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MODIFICATION OF AUTOMATIC POUCH FILLING MACHINE

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Abstract: The project is to make some modifications in automatic pouch filling machine which is mainly used for filling and the packing of milk pouches. The project is sponsored by Qualitronics industry. The machine was operated by electric power and it was fully automatic this saves a lot of time and cost as it is automatic machine. But, the main problem lies in the rural areas where the electric power supply is not available consistently . Also, due to the frequent power cuts and fluctuating supply of electricity , it becomes very difficult to operate the machine in the rural areas and to achieve desired production.

The main purpose of modification is to make the machine manually operated , so that the operator can continue his work without any interruption caused by power failure.

In this project, we narrowed our focus to make the, horizontal sealing mechanism manually operated as it consumes maximum amount of power. We designed a suitable mechanism which consists of pulley, gears, and cam operated correspondingly with hand lever. The design is specified and by theoretical calculations it was found to be correct. The mechanism that we designed has higher F.O.S and also it is small and compact with the minimum effort applied at handle the mechanical advantage is gained at the gears , pulleys and we get required high torque at the Cam and follower.

I. INTRODUCTION

In today's world of artificial intelligence and automation almost each and everything is controlled by a set of programmes designed for the regarding purpose. Humans are so much surrounded by technology that it has become an unavoidable part of even daily routine and can't be neglected.

But even technology need some energy tool to perform the task allotted to it. Whereas in some part due to some unfavourable conditions continues flow of energy becomes difficult to be transferred. Thus there are comes discontinuities in the regarding work. When this energy is in the form of electricity the problem is comes to remote/ruler areas where there is continues supply of electricity is not available. This leads loss of production and also have great impact in various fields of industry such as financial, sales and so on. This makes the industry difficult to survive that industry in the race of market where other industries can defeat such small scale industries easily.

Hence, to not only survive but also to have a consistent stability in the market as well as safe future of the industry. It is necessary to have a solution for every problem and cope up with other industries. Hence in this project we find an alternative solution for such problem.

The project is mainly focused on to perform the function of milk pouch packaging machine manually. The currently available milk pouch packaging machines are fully automatic and are electrically operated. These machines mainly consists assemblies of -1) Horizontal and vertical pouch sealing system. 2) Milk dosing system. 3) Roll folding system. 4) Cam operated slide. 5) Machine structure. Also, the machine consists various components such as pulleys, gears, cam, springs, sensors which are of mild steel and stainless steel are fixed in machine frame which is made up of M.S Cast steel. In this machine motor is fitted with small pulley on its shaft. The smaller pulley is connected to large pulley by means of open belt drive. The ratio of pulley drive is 1:2. The larger pulley is connected input of gearbox shaft having ratio 1:30. The gearbox is also connected to clutch box by chain drive. The cam is mounted on the output of the gearbox shaft which is supported in the pillow block. When machine is energized the motor starts rotating and thus pulleys connected to motor by belt drive also starts rotating due to which gearbox is operated. The gearbox drives the shaft on which cam is mounted. The cam starts start rotating, and at the extreme position of cam roller is moved forward in order to move the slide. The slides operate the drive through the cam and cutting action is done. In our mechanism the motor is replaced by handle as to make the machine manually operated.

II. CALCULATIONS

➢ 2.1 design of shafts

Material for Shafts: -

The material used for ordinary shafts is carbon steel of grades 40 C 8, 45 C 8, 50 C 4 and 50 C 12. The mechanical properties of these grades of carbon steel are given in the following table.

Indian standard designation	Ultimate tensile strength, MPa	Yield strength, MPa
40 C 8	560 - 670	320
45 C 8	610 - 700	350
50 C 4	640 - 760	370
50 C 12	700 Min.	390

From above table we have selected following material for our designed mechanism: -

45 C 8 (Carbon Steel)

the shaft used in our designed mechanism have following stresses: -

$\sigma_t = \sigma_c = 0.36 \times 610 = 219.6$ N/mm^2 OR $\sigma_t = \sigma_c = 0.6 \times 350 = 210$ N/mm^2

therefore,

$\sigma_t = \sigma_c = 0.6 \times 350 = 210$ N/mm^2 is taken because it is less.

Also,

$\tau = 0.18 \times 610 = 109.8$ N/mm^2 OR $\tau = 0.3 \times 350 = 105$ N/mm^2

therefore,

$\tau = 0.3 \times 350 = 105$ N/mm^2 is taken because it is less.

- **2.1.1 Design of Shaft: - 1**

This shaft consists of a small pulley (A) as shown in figure. So before designing a shaft, let's find out the tension in belts (T_1 & T_2) of pully i.e. load acting on shaft by pully (W).

