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# THE OPEN UNIVERSITY OF SRI LANKA

FACULTY OF ENGINEERING TECHNOLOGY

DIPLOMA IN TECHNOLOGY – LEVEL 04

FINAL EXAMINATION – 2010/2011

## *MEX 4231 – ELEMENTARY MACHINE DESIGN – PAPER I*

***DATE*** : ***MARCH 04, 2011***

***TIME*** : ***1430HRS. – 1630 HRS.***

***DURATION*** : ***TWO HOURS***

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***READ THE FOLLOWING INSTRUCTIONS CAREFULLY BEFORE ANSWERING THE QUESTION PAPER***

1. *This question paper has Two Parts,* ***Part A*** *and* ***Part B****. Part A has* ***three (03)*** *questions and Part B has* ***five (05)*** *questions. Answer* ***only four (04)*** *questions selecting* ***at least one (01) question*** *from Part A, but* ***not more than Three (03)*** *questions from Part B.*

2. ***All questions carry equal marks.***

3. *Read the questions carefully before you start answering each question.*

4. *Write the relevant question number at the beginning of the answer.*

5. *Before submitting your answer script, fill the box on the front page by writing the question numbers for which you have already answered.*

6. *Some questions require design data. They are provided to you separately. Do not write anything on these data books. At the end of the examination return them to the supervisor.*

7. ***Assume any missing dimensions or design data. All such assumptions shall be clearly stated appropriately in the relevant answers.***

8. *Any sketches that you provide to explain your answer shall be neatly drawn and labeled.*

9. *Use the charts, catalogues, data sheets and tables available and return them to the supervisor at the end of the examination.*

10. *Use SI Units in all calculations, and present the answers underlined with correct units.*

## *PART A*

***Question 01.***

a. Discuss the primary objectives of maintaining interchangeability in mass production of a particular component. With a suitable example explain how interchangeability is employed in design practice.

b. Briefly define or explain within the context of machines, the terms of each pair given below distinctly.
   i. Reliability and safety factor.
   ii. Ergonomics conditions and operator safety.

c. Discuss how vibration can occur in machines which are in operation and the effects of vibration on machines. Quote relevant examples to explain your answer.

***Question 02.***

Write answers to any four (04) of the following:

a. Compare and contrast shafts and axles considering their design and practical applications.

b. Explain the self locking condition of screw threads.

c. What are the categories of engineering fits between two mating components? Explain your answer with relevant sketches.

d. Explain $L_{10}$ life of a rolling element bearings.

e. What is preloading of a bolt and the necessity of preloading a bolt?

***Question 03.***

a. Some technological innovations have altered the course of history and greatly influenced in further developments in engineering. Explain such an innovation using an appropriate example.

b. Discuss the changes in the traditional design process with the introduction of computer aided design and computer aided manufacturing.

c. Consider a latest engineering product and explain the important features in its design that the designer has considered related to environment, ergonomics, sustainability and economic factors. Also explain/justify clearly, why it is important to consider the above factors.

## *PART B*

***Question 04.***

Pulleys are used to transmit power from one shaft to another by means of belts or sometimes ropes. These pulleys may be made from cast iron, cast steel or processed steel. Fig.4 shows an overhung cast iron pulley with 6 arms mounted on to a shaft, where a leather belt of density 1,000 $kg/m^3$ is used to transmit power of 40 $kW$ at 250$rpm$.

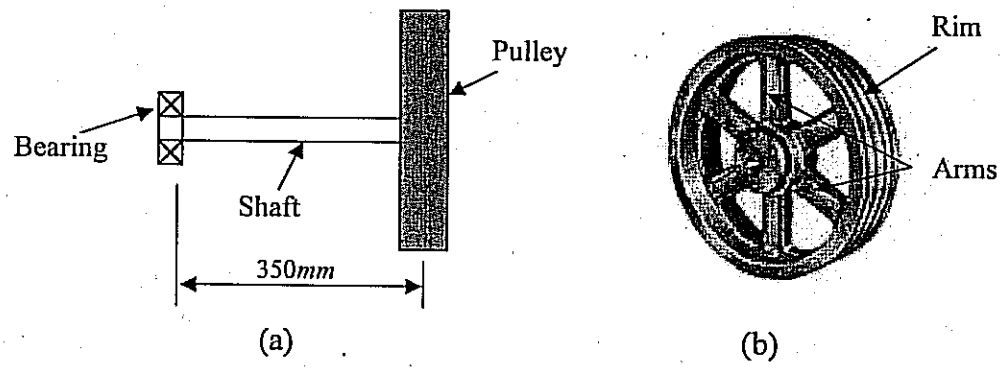



Fig.4: (a) Sketch of a pulley arrangement, (b) Different types pulleys

**Following design data are given;**

• Density of cast iron is 7200$kg/m^3$, • Maximum tensile stress of belt material is 2.6$MPa$, • Tensile stress of pulley arm material is 20$MPa$, • Tensile stress of pulley rim is 20$MPa$, • Shaft is equally strong in tension and compression which is 75$MPa$, • Shear stress of key is 50$MPa$ • Thickness of the belt 10$mm$, • Friction coefficient is 0.25.

Calculate the, *1*. Diameter of the pulley
*2*. Width of the belt
*3*. Diameter of the shaft
*4*. Dimensions of the key
*5*. Dimensions of the arms, which have elliptical cross section with minor diameter is half the major diameter.

