x20/2 = 0.0771x20/2 = 7.07mm

Blade thickness = 2mm

Hence Tk = 0.07 - 2 = 5.07 = 5.1

For 60^0 angle of inclination of blade

Tk = sin60x20/2 = 0.8660x20/2 8.66mm

Blade thickness = 2mm

Tk =8.66-2 = 6.66mm

For 90^0 angle of inclination of blade

Tk=Sin90 x 20/2 = 20/2 = 10mm

Blade thickness = 2mm

TK = 10-2= 8mm

3.3 Determination of power requirement of the ginger slicer

Force required is given by Jekendra and Singh (1991)

$F = d.L.\delta$ 3.4

Where

D = thickness of blade (Disc), m

L = diametere of blade (Disc), m

δ = yield stress of the material, N/m^2

but

d = 2mm, = 0.002m

L = 90mm = 0.09m

$\delta = 1853.5N/m^2$

F = 0.002x0.09x1853.5

= 0.334N

power = Force x Velocity

$F \times V$

$2\pi NR/60$

where

N = speed of impeller required (rpm)

R = Radius of blade,m

$V = 2 \times 3.142 \times 350 \times 0.045/60$

$= 1.65$m/s

power = 0.334×1.65

$= 0.55$W

This is the power required to slice a ginger

According to Salvenchy (1992), the power an average man can genrate is 83W

Hence, it can also be operated manually

But to increase rate of production, A 2hp, 1450rpm electric motor is selected.

There will be speed reduction in the operation

Speed ratio = N_1/N_2

Where

N_1= speed of motor, rpm

N_2 = speed of impeller, rpm

Speed ratio = 1450/350 = 29/7

i.e 29:7

4:1

3.4 SELECTION OF THE DRIVER AND DRIVEN PULLEY

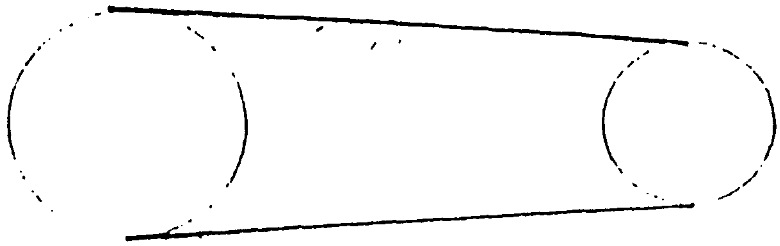


Fig 3.2 Drive and Driven Pulley

Where

α_1 = angle of contact small pulley (deg)

α_2 = angle of contact large pulley (deg)

C = centre of distance (m)

r = Radius of small pulley (m)

r = Radius of large pulley (m)

The relationship between motor pulley speed, diametre & the shaft pulley speed, shaft diametre is given below:

$N_1 D_1 = N_2 D_2$ 3.6

Where

N_1 = Motor speed = 1450rpm

D_1 = Motor pulley diametre = 80mm

N_2 = Shaft pulley speed required = 350rpm

D_2 = Shaft pulley diametre

D_2 = ? $N_1 D_1 / N_2$ = 1450x80/350 = 331.4 = 331mm

3.5.1 Determination of the centre distance

Since an adjustment of the centre is provided, the minimum and maximum centre distance cannot be restricted by the Machine / component in anyway.

Though, a reasonable value of centre distance is given by the larger of $C=3R_1 + R_2$ or $C = 2R$

So, the chosen equation is below, because it gives a larger value

$C = 3R_1 + R_2$ 3.7

Where

R_1 = radius of small pulley (mm) = 40mm

R_2 = radius of larger pulley ((mm) = 165.5mm

$C = 3(40) + 165.5 = 285.5$mm

Calculation of C min. & C max.

i. Minimum centre distance (min)

R = Norminal belt thickness (mm) 11mm

$C_{min} = 0.55\,(D_2 + D_1) + t$ 3.8

$= 0.55\,(331+80) + 11 + 237.05 = 237$mm

$C_{min} = 237$mm

ii. Maximum centre distance (Cmax)

$C_{max} = 2(D_2 + D_1) = 822$mm

$C_{max} = 822$mm

The adjustment would be provided in this form

$C_{min} < c < C_{max}$.

3.5.1 Calculation of the total pitch lengh of the belt

$L = 2C+\pi(R_1 + R_2)+(R_2-R_1)^2/C$ 3.9

Where

C = Centre distance (m)

R_1= Radius of small pulley (m)

R_2= Radius of large pulley (m)

L = Pitch length of the belt (m)

$= 2(285.5) + 3.142\ (40 + 165.5) + (165.5 - 40)^2/285$

L = 1271.9mm

3.5.2 Calculation of the distance from base to centre of gravity

$X = t(w_2+2w_1)/3(w_2+w_1)$ 3.10

Where

t = Norminal thickness of the V – belt = 11mm

W_2 = bottom width of the V – belt = 8mm

W_1 = Top width of the V – belt = 17mm

X = 11(8+7) /3(8+7) = 6.16mm

3.6 DESIGN OF BELT

V - Belt is selectd for the following reasons

i. Reduces Vibration

ii. Reduces shock transmission

iii. Relatively quiet in operation

iv. Offers a maximum of versatility as power transmission element.

v. Can operate with low tension to transmit high torque

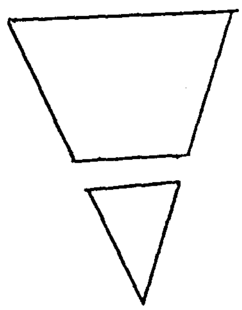


Fig 3.3 V – belt drive.

For V – belt selection, the following dimension and parametres are obtained.

Norminal Top width, w = 17mm

Norminal thickness, t = 11mm

Shear grove angle $\theta = 40^0$

Density of rubber belt = 1250 kg/m^2 (0.045 ib/m) Daron (1975)

Coefficient of friction between pulley and belt = 0.3

Maximum allowable stress for rubber belt = 250psi

Required speed at athe impeller shaft = 350rpm

Assumed diametre of small pulley = 80mm

Speed of motor = 1450 rpm

Width per metre, kgf = 0.49 hgt

Small diametre correction factor for arc of cantact variation Fc = 1.1

The belt speed is given as

$S = \pi\, dN$ 3.11

Where

N = speed of the motor 1450rpm

d = dp = pitch diametre = 80mm

S = 3.142x0.08x1450/60 = 6.07m/s = 6.1m/s.

Calculation of the equivalent pitch diametre

$dc = dp \times Fb$ 3.12

$dc = 0.08 \times 1.13 = 0.0904m$

$dc = 90.4mm$

The value of dc is satisfactory since it is greater than 80mm

To determine the bottom width of the belt

$W_2 = tw_1/2T$ 3.13

Where

t = Norminal thickness f the belt (mm)

T = Thickness of the belt (mm)

W_1 = Top width of the belt (mm)

$T = 0.5w_1/2 \tan \beta$

$\beta = (180 - 0)/2 = (180 - 40)/2 = 70^0$

$T = 0.5x17/2 \tan 70 - 11.68mm$

$W_2 = 11x17^0/2x11.68 = 8mm$

To determine the weight of the belt

$M = \rho v$ 3.14

Where

ρ = density of the belt (kg/m^2) = $1250kg/m^2$

V = Volume of the belt (m^2)

A = Belt cross section area (m^2)

$A = (w_1 + w_2)\ t/2$ 3.15

$= (17+8)_2 = 137.5mm^2$

$A = 137.5mm^2$

The weight of the belt is given by

$W = mg$

$= 1250 \times 137.5 \times 10^{-10} \times 9.81 = 1.69N/m$

Determination of angle of contact

$\sin \beta = R_2 - R_1 / C$ 3.16

Where

R_2 = pitch radius of the shaft pulley (mm)

R_1 = pitch radius of the motor pulley (mm)

C = centre distance (mm)

β = angle between the arc of contact and pulley centre.