We know that

$$\frac{T1}{T2} = e^{\mu\theta} \quad \ldots\ldots\ldots (1)$$

Let's find θ,

$$\text{Sin } \alpha = \frac{r1 + r2}{x} = \frac{75 - 37.5}{196.78} \quad \alpha = 10.99^o$$

Therefore,

$$\theta = [180 - (2 \times 10.99)] \times \frac{\pi}{180} = 2.758 \text{ radians} \quad \ldots\ldots\ldots\ldots\ldots (\theta = 158.02^0)$$

Now putting above values in equation 1, we get

$$\frac{T1}{T2} = e^{0.3x2.758} = 2.287 \quad T_1 = 2.287T_2$$

Torque transmitted on driver pulley: -

$T = (T_1 - T_2) \times r_1 = 473.87$ N

Therefore,

Load acting on 1st shaft (W) = $T_1 + T_2 = 681.07$ N Take it (W) = 700 N

Bending Moment of Shaft: -

By calculation,

$M_C = 6885$ N.mm = 6.9 Nm.

$M_D = -2607$ N.mm = -2.61 Nm. …. (-ve sign indicates hogging)

We know that,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{6.9^2 + 10^2} = 12.149 \text{ Nm.}$$

We also know that,

$$T_e = \frac{\pi}{16} \times \tau \times d^3 \quad 12149.48 = \frac{\pi}{16} \times 105 \times d^3 \quad d = 8.38 \text{ mm} = 10 \text{ mm}$$

➢ 2.2.2 Design of Shaft: - 2

Load acting on shaft due to pulley (W) = 700 N

Let's find out the load acting on shaft due to gear,

Pressure angle of gear = 20^0 ……. (standard)

$$\text{Tangential force } (F_t) = \frac{2T}{D} = \frac{2 X 20 X 10^3}{40} = 1000 \text{ N}$$

$$\text{Radial Load (W)} = \frac{F_t}{Cos\alpha} = \frac{1000}{Cos20^0} = 1064.18 \text{ N}$$

Bending moment of shaft: -

by calculation

$R_B = 958.86$ N

$R_A = 1764.18 - 958.86 = 805.32$ N

For bending moment,

$M_A = M_B = 0$ &

By calculation,

$M_C = 8687.7$ N.mm = 8.688 Nm.

$M_D = -(1.2 \times 10^{-3}) = 1.2 \times 10^{-}$

We know that,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{8.688^2 + 20^2} = 21.805 \text{ Nm.}$$

We also know that,

$T_e = \frac{\pi}{16}$ x τ x d^3 21805 = $\frac{\pi}{16}$ x 105 x d^3 d = 12 mm

2.3.3 Design of shaft: 3

Tangential force acting on gear (F_t) = 1000N
Radial force acting on gear (W) = 1064.18 N
Radial force acting on cam (W) = 951.19 N
Axial force acting on cam = 0 N
Bending moment of shaft 3,
By calculations,
R_B = 807.33 N
R_A = 1208.04 N
For bending moment
M_C = -0.1 N.mm (hogging)
M_D = -0.1 N.mm (hogging)
Now,

$T_e = \sqrt{M^2 + T^2} = \sqrt{0.1^2 + (50\,x\,10\text{^}3)^2}$ = 50000 N.mm

We also know that,

$T_e = \frac{\pi}{16}$ x τ x d^3 50000= $\frac{\pi}{16}$ x 105 x d^3 d = 13.435 m d = 14 mm

Taking diameter of all 3 shafts = 20 mm which is suitable for our design having material 45c8 (carbon steel) and permissible tensile stress 210 N/mm^2.

2.2 design of hand liver

- Material: **Wrought Iron**
- Permissible shear stress **(τ) = 60 N/mm^2**
- Permissible tensile stress **(σ_t) = 70 N/m**
- Diameter of shaft **(d) = 20mm**

1. Design of shaft

We know that,

T = $\frac{\pi}{16}$ x τ x d^3 10 x 10^3 = $\frac{\pi}{16}$ x τ x 20^3 **τ = 6.63 N/mm^2**

It is less than permissible shear stress hence, **Design is safe.**

2. Diameter of Boss

By assumption, We have, d_1 = 1.6d = 1.6 x 20 **d_1 = 32mm**
Thickness of boss (t_2) = 0.3d = 0.3 x 20 **t_2 = 6mm**

3. Length of boss (l_2) = 1.25d **= 25mm**

Now check it for tensile stress,

P x L = l_2 x t_2 x σ_t ($\frac{d+t2}{2}$) 40 x 250 = 25 x 6 x σ_t ($\frac{20+6}{2}$) σ_t = **5.13 N/mm^2**