***Question 05.***

Show that the torque (T) required to raise a load (W) by a square threaded screw and nut pair can be derived as,

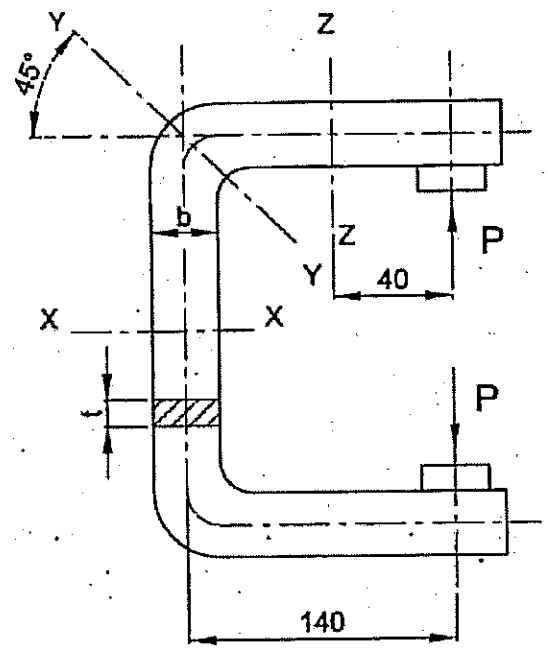
$$T = W \tan(\alpha + \phi) \cdot \frac{d}{2}$$

Where, $\alpha$ is the helix angle, $\phi$ is the friction angle and $d$ is the mean diameter of the screw.

The cutter of a broaching machine is pulled by a square threaded screw of 55$mm$ external diameter and 10$mm$ pitch. The operating nut which takes an axial load of 400$N$ to remove metal on a flat surface, has an internal diameter of 60$mm$ and external diameter of 90$mm$. If the cutting speed is 6$m/min$, determine the power required to rotate the operating nut. Also find the efficiency of the screw.

You may consider the coefficient of friction for all contact surfaces of the nut as 0.15.

*Question 06.*




All dimensions in mm

Fig.6

A C-clamp shown in Fig.6 is made of steel casting which has an allowable stress of $100MN/m^2$. The cross section of the clamp at section *X-X* is rectangular; its width (*b*) is equal to twice the thickness (*t*). If the clamp carries a load (*P*) of 25*kN*, determine,

a. the dimensions (b and t) of the clamp

b. the stresses at sections Y-Y and Z-Z sections.

*Question 07.*

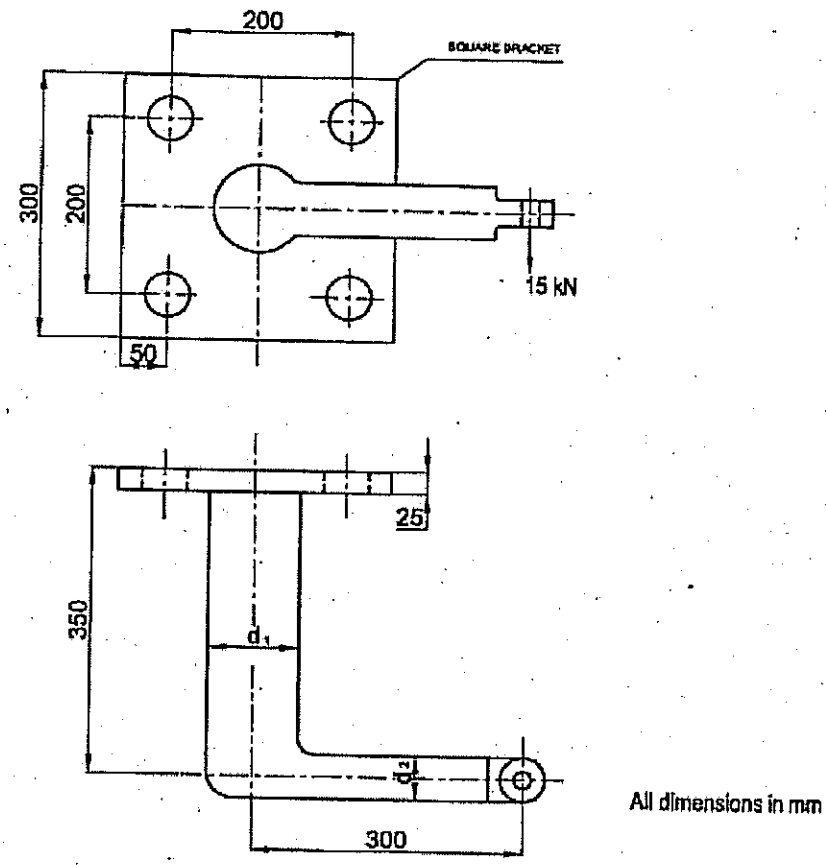



Fig.7

Four bolts are used to retain a solid bracket carrying a vertical load of $15kN$, which acts through the centre of the hole, as shown in the Fig.7. Two circular arms with diameters $d_1$ and $d_2$ are used to obtain the relevant extension to the load.

Calculate,

a. the diameters $d_1$ and $d_2$ of the circular arms if the permissible stresses are $100MN/m^2$ in tension and $60MN/m^2$ in shear.

b. the forces acting on the bolts and also the bolt diameter.

