Modeling and Design of a Prototype Footstep Power Generating Machine

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Abstract

This paper presents a non-conventional method of electrical power generation through the design, modelling and fabrication of a footstep power generating system. It is configured to generate electricity as long as there is a foot force on it. The system consists mainly of a connecting rod, gears, u-shaped shaft, and an alternator. The component members were designed, while static and fatigue analyses were carried out using Autodesk Inventor software. The static and fatigue analysis of both spring and footboard revealed yield strength of 207 *Gpa* and a factor of safety of 12 for spring and 6.58 for footboard. Basic manufacturing processes were employed during the construction of the machine. The results show an output power of 0.912W, voltage of 1.52V, and an efficiency of 21% when an average mass of 62kg acted on it. The root mean square value greater than 0.9 shows a strong correlation of the predictive model.

Keywords

Power Generation, Footstep, Autodesk, Electrical Energy

1. Introduction

Electricity is the most convenient and versatile form of energy and in high demand, it is growing at a faster rate than other forms of energy and plays a crucial role in the industrial and agricultural sectors [1]. Electricity is most often generated at a power station by electromechanical systems, primarily driven by heat engines fueled by combustion or nuclear fission but also by other means such as the kinetic energy of flowing water and wind. Other energy sources include solar photovoltaic and geothermal power [2].

The fundamental principles of electricity generation were discovered in the 1820s and early 1830s by British scientist Michael Faraday [3]. His method is still in use today. Electricity can be generated by the movement of a loop of wire, or disc of copper between the poles of a magnet [4].

The consumption of electricity is an index of the productivity and growth of a country. Therefore, power development has been given high priority in development programs [5]. The increase of the human population in the country has necessitated and multiplied the need for use of energy to do work and to improve well-being than ever existed millions of years ago. As a result, many resources are being wasted with attendant consequences on health and the environment [6]. So, non-conventional energy is very essential at this time to any nation. Walking is a common activity performed by people daily. There is a loss in kinetic energy due to the weight transfer to the surface of the foot during walking. Therefore, the energy of the person from the footstep can be converted to electrical energy. Human power is an attractive energy source. Footstep power generating system is a nonconventional/renewable source of energy that reduces environmental pollution (such as air pollution caused by burning of fossil fuels and improves public health)

thereby reducing premature mortalities due to pollution and saving associated health costs [7].

2. Design Analysis and Calculations

2.1. Spring Design

The average mass (m) of humans is $m = 62\text{kg}$ [8].

From the century spring corporation (CSC) catalog standard for steel with a modulus of rigidity (G) of 80GPa, the outer diameter (OD) is 36mm. similarly; the inner diameter (ID) is 30mm

Figure 1. Spring Cross Section.

The spring wire diameter (d) in Figure 1 can be calculated using the formula

$$O.D = I.D + 2d \quad (1)$$

$$36 = 2d + 30$$

$$\rightarrow d = \frac{36 - 30}{2} = 3\text{mm}$$

$$d = 3\text{mm}$$

Force (F) = mass (m) $\times$ acceleration due to gravity (g)

$$F = mg \quad (2)$$

$$F = 62 \times 9.81 = 608.22\text{N}$$

Four springs are to be used in this design, so the average force that'll act on each spring is given as:

$$\text{Average Load} = \frac{608.22}{4} = 152.055\text{N}$$

Let the deflection resulting from the average load of 152.055N be 90mm

$$K = \frac{F}{\delta} \quad (3)$$

$$K = \frac{152.055}{90} = 1.7\text{N/mm}$$

where

$$K = \text{spring rate}, F = \text{load}; \delta = \text{deflection}$$

Number of coils (N) required for the spring is calculated thus:

$$K = \frac{Gd^4}{8 \times D^3 \times N} \rightarrow N = \frac{Gd^4}{8KD^3} \quad (4)$$

[9]

Where, mean diameter (D) = O.D – d = 33mm

O.D = outer diameter and d = wire diameter

$$\text{Hence, } N = \frac{80 \times 10^3 \times 3^4}{8 \times 1.7 \times 33^3} = 13.6$$