$\beta = \sin^{-1} (165.5-40) / 285.5 = 26.07 = 26.1$

For small pulley, the angle of contact

$\alpha_1 = 180 - 2\beta$ 3.17

$= 180-2(26.1) = 127.8 = 128^0$

For large pulley, the angle of contact

$\alpha_2 = 180 + 2\beta$ 3.18

$= 180 + 2(26.1)\ 232^0.2$

3.6.1 To determine the load carrying capacity of pulley for V – belt in the groove,

It uses the criteria below

$e^{f\alpha/\sin\frac{1}{2}\theta}$ (Schaum's Series)

Where

X = angle of contact of the small pulley (deg)

θ = groove angle of the puley (deg)

F = 0.3

$\theta = 40^0$

$e^{f\alpha/\sin\frac{1}{2}\theta} = e^{0.3 \times 127 \times \pi/180}/\sin\frac{1}{2}40^0$

= 6.99 = 70

Note

The small pulley and large pulley are grooved

Determination of the belt tension at the tight side

If the belt is transmitting at its maximum capacity, the tension T^1 on tight side is given as

T_1 = Maximum belt stress x Area of the belt

= 250 x 211 x 4.4

= 232.1N

Determination of the belt tension at the slack side

$T_1 - Mv^2/g/T_2 - Mv^2/g = e^{f\alpha/\sin\frac{1}{2}\theta}$

Where

M = weight of 1m belt N

V = velocity of belt in m/sec

g = acceleration due to gravity

But the centrifugal force is given as

$Tc = Mv^2/g$

But $M = 12(0.211x0.045) = 0.11395N$

$V = \pi dn/60 = 3.14x0.08x1450/60 = 6.07m/s$

$Tc=0.11325x1.072/9.81 = 6.23N$

From equation 3.18

$T_2 = t_1 - Mv^2g = e^{f\alpha \sin^{1}{}_2 0} Mv^2 /g/ e^{f\alpha \sin^{1}{}_2 0}$

$= 232.1 - 6.23 = 7 x6.23/7 = 38.5N$

$T_2 = 38.5N$

Determination of the maximum allowable stress on the belt

$S_2 = S_1 - m^1V^2 + e^{f\alpha /\sin^{1}{}_2 0} m_1V^2/g/ e^{f\alpha \sin^{1}{}_2 0}$ 3.19

Where

M^1 = weight of 1m belt 1 in^2 in cross section

S_1 = maximum allowable stress N/m^2

S_2 = stress in slack side belt N/m^2

$V \quad = 6.\text{im/s}$

$g \quad = 9.8\text{m/s}$

Also

$T_1 - T_2/S_1 - S_2 = A$ 3.20

From equation 28

$S_2 = S_1A - T_1 - T_2/A$ 3.21

Equate 27 & 29

$S_1 - M^1V2 + e^{f\alpha/\sin ' : 0} M_1V2/S/e^{f\alpha \sin ' : 0}\ S_1A - T_1 + T_2/A$

$S_1 - 6.7+7x6.7/7 = S_10.211 - 52.75 \times 9.63 / 0.211$

$(S_1 - 40.2)0.211 = 7(S_10.211 - 43.12)$

$0.211S_1 - 8.44 = 1.48S_1 - 301.84$

$301.84 - 8.44 = 1.48S_1 - 0.211S_1$

$293.4 = 1.269S_1$

$S_1 = 293x4.4x10^4/1.2x6.45$

$= 1.68\text{MN/m}^2$

Determination of the stress at the slack side of the belt

From equation 27

$S_2 = S_1 - M^1V^2g = e^{f\alpha/\sin ' : 0}\ M^1V^2/s/e^{f\alpha \sin ' : 0}$

$= 250\text{-}6.7x6.7/7$

$S_2 = 41.5x44/6.45x10^{-4} = 0.28x10^6\text{N/m}^2$

$S_2 = 0.28\text{MN/m}^2$

Determination of the torque at the section of the belt can be represented as the twisting moment of ulley on shaft.

$M_E = (T_1 - T_2) R_1$ 3.22

$= (232.1 - 36.5) \times 165.5/1000$

$= 32.04 = 32Nm$

Calculation of horse power per belt in Kilowatt (kw)

$P = (T_1 - T_2) V/1000$ 3.23

Where V = belt speed, m/s

T_1 = tension on tight side of belt N1

T_2 = tension on slack side of belt N1

$P = (232.1 - 38.5) \times 6.1/1000 = 1.18$hp/belt

Determination of the number of belt required

Number of belt = Motor power in Kw/Power transmitted per belt

$= 1.5/1.18 = 1.25$

$T2_2$ (2) belts will be used

The Resultant Force at the pulley causing bearing reaction is given as

$T_1 + T_2 = 232.1 + 38.5 = 271.5N$

3.7.0 GEAR DESIGN

Spurs gears, were used in the Design

Spur gear provide a positive means of transmitting power between parallel shafts

At a constant angular velocity ratio.

Circular pitch Pc is the distance from a point on one tooth to a corresponding

Point on an adjascent tooth, measured on the pitch circle.

$Pc = \pi D/N$ 3.24

Where

D = Pitch diametre of the gear, mm

N = Number of teeth on the gear

Module, M is the pitch diametre divided by the Number of Teeth on the gear

$M = D/N$ 3.25

$M = Pc/\pi$ 3.26

Line of action is a line normal to a pair of mating tooth profile at their point of

contact

Pressure angle θ is the point of tangency of the pitch circles.

Angular velocity ratio (or transmission ratio) is the ratio of the angular velocity of

the pinion to the angular velocity of the mating gear.

Angular ratio $W = Ng/Np = Dg/Dp$ 3.27

3.7.1 GEAR CALCULATIONS.

From equation 3.24

i. $P_c = \pi\ d/N$ $\quad$ D = 70mm

N = 28

P_c = 3.142 x70/23 = 9.526 = 9.5mm

ii. From equation 3.25

M = D/N = 70/23 = 3

ii. From equation 3.27

$W = N_g/N_p$ = 23/23

i.e 1:1 gear, pinion ration

iii. Contact angle = 20^0 full depth involute

iv. Addendum = M -= 3mm

v. Minimum dedendum = 1.157mm = 1.157x3 = 3.471 = 3.5mm

vi. Whole depth = 2.157m –2.157x3 = 6.471 = 6.5mm

vii. Clearance = 0.157m = 0.157x3 = 0.471 = 0.5mm

viii. Outside diametre = D+2 adendum = 70+2x3 = 76mm

ix. Radius of base circle = pitch radius x cos θ = 35 cos 20 = 342.8mm

x. Diametre of base = base circle radius x2 = 32.8x2 = 65.6mm

xi. Root diametre = outside diametre – 2x whole depth = 76 – 2 x 6.5 = 63mm

Xiii Interference is avoided if the addendum radius of gear is

$\leq$ (base circle radius)^2 + (Centre distance)^2 (Sin θ)^2

$\leq$ (32.8)2 + (1/2(70+70)^2 (sin 20)^2

$\leq$ 794.68 = 4900 x 0.11698

$\leq$ 34.87mm

Since the addendum radius = 76/2 = 33mm, then the Design is without interference, hence it is safe.

