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Structural Analysis of Power Generation through Footstep

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Abstract: Energy generation has been a serious concern in present scenario. Efforts are being made to develop such systems which could serve for the non-renewable sources of energy in hitherto there comes a requirement to mechanically design such systems. The system designed uses a defined mechanism that comprises of rack and pinion, gear assembly, and springs. Dimensioning of each and every component is done using mechanical data handbooks that involves the analysis of failure criterion i.e. in shear, etc. in various components that could happen during working of system under loading of 75 kg and above. In addition, weld design is also calculated. Simulation depicts the equivalent stress distribution in the designed spring and weld. On considering all governing factors a safer design of mechanical footstep power generator is concluded.

Keywords — Equivalent stress distribution, footstep power generator, simulation of spring & weld and weld design.

I. INTRODUCTION

The demand of energy has been increasing since its very first use i.e. at the time of Stone Age. As the development started through the years, demand of energy has increased rapidly vis-à-vis new discoveries in field of harvesting energy resources were made and the process is continued till date [1]. Mechanical footstep power generator is one such system. Power generation through footstep does not rely on any conventional sources of energy which is one of the advantages of this system and this system can easily be installed in any public places to obtain maximum output in the form of electrical energy. The only source of energy for this system is human locomotion. Countries like India and China which not only have large population but also have high population density, in such situation public places like railway stations, bus stands, etc. become hotspot for the human locomotion and this human locomotion provides a great chance to harvest this bio-energy [2].

With such a huge human locomotion, it becomes important to design a system which could serve as a durable source to harvest energy. Design of mechanical footstep power generator takes in account a lot of factors regarding variable loads of footstep, places of application, number of people passing over the system, amount of energy being harvested in each step, safety factors that come into play, etc. [3].

This system has been designed with all mechanical aspects to widen its application; failure analysis is also carried out in order to design a safer system under variable loading conditions [4].

II. WORKING PRINCIPLE

The mechanical footstep power generator uses human footstep energy and converts it to electrical energy, and this conversion of mechanical energy to electrical energy uses a proper designed mechanism (Fig. 1).

When a person steps on the stepping platform, then using rack and pinion arrangement, the upper platform pushed down, this causes the rotation of shaft on which the pinion is mounted. The shaft is connected to the DC generator using a geared arrangement and further causes rotation of generator's shaft to harvest energy. As the person puts off his step, the spring arrangement helps the system to retract to its initial position. In this way the energy is harvested via to and fro motion.

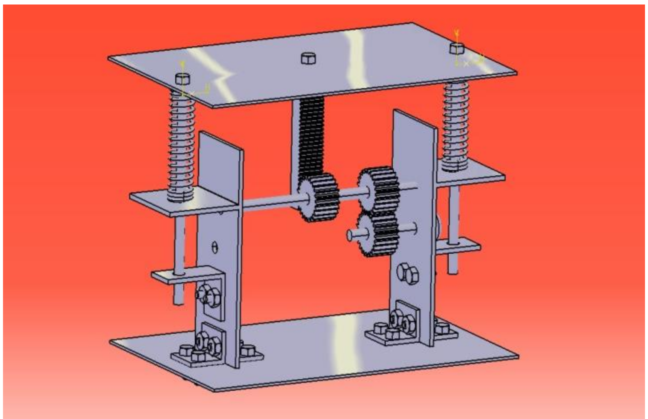

Figure 1: Mechanical footstep power generator.

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III. DESIGNING

Design of Mechanical Footstep Power Generator involves the designing of its various components i.e. rack and pinion, shaft, gear assembly, weld, and springs. The designing has been done using various design data books [8 - 10].

Material selection of various components in the design of mechanical footstep power generator has been done with a foresight of its availability and economy.

3.1 POWER TRANSMISSION

Calculation of power transmission lays emphasis on the load which is applied through human footstep. On an average a load of 75 kg per person is assumed for designing purpose and a maximum displacement of 7 cm is taken, to consider transmission of power through human footstep [5 - 6].

A load of 75 kg provides 687 N of force. Then power transmitted is work done per unit time i.e. P = 48.09 W, for this power transmission the length of rack is 145 mm.

3.2 DESIGN OF GEAR ASSEMBLY

Gears are the toothed wheels or multi-lobed cams, which transmit power and motion from one shaft to another by means of successive engagement of teeth. Gears are used to transmit large power, the motion can be transmitted at very low velocity, also the efficiency of gear drives are very high and it can be upto 99% in case of spur gears [9]. In spur gears, teeth are cut parallel to the axis of the shaft, spur gears are used only when the shafts are parallel. An assembly of gears is to be designed for a designed torque of 41.33 Nm, taking a service factor (C_s) of 1.8, considering continuous operation under medium shocks.