*Question 08.*

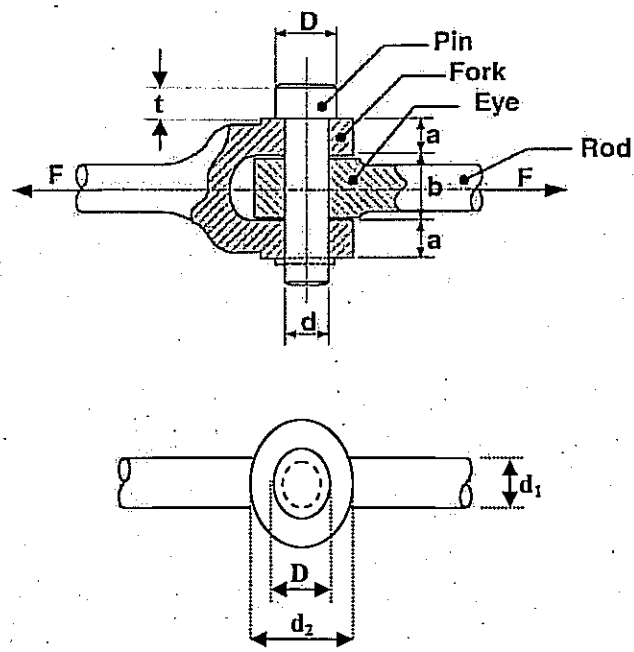



Fig.8: Sketch of a Knuckle joint

Fig.8 shows a Knuckle joint. If the design strengths are 75*MPa* in tension, 60*MPa* in shear and 150*MPa* in compression, design the knuckle joint completely to transmit a tensile load of 150*kN*.

***Note:*** *If the diameter of the Rod is $d_1$ as shown in the Fig.6, the other related dimensions can be fixed empirically as;*

*Diameter of the pin, $d = d_1$*
*Outer diameter of the eye, $d_2 = 2d_1$*
*Diameter of the Pin head and the collar, $D = 1.5\ d_1$*
*Thickness of the eye (rod end), $b = 1.25d_1$*
*Thickness of the fork, $a = 0.75d_1$*
*Thickness of the pin head, $t = 0.5d_1$*

***<u>You should check for following strengths</u>****, 1. tension of the solid rod, 2. failure of the knuckle pin in shear 3. eye (rod end) in tension, 4. eye (rod end) in shear, 5. eye (rod end) in crushing, 6. fork ends in tension, 7. fork ends in shear, 8. fork end in crushing.*

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## FACULTY OF ENGINEERING TECHNOLOGY

## DIPLOMA IN TECHNOLOGY – LEVEL 04

**FINAL EXAMINATION – 2010/2011**

***MEX4231 – ELEMENTARY MACHINE DESIGN – PAPER II***

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The Open University of Sri Lanka

***DATE*** : *MARCH 04, 2011*
***TIME*** : *0930HRS. – 1330HRS.*
***DURATION*** : *FOUR HOURS*

***READ THE FOLLOWING INSTRUCTIONS CAREFULLY BEFORE ANSWERING THE QUESTION PAPER***

1. *This question paper has only one question. Answer all parts of the question.*
2. *Devote 15 to 20 minutes to read and understand the question carefully.*
3. ***Wherever appropriate, use catalogues, tables, data sheets and charts provided to you at the examination hall. State in the answer clearly if you extract and use any data or information from such literature to support design calculation.***
4. *At the end of the examination, hand over all such literature to the supervisor or an invigilator.*
5. ***Any missing data may be sensibly and reasonably assumed, provided that such data are clearly stated with reasons to accept them.***
6. *Any ideas/opinions presented in the form of neatly drawn sketches in place of written presentation are encouraged.*
7. *Any results from calculation should be presented with their correct units, unless they are dimensionless. All such results should be underlined.*
8. ***It is important that candidates answer all parts of the question listed in that given order in the question.***
9. ***Use SI units in all calculations and present the answer underlined with correct units.***