3.7.2 FORCE ACTING ON THE GEAR

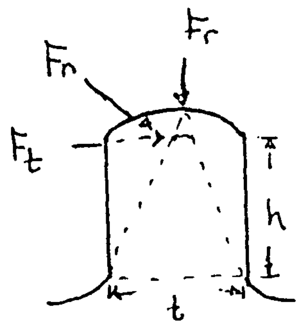


Fig 3.4 **Forces acting on a gear tooth**

Where

F_r = radial force (N)

Fn = Total force (load) neglecting friction (N)

Ft = Tangential force, N

Ft = Mt/r 3.20

Where

Mt = power transmitted x 60/gear speed x 2 π = 90000/2199.4 = 40.9Nm

Ft = 40.9/0.03 = 1168N

$F_r = F_t \tan\theta$ 3.29

$= 1168 \tan 20 = 425.2N$

$Fn = (F_t)^2 + (F_r)^{2^{\wedge}1/2} = (1364)^2)^{\wedge 1/2} = 1860496 = 246512.3 = 145.6N$

3.7.3 Determination of the strength of Gear teeth

Using Lewis equation

The induced bending stress at the base of tooth is given as

$S_b = F/bp_cY$ 3.30

Where

F = transmitted force, N

S_b= induced bending stress N/m^2

B = Face width, m

Y = Form factor = 0.0106

P_c= Pitch circle.

But $b=KPc$, where K = Constant: 4 upper limit

$B = 4x9.5 = 38.1 = 0.0381mm$

$S_b = 1168/0.0386x0.0095x0.106 = 30.44x106N/m^2$

3.7.4 To determine the allowable Tooth Stress

It virtually depends upon the selected material and pitch line velocity.

For spurs gear, Barth's equation can be used.

$S_a = So\ (3/3+v)$ for less than 10m/s 3.31

Where

S_a = allowable tooth stress, N/m^2

S_o = the endurance strength of material, dependent on average stress concentration values of gear material (N/m^2)

V = Pitch line velocity (m/s)

$V = (S_g x 2 \pi)/60(1/2(N_g xM/1000)$ 3.32

Where

S_g = speed of gear (rpm)

N_g = Number of teeth

M = Module.

$V = (350x2\pi)/60(1/2(23x3/1000)$

$= 36.7x0.03 = 1.2m/s$

S_o for the steel pinion and gear is 00MN/M^2

$S_a = 200x10^6\ (3/3+1.2) = 146x10^6N$

$= 146MN/m^2$

since the induced bending stress is < allowable stress

i.e $S_b < S_a$, therefore the design is satisfactory.

3.7.5 Determination of Dynamic Tooth Load

The dynamic toth is as a result of the inaccuracies of the tooth profiles, spacing, misalignment in mounting and tooth deflection under load. Hence there will be a change of velocity which produces dynamic force other than the transmitted force. By buckingham equation.

$Fd = 21V(bc+F)/21V+\sqrt{(bc+F)}+F$ 3.33

Where Fd = dynamic load, N

V = pitch line velocity. m/s

B = face width, m

F = transmitted force,N

C = a constant, in N/m, which depends on the tooth form, material and degree of accuracy with which the tooth is cut. It is known as deformation factor.

For 0.13mm permissible error C = 1482KN from tableII

$`Fd=21x1.2(0.0376x1432x10^3+1168/21x1.2+$

$\sqrt{(0.0376x142x10^3+1168)}+1168$

= 1417730.84/287.6 x 1168

Fd = 6582N

=6.582KN.

Calculation of the allowable endurance load F_o

$F_o = S_o$ by P_c..3.34

Where

S_o = as defined earlier

$B,y\&p_c$ = as defined earlier

$F_o = 200x10^6x0.0376x102x0.0094$

F_o = 7673N = 7.67KN

$Fd<F_o$, hence the design is satisfactory

3.7.5 Determination of wear/Allowable tooth load

To ensure the durability of the gear pair, the tooth profile must not have excessive contact stress. The expression by Buckinham can be used.

$F_w = D_p bkQ$ 3.35

Where

F_w = allowable/wear load, N

D_p = pitch diametre of gear, m

K = stress factor for falique , N/m^2

$Q = 2Ng/(N_{p+}+N_g)$ 3.36

Where

N_g = Number of gear teeth = 23

N_p = Number of pinion teeth = 23

$K_,$ = 2553KN/m^2 at Borinell hardness Number (BHN) 400 (steel & gear pinoin.)

20^0 full depth involute.

Q=2(23)/23+23 = 1.0

D_p = 70mm = 0.07m

B = 38.1mm = 0.0381m

F_w = 0.07x0.0381x2553000x1.0 from equation 3.37

F_w = 6808N = 6.81KN

The allowable load F_w dynamic load fd

Hence the design is satisfactory.

i.e 6.81KN > 6.58KN.

3.8.0 DESIGN OF SHAFT

Assumed mass of ginger fed into the shaft =3kg

Density of steel (stainless) = 7480kg/m^3

Assumed diametre of shaft = 20mm

Weight of disc = $\rho v_g n$ 3.38

Where

P = density of disc

$V = 4/3\ \pi\ (r_1 - r_2)^3$

R_1 = radius of outside sphere, m

R_2 = Radius of inside sphere,m

N = Number of disc

Weight of disc = $7840 \times 4/3.142(0.045-0.016)^3 \times 9.81 \times 5$

= 62.8N

Uniformly distributed load on shaft = weight of disc + weight of ginger/Active length of shaft.

= 62.8+3x9.81/0.18

= 512N/m

FREE BODY DIAGRAM OF VERTICAL FORCE

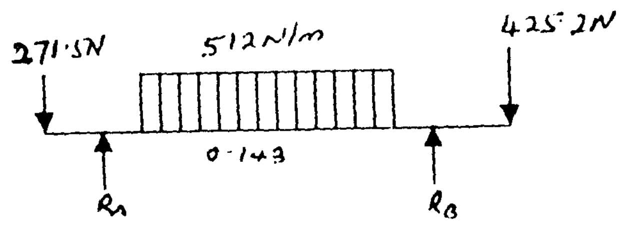


Fig 3.5. Free body diagram of Vertical forces

$\Sigma fy = - R_B - R_B + 271.5 + (512 \times 0.14) + 425.2 = 0$

$R_B - R_B + 768.4 = 0$

$R_B = 768.4 - R_B$ 3.39

Taking moment about R_B

$\Sigma my = 0.18 + 512 \times 0.141 \times 0.16/2 + 271.5 \times 0.27 - 425.2 \times 0.06 = 0$

$= 0.18\ R_B + 5.7 + 73.3 - 25.5 = 0$

$R_B = 53.5/0.18 = 297N$

From equation 3.36

$R_B = 768.4 - 297 = 471.4N$

SHEAR FORCE CALCULATION

$0<x<0.09+Q+271.5$, $Q=-271.5N$

$0.09<x<0.14+Q+271.5-297$, $Q=+25.5$

$0.14<x<0.33+Q+271.5-297+512 \times 0.14-471.4+425.5$

$Q=-2N$

BENDING MOMENT CALCULATION

0<x<0.09   +271.5x+M=0

M=-271.5(0.09)=-24.4NM

0.09<x<0.11   +271.5x-297(x-0.09)+M=0

+271.5(0.11)-297(0.11-0.09)+M=0

M=-23.96NM

0.11<x<0.25   +271.5x-297(x-0.09)+512(x-0.11)(x-0.09/2)+M=0

+271.5(0.25)-297(0.25-0.09)+512(0.25-0.11)(0.25-0.11)(0.25-09/2)+M=0

69.7-47.5+5.7+M=0

M=-26NM

0.25<x<0.33   +271.5x-297(x-0.09)+512(x-0.14)(x-0.09/2)-471(x-0.25)

+425.5(x-0.253)+M=0

+271.5(0.33)-297(0.33-0.09)+512(0.25-0.19)(0.25-0.09/2-471(0.33-

0.25)+425.5(0.33-0.25)=M=0

M=-23.3N

FREE BODY DIAGRAM OF HORIZONTAL FORCES

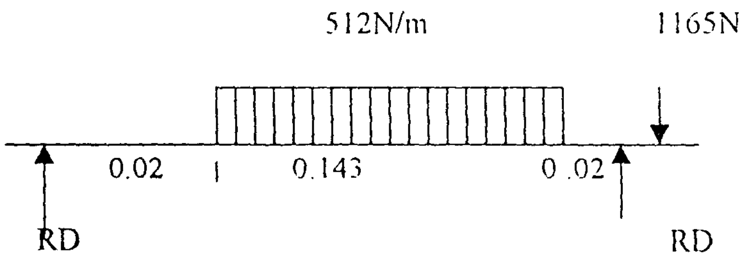


Fig 3.6 Free body diagram of Horizontal Forces.