3.2.1 Minimum number of teeth

Relation between the number of teeth (Z_1 and Z_2) to determine the minimum number of teeth on gear of involute gear profile without interference is given by the following equation and the numbers of teeth ($Z_1 = Z_2$) are calculated as 14.

$$Z_1 = \frac{2 * K_1}{sin^2(\alpha)}$$

3.2.2 Material selection

The endurance strength of the tooth in a gear is the deciding factor, when the gear tooth is subjected to fluctuating forces. The gear material should have sufficient strength to resist failure due to breakage of the gear tooth. In the designing of the gear assembly C40 has been taken into consideration and used.

The Lewis form factor (y_{pinion}) and Lewis factor (Y_{pinion}) for 20^0 stub teeth system is given by

$$y_{pinion} = 0.11$$

$$Y_{pinion} = \pi * y_{pinion} = 0.34$$

3.2.3 Calculation of module

The module specifies the size of the gear tooth. In this case the central distance between the gear assembly is fixed (44 mm). Lewis equation that is used to determine the module is considered as the basic equation in the designing of gears. The driving force or tangential load at the pitch line is given by (Eq.1) i.e. F_t = 1878.7 N. Face width of gear (b = 9.5 * m) is used. Barth's formula for ordinary cut gears is used to calculate the velocity factor (C_v = 0.98). On calculation a module m = 2 is obtained.

$$F_t = \sigma_d * C_v * b * Y_{pinion} * m \quad (1)$$

$$F_t = \frac{2 * T_{d1}}{D_1}$$

$$C_v = \frac{3.05}{3.05 + v}$$

3.2.4 Checking under static load

The gear subjected to static load during its operation is required to be calculated, for this the induced stress in the gear profile is required to be determined and for the safe function of gear i.e. without failure during static loading, the induced stress must be less than the yield strength of the gear material. Here using (Eq.2), $\sigma_{induced}$ is 145 MPa which is less than 207 MPa, thus the gear does not fail under designed static loading condition.

$$\sigma_{induced} = \frac{F_t}{c_v b \pi y m} \quad (2)$$

3.2.5 Designing and checking under dynamic load

Buckingham equation (Eq.3) is used to find the maximum dynamic load in the gear tooth. Dynamic factor depending upon machining errors is calculated using (Eq.4) i.e. C = 387931.1 for which k_1 is 8.70 for 20^0 stub tooth, the measured error in action between gears, e = 0.025 mm. K_3 is taken as 20.67, on calculation the value of dynamic load is 1970 N.

$$F_d = F_t + \frac{K_3 * v * (C * b + F_t)}{K_3 v * + \sqrt{C * b + F_t}} \quad (3)$$

$$C = \frac{e}{k_1(\frac{1}{E_1} + \frac{1}{E_2})} \quad (4)$$

To check the design of gear under dynamic loading conditions, dynamic strength of the gear is required to be checked which follows (Eq.5). For the system to work safe under dynamic loading condition, the dynamic strength (F_s = 2674.4 N) should be greater than maximum dynamic load (F_d = 1970 N).

$$F_s = \sigma_d * b * Y * m \geq F_d \quad (5)$$

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3.2.6 Checking in endurance

The beam strength of teeth or the endurance strength of pinion is determined using (Eq.6), where σ_{en} = 1.75 * σ_d. On calculating Fen is 4757 N. Safety of the gear design is verified using (Eq.7).

$$F_{en} = \sigma_{en} * b * Y_{pinion} * m \tag{6}$$

$$\frac{F_{en}}{F_d} = 2.5 > C_s = 1.8 \tag{7}$$

3.2.7 Checking in wear

Wearing of gear profile is one of the important factor that is taken into consideration during design stage. The limiting load for wear is calculated using (Eq.8), where ratio factor i.e. Q is 1 and load stress factor which depends on the maximum fatigue limit is K = 0.3 using (Eq.9) and surface endurance limit is σ_{es} = 342.5 MPa.

$$F_w = D_1 * b * Q * K \geq F_d \tag{8}$$

$$K = \frac{\sigma^2{}_{es} * \sin\alpha}{1.4}\left[\frac{1}{E_1} + \frac{1}{E_2}\right] \tag{9}$$

On calculating, the limiting load for wear of gear i.e. F_w = 228.14 N which violates (Eq.8). In order to make the gear safe under wear, the maximum fatigue limit is required to be recalculated i.e. K' = 2.3, thus for the gear to work without wearing the gear should have the Brinell Hardness Number should be atleast 450 [8].

