Structural dynamics effect on voltage generation from dual coupled cantilever based piezoelectric vibration energy harvester system

Khoo Shin Yeena,b, Zainab Shakir Radeefa, Ong Zhi Chaoa,b, Yu-Hsi Huangc, Chong Wen Tonga, Zubaidah Ismaild,b

aDepartment of Mechanical Engineering, Faculty of Engineering, University of Malaya, 50603 Kuala Lumpur, Malaysia
bAdvanced Shock and Vibration Research Group, Applied Vibration Laboratory, Block R, Faculty of Engineering, University of Malaya, Malaysia
cAdvanced Shock and Vibration Research Group, Applied Vibration Laboratory, Block R, Faculty of Engineering, University of Malaya, Malaysia

A R T I C L E   I N F O

Article history:
Received 20 September 2016
Received in revised form 7 April 2017
Accepted 3 May 2017
Available online 6 May 2017

Keywords:
Cantilever structure
Location selection
Modal analysis
Operating deflection shape analysis
Piezoelectric location
Vibration harvesting

A B S T R A C T

This article investigates the effect of structural dynamics on voltage generation from a novel dual coupled cantilever based piezoelectric vibration energy harvester (PVEH) system. Two non-destructive vibration techniques using the EMA and ODS analysis techniques have been integrated into the location selection scheme for enhancing vibration energy harvesting purpose. The location selection scheme is based on a measurement procedure on both harvester and its host structure to identify the optimal location. The results shows that the proposed cantilever PVEH is able to harvest high voltage in the high displacement region. An optimal location, (i.e. maximum vibration points/anti-nodal points) determined by the location selection scheme could yield 33.3% improvement in harvested voltage, as compared to baseline voltage. Meanwhile, if the piezoelectric plate is placed at any minimal vibration points or nodal point on the structure as determined by the location selection scheme, a significant reduction (>70%) in harvested voltage is observed.

© 2017 Elsevier Ltd. All rights reserved.

1. Introduction

Energy harvesting from mechanical vibrations in vibrating environments, especially vibrating machinery and vibrating structure has received intense attention as a new energy solution, to replace traditional power source, (i.e. batteries and power grid), due to the great demand of self-powered system such as micro-scale electronics and wireless sensor network. Generally, there are three main transduction mechanisms to convert the unwanted vibration energy to electrical energy: piezoelectric, electromagnetic and electrostatic mechanisms. Among these electromechanical transducers, piezoelectric generators has received the most attention due to its higher energy storage density for micro-scale application, (i.e. approximate 35.4 mJ/cm³ of practical values and 335 mJ/cm³ of aggressive value) and it is easy to integrate into a system, as reported in the study of [1] and [2] respectively. Detailed comparisons of the performance between various generators were studied by [3–5].

Plenty of research is ongoing toward improving the power output of piezoelectric vibration energy harvester (PVEH) from a given vibration source, to realize its practical usage. Important limitation factors that are affecting the generation of electrical power can be listed as follow: frequency matching, properties of piezoelectric material, proof mass, harvesting circuit, strain distribution and location of PVEH. Inappropriate design without considering these limitation factors may cause high power loss in terms of mechanical loss, mechanical-electrical transduction loss and electrical loss. The relationship between the limitation factors and power loss can be found in the study by [6]. Among these parameters, ‘frequency matching’ factor has received the most attention from the researchers. It is well known that vibration energy harvester mechanically resonates at a frequency that coincides with its natural frequency, hence generates maximum electrical power. However, the power output decreases significantly when the excitation frequency slightly shifts from resonant point. Liu et al. [7] showed that resonance based vibration energy harvester has a narrow useful bandwidth, (i.e. the available bandwidth is within ±2 to 3 Hz to maintain voltage output approximately at 70% of its maximum value or greater). Conventional piezoelectric harvester performs greatly when the excitation frequency is known in a priori, however, it may suffer significant power loss when the excitation
frequency is unknown or the vibration environment varies with time.

In the case of unknown excitation frequency or time-variant vibration environment, it is particularly useful to implement frequency tuning strategy. Roundy et al. [1] demonstrated several frequency tuning methods to obtain maximum power output. They proposed self- and adaptive-tuning approaches such as utilization of active and passive tuning actuators, (i.e. electronic spring and moveable clamp) or preload method, in order to adjust and match the natural frequency of the system with the driving/excitation frequency. Besides that, they demonstrated that a wider frequency bandwidth can be realized by using a multiple proof masses concept. However, the design is accompanied by some limitations such as additional system, (i.e. actuator) is required and an increase of the piezoelectric material’s volume, (i.e. higher harvester size and cost). Furthermore, Muriuki and Clark [8] presented a shunt method to tune the natural frequency of piezoelectric resonator. In this study, complex electronic and actuator system with the implementation of the shunt capacitor concept was used.

