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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 ɸ AC motor
RPM=6500
HP=(1/12)=(1/12)*746=62.166(W)
Power=62.166x10-3 kW=62.166W
T=Motor Torque
$P=2\pi NT/(60*1000)$
$62.166=2\pi x6500xT/(60*1000)$
T=0.09133N-m

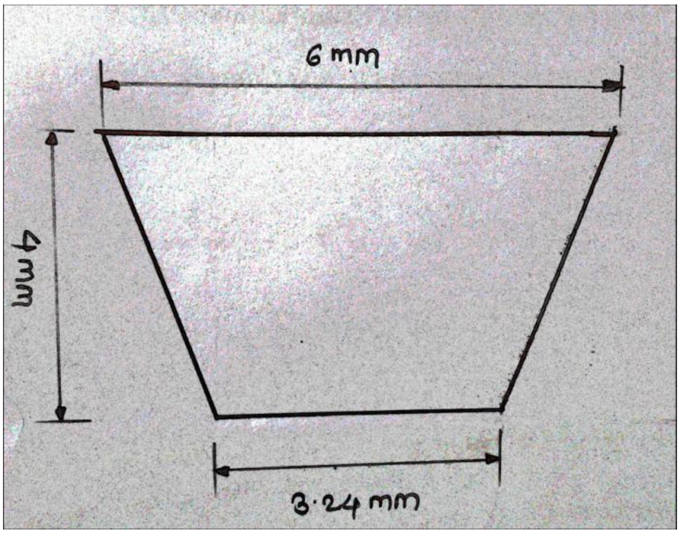
3.1.1.Selection of Power Transmission System:-

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.β
-Diameter of driver pulley,d=20mm
Now,
V=Linear speed of belt,
$V=\pi dn/60000$
$=\pi x20x6500/60000$
V=6.8067m/s
d=diameter of input pulley,(mm)
D= diameter of output pulley,(mm)
n=speed of motor,(mm)
β=angle of groove,
ρ=density of belt material,
c=centre distance between two pulleys,(mm)
L=length of belt,(mm)
Θ=angle of lap,(rad)
b=width of belt,(mm)
A=area of cross section of belt,(mm2)
Selection of material:-
Max. allowable tension=200N=F_{t1}
Assume coefficient of friction=0.23
D=120mm,d=20mm,n=6500rpm, $\rho=970kg/m^3$, $2\beta=38°$
G=Reduction ratio
G=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:-
$L=2C+\pi(D+d)/2+(D-d)^2/4C$
$=2x1700+ \pi(120+20)/2+(120-20)^2/(4x1700)$
=3621.38 mm
L=3622 mm
Now
$\theta= \pi-2d$
Where,
$\alpha=\sin^{-1}(D-d/2C)$

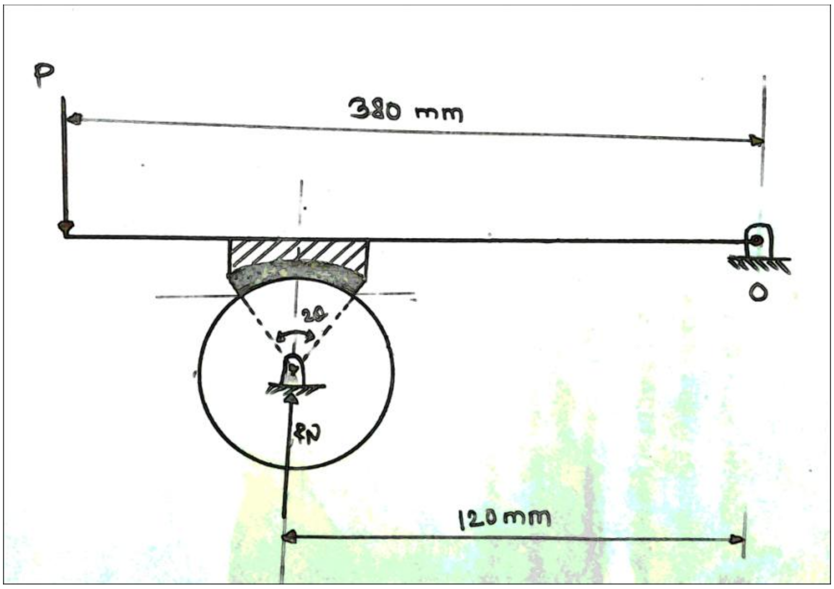
$$\alpha = \sin^{-1}(120-20/2x1700)$$

$\alpha=1.6854°=0.0294$ rad
Now
θ=angle of lap
$=\pi-2d= \pi-(2x0.0294)$
Θ=3.08276 rad
b=width of belt
$b=6-2(4\tan\alpha)$
=6-2(4xtan 19)
b=3.24 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



