Energy Harvesting From a Crane System by a Piezoelectric Energy Harvester: Two Concentrated Moving Masses Approach

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***Abstract*—Energy harvesting using a piezoelectric energy harvester (PEH) from the linear vibration of a crane system is presented in this paper. The PEH is designed in the form of a piezoelectricity lever-arm mechanism, and mounted in the midpoint of the top beam of the crane framework through a spring. Two concentrated moving masses traversing on the top beam of the crane framework excite the PEH and result in useful electrical energy. The numerical results show the electric power can be harvested up to 0.5 W, 37.5 W and 4.7 mW in axial, vertical and lateral displacements, respectively with a span of 12 m of the top beam of the crane framework at the trolley speed of 3.7 m/s, and the payload mass of 1200 kg.**

***Index Terms*—crane system, piezoelectric, lever-arm mechanism, electric power**

I. Introduction

Crane systems are widely used in ports, factories, construction sites and other places. They have tasks to do repetitive motions such as: hoisting, transporting (longitudinal or transverse motions), and lowering the heavy payload, as well as combination of each motion. As a combination of structural steel and mechanical system, they need to be designed and built rigidly to keep up the lifting capacity and productivity. However, as the lifting capacity and the size of crane systems are getting heavier and bigger, vibration becomes more pronounced during their operations. The source comes from when the payload is traversed or hoisted along the span of the top beam of crane framework, it will excite the crane framework resulting vibration where the magnitude and position vary in both time and space [1]. It affects significantly the productivity in terms of capability of transporting heavy payload.

Since the crane systems are massive available, then the vibration of the crane systems must be utilized. It can be a promising renewable energy to be harvested. Harvesting energy in dynamic systems has evoked great interests of numerous researchers in recent years [2]-[12]. To the best of authors's knowledge, the use of piezoelectric energy harvester (PEH) has not been performed and reported in any works related with harvesting of vibration of a crane system. Hence, there is an urgent need of developing piezoelectric energy harvester (PEH) for harvesting the vibration of the crane systems, and this paper is addressed for such a purpose.

The rest of this paper is organized as follows: Section 2 describes about the design and mathematical modeling of the system. Results and discussions are presented in Section 3. Finally, conclusion is put on Section 4.

II. Mathematical Modelling of a PEH Mounted on a Crane System

There are different types of crane such as overhead crane, gantry crane, tower crane and various special purpose cranes. However, their basic components are similar. In this paper, a gantry crane system is chosen for energy harvesting by piezoelectric energy harvester (PEH), where it is mounted onto the center point of top beam of the crane framework. Fig. 1 describes the overall system (a gantry crane with a PEH). The figure shows that energy harvesting from the gantry crane by a PEH can be seen a coupled mechanical-electrical system. It can be considered as two main structures, namely super- and sub-structure. The earlier is the gantry crane system while the latter is the PEH.

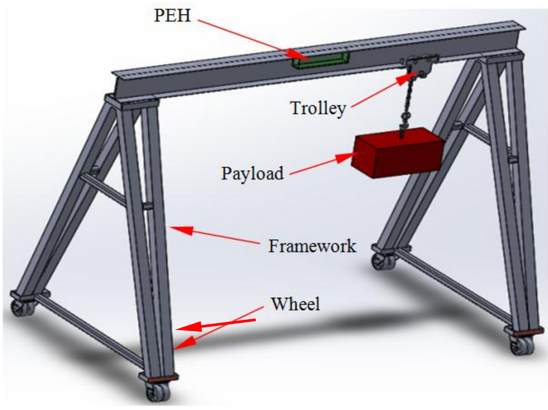

Figure 1. A gantry crane system with a PEH.

