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NANYANG
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**SMALL-SCALE WIND ENERGY
HARVESTING USING PIEZOELECTRIC
MATERIALS**

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SCHOOL OF CIVIL AND ENVIRONMENTAL ENGINEERING

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SMALL-SCALE WIND ENERGY HARVESTING USING PIEZOELECTRIC MATERIALS

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SUMMARY

In the past few years, explosive studies have been conducted on energy harvesting in both research and industrial communities. The interest comes from the development of low-power-consumption small-scale electronics like wireless sensors and portable electric devices, which can be self-powered by harvesting electrical energy from various ambient sources. Wind energy is ubiquitous in both outdoor and indoor environments, yet small-scale energy harvesting aimed to harness power from wind flows with low speed ($<10\text{m/s}$) has received only limited attention. An extensive literature review on the fundamentals and the state-of-the-art techniques of small scale wind energy harvesting is presented, with a focus on those based on the piezoelectric transduction mechanism. Theoretical modeling methods, available harvester structure designs as well as efficiency enhancing techniques reported in the literature are reviewed in detail.

A comparative study of different tip cross-sections for small-scale wind energy harvesting based on galloping phenomenon is presented. An analytical model is established and verified by the experimental results. It is recommended that, for the very first time as the author knows, the square section should be used for small wind galloping piezoelectric energy harvesters (GPEHs). Subsequently, comparison study on analytical models of GPEHs shows that the quasi-steady hypothesis is applicable for galloping force formulation in GPEHs. A single-mode is sufficient to represent the dynamics as the GPEH oscillates constantly at its fundamental mode. The distributed parameter model owns a more rational representation of the aerodynamic force, while the one degree-of-freedom (1DOF) lumped parameter model gives a better prediction on the cut-in wind speed. A parametric study is carried out for further parameter optimization of harvesters for power output enhancement.

In order to enhance the output power in low wind speeds, a two degree-of-freedom (2DOF) piezoelectric aeroelastic energy harvester with a cut-out cantilever and two magnets is proposed, which is demonstrated experimentally to have a lower cut-in

wind speed of 1m/s and higher power output than the conventional 1DOF harvester up to 4.5m/s. Subsequently, an easy but quite effective method to significantly enhance the power generation capability of aeroelastic energy harvesters is presented by attaching a beam stiffener to the substrate of the harvester as an electromechanical coupling magnifier. It is shown to be effective for all the three considered types of harvesters based on galloping, vortex-induced vibration and flutter, leading to a superior performance over the conventional designs without the beam stiffener, with dozens of times increase in power, almost 100% increase in the power extraction efficiency, yet with comparable or even smaller transverse displacement.

In order to enhance the power extraction efficiency in the aspect of interface circuit, the feasibility of employing the synchronized charge extraction (SCE) interface for a GPEH is investigated. The performance of SCE circuit is compared with a standard circuit, revealing three main advantages of SCE in GPEH system: first, the output power from SCE is independent of electrical load, eliminating the requirement of impedance matching and thus ensuring the flexibility of adjusting the GPEH system for practical applications; second, the SCE circuit helps to save piezoelectric materials by 75% compared to the standard circuit; third, the displacement amplitude of GPEHs with SCE is much smaller, alleviating the fatigue problem and enhancing the durability of the harvesting system. Finally, a theoretical criterion is proposed to determine the applicable region of SCE in GPEH. Subsequently, the analytical solutions of responses of a GPEH are derived. Three different interfacing circuits, including the simple AC, standard and SCE circuits, are considered in the analysis, with which the explicit expressions of power, voltage and displacement amplitude are formulated. The proposed theoretical solutions provide significant guidelines for accurate evaluation of effectiveness of GPEHs and the scheme of normalization makes it convenient to compare devices with various parameters.

The last investigative work focuses on the harvesting capability of a GPEH using the synchronized switching harvesting on inductor (SSHI) interfaces, with

preliminary experimental and simulation results presented. In both wind tunnel experiment and circuit simulation, the superiority of parallel SSHI is confirmed while series SSHI demands further investigation. The power output is increased by 43.75% with parallel SSHI compared to the standard circuit at a wind speed of 6m/s.

Last but not least, future work strategies are proposed for relative fields involving efficient small-scale wind energy harvesting, like developing multi-directional wind energy harvester and portable power generator, harvesting energy from random vibration and wind flow by installing harvesters on vehicles, etc. The prospective impact of the work presented in this thesis paves the path towards self-powered smart electronic systems which are essential for a sustainable and smart city in the future.

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CHAPTER 1 INTRODUCTION

1.1 Background

Microsystem technologies have been growing rapidly in recent years, such as wireless sensors and portable electronic devices. However, the reliance on traditional electrochemical batteries has become a significant obstacle for various miniaturized electronic devices. Problems of using the electrochemical batteries include the time consuming maintenance, expensive replacement, limited lifespan as well as periodic disposal work and induced environmental pollutions. Moreover, the bulky batteries generate a huge limitation for miniaturization in micro electromechanical systems (MEMS). To overcome these issues, some researchers have started obtaining electrical power from the ambient energy sources surrounding the devices, which is the concept of energy harvesting.

The untapped ambient energy sources, like solar, wind, thermal energy, mechanical vibration and human activities, can be scavenged as an alternative power supply to implement self-powered electronic systems like self-powered wireless sensor networks (WSNs). Studies on harvesting energy from these potential energy sources have flourished in recent years. Wind turbines have been widely employed for large scale wind energy harvesting when strong wind is available, e.g., in the offshore environments, yet they are much less efficient when scaled down to harness small wind energy. On the other hand, small-scale wind energy is a pervasive energy source, such as the wind flows in the ventilation and air conditioning systems, on the side of highways with high-speed travelling vehicles, in the wind gusts around the unmanned aerial vehicles, and highly urban areas with limited open space, etc. Harvesting such kind of small-scale wind energy can serve as an alternative power supply for WSNs and also fit the goal of sustainable development to create intelligent, autonomous and eco-friendly systems. The work presented in this thesis focuses on small-scale wind energy harvesting.

Aeroelastic instability phenomenon happens to a specific bluff body subjected to

wind flows via fluid-structure interaction, resulting in large amplitude limit cycle oscillations. The oscillating wind-induced vibrations can cause damages to the building structures or the aircrafts and hence are usually undesired in civil and aerospace engineering. However, they can be beneficially transferred into electricity using specific vibration-to-electricity transduction mechanisms, such as the electrostatic, electromagnetic, piezoelectric and magnetostrictive mechanisms, among which the piezoelectric transduction has attracted great interest due to its high power density, ease of application and non-reliance on external magnetic field or initial voltage. The oscillating mechanical strain energy during vibrations can be converted into electrical energy via direct piezoelectric effect, either in the 33 working mode or 31 working mode of the piezoelectric material. The 31 mode is mostly adopted due to its low resonant frequency which can be more likely matched by the wind-induced vibrations. The work on energy harvesting presented in this thesis uses the piezoelectric transduction mechanism in the 31 mode.

Aeroelastic instabilities, such as vortex-induced vibrations, galloping, flutter, wake-induced oscillations (like wake galloping) and turbulence-induced vibrations, can be obtained with proper designs of the harvester structure. In particular, galloping owns its advantage of large oscillation amplitude and the ability of oscillating in infinite range of wind speeds, making it a prospective choice to obtain structural vibrations for energy harvesting purpose. The power generation process based on aeroelastic instabilities incorporates a series of mutual coupling behaviors, i.e., the aeroelectromechanical coupling. The nonlinear aerodynamic force induces mechanical vibrations through the fluid-structure coupling, and then the mechanical strain energy is converted into electricity through the direct piezoelectric effects along with the electromechanical coupling. Moreover, due to the converse piezoelectric effects, mutual interaction between the fluid and the piezoelectric aspects is induced through the mechanical strain, although this interaction is insignificant. In order to comprehensively study the mutual coupling behavior and accurately predict the mechanical and electrical responses, coupled analytical modeling for small-scale wind energy harvesters is required, which should include coupled electromechanical and aerodynamic modeling parts. Moreover, the

influence of various design parameters on the power generation process deserves careful analysis in order to efficiently optimize the harvester design.

Properly modified harvester structure designs can enhance the power generation efficiency, yet at the same time require further efforts to develop the analytical models. Moreover, the harvested energy is supposed to be utilized in WSNs and other electronic devices, or to be stored in electrical energy storage systems. In either scenario, the DC signal is required. Therefore, an interface circuit for AC-DC rectification and regulation need be implemented for wind energy harvesters before they can be employed in real applications. Nevertheless, involving the advanced nonlinear power extraction interface circuit further complicates the mutual coupling behaviors between the fluid, structure and electric circuit, requiring further analysis to study the system responses. In a word, research efforts on comprehensive analytical modeling, innovative mechanical design and efficiency enhancement of small-scale wind energy harvesters are greatly desired.

1.2 Research objectives

Small-scale wind energy harvesting aims at powering wireless sensors and other low-power electronic devices mounted in remote areas with harnessable wind flow energy, thus implementing totally self-powered electronic systems. The objectives of the studies presented in this thesis include the development and comprehensive comparison and evaluation of analytical modeling methods, as well as the pursuit of various approaches to enhance the power generation performance of small-scale wind energy harvesters. Regarding the performance enhancement of aeroelastic energy harvesters, the main targets include reducing the cut-in wind speed, enlarging the effective wind speed range and increasing the harvested power at the target wind speed. In this research, effects on performance enhancement are devoted into structural modifications and interface circuit optimizations. The major objective of developing such kind of small-scale wind energy harvesters is to implement totally self-powered WSNs, both in indoor and outdoor environments. Possible wind energy sources can be either from the flows in the heating, ventilation and air conditioning (HVAC) system inside buildings, or the outdoor natural winds. These self-powered WSNs can be used to realize automatic structural health

monitoring, environmental monitoring, or a forever connected smart office and smart city. Figure 1.1 illustrates the concept of these potential applications.

Figure 1.1 Potential application areas of the proposed and analyzed wind energy harvesters. (a) Indoor environment and (b) outdoor environment

1.3 Original contributions

The original contributions of this research can be summarized as follows.

(1) An extensive literature review on the fundamentals and the state-of-the-art techniques of small scale wind energy harvesting is presented, with a focus on those based on the piezoelectric transduction mechanisms. Various aeroelastic instabilities, including the vortex-induced vibrations, galloping, aeroelastic flutter, wake-induced oscillations and turbulence-induced vibrations, are considered. Theoretical modeling methods for aeroelastic energy harvesters are summarized and discussed in detail, which are classified into mathematical modeling, equivalent circuit modeling and computational fluid dynamics simulation. Designs of harvester structures as well as interface circuits, and performance enhancing techniques are also reviewed in detail. Studies of implementing self-powered wireless sensing systems with small-scale wind energy harvesters as power suppliers are summarized.

(2) A comparative study of different tip cross-sections for small-scale wind energy harvesting based on galloping phenomenon is presented. A prototype device is fabricated with a piezoelectric cantilever and a tip body with various cross-section profiles (square, rectangle, triangle and D-shape) and tested in a wind tunnel. Experimental results demonstrate the superiority of the square-sectioned tip for the low cut-in wind speed of 2.5m/s and the high peak power of 8.4mW. An analytical model is established and verified by the experimental results. It is recommended that, for the very first time as the author knows, the square section should be used for small wind galloping energy harvesters.

(3) In order to better understand the electroaeroelastic behavior and further improve the galloping piezoelectric energy harvester (GPEH), an effective analytical model is required, which needs to incorporate both the electromechanical coupling and aerodynamic force. Available electromechanical models for GPEH include the lumped parameter one-degree-of-freedom (1DOF) model, approximated distributed parameter model based on the Rayleigh-Ritz discretization, and distributed parameter model with the Euler-Bernoulli beam representation. Each modeling

method has its own advantages. The corresponding aerodynamic models are formulated using quasi-steady hypothesis (QSH). The 1DOF model, the Euler-Bernoulli distributed parameter model using single mode and the Euler-Bernoulli distributed parameter model using multi-modes are compared and validated with the experimental results. Based on the comparison and validation, the most effective model is employed for the subsequent parametric study. The effects of load resistance, wind exposure area of the bluff body, mass of the bluff body and length of the piezoelectric sheets on the power output are investigated. These simulations can be exploited for designing and optimizing GPEHs for better performance.

(4) Most piezoelectric aeroelastic energy harvesters operate effectively only at high wind speeds or within a narrow speed range. To overcome this issue, a two degree-of-freedom (2DOF) piezoelectric aeroelastic energy harvester with a cut-out cantilever and two magnets is proposed. Translational galloping is induced with a square cross-sectioned bluff body attached at the cantilever tip. Magnetic interaction is introduced to create stiffness nonlinearity. The proposed device is demonstrated by wind tunnel tests to have a lower cut-in wind speed of 1m/s and higher power output than the conventional 1DOF harvester in the low wind speed range up to 4.5m/s.

(5) An easy but quite effective method to significantly enhance the power generation capability of aeroelastic energy harvesters is presented. The method is to attach a beam stiffener to the substrate of the harvester, which works as an electromechanical coupling magnifier. It is shown to be effective for all the three considered types of harvesters based on galloping, vortex-induced vibration and flutter, leading to a superior performance over the conventional designs without the beam stiffener, with dozens of times increase in power, almost 100% increase in the power extraction efficiency, yet with comparable or even smaller transverse displacement. Choice guidelines of optimal types of energy harvesters are also suggested based on the given wind situations where the electronic device is located.

(6) Considerable efforts have been devoted to optimizing the interface circuit to boost the power output from vibration-based piezoelectric energy harvesters (VPEH), such as impedance matching, synchronized charge extraction (SCE) and synchronized switching harvesting on inductor (SSHI). However, application of these circuits in aeroelastic energy harvesting has received far less attentions. With an experimentally validated equivalent circuit model (ECM), the feasibility of employing the SCE interface for a GPEH is investigated. The performance of SCE circuit is compared with a standard circuit, revealing three main advantages of SCE in GPEH system: first, the output power from SCE is independent of electrical load, eliminating the requirement of impedance matching and thus ensuring the flexibility of adjusting the GPEH system for practical applications; second, the SCE circuit helps to save piezoelectric materials by 75% compared to the standard circuit; third, the displacement amplitude of GPEHs with SCE is much smaller, alleviating the fatigue problem and enhancing the durability of the harvesting system. Finally, a theoretical criterion is proposed to determine the applicable region of SCE in GPEH.

(7) Very little efforts have been devoted to deriving explicit output responses of aeroelastic piezoelectric energy harvesters, especially for the cases involving sophisticated interface circuits. The analytical solutions of responses of a GPEH are proposed in this work. Three different interfacing circuits, including the simple AC, standard and SCE circuits, are considered in the analysis, with which the explicit expressions of power, voltage and displacement amplitude are derived. The optimal load and coupling are calculated for maximum power generation. The cut-in wind speeds for these circuits are also formulated. Wind tunnel experiments based on a prototype of GPEH with a square sectioned bluff body and circuit simulation based on the equivalent circuit model are carried out to validate the analysis. Recommendations of the applicability of different circuits are provided based on the observed behaviors of the circuits. The proposed theoretical solutions provide significant guidelines for accurate evaluation of effectiveness of GPEHs and the scheme of normalization makes it convenient to compare devices with various parameters.

(8) The harvesting capability of a GPEH using the SSHI interfaces is investigated, with a focus on comparing the performances of Series SSHI (S-SSHI) and Parallel SSHI (P-SSHI) with that of a standard DC interface. In both wind tunnel experiment and circuit simulation, the superiority of P-SSHI is confirmed while S-SSHI demands further investigation. The power output is increased by 43.75% with P-SSHI compared to the standard circuit at a wind speed of 6m/s.

1.4 Organization of Thesis

The thesis is divided into nine chapters including this first introductory chapter. From Chapter 2 to Chapter 8, each chapter presents a complete and independent work, while the contents are related with each other. Chapter 2 presents an extensive literature review on the state-of-the-art small scale wind energy harvesting approaches with a focus on those using piezoelectric materials. Chapter 3 presents a comparative study of different tip cross-sections for galloping piezoelectric energy harvesting, and provides the experimentally validated conclusion that the square section should be used for small wind galloping energy harvesters. Chapter 4 performs a thorough evaluation on the effectiveness and accuracy of the 1DOF model and the Euler-Bernoulli distributed parameter model of GPEHs, with the aerodynamic models formulated using quasi-steady hypothesis. Chapter 5 and Chapter 6 present efficiency enhancing techniques for small wind energy harvesters in the aspect of structural modifications. In Chapter 5, a 2DOF GPEH with a cut-out cantilever and two magnets is proposed, which is experimentally demonstrated to have a lower cut-in wind speed and higher power output than the conventional 1DOF harvester in the low wind speed range. In Chapter 6, the proposed approach of adding a beam stiffener onto the harvester's cantilever is shown to be effective for all the three considered types of harvesters based on galloping, vortex-induced vibration and flutter, with dozens of times increase in power and almost 100% increase in the power extraction efficiency over the conventional counterparts. Chapter 7 and Chapter 8 present approaches of enhancing power extraction efficiency of small wind energy harvesters by optimizing the interface circuits. In Chapter 7, the feasibility of enhancing the

output power of a GPEH with the SCE interface is investigated based on experiment and circuit simulation, showing great advantages of the SCE in GPEH system. Chapter 8 further derives the explicit analytical solutions of the mechanical and electrical responses of a GPEH, which enables fast and accurate evaluations of effectiveness of GPEHs and facilitates the optimization process with various interfaces including the simple AC, standard and SCE circuits. Finally, Chapter 9 summarizes the research presented in this thesis and provides recommendations for the future work. In particular, the preliminary experiment and circuit simulation results for a GPEH system integrated with the SSHI is provided and discussed. The outline of this thesis is shown in Figure 1.2.

Figure 1.2 Thesis outline

CHAPTER 2 LITERATURE REVIEW¹

2.1 Overview on energy harvesting

In order to sustain the operation of off-grid wireless sensor networks (WSNs) and other low-power electronic devices, the past decades have seen a dramatic rise in the interdisciplinary research area of “energy harvesting”. The ultimate objective of energy harvesting is to realize automatically self-powered WSNs and supply alternate power for small electronics, liberating them from the inconvenient replacement or recharge of the batteries, reducing the high cost of manual maintenance and eliminating the experimental pollution associated with the disposal of the chemical batteries.

The concept of energy harvesting aims at converting the ambient energy surrounding the electronic devices into electricity at a “small scale”, usually at a level of milliwatt or even less. Due to advancement in electric circuit techniques, the size and power consumption of the electronics are greatly reduced, such that the “small-scaled” harvested energy is already sufficient to continuously run a low-power electronic device or to power a single duty cycle operation for high-power device.

2.1.1 Available sources for energy harvesting

Solar energy, in the form of radiant light and heat from the sun, is commonly employed for energy harvesting for the past few decades. In direct sunlight, high power density can be obtained by solar energy converters such as the solar panels, solar photovoltaics, or artificial photosynthesis (World Energy Outlook 2011). Large power capacity is available from the sunlight. In 2010, total global capacity for solar energy production reached 39.8 GW (Solar Energy Facts). The disadvantage

¹ Content based on this chapter will be submitted for publication as “Zhao, L. and Yang, Y. (2015) Small-scale wind energy harvesting: modeling, design, efficiency enhancement and applications in self-powered wireless sensing systems”.

of solar energy harvesting is that it depends strongly on the degree of sun exposure. In situations where the sunlight is weak or not applicable, such as in the indoor environments or at night, the power density becomes quite low or completely vanishes.

Thermo gradient is another source for energy harvesting, which is based on heat flux caused by temperature differences. The potential of harvesting energy from thermal gradient was firstly discovered by Thomas J Seebeck around two centuries ago in 181, who observed a voltage generation from a thermal gradient between two dissimilar conductors (Thomas Johann Seebeck, Wikipedia). Thermoelectric material is the key for thermoelectric energy harvesting. Strasser *et al.* (2003) reported a micro-scale thermoelectric generator in the size of 1cm^2 with a peak power of $1\mu\text{W}$ achieved at a temperature drop of 5K .

