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Design and Analysis of Piezoelectric Energy Harvester for Wind Turbine

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Abstract - In this work, a scissor jack mechanism is proposed as a piezo energy harvester for wind turbine applications. The harvester works based on the principle of conversion of rotational motion of input shaft into linear vibration of the piezoelectric patch through a scotch yoke mechanism. The input force is magnified using a scissor jack mechanism. A mathematical model has been developed to compute the average electric power. The effect of parameters such as spring stiffness, angular velocity, scissor jack angle, the thickness of the piezoelectric bar, and the area of the piezoelectric bar has been considered on the generated power. Maximum power of $173.79\ W$ is obtained from the harvester at a wind speed of 7.5 m/s

Keywords: energy harvesting, piezoelectric material, scissor jack mechanism, scotch-yoke, wind turbine

I. INTRODUCTION

Continuously depleting traditional energy sources and their harmful effects on the environment have increased the attention of scientists to search for the alternative and environment-friendly sources of power generation. Harvesting the energy from the solar, wind, or water current through piezoelectric material are some of the alternatives. Similarly, small-power sensors, actuators, and wireless communications devices run on batteries [1]. Replacement of batteries in such low power based wireless sensors, implantable therapeutical tactics, and structural fitness vigilance system is a tedious task and sometimes impractical [2]. In such cases, a harvester can be used which on one hand scavenges or harvests the energy from ambient and other hand recharge the batteries. Piezoelectric harvesters are considered as having high energy density, ingenious design, convenient manufacturing processes, and affordable cost [3]. A windmill type harvester made of piezoelectric composite in the form of a cantilever was given by Tien and Goo [4]. To excite the harvester, an exciter teeth fan is attached to the hub windmill. The generated voltage and power from the prototype of the harvester are 26 V and 8.5 mW respectively. Khameneifar et al. [5] analyzed and designed a vibration-based energy generator using piezoelectric material for angular motion utilities. The rotation of the hub is converted into the vibration in the beam due to the gravitational effect of tip mass. Both PVDF and PZT have been used to harvest the energy at optimum load resistance. The maximum power of 6.4 mW calculated at shaft speed 138 rad/s using PZT material. Rezaei-Hosseinabadi et al. Kishore et al [6] depicted an ultra-low speed windmill. The windmill consists of a 72 mm diameter horizontal axis wind turbine rotor with 12 alternating polarity magnets around its periphery and a 60 mm × 20 mm × 0.7 mm piezoelectric bimorph element having a magnet at its tip. They reported that the electric power of 450 μW can be produced at a wind speed of 1.877 m/s. Wu et al. [7] proposed a cantilever crosswind energy harvester. The vortex shedding phenomenon has been used to generate vibration of the cantilever. Effect of length, location of PZT slabs, and proof mass on the power analyzed. At the resonant frequency, maximum output power 2W is calculated. Xie et al. [8] designed a ring piezoelectric energy harvester having an inner and outer concentric ring. The inner ring rotates with some frequency while the outer ring is stationary during the operation. Relative motions between these rings produce a periodic magnetic force on the piezoelectric patches and subsequently potential difference between the surfaces of the piezoelectric slabs. [9] evolved a piezoelectric harvester based on wind turbines without a contact vibration system. The results show that power 5274.8 W has been generated at a radius of 0.5 m of the harvester. Tao et al. [10] evolved a new design of harvester which extracts wind energy and converted it into mechanical vibration via a scotch yoke mechanism. Finally, periodic mechanical vibration generates electric power. Through simulation power, up to 150 W can be obtained at 7.2 m/s velocity of wind with a radius of blades 1 m. Narolia et al. [11] developed a parallel stationary and rotary plate coaxial piezoelectric energy harvester. To calculate the RMS value of power a simple mathematical model has been given and the effects of geometrical and process parameters on the harvester have been studied. It is observed that the optimum power output of 1.3572 W can be calculated with11 mm radius plate harvester. A new design of PVDF cantilever energy harvester excited by swirling airflow proposed by Stamatellou and Kalfas [12]. To generate the vibration through the swirling flow of air the PVDF cantilever is extended by a plastic sheet. When

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the harvester comes into the stream of swirling airflow it generates 3µW of power at 2 m/s flow of air with a load resistance of 150 Ω. Narolia et al. [13] proposed a new design of harvester based on piezoelectric, exited by attractive magnetic forces. To enhance the magnitude of the force on piezo patches a leverage system has been used. To determine the output power simulation has been carried out. The approximate RMS of electric power is 31.1432 W calculated.