Frequency tuning for broadband energy harvesting is a major challenge in the piezoelectric energy harvesting technology. The approaches used to overcome this limitation are discussed here. Liu et al. [7] examined three cantilever typed piezoelectric generators with closely spaced natural frequencies, which were connected in series circuit and the cantilever configuration was in array arrangement. They found that this array arrangement enhanced the overlapping effect, which was effective to obtain a wider bandwidth. Moreover, they obtained higher power output, (i.e. 26% increment) by using electrical connection after AC-DC rectification, as compared to direct serial connection from all output. Wah et al. [9] highlighted that ripples between peaks in the frequency spectrum for a multiple cantilever beams energy harvester may cause undesirable results. They proposed to alter the natural frequency of each cantilever beam to have a close frequency ratio in order to increase the available bandwidth and reduce the ripples. Xiao et al. [10] provided a guideline for optimizing the harvesting power and bandwidth. A parameter study of a two DOF PVEH system, which was operating under various masses and frequency ratios had been demonstrated. They found that while keeping the total mass and the mass ratio of the oscillators constant, an increment in the numbers of DOF will improve the harvested power as well as shifted the first vibration mode to a lower frequency region.

On the other hands, several researchers focused on nonlinear technique to improve the broadband characteristic of PVEH. Mann and Sims [11] proposed a novel magnetic integrated PVEH system that engaging the system's nonlinear response, with a much larger range of frequencies as compared to linear energy harvesting, hence improving the energy harvested. Ferrari et al. [12] implemented bistable nonlinear oscillator using a magnet, where two equilibrium positions were existing. It rapidly switched between these equilibrium positions under proper mechanical excitations in order to increase the velocity as well as the power output. Furthermore, the implementation of both multiple degrees of freedom (MDOF) and nonlinear techniques in the piezoelectric harvesting system can further improve the performance of harvester as demonstrated in [13,14].

Strain distribution is another important design parameter to enhance power scavenging. Theoretically, PVEH with uniform strain distribution is preferable due to higher utilization of its potential energy. In fact, this can be achieved by altering the geometry of the PVEH. For example, Baker et al. [15] extended the works from Roundy et al. [1] which addressed the uneven strain distribution issue for a conventional cantilever rectangular piezoelectric beam. They proposed a trapezoidal piezoelectric stack PVEH, which had a more uniform strain distribution. The result showed that it was able to increase 30% of the output power per unit volume. On the other hand, Kim et al. [16] developed a ‘cymbal’ PVEH which was in circular configuration and it was designed for high force applications. The result showed that the stress applied to the piezoelectric material was more evenly distributed than in a conventional stack configuration. Moreover, Mateu and Moll [17] found that triangular cantilever generated more power than rectangular beam.

Among all of the discussed limiting factors related to mechanical loss, ‘location’ factor has attracted little attention from researchers, however, it is a crucial factor that cannot be neglected, to successfully integrate a harvester to a vibrating system. As a rule of thumb, good PVEH location must be rich of vibration energy, where it should be selected at high strain region since voltage generated is proportional to the strain level. Eggborn [18] conducted a parametric study on the PVEH location on a beam. They found that maximum strain energy occurred at the clamped end of the beam, which in turn generated maximum power, compared to zero power generation on the free end of the beam due to zero strain energy occurred. Dutot et al. [19] emphasized that there is a need to check the total area under the strain curve. If the mode shape induced by the host structure causes the total area to decrease, the total power will decrease too which is known as a strain cancellation phenomenon. However, if this effect is taken into account, power benefit due to strain accumulation is possible. Erturk and Inman [20] and Erturk et al. [21] further examined the concept of the effects of strain mode shapes on the harvested energy. Their research outcomes verified that the voltage output depends on the area under the strain curve. In their testing, the numbers of nodes of a system undergoing nth mode of vibration are n−1, n and n+1 for pinned-pinned, clamped-clamped and clamped-pinned boundary conditions respectively. Strain nodes exist in higher modes would cause the strain to change the sign, (i.e. the resulting voltages in the electrodes to be 180 deg out of phase), therefore it reduces the harvested voltage. They suggested to use segmented electrodes instead of continuous electrodes, which avoided the power reduction due to strain node/strain cancellation effect. Liao and Sodano [22] further examined the effect of PVEH placement on damping of vibration modes and the effect of patch size on optimal placement. It was observed that the mass loading effect for all patches, (i.e. 2%, 15% and 30–70% of area coverage for small-size, mid-size and large-size patches respectively) is significant in the system, which shifts the natural frequency and mode shape. Hence the optimal locations are affected as the patch size changes.