$\therefore$ Number of turns of the spring (N) = 14

The torque (T) acting on the springs is given as:

$$T = F \times D/2 \quad (5)$$

[9]

$$T = 608.22 \times 18$$

$$T = 10947.96 \text{ N} - \text{mm}$$

The shear stress in the spring is given thus:

$$\frac{T}{J} = \frac{\tau}{r} \quad (6)$$

[9]

$$\rightarrow \tau = \frac{Tr}{J}$$

where

$$J = \frac{\pi d^4}{32} = 7.95$$

J is polar moment of inertia of spring wire

$$r = D/2 = 18$$

r is the radius of wire diameter

$$\tau = \frac{10947.96 \times 18}{7.95} = 24787.83 \text{ N/mm}^2$$

2.2. Bearing Design

From the gear specifications, its approximate weight = 25.9N and the weight of the connecting rod = 5N. These weights constitute

Figure 2. Force Diagram.

the forces acting on the shaft on the bearing illustrated in Figure 2.

$$\sum \uparrow = R_A + R_B - 25.9 - 5$$

$$\rightarrow R_A + R_B = 30.9 \quad (7)$$

Where R_A and R_B are reactions at points A and B, respectively.

The moment about point A is given by:

$$\sum M_A = 0 \quad (8)$$

$$R_B \times 32 = 25.9 \times 1 + 5 \times 16$$

$$R_B = \frac{25.9 + 80}{32} = 3.31$$

Substitute $R_B = 3.31$ into equation (1)

$$R_A = 30.9 - R_B = 27.59 \text{ N}$$

Since R_A and R_B are 27.59 and 3.31 respectively. The bearing loads are small; therefore, the recommended bearing is a journal bearing. Table 1 shows the bearing specification.

Table 1. Specification of Ball Bearing R1804

Description	Value
Bore(d)	26mm
Outer Diameter (OD)	42mm
Width (B)	7mm

Ball No.	14
Ball Size	3.3mm
Dynamic Load Rating (C)	750
Static Load Rating (C ₀)	480

$$\text{Angle of Contact } C_0 = f_0 i Z D^2 \cos \alpha \quad (9)$$

[10]

Where;

i= Number of rows of balls in any one bearing = 1

Z = Number of ball per row = 14

D= Diameter of balls = 3.3mm

 α = Angle of contact f_0 = A factor depending upon the type of bearing = 3.33 for ball bearing

$$\cos \alpha = \frac{C_0}{f_0 \times i \times z \times D^2}$$

$$\cos \alpha = \frac{480}{3.33 \times 1 \times 14 \times 3.3^2}$$

$$\alpha = 19.56^\circ$$

The life of the bearing is given by the formula below

$$L_1 = \left(\frac{C}{W_1} \right)^3 \times 10^6 \text{ revolutions} \quad (10)$$

[9]

where,

L = bearing life

C = dynamic load rating = 750

W = dynamic load

k = 3 for ball bearing.

For Section A:

W = R_A = 27.59N

$$L = \left(\frac{C}{W} \right)^3 \times 10^6 \text{ revolutions} = \left(\frac{750}{27.59} \right)^3 \times 10^6 \text{ revolutions} = 2.0 \times 10^{10} \text{ revolutions}$$

[9]

For Section B:

$$W = R_B = 3.31$$

$$L = \left(\frac{C}{W} \right)^k \times 10^6 \text{ revolutions}$$

$$L = \left(\frac{750}{3.31} \right)^3 \times 10^6 \text{ revolutions}$$

$$= 1.16 \times 10^{13} \text{ revolutions}$$

The resulting calculation above shows that the bearing chosen is suitable for the project as it has many revolutions to undergo before failure and the bearings used are journal bearings because it is readily available.

2.3. Shaft Design

The specifications of the shaft below were assumed with respect to its availability and the resulting deflection calculated

Let the length of the shaft (L) be 320mm

Diameter of the shaft = 24mm

Selected shaft material is steel.

Modulus of Rigidity of steel (G) = 80 GPa

The deflection that the shaft will experience is given by the equation below

$$\delta = \frac{FL^3}{3GI} \quad (11)$$

[10]

where;

F = W = shaft load

$$\begin{aligned} \text{Moment of Inertia } I &= \frac{\pi d^4}{64} = \frac{\pi \times 24^4}{64} = 16286.02 \\ \therefore \delta &= \frac{30.9 \times 320^3}{3 \times 80 \times 10^3 \times 16286.02} = 0.259 \text{ mm} \end{aligned}$$

This means that the material of the shaft is suitable for the project because its deflection as a result of the weight is within limit.

