DESIGN AND FABRICATION OF SYSTEM FOR SMALL SCALE POWER GENERATION USING SPEED BRAKER

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Abstract: *With the increasing exhausts of fossils fuels and its adverse environmental effects, clean energy production has become a quintessential demand of the hour. In the other hand, a huge amount of kinetic energy is wasted on our roads which can be used for clean energy production and various other purposes. This paper presents the design of components like spring, rack and pinion and Shaft to make the speed breaker able to handle the load of max 400 kg along with the experimental study which shows that the power to 1.2 to 28.6 watt was produced when the load of 60 to 240kg were applied at a different interval. Hence, we concluded that this method could be a good source of a non-conventional form of producing energy.*

Key Words: *clean energy, design, Shaft, speed breaker, Non-conventional*

1. INTRODUCTION

As a substantial amount of energy is wasted on speed breaker by vehicles moving on it, also to control the speed of vehicles speed breaker is required. The yearly, vehicle growth rate has also been increasing significantly in Nepal. The motivation to build a device to utilize the energy that is being wasted form moving vehicle to useful work is because the amount of traffic and consequently the Number of speed breaker on roads is increasing [1]. In the context of Nepal, as for July 2019, the peak load is 1,160 MW and per capita consumption to be 245 kwh, and by next five years, it is expected to be 1500 kwh [2]. In FY 2018/19, the growth in the economy is anticipated to hold out 6.5 per cent. On top of the 90 billion in petroleum, Nepal used up almost 20 billion in getting electricity from India [3]. The social and economic development of Nepal is affected by inefficient energy supply. In such a situation, the use of the alternate system would be the most useful one. So more investments and efforts should also be done to the alternate and pure form of energy. This paper is focused mostly on the design of such an energy harnessing system and on the principle of converting kinetic energy into electrical energy.

2. METHOD AND MATERIAL

2.1 Working

The primary function of this system is to obtain electrical energy by the use of kinetic energy of moving vehicle. Here, initially, the load is applied on the speed hump. Due to the application of load, the spring gets compressed, and the rack attached to the hump starts to move downward. The rack undergoes reciprocating motion. This reciprocating motion of rack is converted into rotatory motion by the pinion connected with rack. The pinion is mounted on the Shaft so, as the pinion rotates the shaft-mounted pinion also starts rotating. To transfer the speed, the larger sprocket is attached to the pinion mounted Shaft and is connected to the smaller sprocket (which acts as freewheel). The connection is done with a chain drive. Multiplication of seed takes place with the transfer of speed from larger to smaller sprocket. So, the smaller sprocket mounted shaft starts to rotate with the greater speed, which in turn rotates the flywheel that is attached to the same Shaft. The flywheel continues to store the energy. This high speed of Shaft provides sufficient energy to rotate the generator, thereby producing D.C. current.

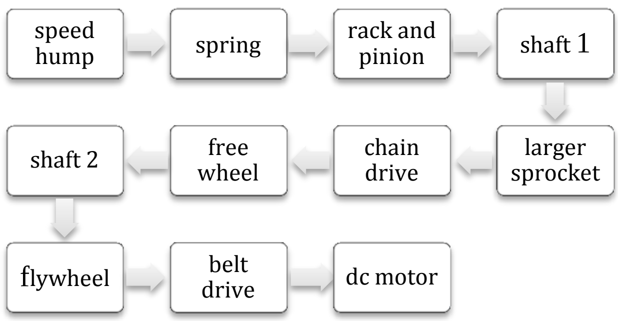

Fig-1: power flow

2.2Design

In this design, top to the bottom method is used, the vertical displacement of the bumper plate (compressed length of spring) is taken as the main design parameter, and the design of other components is done subsequently.

We are considering the mass between 300-400 kg (generally for motorbikes) for our system. Considering this, various parameters were calculated to determine the actual dimensions of the components like spring, rack and pinion, Shaft, and sprocket so that it could hold the supposed load. Finally, the model is drawn using CATIA V5 with respective dimensions followed by material collection fabrication and assembly.