Selection of PVEH location on the MDOF harvester is described here. Cornwell et al. [23] demonstrated the usage of tuned vibration absorber, (i.e. auxiliary structure) to enhance the power harvesting. Placement of PVEH on the auxiliary structure can generate more power rather than host structure. Their results showed that the harvested voltage is proportional to strain, and electrical power is proportional to voltage squared. Tang & Yang [24] and Xiao et al. [10] investigated the effect of the location of the piezoelectric element on the harvesting performance in a two DOF PVEH system. Tang and Yang [24] showed that the configuration of piezoelectric patch attached to the primary oscillator mass has a better harvesting performance compared to the configuration of piezoelectric patch attached to the auxiliary oscillator mass. Xiao et al. [10] further improved the configuration by inserting additional piezoelectric elements between every two nearby oscillators to maximize the scavenging power. The examples above showed that the optimal PVEH placement is crucial for practical implementation in real system.

Previous research showed that location selection of the electrodes/the piezoelectric sensor on a host/auxiliary structure was a very important factor to maximize the harvesting power. Failure
to address it may suffer zero harvesting power. Thus, most of the study focused on the implementation of PVEH patch in the high strain region for harvesting the maximum voltage output. For example, Eggborn [18], Erturk et al. [21] and Liao & Sodano [22], proposed that PVEH patch should be placed at the clamped end (location a) of the cantilever beam for maximum power generation, as shown in Fig. 1. This was because the location had the highest strain and the power was proportional to the strain. The PVEH patch generated poor power at the free end (location z) as it had the lowest strain, (i.e. strain node). The limitation of the PVEH patch was mentioned in [18], where the harvester was not possible to be placed at the clamped end of beam in some applications and it is possible that the harvester was placed at the strain node. Therefore, there is a need to find a new placement method to solve this issue. Moreover, there is no study on the effect of cantilever PVEH location to the power generation so far in the knowledge of the authors. Thus, motivation to fill this literature gap is created in this study.

Much of the research up to this point only focused on the characterization/improving the efficiency of PVEH [2]. Most of them [1,6,7,10–12,14,15,19,24–26] investigated the dynamics effect of the PVEH alone without including the host structure. To bring the concept of creating self-powered electronic with PVEH system closer to reality, there is a need to investigate the dynamics effect from both PVEH auxiliary structure and its vibrating host structure. In this paper, the structural dynamics effect of the vibrating host structure towards the harvester performance will be investigated.

Two essential vibration tools, namely Experimental Modal Analysis (EMA) and Operating Deflection Shape (ODS) analysis will be adopted to experimentally identify the optimal location for vibration energy harvesting on a dual coupled cantilever based PVEH system. Furthermore, this study reveals a novel PVEH configuration, (i.e. cantilever PVEH) that producing high voltage in the low strain/high acceleration region, which solve the limitation of the conventional PVEH patch. The effect of cantilever PVEH location to the power generation will be investigated throughout the study.

2. Theoretical background

2.1. Mathematical formulation

In this study, the dual coupled cantilever based PVEH system is designed to harvest the wasted vibration energy in high response (acceleration/displacement) region. The governing equation of the vibration problem can be described as Eq. (1).

$$\mathbf{M}\ddot{x} + \mathbf{C}\dot{x} + \mathbf{K}x = \mathbf{Q}$$ (1)

where M, C and K are mass, damping and stiffness matrices respectively. X, \(\dot{X}\) and \(\ddot{X}\) are acceleration, velocity and displacement vectors respectively. \(\mathbf{Q}\) is the forcing vector.

Considering periodic excitation of a shaker, the forcing function can be assumed as \(\mathbf{Q} = \mathbf{Q}_0 \cos \omega t\). Then, the spatial coordinate equation of motion above can be uncoupled into the generalized/principal coordinate as follows.

$$\ddot{x}_{p,r} + 2\xi_r \dot{x}_{p,r} + \omega_n^2 x_{p,r} = q_{p,r} \cos \omega t, \quad i = 1, 2, 3, \ldots, mth DOF$$ (2)

where \(x_{p,r}, \dot{x}_{p,r}, \ddot{x}_{p,r}\) are the \(r^{th}\) mode steady-state displacement, velocity and acceleration respectively in principal coordinate. Decay rate of \(r^{th}\) mode, \(\xi_r\), is equal to damping ratio, \(\zeta_r\) multiply with undamped natural frequency, \(\omega_n\). \(q_{p,r}\) is the \(r^{th}\) mode forcing function in principal coordinate. \(\omega\) is the excitation frequency of the forcing function in rad/s.