$\sum fy = R_B - R_B + 512(o.14)+1168 + 0$

- $R_B - R_B + 1241 + 0$

- $R_B = 1241 - R_B$

Taking moment about R_B

$\sum = 0.18+512(0.14)(0.16/2)-1168(0.06)=0$

- $0.18\ R_B + 5.97-70.1 = 0$

$R_B = 64.11/018 = 350N$

$R_B = 1241 - 3500=891N$

SHEAR FORCE AND BENDING MOMENT CALCULATION

Shear force

$0<x< 0.02$ $\quad +Q-(-350) = 0 \quad Q = 350N$

$0.02<x< 0.163 + Q-(350)+512(x-0.02) = 0$

$Q+350+73.2 = o$

$Q=-423.2N$

$0.163<x<0.245$ $+Q-(-350)+512(x0.1)-891+1168=0$

$Q+350+512(0.143)-891+1168+0$

$Q=-700N$

Bending Moment

$0<x<0.02$ $-(-350)x+M=0$

$M+350(x)$

$M+350(0.02)$

$M=-7Nm$

$0.02<x<0.163$ $-(-350)+512(x-0.02)(x-0.02)/2)+M=0$

$350+512(0.163-0.02)(0.163-0.02)/2+M=0$

$M=-67.3NM$

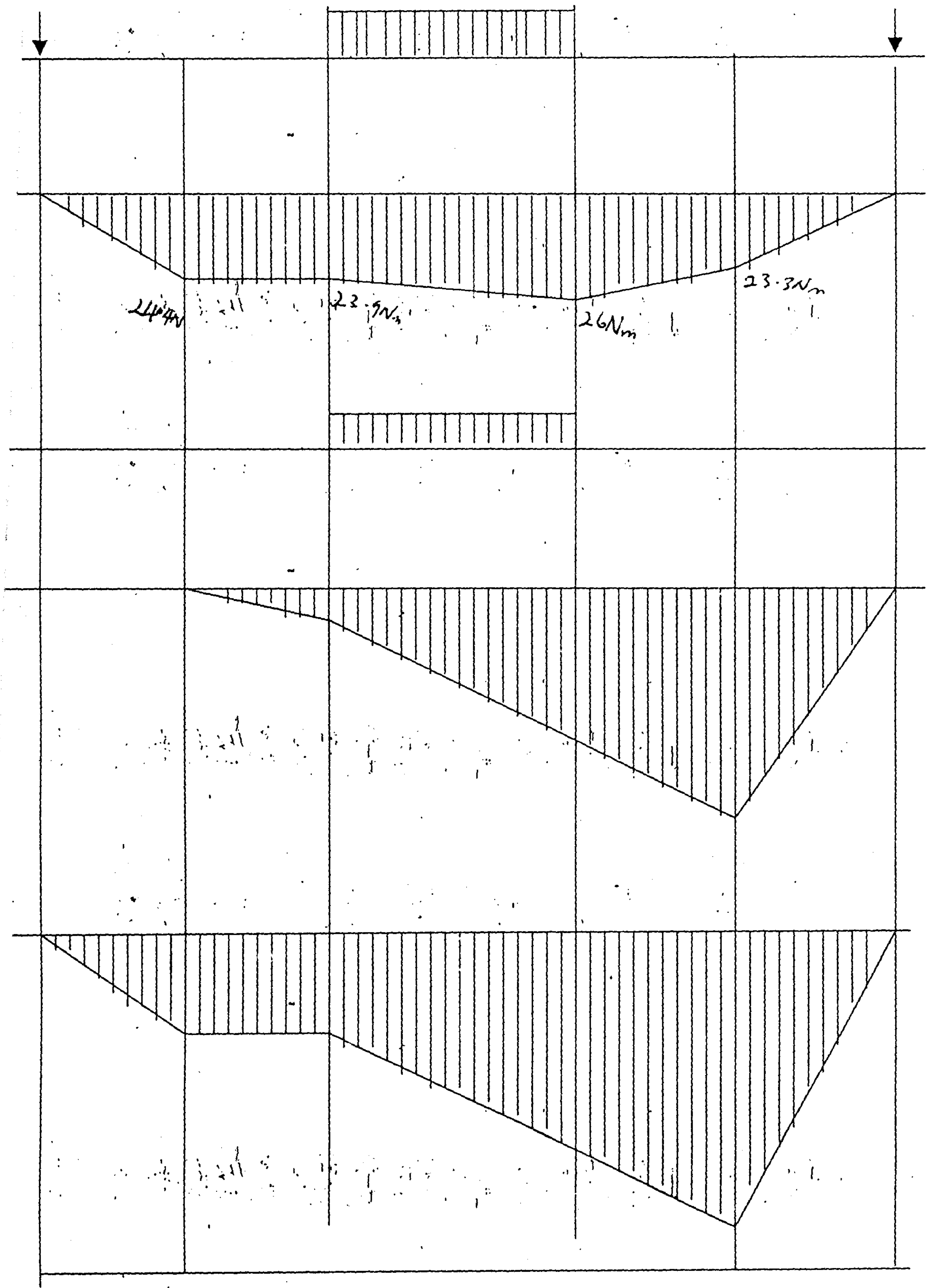
$0.163<x<0.243$ $-(-350)x512(x-0.1)(x-0.163)/2-891(x-0.163)+1168(x-0.163)+M=0$