A. Design of PEH for a Crane System

Realistic model of the PEH is a lever-arm mechanism which is adopted from [3]-[9] and depicted in Fig. 2. This

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sub-structure is made from steel with a length, width and height of ℓ_b, w_b, h_b, respectively as well as material properties: mass density and Young's modulus of ρ_b, E_b, respectively. The figure illustrates that the lever-arm mechanism is designed in a cantilever beam which is hinged in a certain distance in such that it has arm length ratio, $n = \ell_1/\ell_2$, $n \geq 1$.

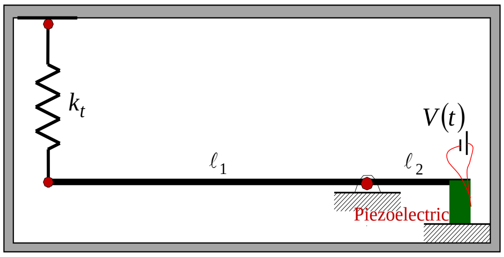


Figure 2. PEH for a crane system.

The ratio is designed in order such that it can magnify the excitation force in the short-moment arm with respect to the long-moment arm of the lever-arm mechanism. The magnified force will deform the piezoelectric bar which is firmly bonded in the end of short-moment arm. The spring stiffness of k_t is chosen in order such that it enhances the excitation force applied to the lever-arm mechanism. Deformation of the piezoelectric bar will generate the useful electrical power.

Dynamic of piezoelectricity lever-arm mechanism in Fig. 2 can be modelled as a translational lumped mass and shown in Fig. 3. Its equivalent mass, m_e comes from the material density, ρ_b and the cross-section, A_b of the cantilever beam. It can be calculated as,

$$m_e = \rho_b A_b \ell_1 n, \quad (1)$$

Equivalent spring constant, k_e in Fig. 3 is combination of spring constant of the cantilever beam and piezoelectric in series arrangement which is denoted by k_c and k_p, respectively. Spring constant of the cantilever, k_c is a flexural stiffness which depends on the Young's modulus of the material of the cantilever, E_c and cross-section property which is scaled by the arm length ratio, n.

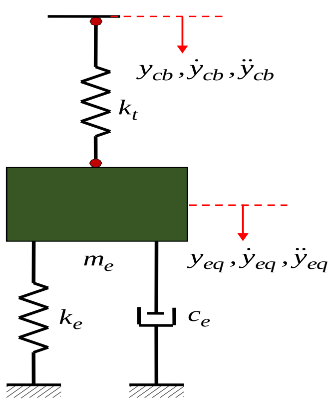


Figure 3. Equivalent model of lever-arm mechanism and piezoelectric bar.

Spring constant of for piezoelectric, k_p is an axial stiffness which depends on the Young's modulus of the material of the piezoelectric. Equivalent spring constant, k_e can be calculated as follows,

$$k_e = \frac{k_c k_p}{(k_c + k_p)}, k_c = \frac{E_c s_1 s_2^3}{4(\ell_1 n)^3}, k_p = \frac{E_p a b}{(n^2 h)}. \quad (2a)$$

Equivalent damping coefficient, c_e in Fig. 3 is a sum of the mechanical and electrical damping coefficients which is denoted by c_c and c_p, respectively. Damping coefficient, c_e can be calculated as follows,

$$c_e = c_c + c_p. \quad (2b)$$

B. Structural Dynamic of a Crane System

Based on Figs. 2-3, schematic model of PEH Mounted on a gantry crane system and its finite element (FE) model are depicted in Fig. 4. Structural dynamic of the overall system is modeled through moving finite element method (MFEM) which has been developed by [1] and [13]-[14] for crane systems. In deriving the MFE model, some simplifications are made for the sake of numerical approach such as:

1. Trolley and the payload are treated as two lumped masses which move in the top beam of the crane framework, and called as two concentrated moving masses. They are considered sufficient to represent the weight of trolley and payload which is spaced with a certain distance.
2. Since the type of moving load used is moving mass, then there is interaction between the moving load and the crane framework.
3. Dynamics of payload in terms of swing angle and hoist system are not considered in the mathematical model.