The ambient vibration around the devices is also a promising energy source, which has been the focus of many recent researches on energy harvesting. Low-level mechanical vibrations are pervasive in the environment while high-level vibrations are available on machinery and vehicles. To harness energy from these “preexisting” vibrations is termed vibration energy harvesting. Vibration energy harvesters have been heavily reported in the literature with various transduction mechanisms. The main limit in vibration-based energy harvesters lies in their narrow operating bandwidth, because most harvesters perform well only around their resonant frequencies. Yet the environmental vibrations are usually random and frequency-variant within a wide frequency spectrum. Numerous efforts have been devoted to enhancing the power efficiency and broadening the bandwidth of vibration energy harvesters.

Mechanical energy from various human activities provides an alternative option for powering wearable electronic devices, such as body sensing networks for human health monitoring, biomedical devices like pacemakers, or digital entertainment devices like MP3 player. Some designs of energy harvesters based on human activities are summarized in the review paper by Anton and Sodano (2007). The

available energy level varies among different types of activities. According to González *et al* (2002), the continuous activities such as blood flow can generate 0.93W of power, while the daily discontinuous activities like walking can generate up to 67W of power.

Furthermore, fluid energy is a ubiquitous energy source existing in the ocean waves, natural wind in the outdoor environments, flows generated from human activities like daily motions and vehicle driving, and flows in the indoor heating and ventilation air conditioning ducts, etc. The technique of harvesting fluid energy in large scale turbines has been developed very well, which can generate high-level power in kilo or mega watt in places where rapid water flow or strong wind is available, like the offshore environment. A recent wind turbine model of V164-8.0 MW developed by “Vestas” is regarded as the world’s biggest wind turbine, which has a 163m diameter of rotor, a 21.124m² swept area, and an 8.0MW power output (Vestas V164-8.0 nacelle and hub). According to Global Wind Energy Council (GWEC), 4.46×10^{11} KWH of wind energy was generated worldwide in 2010, and around 2.5% of world electricity is supplied by wind power in these days (Global wind energy council, wind in numbers). Nevertheless, small scale fluid energy harvesting for powering small electronics has received only limited attention. As the power requirement of a single wireless sensor node is decreased down to mW level, some miniature designs have been reported to harvest small scale fluid energy at the level of mW or uW. Although the electromagnetic wind turbines perform well in large scale, miniaturization of such structures results in greatly reduced efficiency and increased manufacturing complexity with relatively large mechanical loss due to the relatively high viscous drag at low wind speeds.

This thesis focuses on harnessing small scale energy from uniform wind flow. The energy conversion techniques for small wind energy harvesting are based on the flow-induced vibrations. Kinetic energy in the wind is transferred to mechanical vibration energy first, it is then converted to electricity via a specific vibration-electricity transduction mechanism. In the following sections, the vibration-electricity transduction mechanisms are introduced first, followed by a

detailed review of modeling, design, efficiency improvement, and practical integration with wireless sensor networks of small wind energy harvesters.

2.1.2 Transduction mechanisms

Various small wind energy harvesters conduct power conversion based on seeking and enlarging flow-induced mechanical vibrations. The vibration energy is further converted into electricity using vibration energy harvesting approaches. There are mainly three basic vibration-to-electricity conversion mechanisms: electrostatic, electromagnetic and piezoelectric conversions.

Electrostatic energy harvesters convert vibration energy into electricity via a variable capacitor, or varactor. The harvester is usually designed in the way such that the gap distance or overlap area of a parallel-plate capacitor is varied due to vibrations. Voltage increases as capacitance decreases if the charge is constrained by keeping the capacitor open circuited, so that the potential energy stored in the capacitor is increased; otherwise, the charge is moved out of the capacitor, yielding a net harvesting current, if the voltage across the capacitor is constrained (Meninger et al., 2001; Roundy et al., 2003; Torres and Rincón-Mora, 2009). Electrostatic transduction mechanism is convenient to be integrated into micro power generating systems like MEMS (Sterken et al., 2004; Mitcheson et al., 2004; Beeby et al., 2006), which is a main advantage of this mechanism, yet it is associated with a shortcoming that it always requires an initial polarizing voltage or charge source.

Electromagnetic energy harvesters generate electrical current in a conductor, which is usually a coil, placed in a magnetic field (Faraday's law) (Amirtharajah and Chandrakasan, 1998; Elvin and Elvin, 2011; Glynn-Jones et al., 2004). Relative movement between the coil and magnet or the change in the magnetic field will induce current in the coil. Most electromagnetic energy harvesters take the form of the former way, which consist of a coil, permanent magnets and an oscillating cantilever. Either the coil or the magnets can be fixed to the cantilever in order to pursue the relative movement between each other (Beeby et al., 2006). An example of such electromagnetic energy harvester proposed by El-Hami et al (2001)

generated a power of over 1mW with a harvester volume of 240mm³ at a vibration frequency of 320Hz. The advantage of electromagnetic energy harvester is that it requires no external voltage source, and is capable of generating high-level current output. The drawback is that the output voltage is quite low (0.1-0.2V). Further voltage regulation process is necessary to make such a low voltage applicable for electronic terminals (Cook-Chennault et al., 2008). Moreover, although it performs with great success in bulk size, wafer-scale electromagnetic energy harvesters are difficult to achieve because of the relatively poor properties of planar magnets (Beeby et al., 2006).

Energy harvesting using piezoelectric materials has seen a dramatic rise in the past few years due to the high power density and simple configuration of piezoelectric energy harvesters, which usually take the form of a cantilever beam shown in Figure 2.1. The energy conversion is realized using a special characteristic of the material called “piezoelectricity”, which produces a voltage across the material when it is subjected to mechanical strain, thus transferring the strain energy into electricity. Piezoelectric transduction has higher power density and output voltage compared to the aforementioned two mechanisms. Moreover, it is easy to be integrated in micro-scale power generating systems like MEMS (Jeon et al., 2005; Dutoit et al., 2005; Lu et al., 2004). Nano-scale piezoelectric energy harvester is also proposed by Wang (2011). These advantages of piezoelectric transduction spur tremendous research interests in designing efficient piezoelectric energy harvesters (Roundy et al., 2003; Anton and Sodano, 2007). Many examples of vibration energy harvesters using piezoelectric transduction mechanism can be found in the literature (Roundy and Wright, 2004; Anton and Sodano, 2007; Cook-Chennault et al., 2008; Tang et al., 2010; Castagnetti, 2012). Due to the aforementioned advantages, the study of small wind energy harvesting presented in this thesis is conducted using piezoelectric transduction mechanism.

Figure 2.1 A typical vibration piezoelectric energy harvester

2.1.3 Fundamentals of piezoelectric energy harvesting

Before proceeding to the working principle of small wind piezoelectric energy harvesters, this section presents the basic theory of piezoelectricity and general properties of piezoelectric materials, including the piezoelectric constitutive relations, commonly used piezoelectric materials and the working modes for piezoelectric transduction mechanism.

2.1.3.1 Piezoelectricity

Piezoelectricity was first discovered in 1880 by Pierre and Paul-Jacques Curie. They found out that when a mechanical stress was applied on certain crystals, electrical charges appeared, with the voltage being proportional to the applied mechanical stress. Vice versa, mechanical strain will occur when they are subjected to electric fields. These behaviors are labeled as direct and converse piezoelectric effects, respectively. Crystals displaying such behaviors include tourmaline, tourmaline, topaz, quartz, Rochelle salt and cane sugar (Piezoelectric materials (online)).

The intrinsic piezoelectricity of piezoelectric materials is achieved by the “poling” treatment, as shown in Figure 2.2. For an “unpoled” piezoelectric material, the dipole directions are random. During poling, a very high electric field is applied in the direction of “poling direction” at a temperature called Curie temperature (T_{Curie}), and all the dipoles are oriented in the direction of the field. Due to the pinning effect produced by microscopic defects in the crystalline lattice, when removing the electric field and decreasing the temperature, most dipoles keep roughly in

alignment with the poling direction, resulting in a permanent piezoelectric effect.

Figure 2.2 Poling process of piezoelectric materials: (a) before poling, $T < T_{\text{Curie}}$, (b) during poling, $T = T_{\text{Curie}}$ and (c) after poling, $T > T_{\text{Curie}}$.

When mechanical displacement is applied to the piezoelectric material, the mechanical stress results in an electrical field and charge displacement, this phenomenon is caused by the direct piezoelectric effect, and usually termed as the “*sensing mode*” of piezoelectric operation. Vice versa, when an electric field is applied to the piezoelectric material, mechanical deformation will be induced by the converse piezoelectric effect. This interaction is usually termed as the “*actuating mode*” of piezoelectric operation.

In practice, different forms of piezoelectric materials are commonly used, with some of them being commercially available. These commonly used forms include single crystal (e.g. quartz), piezoceramic (e.g. lead zirconate titanate or PZT), thin film (e.g. sputtered zinc oxide), screen printable thick-films based upon piezoceramic powders and polymeric materials such as polyvinylidene fluoride (PVDF) (Beeby et al., 2006).

The most commonly used type of piezoelectric material for energy harvesting is the lead zirconate titanate (PZT) shown in Figure 2.3(a). PZT has high power conversion efficiency, high Young’s modulus, but is extremely brittle, and weak under tensile stresses. Therefore, it is not suitable for some circumstances where a curvature in the piezoelectric material would be induced (Sodano et al., 2004).

Polyvinylidene fluoride (PVDF) is a piezoelectric polymer with considerable flexibility (Figure 2.3(b)). However, its efficiency for power generation is quite low. An initial applied stress or deformation is able to increase its power generation efficiency. A composite material consisting of active piezoelectric fibers embedded in a polymeric matrix phase is another commonly used material. The polymer greatly increases the flexibility of the piezo-fibers, and provides an electrical insulation around the piezoelectric material. Commercially available piezoelectric composites include active fiber composites (AFC), macro fiber composites (MFC), and a patch transducer called DuraAct, as shown in Figure 2.3(c), 2.3(d) and 2.3(e), respectively.

Figure 2.3 Different types of piezoelectric materials: (a) PZT (Picture from PI Ceramic), (b) PVDF (Picture from Piezotech S.A.S) and (c) AFC (Picture from MSI.inc) (d) MFC (Picture from Smart Material Corp.) (e) DuraAct (Picture from PI Ceramic)

2.1.3.2 Piezoelectric constants and coupling mode

The piezoelectric activities of a specific type of piezoelectric material are affected by a series of piezoelectric constants. The piezoelectric strain constant d is a parameter relating the developed mechanical strain and the applied electric field, defined as (Beeby et al., 2006)

$$d = \frac{\text{strain developed}}{\text{applied electric field}} m/V \quad (2.1)$$

The electromechanical coupling coefficient k is another key factor determining the power conversion efficiency of the piezoelectric material. It can be defined as (Beeby et al., 2006)

$$k_{ij}^2 = \frac{W_i^e}{W_j^m} \quad (2.2)$$

where W_i^e is the generated electrical energy in the i axis, while W_j^m is the input mechanical energy in the j axis. Other constants include the permittivity of the piezoelectric material, ϵ , which is the dielectric displacement per unit electric field, the compliance s , which is the strain produced per unit of stress, and the piezoelectric voltage constant g , which is the electric field per unit of mechanical stress. More complete descriptions of the piezoelectric constants are given in the IEEE standards.

The linear constitutive equations for piezoelectric materials can be expressed as (Roundy et al., 2003)

$$\begin{cases} \delta = \sigma/Y + dE \\ D = \epsilon E + d\sigma \end{cases} \quad (2.3)$$

where δ , σ , Y , D and E denotes the matrix for the mechanical strain, mechanical stress, Young's modulus, electrical displacement and electric field, respectively. Other parameters have been mentioned in the above paragraph. The values of the above piezoelectric constants, however, are not constant over time, but may vary with age, stress, and temperature, etc.

Because piezoelectric materials are typically anisotropic, the values of the abovementioned piezoelectric constants differ in different directions and are related to the orientation of the poling and the direction of the applied mechanical stress. As a result, there are different electromechanical coupling modes of piezoelectric materials. Generally there exist the following two coupling modes: 33 mode and 31 mode, as shown in Figure 2.4. In the 33 mode, the force is in the same direction as the poling direction, such as the compression of a piezoelectric block that is poled on its top and bottom surfaces; while in the 31 mode, the force is applied in the direction perpendicular to the poling direction. When a piezoelectric patch is

bonded to the surface of a substrate and undergoes bending deformation, it is generally operating in the 31 mode. A bimorph/unimorph cantilever is an example of such case.

The 33 mode gives a higher electromechanical coupling than the 31 mode. Nevertheless, it is compared by Roundy *et al.* (2003) that the 31 mode has two superiorities over the 33 mode, making it dominant in the practical designs of energy harvester. First, with the 31 mode, the harvester is easy to be tuned to work at a low resonant frequency, thus better matching the ambient vibration frequencies in a natural environment. Secondly, the harvesting system operating in the 31 mode is more compliant, easy to obtain larger mechanical strains given a small input force. Due to the aforementioned advantages, in this thesis we exploit the 31 mode in the design of small wind piezoelectric energy harvesters.

Figure 2.4 Two modes of operation of piezoelectric materials depending on directions of polarization and mechanical stress

2.2 Small-scale wind energy harvesting using piezoelectric materials

In order to provide a small scale but continuous power supply for small electronic devices like the WSNs, some research has been conducted to miniaturize the electromagnetic wind turbine into centimeter scale (Howey et al., 2011; Rancourt et al., 2007; Bansal et al., 2009; Bressers et al., 2010; Xu et al., 2010). Like the large scale wind turbines, these miniaturized wind turbines operate with the Faraday's law, consisting of rotating blades around a shaft and generating power with permanent magnets and coils through electromagnetic coupling.

When a traditional electromagnetic wind turbine is scaled down, system complexity and manufacturing difficulty will be increased. The power generation efficiency is lowered down due to the relatively high mechanical friction and viscous drag on the blades at low Reynolds numbers (Kwon, 2010; Karami et al., 2013). In view of the relatively high power density of piezoelectric transduction at small scale, some researchers have proposed designs of miniaturized windmill/turbine using piezoelectric transduction (Priya et al., 2005; Chen et al., 2006; Tien and Goo, 2010; Karami et al., 2013). In these designs, piezoelectric transducers are excited through impacts with the rotating shafts or blades in the wind, transferring the impact-induced vibration energy into electricity.

Recently, much effort has been made to harvest small wind energy through the aeroelasticity phenomena. When structures are subjected to wind flows, aeroelastic instabilities will occur, such as vortex-induced vibration, galloping, flutter, turbulence-induced vibration, wake galloping and buffeting. In the field of civil or aerospace engineering, this kind of instabilities is usually undesired. The famous Tacoma Narrows Bridge collapsed down in 1940 due to these aerodynamic instability phenomena. However, the energy of structural vibrations induced by aeroelastic instability can be beneficially converted to electricity using the vibration-based piezoelectric energy harvesting approach. In an aeroelastic energy harvester, piezoelectric transducers are not excited by blades as in a turbine, but directly interact with wind via a specific type of aeroelasticity. This thesis focuses on small wind piezoelectric energy harvesting by exploiting such kinds of aeroelasticity phenomena.

2.2.1 Aeroelastic instabilities

In this section, the mechanisms and characteristics of different types of aeroelastic instabilities are briefly introduced, including the vortex-induced vibration, galloping, flutter, wake galloping and turbulence-induced vibration.

2.2.1.1 Vortex-induced vibration

Vortex-induced vibration (VIV) is one of the most classic aeroelastic instability phenomena. It is frequently observed in many engineering structures, such as tall buildings, slender chimneys, electric power lines, cables on bridges, marine cables, stacks, heat exchangers, offshore structures and other aerodynamic and hydrodynamic structures.

When a bluff body is subjected to a steady and uniform flow with high enough Reynolds number like 100, the effect of viscosity of the flow is only dominant near the boundary of the bluff body. The flow is separated by the bluff body, forming two shear layers on each side, from which vortices are formed periodically (Williamson, 1996; Païdoussis *et al.* 2011; Facchinetti *et al.*, 2004). The creation process of the continuous alternating vortices in the downstream of the bluff body is called “vortex shedding”. Figure 2.5 gives a brief schematic of vortex shedding behind a bluff body. The behavior of vortex shedding has been extensively studied both theoretically and experimentally. For a thorough literature review, interested readers are referred to the review paper of Williamson (1996). The alternating vortices are shed to the wake at the “vortex shedding frequency” ω_f , defined as

$$\omega_f = 2\pi St \frac{U}{L} \quad (2.4)$$

where l is the reference length scale which is generally taken as the cross-flow frontal dimension of the bluff body; U is the wind velocity; and St is the Strouhal number. For some section shapes commonly employed in civil structures, the corresponding Strouhal numbers are given in Figure 2.6, in the limit of large Reynolds numbers (say 1000) (Païdoussis *et al.* 2011).

Figure 2.5 Schematic of vortex shedding

Figure 2.6 Strouhal number for some commonly employed section shapes in civil

structures at large Reynolds number

Figure 2.7 Typical response of amplitude of oscillation versus wind speed for VIV

Under the action of vortex shedding, the bluff body will undergo, usually periodic, oscillations, which is the so called VIV, resulting in a complex interaction between the bluff body and the vortices. Two main features of VIV are summarized by Barrero-Gil et al. (2011). Firstly, there is a lock-in regime of wind speed, where the vortex shedding frequency is kept synchronized with the oscillation frequency associated with large amplitude oscillations; secondly, hysteresis appears in the displacement response of a cylinder undergoing VIV. More detailed research on VIV phenomenon has been reviewed by Sarpkaya (2004) and Williamson and Govardhan (2004). Typically, the range of wind speed giving significant vortex-induced vibrations for effective power generation is narrow, which remains a main constraint for VIV-based energy harvesting. A typical response curve of oscillation amplitude versus wind speed for VIV in a steady flow is depicted in Figure 2.7.

2.2.1.2 Galloping

Figure 2.8 Schematic of a bluff body subjected to wind flows

Translational galloping is a self-excited phenomenon giving rise to large amplitude oscillations of bluff body when subjected to wind flows. However, it is not really “self-excited”, but the governing equation of the vibration due to galloping can be written in a way that the input aerodynamic force is hidden, making the motion seem to be self-excited.

Assume that a bluff body is mounted on a spring as shown in Figure 2.8. Wind flows from the right to the left with a velocity of U . The governing equation of the galloping motion can be written as

$$M\ddot{w} + C\dot{w} + Kw = F_z \quad (2.5)$$

where w is the vertical position of the mass center of the bluff body; M is the mass of the bluff body; C is the damping coefficient; K is the stiffness of the spring and F_z is the aerodynamic force. The overdot denotes differentiation with respect to time t . F_z depends on the geometry properties of the bluff body, the wind velocity U and the bluff body vibration velocity $\dot{w}$. F_z can be expressed as a polynomial function of the angle of attack α , and for simplicity, it can be written as

$$F_z = XU^2(A\alpha - B\alpha^3) \quad (2.6)$$

where X is a constant related to the air density and the geometry properties of the bluff body, and A and B are both positive empirical coefficients. Quasi-steady aerodynamics is applicable here because the frequency of vibration caused by galloping is low enough (Païdoussis *et al.* 2011). If the bluff body undergoes only translational oscillation without rotation, the angle of attack α can be expressed as

$$\alpha = \frac{\dot{w}}{U} \quad (2.7)$$

Substituting Eq. (2.6) and (2.7) into Eq. (2.5), and dividing both sides of Eq. (2.5) by M yields

$$\ddot{w} + (2\zeta\omega_n - \frac{XA}{M}U + \frac{XB}{MU}\dot{w}^2)\dot{w} + \omega_n^2 w = 0 \quad (2.8)$$

where ζ is the damping ratio and ω_n is the natural frequency of the system. In such a way as in Eq. (2.8), the aerodynamic force can be considered as an effective damping and is “hidden” as mentioned before.