It is less than permissible tensile stress hence, **Design is safe.**

4. Design of key

Material: **45 C 8** (Similar to Shaft Material)

σ_{ult} = 700N/mm^2
σ_{el} = 350 N/mm^2
σ_c = 0.36 x 700 = 252 N/mm^2
σ_c = 0.36 x 350 = 210 N/mm2
hence **σ_c** is taken as **210 N/mm^2** (it is less)
τ = 0.3 x 350 = 105 N/mm^2
τ = 0.18 x 700 = 126 N/ mm^2
hence **τ** is taken **105 N/mm^2** (it is less)
From Table,
For shaft of 20mm diameter
we find out the width and thickness of key,
Width of key (w)= 8mm, Thickness of key (t) = 7mm
The length of key is obtained by considering the key in shearing and crushing. Let, **L = length of the key**
Now considering length of key, **T = l x w x τ x $\frac{d}{2}$** 10 x 10^3 = l x 8 x 105 x $\frac{20}{2}$ l = 11.9 mm , Taken as **12 mm**
Checking it for Crushing

T = l x$\frac{t}{2}$ x σ_c x$\frac{d}{2}$ 10 x 10^3 = 12 x $\frac{7}{2}$ x σ_c x $\frac{20}{2}$ **σ_c = 23.80 N/mm^2**

It is less than permissible crushing stress; hence, **design is safe.**

5. Cross section of lever near the boss

It is done by considering the bending of lever
t = 6 mm B = 5t = 30mm
We know that bending moment on the lever
M = P x L = 40 x 250 = **10000 N.mm**
Section modulus **(z) = $\frac{1}{6}$ x t x B^2 = 900 mm^3**

$\sigma_b = \frac{M}{Z} = \frac{10000}{900}$ = **11.11 N.mm^2** hence, **design is safe.**

- **2.3 Selection of pulley and belt**

The V belts are probably the most common means of transmitting power between fractional horse power motors to machines. Mostly, the driver and driven pulleys lies in the same vertical plane. There is an upper limit on the centre distance or belt length. Long centre distances are not recommended because the excessive vibrations of slack side flutters and shortens the belt life. In general, the centre distance should not be greater than 3 times the sum of diameters of input and output pulleys. Since the v belt is short, it is subjected to action of load and fatigue a greater number of times, further, its ability in absorbing shocks is poor.

The advantage with belt drives is that they reduce vibrations and shock transmission, since the belts are elastic and usually quite long. These properties play an important part in absorbing shock loads and isolating the effects of vibrations. This aspect is very important for the life of machine.

Selection of pulley,

Diameter of pulley A **(d_1) = 75 mm** (smaller pulley)
Diameter of pulley B **(d_2) = 150 mm** (bigger pulley)
Type of V- Belt = **A**
Length of belt **(L)= 747mm** (standard value)
We know that,

$$\mathbf{L = \frac{\pi}{2} (d_1 + d_2) + (2x) + [\frac{d_1 - d_2}{4x}]^2}$$

By putting values in above equation,

$747 = \frac{\pi}{2} (75 + 150) + (2x) + [\frac{75 - 150}{4x}]^2$

$747 = 353.43 + (2x) + [\frac{5625}{16x^2}]$

$393.57 = 2x + \frac{351.56}{x^2}$

$393.57x^2 = 2x^3 = 351.56$

$2x^3 – 393.57x^2 + 351.56 = 0$

By equating above equation, we get

x = 196.78mm

As per the above information, the centre distance should not be greater than 3 times the sum of diameters of input (**d_1**) and output pulley (**d_2**).

Let's check it,

sum of diameters = 3 (d_1 + d_2)
= 3 (150 +75) = 3 x 225 **= 675mm**

196.78mm < 675mm

Hence distance between centres of two pulleys is lies within limit.

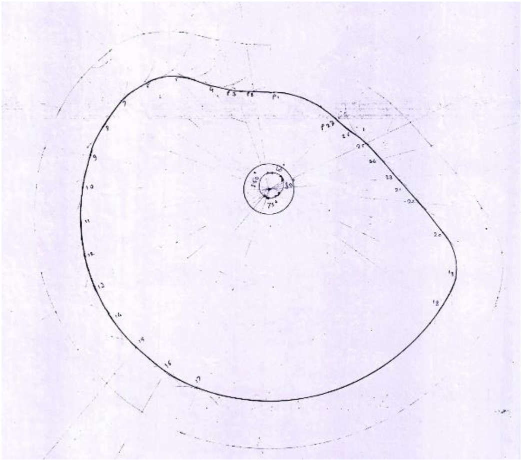
Table 20.1. Dimensions of standard V-belts according to IS : 2494–1974.