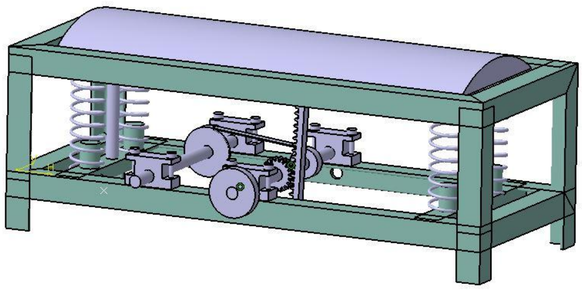

Fig-2: 3D design of the complete system

2.3 DESIGN DETAILS

2.3.1Specification of frame

It is the supportive structure on which the entire components are mounted.

Type of material – mild steel

Size 1000×450×500mm

2.3.2 Speed hump

It is the uppermost part of the mechanism on which the actual speed breakers lie. It also has guidance pipes for proper support and alignment with the frame.

Table-1: design specification of speed hump

S.N	Descriptions	Remarks
1.	Length of hump	925 mm
2.	Breadth of hump	400 mm
3.	Height of hump	10 mm
4.	Length of guiding rod	250 mm
5.	The diameter of the guiding rod	10 mm
6.	Type of Material	Stainless steel

2.3.3 Spring

Design Considerations for spring

For this, we have selected music wire ASTM A228 because of its wide availability, toughness and also due to its feature to withstand higher stress under frequent loading condition.

Diameter of wire (d) = 6.5mm

Outside Diameter (Do) = 64 mm

Type of ends: squared, closed with ends ground

Design calculation

Mean coil diameter (D) = $D_0 - d$ = 57.5 mm

Spring index (c) = D/d = 8.84

Now, for calculation of active coil from [3]

$N_a = Nt - 2 = 9$

Where N_t = total coils

Na= Number of active coils

For the value of G, using [3]

G=80 Gpa

The spring rate of scale (K): $K = d^4G/8D^3N$ [4]

□ K=14.40 N/mm

For minimum tensile strength,

A=0.145, m=2211, d=6.5 mm

$S_{ut} = A/(d_m)$ = 750.3 M pa [5]

Force on the spring:

$K_B = (4C + 2) / (4C - 3)$ where, K_B= stress concentration factor

Maximum Force the spring can withstand [6]

$F_{max} = (\square\, d3) / (8KBD)$ =1516.59 N

So, maximum Force a spring can withstand is 1516.59 N

And, its corresponding length for maximum compression is

y= F/k=10.8 cm

So, our design calculation will be based upon maximum Force each spring will withstand and maximum compression length.

2.3.4 Rack and Pinion

Design consideration

To avoid noise and heat problem at low speed and to avoid axial thrust due to helical gear spur gear is used. A little consideration and research show that 20° Full-depth involute system is mostly used in the Rack and

Pinion system; hence our calculation will be based on this system. Also, after doing some iterations and also keeping the cost-effectiveness of product in mind, we selected Case Hardened Carbon steel and speed of rack and acceleration time are also considered as below.

The Pitch Diameter of pinion 50mm

Mass of the container = 400Kg (suppose)

Speed of the Rack (v) = 0.3m/s(suppose)

Acceleration Time =2 sec (suppose)

Load Service Factor C_s [7]= 1.8

Calculation

Velocity Factor C_v[8] =3/(3+v)

Tangential load (F_t) = μmg = 1177.2 N

Centrifugal force(F_c) = ma = 1177.2 N

Total load (F_T) =F_t+F_c = 1217.2 N

Effective load (force) due to velocity factor [9] = F_T*C_s/C_v

= 2410.056 N

For full depth involute system, minimum Number of teeth on pinion of $20^0$to avoid interference would be 18

- The pitch diameter of Pinion (D) – 50 mm (As per the space limitations)
- No of Teeth on Pinion – 24(This value is taken after few iterations of tooth stress and also keeping in mind that module value should be minimum for calculated tangential Force)

We Know That,

Module(m) = D/T =2

We have fixed the pitch circle diameter; we have calculated the module and Number of teeth. Also, we have calculated the Total tangential Force.