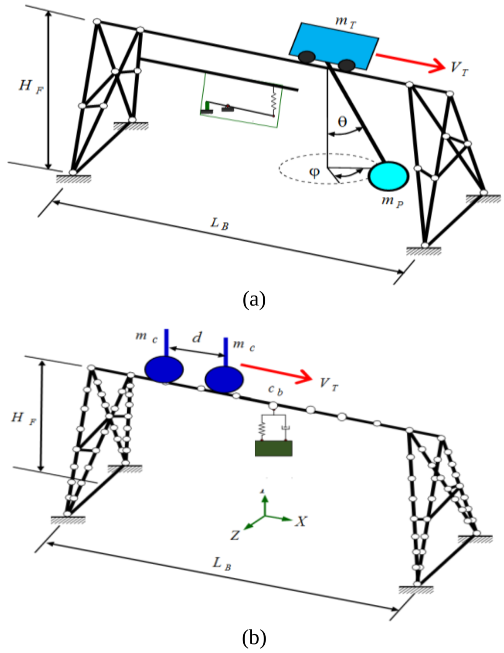


Figure 4. Model of gantry crane system with a PEH (a) schematic model (b) FE model of crane framework induced by two moving masses.

MFEM for gantry crane system has been derived by [1], however, it revisited here for a concise notation. Equation of motion for MDoF structural system, geometrically and materially linear dynamic is represented as follows,

$$[M_{st}]\{\ddot{q}_{st}(t)\}+[C_{st}]\{\dot{q}_{st}(t)\}+[K_{st}]\{q_{st}(t)\}=\{F_{st}\}(t). \quad (3)$$

Terms $[M_{st}],[C_{st}],[K_{st}]$ are representing global mass, damping and stiffness matrices of the crane framework, $\{q\}$ is the nodal displacement vector and its time derivative while $\{F_{st}(t)\}$ is the external force vector.

Since the masses traverses to the every node along the top beam of crane framework, make its position is time-variant as illustrated by Fig. 5. Terms $\{N_k\}_u$ $u=k=1,7$, $\{N_k\}_v$, $v=k=2,6,8,12$ and $\{N_k\}_w$, $w=k=3,5,9,11$ are shape functions associated with translation degree of freedoms in three directions axial (x), vertical (y) and lateral (z) and terms $\{d_{sk}\}_u$, $\{d_{sk}\}_v$, $\{d_{sk}\}_w$ are displacements in three directions. Terms $(u(x,t),v(x,t),w(x,t))$ depend on position x and time t. Terms $d_{si}=(i=1-12)$ are the displacements for the nodes of the space frame element at which the moving masses locate. Figure 5 suggests that the overall matrices property of (3) is time-dependent, and it can be rewritten as in (4). Details about the calculation of local position of the two concentrated moving masses, x_s from the left end of the *s*-th beam element, at time t, as seen from Fig. 4 in the length of the *s*-th beam element, ℓ_x can be referred in [14].

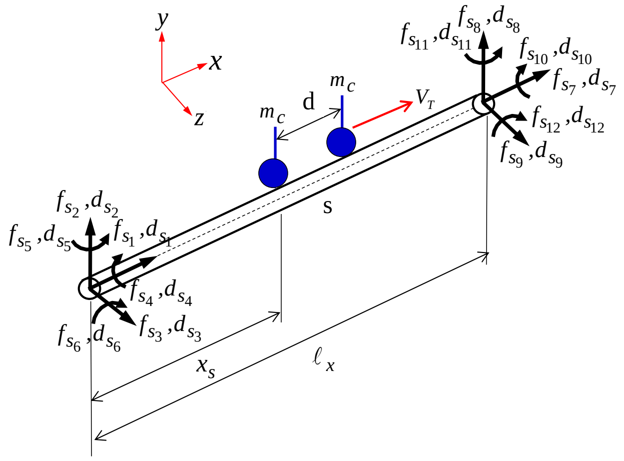


Figure 5. Equivalent nodal forces of the *s*-th space frame element.