3.3 DESIGN OF SPRING

A spring is an elastic machine element, which deflects under the action of load and returns to its original shape once the load is removed. Among some of the important applications of springs, these are used to apply force and control motion; even these are used to absorb shocks and vibrations. Helical springs are the most popular and widely applied springs [9]. These helical springs are made usually circular in cross section and bent in the form of helix under applied load. Helical Compression Springs, of square ends and ground type are used. An axial force is taken into consideration to be applied on stepping platform, which weighs 2.5 kg (24.525 N).

For designing springs a total load of 80 kg (784.8 N) is taken into consideration. The maximum deflection is fixed upto 50mm under applied axial load. Springs used are made of Hard drawn ASTM A227, with spring index 8.5, m = 0.190, A = 1783, G = 78450 MPa [7]. Permissible shear stress is determined using (Eq.10), which is recommended by Indian Standard 4454-1981 [9]. Wahl factor (Eq.11) is used to find out resultant stresses (Eq.12) concentration in a spring i.e. K = 1.17235. The wire diameter of spring is calculated as d = 3.83 mm.

$$\tau = 0.5\, S_{ut} \tag{10}$$

$$S_{ut} = \frac{A}{d^m}$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \tag{11}$$

$$\tau = K\left[\frac{8PC}{\pi d^2}\right] \tag{12}$$

Numbers of active coils are determined using (Eq.13), and the total numbers of coils are 18, with a solid length of ~69 mm. For the free length of spring, deflection is calculated using (Eq.13), i.e. δ = 50.29 mm. Required spring rate, k = 7.5 N/mm is calculated using (Eq.13) and actual spring rate, k' = 7.95 N/mm is calculated using (Eq.13).

$$\delta = \frac{8PND^3}{Gd^4} \tag{13}$$

The designing of spring has a resulted in a spring of wire diameter of 3.83 mm, number of coils are 18, with ~ 120 mm of free length and actual spring rate is more 6% than required spring rate. Fig. 2, shows the equivalent stress distribution in the designed welding region.

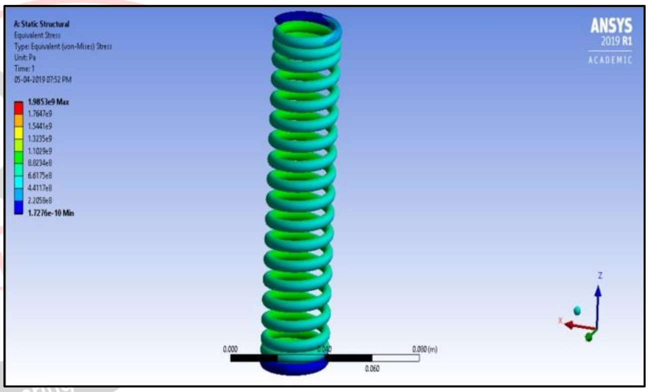

Figure 2: Equivalent stress distribution in the spring.

3.4 DESIGN OF WELD

Welding is a process of joining metallic parts by heating to a suitable temperature with or without the application of pressure. It is one of the economical and efficient methods for obtaining a permanent joint. A design of T joint weld of weld material 30C8 is required for a weld length 200 mm. This T-shaped weld is applied with an eccentric load of 400N at an eccentricity of 65 mm. According to MSST, τpermissble is 135 MPa. The primary shear stress (Eq.14) and bending stress (Eq.15) on the weld are 2/t N/mm2 and 43.333/t N/mm2, respectively. Throat of weld is calculated using (Eq.16).

$$\tau_1 = \frac{P}{A} \tag{14}$$

$$\sigma_b = \frac{M_b}{Z} \tag{15}$$

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$$\tau = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau_1^2} \tag{16}$$

The designed weld has a throat (t) of ~ 0.2 mm and leg (h) of ~ 0.3 mm. Fig. 3, shows the equivalent stress distribution in the designed welding region.

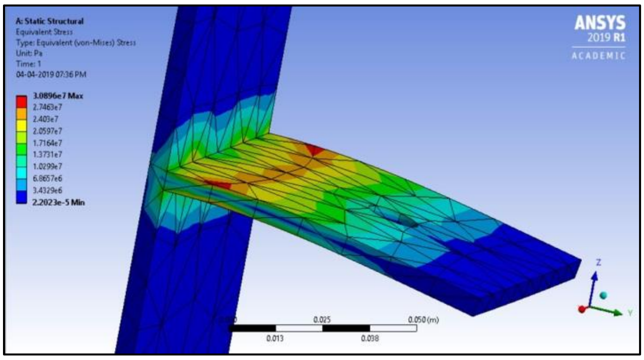

Figure 3: Equivalent stress distribution in the welding region.