By solving Eq. (2), we obtain the steady-state response in principal coordinate as follows.

$$x_{p,r} = \frac{q_{p,r}}{\omega_n^2} \beta_r \cos(\omega t - \theta_r), \quad r = 1, 2, 3, \ldots, nth mode$$ (3)

in which the magnification factor,

$$\beta_r = \frac{1}{\sqrt{[1 - \omega^2/\omega_n^2]^2 + [2\zeta_r \omega/\omega_n]^2}}$$ (4)

and the phase angle,

$$\theta_r = \tan^{-1} \left[ \frac{2\zeta_r \omega/\omega_n}{1 - \omega^2/\omega_n^2} \right]$$ (5)

Hence, back transformation is performed to obtain the contribution of the considered mode.

$$X_{i,r} = \phi_{i,r} x_{p,r}$$ (6)

where \(\phi_{i,r}\) is the \(r^{th}\) mode normalized mode shape at \(i^{th}\) DOF of the structure. The total response of the structure is the combination of response contributed from each mode. It is given by Eq. (7).
Theoretically, cantilever PVEH should be placed at the location with the highest displacement, \( \max(\mid X \mid) \) or the highest acceleration, \( \max(\ddot{X}) \) to harvest maximum power.

### 2.2. Factors affecting the structural response

In order to obtain the optimal location for the PVEH placement, it is crucial that one has to first evaluate the dynamic characteristics of the given structure, namely natural frequencies, mode shapes and modal damping, as illustrated in Eqs. (3) and (7). One should be able to analyze the structural response by considering the following factors: (1) Closeness of the natural frequencies to the excitation frequencies; (2) Amount of damping of the resonance modes; (3) Interaction between the distribution of excitation forces and the mode shapes. Detailed description of the factors are given below.

**Factor (1)** – Structural response increase especially when the excitation frequency close to the natural frequency of the system. For the resonant case, let the excitation frequency matches the 1st mode natural frequency \( (\omega = \omega_1) \). The magnification factor of 1st mode is the highest compared to other mode. Hence, the total response of the structure is mainly contributed by Mode 1, i.e. \( \dot{X} \approx \phi_1 X_{p,1} \). For non-resonant case, let excitation frequency to be somewhere near the natural frequencies of 1st–3rd modes. The total response of the structure is mainly contributed by Modes 1–3, i.e. \( \dot{X} \approx \phi_1 X_{p,1} + \phi_2 X_{p,2} + \phi_3 X_{p,3} \).

**Factor (2)** – Modal damping plays an important role to reduce the vibration especially when resonance occurs. When \( \omega = \omega_0 \), the magnification factor, \( \beta_r = \frac{1}{\sqrt{1 + \xi^2}} \). As the damping ratio increases, the magnification factor will be reduced and hence it reduces the total response of the structure. Note that damping does not cause much change in the response when it away from resonance.

**Factor (3)** – The generalized force, \( q_{r,r} \) is the 3rd factor that affecting the response of the structure, where \( q_{r,r} = \phi_1 Q_1 + \phi_2 Q_2 + \cdots + \phi_m Q_m \): \( r = 1, 2, \ldots, m \)th mode. The interaction between the distribution of excitation forces and the mode shapes can be separated mainly into two categories: symmetrical or anti-

\[
X = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_m \end{bmatrix} = \begin{bmatrix} \phi_{1,1} X_{p,1} + \phi_{1,2} X_{p,2} + \cdots + \phi_{1,n} X_{p,n} \\ \phi_{2,1} X_{p,1} + \phi_{2,2} X_{p,2} + \cdots + \phi_{2,n} X_{p,n} \\ \vdots \\ \phi_{m,1} X_{p,1} + \phi_{m,2} X_{p,2} + \cdots + \phi_{m,n} X_{p,n} \end{bmatrix}
\]
symmetrical. In general, if the force vector is non-symmetrical/symmetrical with the shape vector, zero/non-zero generalized force will be obtained. Zero generalized force will eventually produce minimal structural response.