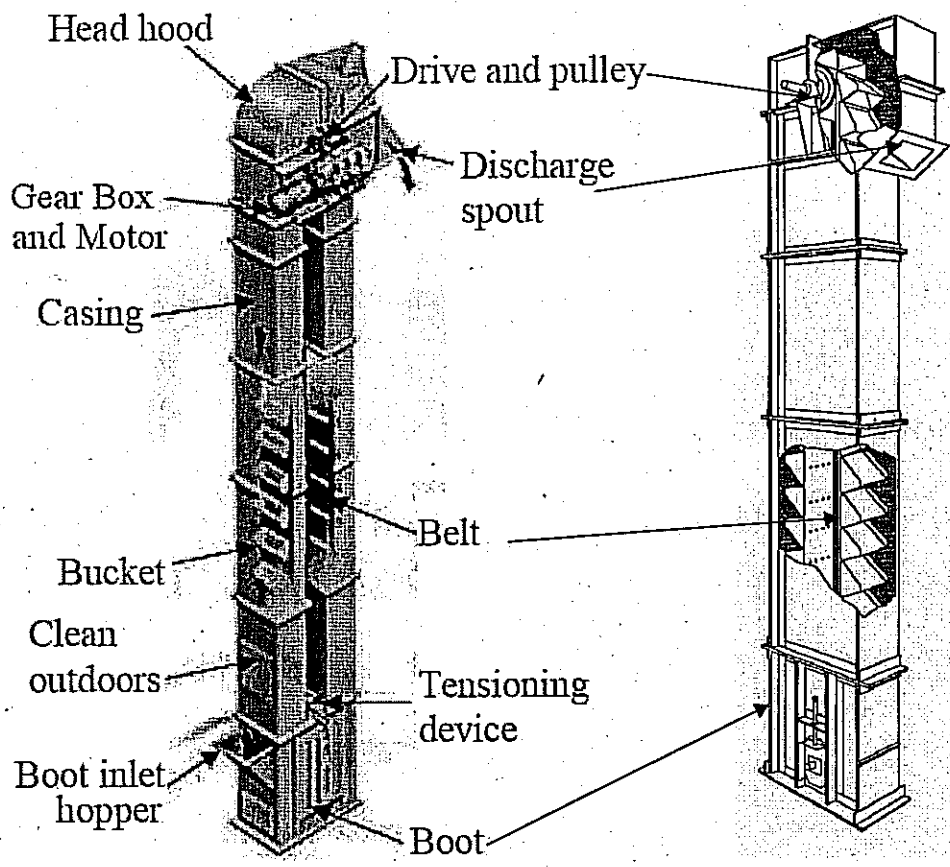



Fig.1: Main Components of Bucket Elevator

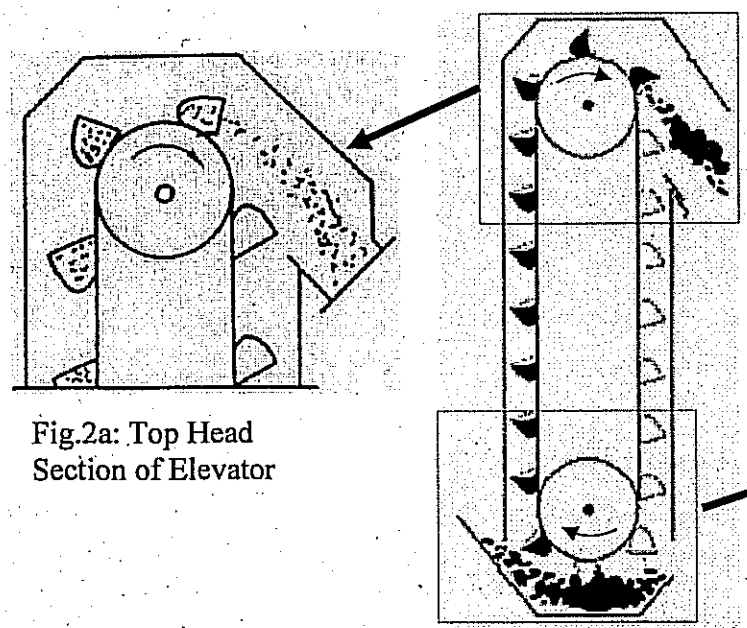


Fig.2a: Top Head Section of Elevator

Fig.2b: Schematic diagram of the Elevator

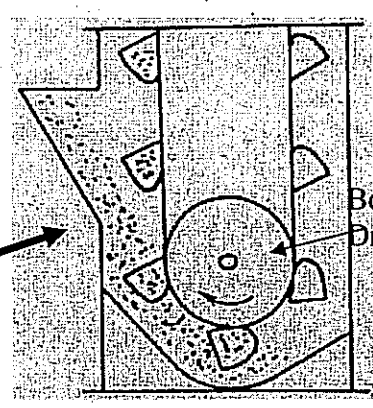



Fig.2c: Bottom Boot Section of Elevator

Fig.2: Bucket Elevator

## *Introduction to the Question*

In the paddy post harvest system, it is a common practice that paddy is transported, or conveyed from one place to another. Traditionally, the method of transportation is manual. However, manual methods have been replaced by bucket elevators and conveyors.

Paddy is a highly abrasive material and it causes excessive and rapid wear on mechanical parts such as screw, chain, pneumatic conveyors, etc. For this reason, bucket elevators and belt conveyors (both using rubber-covered belts) are preferred to. Not only the latter is economical, but has longer operational life.

Bucket elevator, shown in Fig.1, is a lifting device that can be used to lift bulk materials from one elevation to another and Sri Lankan rice milling industry uses bucket elevator y to elevate the rice from ground level to parboiling tanks, where the paddy is boiled and directed to required locations by belt conveyors, as shown in Fig.3.

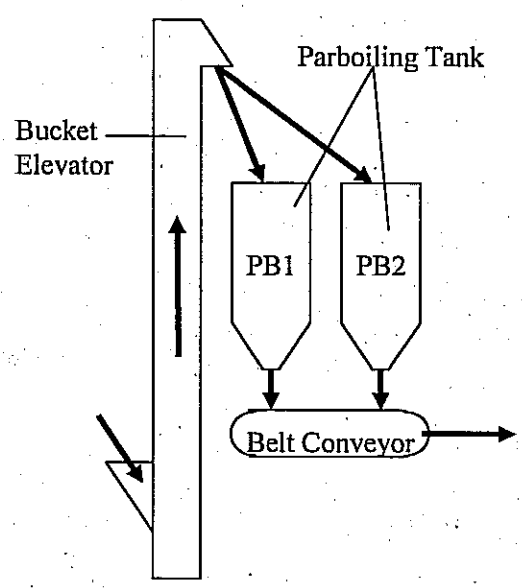



Fig.3: Schematic diagram of the paddy flow, shown by arrows.