Fig

Area of cross section of belt
A=1/2(6+3.24)x4
A=18.48 mm^2
Now mass of belt per unit length
But,
t=Thickness of belt=4 mm
m=ρx(b/1000)x(tx1000)x1
=970x(6/1000)x(4/1000)x1
m=0.2380 kg/m
Now
$Fc=mV^2$
Where
Fc=Centrifugal tension,(N)
V=Linear speed of belt,(m/s)
$Fc=mV^2$
$=0.2380+(6.8067)^2$
Fc=10.6561 N
Now
$F_1=F_{t1}-F_c$
Where,
F_1=Tension in tight side,(N)
F_2= Tension in slack side,(N)
F_{t1}= Max. allowable tension in belt=200 N
F_c=Centrifugal tension,(N)
$F_1=F_{t1}-F_c$
=200-10.6561
=189.3439 N
F_1=190 N (Eq^n A)
Using formula
$F_1/ F_2=e^{(\mu\theta/\sin\beta)}$
$= e^{(0.23x3.082/\sin19)}$
F_1/ F_2=2.1773
From Eq^n A
F_1=190 N
F_2=87.2663 N
Design Of Brake:-



Drum material = mild steel
Density of material= $7800 kg/m^3$
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
$P_i=0.50\ N/mm^2$
$=520\ N/m^2$ (484 by V.B.Bhandari)
Now,
$R_N=(1/2).P_{max}.b.r.(2\theta+\sin 2\theta)$
Where,
R_N=Normal Reaction,(N)
P_{max}=Maximum Internal pressure,(N/m^2)
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,N

-Distance between frictional force from centre of drum=35,mm

-Distance between drum line and fulkcrum 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,N-m

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

Taking moment about at fulkcrum point or fix point,

$R_N*120-F_t*35-P*380=0$

$1413*120-(\mu*R_N)*35-P*380=0$

$1413*120-(0.23*1413*35)-P*380=0$

P=413.6743,N

P=Total bearing force ,N

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=voiume of drum,m^3

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

$=2.6609*10^{-4}+2.3961*10^{-4}$

$V=5.057*10^{-4}$,m^3

Now,

mass of drum,

m=v*ρ

$v=5.057*10^{-4}*7800$

V=3.9475 ,kg

Weight of drum,

W=3.9475/9.81

W=0.40240 ,N

Forces acting on brake drum:

-Total downword 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

$F_t=\mu.R_N$

$F_t=0.23*1413.93$

$F_t=353.48$,

-Toal radial force(F_r)applied on brake drum,

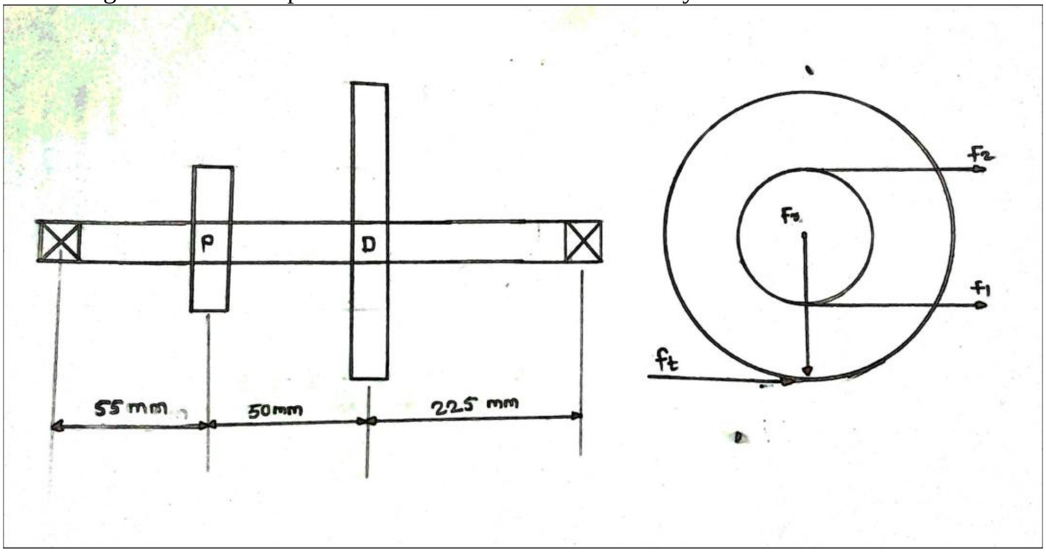
$F_r=P=414$,N

3.3.Design Of Shaft:-

Table -3.1 Permisible Values Of Shear Stress

Description	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
I. EN 24	720	380

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 form various relation.