$350(0.243(0.243)+512(0.143)(0.04)-891(0.08)+1168(0.08)+M=0$

$8.1+2.93-71.28+93.44+M=0$

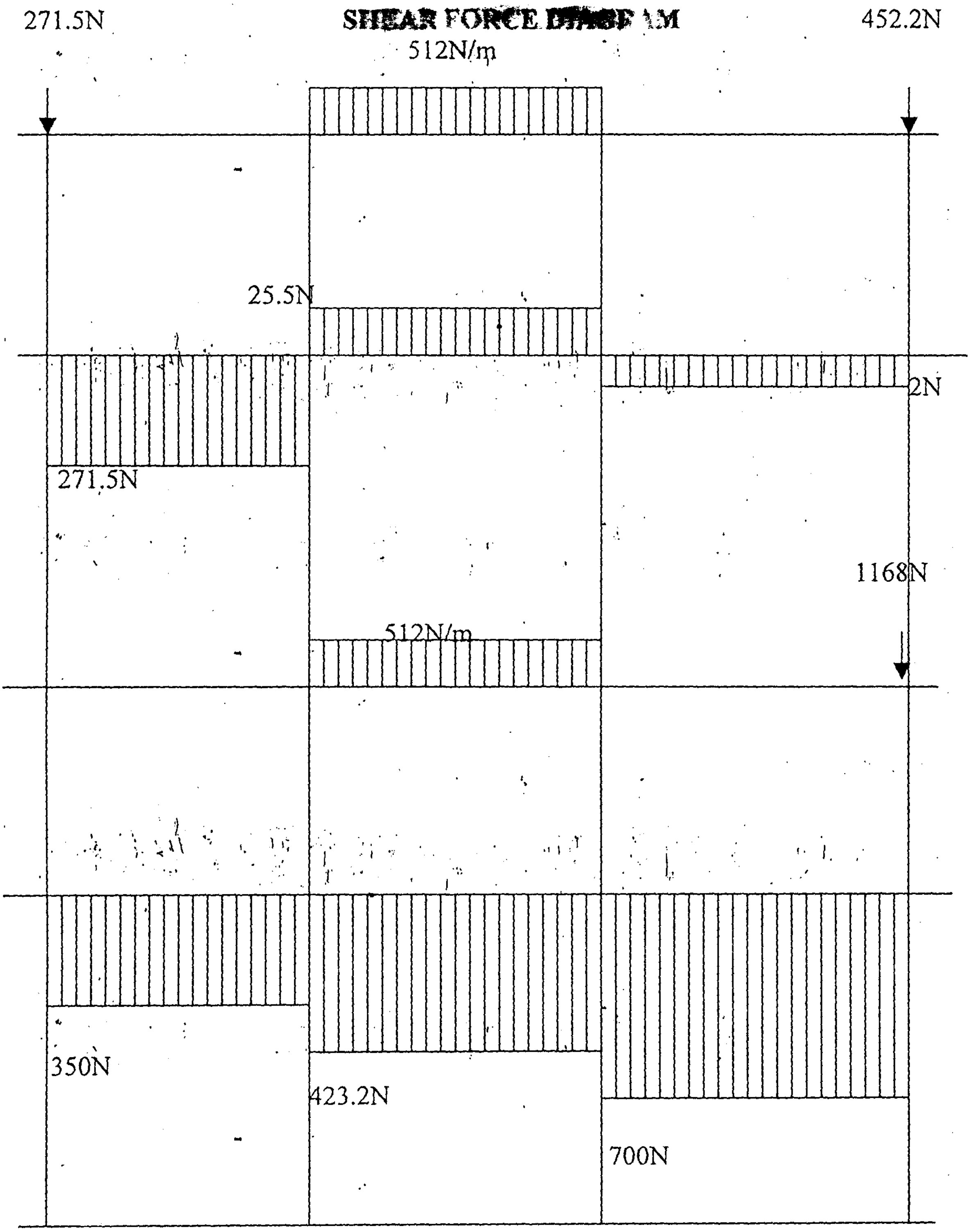
$M=-110.2Nm$

BENDING MOMENT DIAGRAM



Bending Moment Diagram

SHEAR FORCE DIAGRAM



Shear Force Diagram

3.8.1 Determination of torsional stress on the shaft

For torsional loads, the torsional stress τxy is given as (for solid shaft) below

$\tau xy = 16mt/\pi d^3$..3.40

where

Mt=torsional moment; N/m

d=shaft diametre,m

but $d^3 = 16/\pi sp\ (K_b M_b)^2 + (K_t M_t)^2$..................................... ...3.41

(ASME code equation for solid shaft)

where K_b=combined shock and fatique factor applied to torsional moment 1.5 for rotating shaft where load is gradualy applied

K_t = combined shock and fatique factor applied to torsional moment 1.0 for rotating shaft of gradually applied load

S_p = allowable stress = = (600pss) $40MN/m^2$ for shaft key way

M_b = maximum bending moment, NM

$= (23)^2 + (110)^2 = 112NM$

$d^3 = 16/3.142x40x10^6\ (1.5x112)(1x40.9)^2$

$d^3 = 16x172.9x10^{-6}/3.142x40 = 2766.4x10^{-6}/125.7$

d=0.028023m

d = 28mm

3.8.2 DESIGN OF SHAFT FOR TORSIONAL RIGIDITY

This is based on the permissible angle of twist

$\theta=584M_tL/Gd^4$ 3.42

where

θ = angle of twist, deg

L = length of shaft,m

M_t= forsional moment,Nm

G = fortional modules of elasticity,N/m^2

d = shaft diametre,m

$\theta=584x40.9x0.35/80x10^9x(0.028)^4$

$\theta=0.170^0$twist

3.8.3 DESIGN OF SHAFT FOR LATERIAL RIGIDITY

This is based on the permissible lateral deflection for proper bearing operation, accurate performance, satisfactory gear took action, shaft alignment etc.

$D^2y/dx^2 = Mb/EI$ 3.43

Where

Mb =maximum bending moment,Nm

E = modules of Elasticity.N/m^2

I = rectangular moment of inertia,m^4

From equation 3.43

D^2y=I/EI $Mbdx^2$ 3.44

$\int^{y}_{0}\int^{y}_{0} d^2_y = 1/TI \int^{xi}_{0}\int^{xi}_{0} Mbdx^2$ 3.45

i. For Deflection as a result of vertical forces on the shaft the bending moment id given as

$Mb = -271.5(x)$ 3.46

Applying to equation 3.42

$\int^{y}_{0}\int^{y}_{0} d^2y = 1/EI \int^{x}_{0}\int^{x}_{0} -271.5xdx^2$

$\int^{y}_{0} dy = 1/EI \int^{0.09}_{0} -271.5x^2 dx$

$y = 1/6tI \left|(-271.5x^3)\right|^{0.05}_{0}$

$y = -271.5(0.09)^3/6EI$

$y = -45.25x0.000729/EI$

but $I = \pi d^4/64 = 3.142(0.028)^4/64 = 3.0x10^{-8}m^4$

$E = 200GN/m^2 = 200x10^9N/m^2$

$Y = 0.03299/200x3x10^{9-8} = 0.03299/6000$

$Y = -5.5x10^{-6}m$
$Y = -0.0055mm$

ii. The Deflection as a result of horizontal forces on the shaft the maximum bending moment is given as

$M = -256x^2 - 553.7x + 43.1$ 3.47

Applying in equation 3.45

$\int^{y}_{0}\int^{y}_{0} d^2{}_y{}^2 = 1/EI \int^{0.243}_{0}\int^{0.243}_{0} - 256x^2 - 553.7x = 43.1dx^2$

$\int^{y}_{0} dy = 1/EI \int^{0.243}_{0} (-256x^3/3 - 553.7x^2/2 + 43.1x)dx$

$y = 1/EI \left|-256x^4/12 - 553.7x^3/6 + 43.1x^2/2\right|^{0.243}_{0}$

$y = 1/EI\left|-0.0744 - 1.324 + 1.273\right|$

$y = -0.1254/200x3x10^{9-8} = 0.2x10^{-4}m$

$y = 0.02mm$

3.9 BEARING SELECTION

i. To select desired design life

For Agricultural equipment = 3000-6000hrs

II. To calculate equivalent radial load, P

$P=XR+YT$ 3.48

Where

R = radial load, N

T = Thrust axial load,N

X and Y = radial and thrust factor

For single row ball bearing

$X_1=1.0\ Y=0$

But $R=P/2\pi Nr$ 3.49

Where

P = Power transmitted, w = 1500w

N = Speed of shaft, rpm = 350rpm

r = Radius of shaft, m = 0.028/2 = 0.014m

R = 1500/2x3.142 x 350 x 0.014 = 48.7N

III. To determine the thrust load, T then we analyse the load on the bearing

a. Weight of disc = 57N

b. Weight of ginger = 29.4N

c. Weight of pulley = 19.6N

d. Weight of gear = 7.8N

$T = 57+29.4+19.6+7.8=113.8N$.

Hence, the equivalent Radial load, P will be

From equation 3.48

$P = 1x48.7+0x113.8 = 48.7N$

IV. To determine the required (load) capacity cr

$G=PL_hN)^{1/k}/Z$.. 3.50

Where

K = Constant (for basll bearing Z = 25.6)

N = Speed of shaft = 350rpm

L_h = life of bearing = 4500

P = equivalent Radial load = 48.7N

$Cr = 48.7(4500x350)^{1/3}/25.6= 221.3N$

V. Calculation of the rated life of the bearing in hours

$L_h = 16700/N\ (c/p)^k$..3.51

$= 16700/350\ (221.3/48.7) = 47.7(93.8)$

$L_h = 4475.8 \approx 4476hrs$

VI. The required life of bearing in million revolution L, is given as

L = number of rev/sec of shaft x operating life of bearing

L = 350/60 x 4476 = 1566600/60 Mrv.

L = 26110Mrv.