The torque T acting on the shaft is given by the equation

$$T = F \times d/2$$

[10]

Where the force F on the shaft is 30.9

$$= 30.9 \times 12 = 370 \text{ Nmm}$$

The shear stress is calculated using the formula:

$$\frac{T}{J} = \frac{\tau}{r}$$

[9]

$$\rightarrow \tau = \frac{Tr}{J}$$

Where,

$$J = \frac{\pi d^4}{32} = 32572.03$$

Hence,

$$\tau = \frac{370.8 \times 24/2}{32572.03} = 0.1366 \text{ N/mm}^2$$

To determine depth value of the shaft “h”. The maximum compression allowable on the spring is 120mm. So, let the maximum vertical deflection of the spring be $X = 120 \text{ mm}$. When rotated, the u-shaped shaft creates a virtual circle.

NOTE: The depth of shaft was determined with respect to the extent of compression of the spring, this is in order to enable the shaft make full revolutions.

Now, let points A and B be just a little more than 180° say 185° apart. The angular distance S (length of arc subtended by 185° of the circle) between these points should be less than or equal to the maximum spring deflection.

Hence, calculating for $h = r$

$$\begin{aligned} X &= S = 120 \text{ mm} \\ \theta &= 185^\circ \text{ and } S = \text{length of arc} \\ \therefore S &= \frac{\theta}{360} \times 2\pi r \\ \rightarrow r &= \frac{360 \times S}{2\pi\theta} = \frac{360 \times 120}{2 \times \pi \times 180} = 37.165 \text{ mm} \end{aligned} \quad (12)$$

Therefore,

$$h = r = 37.165 \text{ mm}$$

It is important to note that the vertical deflection of the spring is tactically calculated to make the shaft rotate to approximately 185° , the spring compression stops here and the completion of the shaft rotation is achieved by the spring

return, thereby completing a revolution of 360° .

2.4. Gear Design

The aim is to get a gear ratio of 4:1, the specifications of gears were chosen due to its availability

For the driving gear,

Number of teeth = 36

Pitch circle diameter = 106mm

Inner diameter = 24mm

For the driven gear,

Number of teeth = 9

Pitch circle diameter = 24mm

Inner diameter = 10mm

$$\text{Circular Pitch } (P_c) = \pi D/T \quad (13)$$

Where,

D = Diameter of the pitch circle = 106mm

T = Number of teeth on the wheel = 36

$$\therefore P_c = \frac{\pi \times 106}{36} = 9.25\text{mm}$$

$$\text{Diametral Pitch } (P_d) = \frac{T}{D}$$

$$\therefore P_d = \frac{36}{106} = 0.34 \text{ teeth/mm}$$

$$\text{Module } (M) = \frac{D}{T} \quad (14)$$

$$\therefore M = \frac{106}{36} = 3\text{mm}$$

Table 2 shows the standard proportion for gear systems of 20° full depth involute.

Table 2. Standard Proportion for Gear Systems of 20° Full Depth Involute

S/n	Particulars	20° full depth involute system
1	Addendum	$1 \times 3 = 3\text{mm}$
2	Dedendum	$1.25 \times 3 = 3.75\text{mm}$
3	Working depth	$2 \times 3 = 6\text{mm}$
4	Minimum total depth	$2.25 \times 3 = 6.75\text{mm}$
5	Tooth thickness	$1.5708 \times 3 = 4.71\text{mm}$
6	Minimum clearance	$0.25 \times 3 = 0.75\text{mm}$
7	Fillet radius at root	$0.4 \times 3 = 1.2\text{mm}$

$$\text{Velocity ratio } (VR) = \frac{T_1}{T_2} \quad (15)$$

where T_1 = Number of teeth of small gear = 9

T_2 = Number of teeth of big gear = 36

$$VR = \frac{9}{36} = 0.25$$

Check for interference

$$T_1 \geq \frac{2aw \frac{1}{T_2} P_d}{\sqrt{1 + \frac{1}{T_2} \left(\frac{1}{T_2} + 2 \right) \sin^2 \phi} - 1} \quad (16)$$

where

T_1 = the number of teeth of the small gear

$$P_d = 0.34 \text{ teeth/mm}$$

$$aw = M = \text{Module} = 3\text{mm}$$

$$\varphi = 20^\circ$$

$$T_1 \geq \frac{2 \times 3 \times \frac{1}{36} \times 0.34}{\sqrt{1 + \frac{1}{36} \left(\frac{1}{36} + 2 \right) \sin^2 20} - 1}$$

$$T_1 \geq 0.7$$

This means that there is no interference.

3. Fabrication and Testing

The 3-D drawing and constructed figure of the footstep power generating machine is shown in Figure 3 (a) and Figure 3 (b)

Figure 3. Footstep Power Generating Machine (a) 3D model (b) Fabricated.