First, the bluff body is set still. With any small disturbance, $\dot{w}$ is arbitrarily small, so we only need to consider the first two terms of the damping in the brackets in Eq. (2.8). For sufficiently small U , the term $(2\zeta\omega_n - XAU/M)$ is positive, thus the damping of the system is positive and the oscillations will be damped to the zero equilibrium. When the wind velocity increases and exceeds a certain value, the term $(2\zeta\omega_n - XAU/M)$ will become negative, giving rise to self-excited oscillation of the bluff body, which is usually called a Horf bifurcation. When $\dot{w}$ is large enough due to the increasing amplitude of oscillation, the third term of damping expression $(XB/MU)\dot{w}^2$ should be taken into account, making the overall damping non-negative. When the overall damping reaches zero, the vibration amplitude will be stable and the limit cycle oscillation will occur. Due to the self-excited and self-limiting characteristics of galloping, it is deemed a prospective energy source for energy harvesting. Figure 2.9 shows the typical response curves of galloping amplitude versus wind speed. The form of responses mainly depends on the cross section shape of the bluff body and the flow condition, i.e., smooth flow or turbulent flow. Besides the supercritical response that is self-excited as discussed above, there exists the subcritical galloping response (Figure 2.9(c)), where a large initial perturbation is necessary to induce galloping. This type of galloping is a feasible choice for energy harvesting if external perturbations are available and the ambient wind speed is decreasing. However, if these conditions are not satisfied, the self-excited supercritical galloping is the superior choice. Some cross sections also display a hysteresis region as shown in Figure 2.9(b).

Figure 2.9 Typical response of amplitude of oscillation versus wind speed for galloping (a) supercritical bifurcation (b) supercritical bifurcation with hysteresis and (c) subcritical bifurcation.

The criterion for self-excited galloping (i.e., without the requirement of an external perturbation) is expressed as (Den Hartog, 1956),

$$\frac{\partial C_{Fz}}{\partial \alpha} > 0 \quad (2.9)$$

where C_{Fz} is the total aerodynamic force coefficient in the direction normal to the incoming flow (Païdoussis *et al.* 2011).

2.2.1.3 Flutter

The concept of exploiting flutter oscillation in a mill wing as a possible power generator can be traced back to several decades ago (Ly and Chasteau, 1981; McKinney and DeLaurier, 1981; Schmidt, 1992; Jones and Platzer, 1999). Flutter instability was initially studied in the field of aeronautics. The flutter instabilities of aircraft wings and empennage structures are common examples of this type of phenomena, which can cause severe damage to the flight vehicles.

There are several forms of flutter instabilities, such as the cross-flow flutter of a cantilever or a flexible belt (De Marqui et al., 2010; Humdinger Wind Energy, Windbelt Innovation), the flutter of a cantilevered plate in axial flow (Huang 1995; Tang et al., 2003), and the flutter of an airfoil with coupled torsion and bending motions (Theodorsen, 1934; Peters et al., 1995; Hodges and Pierce, 2002). All the above types of flutter have been employed to harvest the flow energy. Among these types of flutter, the flutter of an airfoil has been studied with the greatest enthusiasm. In this section, we focus on the mechanism of the aeroelastic flutter of an airfoil which undergoes pitch and plunge motions simultaneously.

A formal definition of aeroelastic flutter is described in the book of Hodges and Pierce (2002) as: a dynamic instability of a flight vehicle associated with the interaction of aerodynamic, elastic, and inertial forces. It is a self-excited oscillatory motions caused by the aerodynamics forces coupling with the natural modes of vibration. The magnitude of oscillation increases with the wind speed once it surpasses the critical value, which is the so called flutter speed. Typical response

curves of amplitude versus wind speed for flutter in a steady flow is depicted in Figure 2.10. As shown in the responses, flutter amplitude can either infinitely increase with wind speed associated with supercritical bifurcation, or gradually diminish or even vanish at a cut-out wind speed associated with subcritical bifurcation, depending on the parameters of the structural system (Bryant et al., 2012b; Abdelkefi et al., 2012a). A typical section subject to aeroelastic flutter is an airfoil, with pitch and plunge modes as shown in Figure 2.11. Flutter occurs when the two modal frequencies of the corresponding torsion and bending modes coalesce with each other. Since it results from the convergence of two structural modes, it is also referred to as the coupled mode flutter (Hodges and Pierce, 2002). The critical condition at the stability boundary is called the flutter boundary, at which the corresponding wind speed is called the flutter speed. Below the flutter speed, the system will always return to stable status because the airflow provides positive damping. Above the flutter speed, however, any small perturbations can cause exponentially increased amplitude of oscillations due to the flow induced negative damping. Finally, because of the nonlinearity in the system (material, geometric or aerodynamic nonlinearity), the amplitude of oscillation will become constant and the system undergoes the limit cycle oscillation.

Figure 2.10 Typical responses of amplitude of oscillation versus wind speed for flutter: (a) supercritical bifurcation and (b) subcritical bifurcation.

Figure 2.11 Schematic of a typical flapping wing undergoing pitch and plunge motions, i.e., θ and h (figure from Hodges and Pierce, (2002)).

From a quantitative perspective, the flutter boundary depends on the real and imaginary parts of the two complex conjugate pairs of eigenvalues, e.g., $\Gamma_1 \pm i\Omega_1$ and $\Gamma_2 \pm i\Omega_2$, corresponding to the bending and rotation modes using the famous p method (Hodges and Pierce, 2002). The negative of the real component $-\Gamma$ stands for the modal damping; while the imaginary component Ω stands for the modal frequency. When the wind speed increases from zero, the values of the two imaginary components gradually approach each other, while the values of the real components are both negative. At the flutter boundary, the two imaginary components coalesce with each other but do not fully converge, while one of the two real components becomes zero. This point represents that the system no longer undergoes decaying damped oscillations under external perturbations, but is associated with growing amplitude oscillations. Fully coupled pitch-plunge oscillations occur, with the flow energy being converted into vibration energy. Figure 2.12 shows the typical variations of modal frequency and modal damping with wind speed obtained using the p method.

Figure 2.12 Typical variations of modal frequency and modal damping with wind speed

2.2.1.4 Wake-induced oscillations

Wake-induced oscillations occur to pairs or groups of cylinders, when one or more cylinders are positioned in the wake of one windward cylinder. In engineering applications, this type of oscillations frequently happen to bundles of transmission lines, twin cables in the offshore structures, twin slender chimneys and arrays of heat-exchanger tubes, etc. The aerodynamic characteristics of the vibrations of the leeward cylinder (or cylinders) depend significantly on the arrangement of cylinders as well as the Reynolds number. Specifically, for twin cylinders, the spacing ratio L/D , with L indicating the distance between centers of cylinders and D being the diameter of the cylinder, is a key parameter that determines the characteristics of wakes and the induced vibrations.

Interference galloping

Interference galloping occurs to twin cylinders closely located to each other, say, $L/D \leq 3$. As indicated by Ruscheweyh (1983), the interference galloping is a self-excited oscillation phenomenon, with the onset wind speed depending on several factors like mass-damping parameter, spacing of the cylinders and the specific interference galloping criterion. It was found that large amplitude oscillation of the leeward cylinder is induced by the accelerated gap flow and the

surrounding accelerated outside flow (Shiraishi, 1986). The initial vertical position of the leeward cylinder relative to the windward one has a great influence on the dynamic characteristics.

Wake galloping

Wake galloping also occurs to twin cylinders, which are often two circular cylinders of tandem arrangement (i.e., the windward cylinder lies along the centerline of the wake of the leeward cylinder), causing periodic oscillations of the leeward cylinder due to the wake interference. It is found that wake galloping can happen when L/D is between 1.5 and 6 (Tokoro et al., 2000). A schematic of wake galloping is shown in Figure 2.13(a). The amplitude of wake galloping also increases unlimitedly with wind speed (Jung and Lee, 2011), like the case of galloping shown in Figure 2.9. Oscillations due to wake galloping have been successfully utilized to harvest the flow energy by several researchers (Jung and Lee, 2011; Hobbs and Hu, 2012; Abdelkefi et al., 2013d). Due to the complexity of the wakes and the mutual effects of numerous parameters, these studies were mainly conducted via experimental methods. Details of the harvester design will be reviewed in Section 2.3.

Figure 2.13 (a) Schematic of wake galloping of parallel cylinders (Jung and Lee, 2011) (b) typical trajectory of the leeward cylinder during wake-induced flutter (Païdoussis *et al.* 2011).

Wake- induced flutter

Wake-induced flutter occurs to the twin or groups of cylinders with large gaps, i.e., $L/D=10\sim20$ (Tokoro et al., 2000). A typical engineering example of this type of instability is the coupled movement of bundles of transmission conductors. Unlike galloping which is a one degree-of-freedom (1DOF) damp controlled instability, wake-induced flutter is governed by aerodynamic stiffness and requires at least two degree-of-freedom (2DOF) (Païdoussis *et al.* 2011). Near the instability boundary, the two modes (the in-plane and out-of-plane modes) coalesce with each other, like the case of flutter, which gives it the name “wake-induced flutter”. The trajectory of the vibration of a flexibly mounted leeward cylinder in the wake of a fixed windward cylinder is usually an ellipse shown in Figure 2.13(b). Due to its potential to cause serious damages in engineering applications with its associated large amplitude of vibration, extensive experimental and theoretical studies have been conducted to analyze the dynamic characteristics (Païdoussis *et al.* 2011). Up to now, this kind of wind induced instability has not been applied in energy harvesting studies. However, it is definitely a feasible alternative to extract the flow energy.

2.2.1.5 Turbulence-induced vibration

The above mentioned aeroelastic instabilities, including VIV, galloping, flutter and wake galloping, have one important common feature that their occurrences require the wind speed to be beyond the critical wind speed, i.e., the cut-in or onset wind speed. Turbulence-induced vibration (TIV), however, is unavoidable as long as the turbulent flow is in contact with an elastic structure (Au-Yang, 2001; Hobeck, 2014). This makes it an advantageous alternative instability source for energy harvesting purpose.

TIV frequently occurs to natural vegetations like tree leaves, wheat and grass. Inspired by the motions of grass under turbulent flow, Hobeck and Inman (2012a) developed a “piezoelectric grass harvester”, which consists of an array of vertically erected piezoelectric cantilevers. These cantilevers undergo vigorous vibrations under proper turbulent flow conditions, converting the induced strain energy into electricity through piezoelectric transduction.

Large amplitude oscillations induced by turbulence occur at the wind speed where the vortex shedding frequency matches the fundamental frequency of the elastic structure to induce resonance (Akaydin et al., 2010; Hobeck and Inman, 2012a). Therefore, there exists a critical wind speed for the maximum amplitude of displacement. Yet the amplitude will never be zero when the wind speed is below or beyond this critical value. Figure 2.14 shows the typical response of the amplitude of oscillation versus wind speed for TIV.

Figure 2.14 Typical response of amplitude of oscillation versus wind speed for TIV

2.2.2 Mathematical modeling - Part I: electromechanical model

Various modeling methods for aeroelastic piezoelectric energy harvesters have been proposed in the literature. Generally, these methods can be classified into three categories: the mathematical modeling method, the equivalent circuit modeling method, and the computational fluid dynamics method. This section presents the mathematical modeling method for aeroelastic piezoelectric energy harvesters.

The coupling effects for aeroelastic piezoelectric energy harvesters contain two parts: the electromechanical coupling between the piezoelectric material and the mechanical structure, and the aeroelastic coupling between the mechanical structure and the incoming flow. Therefore, the mathematical model for aeroelastic piezoelectric energy harvesters also includes two parts: the electromechanical model and the aerodynamic model.

Electromechanical modeling methods have been widely studied for vibration-based piezoelectric energy harvesting. For aeroelastic piezoelectric energy harvesting, the analysis of the electromechanical coupling is based on the vibration energy

harvesting techniques. Electromechanical models include the uncoupled and coupled 1DOF model, which is also called the lumped parameter model, the uncoupled and coupled distributed parameter model, and the approximate distributed parameter model (Reyleigh-Ritz approach based model). An electromechanical model generally includes two equations: the mechanical equation, and the electrical circuit equation. Erturk and Inman (2008c) discussed the issues in the electromechanical modeling of vibration-based piezoelectric energy harvesters. The existing models are presented in this section.

2.2.2.1 Lumped parameter model

Uncoupled 1DOF model

Representing the mechanical domain as a mass-spring-damper system (1DOF model) can obtain useful fundamental insights of the energy harvester. This method was firstly applied to electromagnetic generators by Williams and Yates (1996). Considering a magnetic seismic mass moving inside a coil as the microelectric generator shown in Figure 2.15, the governing equation for this system is

$$m\ddot{z} + c\dot{z} + kz = -m\ddot{y} \quad (2.10)$$

where y is the base excitation; z is the displacement of the seismic mass m relative to the base excitation; k is the spring constant; and c is the total damping, which contains the mechanical damping and electrical damping induced by electromagnetic energy harvesting. Regarding the coupling effect as viscous damping is suitable for electromagnetic energy harvesting, but is not proper for piezoelectric energy harvesting, since the effect of piezoelectric coupling is more sophisticated than the simple viscous damping effect.

Figure 2.15 Schematic of an electromagnetic generator (Williams and Yates, 1996).

Coupled 1DOF model

Dutoit *et al.* (2005) proposed a coupled 1DOF model for a piezoelectric energy harvester working in the 33 mode (Figure 2.16). The governing equations are

$$\ddot{w} + 2\zeta_m \omega_n \dot{w} + \omega_n^2 w - \omega_n^2 d_{33} v = -\ddot{w}_B \quad (2.11)$$

$$R_{eq} C_P \dot{v} + v + m_{eff} R_{eq} d_{33} \omega_n^2 \dot{w} = 0 \quad (2.12)$$

where w_B is the base displacement; w is the displacement of the proof mass relative to the base; v is the voltage output; m_{eff} is the effective mass; ζ is the mechanical damping ratio; ω_n is the undamped natural frequency; and C_P is the capacitance of the piezoceramic. The backward coupling effect of the electric output is treated as $-\omega_n^2 d_{33} v$, making the model a “coupled” one. Equation (2.11) is the mechanical equation of motion, and Eq. (2.12) is the electrical circuit equation.

Figure 2.16 Schematic of 1DOF harvester proposed by Dutoit *et al.* (2005).

1DOF correction factor

A piezoelectric energy harvester usually consists of a cantilever beam connected with a tip mass (proof mass) at the free end. Erturk and Inman (2008c) showed that if the proof mass of the harvester is not much larger than the mass of the cantilevered beam, the uncoupled and coupled 1DOF models underestimate the power output due to the inaccurate consideration of the contribution of the distributed beam mass to the excitation amplitude. The uncoupled 1DOF model should be modified with correction factors as

$$m \ddot{z} + c \dot{z} + kz = -\mu_1 m \ddot{y} \quad (2.13)$$

$$m \ddot{z} + c \dot{z} + kz = -\kappa_1 m \ddot{y} \quad (2.14)$$

where μ_1 and κ_1 are the correction factors for transverse vibrations and longitudinal vibrations, respectively, given by

$$\mu_1 = \frac{(M_t/mL)^2 + 0.603(M_t/mL) + 0.08955}{(M_t/mL)^2 + 0.4637(M_t/mL) + 0.05718} \quad (2.15)$$

$$\kappa_1 = \frac{(M_t/mL)^2 + 0.7664(M_t/mL) + 0.2049}{(M_t/mL)^2 + 0.6005(M_t/mL) + 0.161} \quad (2.16)$$

where M_t is the proof mass; m is the distributed mass of the cantilever and L is the cantilever length. Moreover, applying κ_1 to the coupled 1DOF model, the mechanical equation of motion Eq. (2.11) becomes

$$\ddot{w} + 2\zeta_m \omega_n \dot{w} + \omega_n^2 w - \omega_n^2 d_{33} v = -\kappa_1 \ddot{w}_B \quad (2.17)$$

The subscript 1 stands for the fundamental mode.

2.2.2.2 Distributed parameter model

In order to take the effects of higher vibration modes into account, Erturk and Inman (2008a) presented a distributed parameter model for a unimorph vibration energy harvester without the proof mass shown in Figure 2.17. The governing mechanical equation is

$$\begin{aligned} \frac{\partial^2}{\partial x^2} \left[YI \frac{\partial^2 w_{rel}(x,t)}{\partial x^2} \right] + c_s I \frac{\partial^5 w_{rel}(x,t)}{\partial x^4 \partial t} + c_a \frac{\partial w_{rel}(x,t)}{\partial t} + m \frac{\partial^2 w_{rel}(x,t)}{\partial t^2} \\ + \theta v(t) \left[\frac{d\delta(x-x_1)}{dx} - \frac{d\delta(x-x_2)}{dx} \right] = -m \frac{\partial^2 w_b(x,t)}{\partial t^2} - c_a \frac{\partial w_b(x,t)}{\partial t} \end{aligned} \quad (2.18)$$

where YI is the average bending stiffness; I is the equivalent area moment of inertia of the composite cross section; L is the length of the beam; m is the mass per unit length; θ is the electromechanical coupling coefficient; c_s and c_a are the strain rate damping coefficient and viscous air damping coefficient, respectively; w_b and w_{rel} are the base excitation and the deflection relative to the base motion, respectively; v is the output voltage of the energy harvester; and $\delta(x)$ is the Dirac delta function.

Figure 2.17 Unimorph piezoelectric energy harvester under base excitation (Erturk and Inman, 2008a)

Substituting the modal expansion of the relative deflection given by

$w_{rel}(x,t) = \sum_{r=1}^{\infty} \phi_r(x) \eta_r(t)$, the coupled mechanical equation in modal coordinates can

be obtained as

$$\ddot{\eta}_r(t) + 2\zeta_r \omega_r \dot{\eta}_r(t) + \omega_r^2 \eta_r(t) + \chi_r v(t) = -\frac{\partial^2 w_b(x,t)}{\partial t^2} \left(m \int_0^L \phi_r(L) dx \right) \quad (2.19)$$

Considering the simple electrical circuit consisting of a resistive load only, the coupled electrical circuit equation is

$$\frac{v(t)}{R_l} + C_p \dot{v}(t) - \sum_{r=1}^{\infty} \phi_r \dot{\eta}_r(t) = 0 \quad (2.20)$$

In Eqs. (2.19) and (2.20), $\phi_r(x)$ and $\eta_r(t)$ are the modal eigenfunction and modal coordinate, respectively, and χ_r and ϕ_r are both electromechanical coupling coefficients.

2.2.2.3 Rayleigh-Ritz type of approximate distributed parameter model

For harvesters with more complicated structures than a uniform cantilever beam, the deviation of accurate analytical solutions would be either more cumbersome or impossible. An approximate coupled distributed parameter model based on the Rayleigh-Ritz method was proposed by several researchers (Dutoit *et al.*, 2005; Sodano *et al.*, 2004). Elvin and Elvin (2009b) presented the Rayleigh-Ritz type coupled electromechanical equations given by

$$\mathbf{M}\ddot{\mathbf{r}} + \mathbf{C}\dot{\mathbf{r}} + \mathbf{K}\mathbf{r} - \mathbf{\Theta}\mathbf{v} = -\ddot{w}_b \left(\int_0^L \mathbf{m}\boldsymbol{\phi}(x) dx + M_t \boldsymbol{\phi}(L) \right) \quad (2.21)$$

$$\mathbf{\Theta}^T \dot{\mathbf{r}} + \mathbf{C}\mathbf{v} = \mathbf{I} \quad (2.22)$$

where $\mathbf{M}$, $\mathbf{C}$, $\mathbf{K}$ and $\mathbf{\Theta}$ are the mass, damping, stiffness and piezoelectric coupling matrices, respectively; $\mathbf{r}$ is the displacement vector; $\mathbf{m}$ is the mass distribution per unit length; M_t is the proof mass; w_b is the base excitation; $\boldsymbol{\phi}$ is the vector of assumed mode shape which could be any admissible function; and $\mathbf{I}$ and $\mathbf{v}$ are the current and voltage vectors, respectively.