Now we have to calculate one more important parameter- Tooth Face Width (b)

Using lewis equation,

Ft=σbπym [10]

Where,

σ= permissible stress

b= face width

y= lewis form factor =$0.15-\frac{0.841}{T}$ = 0.13995

We know that the ultimate stress of carbon steel ranges from 375 - 400 MPa.

Let us consider factor of safety be 2,

$\sigma_p = \sigma_u/2$ =190 Mpa

since Ft= σ_p*bπym so, b = 14.4 mm (approx.)

Conclusion

Pitch Circle Diameter of Pinion- 50mm

No of Teeth – 24

Module – 2

Tooth Face width – 14.4 mm

Calculation for pinion

Rest of the parameters like Addendum, Dedendum, Working Depth, Tooth Thickness, Minimum Clearance are calculated as below,

Pressure angle of pinion (Φ) = 20 (suppose)

Module = Diameter of pitch circle/ total number of teeth

= 2 mm

(Radius)pitch circle (r)=25mm

Addendum (a)=2 mm

(Radius) addendum circle (ra) = addendum+r= 27mm

(Length)path of contact [11]=(a/sinΦ)+{[ra ^2–(rsinΦ) ^2]} ^0.5 - r sin Φ = 13.9 mm

Minimum number of teeth of contact= Length of arc of contact/circular pitch

Since, (Length)arc of contact= Length of path of contact / cos Ø =14.95mm

So, minimum number of teeth of contact= 3 teeth For gear, circumference = 2 π R=157.45mm

(Length)rack =157.45+250=407.45mm

2.3.5 Shaft

For shaft 1

Design consideration

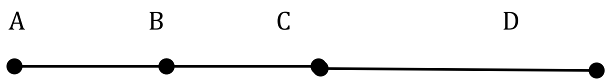

Fig-3: Shaft 1

Point A= Bearing 1 AB= 100mm

Point B= Large sprocket AC= 200 mm

Point C= Pinion AD= 400mm

Point D= Bearing 2

According to the availability of material and commonly used type, we used Cast Iron to design the Shaft.

We know that,

The ultimate tensile stress of Carbon Steel (S_{ut}) = 270Mpa [12]

Velocity of rack (V_R) = 1/3 m/s (suppose)

Calculation

$F_R/T_P = V_R/W_P = r_p$[13]

where,

r_p= pitch radius pinion

W_P =angular velocity of pinion = 14 rad/sec

We know: W_P= 2 πN/60

N_1= 230rpm = speed of Shaft

Torque (T_P)= Ft* r_p = 60.2 Nm

Now,

Normal load acting on pinion (W_c)= Ft/cos20 =2564.87 N

Vertical component (W_{cv}) = W_c Cos20 = 2410N

Horizontal component (W_{ch}) = W_c Sin20 = 824.26N

For Large sprocket

Number of teeth = 40

Diameter of large sprocket (D_s) = 162 mm

For chain drive, torque acting will be T= (T_1-T_2) Rs

i.e (T_1-T_2) = 796.3 N

where, Rs is the radius of large sprocket

angle of wrap = 180 degree

$T_1/T_2=e^{0.2*3.1416}$[14]