The design of the footstep power generating machine includes essential components such as: The foot board where the force is applied, helical compression spring to return the system to its original position when the load is removed, shaft to transmit power to the gear. Connecting rod to convert reciprocating motion into rotational motion. Other equipment includes: gears, bearing, D.C. generator, multimeter and light emitting diode (LED) bulbs. The selected materials are discussed in Table 3.

Table 3. Materials Listing and Specifications

<i>Name of component</i>	<i>Specification (mm)</i>	<i>Material used</i>
Plate	400 x300x4	Mild Steel
Base	400x300x4	Mild Steel
Connecting rod	Length = 120 Bore = 24	Steel
Spring	Outer diameter = 36 Inner diameter = 30 Wire diameter = 3	Steel
Driving gear	- No of teeth = 36 - Type - spur gear	Steel

Driven gear	- No of teeth = 9 - Type - spur gear	Mild steel
U shaped shaft	- length = 320mm - thickness = 20mm	Steel
Straight shaft	- length = 200mm - diameter = 10mm	Steel
D. C generator	- Type - dynamo - voltage = 12 v	
LED bulb	-voltage = 3 v	

4. Results and Discussion

The results got from the design calculations are summarized in Table 4.

Table 4. Design Results

Component	Design Results
Spring	Spring rate = 1.7N/mm Number of turns = 14 Torque = 10947.96N-mm Shear stress = 24787.83N/mm ²
Shaft	Deflection = 0.119mm Torque = 146.4N-mm Shear stress = 0.0539N/mm ²
Bearing	Angle of contact = 20° Reaction on bearing A = 9.47N Reaction on bearing B = 2.73N Life of bearing A = 4.96×10^6 revolutions Life of bearing B = 2.07×10^{13} revolutions
Gear	Module = 3 velocity ratio = 0.25 Interference = 0.7

4.1. Static Analysis of Spring and Footboard

The static analysis of some of the components of the footstep power generating machine was done. The results got from the analysis of the necessary components are presented in tables and graphs.

4.2. Spring Analysis Results

The spring's physical properties, mechanical properties and force distributions in the X, Y and Z co-ordinates are shown in Tables 5 and 6 respectively.

Table 5. Physical & Mechanical Properties of Spring

Material	Steel
Density	7.85 g/cm ³
Mass	0.145351 kg
Area	21,777.9 mm ²
Volume	18,516 mm ³
Center of Gravity	$x = 0.000223447$ mm $y = 135.002$ mm $z = -0.29917$ mm
Yield Strength	207 MPa

Ultimate Tensile Strength	345 MPa
Young's Modulus	210 GPa
Poisson's Ratio	0.3
Shear Modulus	80.7692 GPa

Table 6. Spring Force Values

Load Type	Force
Magnitude	245.25 N
Vector X	22.628 N
Vector Y	-236.883 N
Vector Z	59.348 N

4.2.1 Von Mises Stress

The figure below shows the von Mises stress in different regions of the spring when a load of 100kg is placed on the foot bar. The Von Mises stress of the spring is shown in Figure 4.

Figure 4. Von Mises Stress in the spring.

The Finite Element Analysis done using Autodesk Inventor software is summarized in Table 7.

Table 7. Spring Static Analysis Summary

Name	Minimum	Maximum
Volume	18,516.3 mm ³	
Mass	0.145353 kg	
Von Mises Stress	0.00869094 MPa	3.11667 MPa
1 st Principal Stress	-0.137906 MPa	1.86763 MPa
3 rd Principal Stress	-1.73083 MPa	0.150263 MPa
Displacement	0 mm	0.000273822 mm
Safety Factor	15	15
Stress XX	-0.988447 MPa	0.990931 MPa
Stress XY	-1.73781 MPa	0.287367 MPa
Stress XZ	-0.528229 MPa	0.542401 MPa
Stress YY	-0.578698 MPa	0.584505 MPa
Stress YZ	-0.464399 MPa	0.412399 MPa
Stress ZZ	-0.969968 MPa	0.988355 MPa
X Displacement	-1.0156 × 10 ⁻⁴ mm	1.0238 × 10 ⁻⁴ mm
Y Displacement	-2.7381 × 10 ⁻³ mm	6.8791 × 10 ⁻⁶ mm
Z Displacement	-7.842 × 10 ⁻⁵ mm	7.8827 × 10 ⁻⁵ mm
Equiv. Strain	3.6103 × 10 ⁻⁸	1.2863 × 10 ⁻⁵

