

# Automated food sorting machine.

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## ABSTRACT

This project deals with development of a food sorting machine. The fabricated machine employed food weights as a classification factor to sort the food in different categories. A load cell with rated capacity equal to 2 kg and 1 mv/v sensitivity is employed in measuring system and using hydraulic cylinder. The profiteered load cell output signal is amplified before being applied to the control unit in order to be easily detected by microcontroller. Peak signal to noise ratio (PSNR) criterion is employed to analyze the machine performance which aids its improvement. The PSNR results as well as experimental results specify that vibration could significantly affect machine performance. To overcome this issue, the fabricated machine divided into two parts of measuring and transferring units with separate chassis.

## INTRODUCTION

Agriculture sector plays an important role in economic development of India. As compared with development in electronic and automobile sector, development in agriculture sector is very slow. Hence, there is a need to come up with some novel techniques so as to fore front the agriculture sector again. As food plays vital role in day today's animal life, sorting of food according to different animal is necessary in evaluating agricultural produce, meeting quality standards and increasing market value. It is also helpful in planning and packaging. In India, human power in agricultural sector is widely used. If the sorting and grading of food is done through manual techniques, the process will be too slow and sometimes it will be error prone. After the harvest, animal food and grass of all types have to be sorted, packaged and transported. A wide range of technologies have been developed or refined over the years for sorting according to density, weight, quantity.

Automation is increasingly getting important in the food sorting process because computers or machines are capable of handling repetitive task quickly and effectively. Thus machines are also capable to sort food according to the grades, weights and quantity without mistakes.

In this automation system, which comprises of mechanical hydraulic structure in addition with electronics segment separately, is designed to be used in small agricultural animal purpose. There are several reasons to need this machine as a solution for agriculture industrial problems. Nowadays, usage of human power especially in agriculture sector is critically and widely use. One of these project objectives is replacing man power with machine. Usually a lot of human error occurs during the process of fruit sorting. Therefore this system is proposed to minimize or overcome those errors. Usually people can work around 7-8 hours per day. Working more than this period sometimes, makes the workers lose their focus and to concentrate on the job becomes challenging for them.

Automation systems now a day are chosen to overcome this problem and moreover the designed system produces efficient and high productive results. Usually in industries, designed machines are big in size, therefore are not portable. The aim is to design a fruit sorting machine which is portable. For this, the main task is to integrate AVR microcontroller as a main control system with entire electro-pneumatic system. ATmega16 microcontroller is used to control the sequence of operations performed by the system. This project is divided into two major sections. First is mechanical part and another one is electronics segment.

## AIM

Most of times when food arrives from the plantation to their processing units, they are found in tainted condition. They need to be washed first before processing. In this proposed system we can add an arrangement which will wash the fruits before getting to animals. In addition to this design which sorts foods on the basis of weight, we can also develop a structure in it which will sort the food on the basis of its size as well before sorting it on the basis of weight. This will increase the accuracy of sorting and hence the overall efficiency of whole sorter

## CALCULATION

### Bearing selection calculation

Isi no.	Bearing basic design no.	d	D <sub>1</sub>	D	D <sub>2</sub>	B	C(KN)	C <sub>0</sub> (KN)
20 A CO <sub>2</sub>	6202	15	21.7	35	30.5	11	7.8	3.8

$$P = XF_r + YF_a$$

Where,

P=equivalent dynamic load(N)

X=radial load constant

F<sub>r</sub>=radial load (N)

Y= axial load constant

F<sub>a</sub>=axial load(N)

In our case;

Radial load,

$$F_r = T_1 + T_2$$

$$= 124.4 + 16$$

$$F_r = 140.4 \text{ N}$$

$$F_a = 0$$

$$P = 1 * 140.4 \text{ N}$$

$$L = (C/P)^P$$

Considering 4000 working hours

Here,

n=no of rev./min.

L=length in meter

h=depth in meter

$$L = \frac{60nLh}{10^6}$$

$$L = 240 \text{ m rev.}$$

$$L = (C/P)^P$$

$$240 = (C/140.4)^3$$

$$C = 8.725 \text{ KN}$$

As, required dynamic loading is less than the rated dynamic capacity of the bearing

$$6 \text{ KN} < 12 \text{ KN}$$

Hence, bearing is safe.