T_1 =1711.58N and T_2=915.28N

Horizontal load acting at B = T_1 + T_2 = 2626.86N

T_1 =1711.58N and T_2=915.28N

Horizontal load acting at B = T_1 + T_2 = 2626.86N

Vertical load at B = 0

Load diagram

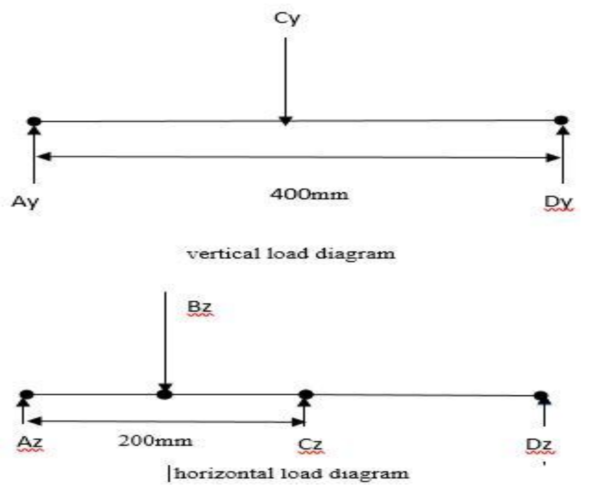

Fig-4: Load Diagram for shaft 1

From above diagram

A_y+D_y = 2573.9

A_y=1286.95N and D_y= 1286.9N

Here, A_y,D_y= vertical load on bearing 1 and 2 respectively

A_z +C_z +2293 = 0

A_z= -600.7N = 600.7N and D_z= -1692.3 = 1692.3 ↑

A_z, D_z =horizontal load on bearing 1, pinion and bearing 2 respectively

From fig 3

For vertical load

M_b= 128.9 Nm

M_c= 257.38 Nm

Here, Mb and Mc are moment at point B and C

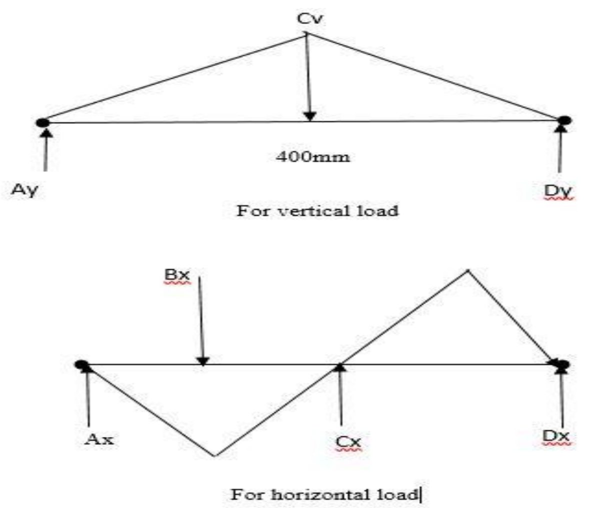

Fig-5: BMD for shaft 1

For horizontal load

M_B=60Nm and Mc=-142Nm

At point B

M_B =142.18Nm and

At point C

Mc = 293.9 Nm

M_{max} = 293.9Nm

If kt and km be the integrated fatigue and shock factor of torsion and bending, where

Km= 1.5 and kt= 1

Now, For maximum tensile strength [15]

⍰ max = (16/ πd3) × √ (km* M)^2 +(Kt×T)^2 96*10^6 =2269.9/d^3

So, required diameter of Shaft (d)= 28.7 mm

2.3.6 Sprocket

The relation between the speed of rotation and teeth of the small and large sprocket is given as, [16]

$N_1/N_2 = T_2/T_1$

Where, T_1, T_2 = teeth of smaller and larger sprocket

N_1, N_2 = Rotational speed of small and large sprocket respectively

N_2 =400 rpm

The average velocity of the chain (v) = $\frac{3.14*N*D}{60}$

= 0.13 m/s

Pitch of the chain (P) =12.8mm

The factor of safety of chain drive [17] = $\frac{Breaking\ Strength}{Total\ Tension\ on\ driving\ side}$

Breaking strength (W_B) = 106*P^2(for roller chain)

= 15264 N

Factor of Safety = W_B / W

=15264/ 2626.89 > 7 {hence, satisfied for roller chain}

Power Transmitted by chains:

If V be the velocity of chain and n be the factor of safety, then

On the basic of breaking load, [18]

P=W_B*V/n*K_s = 211.6 W

Here, ks= k1,k2k3 and k1= load factor

k2= lubrication factor k3= rating factor

Tangential driving force by chain on sprocket (Ft)s=1933.33N

2.3.7 shaft 2

Design consideration

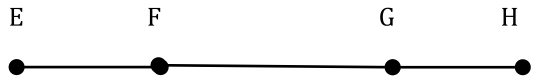

Fig-6: Shaft 2

Point E= Bearing 3 EF= 100mm

Point F= Small Sprocket EG= 400mm

Point G= Bearing 4 GH= 50 mm

Point H= Flywheel

tangential force by chain on sprocket Ft=1933.33N

Calculation

Normal load acting on tooth gear: W_F =1933.33/ cos20 = 2057.40N

Load acting normally at 20 degree to the vertical

Vertical component of W_F =vertical load acting on shaft

Vertical load at point at B W_{FV}=W_F*Cos20 = 1933.33N

Horizontal load component for W_{FH}= W_F * sin20= 704.53 Also, let the mass of flywheel which is at 5cm away from bearing G be 3.5 Kg