Type of belt	Power ranges in kW	Minimum pitch diameter of pulley (D) mm	Top width (b) mm	Thickness (t) mm	Weight per metre length in newton
A	0.7 – 3.5	75	13	8	1.06
B	2 – 15	125	17	11	1.89
C	7.5 – 75	200	22	14	3.43
D	20 – 150	355	32	19	5.96
E	30 – 350	500	38	23	–

Table 20.2. Dimensions of standard V-grooved pulleys according to IS : 2494–1974.
(All dimensions in mm)

Type of belt	w	d	a	c	f	e	No. of sheave grooves (n)	Groove angle (2β) in degrees
A	11	12	3.3	8.7	10	15	6	32, 34, 38
B	14	15	4.2	10.8	12.5	19	9	32, 34, 38
C	19	20	5.7	14.3	17	25.5	14	34, 36, 38
D	27	28	8.1	19.9	24	37	14	34, 36, 38
E	32	33	9.6	23.4	29	44.5	20	–

- **2.4 design of cam**



Our designed cam profile

Rise:-

Point No	Distance O2 - 24	y	α
0-1	15.5	1.5	11.60
1-2	51.5	6	32.77
2-3	51	11.2	30.90
3-4	41	11.1	25.72
4-5	33.5	6.4	22.62
5-6	18	4.3	12.95
6-7	21	3	15.25
7-8	10.5	2.7	7.795
8-9	14	2.3	10.39
9-10	13.5	2.8	10
10-11	14.5	2.5	10.73
11-12	12.5	1.8	9.36
12-13	8	1.5	6.05
13-14	5.5	1.1	4.19
14-15	7.5	1.3	5.69
15-16	3.5	0.8	2.68

Dwell:-

Point No	Distance O2 - 24	y	α
0-1	16	1.3	12
1-2	51.5	6	32.74
2-3	74	14.7	39.84
3-4	58	13.5	33.54
4-5	38	10.3	24.26
5-6	24.5	6.4	16.95
6-7	22	4.4	15.67
7-8	9	3.4	6.63

- **2.4.1 Inertia Forces Acting on Horizontal Jaw During Its Movement**

An inertia force is a force that resist a change in velocity of an object. It is equal to - and in the opposite direction of - an applied force, as well as a resistive force.

The concept is based on Newtons laws of motion, including the law of inertia and the action-reaction law, which states that,

I. Every object in a state of uniform motion tends to remain in that state of motion unless an external force is applied to it (Law of inertia)
II. The relationship between an objects mass **m**, its acceleration **a**, and the applied force **F**, is: **F = ma**
III. If a force is applied to an object, there is an equal and opposite reaction (Action-Reaction Law)

That equal and opposite reaction is called the inertia force. It is equal to **-F = ma**

2.4.1.1 Calculation of velocity,

The velocity of an object is the rate of change of its position with respect to a frame of reference, and is a function of time.

Velocity is a physical vector quantity; both magnitude and direction are needed to define it.

We know that,

$$\textbf{velocity}=\frac{\textit{Distance}}{\textit{Time}}$$

where, **Distance** = Vertical distance between two consecutive points on cam profile displacement diagram

Time = time taken by cam per 10-degree revolution. (**0.2084sec)**

Velocity of cam during rise per 10 degree

Example, For point 0-1:

$V_{0-1} = \frac{d}{t} = \frac{1.5}{0.2084}$ **V_{0-1} = 7.19mm/sec**

The table shows different velocities of sliding jaw at various points of contact per 10^0 on cam profile during forward movement of cam: -

0-1 = 7.19 mm/sec	8-9 = 11.036 mm/sec
1-2 = 28.79 mm/sec	9-10 = 11.996 mm/sec
2-3 = 53.74 mm/sec	10-11 = 11.996 mm/sec
3-4 = 53.26 mm/sec	11-12 = 8.637 mm/sec
4-5 = 30.71 mm/sec	12-13 = 7.198 mm/sec
5-6 = 20.63 mm/sec	13-14 = 5.278 mm/sec
6-7 = 14.39 mm/sec	14=15 = 6.238 mm/sec
7-8 = 12.96 mm/sec	15-16 = 3.838 mm/sec

The table shows different velocities of sliding jaw at various points of contact per 10^0 on cam profile during return movement of cam: -

0-1 = 6.238 mm/sec	4-5 = 49.42 mm/sec
1-2 = 28.79 mm/sec	5-6 = 30.71 mm/sec
2-3 = 70.537 mm/sec	6-7 = 21.11 mm/sec
3-4 = 64.78 mm/sec	7-8 = 16.31 mm/sec

2.4.1.2 Linear Acceleration of Cam

- Accelerations are vector quantities (they have magnitude and direction) and add according to the parallelogram law.