## DESIGN PROCEDURE

Given data

$$P = 1/15 \text{ HP} = 50 \text{ Watt}$$

$$N_1 = 6000 \text{ RPM}$$

$$G = 5$$

$$P = \frac{2\pi NT}{60 * 1000}$$

$$\therefore T_1 = 79.6178 \text{ N-m}$$

$$(T_2/T_1) = G$$

$$(T_2/79.6178) = 5$$

$$\therefore T_2 = 398.089 \text{ N-m}$$

$$T_{\text{design}} = T_2 = 398.089 \text{ Nm}$$

## DESIGN OF INPUT SHAFT

### MATERIAL SELECTION:

DESIGNATION=40 cr1

$$UTS = 1100 \text{ MPa}$$

$$Y_t = 900 \text{ MPa}$$

As per ASME code,

$$F_s = 0.18 * UTS = 198 \text{ N/mm}^2$$

Or

$$F_s = 0.3 * Y_t = 270 \text{ N/mm}^2$$

considering the minimum of above two values:

$$\therefore F_{s_{\text{max}}} = 198 \text{ N/mm}^2$$

Reducing 25%

$$F_{s_{\text{max}}} = 148.5 \text{ N/mm}^2$$

$$T=79.6178\text{N-m}$$

$$\therefore T=79.6178*10^3 \text{ N-mm}$$

Assuming 25% overload

$$\therefore T_{\text{design}}=1.25*T=99.5222*10^3 \text{ N-mm}$$

Check for torsional shear of shaft:

Assuming  $d=16\text{mm}$

$$F_s(\text{actual})=\frac{16*T_{\text{design}}}{\pi*d^3}$$

$$F_s(\text{actual})=\frac{16*99.5222*10^3}{\pi*16^3}$$

$$F_s(\text{actual})=123.884 \text{ N/mm}^2$$

$$A_s, F_s(\text{actual})<F_{s\text{max}}$$

Hence ,design is safe.

## Calculation of gears and belts

### Design of belt drive

Motor input specification

Voltage=12V

Current=1Amp.

Step angle= $1.8^0$

RPM=30

Calculation of motor power

$$P=V*I$$

$$=12*1$$

$$P=12\text{watt}$$

### Torque

$$\text{Power}=\frac{2\pi NT}{60}$$

$$12=\frac{2\pi*30*T}{60}$$

$$T=3.81\text{Nm.}$$

### Design of pulley

Inputs specification

Diameter of larger pulley  $D=27\text{mm}$

Diameter of smaller pulley  $d=22\text{mm}$

$C=150\text{mm}$

Speed of larger pulley  $N=30\text{RPM}$

Input torque  $T_1=3.81\text{Nm}$

$$\frac{D}{d}=\frac{N}{n}$$

$$\frac{27}{22}=\frac{30}{n}$$

$$n=24.44\text{RPM}$$

output torque transmitted,

$$T_2=\frac{12*60}{2\pi*24.44}$$

$$T_2=4.6910\text{Nm}$$

Contact angle,

$$\alpha=180-2\sin^{-1}(D-d/2C)$$

$$\alpha=180-2\sin^{-1}(27-22/150)$$

$$\alpha=176.179^0$$



**design of belt:**

belt velocity

$$V = \frac{\pi ND}{60 \times 1000}$$

$$V = \frac{27 \times \pi \times 30}{60 \times 1000}$$

$$V = 0.0424 \text{ m/sec}$$

**Length of belt**

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

$$= 2 \times 150 + \frac{\pi(27+22)}{2} + \frac{(27-22)^2}{4 \times 150}$$

$$= 300 + 24.5 + 0.00833$$

$$L = 324.5083 \text{ mm}$$

**Design of shaft****Shaft:**

Shaft is a common and machine element. It is a rotating member in journal shaft is supported on bearing and it rotates a set of gears or pulleys for the purpose of power transmission. The shaft is generally acted upon by bending moment, torsion and axial force.

**Std. sizes of shaft:**

Typical size of shaft that are available in the market are,

- Upto 25mm .....0.5mm increments
- 25 to 50mm.....1mm increments
- 50 to 100mm.....2mm increments
- 100 to 200mm.....5mm increments

**Material for shaft:**

The ferrous, non-ferrous material and non-metals are used as shaft materials depending on the applications.