A brief discussion about the literature shows that most of the energy harvesters are used to supply energy to small power-consuming devices meanwhile some of them can generate a huge amount of power at a reasonable rate. Hence, it is needed to develop a piezoelectric energy harvester having reliable, robust which is capable to harvest a sufficient amount of energy from wind or water current. Narolia et al. [14] presented a rotary piezoelectric energy harvester to harvest the energy from the rotary motion objects. They used a magnetic force to generate the strain in the PZT patch with d_{33} excitation mode. To verify the analytical model experiments were performed and results showed that the power of 14.48 nW can be harvested at 2100 rotation per minute with 0.3126 N magnetic force.

In this work, a design with a new concept of energy harvesting technique from rotating objects is proposed. The rotary motion of the wind or water turbine is transformed into the translating motion of a slotted rod via a scotch yoke mechanism. Both ends of the slotted rod are connected to the spring for producing vibrations of the piezoelectric scissor jack mechanism. The power yield from the wind or water turbine is directly transformed into an employed force on the PZT patches. The magnification of applied force is carried out through a scissor jack mechanism to achieve high-efficiency harvesting techniques.

Design and modeling of harvester

The proposed design and working principle of the harvester are illustrated in Fig. 1 (a) and (b). The major components of the harvester are a windmill, scotch yoke mechanism, two piezoelectric scissor jacks, and two springs with spring constant k_s . The scotch yoke mechanism as shown in Fig.2, is used to convert the rotary motion into a translated motion [15]. A sliding pair of slotted rod and cylinder mounted on the rotating wheel confirms the motion only in the normal direction to the shaft axis. When the shaft rotates with an angular velocity, ω, the eccentricity, Y, of the cylinder and wheel, displaces the end of the slotted rod from the starting point in time, t by an amount $Z_s = Y \sin \omega t$.

Fig. 3 (a) shows the piezoelectric-scissor jack mechanism consisting of four arms of uniform length. The lower ends of the upper two arms are hinged at fixed and sliding blocks for making turning pair and the upper ends of arms are hinged together to form self-turning pair. The material of the arms of a scissor jack is steel and the cross-section is square in plausibility with a side of the square is s and the length of the arm is l.

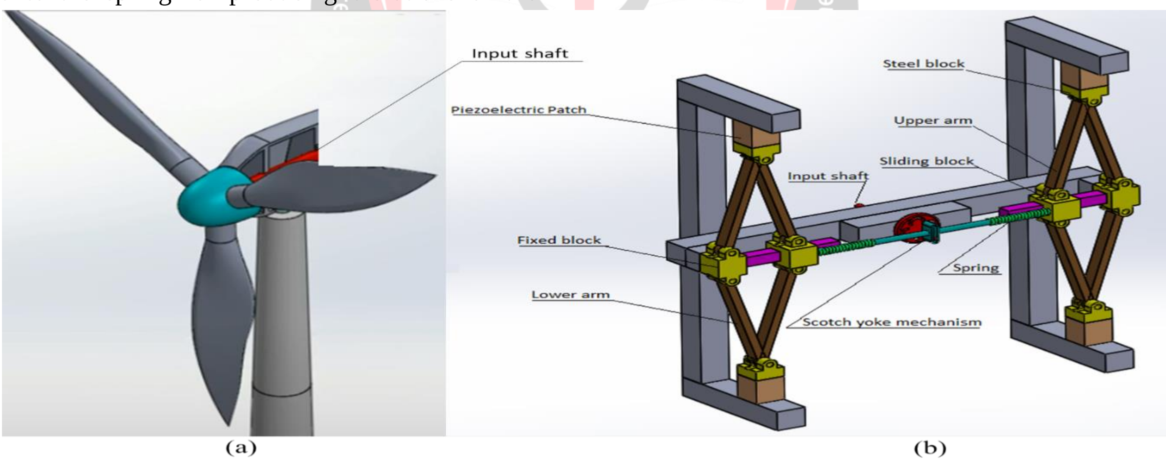

Fig. 1. Simple configuration of wind turbine and scissor jack mechanism energy harvester