Detailed comparison and discussion on electromechanical modeling method involved in small wind piezoelectric energy harvesting based on galloping will be presented in Chapter 4.

2.2.3 Mathematical modeling - Part II: aerodynamic model

In the above section, the electromechanical models are composed of the coupled mechanical equation and the circuit equation. The right hand side of the coupled mechanical equations is the forcing term due to base excitations. However, for aeroelastic piezoelectric energy harvesting, the forcing term corresponds to the aerodynamic force, usually at the free end, exerted by the incoming flow on the harvester. The mathematical model for the aerodynamic force depends on the specific aeroelastic instability phenomenon based on which the aeroelastic energy harvester is designed to operate.

There have been many studies in the literature on the mathematical modeling of different types of aeroelastities, by researchers in both the area of aerodynamics and the area of small scale wind energy harvesting (Païdoussis et al., 2011; Hodges and Pierce, 2002; Bryant and Garcia, 2011; Williamson, 1996; Sarpkaya, 2004; Facchinetti et al., 2004; Barrero-Gil et al., 2011; Sirohi and Mahadik, 2011; Zhao et al., 2013). This section presents the most classic aerodynamic modeling methods for each type of aeroelastic instability, which are most frequently employed to calculate the response of the aeroelastic energy harvesters. Other related methods will be included in the review of different designs of aeroelastic energy harvesters in Section 2.3.

2.2.3.1 Aerodynamic model for vortex-induced vibration

Figure 2.18(a) shows the schematic of a typical energy harvester based on vortex induced vibration (VIV), which consists of a cylinder as the bluff body undergoing VIV, and a piezoelectric cantilever fixed at one end as the flexible support. The complex aerodynamic force on the bluff body due to vortex shedding has been enthusiastically studied for many years. Early studies on VIV have mainly focused

on 2D domain features, i.e., without considering the difference of elastic deformations along the cylinder's axial direction. In order to perform a preliminary evaluation on the attainable maximum power conversion efficiency of a VIV based harvester (Figure 2.18(a)), Barrero-Gil *et al.* (2011) presented a simplified mathematical model for a cylinder undergoing VIV, supported with a spring and damper. The schematic is shown in Figure 2.18(b). The equation of motion of the cylinder in the cross-flow direction is

$$m(\ddot{y} + 2\zeta\omega_n\dot{y} + \omega_n^2 y) = F_y(t) \quad (2.23)$$

where m is the mass per length; ζ is the damping ratio; ω_n is the fundamental frequency; F_y is the aerodynamic force per length exerted on the cylinder due to vortex shedding, expressed as

$$F_y(t) = \frac{1}{2}\rho_a U^2 D C_y(t) = \frac{1}{2}\rho_a U^2 D C_y \sin(2\pi f t + \varphi) \quad (2.24)$$

where ρ_a is the air density; U is the wind speed; D is the characteristic dimension (diameter here) of the cylinder; f is the oscillation frequency in Hz; φ is the phase difference between the aerodynamic force and the cylinder displacement; and C_y is the aerodynamic force coefficient. The steady state harmonic vibration can be depicted as

$$y(t) = A \sin(2\pi f t) \quad (2.25)$$

By substituting Eq. (2.24) into (2.23), the amplitude and oscillation frequency of vortex induced vibration were obtained and further normalized into dimensionless form as

$$A^* = \frac{C_y \sin \varphi}{16\pi^2 m^* \zeta} \left(\frac{U^{*2}}{f^*} \right) \quad (2.26)$$

$$f^* = \sqrt{1 - \frac{C_y \cos \varphi U^{*2}}{8\pi^2 m^* A^*}} \quad (2.27)$$

Employing the experimental measured results of the coefficients of $C_y \sin \varphi$ and $C_y \cos \varphi$ in the plane (A^*, V^*) ($V^* = U^*/f^*$), Barrero-Gil *et al.* (2011) iteratively calculated the normalized amplitude and frequency of VIV in Eqs. (2.26) and (2.27). For each fixed value of reduced wind speed V^* , A^* and f^* are increased by the respective step size, with $C_y \sin \varphi$ and $C_y \cos \varphi$ for a specific pair of (A^*, V^*) calculated by 2D spline interpolation. The process is finished until Eqs. (2.26) and (2.27) are

satisfied within a prescribed tolerance.

Figure 2.18 Schematic of (a) a typical VIV based energy harvester and (b) a cylinder undergoing VIV (Barrero-Gil *et al.*, 2011)

Besides the simplified model of Barrero-Gil *et al.* (2011), the phenomenological models based on wake oscillators have been extensively used and modified to simulate the near wake vortex shedding dynamics (Balasubramanian *et al.*, 2000; Facchinetti *et al.*, 2002). Besides simulating the 2D domain dynamics of VIV, wake oscillators have been frequently employed to study the 3D domain features of slender structures subject to VIV, such as tensioned cables, where the dynamics along the span wise direction has also been considered. Here we introduce the coupled models in the work of Facchinetti *et al.* (2004). A nonlinear wake oscillator described in a van der Pol equation is coupled with the structure oscillator, which describes the equation of motion of the bluff body. The schematic of the flexibly mounted cylinder is similar to that in Figure 2.18(b). In Chapter 6, this wake oscillator-based model is employed to predict the power response of a VIV harvester and verify the effectiveness of the proposed strategy of enhancing the power output with a beam stiffener (Zhao and Yang, 2015).

The structure oscillator is given by

$$(m + m_f) \left(\ddot{y} + 2\zeta\omega_n\dot{y} + \frac{r_f}{m + m_f}\dot{y} + \omega_n^2 y \right) = S \quad (2.28)$$

Besides the parameters that have already been described in Eq. (2.23), m_f and r_f are the distributed fluid-added mass and damping, respectively, expressed as

$$m_f = \frac{C_M \rho D^2 \pi}{4} \quad (2.29)$$

$$r_f = \frac{C_D}{4\pi St} \omega_f \rho D^2 \quad (2.30)$$

where C_M is the fluid added mass coefficient and set as $C_M=1$ for a circular cross section; C_D is the mean sectional drag coefficient and set as $C_D=2.0$ in the range of Re : $300 < Re < 1.5 \times 10^5$. S is the vortex shedding induced alternating force which is coupled with the wake oscillator. The nonlinear wake oscillator is described in the van der Pol equation

$$\ddot{q} + \varepsilon \omega_f [q^2 - 1] \dot{q} + \omega_f^2 q = F \quad (2.31)$$

where q is a dimensionless wake variable, ε is a constant van der Pol parameter, and F is the forcing term of the wake oscillator. The structure-fluid coupling terms, S and F , are related to q and acceleration, as

$$S = \frac{1}{4} \rho D U^2 C_{L0} q, \quad F = \frac{A}{D} \ddot{y} \quad (2.32)$$

where C_{L0} is the reference lift coefficient on a fixed cylinder undergoing vortex shedding, set as $C_{L0}=0.3$ in the range of $300 < Re < 1.5 \times 10^5$; and A is a constant scaling parameter of the F . It should be mentioned that according to Facchinetti et al. (2004), there exist other two coupling forms besides the acceleration coupling introduced here, which are, respectively, the displacement coupling $F=(A/D)y$, and the velocity coupling $F=(A/D) \dot{y}$. The mentioned constant parameters, ε and A , can be obtained from experiments and are suggested being taken as $\varepsilon=0.3$ and $A=12$ for all the above three coupling methods (Facchinetti et al., 2004).

2.2.3.2 Aerodynamic model for galloping

Consider a bluff body undergoing galloping shown in Figure 2.8. The aerodynamic model for galloping is based on the quasi-static hypothesis (Den Hartog, 1956; Paidoussis *et al.*, 2011), which is applicable to most cases of galloping since the characteristic timescale of flow (U/h) is small compared to the characteristic timescale of oscillation ($2\pi/\omega_n$). Mathematical modeling of the aerodynamic force due to galloping is given by

$$F_{tip} = \frac{1}{2} \rho_a h l U^2 C_{Fz} \quad (2.33)$$

where h is the frontal dimension facing the wind flow; l is the length of the tip body; and C_{Fz} is the total aerodynamic force coefficient. It is convenient to express C_{Fz} with a polynomial expansion, which is usually obtained by fitting to the experimental results, given by (Païdoussis *et al.*, 2011)

$$C_{Fz} = \sum_i A_r \left(\frac{\dot{w}}{U} \right)^r, r = 1, 2, 3, \dots \quad (2.34)$$

where A_r is the empirical coefficient for the polynomial fitting. For numerical values of A_r for several cross section shapes in smooth flow, readers are referred to Païdoussis *et al.* (2011) after the work of Novak (1969) and Novak and Tanaka (1974).

It should be noted that turbulence in the flow varies the values of A_r . Take the D-section for instance, referring to the galloping criterion in Eq. (2.9), $A_1 = \partial C_{Fz} / \partial \alpha = -0.097431 < 0$ in smooth flow, so the D-section is not able to reach “self-excited” galloping, but an external large perturbation is necessary. However, if the flow condition is changed to be with 11% turbulence density, A_1 becomes $0.79 > 0$ for the D-section (Barrero-Gil *et al.*, 2010), which enables it to undergo self excited galloping. Moreover, the degree of the expansion polynomial of C_{Fz} affects the response of galloping. For example, if a fifth or seventh-order polynomial representation of C_{Fz} for a square section is used, the predicted response of vibration amplitude versus wind speed shows a hysteresis phenomenon in a specific portion, as shown in Figure 2.9(b). However, if a third-order polynomial is used, the hysteresis region disappears, with the response similar to that in Figure 2.9(a). We have determined that a third-order polynomial predicts acceptable responses since the hysteresis phenomenon is hard to observe in the experiment of galloping-based harvester (Chapter 4).

Besides using the expansion polynomial of C_{Fz} to calculate the galloping force, another method is to perform a table lookup to find the corresponding values of lift or drag coefficients at a specific angle of attack directly from the measured data. Sirohi and Mahadik (2011, 2012) employed this table lookup method to calculate

the galloping forces on a triangular section and a D-section for the purpose of wind energy harvesting.

2.2.3.3 Aerodynamic model for flutter

As mentioned in Section 2.2.1.3, among the frequently studied three forms of flutter, i.e., cross flow flutter of plates or belts, axial flow flutter of cantilevered plates, and modal convergence flutter of airfoils, the airfoil flutter has attracted the most interests for energy harvesting. In this section, we introduce the theoretical modeling methods that are suitable for predicting the power response of an energy harvester based on the aeroelastic flutter of an airfoil. The aerodynamic models are classified into two main categories: the linear aerodynamic model and the nonlinear aerodynamic model. The former is often exploited to conduct the flutter boundary analysis, while the latter is required to analyze the limit cycle oscillations beyond the flutter boundary. The power output from an airfoil flutter based energy harvester can be readily calculated by incorporating the aforementioned electromechanical model with one of the following aerodynamic models.

(1) Linear aerodynamics

- **Steady flow theory**

As shown in Figure 2.11, a typical airfoil section subjected to wind flows is elastically mounted using a compression-extension spring and a torsion spring. The four points of Q, C, P and T represent, respectively, the quarter-chord which is assumed to be the aerodynamic center, the center of mass, the reference point where the plunge displacement is measured, and the three-quarter-chord (Hodges and Pierce, 2002). The equations of motion of the airfoil section are given by

$$m\ddot{h} + mbx_{\theta}\ddot{\theta} + d_h\dot{h} + k_h h = -L \quad (2.35)$$

$$I_P\ddot{\theta} + mbx_{\theta}\ddot{h} + d_{\theta}\dot{\theta} + k_{\theta}\theta = M_{\frac{1}{4}} + b(\frac{1}{2} + a)L \quad (2.36)$$

where h and θ are the plunge displacement and pitch displacement, respectively; m is the mass per length in the span direction; b is the semichord length; I_P is the moment of inertia per length about the reference point; d_h and d_{θ} are the damping per length in the plunge and pitch degrees of freedom, respectively; k_h and k_{θ} are the

stiffness per length in the plunge and pitch degrees of freedom, respectively; a is the dimensionless parameter that is used to determine the location of the reference point from the leading edge; and x_θ is the dimensionless chordwise offset of the center of mass from the reference point, denoted by $x_\theta = e - a$, with e determining the location of the center of mass (Figure 2.11). L and $M_{1/4}$ are the aerodynamic lift per length and the aerodynamic pitching moment per length about the quarter-chord, respectively. With the steady flow theory, they are calculated by

$$L = \frac{1}{2} \rho_a (2b) U^2 C_L = 2\pi \rho_a b U^2 \theta \quad (2.37)$$

$$M_{1/4} = 0 \quad (2.38)$$

where C_L is the effective lift coefficient, which is taken to be $C_L = 2\pi\theta$ when steady flow theory for a thin airfoil is employed. The angle of attack is simply taken as the instantaneous pitch angle θ . The lift-curve slope is taken to be 2π .

• Unsteady flow theory

The steady flow theory comes in a quite simple form. However, it has unacceptable deficiency in predicting the flutter boundary and modal frequencies at the boundary. Unsteady aerodynamic effects are important due to at least the following three facts, as summarized by Hodges and Pierce (2002). First, the direction of relative wind vector is not fixed in space due to the airfoil's oscillatory motion, changing the effective angle of attack; second, vortex shedding at the trailing edge is induced by the airfoil's oscillation, the downwash from which also changes the effective angle of attack; third, the apparent mass and inertia effects should be considered because the air particles surrounding the airfoil are accelerated by its oscillatory motion. In an unsteady flow model, both circulatory and noncirculatory terms should be included. In the literature, the most commonly studied unsteady flow theories include the Theodorsen's unsteady thin-airfoil theory (Theodorsen, 1934) and the finite-state unsteady thin-airfoil theory of Peters et al. (1995).

a) Theodorsen's unsteady thin-airfoil theory

The unsteady flow theory derived by Theodorsen (1934) assumes that the airfoil undergoes small amplitude harmonic oscillations in incompressible flow. This

theory has been frequently employed in predicting the power output responses of airfoil flutter based energy harvesters at the flutter boundary (Erturk et al., 2010; Sousa et al., 2011; De Marqui and Erturk, 2012). The aerodynamic lift includes both circulatory and noncirculatory terms, while the aerodynamic pitching moment about the quarter-chord includes noncirculatory term only. The expressions are given by

$$L = 2\pi\rho_a bUC(k) \left[\dot{h} + U\theta + b\left(\frac{1}{2} - a\right)\dot{\theta} \right] + \pi\rho_a b^2 (\ddot{h} + U\dot{\theta} - ba\ddot{\theta}) \quad (2.39)$$

$$M_{\frac{1}{4}} = -\pi\rho_a b^3 \left[\frac{1}{2}\ddot{h} + U\dot{\theta} + b\left(\frac{1}{8} - \frac{a}{2}\right)\ddot{\theta} \right] \quad (2.40)$$

The first term in Equation (2.39) accounts for the circulatory effect, while the second term in Equation (2.39) and the term in Equation (2.40) account for the noncirculatory effects. The circulatory lift is the most significant among all these terms. Comparing Equation (2.39) with (2.37), it can be inferred that an effective angle of attack is introduced, given by

$$\alpha = C(k) \left[\theta + \frac{\dot{h}}{U} + \frac{b}{U} \left(\frac{1}{2} - a \right) \dot{\theta} \right] \quad (2.41)$$

This effective angle of attack is calculated at the three-quarter chord, point T in Figure 2.11, and takes in account the induced flow over the chord. The lift-deficiency function $C(k)$ is a complex-valued function of the reduced frequency $k=b\omega/U$ with ω being the frequency of motion. The involvement of $C(k)$ decreases the magnitude of the unsteady lift when compared to the value obtained directly from the steady flow theory, and introduces a phase difference between the plunge and pitch motions.

It is worth noting that the quasi-steady linear flow model, which is obtained by taking $C(k)$ to be unity in Equation (2.39), is a simplified and commonly used model to approximately predict the time dependent responses of the airfoil at the flutter boundary. The airfoil is assumed to have the same aerodynamic characteristics as the one moving with constant pitching and plunging velocity equal to the instantaneous values (Fung 1955). This is only acceptable for the situations where the characteristic timescale of flow is small compared to the characteristic timescale of oscillation as clarified in the galloping model.

b) Finite-state unsteady thin-airfoil theory of Peters et al.

Unlike the Theodorsen's theory, the finite-state theory of Peters et al. (1995) does not assume simple harmonic oscillations. As a time-domain model with state-space form representation, it is capable of calculating eigenvalues below the flutter speed and applicable in designs for active control of flutter. It has also been employed in the study of flutter energy harvesting at the flutter boundary (Bryant and Garcia, 2009, 2011). The accuracy and validity of this model have been confirmed by the wind tunnel test of their harvester prototype. The circulatory effects of vortex shedding are approximated by introducing an induced-flow term, i.e., the average induced-flow velocity λ_0 , which is further represented with a set of time-domain differential equations. The lift and pitching moment are expressed as

$$L = 2\pi\rho_a bU \left[\dot{h} + U\theta + b \left(\frac{1}{2} - a \right) \dot{\theta} - \lambda_0 \right] + \pi\rho_a b^2 \left(\ddot{h} + U\dot{\theta} - ba\ddot{\theta} \right) \quad (2.42)$$

$$M_{\frac{1}{4}} = -\pi\rho_a b^3 \left[\frac{1}{2} \ddot{h} + U\dot{\theta} + b \left(\frac{1}{8} - \frac{a}{2} \right) \ddot{\theta} \right] \quad (2.43)$$

It can be noted that the expression of the pitching moment is the same with that in the Theodorsen's theory. λ_0 is represented in terms of N induced-flow states λ_n as

$$\lambda_0 \approx \frac{1}{2} \sum_{n=1}^N b_n \lambda_n \quad (2.44)$$

where λ_n can be obtained from a set of N differential equations given by

$$[A]\{\dot{\lambda}\} + \frac{U}{b}\{\lambda\} = \{c\} \left[\ddot{h} + U\dot{\theta} + b \left(\frac{1}{2} - a \right) \ddot{\theta} \right] \quad (2.45)$$

The expressions of the related matrices are given as follows:

$$[A] = [D] + \{d\}\{e\}^T + \{c\}\{d\}^T + \frac{1}{2}\{c\}\{e\}^T \quad (2.46)$$

$$D_{nm} = \begin{cases} \frac{1}{2n} & n = m + 1 \\ -\frac{1}{2n} & n = m - 1 \\ 0 & n \neq m \pm 1 \end{cases} \quad (2.47)$$

$$e_n = \begin{cases} (-1)^{n-1} \frac{(N+n-1)!}{(N-n-1)! (n!)^2} \frac{1}{n} & n \neq N \\ (-1)^{n-1} & n = N \end{cases} \quad (2.48)$$

$$d_n = \begin{cases} \frac{1}{2} & n = 1 \\ 0 & n \neq 1 \end{cases} \quad (2.49)$$

$$c_n = \frac{2}{n} \quad (2.50)$$

(2) Nonlinear aerodynamics

Although the aforementioned sophisticated aerodynamic models like those of Theodorsen and Peters et al are adequate to predict aeroelastic and power output responses of flutter based harvesters near the flutter boundary, they are both based on linear assumptions, where the angle of attack is regarded to remain small, i.e., below the critical angle of attack to induce stall, and the attached flow over the airfoil is dominant. With this assumption, the amplitude of oscillation grows exponentially at a wind speed above the flutter speed. Theoretical modeling of flutter energy harvesters with the Theodorsen's or finite-state linear theory is merely able to predict the power response just above the flutter speed. To accurately study the aeroelastic as well as power output responses of a harvester undergoing limit cycle oscillations far away from the flutter boundary, nonlinearity has to be introduced into the model. Either structural nonlinearity (i.e., material nonlinearity and geometry nonlinearity) or aerodynamic nonlinearity can induce limit cycle oscillations above the flutter boundary. Structural nonlinearities can be due to large displacement of wings, loose linkages and worn hinges, and nonlinearities in stiffness properties of other components. Classical types of stiffness nonlinearities like the cubic nonlinearity, hysteresis nonlinearity and freeplay nonlinearity have drawn wide attentions in the research of aeroelasticity in aircraft (Woolston et al., 1957; Zhao and Yang, 1990; Dugundji, 1992; Lee et al., 1999; Dowell et al., 2003; Xiang et al., 2014). Inspired from this, in flutter based energy harvesting, some researchers considered the stiffness nonlinearity in the analysis of power responses

(Sousa et al., 2011; Abdelkefi et al., 2012a, 2012b, 2012c). On the other hand, the aerodynamic nonlinearities arise from the stalling phenomenon, where the separated airflow over the wing becomes dominant, and the aerodynamic lift force begins to decrease (Fung, 1955; Balakrishnan, 2012; Dowell, 2015). The critical angle of attack where flutter stall occurs is typically in the range of 8 to 20 degrees. The flap rotation of the airfoil for a flutter based energy harvester is most likely to surpass this critical value (Bryant and Garcia, 2011), making the nonlinear aerodynamic modeling necessary for accurate power output prediction. Next, we introduce typical nonlinear aerodynamic modeling methods in the literature.