**Design consideration for shaft:**

The stress at any point on the shaft depends on the nature of load acting on it. The stresses may be presented as follows:

bending stresses ( $\sigma_b$ )

$$\sigma_b = \frac{32M}{\pi d_0^3 (1-k^4)}$$

$\therefore M$  = bending moment at the point of intersection

$d_0$  = outer diameter of shaft

$k$  = ratio of inner to outer diameter of shaft

( $k=0$  for solid shaft)

Axial stresses ( $\sigma_a$ )

$$\sigma_a = \frac{4\alpha F}{\pi d_0^2 (1-k^2)}$$

$\therefore F$  = axial force (Tensile or Compressive)

$\alpha$  = column action factor (=1 for tensile load)

Stresses due to torsion ( $\tau_{xy}$ )

$$\tau_{xy} = \frac{16T}{\pi d_0^3 (1-k^4)}$$

$\therefore T$  = torque on the shaft

$\tau_{xy}$  = shear stress due to torsion

combine bending and axial stress:

both bending and axial stresses are normal stresses, hence the normal stress is given by,

$$\sigma_x = \frac{32M}{\pi d_0^3(1-k^2)} \pm \frac{4\alpha F}{\pi d_0^3(1-k^4)}$$

The net normal stress can be either positive or negative. normally, shear stress due to torsion is only considered in shaft and shear stress due to load on the shaft is neglected.

Maximum shear stress theory:

Design of the shaft mostly use maximum shear stress theory. its state that a machine member fails when the maximum shear stress at a point exceed maximum allowable shear stress for the shaft material

$$\therefore \tau_{\max} = \tau_{\text{allowable}} = \sqrt{(\sigma_x/2)^2 + \tau_{xy}^2}$$

Substituting the value of  $\sigma_x$  and  $\tau_{xy}$  in the above equation, the final form is

$$\tau_{\text{allowable}} = \frac{16}{\pi d_0^3(1-k^4)} \sqrt{(M + \alpha F d_0(1+k^2)/8)^2 + T^2} \quad \dots\dots\dots (1)$$

Therefore the shaft diameter can be calculated in term of external load and material properties.

#### ASME design code :

The shaft are normally acted upon by gradual and sudden loads. Hence, the equation (1) is modified in ASME code by suitable load factors,

$$\tau_{\text{allowable}} = \frac{16}{\pi d_0^3(1-k^4)} \sqrt{(MC_{bm} + \alpha F d_0(1+k^2)/8)^2 + (C_t T)^2} \quad \dots\dots\dots (2)$$

Where,  $C_{bm}$  &  $C_t$  are bending and torsional factor

	$C_{bm}$	$C_t$
For stationary shaft:		
1.load applied gradually	1	1
2.load applied suddenly	1.5-2	1.5-2
For rotating shaft		
1.load applied gradually	1.5	1
2.load applied suddenly	1.5-2	1-1.5

ASME code foe steel purchased under definite specification :

30% of the yield strength but not over 18% of the ultimate strength in tension for shaft with keyways. This values are to be reduced by 25% for the presence of keyways.

The equation (1) & (2) are commonly used to determine shaft diameter.

#### Design of Shaft:

For a main shaft which is a power generator, power is given as,

$$P = F \times V \quad \dots\dots\dots (1)$$

Our whole assembly will have weight approximately equal to 60kilograms. Thus total force acting will be on 5 wheels. Out of those 4 wheels we have maximum load acting on rear wheels mounted on shaft. This shaft is subjected to approximately 50 kilograms of load. So force acting on shaft is given by,