1 st Principal Strain	3.0676×10^{-8}	1.1330×10^{-5}
3 rd Principal Strain	-1.0945×10^{-5}	-2.625×10^{-8}
Strain XX	-4.6810×10^{-6}	4.6484×10^{-6}
Strain XY	-1.0757×10^{-5}	1.7789×10^{-6}
Strain XZ	-3.2699×10^{-6}	3.3577×10^{-6}
Strain YY	-2.1195×10^{-6}	2.1411×10^{-6}
Strain YZ	-2.8748×10^{-6}	2.5529×10^{-6}
Strain ZZ	-4.5782×10^{-6}	4.6560×10^{-6}

4.2.2 Footboard Analysis

The footboard's physical, mechanical properties and force distributions in the X, Y and Z co-ordinates are listed in Tables 8 and 9 respectively.

Table 8. Physical & Mechanical Properties of Footboard

Material	Steel, Mild
Density	7.85 g/cm^3
Mass	5.19984 kg
Area	$337,760 \text{ mm}^2$
Volume	$662,400 \text{ mm}^3$
Center of Gravity	$x = 230 \text{ mm}$ $y = 180 \text{ mm}$ $z = -2 \text{ mm}$
Yield Strength	207 MPa
Ultimate Tensile Strength	345 MPa
Young's Modulus	220 GPa
Poisson's Ratio	0.275
Shear Modulus	86.2745 GPa

Table 9. Footboard Force Values

Load Type	Force
Magnitude	981.000 N
Vector X	0.000 N
Vector Y	0.000 N
Vector Z	-981.000 N

The Von Mises stress distribution over the footboard on application of 981 N force (100kg) is shown in figure 5

Figure 5. Von Misses Stress Distribution in Footboard.

The corresponding displacement distribution is shown in Figure 6.

Figure 6. Footboard Factor of Safety.

The Finite Element Analysis is summarized in Table 10.

Table 10. Footboard Static Analysis Summary

Name	Minimum	Maximum
Volume	662,400 mm ³	
Mass	5.19984 kg	
Von Mises Stress	0.176792 MPa	31.4377 MPa
1 st Principal Stress	-15.5508 MPa	40.3576 MPa
3 rd Principal Stress	-41.2078 MPa	15.2345 MPa
Displacement	0 mm	0.587365 mm
Safety Factor	6.585	15
Stress XX	-41.1502 MPa	40.3042 MPa
Stress XY	-2.75368 MPa	2.7746 MPa
Stress XZ	-2.06052 MPa	1.95488 MPa
Stress YY	-15.6087 MPa	15.2878 MPa
Stress YZ	-0.997642 MPa	0.758438 MPa
Stress ZZ	-15.6087 MPa	15.2878 MPa
X Displacement	-7.9677 × 10 ³ mm	7.9663 × 10 ³ mm
Y Displacement	-1.8009 × 10 ⁻³ mm	1.7989 × 10 ⁻³ mm
Z Displacement	-0.587365 mm	0 mm
Equiv. Strain	7.0808 × 10 ⁻⁷	1.2864 × 10 ⁻⁴
1 st Principal Strain	9.8645 × 10 ⁻⁷	1.5024 × 10 ⁻⁴
3 rd Principal Strain	-1.4953 × 10 ⁻⁴	2.6521 × 10 ⁻⁶
Strain XX	-1.4914 × 10 ⁻⁴	1.5024 × 10 ⁻⁴
Strain XY	-1.5959 × 10 ⁻⁵	1.6080 × 10 ⁻⁵
Strain XZ	-1.1942 × 10 ⁻⁵	1.1329 × 10 ⁻⁵
Strain YY	-2.5496 × 10 ⁻⁵	2.4479 × 10 ⁻⁵
Strain YZ	-5.7818 × 10 ⁻⁶	4.3955 × 10 ⁻⁶
Strain ZZ	-5.6327 × 10 ⁻⁵	5.2795 × 10 ⁻⁵

The corresponding factor of safety distribution is shown in Figure 7.

Figure 7. Footboard Factor of Safety.

4.3. Experimentation and Evaluation

In this section, a detailed evaluation of experimental test conducted on the manufactured footstep generator is presented. Based on the test results, a correlation was carried out to generate a predictive model for the prediction of voltage and power beyond experimental limits. The power input and output evaluations are presented in sections 4.3.1 and 4.3.2 respectively.