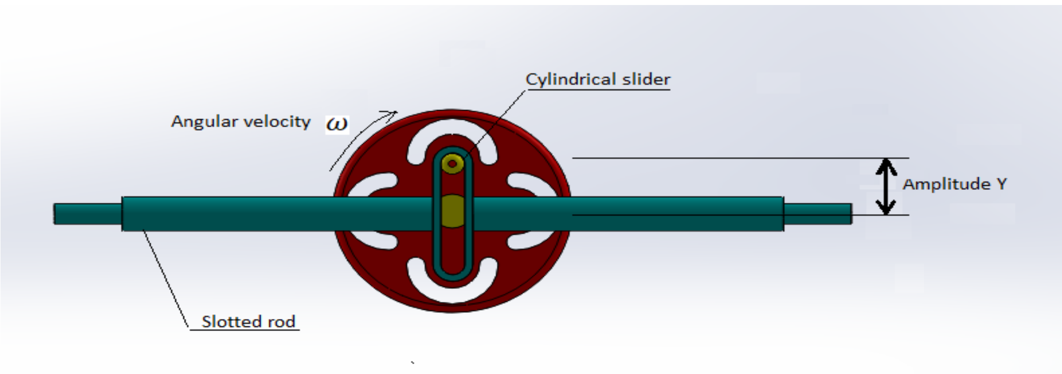

Fig. 2. Scotch yoke mechanism

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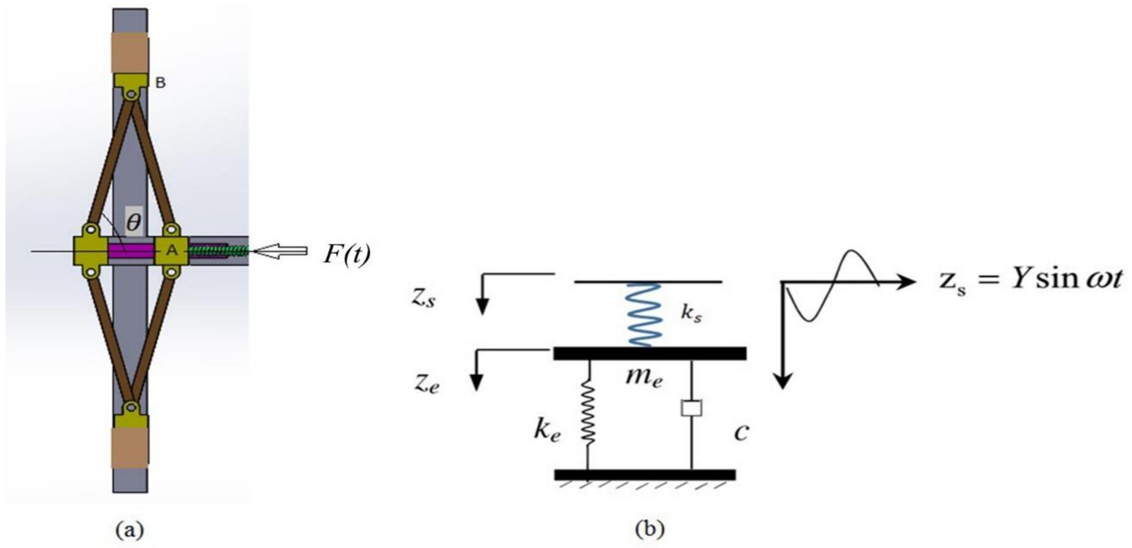

Fig. 3. (a) The scissor jack mechanism; (b) the equivalent damped mass-spring model.

The dimension of arms is selected as $s = 10\ mm$ and $l = 150\ mm$. A steel block of length a_b, width $b_{b,}$ height h_b, and young's modulus $E_{s,}$ is firmly bonded between the hinged pair of arms and the piezoelectric patch with length, width, height, and modulus of elasticity, of a_p, b_p, h_p, and E_p respectively. The rigid connection among the turning pair of arms, steel block, and piezoelectric patch make sure that there is no relative motion. The same configuration has been used for the lower arms of a scissor jack. Since both the springs of the slotted rod are linked to the piezoelectric scissor jack devices, the angular motion of the input shaft is then transformed into a spring force $F_s(t)$ at point A and amplified by $\tan\theta$, (θ = Angle between the axis of the slotted rod and arm of scissor jack) times at the point B on the PZT bar. The compressive/tensile force on the piezoelectric bar produces strain and then the electric power.

For modeling, the system, one of the piezoelectric scissor jacks is selected as an example and arms of the scissor jack are assumed to as rigid bodies having no deformation and considering a single-degree-spring-mass system with damping, as depicted in Fig. 3(b) [16], [17]. The spring force is magnified by the scissor jack mechanism and is applied to the piezoelectric patch via a steel block. The equivalent mass (m_e), spring stiffness (k_e), and coefficient of damping (c) can be calculated from the dimensions and material properties of the sliding block, steel block, and piezoelectric patch.

In the designed model, the steel block is firmly attached to the scissor jack at the hinge joint A and the piezoelectric patch at point B. The equivalently mass $m_{e,}$ of the sliding block can be calculated as:

$$m_e = [(2hd + \tfrac{\pi}{4}\, d^2)b_{so} + l_s(b_{so}h_{so} - b_{si}h_{si})]\rho_s, \quad (1)$$

where $h = 20\ mm$, is the height and $d = 20\ mm$, is the diameter of an extruded portion of sliding block $l_{s,} = 50\ mm, b_{so} = 40$ and $h_{so} = 40\ mm$, is the length, width, and height of the sliding block respectively. $b_{si} = 20\ mm, h_{si} = 20\ mm$, is the width and height of the slot inside the sliding block and ρ_s is the density of the sliding block.