$$F = m \times g \quad \dots\dots\dots (2)$$

Putting  $m=50\text{kg}$

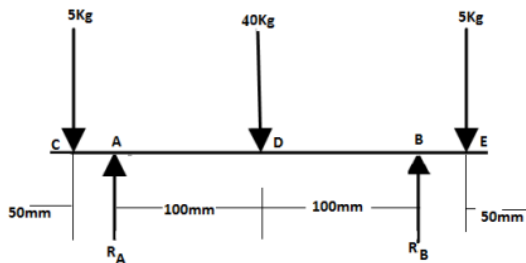


Fig. 8 Free body diagram

$$g = 9.81 \text{ m/s}^2$$

Thus

$$F = 50 \times 9.81 = 490.5 \text{ N}$$

Velocity is found out to be 10 cm/s i.e.  $V = 0.10 \text{ m/s}$

Thus

$$\text{Power, } P = 490.5 \times 0.10 = 49.05 \text{ watts}$$

$$\text{We know that torque is given as, } T = \frac{P \times 60}{2\pi n}$$

$$\text{Assuming No. of Revolution, } n = 50 \text{ rpm}$$

$$\text{Thus, we have Torque, } T = \frac{49.05 \times 60}{2\pi \times 25} = 9.36 \times 10^3 \text{ N-mm}$$

For a given shaft we have from diagram,

Vertical reactions at wheels i.e. fixed supports,

$$\begin{aligned} R_A = R_B &= (5 + 40 + 5) / 2 \\ &= 25 \text{ kg} \\ &= 25 \times 9.81 \\ &= 245.25 \text{ N} \end{aligned}$$

From bending moment diagram, maximum bending moment is found to be  $M = 1750 \text{ Kg-mm} = 17.167 \times 10^3 \text{ N-mm}$

The resultant moment on a given shaft is given as

$$\begin{aligned} MR &= (M + T)^{1/2} \\ &= ((17.167 \times 10^3)^2 + (9.36 \times 10^3)^2)^{1/2} \\ &= 19.552 \times 10^3 \text{ N-mm} \end{aligned}$$

Also we know that shaft diameter is given as,

$$d = [(MR \times 16) / (\pi \tau)]^{1/3}$$

#### SHEAR STRESS VALUES

Service Condition	$\tau_s$ (MPa)
Heavily loaded short shafts carrying no axial load	48-106
Multiple bearing long shafts carrying no axial load	13-22
Axially loaded shafts (bevel gear drive or helical gear drive)	8-10
Shafts working under heavy overloads (stone crushers, etc.)	4.5-5.3

Consider shear stress,  $\tau = 50 \text{ MPa}$

$$d = [((19.552 \times 10^3) \times 16) / (\pi \times 50)]^{1/3}$$

$$d = 12.581 \text{ mm}$$

This is ideal diameter of shaft which is needed. Since a shaft may be subjected to extra load as it has to work in rough conditions and from availability point of view, we chose a safe diameter from DDHB (Table 3.5a) of standard shaft diameter of 15 mm.

Thus diameter of shaft,  $d = 15 \text{ mm}$

#### CALCULATION OF WELDED JOINT:-

Checking the strength of the welded joints for safety

The transverse fillet weld welds the side plate and the edge stiffness plates,

The maximum load which the plate can carry for transverse fillet weld is

$$P = 0.707 \times S \times L \times t$$

Where,

S = factor of safety,

L = contact length = 25mm

The load of shear along with the friction is 50 kg = 500N

Hence,  $500 = 0.707 \times 3 \times 35 \times ft$

Hence let us find the safe value of 'ft'

$$\text{Therefore } ft = \frac{500}{0.707 \times 3 \times 35}$$

$$ft = 6.73536 \text{ N/mm}^2$$

Since the calculated value of the tensile load is very smaller than The permissible value as  $ft=56 \text{ N/mm}^2$ . Hence welded joint is safe.

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#### REFERENCES

- [1] S. M. Metev and V. P. Veiko, *Laser Assisted Microtechnology*, 2nd ed., R. M. Osgood, Jr., Ed. Berlin, Germany: Springer-Verlag, 1998.
- [2] J. Breckling, Ed., *The Analysis of Directional Time Series: Applications to Wind Speed and Direction*, ser. Lecture Notes in Statistics. Berlin, Germany: Springer, 1989, vol. 61.
- [3] S. Zhang, C. Zhu, J. K. O. Sin, and P. K. T. Mok, "A novel ultrathin elevated channel low-temperature poly-Si TFT," *IEEE Electron Device Lett.*, vol. 20, pp. 569–571, Nov. 1999.
- [4] *Wireless LAN Medium Access Control (MAC) and Physical Layer (PHY) Specification*, IEEE Std. 802.11, 1997.