4.3.1 Power Input

Let us consider,

The average mass of human moving over the machine = 62kg [8]

The deflection caused by 62kg = 9cm

$$\text{Input power} = \text{workdone}/\text{time} \quad (17)$$

$$\text{Workdone} = \text{force} \times \text{distance}$$

$$\text{Force} = \text{mass} (a) \times \text{accl. due to gravity} (g)$$

$$\text{Force} = 62 \times 9.81 = 608.22 \text{ N}$$

Hence,

$$\text{Workdone} = 608.22 \times 0.09 = 54.73 \text{ Nm}$$

$$\text{Input power} = \frac{54.73}{60} = 0.912 \text{ watts}$$

4.3.2 Power Output

$$\text{Power output} = \text{current} (I) \times \text{voltage} (V) \quad (18)$$

$$\text{But } V = IR \text{ (Where } R = \text{resistance)}$$

Therefore,

$$\text{Power} = \frac{V^2}{R}$$

The dynamo used has specification of 6V, 3 watts

$$R = \frac{V^2}{P} = \frac{6^2}{3} = 12\Omega$$

But the maximum voltage produced = 1.52V

Therefore,

$$\text{Power} = IV = \frac{V^2}{R} = \frac{1.52^2}{12} = 0.193 \text{ watts}$$

$$\text{Efficiency} = \frac{\text{Output power}}{\text{Input power}} \times 100\% \quad (19)$$

$$\eta = \frac{0.913}{0.912} \times 100 = 21.162\%$$

The values of the corresponding load, voltage and power readings from the experiment are displayed in Table 11 and Figure 8 respectively.

Table 11. Corresponding Load, Voltage and Power Values

Load (kg)	Voltage (V)	Power (W)
50	0.5	0.021
62	0.80	0.053
75	1.52	0.193
85	1.60	0.213
100	1.60	0.213

From Figure 8, an increase in load on the footstep power machine shows direct proportionality from 50 *kg* to 80 *kg* but differs beyond 80 *kg* as any increase in load shows no appreciable increase in neither voltage nor power. The above phenomena are attributed to the capacity of the installed alternator, since it cannot deliver beyond 1.6 v or 0.22 *watts*. Figures 9 and 10 represent the curve fitting for the generated voltage and power out puts, where predictive models were generated to give an idea of the performance of the footstep machine beyond experimental boundaries. Figure 11 shows the interrelationship between voltage, power and efficiency and how they influence each other with significant level of 0.05. The root mean square values for both voltage and power output greater than 0.9 shows a strong correlation.

Figure 8. Experimental Results of Power and Voltage output due to applied load.

Figure 9. Curve fitting for Voltage output due to applied load.

Figure 10. Curve fitting for Power Output due to applied load.

Figure 11. Curve fitting for Power Output due to applied load.

5. Conclusion

The static analysis of both the spring and footboard reveals that an increase in load results in an increase in the von Mises stress, displacement, 1st Principal stress, and 3rd Principal stress of the material. Also, the factor of safety of the material was found to inversely proportional to load (see Figures 5, 6 and 7)

The yield strength and maximum stress of the footboard when a force of 981 N was applied were 207MPa and 40.3576MPa, respectively. The values above indicate the safety of the footboard material (mild steel) since the stress in the footboard is less than the yield strength of the material with corresponding factor of safety of 6.58445.

As observed from Figure 7, points of maximum stress concentrations in the footboard are at the points where fixed constraints were applied to the board. The minimum stress in the footboard is at the point of application of the load. As shown in Figure 6, there was no displacement at the points on the footboard where the fixed constraints were applied, but this displacement increases from zero at the points of fixed constraints to a maximum value of 0.5874 at the point of application of the load which is the point of maximum displacement from the fixed ends.

Steel has a yield strength of 207MPa. The spring used in this design is made from steel, and the maximum stress concentration in the spring when acted on by a 245.250N force as 3.11667MPa (that is the von Mises Stress). This is seen in Table 4. It is obvious that the material is in a safe zone. This is because the stress in the plate is less than the yield strength of the material.

In testing the setup, different weights were used on the prototype footstep power generating machine to get the output voltage using a multimeter. Table 11 shows the result of the experiments conducted on the prototype system. It was observed that the voltage and hence power readings kept changing with the change in the applied load, but this was discontinued from when a load of 85kg was used, at this load value, the corresponding voltage and power values remained constant. This was because, at this point, the spring had attained its maximum vertical deflection. Moreover, the capacity of the generator could be another contributing factor since it cannot generate power beyond its maximum capacity. Due to the above limitations, a correlation was carried out for predicting the voltage and corresponding power with both the R^2 -value greater than 0.9, indicating a strong correlation.

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