The force acting at point B propel an elastic axial deformation of steel block and piezoelectric patch, the equivalent spring stiffness, k_e as shown in Fig. 3(b) can be calculated as [18]:

$$k_e = k_b k_p/(k_b + k_b), \qquad k_b = E_s a_b b_b/(\tan\theta^2\, h_s), \qquad k_p = E_p a_p b_p/\left(\tan\theta^2\, h_p\right) \quad (2)$$

where, E_p, k_p, E_s, k_b are modulus of elasticity and spring constant of the piezoelectric patch and steel block, respectively.

The magnitude of equivalent damping coefficient (c), can be represented as a summation of electrical and mechanical damping coefficients, as shown in Fig. 3 (b):

$$c = c_e + c_m \quad (3)$$

The electrical damping is due to the equivalent electric resistance offered by the piezoelectric patch during the conversion of work into electricity [19]. The magnitude of c_e can be given as [20]:

$$c_e = \tan\theta^2\, d_{33}^2 k_e^2/(\pi^2 c_v f) \quad (4)$$

Where, d_{33} is the strain coefficient of piezoelectric material in the direction of poling; c_v is the electric capacitance of the piezoelectric patch; f is the system's natural frequency.

With electrical damping, there is viscous air friction on the steel block when the vibration takes place and inherently internal structure damping force [21], a mechanical damping coefficient can be expressed as:

$$c_m = 2\zeta\sqrt{k_b m_e} \quad (5)$$

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where ζ is the ratio of damping.

After getting the value of mass m_e, spring constant k_e, and coefficient of damping c, the motion of the mass m_e can be expressed using Newton's second law as:

$$m_e\ddot{z}_e + c\dot{z}_e + k_e z_e = (z_s - z_e)k_s \tag{6}$$

Where,

z_s = slotted rod displacement

z_e = displacement of equivalent mass m_e

As we know that the translation pace of the slotted rod can be defined as $z_s = Y\sin\omega t$. Therefore, equation (6) can be written as:

$$m_e\ddot{z}_e + c\dot{z}_e + (k_e+k_s)z_e = z_s k_s = k_s Y\sin\omega t \tag{7}$$

The above expression considering as a single-spring -mass system with damping subjected to an equivalent sinusoidal force:

$$m_e\ddot{z}_e + c\dot{z}_e + kz_e = F\sin\omega t \tag{8}$$

where $k = k_e + k_s$ and $F = k_s Y$. The displacement of the equivalent mass m_e can be obtained as [22]:

$$z_e(t) = A\sin(\omega t - \phi) \tag{9}$$

where A = highest amplitude of $z_e(t)$;

ϕ = phase angle and given as:

$$A = \frac{F}{\sqrt{(k-m_e\omega^2)^2+(c\omega)^2}},\ \tan\phi = \frac{c\omega}{k-m_e\omega^2} \tag{10}$$

Hence the displacement $z(t)$, can be expressed as:

$$z(t) = z_s(t) - z_e(t) \tag{11}$$

The applied magnify force $F_m(t)$, on the piezoelectric patch can be calculated as:

$$F_m(t) = \tan\theta\left[k_s\left(z_s(t) - z_e(t)\right) - c\dot{z}_e(t)\right] \tag{12}$$

The yielding of periodic surface charge, voltage, and current from the piezoelectric patches can be given as [23]:

$$Q(t) = d_{33}F_m(t) = d_{33}\tan\theta\left[k_s\left(z_s(t) - z_e(t)\right) - c\dot{z}_e(t)\right] \tag{13a}$$

$$V(t) = d_{33}\tan\theta\left[k_s\left(z_s(t) - z_e(t)\right) - c\dot{z}_e(t)\right]/c_v \tag{13b}$$

$$I(t) = d_{33}\tan\theta\left[k_s\left(\dot{z}_s(t) - \dot{z}_e(t)\right) - c\ddot{z}_e(t)\right] \tag{13c}$$

where,

c_v = electric capacitance of the PZT material is given as: [13]

$$c_v = c_v' \times a_p \times b_p \times 0.0001/(0.01 \times 0.01 \times h_p) \tag{14}$$

The electrical power generated by every PZT bar of the harvester is given as:

$$P_e(t) = V(t).I(t) \tag{15}$$

$$P_e(t) = d_{33}^2(\tan\theta)^2\left[k_s\left(z_s(t) - z_e(t)\right) - c\dot{z}_e(t)\right]\left[k_s\left(\dot{z}_s(t) - \dot{z}_e(t)\right) - c\ddot{z}_e(t)\right]/c_v \tag{16}$$