3. Problem formulation

Structural dynamics analysis of a dual coupled cantilever based PVEH system shown in Fig. 2 is considered in this section. Two non-destructive vibration techniques using the EMA and ODS analysis techniques will be integrated into the location selection scheme for enhancing vibration energy harvesting purpose. The location selection scheme is based on a measurement procedure on both harvester and its host structure to identify the optimal location as well as to avoid the poor location with low harvesting energy. By comparing the baseline voltage generated purely by the harvester and the actual voltage generated by the harvester when it is mounted on its vibrating host structure, the structural dynamics effect, (i.e. from the vibrating host structure) on the producing voltage will be evaluated.

3.1. Dual coupled cantilever based PVEH system

A dual coupled cantilever based PVEH system is a piezoelectric harvesting system that is mainly comprised of two cantilever typed structures: (1) Cantilever beam as the vibrating host structure; (2) Piezoelectric cantilever plate/harvester as the auxiliary structure as shown in Fig. 2. The piezoelectric plate or PVEH is coupled and attached to the host structure by using magnets. In this
study, the host structure is made of steel, which has dimensions of 450 mm in length, 5 mm in thickness, and 20 mm in width. The host structure is fixed at one of its ends, while the other end is free to move, (i.e. cantilever configuration). The PVEH is made of piezo-ceramic material, (i.e. lead zirconate titanate or PZT), which has dimensions of 63.5 mm in length, 0.51 mm in thickness, and 31.8 mm in width. The detail of the properties and performance regarding this PVEH (model: T220-A4-503Y-) is given by the manufacturer (Piezo System, Inc.) from the following website (http://www.piezo.com/prodbg1brass.html). Similar to the boundary condition of host structure, the PVEH acts as a cantilever where one of its ends is clamped.

The mass loading effect occurs when an additional mass is added to a system which changes the dynamic characteristic of the system. This happens when we shifts the position of the PVEH auxiliary structure from Point 1 to Point 9 in order to examine the dynamic characteristic of the piezoelectric plate using ESPI.

**Fig. 5.** Dynamic characteristic of the piezoelectric plate using ESPI.

**Fig. 6.** FRF of cantilever beam using impact testing.
performance of the harvester in various positions. To avoid the mass loading effect, dummy masses with equivalent weight as PVEH have been attached to the host structure, i.e. weight of each dummy mass is equal to the total weight (200 N) of PVEH. The position of the dummy mass will be switched with the PVEH when the PVEH is moving from one point to another. This ensures constant mass loading distribution of the system during the experiment. As a result, data consistency (time-invariant dynamic characteristic) throughout the measurement can be achieved.

3.2. Electronic Speckle Pattern Interferometry (ESPI)

Electronic Speckle Pattern Interferometry (ESPI) is an experimental technique for measuring both transverse (out-of-plane) and planar (in-plane) vibrations using non-contact optical techniques. A self-arranged optical setup is possible, where the undesired measurement direction of laser beams can be blocked in order to switch from transverse to planar measurement arrangement or vice versa. ESPI technique has the advantages of real-time, non-contact and high precision measurement. Detailed experimental setup and the performance comparison of ESPI technique over conventional measurement techniques utilizing laser Doppler vibrometer and impedance analyzer as well as the finite element method can be found in the study of Huang & Ma [25] and Huang et al. [26]. In this study, the ESPI technique is used for measuring the resonant frequency and its corresponding mode shape for transverse vibration. Planar vibration is not considered in this study due to the nature of the excitation in the harvesting system, (i.e. base excitation induced transverse vibration to the PVEH system).

3.3. Experimental Modal Analysis (EMA)

EMA is an experimental technique commonly used to identify dynamic characteristic of elastic structures. It often requires the system to be in a complete ‘shut down’ or ‘non-operating’ state. This means that no unaccountable excitation force will be induced into the system, except the designed and measurable force such as impact excitation from the modally tuned hammer (known as impact testing) or random excitation from shaker (known as shaker testing). The responses of the system induced by the designed force will be recorded as well. Transfer functions (also known as Frequency Response Functions (FRFs)) are later obtained by considering the cross-correlated functions and auto-correlated functions between the measured input forces and output responses as shown in Eq. (8).

\[ H_{ij}(\omega) = \frac{B_i(\omega)A_j(\omega)}{A_i(\omega)A_j(\omega)} \]  

where \( H_{ij}(\omega) \) = transfer function due to \( i \)th DOF and \( j \)th DOF of the locations of output response and input force respectively, at the corresponding frequency, \( \omega \). Both output response, \( B \) and input force, \( A \) are complex value in the frequency domain. The transfer function is based on \( H_{1} \) estimator, where the numerator part represents the cross-correlated function between output response and the complex conjugate of input force, while the denominator part represents the auto-correlated function between the input force and the complex conjugate of input force.