Calculations of linear acceleration: -

Acceleration = $\frac{Velocity}{Time}$

Where, **Velocity =** Velocity of sliding jaw.

Time = Time taken by cam per 10-degree revolution. (**0.2084sec)**

Acceleration of cam during rise per 10 degree

Example, For point 0-1:

$A_{0-1} = \frac{V}{T} = \frac{7.19 \, X \, 10^{-3}}{0.2084}$ **A_{0-1} = 0.0345 m/sec^2**

The table shows different velocities of sliding jaw at various points of contact per 10^0 on cam profile during forward movement of cam: -

0-1 = 0.0345 m/sec^2	8-9 = 0.00923 m/sec^2
1-2 = 0.10076 m/sec^2	9-10 = 0.00460 m/sec^2
2-3 = 0.11972 m/sec^2	10-11 = 0.00460 m/sec^2
3-4 = 0.00230 m/sec^2	11-12 = 0.01612 m/sec^2
4-5 = 0.10820 m/sec^2	12-13 = 0.00690 m/sec^2
5-6 = 0.04836 m/sec^2	13-14 = 0.00921 m/sec^2
6-7 = 0.02994 m/sec^2	14-15 = 0.00485 m/sec^2
7-8 = 0.00686 m/sec^2	15-16 = 0.01156 m/sec^2

The table shows different acceleration of sliding jaw at various points of contact per 10^0 on cam profile during return movement of cam: -

0-1 = 0.02994 m/sec^2	4-5 = 0.07370 m/sec^2
1-2 = 0.10820 m/sec^2	5-6 = 0.08978 m/sec^2
2-3 = 0.20033 m/sec^{2v}	6-7 = 0.04606 m/sec^2
3-4 = 0.02764 m/sec^2	7-8 = 0.02303 m/sec^2

Inertia force calculations: -

Inertia force = mass x acceleration

Mass= total mass of roller follower and sliding jaw **(3.25 kg)**

Total inertia force acting on follower and sliding jaw during rise per 10 degree

Example

Inertia force (F) = m x a

For point 0-1:

Inertia force (F) = 3.25 x 0.0345

= **0.1121 N**

The table shows different inertia forces exerted by sliding jaw at various points of contact per 10^0 on cam profile during forward movement of cam: -

0-1 = 0.1121 N	8-9 = 0.030 N
1-2 = 0.3275 N	9-10 = 0.01495 N
2-3 = 0.3891 N	10-11 = 0.01495 N
3-4 = 0.00748 N	11-12 = 0.05240 N
4-5 = 0.352 N	12-13 = 0.02242 N
5-6 = 0.1572 N	13-14 = 0.02994 N
6-7 = 0.09731 N	14-15 = 0.01576 N
7-8 = 0.02290 N	15-16 = 0.03757 N

The table shows different inertia forces exerted by sliding jaw at various points of contact per 10^0 on cam profile during return movement of cam: -

0-1=0.09731 N	4-5=0.23853 N
1-2=0.35165 N	5-6=0.29179 N
2-3=0.65107 N	6-7=0.14970 N
3-4=0.08983 N	7-8=0.07485 N

➢ 2.4.2 Normal and tangential forces acting on cam

For normal force (F^y)= F_n cos α

For tangential force (F_t) = F_n sin α

Where, F_n = summation of all forces acting on cam.

Normal force is summation of inertia force, spring force and friction force.

- **Force acting on the cam during rise = F_{st} + F_i + F_f**

Where, F_{st} = force acting on cam due to tension spring

F_i = inertia force

F_f = friction force {10 % of (F_{st} + F_i)}

Therefore, Force acting on the cam during rise = 39.58 + 0.66 + 4.02 **Rise force = 44.26 N**

- **Force acting on cam during dwell = F_{st} + F_i + F_f + F_{sc}**

Where , F_{sc} = force acting on cam due to compression spring

F_f = friction force {10 % of (F_{st} + F_i + F_{sc})}

Therefore, Force acting on the cam during dwell = 39.58 + 0.66 + 86.47 + 824.5 **Dwell force = 951.21 N**

- **Force acting on the cam during return = - (F_{ts} + F_i + F_f)**

Therefore, Force acting on the cam during return = - (39.58 + 0.66 + 4.02) **Return force** = **- 44.26 N**

Force acting on the cam during return is taken negative because during return instead of applying force on follower by cam here follower apply the force on cam due to retraction of spring.