In this generator there are four piezoelectric patches included, hence the power produced by four bars of PZT would be;

$$P_T(t) = 4d_{33}^2(\tan\theta)^2\left[k_s\left(z_s(t) - z_e(t)\right) - c\dot{z}_e(t)\right]\left[k_s\left(\dot{z}_s(t) - \dot{z}_e(t)\right) - c\ddot{z}_e(t)\right]/c_v \tag{17}$$

The root means square of the produced electric power during 0 to t could be expressed as:

$$P_T^{rms} = 4\sqrt{\frac{1}{\tau}\int_0^{\tau} P_e^2(t)dt} \tag{18}$$

Once the formulation mathematical model is completed, the generated voltage and current from the PZT bars can be calculated, and subsequently the root mean square of the power.

Considering that the piezoelectric patches are made of PZT-5H (lead zirconate titanate) while the steel blocks, sliding blocks, and springs are made of steel. Properties of the material used for the harvester and dimensions of the scissor jack device are given in Tables 1 and 2 respectively. A wind turbine model based on aerodynamic has been adopted to design an efficient and effective energy harvesting device. The output of mechanical power from the wind turbine, P_m, is given by [24]:

$$P_m = \frac{1}{2}\rho\pi R^2 v_W^3 C_P(\lambda) \tag{19}$$

Where ρ the density of the air, at NTP the value of this is equals $1.225\,kg/m^3$; R is the radius of the turbine; V_W is the speed of the wind; $C_P(\lambda)$ is the turbine's power coefficient, can be determined by the Tip-Speed Ratio λ. The tip-speed ratio λ is defined as the ratio of the blade tip speed to the wind speed V_W:

$$\lambda = \frac{R\omega}{v_W} \tag{20}$$

An empirical relation between the power coefficient C_p and λ can be given as [25]:

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$$C_P(\lambda) = 0.73\left(\frac{151}{\lambda} - 13.65\right)e^{\left(\frac{-18.4}{\lambda}+0.055\right)} \tag{21}$$

The rotation of the blades of the wind turbine (horizontal-axis) starts just as the input torque, T, dominates the total resistive torque due to generator and drive train, T_R. The cut-in speed is defined as the speed of the wind when the input and resistive torque become equal. A standard blade elementary theory has been used to determine the cut-in speed by Wood [26] expressed as:

$$U_C = \left(\frac{2T_R}{N\rho R^3 I_{cp}}\right)^{1/2} \tag{22}$$

Where U_C = Cut-in speed, T_{Rmax} = $k_s Y^2$ = Resistive torque, N = Number of blades, ρ = Air density, and I_{cp} = Chord-pitch integral.

For the same number of blades from Eq. (22), it is observed that, with increasing the value of T_R, the cut-in speed increases. Hence, to establish the wind turbine in regions of repugnant wind low value of T_R necessary to harvest the energy. Additionally, cut-in speed can be decreased by rising the radius and number of the blades for the same resistive torque.

Table 1 Dimensions and material properties of PZT-5H

K_{33}	d_{33}(C/N)	E_p (N/m^2)	a_p(mm)	b_p(mm)	h_p (mm)
0.78	6e-10	53e9	20	20	10-100
C'_v	0.375 for the PZT bar having dimensions of a_p= 0.01 m, b_p = 0.01 m, h_p = 0.0001 m,[7]				

Table 2 Dimensions and properties of the material of the scissor jack device.

E_s(N/m^2)	ρ_s (kg/m^3)	a_b(mm)	b_b (mm)	h_b (mm)	ζ	θ(degree)
210e9	8050	20	20	10	0.0017	70-80

Effect of Various Parameters on Harvested Power

After obtaining the relation of root mean square for electric power due to rotary motion, the effectiveness of the harvester has been investigated. The effect of the spring stiffness, eccentricity, scissor jack angle, speed of the turbine, the thickness of the PZT bar, and length (width) of the PZT bar on the output power has been studied. For this purpose, a wind turbine with three-blades and a blade tip radius of 1 m has been considered. The extracted mechanical power is calculated by the empirical relation as given in Eq. (19), using the aerodynamic model at various resistive torque. The effects of various parameters on harvested electrical power are studied.

1.1 Effect of Stiffness of Spring

Analytical formulation indicates that the execution of the harvester mainly depends on the stiffness of the spring and the eccentricity of the scotch yoke mechanism. It is considered that the wind turbine is situated in areas of repugnant wind with a cut-in speed of 4.75 m/s for a turbine with three blades of 1 m length [27]. Fig. 4 shows the influence of the spring stiffness k_s, on the output electric power corresponding to the scissor jack angle, speed of the turbine, thickness, and side of the PZT bar as 80 degrees, 50 rad/s, 100 mm, and 20 mm respectively. The generator resistive torque, $k_s Y^2$, is considered in the range between 0.4 N/m to 1.2 N/m. From Fig. 4, it is observed that electric power rises nonlinearly with an increase in the stiffness of the spring. At the resistive torque of 0.4 N/m, the RMS of the power varies from 2.006 W to 69.51 W, corresponding to a stiffness value between $100\ kN/m^2$ to $400\ kN/m^2$. Similarly, at the resistive torque of 1.2 N/m, the RMS of the power varies from 6.0019 W to 208.55 W with the same variation in stiffness of the spring.