In this study, impact testing with Single Input Multiple Output (SIMO) approach is used where multiple accelerometer sensors are mounted on the desired discrete set of geometrical positions, which are sufficient to describe the vibration distribution of the structure due to a single input force. In this case, the obtained FRF is called accelerance FRF. The input force location is selected by using driving point FRFs measurement, (i.e. measurement of the \( H_{xxy} \) where the input response is at the same location as an output force for all the possible force locations). Driving point FRFs with the maximum number of strong (large magnitude) resonance peaks is then identified as the potential reference force [27]. SIMO approach is selected for its time efficiency. Most importantly, it does not have a problem with a mass loading issue where high measurement accuracy can be achieved. Once FRFs are obtained, various curve fitting algorithms are then used to extract the modal parameters, namely natural frequencies, mode shapes and modal damping as follows.

\[ H_{ij}(\omega) = \sum_{i=1}^{n} \frac{\phi_i \phi_j}{(\omega_{i0}^2 - \omega^2) + (2\zeta_i \omega_{i0} \omega)} \]

where \( H_{ij}(\omega) \) = transfer function due to \( i \)th DOF and \( j \)th DOF of the locations of output response and input force respectively, at the corresponding frequency, \( \omega \). \( \phi_i, \phi_j \) = mode shapes due to \( i \)th DOF and \( j \)th DOF of the locations of output response and input force respectively, at the corresponding \( r \)th mode. \( \omega_{i0}, \zeta_i = r \)th mode natural frequency and damping ratio respectively.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Structural natural frequency (Hz)</th>
<th>Damping (%)</th>
<th>Mode shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12.5</td>
<td>2.77</td>
<td><img src="image" alt="Mode shape view" /></td>
</tr>
<tr>
<td>2</td>
<td>51.8</td>
<td>5.52</td>
<td><img src="image" alt="Mode shape view" /></td>
</tr>
<tr>
<td>3</td>
<td>135</td>
<td>7.3</td>
<td><img src="image" alt="Mode shape view" /></td>
</tr>
</tbody>
</table>
3.4. Operating deflection shape (ODS) analysis

ODS analysis can be defined as any forced motion of 2 or more DOF (point & directions) on a machine or structure [28]. In other words, the vibration pattern of a structure as a function of time or frequency can be animated or visualized by integrating the technique with virtual instrument software. This technique requires the system to be in ‘in-service’ or ‘operating’ state. In general, two methods of measurement are used to acquire the ODS, namely the measurement set method and simultaneous method. The most common measurement set method is the ODS FRF, where it is formed by the magnitude of the auto spectrum of a roving response and the phase of the cross spectrum between the roving response and reference response. The simultaneous method is used where all channels of data are acquired simultaneously by using a multi-channel acquisition system. Both measurement work well for stationary signal such as sinusoidal excitation. Considering the effectiveness of cost and time, time ODS with simultaneous method is implemented in this study. Time ODS data can be converted to the frequency domain by using Fourier transform. Hence, one can link the obtained ODS data to the Eqs. (3)–(7) and dwell deep into the problem. Then the structural dynamics problem can be evaluated and analyzed in a systematic approach.

3.5. Measurement procedure description

A non-invasive measurement and evaluation procedure was applied to a dual coupled cantilever based PVEH system mainly...
in two conditions: (1) Exciting frequency matched the natural frequency of PVEH; (2) Exciting frequency matched the natural frequency of the host structure. These conditions were considered based on the ‘need to know’ whether and how the structural dynamics affects the overall harvesting performance of the PVEH. On the other way, the performance of PVEH without the structural dynamics effect, (i.e. PVEH was directly attached to shaker) was examined as illustrated in Fig. 3. The voltage outputs of the PVEH in two different excitation frequencies were recorded separately so that it will be used as a reference/baseline for further analysis.