- **Quasi steady model based on the effective angle of attack**

The quasi-steady nonlinear aerodynamic model is based on the concept of effective angle of attack (Fung 1955; Strganac et al., 2000) in the aforementioned Theodorsen's linear model. Extra nonlinear terms are simply introduced to it to approximate the aerodynamic lift at large angles of attack. As a common practice, noncirculatory terms are further ignored in this model. The expressions of the lift and moment are given by

$$L = 2\pi\rho_a b U^2 \left\{ \left[\theta + \frac{\dot{h}}{U} + \frac{b}{U} \left(\frac{1}{2} - a \right) \dot{\theta} \right] - c_s \left[\theta + \frac{\dot{h}}{U} + \frac{b}{U} \left(\frac{1}{2} - a \right) \dot{\theta} \right]^3 \right\} \quad (2.51)$$

$$M_{\frac{1}{4}} = 0 \quad (2.52)$$

where c_s is a nonlinear parameter related to the flutter stall and can be determined from the measured lift curve in the wind tunnel experiment. As in the case of linear aerodynamics, the quasi-steady model is only applicable for slow harmonic oscillations with low frequency in subsonic flow, e.g., for a harvester that is designed to own relatively low natural frequencies, say, smaller than 10Hz. This model has been employed in flutter based energy harvesting by Abdelkefi et al. (2012a, 2012b, 2012c) and Bibo and Daqaq (2013a, 2013b).

- **Semi-empirical unsteady nonlinear aerodynamic model based on ordinary differential equations**

One commonly exploited semi-empirical unsteady nonlinear aerodynamic model is

the ONERA model, which was initially developed by Tran and Petot (1981) and Dat and Tran (1983). Modification was made later by Peters (1985) on the effective angle of attack. This model has been applied to extensive studies on the nonlinear characteristics of airfoil flutter (McAlister et al., 1984; Dunn Dugundji, 1992; Chen, 1993; Tang and Dowell, 1996). In this model, the aerodynamic lift on an airfoil subjected to flutter stall is associated with the angle of attack in the form of an ordinary differential equation. Coefficients are determined from the experimental data. The static force curve of the airfoil is employed, with a single lag term introduced to the linear portion where the Theodorsen's linear theory is used, and two lag terms introduced to the stall portion (Chen, 1993).

The ONERA model was first introduced into the study of flutter based energy harvesters by Bryant and Garcia (2011), with very well agreed predictions with experiments achieved for the flutter response and power outputs. The model is described as follows.

$$C_z = C_{z1} + C_{z2} \quad (2.53)$$

$$C_{z1} = s_{z1}\dot{\alpha} + s_{z2}\ddot{\theta} + s_{z3}\dot{\theta} + C_{z\gamma} \quad (2.54)$$

$$\dot{C}_{z\gamma} + \lambda_1 C_{z\gamma} = \lambda_1 a_{0L} (\alpha + \dot{\theta}) + \lambda_2 a_{0L} (\dot{\alpha} + \ddot{\theta}) \quad (2.55)$$

$$\ddot{C}_{z2} + r_1 \dot{C}_{z2} + r_2 C_{z2} = -r_2 \Delta C_z|_{\alpha} - r_3 \Delta \dot{C}_z|_{\alpha} \quad (2.56)$$

Note that the dot is the derivative with respect to the dimensionless time $\tau = Ut/b$. In the above equations, C_z stands for any relative aerodynamic force coefficient, i.e., C_L for lift, C_D for drag or C_M for moment. C_{z1} represents the contribution of the linear force, which further includes the circulatory terms ($C_{z\gamma}$) and the noncirculatory terms; while C_{z2} represents the contribution of the nonlinear force, which has to be considered when the static force curve deviates from the extension of the linear portion, as shown in Figure 2.19. In general, a_{0L} is taken as 2π as the lift-curve slope in the linear portion. The parameters s_{z1} , s_{z2} , s_{z3} , λ_1 , λ_2 , r_1 , r_2 , and r_3 are empirically derived by fitting to results of wind tunnel experiment, of which the identification process has been discussed a lot in the literature (McAlister et al., 1984; Dunn Dugundji, 1992; Chen, 1993).

Figure 2.19 Concept of ONERA unsteady nonlinear aerodynamic model

Besides the ONERA model, there are other semi-empirical models that have yet been but are potential to be employed for power output predictions of flutter based energy harvesters, e.g., the one proposed by Mahajan et al. (1993) which assumes that the aerodynamic forces behave like a damped harmonic oscillator. Recently, a low-order quasi-steady model based on rotational lift and a revised version incorporating dynamic stall was proposed by Gomez et al. (2014). This model was experimentally validated with the scale and shape of the force curves similar to the experimental data, providing an alternate method for modal convergence flutter based energy harvesting.

2.2.3.4 Aerodynamic model for wake galloping

As mentioned in Section 2.2.1.4, due to the complexity of upstream wakes, the characteristics of wake galloping (Tokoro et al., 2000) and the performance of wake galloping based energy harvesters (Jung and Lee, 2011; Hobbs and Hu, 2012; Abdelkefi et al., 2013d) were basically evaluated via experiments. The measured results of Jung and Lee (2011) showed that obvious displacement occurred for configurations with $L=3D$ to $L=6D$, which can serve as a rough guidance for proper arrangement of the twin cylinders. Tokoro et al. (2000) conducted an experimental study on the wake galloping of twin cables and investigated the influence of various parameters on the aerodynamic characteristics including the spacing, Reynolds number, incidence angle of the wind, damping and natural frequency of the system. It was found that the maximum amplitude of vibration occurs at $L/D=4.3$ with an incidence angle of the wind of 15° . Also, the vibration direction is not exactly

normal to the wind flow, but inclined to it with a small angle. For detailed results readers are referred to the work of Tokoro et al. (2000).

As for wake-induced flutter, theoretical models do exist in various forms, including the linear model that predicts the instability boundary as well as the nonlinear model that calculates the amplitude of limit cycle oscillation (Païdoussis et al., 2011). There is an interesting finding that an increase in structural damping does not necessarily reduce the amplitude of oscillation of the leeward cylinder, whereas it increases the extraction of the flow energy. Future designs of energy harvesters based on wake-induced flutter should take this point into account. For a detailed analysis of the characteristics of wake-induced flutter, readers are referred to the review work of Païdoussis et al (2011).

2.2.3.5 Aerodynamic model for turbulence-induced vibration

Theoretical modeling of turbulence-induced force is not as straightforward as that of the aforementioned types of aerodynamic forces, due to the random noises in the turbulent flow. Mean velocity and turbulence density are not sufficient to fully represent the characteristics of the flow, not to mention the characteristics of the induced force. In order to conduct a theoretical analysis of their previous prototypes of piezoelectric grass-typed harvester (Hobeck and Inman, 2012a) mentioned in Section 2.2.1.5, Hobeck and Inman (2014) proposed a statistical model to simulate the distributed turbulence-induced force along a unimorph cantilever. This model requires time-series dynamic pressure measurements from the experiment, which are accomplished using dual pressure probes. Details of the design and analysis of the pressure probes were included in another work of Hobeck and Inman (2012b). Being verified with wind tunnel experiments, the displacement and power responses of their prototype were successfully predicted with this statistically derived aerodynamic model. Here we introduce the basic procedure of this model. For more detailed derivation process, readers are referred to the paper of Hobeck and Inman (2014).

Figure 2.20 Schematic of cantilevered unimorph harvester for statistical model of turbulence-induced force (Hobeck and Inman, 2014).

Figure 2.20 shows the schematic of cantilevered unimorph harvester for the derivation of statistical model. The model is based on the acceptance integral approach proposed by Powell (1958). The acceptance integral is given by

$$J_{mn}(\omega) = \int_0^{L_s} \int_0^{L_s} \phi_m(z) S_p(z, z', \omega) \phi_n(z') dz dz' \quad (2.57)$$

where J is the acceptance; L_s is the length of the cantilever; ϕ is the mode shape; z and z' denote the vertical locations of the measured points along the cantilever; ω is the frequency of oscillation; S_p is the pressure cross-power spectral density (CPSD) which is obtained from experimental measurements using the pressure probes, with its expression given by

$$S_p(z, z', \omega) = \lim_{T \rightarrow \infty} \frac{1}{4\pi T} \int_{-\infty}^{\infty} \left\{ \int_{-T}^T E[p(z, t) p(z', t + \tau)] dt \right\} e^{-j\omega\tau} d\tau \quad (2.58)$$

where p is the measured pressure; E is an expectation of the two pressure signals that must be taken; and T and τ are the sample time and time offset, respectively. With the experimental measurements, the statistically obtained modal distributed turbulence-induced forcing term is given by

$$\bar{\psi}_m(t) = \left(H_m^{-1} \sqrt{2A \int_{-\omega_s}^{\omega_s} |H_m|^2 J_{mm} d\omega} \right) e^{j\omega t} \quad (2.59)$$

where H_m is the modal displacement frequency response function expressed as

$$H_m = \frac{1}{(m_m + \mu_m)(\omega_m^2 - \omega^2 + 2j\zeta_m\omega_m\omega)} \quad (2.60)$$

with m_m , μ_m and ζ_m being the modal beam mass, fluid added mass and modal damping, respectively. Because measured data from experiments are essential, the accuracy of this model significantly depends on the electrical noise and bandwidth limitations of the pressure probes (Hobeck and Inman, 2012b).

2.2.4 Equivalent circuit modeling

Using the mathematical models incorporating the electromechanical model and the aerodynamic model, the electromechanical and fluid-structure coupling behaviors can be readily analyzed when the interface circuit is kept simple, i.e., a pure resistor for power dissipation. However, for practical applications in WSN nodes and other electronics, the interface circuit connected to an energy harvester will be more complex for the process of AC-DC signal rectification and regulation, or for further storage of energy. In such a case, theoretical formulations for the harvesting process become much complicated due to the added nonlinear electronic components in the circuit. To solve this problem, researchers have proposed equivalent circuit models for vibration piezoelectric and electromagnetic energy harvesters (Elvin and Elvin, 2009a, 2009b; Yang and Tang, 2009).

The equivalent circuit model is established based on the analogies between the relationships of parameters in the mechanical and electrical domains. For example, $F = M\ddot{x}$, $F = C\dot{x}$ and $F = Kx$ in the mechanical domain is analogous in the form to $V = L\ddot{q}$, $V = R\dot{q}$ and $V = (1/C_0)q$, respectively, with F , M , C , K and x representing the mechanical force, mass, damping and stiffness, respectively, and V , L , R , C_0 and q representing the voltage, inductance, load resistance, capacitance and electrical charge, respectively. The earliest uncoupled equivalent circuit model simulated the piezoelectric energy harvester with an ideal current source in parallel with its internal capacitance, or with an ideal voltage source in series with its internal capacitance. Elvin and Elvin (2009a) proposed an equivalent circuit model for vibration energy harvester in which the electromechanical coupling was taken

into account. The derivation process was based on the Rayleigh-Ritz model with a series of assumed displacement modes. Later, a coupled finite element–circuit simulation model was proposed by Elvin and Elvin (2009b), which is capable of analyzing complicated mechanical structures and electrical circuits utilizing the powerful tools of finite element analysis (FEA) like ANSYS or ABAQUS and electrical simulators like SPICE. The two parts of simulations were conducted separately, and post-processing of data extraction and transfer was required for the coupling between the two parts. Yang and Tang (2009) proposed an equivalent circuit model where the system parameters can be efficiently obtained from theoretical analysis or FEA. Example studies of energy harvester with both simple and complicated geometries were conducted. The schematic of the multi-mode equivalent circuit model for a vibration energy harvester is shown in Figure 2.21 (Yang and Tang, 2009).

Figure 2.21 Schematic of multi-mode equivalent circuit model for a vibration energy harvester (Yang and Tang, 2009)

As for aeroelastic energy harvesters, a new problem arises since the external aerodynamic forces are displacement-dependent nonlinear forces, unlike in vibration energy harvesting, where the base excitation force is independent of the system displacement and can be easily simulated with a separate voltage source component in the equivalent circuit model (see V_1 , $V_2 \dots V_r$ in Figure 2.21). To solve this problem, Tang et al. (2015) proposed an equivalent circuit representation method for galloping-based piezoelectric energy harvesters, representing the nonlinear aerodynamic force with a user-defined electronic component with a

nonlinear transfer function, as shown in Figure 2.22. The meanings of the symbols in the equivalent circuit model are given in Table 2.1. The aerodynamic force was represented with a voltage source given by

$$V(t) = -\frac{1}{2}\rho_a h l U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\dot{q}(t)}{U} + \beta C V_c(t) \right]^i \quad (2.61)$$

where the term in the bracket indicated the modified angle of attack taking into account the beam rotation at the free end, i.e., modifying Equation (2.7) into $\alpha = \frac{\dot{w}}{U} + \beta w$ with β being an angle coefficient. The voltage V_c across the capacitor C was employed to calculate the charge q with $q = C V_c$. The proposed model was validated with wind tunnel experimental results.

Figure 2.22 Schematic of equivalent circuit representation for a galloping-based piezoelectric energy harvester (Tang et al., 2015)

Table 2.1 Analogy between mechanical and electrical domain (Tang et al., 2015)

Equivalent circuit parameters	Mechanical counterparts
Charge: $q(t)$	Displacement: $w(L_b, t)$
Current: $\dot{q}(t)$	Velocity: $\dot{w}(L_b, t)$
Inductance: L	Effective mass: M
Resistance: R	Effective damping: D
Capacitance: C	Reciprocal of effective stiffness: $1/K$
Ideal transformer turn ratio: N	Electromechanical coupling: θ

Elvin (2014) proposed two approaches, i.e., a system-level approach and a dependent source equivalent approach, to model the behaviors of advanced energy harvesters with nonlinear component, e.g., a vibration energy harvester with nonlinear stiffness (Duffing harvester), or aeroelastic energy harvester with nonlinear aerodynamic force. For a 2DOF flutter-based energy harvester, of which the aerodynamic mechanism has been introduced in Sections 2.2.1.3 and 2.2.3.3, the

equivalent circuit model is shown in Figure 2.23. The governing equation of the harvester was expressed similar to Equations (2.35) and (2.36), with

$$\begin{aligned} m\ddot{x} + D\dot{x} + Kx - \Theta V + (me\ddot{\alpha} - H_1\dot{x} - H_3\alpha) &= F(t) \\ I_\alpha\ddot{\alpha} + D_\alpha\dot{\alpha} + K_\alpha\alpha + (m\ddot{x} - A_1\dot{x} - A_3\alpha) &= M(t) \end{aligned} \quad (2.62)$$

where x , α , V , m , I_α , D , D_α , K , K_α , e and Θ were the plunge displacement, pitch displacement, generated voltage in the piezoelectric element, airfoil mass, airfoil moment of inertia, plunge damping, pitch damping, plunge stiffness, pitch stiffness, offset of the center of mass from the reference point and electromechanical coupling, respectively; $F(t)$ and $M(t)$ were indicated as the aerodynamic lift and moment, of which the formulations were not specified; and H_1 , H_3 , A_1 and A_3 were the aerodynamic force coefficients. With such a governing equation, the equivalent circuit model was established by adding the two nonlinear terms in the parentheses at the left side as two voltage-dependent sources NV_1 and NV_2 . During circuit simulation, they were defined with the voltage across the standard circuit components by

$$\begin{aligned} NV_1 &= \frac{V(I_\alpha)}{I_\alpha} me - H_1 \frac{V(D)}{D} - H_3 \frac{V(\frac{1}{K_\alpha})}{K_\alpha} \\ NV_2 &= \frac{V(m)}{m} me - A_1 \frac{V(D)}{D} - A_3 \frac{V(\frac{1}{K_\alpha})}{K_\alpha} \end{aligned} \quad (2.63)$$

Although the validation process of the proposed equivalent circuit model for the flutter harvester was not provided, a similar equivalent circuit model for a nonlinear electromagnetic Duffing harvester was validated theoretically, verifying the proposed voltage-dependent source equivalent approach.

Figure 2.23 Schematic of equivalent circuit model for a 2DOF flutter-based piezoelectric energy harvester (Elvin, 2014)

2.2.5 Modeling based on computation fluid dynamics (CFD)

Besides the mathematical and equivalent circuit modeling, some researchers have employed computational fluid dynamics (CFD) to simulate the complex interactions between the fluid flow and the solid structure of the energy harvester, utilizing the commercial CFD simulation software like the COMSOL Multiphysics, ANSYS-CFX, ANSYS-FLUENT, etc. It is beyond the scope of this chapter to present and compare the detailed coding and meshing process of various CFD modeling methods, interested readers are referred to the respective software manuals and related technical literatures (COMSOL CFD Module, ANSYS CFX, ANSYS Fluent, Computational fluid dynamics, Wikipedia). Here we introduce the recent flow energy harvesting studies with applications of CFD.

Sivadas and Wickenheiser (2011) conducted a parametric study on a VIV based piezoelectric energy harvester, which consisted of an upstream fixed bluff body and a piezoelectric cantilever attached to its trailing edge. CFD simulations with the COMSOL Multiphysics software were run for different dimensions and shapes of the bluff body, length and thickness of the beam and Reynolds number to investigate their effects on the lock-in bandwidth and output power. It was found that a medium long beam with a length-to-diameter ratio of 2 to 2.5 could induce the maximum strain on the beam because it enabled the vortex streets to form at the right position to excite the beam's fundamental mode. Among the three considered

bluff body shapes, i.e., cylinder, triangle and pentagon, it was shown that the triangular bluff body had a narrow lock-in bandwidth at a low Reynolds number region, while the cylindrical and pentagonal bluff bodies had wider lock-in bandwidths at higher Reynolds number region. Vorticity field for different shapes with a medium beam is shown in Figure 2.24. The cylindrical bluff body was found to generate the highest average power and determined to be the optimal bluff body for VIV harvester. For a cylindrical-bluff-body harvester with a beam length of 0.04m and a diameter of 0.02m at Re range of 300 to 1100, a maximum power of 0.35mW was numerically predicted.

Figure 2.24 Vorticity field plot for a VIV based piezoelectric energy harvester with different bluff body shapes (Sivadas and Wickenheiser, 2011).