$$\begin{aligned}&\left[[M_{st}]+m_c\left\{\{N_k\}^T\{N_k\}\right\}\right]\{\ddot{q}\}+\\&\left[[C_{st}]+2m_c\left(\dot{x}+\ddot{x}t\right)\left\{\{N_k\}^T\{N_k{'}\}\right\}\right]\{\dot{q}\}+\\&\left[\begin{array}{l}[K_{st}]+m_c\left(\dot{x}+\ddot{x}t\right)^2\left\{\{N_k\}^T\{N_k{''}\}\right\}+\\ m_c\ddot{x}\left\{\{N_k\}^T\{N_k{'}\}\right\}\end{array}\right]\{q\}\\&=\left\{\{N_k\}^T\right\}m_c g.\end{aligned} \quad (4)$$

C. Numerical Solver and Harvesting Formulations

The computational scheme for solving (4) is based on Newmark- β method of direct integration. The two parameters are selected as β =0.25 and γ =0.5, which implies a constant average acceleration with unconditional numerical stability. For each integration step, vibration crane framework can be calculated. Details of Newmark- β method can be referred to [1]. After obtaining displacement and velocity of the equivalent model of lever-arm mechanism and piezoelectric bar, electrical charge that can be stored by piezoelectric bar is calculated using following equation [3],

$$Q(t)=d_{33}n\left\{k_t\left(y_{cb}(t)-y_{eq}(t)\right)-c\,\dot{y}_{eq}(t)\right\}. \quad (5)$$

Equation (5) contains some important parameters such as: (a) k_t is stiffness coefficient of the linear spring (b) y_{cb} is displacement of the crane framework in the center point, c_b of the top beam, (c) y_{eq} and $\dot{y}_{eq}$ is displacement and velocity of equivalent model of lever-arm mechanism and piezoelectric bar, and (d) n is magnification factor (arm length ratio) as mentioned in the previous paragraph. Electrical current resulted by piezoelectric bar can be calculated by deriving the electrical charge in (5) with respect to time, and written in (6) as follows,

$$I(t)=d_{33}n\left\{k_t\left(\dot{y}_{cb}(t)-\dot{y}_{eq}(t)\right)-c\,\ddot{y}_{eq}(t)\right\}. \quad (6)$$

Electrical voltage resulted by piezoelectric bar can be calculated by dividing the electrical charge in (5) over the electric capacity of the piezoelectric material, c_a as expressed in (7),

$$V(t)=\frac{d_{33}n\left\{k_t\left(y_{cb}(t)-y_{eq}(t)\right)-c\,\dot{y}_{eq}(t)\right\}}{c_a}. \quad (7)$$

Generated electrical power is then obtained by multiplying the electrical current in (6) and electrical voltage in (7) with the number of piezoelectric used, p and written in (8),

$$P(t)=p\cdot I(t)\cdot v(t). \quad (8)$$

Equations (5)-(8) reflects that the design output of piezoelectric harvester is electrical parameters such as electrical charge, current, voltage and power. In particular, electrical charge (Q) must be produced optimally.

III. Results and Discussions

Based on Fig. 3, crane framework is discretized into 58 elements and 82 nodes where the size and mechanical properties of are tabulated in Table I. Variation of number of elements and nodes is not considered in the simulations. Cross-sections of the members are treated uniform, isotropic and homogeneous materials. The gravitational acceleration is g = 9.81 m/s^2 and time interval is Δt = 0.005 s.