So, Weight of flywheel (W_H)= m*g= 19.62 N

M= mass of flywheel

For horizontal load:

$E_Z+G_Z+W_{FH}$ = 0

E_Z+G_Z +704.53=0

Taking moment at point G

E_Z = -527.65 = 527.65 N↑

G_Z = -272.91= 272.91 N

E_z and G_z = vertical load at bearing 3 and 4

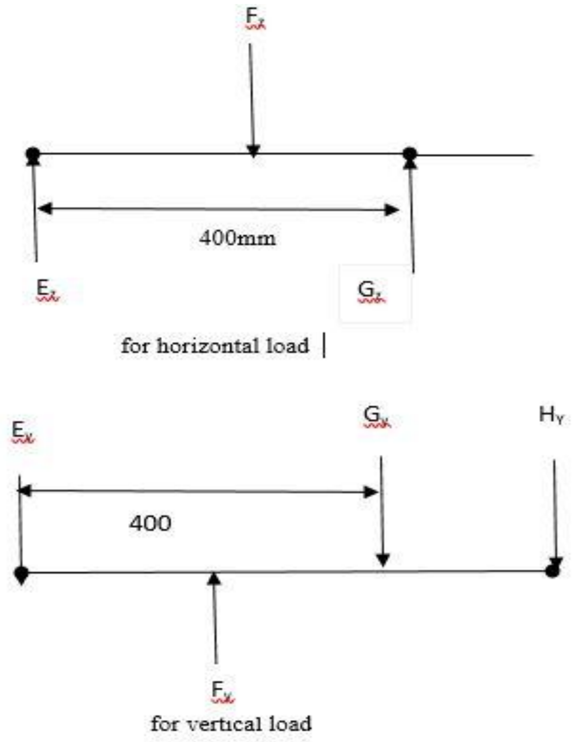

Fig-7: load diagram for Shaft 2

Taking moment at point at C

E_y = -1447.59N = 1447.59N

G_Y =-485.74 N = 485.74N

Here, Gy, Ey = vertical load at bearing 4 and 3

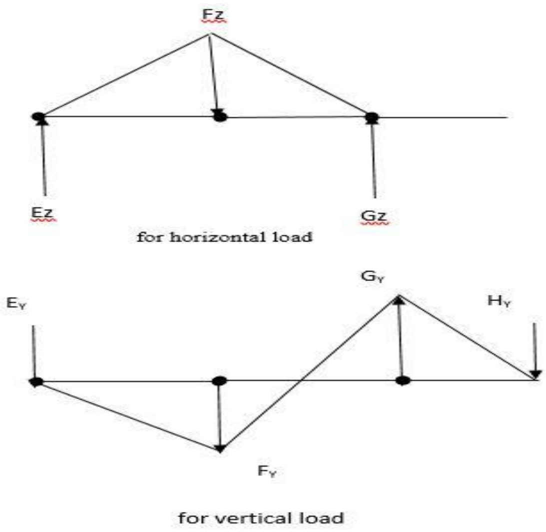

Fig- 8: BMD for shaft 2

Taking a moment at G

M_G= 52.06 Nm

Taking Vertical Load

M_F = 144.7 Nm

M_G= 92 Nm

Here, M_{G} and M_f = bending moment at point F and G

Hence,

The moment at point F,

M_{F} =$\sqrt{2704 + 13156.04}$ = 154 Nm

M_G =92 Nm

Since the maximum bending moment is at point B

∴ $(M_B)_{max}$ = 154 Nm

If k_m and k_t are the combined shock and fatigue factor of bending and torsion respectively then,

Maximum tensile stress [15]

τ_{max} = (16/πd^3) * $\sqrt{(km * m)^2 + (kt * T)^2}$

0.3*275*10^6=1233.92/d^3

So, d = 25mm

Hence the diameter of Shaft two would be 25mm.