VII. The dynamic capacity of the bearing C,

$C = L^{1/k}\ Fr$.. 3.52

Where

F_r = Radial Force, N = 425.2N

L = required life of bearing in Million Revolution

K = Constant (K=3)

$C = 26110^{1/3} \times 425.2 = 12614N$

But in kgf the dynamic capacity is

$12614/9.81 = 1285.8 \approx 12186kgf$

From the above data, then using the General Catalogue published by SKF European Bearing Division, conforming to International Standard Organisation (ISO).

The bearing selected has the following features

ISI No.30 BCO3

SKF 6305

3.10 KEY DESIGN

Keys are used to prevent relative motion between a shaft and the connected member through which torgue is being transmitted. Even though gears, pulley etc. are assembled with an interference fit, it is desirable to use a key designed to transmit the full torgue.

i. Determination of the resistance to the shaft torgue. This is the resisting couple which trends to prevent the key to roll in the fitted key way.

$F = T/r$ 3.53

Where

F = resistance force, N

T = Shaft torgue, N

R = radius of shaft, M

The torgue at the section of the shaft carrying the pulley is given as

$T = (t_1 - t_2)R$ 3.54

Where

T_1 = tension at the tight side of the belt, N

T_2 = tension at the slack side of the belt,N

R = Radius of the pulley.

$T = (232.1 - 3805) x 331/2 x 10^{-3}$

$= 193.6 \times 0.1655 = 32NM$

from equation 3.53

$F = 32/0.014 = 2285.7N$

ii To calculate the length of the key

The torgue that a shaft of diametre d can transmit allowing for a 25% reduction due to stress concentration is

$T = 0.75\pi\, d^3 \int_s / 16$ 3.55

Where

D = diametre of shaft,M

$\int_s$ shearing stress in the key, N/m^2

Also, the torgue that a square key can sustain from the stand point of shear is given as

$T_s = \int_s b/r$ 3.56

Equating3.52&3.53 to have equal strenght of stress

$0.75\pi d^3 \int_s/16 = \int_s blr$

$0.75\pi d^3 \int_s/16 = \int_s blr$

$0.75\pi d^3 \int 16 = blr$

Substituting d/4 for b

$0.75\pi\, d^3 = 16d/4/r$

$L = 0.75\pi/4r = 0.75 \times 3.142 \times (0.028)^2/4 \times 0.014 = 0.03299/m$

$L = 32.99mm \approx 33mm$

ii. The shearing stress $\int_s$ in the key is determined as follows

$S_s = F/bl = Fr/blr = T/blr$ 3.57

$S_s = T/blr = 32/0.028 \times 0.033 \times 0.014 = 9.8 \times 10^6/m^2$

$S_s = 9.8MN/m^2$

Key design is safe since the shear $S_s < S_a$ Allowable shear stress of carbon steel

111. The shear torgue that the key can sustain from the stand point of shear is given as

$T_s = 9.8 \times 10^6 \times 0.028/4 \times 0.033 \times 0.014$

From equation 3.52

$= T_s = 31.96 = 32Nm$

iv To determine the compressive stress S_c in the key

v. $S_c = F/(t/2) L = fr(t/2)Lr = t/(t/2) Lr$ 3.58

T = b for square key

$S_c = T/(t/2)Lr = 32/0.025/8 \times 0.033 \times 0.014 = 19.8\ 10^6 MN/m^2$

Design is satisfactory, since the compressive stress less than the allowable compressive stress of carbon steel.

$S_c < S_{ac}$

vi. To determine the shaft torque that the key can sustain from the stand point of compression is

$T_c = S_c (t/2)Lr$ 3:59

$T_c = 19.8 \times 106\ (0.026/8 \times 0.033 \times 0.014) = 32.016Nm$

$T_c = 32Nm$

A square key can sustain the same shaft torgue from the stand point of shear stress as it can from the stand point compression.

3.11.0 DESIGN OF FRAME

3.11.1 BEAM DESIGN

The total weight on the beamis calculated as follows

$W_B = W_p + W_d + 2W_s + W_m + W_c$ 3.60

Where

W_B = weight on the Beam

W_p = weight of pulley, N

W_g = weight of gear, N

W_d = weight of disc knives, N

W_s = weight of shaft, N

W_m = weight of material (ginger), N

W_c = weight of cylinder, (cutting chamber), N

Density of carbon steel = 7840kg/m^3

Acceleration due to gravity 9.8m/s^2

$W_c = \pi (r_{ic} - r_{2c})^2$ h x s x g 3.61

W_d = 2(4/3π(r1 – r2)3 P x g xn 3.62

Where

N = number of disc knives = 12

R_1 = Major diametreof the disc, m

R_2 = the small diametreof the disc, m

R_{ic} = diametre of the cylinder,m

R_{2c} = small diametre, m

From equation 3.58

$W_c = 3.142(0.065 - 0.014)^2 \times 7840 \times 9.81 \times 0.183 = 115N$

From equation 3.59

$Wd_2\ 2(4/3 \times 3.142(0.04 - 0.014)^3 \times 2700 \times 9.81 \times 12) = 42N$

From equation 3.57

$W_b = 25 + 42 + 115 + 24 + 138 = 360N$

W_b = is regarded as a point load on the Beam

Total load actingon each beam = 360/2 = 180 N

$W_{TB} = 180N$

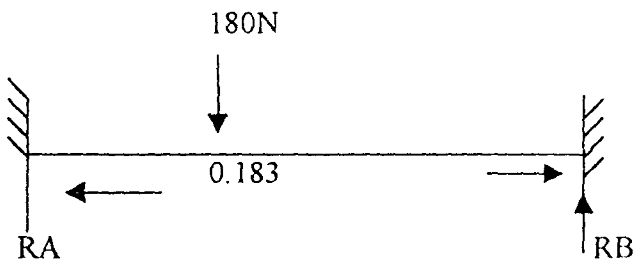


Fig 3.9 Free body diagram of beam.

$\sum fy = R_A - R_B + 180 = 0$

$R_B = 180 - R_A$ 3.63

Taking bending moment at R_B

$\sum My = -0.183RA + 180\ (0.183/2) = 0$

$R_A = 16.5/0.183 = 90N$

$R_B = 180-90 = 90N$

To obtain the maximum bending moment

$0 < x < 0.0915$

$-R_A(x)+m=0$

$M=R_A(x)$

$M=90(0.0915)=8.2Nm$

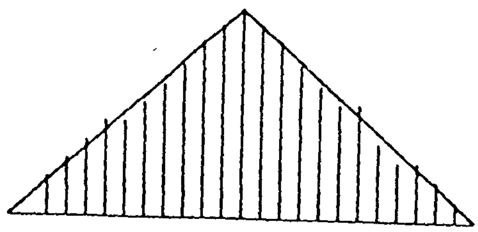


Fig. 3.10 Maximum Bending Moment Diagram

Using the relationship below

$BM/I = \sigma_{a}\mu/y$ 3.64

$BM/\sigma_{a}\mu = I/y$ 3.65

And also

$\sigma_{a}\mu = \sigma_{max}/N$ 3.66

Where

BM = maximum bending moment, Nm

I = Moment of inertia, M^4

$\sigma_{a}\mu$ = allowable maximum stress, N/m^2

σ_{max} = maximum tensile strenght, N/m^2

N = Factor of safety = 2

I/y = Elastic modulus

Tensile strength = $460 MN/m^2$

Thus

$\sigma a\mu = \sigma\ mnax / N = 460/5 = 92\ MN/m^2$

From equation 3.61

$B.M / \sigma a\mu = I/y$

$I/y = 82/92 = 0.087 \times 100\ m^3$

NOTE:Angular Bar of Mild Carbon Steel will be used for the Beam

3.11.2 DESIGN OF COLUMN

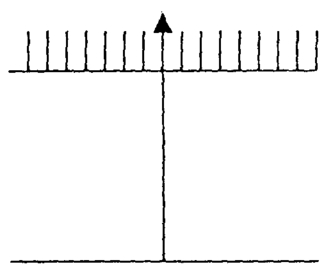


Fig. 3.11. Free body diagram of column.