TABLE I. CRANE FRAMEWORK PROPERTIES [1]

	Top beam Support (Right and Left)	Top beam
Material	Steel	
Young's modulus, E	$2.1e^{11}$ kg/m^2	
Density, ρ	7860 kg/m^3	
Cross-section area, A	$3.45e^{-2}$ m^2	$1.516e^{-2}$ m^2
Inertia, I_{xx}, I_{yy}	$3.139e^{-3}$ m^4, $2.7e^{-3}$ m^4	$8.741e^{-4}$ m^4, $1.76e^{-5}$ m^4
Span of framework, L_B	12 m	
Height of framework, H_F	10.6 m	

TABLE II. PROPERTY AND DIMENSION OF A PIEZOELECTRIC BAR [15]

Item	Symbol	Value	Unit
Piezoelectric number	p	2	-
Piezoelectric dimension	a x b x h	0.015 x 0.015 x 0.1	Meter
Modulus Young	E_p	$7.5e^{10}$	N/m^2
Piezoelectric constant in the polling direction	d_{33}	$3e^{-10}$	C/N
Electrical capacity for the piezoelectric patch	c_v	0.375	Farad

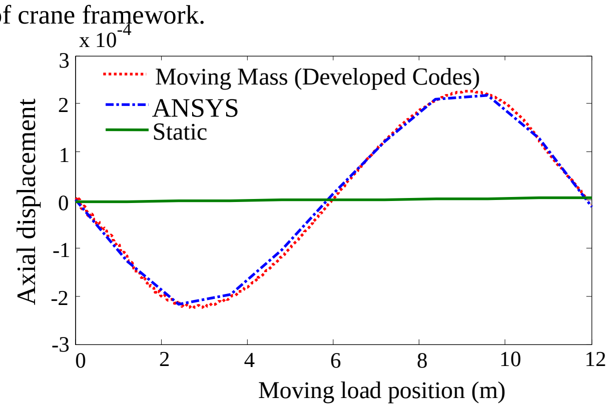
A. *Validation of Developed FE Model*

To validate the above developed FE model under developed numerical codes, ANSYS is taken as a benchmark. Dimensions for crane framework presented in Table I are used and taken from [1]. Validation is performed by simulating one concentrated moving mass that moves with constant velocity $V_T = 1$ m/s with total mass $m = 1200$ kg on the top beam of crane framework. The axial (X), vertical (Y), lateral (Z) displacements at the central point c_b of the top beam are depicted in Fig. 6. The figures show that the value of static displacements in axial (X) and lateral (Z) are very small compared to vertical (Y) displacement. This is to be expected since the strength is weaker in vertical direction so that the deformation of crane framework becomes dominant.

It is also seen that besides the vertical displacement, moving mass has significant effect in axial and lateral displacements compared to their corresponding static displacements. From Fig. 6, dynamic responses between developed codes and ANSYS are in good agreement, where the root mean square (RMS) of error is $8.73e^{-6}$ for axial, $7.8e^{-4}$ for vertical, and $8.56e^{-6}$ for lateral displacements.

B. *Forced Vibration of a Gantry Crane Framework*

For further validation, two moving masses are moved from the left hand side of the top beam using trajectory profile in Fig. 7. The effect of moving mass velocity to both axial and vertical displacements, the velocity $V_1(t)$, $V_2(t)$ and $V_3(t)$ are applied to moving masses as given in Fig. 7 but still stops in the middle of the top beam. The velocity histories are defined in Table II, $(m_T + m_p) =$ 900 kg and $\xi_1 = \xi_1 = 0.005$. It is noted that other forms of velocity histories could have been chosen, but here an arbitrary profiles are chosen to primarily illustrate the effect of moving mass velocity on the dynamic responses of crane framework.