Pobering and Schwesinger (2008) proposed a VIV energy harvester and investigated its behavior with CFD simulations. The development of von Kármán's vortex streets behind three cantilevers that were linearly arranged in a row was shown in Figure 2.25. It was found that the upstream vortices from the prior cantilevers combine with and amplify the following ones, increasing the strain thus power generation capability of the downstream harvester. It can be inferred from this finding that by properly arranging a series of harvesters, the performance of VIV harvesters can be enhanced. Pobering et al. (2009) also conducted CFD simulations to investigate the effect of bluff body shape on the performance of energy harvesters based on VIV. Cylindrical, triangular and hexagonal shapes were considered. Unlike the conclusion of Sivadas and Wickenheiser (2011), it was pointed out that the shape with a very sharp tear-off-edge like the triangular shape gave the best results in terms of the periodicity of vortices and value of the low pressure in the downstream air. The results are shown in Figure 2.26. The COMSOL Multiphysics software was employed for both studies.

Figure 2.25 CFD simulation results of three cantilevers arranged in a row in mean flow direction (Pobering and Schwesinger, 2008)

Figure 2.26 CFD simulation results of pressure development of different bluff bodies (Pobering et al., 2009)

Hobeck et al. (2014) reported the phenomenon of dual cantilever flutter (DCF) during wind tunnel experiments, where two identical cantilevers underwent large amplitude and persistent vibrations when subject to wind flows. They proposed the first documented energy harvesting device based on this DCF phenomenon which consisted of two identical piezoelectric cantilevers, as shown in Figure 2.27(a). It was inferred that dynamics of one beam affected dynamics of the other through fluid coupling. CFD simulations were conducted using shear stress transport (SST) turbulence model with the ANSYS-CFX software for two types of dynamics, i.e., the entrainment dynamics and the flutter dynamics. The former means that when there is no flow, disturbance-induced vibration of one beam will cause the other beam to start to oscillate; while the latter means the constant and identical amplitude but out of phase oscillations of the two beams during DCF. The simulation results of the two types of dynamics are shown in Figure 2.27(b). The meshing zone layouts are shown in Figure 2.28, with the left and right graphs indicating the case without and with relative deflections, respectively. A non-conformal interface was used to separate the two beams into completely independent halves of fluid. The inlet and

outlet in the left graph was changed to open like that in the right graph when modeling the entrainment dynamics. The CFD simulation results for entrainment dynamics successfully captured the experimental measurements, while the results for flutter responses would be validated by future experiments.

Figure 2.27 DCF based piezoelectric energy harvester (a) schematic of harvester device (b) CFD simulation results for entrainment dynamics and flutter responses (Hobeck et al., 2014)

Figure 2.28 Computational domain and meshing zones in CFD model for DCF harvester (Hobeck et al., 2014)

Akaydin et al. (2010) employed FLUENT to conduct CFD simulations for their cantilevered harvester that can harvest energy from highly unsteady wind flows, as shown in Figure 2.29(a). Vortex shedding of an upstream cylinder was chosen to generate the desired unsteady turbulent flow, as shown in Figure 2.29(b). The shear stress transport (SST) $k-\omega$ turbulence model was employed during simulation. The meshing details are shown in the lower graph in Figure 2.29(b). It can be seen that triangular cells were used in the vicinity of the beam in order to facilitate simple dynamic re-meshing at each time step; while in other areas including the cylinder area quadrilateral cells were used. High mesh density was used in the boundary

layers in order to accurately model the viscous effects like vorticity generation. The cantilever with PVDF attached as the top layer underwent oscillations with relatively large amplitude of tip displacement and high operational frequency. Therefore, the shape of the piezoelectric generator was calculated at each predefined time step Δt and the boundary conditions thus the meshes were updated at each step, as shown in Figure 2.29(b).

Figure 2.29 (a) Diagram of the energy harvester in highly unsteady wind flow (b) procedure and meshing details for CFD simulation (Akaydin et al., 2010)

CFD simulations have also been conducted by Mehmood et al. (2013) for their VIV energy harvester and by Kishore et al. (2013) for their small-scale wind energy portable turbine. CFD simulation owns its advantage in giving clear visual observations of flow pressure distribution and structural deflection during the operation of an energy harvester, but it requires significant efforts in developing and

updating meshes and tremendously high simulation time. To shorten the simulation time, in the above mentioned studies the CFD simulations were all conducted with 2D dimensions. Moreover, all the above mentioned CFD simulations in energy harvesting studies solved for the responses in the fluid and structure domains, however, components in the electrical domain, like the external load or more complex interface circuits, were not considered. However, the backward electromechanical coupling effect was either considered separately in the later data process or simplified into damping effect in the simulation. The fluid and mechanical responses were able to be directly obtained through CFD, which were further employed as an input to derive the corresponding electrical response via circuit simulation or analytical calculation of the circuit governing equation.

2.3 Designs of aeroelastic piezoelectric energy harvesters

Many designs of small-scale wind energy harvesters have been reported in the literature, including those in the form of small-scale windmills and turbines, and those based on the aeroelastic instabilities discussed above, like VIV, galloping, flutter, wake-induced oscillation and TIV. In this section, the recent small-scale wind energy harvester designs and their performances will be reviewed and summarized.

2.3.1 Small-scale windmill and wind turbine

Rancourt et al. (2007) investigated the performance of power generation of a centimeter-scale windmill. Power was generated using electromagnetic transduction mechanism. Three prototypes of propellers were tested in the wind tunnel, which were all 4.2cm in diameter with four blades of different pitch angles, as shown in Figure 2.30. The experimental results showed that the “Schmitz theory”, which was developed for large scale wind turbine to determine the optimal tip speed ratio for maximum turbine efficiency (i.e., kinetic power extracted from the wind over the available kinetic flow energy for the area covered by the disk of the propeller), was also valid for small scale wind turbines. However, the power generation efficiency (electrical power output over the available kinetic flow energy for the area covered

by the disk of the propeller) at low wind speed decreased sharply due to the friction in the generator and the internal electric resistance. At a high wind speed of 11.8m/s a large power of 130mW was achieved, corresponding to a power generation efficiency of 9.5%; while a lower power of 2.4mW was obtained at 5.5m/s with a decreased efficiency of 1.85%.

Figure 2.30 Three propellers tested with different pitch angles (Rancourt et al., 2007)

Howey et al. (2011) tested a miniature electromagnetic wind turbine with a diameter of 3.2cm as shown in Figure 2.31(a), which was claimed to be the smallest turbine-based energy harvester reported to date. The turbine consisted of two rotating magnetic rings mounted on the rim of the rotor, and fixed stator coil sandwiched between the magnetic rings. One of the magnetic rings assembled with six-blade rotor is shown in Figure 2.31(b), and details of the configuration of the magnetic ring and the coil are shown in Figure 2.31(c). Wind tunnel experiment found that the cut-in wind speed was 3m/s, below which the turbine could not operate. The test was run up to 10m/s, and a power of 80 μ W to 4.3 mW was achieved. Compared to the device of Rancourt et al. (2007) at a low wind speed of 5.5m/s, similar power generation efficiency was obtained, i.e., 1.35% for 5m/s and 1.52% for 6m/s. A simplified analytical model based on blade element momentum was used, with which the predictions of turbine efficiency (extracted kinetic power / available flow power) agreed with the measured results. It was noted that at wind speeds lower than 7m/s, the generated power was limited by bearing loss while at wind speeds higher than 7m/s, power output was limited by resistive generator loss. Future designs of miniature turbines aimed to harvest energy from low speed flows should pay attention to these two issues.

Figure 2.31 Miniature electromagnetic wind turbine (a) fabricated prototype (b) one magnetic ring fixed on the rim of a six-blade rotor (c) schematic of configuration of fixed coil sandwiched between two rotating rings (Howey et al., 2011)

Both studies of Rancourt et al. (2007) and Howey et al. (2011) show that the major challenge of miniature electromagnetic windmill lies in the greatly decreased power generation efficiency in slow flows. Of course, if a more sophisticated small scale wind turbine can be established incorporating optimized shape of airfoil and proper design of diffuser, the output power can be significantly increased. It was reported by Kishore et al. (2013) that a properly designed small-scale wind energy portable turbine (SWEPT) with a diameter of 39.4cm can generate a power up to 830mW at a wind speed of 5 m/s. Yet this size of the turbine is much larger than those of the small scale harvesters mentioned above, which are mainly smaller than or in the order of 10cm.

Recently small-scale windmills using piezoelectric transduction have shown great potential in efficiently harvesting low speed flow energy. The rotation of the windmill shaftt under wind flows is transferred to oscillatory motion of the piezoelectric transducer. The mechanical transfer is sometimes achieved by direct impact between the piezoelectric cantilever and the cam or blade, with a working principle similar to that of a mechanical stopper (Gu and Livermore, 2011); some other times it is achieved through magnetic interfaction where no contact impact is

required.

Priya et al. (2005) proposed a piezoelectric windmill to harvest energy from low speed wind flows. As shown in Figure 2.32, twelve piezoelectric bimorphs were arranged in a circular array around the circumference of the center shaft of the windmill. Twelve rubber stoppers were connected to the shaft, each of which was in contact with one of the bimorphs. The shaft connected via a cam to a rotating fan was rotated via the camshaft mechanism. When the shaft rotated, the stoppers caused the back and forth movements of the bimorph transducers, generating electrical energy via direct piezoelectric effect. The measured voltage waveform across a $4.6\text{k}\Omega$ load at an oscillatory frequency of 4.2Hz is shown in Figure 2.32(b). Experimentally, a standard circuit was employed and a power of 10.2mW after rectification was obtained at 6Hz and $4.6\text{k}\Omega$. It was found that power was increased with the prestress level and the number of bimorphs, which yet also resulted in increased difficulty in the fan rotation thus caused an increased cut-in wind speed.

Figure 2.32 (a) Schematic of a piezoelectric windmill and (b) waveform of generated voltage across a $4.6\text{k}\Omega$ load at frequency 4.2Hz (Priya et al., 2005)

In a subsequent work, Priya (2005) presented a theoretical model based on bending beam theory of bimorphs and equivalent circuit of capacitor to predict the output power of the above mentioned piezoelectric windmill. Ten bimorphs were used in the experiment. A cut-in wind speed of 4.7mph and a cut-out wind speed of 12mph (above which damage of structure will occur) were measured. A maximum power of 7.5mW was obtained after rectification at 10mph across a load of $6.7\text{k}\Omega$. The measured variation of power with wind speed is shown in Figure 2.33, showing a

linearly increasing part and a saturated part. A linear relationship was given between the saturated frequency (final constant operating frequency at a constant wind speed) of the windmill and the wind speed as $f(\text{Hz}) = -0.93 + 1.29U$, which well captured the experimental observation that the frequency linearly increased with the wind speed. Also, it was found that the generated power almost linearly increased with the frequency. These two observations corresponded to the linearly increasing power versus wind speed in Figure 2.33. It was suggested that the piezoelectric windmill could be a feasible power supply for wireless sensors.

Figure 2.33 Variation of power with wind speed for piezoelectric windmill (Priya, 2005)

Chen et al. (2006) investigated the performance of wind energy harvester with similar working principle to that of the above piezoelectric windmills, but with a rectangular arrangement of piezoelectric transducers, as shown in Figure 2.34. Twelve bimorph transducers were arranged in six rows and two columns, with a gap of 6 mm between each other. A cylindrical rod in between the two columns was connected via a camshaft mechanism to a rotating fan, which caused the up and down movement of the rod. Subsequently, the six hooks on the rod induced back and forth oscillations of the transducers from which electrical energy could be generated. The variation of power with wind speed from experiment was similar to that of Priya (2005). With a load of $1.7\text{k}\Omega$, a cut-in wind speed of 4.7 mph and a cut-out wind speed of 14 mph were measured, with a maximum power of 1.2 mW obtained at 12 mph. Compared to the windmill, this prototype is easy to fabricate and is space efficient with a rectangular-array arrangement of transducers; also, since all the bimorphs are vibrating in phase, combined circuit can be used

eliminating the trouble of using individual processing circuit required in the circular windmill, as summarized by Myers et al. (2007). Yet the power was much lower compared to the circular windmill. To solve this issue, Myers et al. (2007) proposed an optimized rectangular piezoelectric windmill to enhance power output by employing three fan blades to enlarge the covered flow surface and to increase the captured wind energy. The schematic of the windmill and its internal crank-shaft structure are shown in Figure 2.35. The number of piezoelectric transducers was also increased, with two rows containing nine transducers in each row. Moreover, higher rotational inertia was introduced to prevent the windmill from cogging. It was measured that with a small sized prototype of $3 \times 4 \times 5$ inch ($7.62 \times 10.16 \times 12.70$ cm), an enhanced power of the order of 5mW was obtained at 10 mph. It should be noted that for all the above mentioned piezoelectric windmills, the same piezoelectric transducers were used, i.e., APC 855 with dimensions of a single piece of $60 \times 20 \times 0.6$ mm³.

Figure 2.34 Schematic of rectangular piezoelectric windmill (Chen et al., 2006)

Figure 2.35 Schematic of optimized design of rectangular piezoelectric windmill with three fan blades (Myers et al., 2007)

For the above mentioned impact-driven windmills, a challenging issue exists that the frequent impacts not only dissipate some kinetic energy, they also cause fatigue problems of the piezoelectric cantilevers and induce mechanical damages. In order to overcome this shortcoming, some researchers have proposed windmills that do not require direct impacts, but induce oscillations of the transducers through magnetic interactions.

Bressers et al. (2011) proposed a design of piezoelectric wind turbine where the piezoelectric elements were “contact-less” actuated through magnetic interaction. The schematic of the device is shown in Figure 2.36. A vertical axis wind turbine was connected to a disk, to which a series of alternating polarity magnets were attached. The magnets were also attached at the tips of the cantilevered transducers, which underwent harmonic oscillations through alternating attractive/repulsive magnetic force when the blades were rotating in wind flow. Measurement showed that with a two-blade and four-magnet rotor, the cut-in wind speed was lowered down to 2mph and a maximum power of around 1.2mW was obtained at 9mph.

Figure 2.36 Schematic of contact-less piezoelectric wind turbine with piezoelectric elements actuated via magnetic forcing (Bressers et al., 2011)

Karami et al. (2013) proposed a nonlinear piezoelectric wind turbine shown in Figure 2.37. A vertical axis turbine was placed on top of four vertical cantilevered piezoelectric transducers. Two arrangements of tangential configuration and radial configuration were considered. In both configurations, four tip magnets were embedded at the tips of the transducers, while five magnets were attached to the bottom of the rotating disk that was fixed to the blades. Different from the device of Bressers et al. (2011), nonlinearity was introduced to the transducers. The rotation of the blades induced the distance between the disk and tip magnets to continuously

alter, making the transducer alternately undergo bi-stable and mono-stable dynamics. Therefore, the transducer was both directly and parametrically excited. Experiments with no wind flow showed that the power increased with decreasing vertical gap between the disk and tip magnets. With a magnetic gap of 25mm and a load of 247 k Ω , wind tunnel experiments showed that the output power from the tangential configuration (only a single beam was mounted) monotonously increased with the wind speed. The cut-in wind speed was measured to be 2m/s, and a maximum power of 4mW at 10m/s was obtained. While for the radial configuration, nonlinear phenomenon was observed, and the output power was one order of magnitude less than that from the tangential design. It was explained that the direct excitation in the radial configuration was not as significant as that in the tangential configuration. An analytical model was established and quantitatively captured the nonlinear jump phenomenon. It was concluded that the nonlinear parametric excitation and ordinary excitation mechanisms can result in several advantages, including the low cut-in wind speed, high output power and large operational range of wind speed.

Figure 2.37 Schematic of nonlinear piezoelectric wind turbine with (a) tangential configuration and (b) radial configuration (Karami et al., 2013)

2.3.2 Energy harvesters based on vortex-induced vibrations

The concept of using VIV to harvest energy was first investigated in flowing water instead of wind. Allen and Smits (2001) proposed an “energy harvesting eel” to harvest flowing water energy as shown in Figure 2.38(a). The unit consisted of a fixed flat plate as a bluff body with a piezoelectric membrane placed in the wake. The membrane was set free in the downstream. Alternating vortices were shed

on either side of the bluff body, resulting in pressure differential thus forcing the membrane to oscillate with a movement similar to that of a natural eel swimming. Particle image velocimetry (PIV) experiment was conducted to investigate the vorticity pattern formed behind the bluff body, as shown in Figure 2.38(b). Four prototypes were tested in a re-circulating water channel with different types of material and dimensions of the membrane and various Reynolds number. It was found that lock-in phenomenon occurred to the membranes when they oscillated at the same frequency as the undisturbed wake behind the bluff body. The frequency responses of the membrane were found to be independent to the length of the membrane, but significantly sensitive to the bluff body size. Electrical response (i.e., voltage or power) was not measured in this study.

Figure 2.38 (a) Schematic of energy harvesting eel and (b) vorticity field formed surrounding membrane observed in particle image velocimetry experiment (Allen and Smits, 2001).

Taylor et al. (2001) proposed a similar “eel” harvester to harvest energy from flowing water like in the estuary or even in the open ocean. A prototype of 9.5” length, 3” width and 150 μ m thickness was fabricated with the commercial piezoelectric polymer PVDF, totally consisting of 8 segments. The prototype was tested in a flow tank and data on each segment were acquired separately. A peak voltage of around 2.5V was measured for the segment near the head bluff body at a flow speed of 0.5m/s. Measurements also showed that from head to tail, distortion in the voltage output increased while voltage magnitude decreased. The maximum power was achieved when the flapping frequency matched the vortex shedding frequency. It was pointed out that an optimum central layer and active layer thickness deserves further design efforts to obtain the optimum bending stiffness thus the maximum strain. The next step target was to design and deploy a

multi-element-eel system to produce 1W power in a nominal 1m/s water flow.

Robbins et al. (2006) proposed several methods to improve the power extraction efficiency of an energy harvester with flexible piezoelectric material, similar to Figure 2.38(a). It was shown experimentally that the efficiency could be enhanced by adding mass to the free end of the flapping piezoelement. Analytically, it was shown that the use of stronger-coupling flexible piezoelectric materials such as AFCs (active fiber composites) and MFCs (macro fiber composite) could improve the efficiency by a factor of 25. Moreover, both experiment and analysis showed that using the quasi-resonant rectifier to extract electrical energy from piezoelement instead of standard full-wave rectifier could increase the efficiency by a factor of 2.3.

Pobering and Schwesinger (2008) implemented a VIV energy harvester similar to the energy harvesting eel in both the air and water flow. It was roughly determined that available energy densities in flowing media are in the range of 256W/m^2 in air at a wind speed of 10m/s and 1600kW/m^2 in water at a flow speed of 2m/s, respectively. Behind the fixed bluff body, a piezoelectric bimorph cantilever was attached as shown in Figure 2.39 instead of flexible membrane or polymer, in such a way that the cantilever would only deform in the first vibration mode, unlike in the case of the membrane or polymer where undulating waves were generated along the length. The piezoelectric cantilever underwent periodic vibrations under the oscillating pressure differences induced by vortex shedding from the bluff body. A simple analytical model was established to give the design rules and predict power responses, from which the optimal ratio of the cantilever length L over the frontal dimension D of the windward bluff body was calculated to be $L/D=2.125$. Three identical prototypes were fabricated with a cantilever length of 14mm, width of 11.8mm, thickness of 0.35mm and bluff body frontal dimension of 10.35mm. They were arranged in a row and tested in a wind tunnel, with the results shown in Figure 2.40. The wind speed giving the peak power varied for the three prototypes depending on their positions. A highest deflection of $51\mu\text{m}$ was obtained at a wind speed of 40m/s on the second cantilever. A peak voltage of 0.8V with a load

resistance of $1\text{M}\Omega$ and a peak power of around 0.1mW with a matched load resistance of $1.2\text{k}\Omega$ were measured at 45m/s . Measurements in the water were not reported, but it was predicted that lower water flow speed would lead to comparable high power since the energy density in water was around 1000 times higher than that in air. It was concluded that adjacent cantilevers had strong influences on each other which enhanced the deflection and output power.