The table shows different normal and tangential forces acting at various points of contact per 10^0 on cam profile during forward movement of cam: -

0	F cos 0 = 44.26 N	F sin 0 = 0 N
0-1	F cos 11.60 = 43.36	F sin 11.60 = 8.90
1-2	44.26 cos 32.77 = 32.72 N	44.26 sin 32.77 = 23.96 N
2-3	44.26 cos 30.90 = 37.98 N	44.26 sin 30.90 =22.73 N
3-4	44.26 cos 25.72 = 39.88 N	44.26 sin 25.72 = 19.21 N
4-5	44.26 cos 22.62 = 40.86 N	44.26 sin 22.62 = 17.03 N
5-6	44.26 cos 12.95 = 43.14 N	44.26 sin 12.95 = 9.92 N
6-7	44.26 cos 15.25 = 42.7 N	44.26 sin 15.25 = 9.92 N
7-8	44.26 cos 7.795 = 43.85 N	44.26 sin 7.795 = 6 N
8-9	44.26 cos 10.39 = 43.54 N	44.26 sin 10.39 =7.98 N
9-10	44.26 cos 10 = 43.59 N	44.26 sin 10 =7.69 N
10-11	44.26 cos 10.73 = 43.49 N	44.26 sin 10.73 =8.24 N
11-12	44.26 cos 9.36 = 43.68 N	44.26 sin 9.36 = 7.20 N
12-13	44.26 cos 6.05 = 44.02 N	44.26 sin 6.05 = 4.67 N
13-14	951.21 cos 4.19 = 948.47 N	951.21 sin 4.19 =69.50 N
14-15	951.21 cos 5.69 = 946.53 N	951.21 sin 5.69 = 94.31N
15-16	951.21 cos 2.68 = 950.17 N	951.21 cos 2.68 = 44.48 N

The table shows different normal and tangential forces acting at various points of contact per 10^0 on cam profile during return movement of cam: -

0	951.21 cos 0 = 951.21 N	951.21 sin 0 = 0
0-1	951.21 cos12 = 930.73 N	951.21 sin 12 = 197.77 N
1-2	44.26 cos 32.77 = 37.22 N	44.26 sin 32.77 = 23.96 N
2-3	44.26 cos 39.84 = 33.99 N	44.26 sin 39.84 = 28.36 N
3-4	44.26 cos 33.54 = 36.89 N	44.26 sin 33.54 = 24.46 N
4-5	44.26 cos 24.26 = 40.35 N	44.26 sin 24.26 = 18.19 N
5-6	44.26 cos 16.95 = 42.34 N	44.26 sin 16.95 = 12.90 N
6-7	44.26 cos 15.67 = 42.62 N	44.26 sin 15.67 = 11.96 N
7-8	44.26 cos 6.63 = 43.97 N	44.26 sin 6.63 = 5.11 N

➢ 2.4.3 Torque Calculations

$T = f^y_{32}$ (y + Rr + Rc) tan α

Where, y = vertical distance between two consecutives on displacement diagram. (per 10 degree)

R_r = radius of roller

R_c = radius of cam

Torque of cam during rise per 10 degree

Example, For point T_{0-1}

T_{0-1} = 43.36 (1.5 + 24 + 50) tan (11.60) **T_{0-1} = 671.99 N.mm**

The table shows different torque acting at various points of contact per 10^0 on cam profile during forward movement of cam: -

0-1 = 671.99 N.mm	8-9 = 609.12 N.mm
1-2 = 1916.73 N.mm	9-10 = 587.99 N.mm
2-3 = 1936.65 N.mm	10-11 = 630.44 N.mm
3-4 = 1634.78 N.mm	11-12 = 545.75 N.mm
4-5 = 1368.82 N.mm	12-13 = 352.25 N.mm
5-6 = 776.74 N.mm	13-14 = 5218.32 N.mm
6-7 = 896.38 N.mm	14-15 = 7101.50 N.mm
7-8 = 460.42 N.mm	15-16 = 3326.85 N.mm

The table shows different torque acting at various points of contact per 10^0 on cam profile during return movement of cam: -

0-1 = 14896.81 N.mm	4-5 = 1532.98 N.mm
1-2 = 1916.73 N.mm	5-6 = 1037.50 N.mm
2-3 = 2515.50 N.mm	6-7 = 937.34 N.mm
3-4 = 1895.19 N.mm	7-8 = 395.58 N.mm

➢ 2.5 SELECTION OF GEAR

Pre-set values: -

- Module **(m) = 3**
- Pinion gear diameter **(d_1) = 54 mm**
- No. of teeth **(Z_1) = 18**
- Main gear diameter **(d_2) = 135mm**
- No. of teeth **(Z_2) = 45**
- Pressure angle **α = 20^0** ……………. (standard)