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

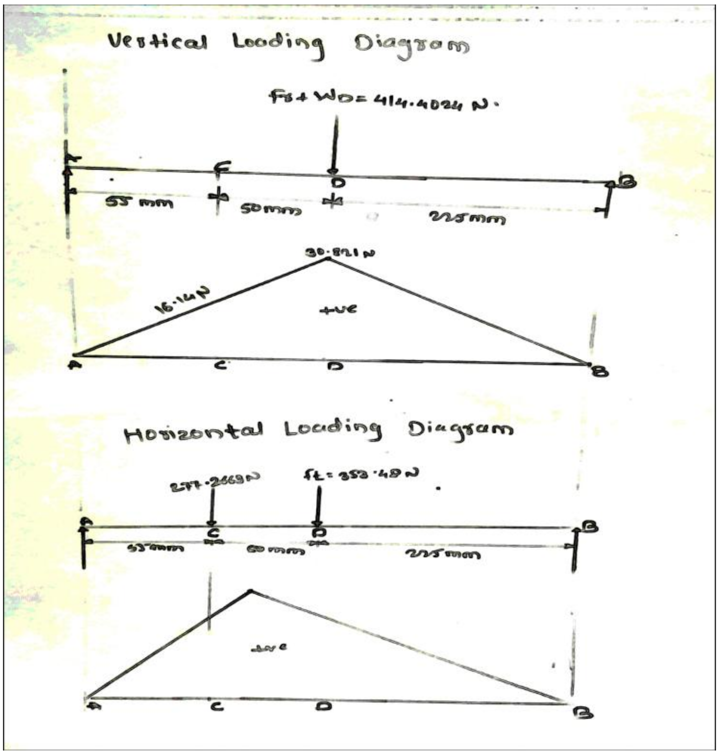
to

780 N/mm^2

Yeild strength=380 N/mm^2

Selecting Ultimate Tensile Strength

S_{ut}=720 N/mm^2
From previous calculation
Forces acted on shaft as
F_r=414 N,F_1=190 N,F_t=325.20 N,F_2=87.2663 N,W_D=0.40241 N
According to ASME code,Allowable Shear Stress for Shaft andf Key material is(τ_{SK}),
τ_{SK}=0.18 S_{yt}
τ_{SK}=0.3 S_{yt}
(Taking whichever is smaller value from above.)
τ_{SK}=0.18 S_{yt}=0.18x720=129.6 N/mm^2
τ_{SK}=0.3 S_{yt}=0.3x380=114 N/mm^2
Smaller of two values is,
τ_{SK}=114 N/mm^2
The allowable shear stress for accounting the keyway effect is(τ_S),
τ_S=0.75x τ_{SK}=0.75x114=85.5 N/mm^2
Brake load OR Brake force applied of rear shaft is suddenly then,
select the shock and fatigue factor
K_b=Shock factor
=1.5-2.0
K_t=Fatigue factor
=1.0-1.5
Then, selecting their values as-
K_t=1.25
K_b=1.75
S_{ut}=Ultimate tensile strength, N/mm^2
S_{yt}=Yeild 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
τ_{SK}=allowable shear stress, N/mm^2
τ_S=Shear stress, N/mm^2