IV. CONCLUSION

Designing of Mechanical Footstep Power Generator has provided an excellent opportunity and a means to gain experience through the applied knowledge. The system developed bridge the gates between institution and industries. The gear assembly has been designed with the factors like static loading, dynamic loading, endurance and wear. The finally manufactured gear should have a Brinell Hardness Number (BHN) of 450 and more in order to sustain wearing conditions. The designed spring of ASTM A227 provides actual spring rate 6% more than the required spring rate. Simulation of spring and weld shows the equivalent stress distribution in the safety range. The system does not depend on any power from the mains and also causes minimum to zero pollution. It is very useful for the places like all roads and as well as all kind of stair case, dense areas that could serve to generate the non-conventional energy like electricity, which can further be harnessed in any form. As the system does not require any non- renewable sources of energy, which further increases its advantages. Operation cost of footstep power generation system is nearly equal to zero. Only operational and maintenance is associated with this system.

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REFERENCES

[1] N. L. Panwar, S. C. Kaushik, and S. Kothari, "Role of renewable energy sources in environmental protection : A review," Renew. Sustain. Energy Rev., vol. 15, no. 3, pp. 1513–1524, 2011.

[2] Z. Peidong, Y. Yanli, and Z. Yonghong, "Opportunities and challenges for renewable energy policy in China," vol. 13, pp. 439–449, 2009.

[3] G. Dhanalakshmi, T. Manjulai, M. Mirunalini, and S. Sangeetha Mary, "Footstep Power Generation System," Int. J. Eng. Comput. Sci., vol. 2, no. 7, pp. 8–13, 2017.

[4] P. P. A. Bhosale, H. Shinde, R. Tahade, and M. Valani, "Design of Foot Step Power Energy Generation Machine," no. June, pp. 943–948, 2017.

[5] N. S. Shenck and J. A. Paradiso, "Energy scavenging with shoe-mounted piezoelectrics," IEEE Micro, vol. 21, no. 3, pp. 30–42, 2001.

[6] S. M. Tom Jose, Binoy Boban, "Electricity Generation From Footsteps; a Regenerative Energy Resource," Int. J. Sci. Res. Publ., vol. 3, no. 3, pp. 498–503, 2013.

[7] Standard specifications for mechanical springs, ASTM A227 .

[8] "Design Data Handbook by K. Mahadevan and K. B. Reddy is utilized, edition 4th (2015)"

[9] "Design of machine elements by V. B. Bhandari, Edition 2017"

[10] "Design Data – Data Book For Engineers, Edition 2005"

NOMENCLATURE

- f_t = Transmitted force
- f_0 = Normal force
- f_r = Resultant force
- α = Pressure angle
- y = Lewis form factor
- b = Face width of gear tooth
- d_0 and d_r = Outer and root diameter of pinion
- $A_d$andD_d = Addendum and dedendum
- P = Power
- T = Torque
- m = Module
- N = Number of revolutions

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- T_1 = Input torque
- T_{d1} = Design torque
- σ_d = Allowable stress
- y_{pinion} = Lewis form factor
- Y_{pinion} = Lewis factor
- F_t = Tangential load
- C_v = Velocity ratio
- v = Pitch line velocity
- D_1andD_2 = Diameter of pinion and gear
- $\sigma_{induced}$ = Induced stress
- F_d = Dynamic load
- C = Dynamic load carrying capacity
- F_{en} = Endurance strength of pinion
- σ_{en} = Endurance limit
- F_w = Limiting load for wear
- K = Load stress factor
- σ_{es} = Surface endurance limit
- τ_1 = Primary shear stress
- σ_b = Bending stress
- t = Throat of weld
- h = Size of weld
- D = Mean diameter of coil
- d = Wire diameter
- N = Number ofactive coils
- N_t = Total number of coils
- l_{solid} = Solid length of spring
- δ = Deflection of spring during compression
- G = Modulus of rigidity
- l_{free} = Free length of spring
- k = Spring rate
- τ = Permissible shear stress
- Z_1 and Z_2 = Minimum number of teeth of pinion and gear.
- τ_{max} = Maximum shear stress w.r.t. MSST.
- F_s = Dynamic strength of gear
- e = Measured error in action between gears

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