Bucket elevators operate by using an endless belt on which rectangular buckets, refer Fig.4, are mounted. The belt revolves between a top and bottom drum and the buckets are carried by the belt. At the bottom the buckets pick up paddy fed into the elevator boot, and at the top the paddy is discharged as the bucket turns downward over the head pulley (top drum). It is a centrifugal discharge, where the speed of the belt around the top pulley (top drum) flings paddy out of the bucket and falls directly into the discharge spout of the upper casing. Spilling of the material is avoided by selecting the correct speeds, corresponding upper pulley diameter and proper adjustment of the discharge spout on the upper casing.

Bucket elevator mainly consists of the following parts

1. Boot section (Lower part) with boot pulley(drum) and paddy inlet hopper (refer Fig.2c).

## *Introduction to the Question*

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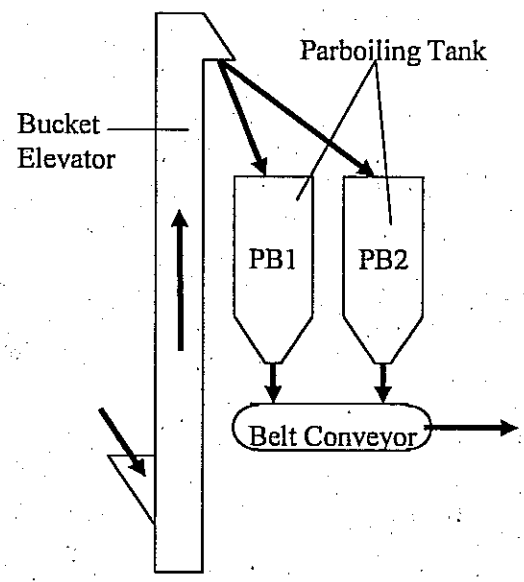



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Bucket elevator mainly consists of the following parts

1. Boot section (Lower part) with boot pulley(drum) and paddy inlet hopper (refer Fig.2c).

2. Head section (Top part) with top pulley(drum), drive mechanism and paddy outlet spout (refer Fig.2a).
3. Belt (bucket carrier) & buckets.
4. Sheet metal housing.

## *Question*

Assume that you are a consulting design engineer and you have been asked to design a bucket elevator to the following basic requirements.

Elevator capacity for paddy - $24 tons/hr$
Elevator height - $7m$
Bulk density of paddy - $576 kg/m^3$

Design the bucket elevator answering the questions given below.

Following details are given from hand books and catalogues to support your design analysis.

- The height of the elevator can be considered as the height from the centre of the boot pulley to centre of the top pulley.

- Arrangement of the suitable bucket is shown in the Fig.4 (a) and (b). The capacity of a bucket when filled to level A-A (refer Fig.4a) is 1,416 $cm^3$. Because of the difficulty of loading the buckets to maximum capacity, it is desirable to have a small tolerance when designing the elevator. The effective capacity is generally considered as 90% of the full capacity. The spacing of adjacent buckets is 165 $mm$ as shown in Fig.4b.