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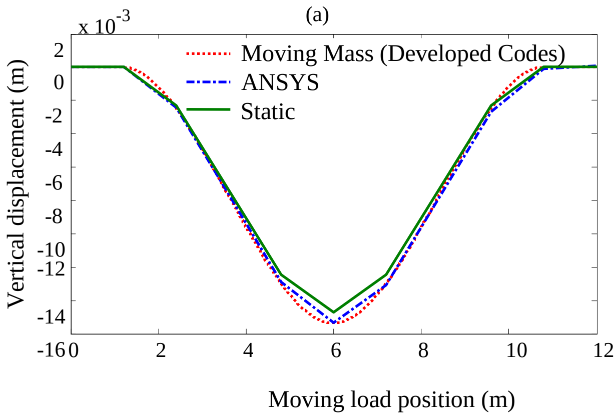

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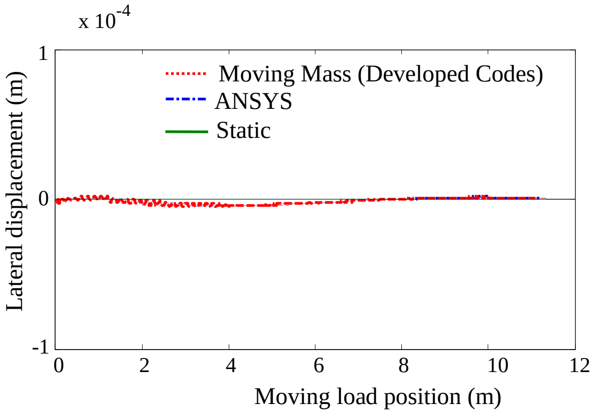

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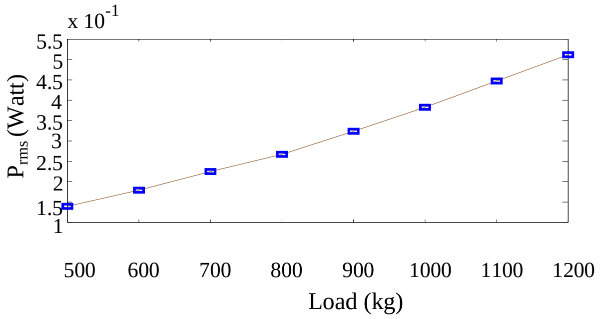
Figure 6. Dynamic displacements of crane framework (a) axial (b) vertical (c) lateral.

Axial (X), vertical (Y) and lateral (Z) displacements of crane framework under velocity variations are given in Figs. 8-10. It is seen that the moving masses propagate along the top beam faster when the velocity is getting higher. Zoom of axial displacements in Fig. 8 as shown in Fig. 9 depict that the increase of moving mass velocity will increase the vibration amplitudes, espescially during acceleration and deceleration phases. It reveals that the velocity of moving load has significant effect on maximum displacement of axial displacement.

C. *Harvested Electric Power in a Crane System*

Based on results in Fig. 6, it can be assessed that axial (X), vertical (Y) and lateral (Z) displacements of crane framework are potential to be harvested. By using equations in (5)-(8), as well as property and dimension of piezoelectric bar tabulated in Table II, electric power harvested using PEH in Fig. 1 can be conducted. For all simulations, ℓ_2 is chosen to be 0.02 m, value of n is fixed to be 6 while damping ratio, $\xi_1 = \xi_1$ is to be 0.005. It should be noted that the harvested electric power is expressed in terms of RMS of electric power, P_{rms}. Parametric study is performed to investigate the ability of PEH in energy harvesting of vibration of crane system. It comprises variations of trolley velocity, V_T, total mass, $(m_T + m_p)$ and spring stiffness, k_t, respectively.

First parametric study is by varying the values of total mass $(m_T + m_p)$ while fixing the masses velocity, V_T to be 1 m/s, the value of k_t is set at $0.75{\cdot}10^4$ N/m. It is seen that the increase of total mass, $(m_T + m_p)$ is followed by the increase of P_{rms} as shown by Fig. 7. However, the increment has a nonlinear form for vertical displacement, and slight nonlinear forms for axial and lateral displacements, respectively. This is to be expected since that the dynamic displacement and its derivatives, $y_{cb}, \dot{y}_{cb}, \ddot{y}_{cb}$ are directly proportional to the total mass according to Eq. (4).



