REGENERATIVE BRAKING SYSTEM

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Abstract — As in today’s world, where there are energy crises and the resources are depleting at a higher rate, there is a need of specific technology that recovers the energy, which gets usually wasted. So, in case of automobiles one of these useful technology is the regenerative braking system. Generally in automobiles whenever the brakes are applied the vehicle comes to a halt and the kinetic energy gets wasted due to friction in the form of kinetic energy. Using regenerative braking system in automobiles enables us to recover the kinetic energy of the vehicle to some extent that is lost during the braking process. In this paper the author discusses two methods of utilising the kinetic energy that is usually wasted by converting it into either electrical energy or into mechanical energy. Regenerative braking system can convert the kinetic energy into electrical energy with help of electric motor. And it can also convert the kinetic energy into mechanical energy.

Keywords—Dynamo, Alternator, convertor, Regeneration, braking system, drum brake, pulley.

I. INTRODUCTION

Mechanical engineering is the branch of engineering that encompasses the generation and application of heat and mechanical power and the design, production, and use of machines and tools. Mechanical engineering also includes the conversion of thermal, chemical and nuclear into mechanical energy using engines and power plants. Mechanical engineers work in many industries, and their work varies by industry and function. Some specialties include applied mechanics; computer-aided design and manufacturing; energy systems; pressure vessels and piping; and heating, refrigeration, and air-conditioning systems. Mechanical engineering is one of the broadest engineering disciplines. Mechanical engineers may work in production operations in manufacturing or agriculture, maintenance, or technical sales. As a mechanical engineers career develops, many are given administrator or managerial positions.

II. MATERIAL SELECTION

As per the market study for material selection on the basis of Strength, Hardness, Weldability, Availability, Machinability and cost it was found out that the Mild Steel (MS) is suitable material for braking system. Also Aluminium material was used for making pulley.

III. DESIGN

3.1. Motor Selection:-

Motor is an Single phase AC motor , Power 50 watt , Speed is continuously variable from 0 to 6000 rpm. The speed of motor is variated by means of an electronic speed variator . Motor is an commutator motor ie, the current to motor is supplied to motor by means of carbon brushes . The power input to motor is varied by changing the current supply to these brushes by the electronic speed variator, thereby the speed is also is changes. Motor is foot mounted and is bolted to the motor base plate welded to the base frame of the indexer table.

3 φ AC motor
RPM=6500
HP=(1/12)=(1/12)*746=62.166(W)
Power=62.166x10^{-3} kW=62.166W
T=Motor Torque
P=2πNT/(60*1000)
62.166=2πx6500xT/(60*1000)
T=0.09133N·m

3.1.1. Selection of Power Transmission System:-

1. Open Belt Drive:-
Select V belt drive.
Advantages of V belt drive:-
-It can transmit large amount of power from one pulley to another ,when two pulleys are relatively close to each other.
-In a V belt, the pulleys are provided with a groove.
Advantages of Open belt drive:-
-Used when shafts are parallel.
-Can not used when an open belt drives used due to small angle of contact on the smaller pulley.
-Used to obtain high velocity ratio and desired belt tension.
Adjusted by changing the position of idler pulley, $\beta$.

- Diameter of driver pulley, $d$=20mm

Now,

$$V = \text{Linear speed of belt},$$

$$V = \pi d n / 60000$$

$$V = 6.8067 \text{m/s}$$

$d$=diameter of input pulley,(mm)

$D$= diameter of output pulley,(mm)

$n$=speed of motor,(mm)

$\beta$=angle of groove,

$\rho$=density of belt material,

$c$=centre distance between two pulleys,(mm)

$L$=length of belt,(mm)

$\Theta$=angle of lap,(rad)

$b$=width of belt,(mm)

$A$=area of cross section of belt,(mm$^2$)

Selection of material:

Max. allowable tension=200N=Ft1

Assume coefficient of friction=0.23

D=120mm, d=20mm, n=6500rpm, $\rho$=970kg/m$^3$, 2$\beta$=38°

$G$=Reduction ratio

$G = \text{Dia. of o/p pulley/Dia. of i/p pulley}$

=120/20

=6

Speed of input shaft=RPM/G

=6500/6

=1083.33rpm

=1084 rpm

Centre distance (c)=1700 mm

Length of belt:-

$L = 2c + \pi (D + d)/2 + (D - d)^2 / 4c$

$L = 2 \times 1700 + \pi (120 + 20)/2 + (120 - 20)^2 / (4 \times 1700)$

$L = 3621.38 \text{ mm}$

$L = 3622 \text{ mm}$

Now

$\theta = \pi - 2d$

$\alpha = \sin^{-1} \left( \frac{D - d}{2C} \right)$

$\alpha = 1.6854^\circ = 0.0294 \text{ rad}$

Now

$\theta = \text{angle of lap}$

=$\pi - 2d = \pi - (2 \times 0.0294)$

$\Theta = 3.08276 \text{ rad}$

$b$=width of belt

$=6 - 2(4 \tan \alpha)$

$=6 - 2(4 \times 19)$

$b = 3.24 \text{ mm}$

Width of belt at bottom side=3.24 mm

Width of belt at top side=6 mm

Depth of belt at top side=4 mm
Area of cross section of belt
\[ A = \frac{1}{2}(6+3.24) \times 4 \]
\[ A = 18.48 \text{ mm}^2 \]

Now mass of belt per unit length
But,
\[ t = \text{Thickness of belt} = 4 \text{ mm} \]
\[ m = \rho \times \left( \frac{b}{1000} \right) \times \left( \frac{tx1000}{x1} \right) \]
\[ = 970 \times \left( \frac{6}{1000} \right) \times \left( \frac{4}{1000} \right) \times 1 \]
\[ m = 0.2380 \text{ kg/m} \]

Now
\[ F_c = mV^2 \]
Where
\[ F_c = \text{Centrifugal tension, (N)} \]
\[ V = \text{Linear speed of belt, (m/s)} \]
\[ F_c = mV^2 \]
\[ = 0.2380 + (6.8067)^2 \]
\[ F_c = 10.6561 \text{ N} \]

Now
\[ F_1 = F_{t1} - F_c \]
Where,
\[ F_1 = \text{Tension in tight side, (N)} \]
\[ F_2 = \text{Tension in slack side, (N)} \]
\[ F_{t1} = \text{Max. allowable tension in belt} = 200 \text{ N} \]
\[ F_c = \text{Centrifugal tension, (N)} \]
\[ F_1 = F_{t1} - F_c \]
\[ = 200 - 10.6561 \]
\[ = 189.3439 \text{ N} \]  
\[ (\text{Eqn A}) \]

Using formula
\[ F_1 / F_2 = e^{\frac{(t+sin\beta)}{0.03}} \]
\[ e^{(0.23 \times 0.082/sin19)} \]
\[ F_1 / F_2 = 2.1773 \]

From Eqn A
\[ F_1 = 190 \text{ N} \]
\[ F_2 = 87.2663 \text{ N} \]

Design Of Brake:
Drum material = mild steel  
Density of material= 7800kg/m³  
Drum dia. D=220 mm  
Thickness of drum,t=7 mm  
Length of drum,h=48 mm  
Diameter of pulley=70 mm  
Thickness of liner or pad=2 mm  
Width of pulley=20 mm  
Assume  
$P_i=$Maximum internal pressure  
$P_i=0.50 \text{N/mm}^2$  
$=520 \text{N/m}^2$  
(484 by V.B.Bhandari)  
Now,  
$R_N=(1/2).P_{\text{max}}.b.r.(2\theta+\sin 2\theta)$  
Where,  
$R_N=$Normal Reaction,(N)  
$P_{\text{max}}=$Maximum Internal pressure,(N/m²)  
$b=$thickness of liner of pad,(m)  
$r=$radius of drum,(m)  