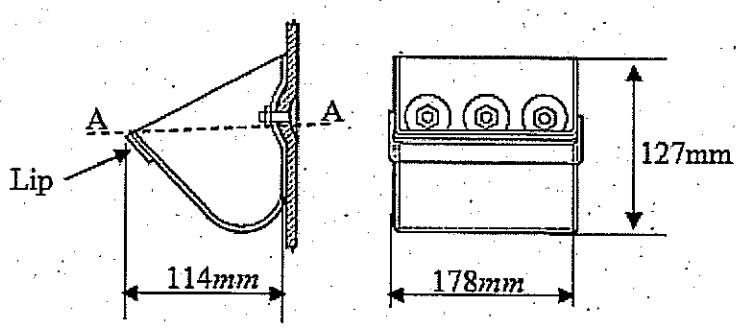



Fig.4a: Dimensions of bucket

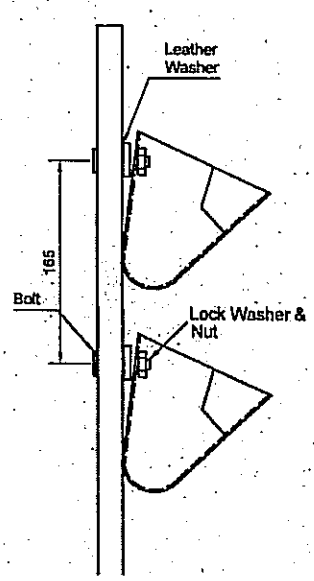



Fig.4b: Arrangement of Buckets

Fig.4: Elevator Bucket

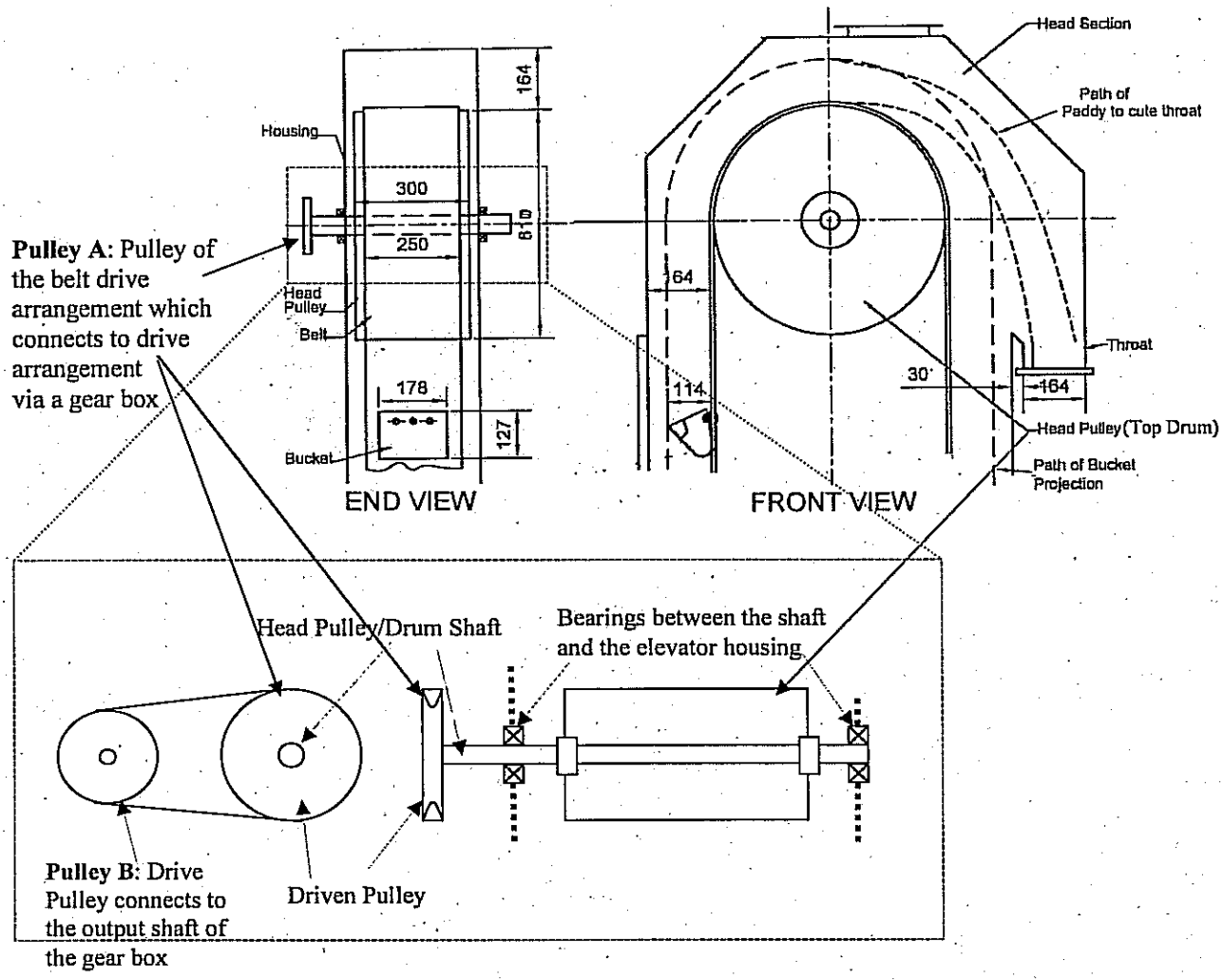



Fig.5: Schematic diagram illustrates the head section and drive arrangement (assume any missing dimensions).

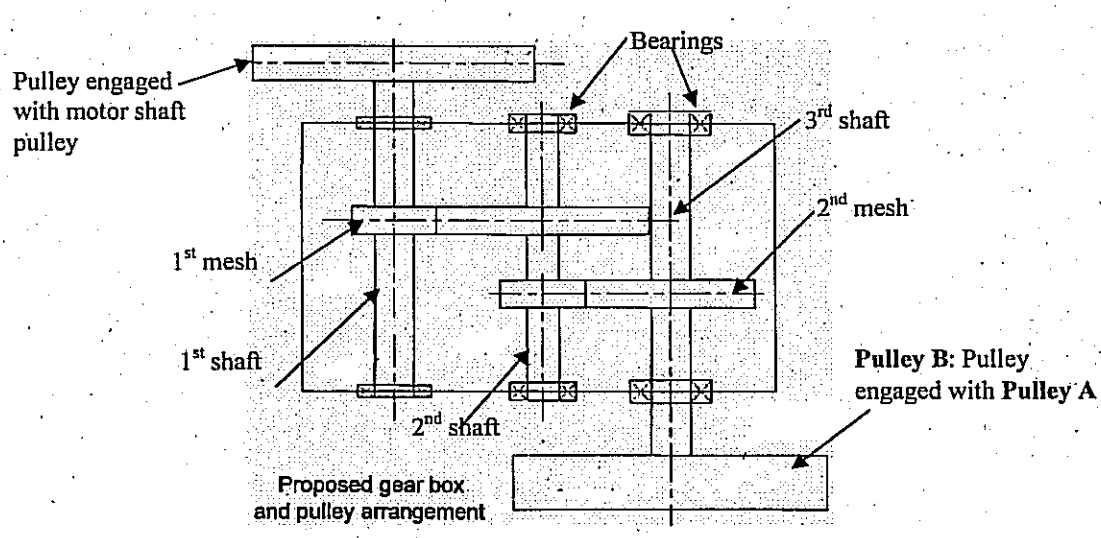



Fig.6: Schematic diagram of the gear box

- The most suitable head pulley (drum) and the boot pulley (drum) have a diameter of 610 *mm*.