Figure 2.39 Schematic of VIV based energy harvester proposed by Pobering and Schwesinger (2008).

Figure 2.40 Variation of deflection with wind speed for three cantilevers (Pobering and Schwesinger, 2008)

In a subsequent study, Pobering et al. (2009) employed the same design as in Figure 2.39 and conducted more tests in both air and water. Three series of prototypes were fabricated. In each series, cantilevers were connected to the fixed bluff body and arranged in a row again. As an example, Figure 2.41 shows the prototypes and experimental configuration for Series 1. The dimensions of a single cantilever in each series were: Series 1, $L=22\text{mm}$, $W=11.8\text{mm}$, $D=10.35\text{mm}$, $T_{PZT}=0.088\text{mm}$; Series 2, $L=22\text{mm}$, $W=11.8\text{mm}$, $D=4.35\text{mm}$, $T_{PZT}=0.088\text{mm}$; and Series 3, $L=38\text{mm}$, $W=18\text{mm}$, $D=6\text{mm}$, $T_{PZT}=0.64\text{mm}$, where W was the cantilever width and T_{PZT} was the total thickness of the piezoelectric layers. Comparison of the wind tunnel experimental results between Series 1 and 2 confirmed the analytical prediction of the optimal geometrical relationship $L/D=2.125$, as shown in Figure 2.42. Subsequently, wind tunnel experiment for Series 3 showed that the electrical

output was enhanced with stapled piezoelectric layers (increased T_{PZT}). It was also found that the cantilevers in a row were able to synchronously oscillate in flowing media with high density like water. With a peak power of 0.055mW obtained from a single piezoelectric layer at 40m/s (Figure 2.42), a total power output of around 1mW can be approximated for Series 1 consisting 9 cantilevers. It was concluded that the power of the proposed harvesting system was sufficient to power sensors logical circuits and wireless data transmission circuits.

Figure 2.41 Prototype and wind tunnel experimental configuration in Series 1, in the right figure, wind flows from right to left (Pobering et al., 2009).

Figure 2.42 Measured voltage and power versus external load with cantilevers of Series 1 and 2 at a wind speed of 40m/s, with index “short” indicating $L/D=2.125$ and “long” indicating $L/D=5.057$ (Pobering et al., 2009).

However, among the above studies on VIV based energy harvesters, no one has investigated the effect of electromechanical coupling on the electrical output or its backward coupling effect on the aeroelastic response. Akaydin et al. (2010a) were among the very first to consider the three-way coupling effects, i.e., the mutual coupling behaviors between the aerodynamics, structural vibration and electrical response. In their study, a new type of energy harvester was proposed to harness flow energy based on the vortex shedding phenomenon as shown in Figure 2.43. A piezoelectric cantilever was put behind a windward cylinder which was fixed as a

bluff body. The downstream end of the cantilever was fixed. The cantilever oscillated in the fully turbulent vortex street formed at high Reynolds number of 14842. Although the harvester was claimed to be designed for harnessing energy from highly unsteady fluid flows, if we consider the windward cylinder as a part the energy harvester the flow in front of the cylinder was smooth and steady as they were placed together in a smooth-flow-generating wind tunnel. Therefore, we review this design in the section of “Energy harvesters based on vortex-induced vibrations”. In this study, performance of the piezoelectric cantilever inside a turbulent boundary layer was also reported, which we will introduce in the section of “Energy harvesters based on turbulent induced vibration”. Experimentally, with a $30\text{mm} \times 16\text{mm} \times 0.2\text{mm}$ cantilever with a piezoelectric layer of PVDF attached on the top surface, and with a cylinder of 30mm in diameter and 1.2m in length fixed in the windward direction, a peak power of $0.4\mu\text{W}$ was measured with a load of $100\text{k}\Omega$ at a wind speed of 7.23m/s , which produced a vortex shedding frequency close to the beam’s resonance frequency around 48.5Hz . Simulations based on CFD were conducted to solve the two dimensional N-S (Navier-Stokes) equations to obtain the aerodynamic pressure, which was substituted into a 1DOF electromechanical model to calculate the voltage output. The electromechanical coupling was assumed to be weak and the backward coupling effect of the voltage generation on the mechanical displacement response was ignored. An explanation of the driving mechanism of the beam’s oscillation was tried to be deduced from the simulation results. The induced flow ahead of the vortex impinges on the beam, and the overpressure resulted from the stagnation region bends the beam; at the same time on the opposite side, the core of another vortex applies suction, driving the beam in the same direction. The mechanism was further confirmed in a subsequent study with more simulations (Akaydin et al., 2010b). It was found experimentally that the face-on configuration where the beam was parallel to the upstream flow (as in Figure 2.43) was the best orientation for the beam. The optimal position of the cantilever was found to be $y=0$ and $x/D=2$ at the free end.

Figure 2.43 (a) Schematic of energy harvester of Akaydin et al. (2010b) and (b) flow visualization that shows vortex induced vibration of piezoelectric cantilever (Akaydin et al., 2010a).

In a subsequent study, Akaydin et al. (2012) proposed a new self-excited VIV energy harvester to improve the output power as shown in Figure 2.44. The cylinder was, different from the previous cases, attached to the free end of an aluminum cantilever with PZT covering the end area. Periodic oscillations occurred in the direction normal to the wind flow due to the vortex shedding. Measured power from wind tunnel experiment is shown in Figure 2.46. A peak power of around 0.1mW of non-rectified power was obtained at a much lower wind speed of 1.192m/s with a matched load of 2.46M Ω . The variation of power as a function of wind speed shown in Figure 2.46 is typical for a VIV energy harvester, with a peak power obtained at the lock-in region where the flow induced vortex shedding frequency matches the mechanical resonance frequency of the harvester. Analytically, voltage output was predicted using a distributed parameter model with experimentally measure strain data, and a non-rigid bonding model on strain transfer from the aluminum substrate to the piezoelectric patch predicted an accurate power (<10% error) than the rigid bonding assumption which was shown to cause significant overestimations (>60%). The aeroelastic interactions were found to play an important role evidenced by the facts that the Strouhal number was decreased from 0.21 for stationary cylinder to 0.105 for the oscillating cylinder and the lift coefficient was increased from less 0.03~0.07 to 0.44. This issue was suggested to be included in the future research. Compared to the previous design in Figure 2.43, the aeroelastic efficiency (i.e., efficiency of converting the flow energy into mechanical energy) was increased from 0.032% to 2.8% and the electromechanical efficiency (i.e., efficiency of converting the mechanical energy into electrical energy)

was increased from 11% to 26%. It was concluded that the modified configuration of the attachment of the cylinder on the cantilever tip and the employment of PZT instead of PVDF greatly enhanced the power generation.

Figure 2.44 (a) Top and side views of self-excited harvester and (b) harvester installed in wind tunnel (Akaydin *et al.*, 2012)

Figure 2.45 Typical variation of output power with wind speed obtained by Akaydin *et al.* (2012).

In order to overcome the narrow operating range of VIV-based harvesters, Weinstein *et al.* (2012) proposed an energy harvester with resonance tuning capabilities, as shown in Figure 2.46. The piezoelectric cantilever was placed behind a cylinder bluff body, and attached with an aerodynamic fin at the tip. Small weights were placed along the fin. Manually adjustment of the weights positions could tune the resonant frequencies of the harvester, making it able to operate at resonance for wind velocities from 2 to 5m/s, as shown in Figure 2.47. A peak power of nearly 5mW was obtained at a wind speed of around 5.5m/s. But the limitation is that this tuning mechanism is not automatic, thus the wind velocity needs to be always stable and the exact wind velocity should be known before the installation of this harvester.

Figure 2.46 (a) Prototype of harvester with manually tunable resonant frequencies and (b) schematic of harvester (Weinstein *et al.*, 2012)

Figure 2.47 Maximum attainable power versus wind speed, tuned to resonance at each point of operation (Weinstein *et al.*, 2012)

Different from the aforementioned studies which investigated the performance of the fabricated VIV energy harvester prototypes based on experiments or numerical simulations, Barrero-Gil *et al.* (2011) attempted to theoretically evaluate the effects of the governing parameters on the power extraction efficiency. The effects of the mass ratio m^* (i.e., the ratio of the mean density of a cylinder bluff body to the density of the surrounding fluid), the mechanical damping ζ and the Reynolds number were investigated with a 1DOF model where fluid forces were introduced from previously published experimental data from forced vibration tests. There was

no specific energy harvester design proposed. The schematic was described with a simple mass-damper-spring system as shown in Figure 2.18(b). Only aeroelastic efficiency, i.e., the efficiency of converting flow energy into mechanical energy, was considered, yet the efficiency of converting flow or mechanical energy into electricity was not considered since specific mechanical-to-electrical transduction mechanism was not indicated. It was shown that the efficiency was greatly influenced by the mass-damping parameter $m^*\zeta$, and there existed an optimal $m^*\zeta$ for peak efficiency at a specific Reynolds number. The range of reduced wind speeds with significant efficiency was found to be mainly governed by m^* . Also, it was found that high efficiency could be achieved for high Reynolds numbers.

Abdelkefi et al. (2012f) conducted a thorough analytical studies on the influences of several parameters on the power level, displacement and synchronization region (i.e., vortex shedding frequency matches the resonance frequency) of a VIV harvester. Similar to the schematic of Barrero-Gil et al. (2011) in Figure 2.18(b), a rigid cylinder was flexibly supported. Piezoelectric transduction mechanism was employed by attaching a piezoelectric transducer to the transverse degree of freedom. A 1DOF electromechanically coupled model was integrated with the model of the aerodynamic lift force, which was expressed by a van der Pol equation. Theoretically, it was found that increasing the load resistance shifted the onset of synchronization to higher wind speeds. A hardening behavior and hysteresis were observed in the displacement, voltage and power responses due to the cubic nonlinearity in the lift coefficient, which deserved a careful analysis in the studies of energy harvesting from VIV.

Similar results were observed in a subsequent study of Mehmood et al. (2013). With the same schematic of Abdelkefi et al. (2012f), the aerodynamic loads due to vortex shedding were obtained using CFD simulations, and were subsequently coupled with the electromechanical model to predict the electrical output. It was found that the region of wind speeds at synchronization was slightly widened when the load resistance increased. An optimum value of load resistance for maximum power output was determined to correspond to the load value with minimum displacement

of the cylinder.

Gao et al. (2013) proposed another configuration of harvester, different from that in Figure 2.44, consisting of a piezoelectric cantilever with a cross-flow cylinder attached to its free end, of which the long axes were arranged in parallel, as shown in Figure 2.48(a). Prototypes were constructed and tested in both laminar flows generated by a wind tunnel and turbulent flows generated by an electric fan (Figure 2.48(b)). Experimentally, it was found that the power output increased with the wind speed and cylinder diameter. Higher voltage and power were generated in the turbulent flow than in the laminar flow. It was concluded that turbulence excitation was the dominant driving mechanism of the harvester, with additional contribution from vortex shedding excitation in the lock-in region. With a piezoelectric cantilever of $31 \times 10\text{mm}$ and a cylinder with a length of 36mm and a diameter of 29.1mm, a peak power of $30\mu\text{W}$ was measured at a wind speed of 5m/s in the fan-generated turbulent flow.

Figure 2.48 Schematic of harvester with piezoelectric cantilever and cylinder connected in parallel (a) experimental setup in a wind tunnel, (b) experimental setup using an electric fan (Gao et al., 2013)

Instead of directly using the impinges of shedded vortices on the piezoelectric membrane or cantilever to induce mechanical oscillations, Tam Nguyen et al. (2013) proposed a miniature pneumatic energy harvester to induce deflections of a piezoelectric film by the fluctuating pressure behind the bluff body due to vortex shedding, as shown in Figure 2.49. Two bluff bodies were placed in the flow channel in a tandem arrangement to enhance the pressure fluctuation behind them. A flexible diaphragm was installed on the wall of the flow channel. A piezoelectric

film was glued to a bulge that was fixed with the flexible diaphragm. The pressure behind the bluff bodies fluctuated due to the vortex shedding, causing the flexible diaphragm as well as the piezoelectric film to deflect upward and downward periodically. A prototype was assembled as shown in Figure 2.49, with an embedded 0.2mm thick polydimethylsiloxane (PDMS) diaphragm on top surface of the flow channel, two triangular bluff bodies with a base length of 4.25mm and an altitude of 2.18mm inserted inside, and a PVDF film of $25\text{mm} \times 13\text{mm} \times 0.205\text{mm}$ glued to the acrylic bulge on top of the diaphragm and the acrylic anchor on top of the flow channel plate. An open-circuit voltage of 14mV and an average power of 0.59nW were measured with the prototype at a wind speed of 20.7m/s. The power is relatively low as compared with other harvesters that have been reviewed. It was suggested that the power can be enhanced by optimizing the blockage ratio (i.e., the ratio of the base length of the bluff body to the height of the flow channel) to 0.33, adjusting the position of the center of the flexible diaphragm to be just above the aft bluff body and using a piezoelectric material with higher piezoelectric constants. This harvester design was recommend to be deployed in the pipelines, tire cavities or machinery by installing a diaphragm on the wall.

Figure 2.49 Schematic of miniature pneumatic energy harvester (upper) and fabricated prototype (below) of Tam Nguyen et al. (2013)

Current studies on energy harvesting from VIV also include the investigation of the interaction of a single and multiple vortices with a cantilever conducted by Goushcha et al. (2014) as an extension of the work by Akaydin et al. (2010a, 2010b). Particle image velocimetry was used to measure the flow field induced by each

controllable vortex and quantify the pressure force on the beam to give a better understanding of the fluid structure interactions. The two driving mechanisms of upstream flow impingement on one side of beam and suction of the vortex core on the opposite side (Akaydin et al., 2010a, 2010b) and the importance of matching the frequency of appearance of vortices with the beam's resonance frequency were demonstrated clearly via the flow visualization, as shown in Figure 2.50.

Figure 2.50 Time dependent strain and flow visualization when vortex arrival rate matches beam's resonance frequency (Goushcha et al., 2014)

Energy harvesting from VIV was also studied by Sanchez-Sanz et al. (2009) with a microresonator and Wang et al. (2014) with the synchronization region demonstrated with CFD.

2.3.3 Energy harvesters based on galloping

It is not until recently that the aeroelastic instability phenomenon of transverse galloping is employed to obtain structural vibrations for energy harvesting purpose. Due to the self-excited and self-limiting characteristics of galloping, it is a prospective energy source for energy harvesting. Moreover, compared to the VIV, galloping owns its advantages of large oscillation amplitude and the ability of oscillating in infinite range of wind speeds, which are preferable for energy harvesting.

Barrero-Gil *et al.* (2010) theoretically analyzed the potential use of galloping to harvest energy using a 1DOF model. The harvesting system was modeled as a

simple mass-spring-damper system. No specific energy harvester design was proposed. The aerodynamic force due to galloping was formulated using a cubic polynomial based on the quasi-steady hypothesis (refer to Eq. (2.34)). The influence of cross-section geometry and mechanical properties on the energy conversion efficiency was investigated. Theoretically, it was found that in order to achieve a high efficiency, the bluff body should have a high aerodynamic coefficient A_1 and a low absolute value of A_3 (generally, $A_1 > 0$, $A_2 = 0$ and $A_3 < 0$). For the mechanical damping, it was determined that a low value of the mass-damper parameter $m\zeta$ should be used, where m is the distributed mass of the bluff body. The natural frequency of the system should be carefully adjusted according to the frontal dimension of the bluff body to increase the power extracted per unit working area, which is the calculated by dividing the extracted power by the area normal to the wind flow covered the bluff body in its oscillation.

A galloping energy harvester consisting of two cantilever beams and a prism with equilateral triangular cross section attached to the free ends was proposed by Sirohi and Mahadik (2011), as shown in Figure 2.51(a). A coupled electromechanical model was established based on the Rayleigh-Ritz method, and the aerodynamic model was based on the quasi-steady hypothesis. Power output versus wind velocity was shown in Figure 2.51(b). Due to the large size and high coupling coefficient of the piezoceramic sheets, a high peak power was achieved as more than 50mW at a wind velocity of 11.6mph (around 5.2m/s) in the laminar flow condition. But an abrupt decrease in output power occurred at 13.6mph, which was not in consistence with the galloping mechanism. The authors attributed this decrease to the large-scale turbulence in the wind tunnel.

Figure 2.51 (a) Schematic of galloping piezoelectric energy harvester and (b) Power output versus wind velocity for triangular tipped harvester (Sirohi and Mahadik, 2011)

Another galloping energy harvester using a tip body with D-shaped cross section connected in parallel with a piezoelectric composite cantilever was developed by Sirohi and Mahadik (2012) (Figure 2.52). The wind flow was generated by an axial fan, which was associated with a highly turbulent profile. The measured voltage generated by the piezoelectric sheets showed that a stable limit cycle oscillation could be obtained for the steady state response (Figure 2.53(a)). Also, the frequency of oscillation was found to be equal to the natural frequency of the cantilever. This is consistent with the galloping mechanism presented in Section 2.2.1.2. A continuously increasing power output with the wind velocity was observed, as shown in Figure 2.53(b).

Figure 2.52 Schematic of a D-shaped galloping energy harvester proposed by Sirohi and Mahadik (2012).

Figure 2.53 (a) Measured voltage in time domain and (b) power output versus wind velocity for D-shaped galloping harvester (Sirohi and Mahadik, 2012)

A comparative study of different bluff body cross-sections for small scale wind energy harvesting based on galloping was conducted by Zhao et al. (2012) and Yang

et al. (2013). Wind tunnel experiment was carried out with a prototype device consisting of a piezoelectric cantilever and a bluff body with various cross-section profiles. Square, rectangle, triangle, and D-shape were considered, as shown in Table 2.2. Figure 2.54 shows an example of the schematic and fabricated prototype of the galloping energy harvester with a square bluff body. Responses of power versus load resistance showed that there existed an optimal load value for maximum output power. Moreover, it was experimentally determined for the first time that the square section generates the largest power and has the lowest cut-in wind speed among all the considered sections, as shown in Figure 2.55. With a $150\text{mm} \times 30\text{mm} \times 0.6\text{mm}$ cantilever and a $40\text{mm} \times 40\text{mm} \times 150\text{mm}$ bluff body, a peak output power of 8.4mW was measured at a wind speed of 8m/s with the optimal load resistance, which is sufficient to power a commercial wireless sensor node. A 1DOF model was used which successfully predicted the power response. Moreover, the analysis with the galloping force represented with a seventh order polynomial predicted a hysteresis region of output power, which was not captured in the experiment due to the unavoidable turbulent flow component in the wind tunnel. It was recommended that the square section should be used for small-scale wind galloping energy harvesters.

Table 2.2 Different cross sections of bluff body in comparative study of Yang et al. (2013)

Section shape
Dimension $h \times d$ (mm)	40×40	40×60	40×26.7	40 (side) 40 (dia.)