The material will fail under excessive load being subjected, andfor this buckingham theory of failure is applied and Euler's critical load equation is also applied.

$P_{cr} = C\pi^2 EI/L^2 = NRA$ 3. 64

Where

P_{cr} = critical load, N

C = Slenderness ratio, depends on the type of material and the design of the ends.

E = Modules of elasticity = $200 \times 10^9\ N/m^2$

I = moment of inertia, M^4

N = factor of safety = 5

Ra - = Reactional forces that has been determined earlier for both end fixed c = 4

Thus

$NRa = C\pi^2 EI/L^2$

$I = NRaL^2/C\pi^2E$ 3.68

$I = 5X\ 113.5\ X\ (0.09)^2\ X\ 10^{-9}/4\ X(3.14)^2\ X\ 200 = 5.8\ X\ 10^{-11}\ M^4 = 58mm^4$

Hence the moment of inertia is $58mm^4$

3.12.0 DESIGN OF THE CUTTING KNIFE (DISC)

The cutting knife (disc) is 20mm thick and it is spherical in shape, sharpened at the edges. It is inclined on the shaft at an angle 30^0 for optimum cutting efficiency. The cutting knives are twelve (12) on each shaft, they are twenty four (24) altogether, they are firmly attached to the shaft by welding.

The area of the cutting knife (Disc) is given as

$$Ad = 4\pi^2 = (r1 - r2)^2 \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.69$$

$\pi 4 \times 3.142\ (0.04 - 0.015)^2$

$= 7.855 \times 1010\text{-}2\ m^2$

$Ad = 7855mm2$

Weight of the Disc can be calculated below

$Wd = \rho \times v \times g$

Where

ρ = dendity of material, kg/m^2 (Al.H.30 = $2700kg/m^3$)

$V = 4/3\ \pi^3$ = volume of the disc, m^3

g = acceleration due to gravity.

$Wd = 2700 \times 4/3\ (r1 - r2) \times 9.81 \times 3.142$

$2700 \times 4/3\ (0.04 - 0.015)3 \times 9.81 \times 3.142$

=1.7N

3.13.0 DESIGN OF HOPPER

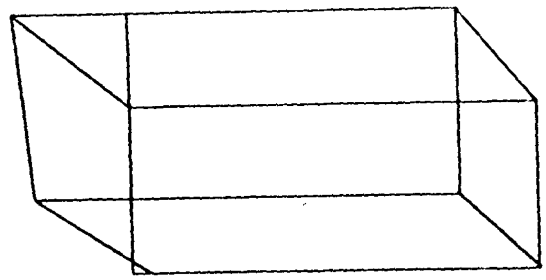


Fig 3.12 The Hopper dimension is shown above, the hopper is at the top of the cylinder (Cutting chamber) it serves as the feed inlet.

The volume of the hopper is given as

$$V = h/3\ (A_1 + A_2 + \sqrt{A_1 \times A_2}) \qquad 3.70$$

Where

V = volume of the hopper, m^3

$A_1 = L_1 \times B_1$ = Area of the top, m^2

$A_2 = L_2 \times B_2$ = Area of the base, m^2

H = height of the hopper, m

L_1 = Length of the top, m

L_2 = Length of the Base, m

B_1 = Breadth of the Top, m

B_2 = Breadth of the base, m

$V = 0.2/3\ (0.2 \times 0.15) + (0.04 \times 0.08) + \sqrt{0.2 \times 0.15 + 0.04 \times 0.08})$

$= 0.067\ (0.03 + 0.0032 + \sqrt{0.03 + 0.0032})$

$= 0.0144 m^3$

3.13.0 DESIGN OF CYLINDER (SLICING CHAMBER)

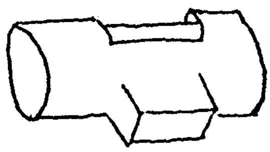


Fig. 14 Slicing Chamber.

The cylinder (slicing chamber) dimension is shown above, though it is not a complete cylinder since, the feed outlet is attached. It serves as the housing for the impeller.

The volume of the cylinder (slicing chamber) is given as

$V = 1/3\ \pi\ (r_1 - r_2)^2\ h/2 + h/3\ (A_1 + A_2 + \sqrt{A_1} + A_2) + B/3 + B_2 + \sqrt{B_1} + B_2$

Where

A_1 A_2, B_1, B_2 = Areas as shown in the figure above

$V = 1/6 \times 3.142\ (0.06 - 0.03)2 \times 0.065 +).15/3\ (0.5 \times 0.183 + 0.65\text{x}0.183 +$

$\sqrt{0.0092} + 0.012 + 0.665/3\ (0.065 \times 0.183 + 0.13 \times 183 + \sqrt{0.0092} + 0.024$

$V = 0.000042 + 0.0083 + 0.0049 = 0.0129\text{m}^3$

CHAPTER FOUR

4.0 THE GINGER SLICER

The slicer is composed of four principal parts, which are the Hopper, the Slicing Chamber, the Feed outlet and the Frame. The hopper has a triangular shape and it is attached directly on top of the slicing chamber. The slicing chamber is where the slicing action is carried out. It consits of the driver and driven shafts, which have some sets of twenty two blades attached on each. Attached directly below the slicing chamber is the feed-outlet, it has a rectangular shape. The hopper, slicing chamber and the feedoutlet are supported by the frame.

4.1 OPERATIONAL PRINCIPLE OF THE MACHINE.

The machine is made up of the inlet, cutting-components and the outlet majorly. The inlet is the hopper. The slicing component comprise of two shaft with sets of sixteen blades on each shaft and the shaft are enclosed in a cylinder slicing chamber and supported by bearing at each end. At one end of the driver shaft is attached pulley and at the other end is pinion but the driven shaft has a gear attached to it at one end, which makes contact with the pinion at the driver shaft, hence power is transmitted to the driven shaft via the driver shaft at a constant speed ratio i.e 1:1.

The source of power is a two (2hp) horse electric motor which by means of a drive belt trnasmits power to the pulley on the driver shaft at a speed ratio of 4:1 (speed reduction).

The driver shaft and driven shaft moves in opposite directions and as the ginger is fed through the hopper, as it falls on the rotating blades it is cut to a thickness of 8mm.

The bladeis inclined at 90^0 to the shaft and are equally spaced at 15mm intervals, and opposite disc are alternatively arranged, the sliced ginger is discharged.

4.2.0 METHODS OF CONSTRUCTION

The designed Machine was fabricated following strictly the design specifications, using the selected materials.

4.2.1 HOPPER

Mild steel of 16-guage was used, the specified dimensions was marked out on the steel sheet, before it was cut off using a cutter/scissors. Then it was welded together to form the Trapezoidal shape of the hopper. The hopper is attached to the semi-cylindrical top of the slicing chamber, to ensure easy access to the cutting components.

4.2.2 SHAFT

Mild steel was used for the two shafts. It was machined to the design specification i.e 25mm x 500mm with a lathe machine

4.2.3 BEARING BRACKET.

Four bearing brackets were provided, two for each shaft. The brackets are made from mild steel of 4mm thick. It was formed into a circular shape in order to house the bearing properly. The bearing brackets are welded to the sides of the slicing chamber.

4.2.4 CUTTING DISC

This was made from mild steel of guage 16, the blades were welded at 90^0 to the shaft. And are sixteen at intervals of 15mm to each other. The blades are circular with an outside diametre of 110mm and inner diametre of 25mm.

4.2.5 DISCHARGE OUTLET

This was made from a mild steel of guage 16, welded to the slicing chamber

4.2.6 PULLEY

The pulley bought was of mild steel and it conforms to the design specification.