$R_N=(1/2)*0.5*0.020*0.110*[2*45*(\pi/180)+\sin(90)]$  

$R_N=1413.9379,\text{N}$  

-Distance between frictional force from centre of drum=35,mm  
-Distance between drum line and fulcrum point O is=120,mm  

**Braking Torque($T_B$) :**  

$T_B=\mu.R_N.r$  

$T_B=0.23*1413.93*0.110$  

$T_B=38.830,\text{N-m}$  

-Distance between applied force and fulcrum point O is 380,mm  

Taking moment about at fulcrum point or fix point,
3.2. Design of Drum:

\[ V = \pi r^2 t + \pi (R^2 - r^2) h, m^3 \]

Where,
- \( R \) = radius of drum, m
- \( t \) = thickness of drum, m
- \( h \) = length of drum, m
- \( V \) = volume of drum, m³

\[ V = \pi (0.110)^2 \times 0.007 + \pi [(0.117)^2 - (0.110)^2] \times 0.048 \]

\[ = 2.6609 \times 10^{-4} + 2.3961 \times 10^{-4} \]

\[ V = 5.057 \times 10^{-4}, m^3 \]

Now, mass of drum,

\[ m = V \rho \]

\[ v = 5.057 \times 10^{-4} \times 7800 \]

\[ V = 3.9475, kg \]

Weight of drum,

\[ W = 3.9475/9.81 \]

\[ W = 0.40240, N \]

Forces acting on brake drum:

- Total downward force applied on shaft = Total bearing force + weight of brake drum

\[ = P + W \]

\[ = 413.6743 + 0.40240 \]

\[ = 414.076, N \]

Therefore total bearing force applied on shaft = 144N

- Total tangential force \( (F_t) \) applied on shaft
\[ F_t = \mu R_N \]
\[ F_t = 0.23 \times 1413.93 \]
\[ F_t = 353.48, \]

Total radial force \( F_t \) applied on brake drum,
\[ F_t = P = 414 \text{N} \]

### 3.3 Design Of Shaft:

**Table -3.1 Permissible Values Of Shear Stress**

<table>
<thead>
<tr>
<th>Description</th>
<th>Ultimate Tensile Strength N/mm(^2)</th>
<th>Yield Strength N/mm(^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I. EN 24</td>
<td>720</td>
<td>380</td>
</tr>
</tbody>
</table>

ASME Code For Design Of Shaft:
Since the loads on most shafts in connected machinery are not constant, it is necessary to make proper allowance for the harmful effects of load fluctuations.
According to ASME code permissible values of shear stress may be calculated from various relations.

**Selection of material:-**
Selection of shaft hard and bear high load as well as withstand of high thrust.

Selecting the material C50

Ultimate Tensile Strength=660 N/mm\(^2\)

to

780 N/mm\(^2\)

Yield strength=380 N/mm\(^2\)
Selecting Ultimate Tensile Strength

$S_u = 720 \text{ N/mm}^2$

From previous calculation

Forces acted on shaft as

$F_1 = 414 \text{ N}, F_2 = 190 \text{ N}, F_2 = 325.20 \text{ N}, F_2 = 87.2663 \text{ N}, W_D = 0.40241 \text{ N}$

According to ASME code, Allowable Shear Stress for Shaft and Key material is ($\tau_{SK}$).

$\tau_{SK} = 0.18 S_yt$

$\tau_{SK} = 0.3 S_yt$

(Taking whichever is smaller value from above.)

$\tau_{SK} = 0.18 S_yt = 0.18 \times 720 = 129.6 \text{ N/mm}^2$

$\tau_{SK} = 0.3 S_yt = 0.3 \times 380 = 114 \text{ N/mm}^2$

Smaller of two values is,

$\tau_{SK} = 114 \text{ N/mm}^2$

The allowable shear stress for accounting the keyway effect is ($\tau_S$).