- The drive mechanism shown in Fig.6 consists of a gear train to obtain appropriate speed reduction. A belt drive system transmits power from the motor to the gear box input shaft, and a second belt drive system transmits power from the output shaft of the gear box to the top drum shaft (drive shaft) of the bucket elevator.

- The speed for centrifugal bucket elevators is usually in the range of 1*m/s* to 2*m/s* to ensure the paddy throws into the chute at the head pulley.

- The total load on the belt (bucket carrier) attributes to the weight of product to be lifted plus the dredging drag load as the bucket scoops up the paddy. The dredging load is assumed to be 25% of the bucket load.

- The buckets are 178 *mm* wide, so the width of the belt can be assumed equal or slightly wider than width of the bucket. Considering the low operational speed, bucket width and projection (which is 114 *mm*), the most suitable belt is a 5-ply 28-oz belt, with a mass of 0.0022 *kg/mm* width/meter length/ply.

- The most suitable head pulley weighs 44 *kg*.

**Answer the following;**

1. Determine the number of buckets needed to pass top of the elevator to achieve the required elevator capacity.

2. Decide the linear speed of the belt.

3. Calculate the theoretical power required to drive the bucket elevator at maximum capacity.

4. Estimate the required motor power needed to drive the elevator, considering the mechanical inefficiencies, hence select a suitable electric motor as the prime mover.

   A proposed arrangement of the drive system is shown in Fig.6. It consists with first V-belt drive system between motor and input shaft of the gear box, and second V-belt drive system which connects output shaft of the gear box and top head shaft.

   i. Define suitable speed reductions at each stage of the drive system.
   ii. Design a suitable V-belt drive system with standard pulleys and select suitable keys for the first belt drive between motor and input shaft of the gear box.
   iii. Design the suitable a pair of spur gear in the gear train completely, from the point of strength, wear and dynamic loads.
   iv. Calculate the minimum diameter of the output shaft of the gear box.

5. Determine the minimum diameter of the top head pulley/drum shaft on which the driven pulley is mounted as shown in Fig.5. Also select suitable rolling element bearings for this Shaft.

6. There are two main problems in these kind of elevators. They are: *1.* in the case of a power failure the belt tends to rotate backwards which cause many difficulties. *2.* the belt tension gradually tend to reduce when operating, which cause many problems. Illustrate suitable mechanisms to prevent the elevator's backward rotation and -a belt tensioner which enable tensioning the bucket carrier belt.

7. This type of belt bucket elevators can fail in a number of ways. D*is*cuss five (5) ways which it can fail.

**The following information is provided in support of the *d*esign analysis, where all data and equation given below may or may not be needed for your calculations. Apart from the data given below you are provided with, *1.* Instruction booklet of spur and helical gear design, *2.* Motor catalog, *3.* Bearing selection manual, *4.* Handbook of metric keys and keyways and *5.* Handbook of V-belt drives.**

Please note that you are given only basic formulae and such formulae need to be further dealt with before applying them appropriately.

Power transmission efficiencies are:

Electric motor : 85%
Belt Drive : 90%
Gear Drive : 95%
Bearings : 95%

Density of steel: $7.8 \times 10^3$ kg/m$^3$

Design shear stress = 0.3 x yield stress in tension
= 0.18 x ultimate stress in tension or which ever smaller.

Design value of the normal stress = 0.36 x ultimate stress in tension
= 0.6 x yield stress in tension
or which ever smaller.

- <u>If necessary select one of the following steels for the shaft material</u>

| AISI Number | Yield strength ($MN/m^2$) | Ultimate tensile strengths ($MN/m^2$) |
|---|---|---|
| 1010 | 303 | 366 |
| 1018 | 373 | 442 |
| 1045 | 532 | 626 |
| 4340 | 683 | 766 |

If key ways are present then values are to be reduced by 25%.

For ductile materials Yield stress in shear = 0.5 x yield stress in tension.

- A shaft having a diameter d, when subjected to combined bending and torsion the equivalent bending, and torsional stresses at any point on the shaft are,

$$\tau_{eq} = \frac{16}{\pi d^3}\left[K_b M + \sqrt{(K_b M)^2 + (K_t T)^2}\right]$$

$$\text{and } \tau_{eq} = \frac{16}{\pi d^3}\left[(K_b M)^2 + (K_t T)^2\right]^{\frac{1}{2}}$$

| | $k_b$ | $k_t$ |
|---|---|---|
| **For Stationery shafts:** | | |
| Load Gradually applied | 1.0 | 1.0 |
| Load Suddenly applied | 1.5 to2.0 | 1.5 to 2.0 |
| **For Stationery shafts:** | | |
| Load Gradually applied | 1.5 | 1.0 |
| Load Suddenly applied (minor shock) | 1.5 to 2.0 | 1.0 to 1.5 |
| Load Suddenly applied (heavy shock) | 2.0 to 3.0 | 1.5 to 3.0 |

$$\sigma_n(\max) = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_n(\min) = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau(\max) = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

- For gears, basic Louis Equation is :

$$F = s\, b\, P_c\, y$$

For ordinary design conditions, the face width is limited to a maximum of 4 times the circular pitch. Letting $b = kP_c$, where the factor, $k \leq 4$,

$$F_d = \frac{21V(bC + F)}{21V + \sqrt{bC + F}} + F$$

Where, $F_d$ = dynamic load, N
$V$ = pitch line velocity, m/s
$b$ = face width, m

$$F = \frac{\text{gear torque}}{\text{pitch diam, of gear}} = 2M_t / D$$

$C$ = a constant, in N/m

$$F_w = D_P\, b\, K\, Q$$

Where, $D_p$ = pitch diameter of smaller gear (pinion), m
$b$ = face width of gear, m
$K$ = stress factor for fatigue, $N/m^2$
$Q = 2N_g/(N_p + N_g)$