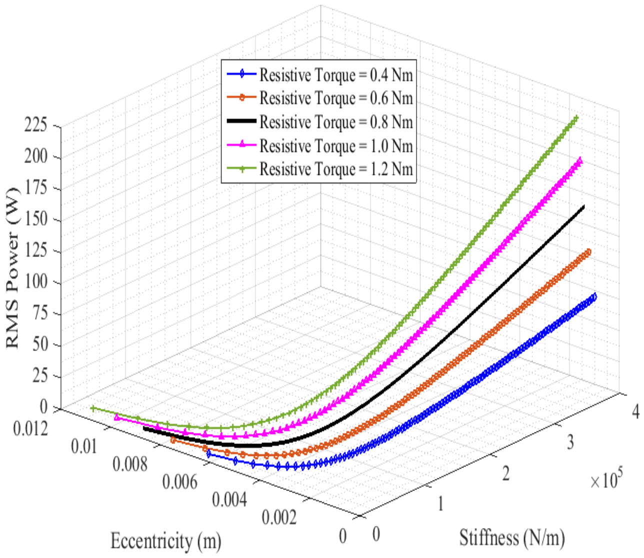

Fig. 4. Variation in RMS of electric power with spring stiffness

1.2 Effect of Speed of Input Shaft

Fig. 5, presents the variation of electric power with the angular speed of the input shaft i.e. wind speed. The spring stiffness, scissor jack angle, and thickness of piezoelectric patches, eccentricity, and width (length) of PZT bars are considered as $400\ kN/m$, $80\ degree$s, and $100mm, 1.6\ mm$ and $20\ mm$, respectively. It is

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observed that the power increases non-linearly from 0 W to 321.32 W with the increase in angular speed from 0 rad/s to 100 rad/s. This is due to fact that an increase in the angular speed of the input shaft leads to a reduction in the period of the applied sinusoidal force, resulting in an increase in the kinetic energy of the harvester and thus producing more electrical power. In the factual design of windmill, more power can be extracted from the wind, if a high value of power coefficient C_P is available. Considering the tip-speed ratio λ = 6.91, the highest magnitude of the $C_P = 0.44$, calculated from Eq. (21). At designed wind velocity, using Eq. (19) and (20), putting the above values, we can get the mechanical power P_m, for angular speed ω.

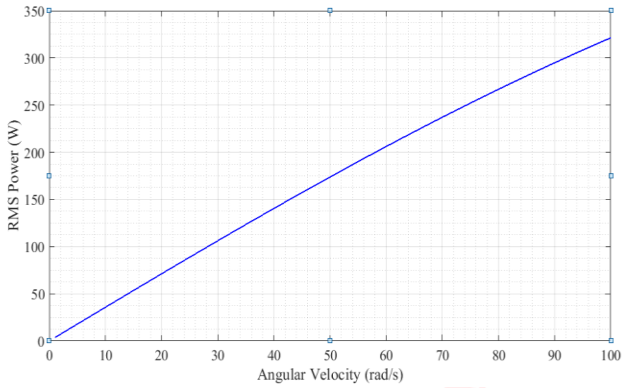

Fig. 5. Effects of Angular Velocity on RMS of electric Power

Rotation of the wind turbine initiates only when the wind velocity dominates the value of cut-in speed. For a particular wind turbine, the value of cut-in speed can be determined using the Eq. (22) at a particular resistive torque. The cut-in speed of windmill of a radius of 1 m, with three blades, taking the chord-pitch integral in Ref.[27], is calculated as 7.5 m/s when $T_R = 1Nm$. Wood [27] concluded that the above-estimated value of cut-in speed is approximate double, hence the actual value of cut-in speed is approximate 4 m/s. Eventually, the speed of wind selected in this work ranges between 5 m/s to 9 m/s to evaluate the efficiency of the harvester as shown in Fig. 6.