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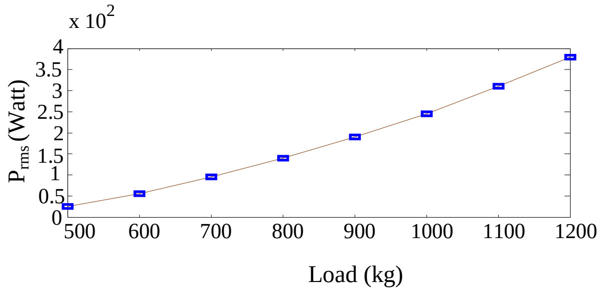

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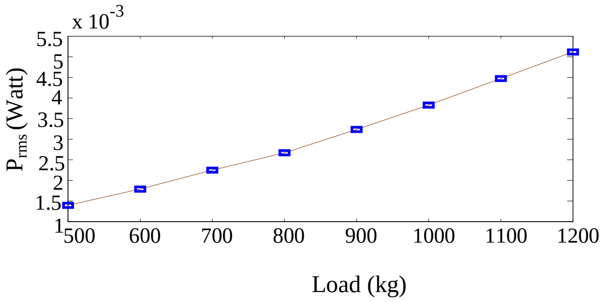

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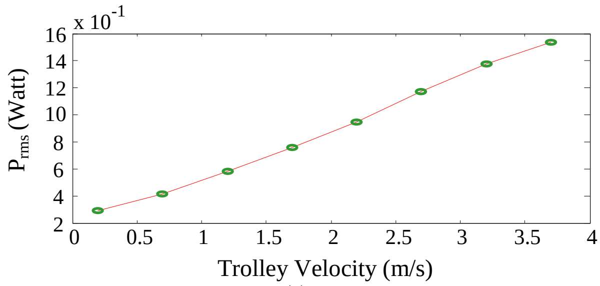
Figure 7. Variation of $(m_T + m_p)$ at $V_T = 1$ m/s.

Second parametric is performed by varying the value of V_T, and fixing the value of $(m_T + m_p)$. It is seen from Fig. 8 that the increase of V_T is followed by the increase of P_{rms}. The increment has a linear form for vertical, while nonlinear forms for axial and lateral displacements, respectively. This finding is owing to the fact that an increase in V_T leads to an increase in dynamic displacement of the crane framework at the c_b of the top beam, and its derivatives, $y_{cb}, \dot{y}_{cb}, \ddot{y}_{cb}$. As a result, the dynamic displacement of the equivalent model of lever-arm mechanism and piezoelectric bar in Fig. 3, and its derivatives, $y_{eq}, \dot{y}_{eq}, \ddot{y}_{eq}$ increases correspondingly. Those increments correspond to the increase of the charge, voltage, current, and generated electric power of the piezoelectric bar as indicated in Eqs. (5)-(8).



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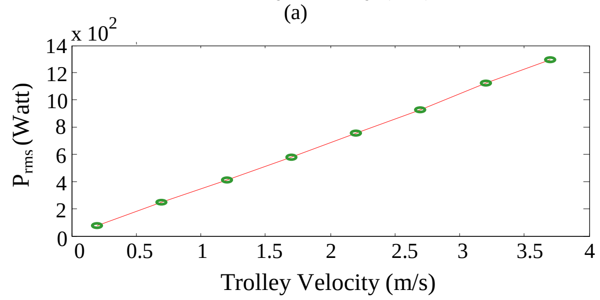

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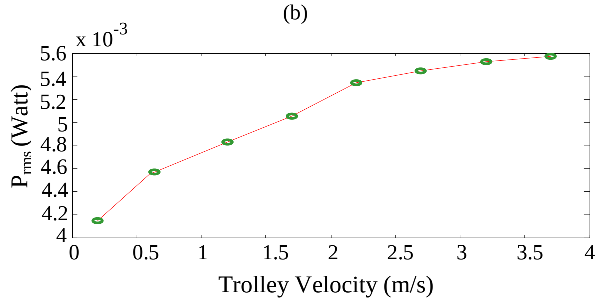