Equipment set-up for MA, ODS analysis and voltage measurement are shown in Fig. 4. The cantilever beam was divided into 10 segments with 9 measurement points equipped with 9 accelerometers (Wilcoxon Research® Integrated Circuit Piezoelectric (ICP) accelerometer model S100C). For MA, a modally tuned PCB® ICP impact hammer model 086C03 was used to excite the cantilever host structure as shown in Fig. 4(a). The force sensor inside the tip of impact hammer will record the force signal and send it to the multi-channel data acquisition (DAQ) system through three four-channel DAQ hardware (i.e, model NI USB 9234) and a compact DAQ chassis, (i.e. model NI CDAQ-9172). All induced acceleration response at all the 9 measurement points will be captured by the DAQ as well. The DAQ system was connected to signal analyzer which was equipped with data acquisition software, (i.e. DASYLab®) and post processing software, (i.e. ME’scoopeVES®) for the MA purpose.

The experimental setup for both ODS analysis and voltage measurement are same as shown in Fig. 4(b), thus their data can be acquired simultaneously. An electromagnetic shaker (B&K LDS® model V101) and its power amplifier (B&K LDS® model PA25E) were used to vibrate the cantilever host structure at desired excitation frequency. A PCB® force sensor model 208C01 was used to monitor the force (7 N ± 1 N) induced by the shaker. Ideally, the shaker’s force was assumed constant throughout the study. Time history of acceleration responses and harvested voltage will be captured by the accelerometers and PVEH, hence transferred the signals to multi-channel DAQ system through DAQ hardware model NI USB 9234 and model NI USB 9263 respectively. The DAQ system was connected to signal analyzer which was equipped with data acquisition software, (i.e. DASYLab®) and post processing software, (i.e. ME’scoopeVES®) for ODS analysis and voltage measurement.

4. Result and discussion

4.1. Vibration characteristic of PVEH

Electronic Speckle Pattern Interferometry (ESPI) was used to measure the transverse vibration characteristic of the PVEH. Laser Doppler Vibrometer (LDV) was used to obtain the displacement spectrum ranging from 0 to 3590 Hz. The test was performed on the piezoelectric plate and the first bending mode was recorded at 80 Hz as shown in Fig. 5. Higher mode of the PVEH is not discussed here as the harvester will be integrated into a system that is operating under low excitation frequency (below 100 Hz).

4.2. Vibration characteristic of cantilever host structure

Meanwhile, MA using impact testing with SIMO approach was performed on the cantilever beam, (i.e. host vibrating structure). From the measured Frequency Response Function (FRF) as shown in Fig. 6, it is observed that there are 3 resonant modes in the frequency range of 0–200 Hz. First three natural frequencies are 12.5 Hz, 51.8 Hz and 135 Hz and their corresponding mode shapes are shown in Table 1 respectively.

4.3. Location selection scheme for piezoelectric energy harvesting

The study presents the selection scheme of optimal location of piezoelectric plate on cantilever beam structure based on vibration mode shape and its ODS. In this study, the piezoelectric plate is placed on the cantilever beam structure subjected to an excitation frequency of 52 Hz. The harvested voltage varies at different location of the cantilever beam. The voltage harvested by the piezoelectric plate and its voltage magnitude ranking is recorded as shown in Fig. 7. It is observed that the harvested voltages at Points 2 and 9 are minimal as compared to other points. Points 1, 6 and 5 harvest maximum peak voltage up to 6.0 Vac (0-peak).
The variation in harvested voltage could be explained based on the vibration mode shape and ODS of the cantilever beam. According to MA result, mode shape at 52 Hz is the second bending mode of the cantilever beam, where the nodal points (points with minimal response) fall on Points 2 and 9 as shown in Fig. 8. This explains that when the piezoelectric plate is placed at Points 2 and Point 9 where the structural vibration is minimal, the voltage harvested by the piezoelectric plate at these two points is minimal.

The similar trends could be observed using ODS of the cantilever beam at 52 Hz as shown in Fig. 9. The dynamic response at 52 Hz is mainly contributed by the second vibration mode of the cantilever beam. It vibrates at the deflection shape which is dominated by the second mode, where Point 2 and Point 9 are with minimal vibration. In addition, the displacement response curve (displacement versus time) as well as its vibration spectrum are provided in Fig. 10. The proposed dual coupled cantilever based PVEH system is able to produce voltage from microvibration, i.e. to produce [0.7 6.0] Vac (0-peak) from rms displacements of [6.0 43.8] µm at 9 examined locations. As illustrated in [29], harvesting energy from minimum mechanical energy at micro movements indicates a good progress toward building a self-powered system.

In summary, vibration pattern obtained from MA or ODS analysis could assist users in determining the optimal location for harvesting the maximum voltage.