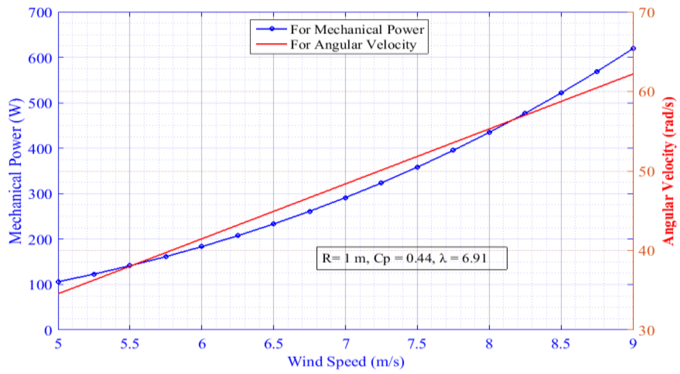

Fig. 6. Effects of wind speed on Mechanical power and angular velocity

In Fig. 6, the mechanical power curve indicates that, at optimal tip speed ratio, the power P_m varies from 106.13 W to 618.99 W when the velocity of wind ranging from 5 m/s to 9 m/s and correspondingly from the angular velocity curve, the angular velocity varies from 34.55 rad/s to 62.19 rad/s. After observing Fig. 5 and Fig.6, it is concluded that the mechanical power of the windmill perennially more than the RMS of output electric power for a particular angular velocity. Unlikely with increasing the angular velocity, the rate of increase of mechanical power is always higher than the rate of increase of electrical power with angular velocity. It shows, the harvester's efficiency is lower at higher angular velocity than at the lower values. The presence of the mechanical and electrical damping at high angular speed in the elements of the harvester and the lower resistive torque are the main reasons behind the low efficiency. To generate more electric power from the wind turbine, a larger blade radius and gearbox are recommended.

1.3 Effect of Scissor jack Angle

Variation in RMS of electric power with the tangent of the scissor jack angle $\tan\theta$ is shown in Fig.7. From Eqs. (13b), (13c) and (16) the electric power is proportional to the square of the tangent of the scissor jack angle. A slight increment in the angle gives a higher lead in the power generation from the harvester. It has been noted that with increasing scissor jack angle the amplitude of the vibration of block, B is decreased than the amplitude of vibration of block A. Fig. 8, represent the displacement of the mass m_e when the scissor jack angle $\theta = 80$ degree, spring stiffness = 400 kN/m, angular velocity = 50 rad/s and eccentricity = 1.6 mm. in Fig. 8, the maximum displacement is 1.12x10^{-4} m is sufficient to use 80 degree scissor jack angle in actual design which is not destroying the PZT bar.

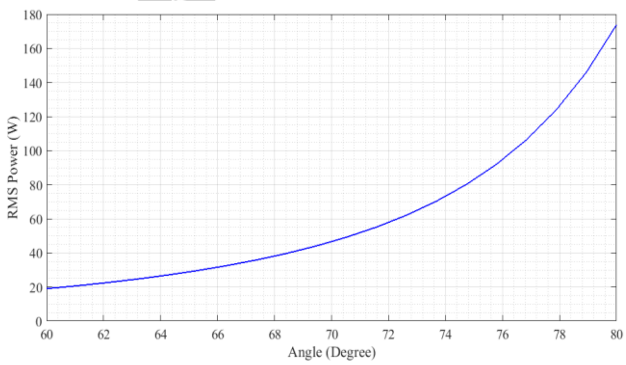

Fig. 7. Mechanical power versus scissor jack angle

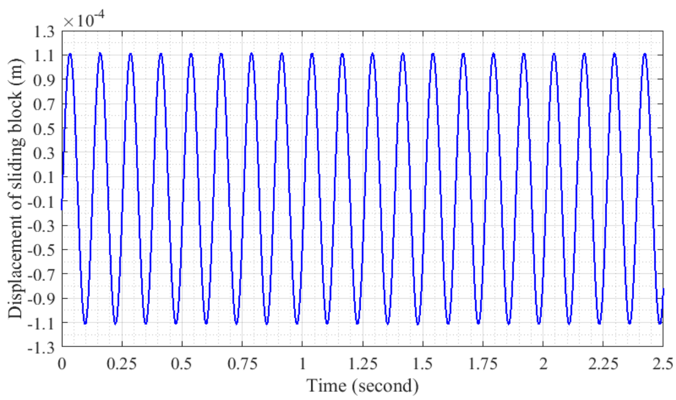

Fig. 8. Displacement of sliding block versus time

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1.4 Effect of thickness of PZT bar

The effect of the thickness of the piezoelectric bar on RMS of the electric power generated by the harvester is shown in Fig.9. Various parameters considered are: angular velocity ω = 50 rad/s, spring stiffness k_s = 400 kN/m, scissor jack angle = 80 degree and length (width) of PZT bar = 0.20 mm. The figure shows that the RMS of the power increase with the increase in the thickness of the piezoelectric bar. The efficacy of the thickness on the power can be interpreted with the help of Eq. (13b) in which power is proportional to the thickness of the PZT bar but from Eq. (2) the spring constant of PZT decreases with increasing thickness of power, hence the resultant effect of thickness as shown in Fig.9 i.e. almost linear variation.