4.2.7 GEAR

The Gear material bought perform effectively, and it conforms to the design specification.

4.2.8 FRAME

The stand was constructed using square or right angle bar of 4mm thickness, and welded to take the design shape

4.3.0 TEST PROCEDURE

After fabrication, performance test was carried out in order to ascertain the efficiency of the machine and also to see if the aim andobjective of the project is met.

The ginger finger were fed through the hopper into the slicing chamber, as the ginger finger come into contact with the blades which are carried by the rotating shaft, the gingers were sliced into slices and collected through the feed oulet. The test was carried out for 60s, for each cutting, the output were collected and weighted.

4.3.1 CALCULATIONA AND ANALYSIS

Testing with ginger finger

The testing was carried out for 60 second, using a stop watch for timing.

Weight of sliced ginger finger before oven drying $W_1 = 0.28kg$

Weightof sliced ginger finger after oven drying $W_2 = 0.03kg$

Moisture content $= (W_1 - W_2) \times 100\%/W1$

$$= (0.28 - 0.030/0.28) \times 100\% = 89.4\%$$

4.3.2 EFFICIENCY OF THE GINGER SLICER

Using the slicer, the slicing efficiency can be obtained using the expression below

$$\text{Efficiency} = \frac{\text{weight of the sliced ginger}}{\text{Weight of the ginger fed into the slicer}}$$

$$= (3.25/60)/\ (5.55/60) = 0.092kg/s \times 100 = 58\%$$

4.3.3 CAPACITY

From the testing of the slicer using the ginger finger the out put per time is obtained as

$$\frac{\text{Weight of ginger fed into the slicer}}{\text{Time taken to slice the ginger}} = 3.25/60 = 0.0542kg/s$$

4.3.4 RESULTS AND DISCUSSION

From the performance of the slicer using the ginger, it was obsered hat almost all the materials fed into the slicer were slashed and collected through outlet. Though some material (ginger) were retained in the slicer chamber.

The economic benefits has to be put into consideration in engineering design. In the design of ginger slicer, the basic factors considered are: the material to be used, the availability and cost of the materials, durability of material and the simplicity of the construction and operation in order to achieve the desired objectives of the project. Since the ginger slicer will be used on food product, hence it is imperative that all the materials that will come in contact with the food medium are resistance to corrosion and do not contaminant the food.

The installation and maintenance of the machine is also considered appropriately, and the result is that the machine is easy to assemble, dis-assemble and maintained by the user.

The performance of the slicer depends on various factors which includes the cross sectioanl areaof the ginger, the moisture content of the ginger, the speed of the blade and the feeding rate of the ginger.

Test 1

Using Ginger

Weight of whole ginger = W_1 = 5.55kg

89.4% Moisture content before oven drying.

Weight of sliced ginger = w_2 = 3.25kg

Weight of oven dried figner = w_3 = 0.45kg

Time of slicing = 60s

Test result

Moisture content = (w_2-w_3/w_2) x 100

= 3.25 – 0.45/3.25 = 89.4%

output/s = w_2/60kg/s

output/hr = 0.0542x3600 = 196kg/hr

input/s = w1/60 kg/s

= 5.55/60 = 0.0925kg/s

input/hr 0.0925 x 3600 = 333kg/hr

Efficency = input/output x 100

= 196/333 x 100 = 58.8 = 59%

Test 2

Using carrot

Weight of sliced carrot = w_1 = 3.76kg

Weight oven dried carrot = w_2 = 1.13kg

Time of slicing = 60s

Test Results

.Moisture cotnent = w2-w3/w2 x 100

= 2.25 – 1.13/2.25 x 100 = 49.7%

output/s = 0.0375 x 3600 = 135kg/hr

input/s w1 60 kg/s = 3.76/60 = 0.063kg/s

Input/hr = 0.063 x3600 = 22.56kg/hr

Efficiency = input/output x 100

= 135/225.6 x 100 = 59.8%

= 59.8 = 60%

4.4.0 COST ANALYSIS

The cost can be classified into three:

i. Cost of material

ii. Cost of labour

iii. Over-head cost

TABLE 1 : COST ANALYSIS

S/N	COMPONENT	MATERIAL	SPECIFICATION (mm)	QTY	UNIT PRICE #:K	PRICE #:K
1	Hopper	Mild steel		1	520.0	520.00
2	Spur gear and pinion	Case-hardening steel	1:1	2	250.00	500.00
3	Shaft	Mild steel	25x510 x25x440	2	300	600.00
4	Bolt and nuts	Mild steel	M_8	4	10.00	40.00
5	Electrode		E_{12}	58	3.00	174.00
6	Blade	Galvanised steel	120x1	28	20.00	560.00
7	Bearing		ISI No30 BCO3 SKF 6305	4	40.00	160.00
8	Bearing Bracket	Mild steel		4	10.00	40.00
9	Frame	Anglebar mildsteel	4x4x40		650.00	650.00
10	Hinges	Mildsteel	38.1x21	2	20.00	40.00
11	Pulley	mildsteel		1	300.0	300.00
12	Blade Housing		180x400x2	1	480.0	480.00
13	Drive belt	Rubber		1	200.0	200.00
14	Keyway		4x7x40	1	300.0	300.00
15	Painting				400.0	400.0
				Total		4964

LABOUR COST

Labour cost is taken to be 40% total material cost

= 40/100 x 4964

N1985.60

Over head cost is 10% of the material cost

=10/100 x 4964

N496.40K

Total cost of fabrication (T.C)

TC = Material cost + Labour cost + Over head cost

TC = N4964 + N1985.60 + 496.40 + N7446

Profit to be 10% of the total cost

= 10/100 x 7445

= N7454.60k

Therefore, price = Total cost + Profit

=N7446 + N744.6

= N8190.60k

CHAPTER FIVE

5.0 CONCLUSION AND RECOMMENDATION

5.1.0 CONCLUSION

The ginger slicer has proved to be very efficient means to slicing ginger at higher production rate.

A better condition of work is obtained by the use of the ginger slicer as against local cutting method. Timeliness of operation is an advantage of using the ginger slicer machine and a resultant reduction in Labour.

Safety is maintained with obvious economic gains. The operation of the ginger slicer is completely hygienic.

5.2.0 RECOMMENDATION.

In order to improve the production of sliced ginger,then certain steps should be taken, like ensuring that the whole slicing components are made of stainless steel. Private enterpreneurs should be encouraged to under take mass production of the ginger slicer machine either in partnership with the Government or solely.

Further investigative study or research should be carried out in order to introduce a conveyance channel or belt that will feed the ginger slicer i.e the ginger slicer should be converted to a processing plant, so that the sliced ginger would be carried or conveyed through a conveyor belt to where it should be dried.

REFERENCES

1. Bater,W.D and Marshal RC (1943): some methods for approsimate production of surface area of gruits, journal of Agricultural Research 66(10) 357 – 373

2. Craword R.J and Benham PP (1987), Mechanics of Gengineering materials, Longman Publishers, London. Pp22-23

3. Deutschman AD, Micheal WJ. And Wilson CE, (1975), Machine design. Theory and practice, Macmillan publisher, New York pp660-669

4. Farm Machinery (1981), Claude Culpin. 11th Edition.

5. Mohsenin, N.N (1984): physical properties of food and Agricultural materials. Gordon and Breach science publishers, second printing.

6. Macrae R Robin and Ms Sadler, (1993) Engyclopaedia of food science, food technology and Nutrition, volume two

7. Nwandikom, G.I and Mittal J.P, (1988): physical properties of yam tuber (D.spp). In Rodos Resnicek ed, physical properties of Agricultural materials and products. Hermphere publishing corporation Newyork, U.S.A

8. Purseglore J.W Brown EG Green CL Robins SRJ, (1981): spices vol. 2 Longman, London.