$N_g$= number of teeth on gear
$N_p$= number of teeth on pinion

and $$K = \frac{s_{es}^2 (\sin\phi)(1/E_p + 1/E_g)}{1.4}$$

Where, $s_{es}$ = surface endurance limit of a gear pair, $N/m^2$
$E_p$ = modulus of elasticity of the pinion material, $N/m^2$
$E_g$ = modulus of elasticity of the gear material, $N/m^2$
$\phi$ = pressure angle.

The surface endurance limit may be estimated from,
$s_{es} = (2.75\ (BHN) - 70)\ MN/m^2$

where, BHN may be approximated by the average Brinell hardness number of the gear and pinion up to a BHN of about 350 for steels.

The wear load $F_w$ is an allowable load and must be greater than the dynamic load $F_d$.

Values of Deformation Factor **C** in kN/m – for dynamic load check.

| **Materials** | | **Involute tooth form** | **Tooth Error - mm** | | | | |
|---|---|---|---|---|---|---|---|
| Pinion | Gear | | 0.01 | 0.02 | 0.04 | 0.06 | 0.08 |
| cast iron | cast iron | 14 ½ ° | 55 | 110 | 220 | 330 | 440 |
| steel | cast iron | 14 ½ ° | 76 | 152 | 304 | 456 | 608 |
| steel | steel | 14 ½ ° | 110 | 220 | 440 | 660 | 880 |
| cast iron | cast iron | 20 ° full depth | 57 | 114 | 228 | 342 | 456 |
| steel | cast iron | 20 ° full depth | 79 | 158 | 316 | 474 | 632 |
| steel | steel | 20 ° full depth | 114 | 228 | 456 | 684 | 912 |
| cast iron | cast iron | 20 ° stub | 59 | 118 | 236 | 354 | 472 |
| steel | cast iron | 20 ° stub | 81 | 162 | 324 | 486 | 648 |
| steel | steel | 20 ° stub | 119 | 238 | 476 | 714 | 952 |

Values for $S_{es}$ as used in the wear load equation depend upon a combination of the gear and pinion materials. Some values for various materials for both $S_{es}$ and k are tabulated.

| **Average Brinell Hardness Number of steel pinion and steel gear** | | **Surface Endurance Limit** $S_{es}$ (MN/m$^2$) | **Stress Fatigue Factor K** (kN/m$^2$) | |
|---|---|---|---|---|
| | | | **14 ½ °** | **20°** |
| 150 | | 342 | 206 | 282 |
| 200 | | 480 | 405 | 555 |
| 250 | | 618 | 673 | 919 |
| 300 | | 755 | 1004 | 1372 |
| 400 | | 1030 | 1869 | 2553 |
| **Brinnel Hardness Number BHN** | | | | |
| Steel Pinion | Gear | | | |
| 150 | C.I. | 342 | 303 | 414 |
| 200 | C.I. | 480 | 600 | 820 |
| 250 | C.I. | 618 | 1000 | 1310 |
| 150 | Phosphor Bronze | 342 | 317 | 427 |
| 200 | Phosphor Bronze | 445 | 503 | 689 |
| C.I. Pinion | C.I. Gear | 549 | 1050 | 1420 |
| C.I. Pinion | C.I. Gear | 618 | 1330 | 1960 |

Form Factors y – for use in Lewis Strength Equation (transformed). $F = s\pi^2 kym^2$ is given in the following table:

| Number of Teeth | 14 ½° Full-Depth Involute of Composite | 20° Full-Depth Involute | 20° Stub Involute |
|---|---|---|---|
| 12 | 0.067 | 0.078 | 0.099 |
| 13 | 0.071 | 0.083 | 1.103 |
| 14 | 0.075 | 0.088 | 0.108 |
| 15 | 0.078 | 0.092 | 0.111 |
| 16 | 0.081 | 0.094 | 0.115 |
| 17 | 0.084 | 0.096 | 0.117 |
| 18 | 0.086 | 0.098 | 0.120 |
| 19 | 0.088 | 0.100 | 0.123 |
| 20 | 0.090 | 0.102 | 0.125 |
| 21 | 0.092 | 0.104 | 0.127 |
| 23 | 0.094 | 0.106 | 0.130 |
| 25 | 0.097 | 0.108 | 0.133 |
| 27 | 0.099 | 0.111 | 0.136 |
| 30 | 0.101 | 0.114 | 0.139 |
| 34 | 0.104 | 0.118 | 0.142 |
| 38 | 0.106 | 0.122 | 0.145 |
| 43 | 0.108 | 0.126 | 0.147 |
| 50 | 0.110 | 0.130 | 0.151 |
| 60 | 0.113 | 0.134 | 0.154 |
| 75 | 0.115 | 0.138 | 0.158 |
| 100 | 0.117 | 0.142 | 0.161 |
| 150 | 0.119 | 0.146 | 0.165 |
| 300 | 0.122 | 0.150 | 0.170 |
| Rack | 0.124 | 0.154 | 0.175 |

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