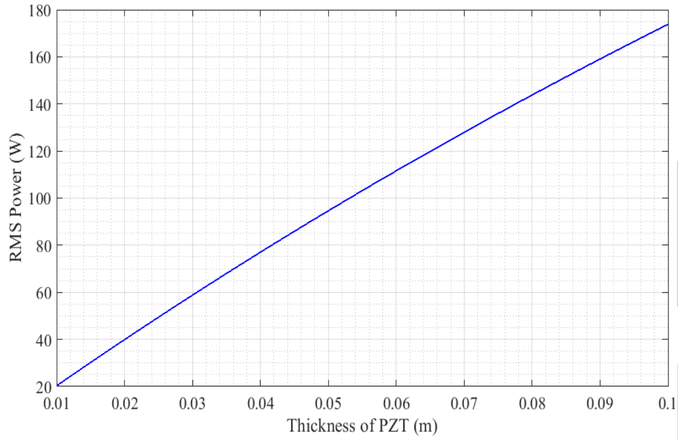

Fig. 9. RMS of generated power versus thickness of PZT bars

1.5 Effect of Length and Width of PZT bar

Fig. 10, demonstrates the variation in RMS of the generated electrical power with the length and width of the PZT bar. ω = 50 rad/s, spring stiffness k_s = 400 kN/m, scissor jack angle = 80 degree and thickness of PZT bar = 0.1 m was fixed for the analysis. It can be observed that the RMS value of the power reduces nonlinearly with an increase in the length and width of the piezoelectric bar. It is also observed that when the length and width increase from 0.015 m to 0.025 mm, the RMS of power decreases from 272.53 W to 118.22 W. From Eqs. (13b) and (16) it can be stated that with an increase in the length and width of the PZT bar, the electric capacity c_v increases which leads to a decrease of the electric voltage. The maximum stress induced into the PZT patch at the peak load as much as low as the yield point of the piezoelectric material [28], hence the theoretical analysis results are convincible.

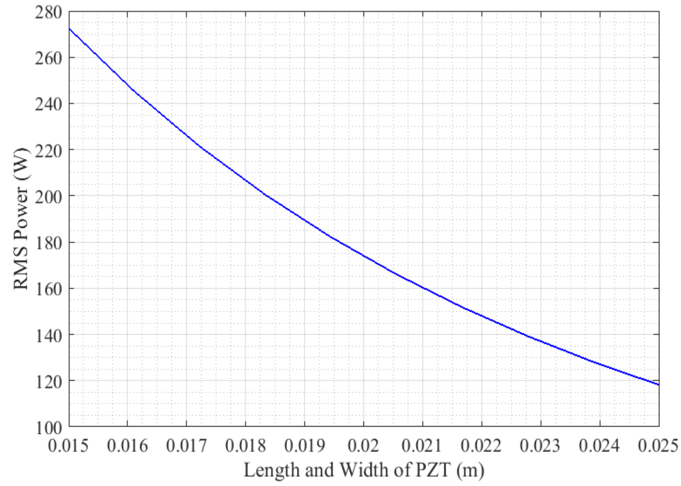

Fig. 10. Variation in RMS of electric power with length and width of PZT bars

II. CONCLUSION

A scissor jack mechanism piezoelectric energy harvester based on a wind turbine is designed to harvest energy from wind. A combination of scissor jack and scotch yoke mechanisms has been used to convert the rotational motion into a linear motion of the slotted block. The kinetic energy of wind produces a rotating thrust on the blade of the turbine, converted and magnified into a periodic force on the PZT bars. A mathematical model is formulated to obtain the output charge, voltage, current, and RMS of electric power from the PZT bars. The proposed scissor jack mechanism helps in magnifying the input force thus enhancing the harvested power also the cut-in speed has been considered in the design of the harvester. From the study, it can be concluded that the RMS value of the produced electric power increases with the increase in the spring stiffness, the velocity of rotation of the turbine shaft (wind speed), scissor jack angle, and thickness of piezoelectric patches and decreases with increase in length and width of the PZT bars. For harvester with geometric parameters $a_p = b_p = 20\,mm,\ h_p = 0.1\,m,\ k_s = 400\frac{kN}{m},\ \omega = 50\frac{rad}{s}, \theta = 80\,degree,$ eccentricity $= 1.6\,mm$ and the wind velocity of 7.5 m/s, the RMS of generated power reaches up to 173.79 W, which is not sufficient to meet the energy requirement of average households (2kW) [29]. In practice, it is proposed that a larger dimension of the harvester and/or large radius of the wind turbine rotor blade may be useful to meet the required energy for the household appliances.

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