2.4 Theoretical power Calculation

Let, Vehicular mass = 300Kg
Since, average standard height for speed breaker = 0.1m
Body weight = Acting force = 300 x 9.8m/s = 2940N

Weight of the Body = Acting force = 300 x 9.8m/s = 2940N

Since Height of Speed breaker= Distance travelled by body=10 cm

On one pushing force, the produced output power = (2940 x 0.1)/60 = 4.9Watt

In one minute, the developed power by a single-vehicle over the breaker arrangement = 294 watt

2.5 Fabrication and Assembly

First of all, a metallic frame of the speed breaker arrangement was prepared. Then two metal rods having 25 mm diameter with ball bearing on both ends are welded on the metallic frame.

Into the supports, holes were drilled, and to house, the bearing boring was done to increase the diameter. Then, on the machined hole hydraulic press fit was done to the bearing. On the metal rod, the sprockets, gears, flywheel were mounted. The metal frame was first to cut into desired shape and size using the metal cutter. Four compression springs are mounted on the speed breaker arrangement, and D.C. generator is mounted at last with the help of a clamp. Also, the hydraulic press fit was done to the gears onto the Shaft. The assembled shafts were then mounted on the bearing supports. The Shaft, gears along with the bearing support were positioned and fixed accurately inside the frame.

Nut bolt was used to fix the frame, and welding was done to fix the metallic hump onto the frame. The upper end of the rack and all four spring were also welded on the inner portion of the hump for fluent reciprocating motion.At most dimensional accuracy, the generator shaft was machined out of M.S. rod. And the hydraulic press was used to press-fit all the ball bearing onto the Shaft. Mild steel plates were used to fabricate the bearing support for the generator shaft. Finally, by the help of nut and bolts, the generator was mounted.

Fig-9: fabricated and assembled system

3. RESULT AND EXPERIMENTAL STUDY

A simple test was conducted where various loads were load applied dynamically for the different interval of time.

During the experimental study, the following steps were taken into consideration:

1. Direction of application and movement of loads was perpendicular to the surface of the hump plate.

2. The application of loads was in such a way that compression occurs uniformly in all four springs.

3. All the measuring instruments like tachometer, voltmeter, and multi-meter were appropriately calibrated.

Firstly, a load of 60 kg was applied on the hump plate dynamically for 1 to 4 times. The load was applied for one, two, three, and four times; That is, the load was applied for one, two, three, and four pushes. The time taken for all the pushes was measured with the help of a stopwatch. At the same time, the rpm of shafts, flywheel, and dc generator was measured with a digital tachometer and voltage, and the current (at no load) were also observed and recorded using a multi-meter. All these data were taken for all Number of load applications or pushes. Similarly, the process is continued and repeated for a load of 120kg and 240 kg.

Fig-10: Experimental process

These observed results are analyzed and tabulated below as:

Table 2: Observed data

No. of load applications	Load (kg)	Time interval of application of load (sec)	Rpm of dc generator (mean)	voltage	current	Power output (watt)
1	60	2.5	235	12	0.1	1.2
2	60	4.4	328	12	0.4	4.8
3	60	6.7	425	12	0.5	6
4	60	8.8	488	12	0.8	9.6
1	120	3	333	12	0.7	8.4
2	120	4.8	420	12	1.1	13.2
3	120	7.3	505	12	1.7	20.4
4	120	9.5	584	12	2.4	28.8
1	240	3.5	395	12	1	12
2	240	6.1	520	12	1.8	21.6
3	240	8.4	762	12	2	24
4	240	10.5	885	12	2.4	28.6

4. CONCLUSION

This work presents fabrication details, design calculation and analysis of small prototype of speed breaker power generation. Also, a brief experimental study shows a maximum power output of 28.6 Watt under the applications of the load of 240 kg at an interval of 10.5 seconds. This is quite a promising implication in case of such a system for further study, scope and implementation as well. However, it was revealed that powers of roughly 1.2 to 28.6 watt could be obtained from the speed-breaker system if the mass of 60 to 240 kg is applied under different intervals of time. This range of power output can be optimized with the use of batteries, specially designed charging circuit, supercapacitors and more robust fabrication in order to obtain the useful amount of power to light streets, traffics lights, electronic sensors and others.

5. ACKNOWLEDGEMENT

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