The novel design of dual coupled cantilever based PVEH system produces high voltage at the high displacement/acceleration region, which behaves differently compared to conventional PVEH patch placement as reported in [18,21,22]. This overcomes the limitation where user must place the PVEH at the high strain region instead of the high acceleration region. This innovative idea gives flexibility for the user to select the desired PVEH placement method. One should select the suitable placement, i.e. configura-

![Fig. 10. Displacement response curve and its spectrum at 52 Hz excitation.](image-url)
tation of PVEH patch and cantilever PVEH) strategically for obtaining the maximum voltage output in the same structure. For example, it is recommended to use the configuration of PVEH patch to harvest energy in low acceleration region, while at the same time, the configuration of cantilever PVEH can be used to harvest energy in high acceleration region as shown in Fig. 11.

4.4. Structural dynamics effects on the harvesting performance of dual coupled cantilever based PVEH system

This study investigates the structural dynamics of the cantilever beam in the effort of optimizing the voltage harvested by the piezoelectric plate, which is placed on the cantilever beam. It is noted that the piezoelectric plate could harvest maximum voltage when it is being excited at its natural frequencies. In the condition of exciting the piezoelectric plate at its first natural frequency, i.e. 80 Hz, the maximum harvested peak voltage is recorded at 24.1 Vac (0-peak) and it is treated as the baseline voltage. In this case, the piezoelectric plate is placed directly on top of the shaker without any interference from structural vibration and the excitation force from the shaker is maintained (Fig. 3). When this piezoelectric plate is placed on the operating cantilever beam at 80 Hz, there is a 42.7% of reduction in harvested peak voltage and it is recorded at 13.8 Vac (0-peak). The reduction could be explained based on the structural dynamics aspect of the cantilever beam. There are no natural peaks being observed at 80 Hz in FRF of the cantilever beam.

![Fig. 11. Selection of PVEH placement method for low and high acceleration regions.](image1)

![Fig. 12. Harvested voltage at 80 Hz excitation for various measurement points in comparison to baseline voltage.](image2)

![Fig. 13. Harvested voltage at 52 Hz excitation for various measurement points in comparison to baseline voltage.](image3)
beam and the effect of mode shape is not taken place. The harvested voltage is mainly contributed by the shaker excitation. Thus, the vibration of the cantilever beam at 80 Hz is very low and even when the piezoelectric plate with the natural frequency of 80 Hz is placed on the cantilever beam. The harvested voltage is significantly reduced as compared to the baseline voltage as shown in Fig. 12.

In real-life application where the structure could not be excited at the piezoelectric plate's natural frequency, selection scheme of optimal location shall be utilized in order to maximize the voltage harvested by the piezoelectric plate. When the cantilever beam is excited at 52 Hz which is far away from the piezoelectric plate's natural frequency, the maximum harvested peak voltage is 6.0 Vac (0-peak) at one of the anti-nodal points of the second mode shape (Point 1). As compared to the baseline voltage at 52 Hz, the maximum peak voltage harvested is only recorded at 4.5 Vac (0-peak), it shows a 33.3% improvement in harvested voltage when the piezoelectric plate is placed at the optimal location, (i.e. the anti-nodal point in the second mode as in modal analysis or maximum vibration point as in ODS). With the location selection scheme, the piezoelectric plate should avoid to be placed at the nodal point of the mode shape or minimum vibration point, i.e. Point 2 (1.2 Vac) and Point 9 (0.7 Vac). This could yield more than 70% reduction in voltage harvested by piezoelectric plate. The results are shown in Fig. 13.

5. Conclusion

Vibration shapes obtained from MA or ODS analysis could be utilized in selecting an optimal location for piezoelectric plate to harvest maximum voltage. Noted that when the piezoelectric plate is placed on a structure under vibration/resonance, the optimal location, (i.e. maximum vibration points/anti-nodal points) determined by the location selection scheme could yield 33.3% improvement in harvested voltage. Meanwhile, if the piezoelectric plate is placed at any minimal vibration points or nodal point on the structure as determined by the location selection scheme, a significant reduction (>70%) in harvested voltage is observed. Furthermore, the effect of the cantilever PVEH location on the power generation has been investigated. The proposed cantilever PVEH is able to harvest high power in the high acceleration region which solves the limitation of the conventional PVEH patch configuration. In conclusion, this study has provided strong evidence that the dynamics of a structure could be studied and utilized before determining the optimal location in harvesting maximum power.

Acknowledgements

This study was supported by Postgraduate Research Fund (PG088-2014B) and University of Malaya Research Grant (RG160-15AET). The suggestions and recommendations from reviewers are gratefully acknowledged.

References