$\tau_S = 0.75 \times \tau_{SK} = 0.75 \times 114 = 85.5 \text{ N/mm}^2$

Brake load OR Brake force applied of rear shaft is suddenly then,

select the shock and fatigue factor

$K_s =$ Shock factor

$= 1.5 - 2.0$

$K_f =$ Fatigue factor

$= 1.0 - 1.5$

Then, selecting their values as-

$K_s = 1.25$

$K_f = 1.75$

$S_u =$ Ultimate tensile strength, N/mm$^2$

$S_y =$ Yield strength, N/mm$^2$

$F_r =$ Radial force, N

$F_t =$ Tangential force, N

$W_D =$ Weight of brake drum, N

$F_1 \& F_2 =$ Tension on tight and slack side respectively, N

$\tau_{SK} =$ Allowable shear stress, N/mm$^2$

$\tau_S =$ Shear stress, N/mm$^2$

3.3.1. Design Of Solid Shaft
Table - 3.8.1 Values Of permissible Tensile Strength

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>720</td>
<td>380</td>
</tr>
</tbody>
</table>

To find out reaction,
Vertical Loading Diagram:-
\[ R_{AV} + R_{BV} = 414.4024 = 0 \]
\[ R_{AV} + R_{BV} = 414.4024 \text{ N} \]
Taking moment about point A
\[ (414.4024 \times 105) - (R_{BV} \times 360) = 0 \]
\[ R_{BV} = 120.8673 \text{ N} \]
\[ R_{AV} = 293.6350 \text{ N} \]
\[ R_{AV} = 120.8673 \text{ N} \]
Horizontal Loading Diagram:-
\[ R_{AH} + R_{BH} = 277.2663 - 353.48 = 0 \]
\[ R_{AH} + R_{BH} = 630.7463 \text{ N} \]
Taking moment about point A
\[ (277.2664 \times 55) + (353.48 \times 105) (R_{BH} \times 360) = 0 \]
\[ R_{BH} = 145.4584 \text{ N} \]
\[ R_{AH} = 630.7463 - 145.4584 = 485.2879 \text{ N} \]
\[ R_{BH} = 145.4584 \text{ N} \]
\[ R_{AH} = 485.2879 \text{ N} \]
Torque on shaft:-
Select maximum torque on shaft,
\[ T = F_t \times (D/2) \]
\[ = 353.4825 \times (220/2) \]
\[ T = 38883.075 \text{ N-mm} \]
Bending moment on shaft
Vertical bending moment at point C & D:
\[ M_{CV} = R_{AV} \times 55 = 233.5350 \times 55 = 16144.425 \text{ N-mm} \]
\[ M_{DV} = R_{BV} \times 255 = 120.8673 \times 255 = 30821.1515 \text{ N-mm} \]
Horizontal bending moment at point C & D:
\[ M_{CH} = R_{AH} \times 55 = 485.2877 \times 55 = 2690.8235 \text{ N-mm} \]
\[ M_{DH} = R_{BH} \times 255 = 145.4584 \times 255 = 37091.892 \text{ N-mm} \]
Resultant bending moment at point C & D:
\[
Mc = \sqrt{MCV^2 + MCH^2}
\]
\[
\sqrt{(16144.425)}
\]
Compared to \( M_c \) and \( M_D \)
Maximum bending moment,
\[ M = M_0 = 48226.0557 \text{ N-mm} \]
\[ M = 48226.0557 \text{ N-mm} \]
Now,
Diameter of shaft
Now, \( \tau_{max} = 16 T_e/\pi d^3 \)
\[ 85.5 = 16 \times 97390.7094/ \pi dx \]
\[ d = 17.9685 \text{ mm} \]
3.4. Design Of Bearing:

Table 3.9.1 Bearing Selection

<table>
<thead>
<tr>
<th>Isi No</th>
<th>Brng Basic Design No (Skf)</th>
<th>D</th>
<th>D1</th>
<th>D2</th>
<th>B</th>
<th>Basic Capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>17</td>
<td>19</td>
<td>35</td>
<td>10</td>
<td>2850</td>
</tr>
</tbody>
</table>

Bearing can carry considerable thrust load apart from radial load and high speed. Selecting Deep Groove Ball Bearing
Deep Groove Ball Bearing take high load as well as thrust load and it can have high speed.
For shaft diameter=17 mm (range of dia. of shaft are 15 mm, 17 mm, 20 mm)
Series=60
Bearing of basic design no. (SKF)=6003
D=35 mm
d=17 mm
D1=19 mm
D2=33 mm
B=10 mm
r=0.5 mm
Basic Capacity,
Static Capacity(Co)=2850 N
W=285 kg
Dynamic Capacity(C)=4650 N
W=465 kg
Maximum permissible speed=20000 rpm
According to bearing design, Select the shaft of diameter is 17 mm.

3.4.2 Design of output power

O/P Pulley dia D=45 mm
i/p pulley dia D=40 mm
Gear reduction ratio = D/d =1.125
Maximum allowable tension =200N
Coefficient of friction = 0.23
Speed of o/p shaft =6500/1.125 =5777.78 rpm.
Let
L =Length of belt mm.
C = Center distance between two pulley ,mm.
θ = Angle of lap.
b = width of belt ,mm.
A = A/S Area of belt ,mm.
Center distance (c) = 1500 mm.
Length of belt
L =2C +π (D+d)/2 + (D−d)^2/4C
L =31330 mm.
Angle of lap θ = π-2α
Where α=0.00167 rad.
Width of belt (b)
\[ b = 6 - 2(4\tan \alpha) \]
\[ = 6 - 2(4\tan \alpha) \]
\[ b = 3.24 \text{ mm}. \]

\[
\text{C/S Area of belt} = 0.5 \times (6 + 3.24) \times 4
\]
\[ = 18.48 \text{ mm}. \]

\[
\text{Mass of belt/Length}
\]
\[ M = \rho \times b / 1000 \times t / 1000 \times 1
\]
\[ = 0.0.125 \]

\[
\text{Width of belt at bottom} = 324 \text{ mm}. \\
\text{Width of belt at top} = 6 \text{ mm}. \\
\text{Depth of belt} = 4 \text{ mm}. \\
\text{Centrifugal tension (} F_c \text{)} = MV^2
\]
\[ F_c = 0.0125 \times (6.8067) \]
\[ F_c = 46.34 \text{ N}. \]

\[
\text{Maximum allowable tensionin belt}
\]
\[ F_{t1} = 200 \text{ N. (by selecting)} \]

Let
\[ F_1 = \text{Tension at tight side, N} \]
\[ F_2 = \text{Tension at slack sight, N}. \]
\[ F_1 = F_{t1} - F_c \]
\[ = 200 - 46.34 \]
\[ = 153.65 \text{ N}. \]

Using formula
\[ \frac{F_1}{F_2} = e^{(\mu \theta / \sin \beta)} \]
\[ F_1 / F_2 = 9.18 \]
\[ F_1 = 154 \text{ N}. \]
\[ F_2 = 16.78 \text{ N}. \]

### III. CONCLUSION

While concluding this part, we feel quite contented in having completed the project assignment well on time. We had enormous practical experience on the manufacturing schedules of the working project model. We are therefore, happy to state that the inculcation of mechanical aptitude proved to be a very useful purpose. We are as such overwhelmingly elated in the arriving at the targeted mission. Undoubtedly the joint venture has had all the merits of interest and zeal shown by all of us the credit goes to the healthy co-ordination of our batch colleague in bringing out a resourceful fulfillment of our assignment described by the university.

### REFERENCES

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