Let's find out the other parameters of gear,

1. **Centre Distance (a)** = $\frac{(Z1+Z2)m}{2} = \frac{(18+45)\,3}{2} = \frac{189}{2}$ = **94.5 mm**
2. **Reference Diameter (d)** = Zm

For pinion = Z_1m = 18 x 3 = **54 mm**

For main gear = Z_2m = 45 x 3 = **135 mm**

3. **Base Diameter (d_b)** = d cos α

For pinion = 54cos 20^0 = **50.74 mm**

For main gear = 135 cos 20^0 = **126.86 mm**

4. **Addendum (h_a)** = 1.00 x m= 1.00 x 3 = **3 mm**
5. **Dedendum** = 1.25 x m = 1.25 x 3 = **3.75 mm**
6. **Tooth depth (h)** = 2.25 x m = 2.25 x 3 = **6.75 mm**
7. **Tip Diameter (d_a)** = d + 2m

For pinion = 54 + (2 x 3) = **60 mm**

For Main gear = 135 + (2 x 3) = **141 mm**

8. **Root Diameter (d_1)** = d – 2.5m

For pinion = 54 – (2.5 x 3) = **46.5 mm**

For main gear = 135 – (2.5 x 3) = **127.5 mm**

9. **Tooth thickness (t)** = 1.5708 x m = 1.5708 x 3 = **4.7124 mm**

➢ 2.6 Springs Used in Machine

1) Jaw supporting springs: -

- Material used for spring is **spring steel.**
- Free length = 80mm
- Initial deflection (x) = 74.5mm ……. (-5.5mm)
- Final compressed length = 71.5mm……. (-3mm)
- Number of active turns = 8
- Total coil turns =10
- Spring wire diameter = 6mm
- Outer diameter of coil = 32mm
- inner diameter of coil = 20mm

Calculations related to jaw supporting springs (compression spring): -

Mean diameter = (OD + ID)/2 = (32+20)/2 = **26mm**

$C = \frac{\text{mean diameter}}{\text{wire diameter}}$ = 26/6 **C = 4.33 mm**

Now, Stiffness (K) = $\frac{Gd}{8c^3 n_a}$

Here, Number of active coils are = 8

G for spring steel is 84 x 10^3 N/mm^2

Stiffness (K) = 84 x 10^3 x 6 / 8 x 4.33^3x 8 **K = 97 N/mm**

But if we add initial and final deflection then the value of (x)will be 5.5+3=8.5mm

Therefore,

F_j= k X (x) =97 X 8.5 **F_j = 824.5 N**

2) follower supporting spring:

- material used for spring is **spring steel.**
- Initial length=212mm
- Maximum length=265mm
- Number of active turns=108
- Number of inactive turns=4 (2 on both sides)
- Outside diameter of coil=16mm
- Inside diameter of coil=12mm
- Wire diameter=2mm

Calculation related to follower supporting spring (tension spring): -

Mean diameter r= (OD + ID)/2 = (16 + 12)/ 2 = **14mm**

C = mean diameter/ wire diameter = 14/2 = **7mm**

Now, Stiffness (K)= $\frac{Gd}{8c^3 n_a}$

Here, Number of active coils are=108

G for spring steel is 84 x 10^3 N/mm^2.

Stiffness (K) = 84 x 10^3 x 2 / 8 x 7^3x 108 **K = 0.56689 N/mm**

But if we add initial and final deflection then the value of (x)will be

X = 53 + 17 = 70 mm

Therefore, F = K x (X) = 0.56689 X 70 = **39.68 N**

➢ 2.7 BEARING SELECTION

Radial Ball Bearings are selected for each shaft.

2.7.1 Bering for shaft 1

Radial load acing on bearing (W_R) = 700 N

Axial load acing on bearing (W_A) = 0

Basic dynamic equivalent radial load acting on bearing (W)

= X.V. WR + Y. WA = 1x1x700 + 0x0 = 700 N

Basic dynamic capacity of bearing (C)

$= W\left(\frac{L}{10^6}\right)^{1/k} = 700\left(\frac{18 X 10^6}{10^6}\right)^{1/3}$ C = 23.27 N= 0.023 KN

2.7.2 Bearing for shaft 2

for pulley

load acting on bearing (W_R) = 700 N

as force is acting at an angle of 45 hence we will resolve this 700N

so, we get 2 components W_R = 494.97 N W_T = 494.97 N

for gear

Tangential load acting on bearing ($W_{t.}$) = $\frac{2T}{D}$ = 1000N

Radial load acting on bearing (W_R) = 1064.18 N

Now resultant force acting on bearing,

$R_A = \sqrt{494.97^2 + 1064.18^2}$ = 1173.66 N

$R_B = \sqrt{494.97^2 + 1000^2}$ = 1115.79 N

W_R = 1173.66 N (maximum load)