:- 

3.3.1.Design Of Solid Shaft

Table - 3.8.1 Values Of permissible Tensile Strength

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
EN 24	720	380

To find out reaction,
Vertical Loading Diagram:-
$R_{AV}+R_{BV}-414.4024=0$
$R_{AV}+R_{BV}-414.4024$ N
Taking moment about point A
$(414.4026x105)-(R_{BV}x360)=0$
$R_{BV}=120.8673$ N
$R_{AV}=414.4026-120.8673$
$R_{AV}=293.6350$ N
$R_{AV}=293.6350$ N
$R_{BV}=120.8673$ N
Horizontal Loading Diagram:-
$R_{AH}+R_{BH}-277.2663-353.48=0$
$R_{AH}+ R_{BH} =630.7463$ N
Taking moment about point A
$(277.2664x55)+(353.48x105)(R_{BH}x360)=0$
$R_{BH}=145.4584$ N
$R_{AH}=630.7463-145.4584$
$=485.2878$ N
$R_{AH}=485.2878$ N
$R_{BH}=145.4584$ N
Torque on shaft:-
Select maximum torque on shaft,
$T=F_t x(D/2)$
$=353.4825x(220/2)$
$T=38883.075$ N-mm
Bending moment on shaft
Vertical bending moment at point C & D:
$M_{CV}= R_{AV}X55=233.5350X55=16144.425$ N-mm
$M_{DV}= R_{BV}X255=120.8673X255=30821.1615$ N-mm
Horizontal bending moment at point C & D:
$M_{CH}= R_{AH}X55=485.2877X55=2690.8235$ N-mm
$M_{DH}= R_{BH}X255=145.4584X255=37091.892$ N-mm
Resultant bending moment at point C & D:

$$\boldsymbol{Mc = \sqrt{MCV^2 + MCH^2}}$$

$$\boldsymbol{\sqrt{(16144.425)}}$$

Compared to M_C and M_D
Maximum bending moment,
$M= M_D=48226.0557$ N-mm
$M=48226.0557$ N-mm
Now,
Diameter of shaft
Now, $\tau_{max}=16\ T_e/\pi d^3$
$85.5=16x97390.7094/\ \pi x d^3$
d=17.9685 mm

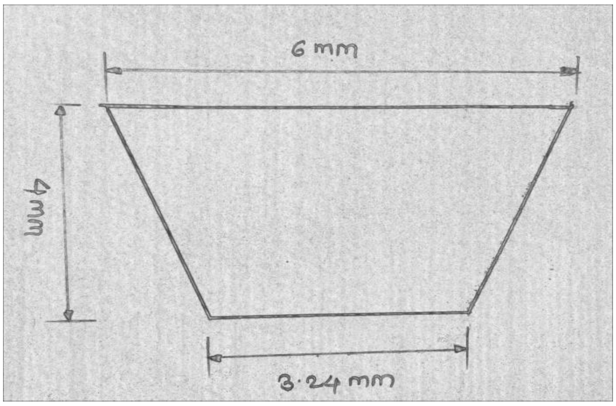
d=18 mm
As Yeild Strength Of Solid Shaft Is 114 N/Mm^2>85.5 N/Mm^2
Design Is Safe.
3.4.Design Of Bearing:-
Table -3.9.1 Bearing Selection

3.4.1.Selection of bearing:

Isi No	Brg Basic Design No (Skf)	D	D1	D	D2	B	Basic Capacity	
							C Kgf	Co Kgf
20a C04	6204	17	19	35	10	10	2850	

Bearing can carrying 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 high speed.
For shaft diameter=17 mm(range of dia. of shaft are 15mm,17mm,20 mm)
Series=60
Bearing of basic design no.(SKF)=6003
D=35 mm
d=17 mm
D_1=19 mm
D_2=33 mm
B=10 mm
r=0.5 mm
Basic Capacity,
Static Capacity(C_o)=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
$L = 2C + \pi (D+d)/2 + (D-d)^2/4C$
L =31330 mm.
Angle of lap $\theta = \pi - 2\alpha$
Where α=0.00167 rad.
Width of belt (b)



$b = 6 - 2(4\tan\alpha)$
$= 6-2(4\tan\alpha)$
$b = 3.24$ mm.
C/S Area of belt = 0.5* (6+3.24)*4
=18.48 mm.
Mass of belt/Length
$M = \rho * b/1000 * t/1000 * 1$
= 0.0.125
Width of belt at bottom =324mm.
Width of belt at top=6mm.
Depth of belt =4 mm.
Centrifugal tension (Fc) $=MV^2$
$F_c = 0.0125*(6.8067)$
$F_c = 46.34$ N.
Maximum allowable tensionin belt
$F_{t1} = 200$ N. (by selecting)
Let
F_1 = Tension at tight side, N
F_2 =Tension at slack sight ,N.
$F_1 = F_{t1} - F_c$
= 200-46.34
= 153.65N
Using formula
$F_1/F_2 = e^{(\mu\theta/\sin\beta)}$
$F_1/F_2 = 9.18$
$F_1 = 154$ N.
$F_2 = 16.78$ N.

III. CONCLUSION

While concluding this part, we fill quite contended 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.

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