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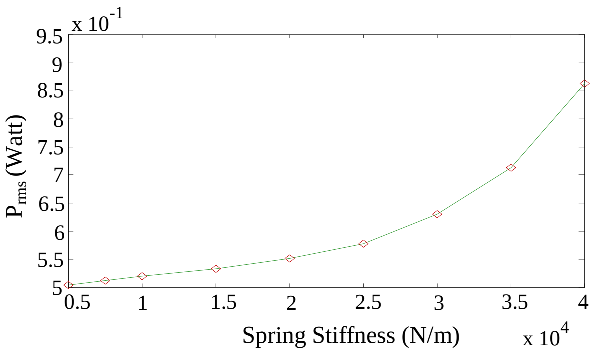
Figure 8. Variation of V_T (a) axial displacement (b) vertical displacement (c) lateral displacement

Last parametric study is performed by varying the spring stiffness, k_t, and fixing the value of V_T at 1 m/s and $(m_T + m_p)$ at 1200 kg. It is found that when the spring stiffness increases, k_t, the P_{rms} also increases nonlinearly for all displacements as revealed from Fig. 9. This finding can be interpreted by the fact that by keeping the V_T, $(m_T + m_p)$ and the size of crane framework to be constant, the force applied to the equivalent model of lever-arm mechanism and piezoelectric bar shown in Fig. 3 increases when the k_t increases.



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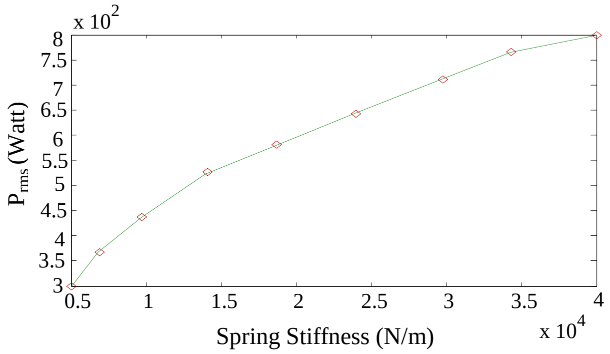

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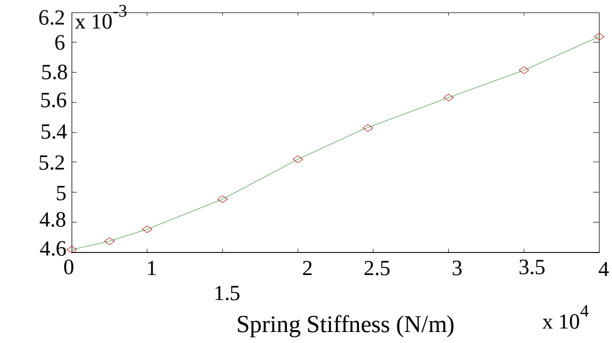

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Figure 9. Variation of k_t (a) axial displacement (b) vertical displacement (c) lateral displacement

IV. Conclusion

In this paper, a design concept of a PEH for a crane system is developed. Structural dynamic of a crane system and dynamics of a PEH are developed to calculate the RMS of the generated electric power. Result findings are as follows:

1. The RMS of the generated electric power increases with an increase in the trolley velocity, payload mass, and spring stiffness.
2. The electric power can be harvested up to 0.5 W, 37.5 W and 4.7 mW in axial, vertical and lateral directions can be harvested for a PEH with a span of 12 m of the top beam of the crane framework at the trolley speed of 1 m/s, and the total mass of 1200 kg.
3. Based on the findings, it is possible to harvest a higher power by increasing the number of PEHs, the size of the crane's structure, the trolley speed, the size of total mass, and other parametric values as long as they are within the safety requirements of the cranes.

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