Hence equivalent dynamic radial load

W = X.V. WR + Y. WA = 1x1x1173.66 + (0x0) = 1173.66 N

Taking FOS = 2

Total load = 2347.32 N

As bearings are on both side the force will be divided equally among them,

Load for each bearing = $\frac{2347.32}{2}$ = 1173.66 N

Hence dynamic capacity of bearing

C = 1173.66 x $(\frac{18 X 10^6}{10^6})^{1/3}$ = 27.64 N = 0.02764 KN

2.7.3 Bearing for shaft 3

For cam

Load acting on bearing = 951.21N

as maximum load is acting at an angle of 0.4 hence we will resolve this 951.19N

so, we get 2 components W_R = 951.19 N W_A = 6.64 N

For gear

The power is transmitted from driving gear is at 20 degree

Hence, by resolving these forces we get 2 components

W_R = 1064.18 N W_T = 1000 N

M_A = 0

= (1064.16 x 80) + (951.19 x 250) – (R_{BV} x 400)

By calculation we get

R_{BV} = 807.33 N

R_{AV} = 1207.67 N

R_{AH} = 1000 N

R_{BH} = 0 N

Resultant forces

$R_A = \sqrt{1207.67^2 + 1000^2}$ = 1567.95 N

$R_B = \sqrt{807.33^2 + 0^2}$ = 807.33 N

Radial force acting on bearing (W_R) = 1567.95 N (maximum force)

Equivalent dynamic radial load

W = X.V. WR + Y. WA = 1x1x1567.95 + 0 x 0 ………. (By neglecting axial load) = 1567.95 N

Taking FOS = 2

Total load = 3135.9 N

As bearings are on both side the force will be divided equally among them,

Load for each bearing = $\frac{3135.9}{2}$ = 1567.95 N

Hence dynamic capacity of bearing

C = 1567.95x $(\frac{18 X 10^6}{10^6})^{1/3}$ = 30.63 N = 0.03063 KN

From above calculations we conclude that bearing **NU 202 ECP Is suitable for our design**.

►

NU 202 ECP

Dimensions

d		15	mm
D		35	mm
B		11	mm
D_1	≈	27.7	mm
F		19.3	mm
$r_{1,2}$	min.	0.6	mm
$r_{3,4}$	min.	0.3	mm
s	max.	1	mm

Abutment dimensions

d_a	min.	17.4	mm
d_a	max.	18.4	mm
$d_{b.}$	min.	21	mm
D_a	max.	31.3	mm
r_a	max.	0.6	mm
r_b	max.	0.3	mm

Calculation data

Basic dynamic load rating	C		6.10	kN
Basic static load rating	C_0		3.55	kN
Fatigue load limit	P_u		1.22	kN
Reference speed			22000	r/min
Limiting speed			26000	r/min
Calculation factor	k_r		0.15	
Limiting value	e		0.2	
Axial load factor	Y		0	

From table of "principle dimensions for radial ball bearing"
We selected following details:

- Bore diameter of bearing = 15mm
- Outside diameter of bearing = 35 mm
- Width of bearing = 11mm
- Bearing number = 202

Now from the table of "basic static and dynamic capacities of various types of radial ball bearings"
We selected following details for bearing number 202 (**single row deep groove ball bearing)**

- Dynamic capacity (C) = 6.10 KN
- Static capacity (C_o) = 3.55 KN
- Life (L) = 30000 hrs (from table)
- Reliability = 90 %

L_{90} = 60 x L x N = 60 x 30000 x 40 = 72 x 10^6 revolutions

Dynamic capacity of bearing by calculations:

$L = (C/W)^k \times 10^6$ rev.

$C = W (L/10^6)^{1/k}$

Where,

C = basic dynamic capacity of ball bearing in km.

L = rating life in millions of revolutions

W = equivalent dynamic load

K = 3, for ball bearing

$C = 951.21 \times [(72 \times 10^6) / 10^6]^{1/3}$ C = 3.957 KN

From calculation the dynamic capacity of ball bearing is 3.957 and this is less than standard dynamic capacity of ball bearing from the table.

Hence,

The ball bearing, we have selected is suitable for our application.

III. FUTURE SCOPE

The mechanism that we designed is limited only for operation of horizontal sealing. By further modification we can make the machine fully manually operated. Also, if the hydraulic system is used instead of mechanical system great amount of mechanical advantage can be achieved which can provide power required to operate whole machine.

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