Figure 2.54 (a) Schematic and (b) fabricated prototype of galloping energy harvester

with a square bluff body (Yang et al., 2013)

Figure 2.55 Wind tunnel experimental results of power versus wind speed at respective optimal load resistance for different bluff body cross sections (Yang et al., 2013)

Abdelkefi et al. (2012g) theoretically investigated the concept of using a galloping square cylinder to harvest energy. The effects of Reynolds number and load resistance on the onset of galloping, amplitude of displacement and output power were analyzed. A normal form solution was provided to validate the numerical solution of the employed 1DOF model, with both solutions confirming that the instability of galloping is a supercritical Hopf bifurcation phenomenon. It was found via numerical calculations that for low Reynolds, the onset of galloping (cut-in wind speed) and output power increased while the displacement decreased with the load resistance, while for high Reynolds, there existed an optimal load with which the output power and the onset of galloping reached the maximum and the displacement was the minimum. Similar findings for high Reynolds were obtained for a triangle sectioned bluff body in a subsequent study of Abdelkefi et al. (2013b) with an Euler-Bernoulli distributed parameter model, and for more cross section shaped bluff bodies with a 1DOF model by Abdelkefi et al. (2013a) and with an Euler-Bernoulli distributed parameter model by Abdelkefi et al. (2014). Theoretically comparing the performances of a square, two isosceles triangles ($\delta=30^\circ$ for one and $\delta=53^\circ$ for the other, with δ being the base angle) and a D-section, the isosceles triangle with $\delta=30^\circ$ was recommended for small wind speeds while the D-section was recommended for high wind speeds. It should be noted that the aerodynamic coefficients used to calculate the galloping force are much sensitive to the flow condition (laminar or turbulent), which has great influences on the

galloping behavior of different cross-sections. For example, turbulence in the flow can stabilize the square section, while it destabilizes the D-section. That is why the D-section cannot oscillate in the wind tunnel as shown by Zhao et al. (2012), but it can gallop when placed in front of an axial fan. The impact of temperature on the power generation performance was studied by Abdelkefi et al. (2013c) since temperature affects the properties of the piezoelectric material. Theoretically, it was found that at a higher temperature, energy can be harvested at lower wind speeds with different load resistances; while at relatively higher wind speeds, more energy can be generated at a lower temperature.

In order to better understand the electroaeroelastic behaviors and provide a guideline to optimize the galloping piezoelectric energy harvester, Zhao et al. (2013) conducted a comparison of different modeling methods to weigh their validity, advantages and disadvantages, and carried out a parametric study to investigate the effects of various parameters regarding the design of the harvester. The 1DOF model, single mode Euler–Bernoulli distributed parameter model and multi-mode Euler–Bernoulli distributed parameter model were compared and validated with wind tunnel experiment on a harvester prototype with a square sectioned bluff body (Figure 2.54). It was found that all these models can successfully predict the variation of the average power with the load resistance and the wind speed, e.g., the results of power versus load at a wind speed of 5m/s shown in Figure 2.56 (a). Especially, higher modes were found not necessary in modeling since minor difference was observed between the single mode and multi-mode Euler–Bernoulli distributed parameter models. It was concluded that the distributed parameter model has a more rational representation of the aerodynamic force while the 1DOF model gives a better prediction of the cut-in wind speed and owns its merit for conveniently obtaining the electromechanical coupling coefficient for a fabricated prototype via direct measurement. The parametric study of the influence of load resistance, wind exposure area and mass of the bluff body, and length of the piezoelectric patches on the cut-in wind speed and the output power showed that the optimum load resistance gave the maximum cut-in wind speed and maximum output power at higher wind speeds, while it gave a valley of output power at low

wind speeds. Increasing the wind exposure area and decreasing the mass of the bluff body can increase the output power and reduce the cut-in wind speed. Moreover, larger length of piezoelectric patches was found to give larger cut-in wind speed and larger growth rate of output power, as shown in Figure 2.56(b). Yet in order to obtain the maximum power density (i.e. power per piezoelectric volume), it was suggested that a medium-long piezoelectric patch be used with careful sweep calculation.

Figure 2.56 (a) Comparison of performance of different modeling methods at a wind speed of 5m/s and (b) influence of length of piezoelectric patches on output power of a galloping harvester (Zhao et al., 2013)

A big issue of small scale wind energy harvesting is that most piezoelectric aeroelastic energy harvesters operate effectively only at high wind speeds or within a narrow speed range. To overcome this issue, i.e., to reduce the cut-in wind speed and enhance output power in the low wind speed range (e.g., lower than 5 m/s) which is typical for heating, ventilation and air conditioning (HVAC) systems' flow condition, Zhao et al. (2014a) proposed a 2DOF piezoelectric galloping energy harvester with a cut-out cantilever and two magnets which induces stiffness nonlinearity of the whole system, as shown in Figure 2.57(a). Wind tunnel experiment confirmed its effectiveness, obtaining a reduced cut-in speed of 1m/s and nearly four time increase in power at 2.5m/s with a magnet gap of 8mm, as compared to the conventional 1DOF harvester, as shown in Figure 2.57(b). The total output power was found to be enhanced in the low wind speed range up to

4.5m/s. It was concluded that the proposed 2DOF galloping harvester is suitable to power wireless sensing nodes for indoor monitoring applications and highly urbanized areas with only low speed wind flows available.

Figure 2.57 (a) Schematic of 2DOF piezoelectric galloping energy harvester for power enhancement at low wind speeds and (b) comparison of total output power for different magnet gap (Δ) configurations.

In a subsequent study of Zhao and Yang (2015a), the electromechanical coupling was greatly increased by bonding a beam stiffener to the cantilever substrate, enhancing the output power and power extraction efficiency for all three types of aeroelastic harvesters based on VIV, galloping and aeroelastic flutter. This will be introduced in Section 2.4.1. Furthermore, in the aspect of interface circuit, Zhao et al. (2014b, 2015a, 2015b) investigated the feasibility of enhancing power generation of a galloping energy harvester with a self-powered synchronous charge extraction (SCE) circuit and a self-powered synchronized switching harvesting on inductor (SSHI) circuit, which will be further reviewed in Section 2.4.2.

Bibo and Daqaq (2014) established a universal relationship between the dimensionless output power and the dimensionless wind speed for galloping energy harvesters, which was shown to be only sensitive to the aerodynamic properties of the bluff body, but independent of the mechanical or electrical design parameters of the harvester, as represented with a unique curve shown in Figure 2.58 (a). This curve significantly facilitates the optimization analysis and comparison of performances of different bluff bodies. Based on the unique curve of each bluff

body, comparison results are shown in Figure 2.58(b) for laminar flows with a maximum turbulence intensity of 5%. It was found that when all harvesters are optimally designed, a squared-section bluff body always outperformed the D-shaped and triangular sectioned bluff bodies, which agrees with the findings of Yang et al. (2013) and Zhao et al. (2012). A 53° isosceles-triangular section harvester was found to outperform the D-shaped one at high wind speeds but underperform the D-shaped one at low wind speeds.

Figure 2.58 (a) Demonstration of unique curve of galloping harvester with square-sectioned bluff body, with different symbols denoting different configurations. (b) Comparison of different sections. Squares for squared section, circles for D-shaped section, and triangles for 53° isosceles-triangular section. Dimensionless power is square of the dimensionless displacement $|y|/y_0$, symbols denote experimental results, and solid curve denotes the theoretical unique curve (Bibo and Daqaq, 2014).

Ewere and Wang (2014) employed the Krylov–Bogoliubov method to obtain analytical approximate solutions for galloping energy harvesters and found that the harvester with a square sectioned bluff body can outperform the rectangular sectioned one for all cases of load resistance and wind speed. In a subsequent study of Ewere et al. (2014), a bump stop was introduced to relieve the fatigue problem of a galloping energy harvester, as shown in Figure 2.59. Using an optimal bump stop design with a gap size of 5mm at the location of 130mm along the beam of length 228mm and a contact surface area of $12.7 \times 40 \text{ mm}^2$, a maximum 20% voltage reduction with substantial 70% reduction in limit cycle oscillation amplitude was observed from the wind tunnel experiment. It was concluded that the service life of a galloping harvester can be significantly improved by incorporating an impact bump stop.

Figure 2.59 Schematic of a galloping harvester incorporated with an impact bump stop (Ewere et al., 2014)

Continuing the feasibility study of harvesting energy from transverse galloping conducted by Barrero-Gil *et al.* (2010), Vicente-Ludlam et al. (2014) carried out a theoretical study of a galloping electromagnetic energy harvester by introducing the electromagnetic transduction mechanism into the 1DOF galloping system, as shown in Figure 2.60(a). It was found that with the load resistance tuned to the optimal value for each wind speed, the power extraction efficiency can remain high over a larger range of wind speeds as compared to a fixed value of the load resistance. In a subsequent study, Vicente-Ludlam et al. (2015) proposed a dual mass galloping electromagnetic energy harvesting system to enhance the energy extraction. Three configurations were investigated, with the electromagnetic generator (i.e., the magnet and coil for electromagnetic induction) located between the secondary mass and a fixed wall for the first configuration as shown in Figure 60(b), between the main and the secondary masses for the second one, and between a fixed wall and the primary mass for the third one. It was shown theoretically that when the mechanical properties were properly adjusted, the first and second configurations can improve the energy extraction efficiency and broaden the effective range of the wind speeds for energy harvesting. A potential practical realization of the first configuration is shown in Figure 2.60(c).

Figure 2.60 (a) Schematic of electromagnetic galloping energy harvester (Vicente-Ludlam et al., 2014); (b) Schematic and (c) a potential practical realization of dual mass harvesting system with the first configuration (A-galloping prism, B-springs, C-a linear guide, D- the secondary mass, E-the coil part and F-a permanent magnet array) (Vicente-Ludlam et al., 2015).

2.3.4 Energy harvesters based on flutter

Numerous designs of energy harvesters based on flutter instabilities have been reported under axial flow conditions (Tang et al., 2009; Dunnmon et al., 2011; Doaré and Michelin, 2011; Shan et al., 2014; Zhao and Ega, 2014; Seo et al., 2015) or cross flow conditions (Li et al., 2011; De Marqui et al., 2011), with flapping airfoil designs (Bryant and Garcia, 2009; Erturk et al., 2010; Abdelkefi et al., 2012b) or tree-inspired (Li et al., 2011) and infrastructure-inspired designs (Kwon, 2010). Examples of flutter based harvesters include the wind belt (Humdinger Wind Energy, Windbelt Innovation) and flutter mill (Flutter mill), etc. The two main types of flutter based energy harvesters include the modal convergence flutter based ones and the cross-flow flutter based ones, which are reviewed in detail in the following sections.

2.3.4.1 Energy harvesters based on modal convergence flutter

Among the explosive studies on small scale wind energy harvesting based on aeroelastic flutter, flapping airfoil or flapping wing based designs have been the most enthusiastically pursued. The mechanism of such flutter, i.e., modal

convergence flutter with coupled pitch-plunge motions, has been introduced in detail in Section 2.2.3.3. Energy harvesting based on airfoil flutter has been reported a few decades ago by Bade (1975) and McKinney and DeLaurier (1981) using electromagnetic transduction mechanism. A patent has been filed by Schmidt (1985) using airfoil blades with piezoelectric transduction mechanism. In the recent years, along with the explosive research on energy harvesting with piezoelectric materials, piezoelectric wind energy harvesting via airfoil flutter has become a hot research area with numerous studies reported.

Bryant and Garcia (2009, 2011) and coauthors (Bryant et al., 2011, 2012a, 2012b, 2013, 2014) are among the very first to investigate the feasibility of piezoelectric energy harvesting with airfoil flutter. An airfoil flutter based energy harvester was first reported by Bryant and Garcia (2009) with both wind tunnel experimental results and theoretical predictions, and further studied with a more detailed theoretical modeling process (Bryant and Garcia (2011)). A piezoelectric bimorph was connected to a rigid airfoil (NACA0012 airfoil profile) at the tip with a revolute joint, permitting both transverse and rotary displacement of the airfoil, as shown in Figure 2.61(a). The section representation of the airfoil showing the transverse and rotational degrees of freedom is similar to Figure 2.11 in Section 2.2.1.3. Theoretical analyses were carried out to predict behaviors of the harvester at the flutter boundary with linear models and during limit cycle oscillations above the flutter boundary with nonlinear models. The linear mechanical model incorporating electromechanical coupling was established using the energy method based on the Hamilton's principle; while the linear aerodynamic model was established based on the finite-state unsteady thin-airfoil theory of Peters et al. (1995). Small angle and attached flow assumptions were taken in the linear models. For the nonlinear models, large flap deflection angles and flow separation effects were taken into account, where the ONERA model, which is a semi-empirical unsteady nonlinear aerodynamic model based on ordinary differential equations, was employed to calculate the nonlinear aerodynamic lift and moment. Both the linear and nonlinear aerodynamic models have been reviewed in detail in Section 2.2.3.3. Wind tunnel experimental results of a fabricated harvester prototype agreed well with the

analytical predictions except for a slight discrepancy of the flutter speed, as shown in Figure 2.61(b). Around the critical wind velocity, a small initial condition could lead to oscillations that decay back to the trivial equilibrium, while a larger initial condition could lead to sustained limit cycle oscillations, revealing a subcritical Hopf bifurcation. The prototype consisted of a $254 \times 25.4 \times 0.381$ mm substrate cantilever attached with two PZT patches of dimension $46.0 \times 20.6 \times 0.254$ mm, and an airfoil with a semichord of 2.97 cm and a span of 13.5 cm. A cut-in wind speed of 1.86 m/s was obtained. A maximum output power of 2.2 mW was delivered to an optimal load of $277 \text{ k}\Omega$ at a wind speed of 7.9 m/s. At the flutter boundary, the real part of the two eigenvalues of the first bending and rotation modes turned from negative to positive, and the imaginary parts started to coalesce. It was concluded that collating and superposing the bender and system resonances can maximize the output power.

Figure 2.61 (a) An airfoil flutter-based piezoelectric energy harvester (b) variation of output power with wind speed (Bryant and Garcia, 2011)

In a subsequent work of Bryant et al. (2011), a parameter study was performed both experimentally and analytically to investigate the influence of several system design parameters on the cut-in wind speed. It was found that the cut-in wind speed could be minimized by modifying the hinge stiffness and the flap mass distribution, yet its variation was less sensitive to the hinge stiffness when large damping was introduced. Later, Bryant et al. (2012a) compared the quasi-steady aerodynamic model with the semi-empirical model considering dynamic stall effects. It was concluded that the quasi-steady model was applicable only for low flapping frequencies, while the dynamic stall model can be used to predict trustworthy

results at high frequencies. Bryant et al. (2012b) also investigated the influence of the compliance of the host structure on the behavior of the airfoil flutter based energy harvester. Experimentally, it was found that a compliant host structure reduced the cut-in wind speed, cut-in frequency and oscillation frequency during the limit cycle oscillation, and shifted the peak power toward the lower wind speeds, as compared to a stiff host structure. Bryant et al. (2013) presented an experimental study on energy harvesting efficiency and found that the peak power density and power extraction efficiency of the flutter energy harvester occurred at a low wind speed, at which the limit cycle oscillation frequency matched the first natural frequency of the piezoelectric structure. Preliminary study of implementing synchronized switching approaches like the synchronized switching and discharging to a storage capacitor through an inductor (SSDCI) technique in an aeroelastic flutter energy harvester was conducted, with a separate microcontroller working as the peak detector. However, the efficiency increasing capability of the SSDCI in flutter energy harvesting was not thoroughly studied. Subsequently, Bryant et al. (2014) experimentally demonstrated the concept of using ambient flow energy harvesting to power aerodynamic control surfaces. With a prototype that produced a power of 40mW at a wind speed of 26m/s, it was shown that the system produced more than 55° of tab deflection over approximately 0.7 second after the storage capacitor was charged for 235 seconds at 32.2 m/s. It was found that the harvester was still able to produce power when the host control surface was rotated to a large angle of attack over 50°, confirming the feasibility of the alternative design of placing the harvester on the control surface itself.

De Marqui and Erturk and coauthors (Erturk et al., 2008; Erturk et al., 2010; Sousa et al., 2011; De Marqui and Erturk, 2012; Dias et al., 2013, 2014) are also among the first to study harnessing flow energy via the aeroelastic flutter of airfoils. The concept of energy harvesting from macro-fiber composites with curved airfoil section was first proposed by Erturk et al. (2008). Later, Erturk et al. (2010) presented an experimentally validated lumped-parameter aeroelastic model for the flutter boundary condition of a piezoaeroelastic airfoil similar to Figure 2.11 in Section 2.2.1.3 with the piezoelectric coupling inserted to the plunge degree of

freedom. The linear lift and moment at flutter boundary were modeled with the Theodorsen's unsteady thin airfoil theory which has been introduced in Section 2.2.3.3. Using a flexibly supported airfoil prototype (Figure 2.62) with a semichord of 0.125m and a span length of 0.5m, a power of 10.7mW was measured at the linear flutter speed of 9.30 m/s with an optimal load of 100k Ω . It was found both theoretically and experimentally that the optimal load gave the maximum flutter speed due to the associated maximum shunt damping effect during power extraction. It was recommended that a nonlinear stiffness component and/or a free play can be incorporated to induce stable limit cycle oscillations above and below (in the case of subcritical Hopf bifurcation as shown in Figure 2.10(b)) the flutter boundary for useful power generation. As discussed in Section 2.2.3.3, in order to obtain stable limit cycle oscillations above or below the flutter boundary, nonlinearity has to be introduced into the system, which can be either structural nonlinearity or aerodynamic nonlinearity. The structural nonlinearity induced by the nonlinear stiffness component and/or a free play suggested by Erturk et al. (2010) and the aerodynamic nonlinearity modeled in Bryant and Garcia (2011) can both ensure the acquisition of stable and large-amplitude limit cycle oscillations beyond the linear flutter speed and harvest energy in a wide wind speed range.

Figure 2.62 Prototype of piezoaeroelastic airfoil proposed by Erturk et al. (2010)

In a subsequent study, Sousa et al. (2011) theoretically and experimentally investigated the advantages of exploiting structural nonlinearities in the piezoaeroelastic energy harvesting system with a schematic similar to that in Figure 2.62. Piezoelectric coupling was introduced to the plunge DOF while structural nonlinearities were introduced to the pitch DOF, aiming to solve the problem of a

linear piezoaeroelastic energy harvester, i.e., having persistent oscillations only at the flutter boundary thus leading to a very limited condition for energy harvesting. Free play nonlinearity and the combination of cubic hardening stiffness and free play nonlinearity were added to the pitch DOF separately. The restoring moments in the pitch DOF are shown in Figure 2.63, with comparison to the restoring moment of the linear configuration. The aerodynamic lift and moment were modeled with an approximation to the Theodorsen's function. It was shown both theoretically and experimentally that the free play nonlinearity reduced the cut-in wind speed by 2m/s and doubled the output power. Theoretically, it was found that the hardening stiffness brought the plunge amplitude to acceptable levels thus increased the upper limit of the operational wind speed beyond the flutter speed. It was concluded that the combined structural nonlinearities can be introduced to enhance the performance of aeroelastic energy harvesters based on piezoelectric and other transduction mechanisms.

Figure 2.63 Introducing structural nonlinearity into piezoaeroelastic system: restoring moments in pitch DOF for linear configuration, free play nonlinearity and combined cubic and free play nonlinearity (Sousa et al., 2011).

De Marqui and Erturk (2012) theoretically analyzed the performance of two airfoil-based aeroelastic energy harvesters using piezoelectric transduction and electromagnetic induction, as shown in Figure 2.64, with piezoelectric and electromagnetic couplings inserted to the plunge DOF separately. It was found that there existed optimal values of load resistance giving the largest flutter speed as well as the maximum output power for the considered range of dimensionless equivalent capacitance and dimensionless electromechanical coupling in the

piezoelectric configuration. For the electromagnetic configuration, increasing the load resistance reduced the flutter speed for any dimensionless inductance or electromechanical coupling, and the value of load resistance that generated the maximum output power matched the internal coil resistance, which agreed with the maximum power transfer theorem.

Figure 2.64 Two airfoil-based aeroelastic energy harvester configurations using (a) piezoelectric transduction and (b) electromagnetic induction (De Marqui and Erturk, 2012)

Subsequently, Dias et al. (2013) theoretically analyzed the performance of a hybrid airfoil-based aeroelastic energy harvester using simultaneous piezoelectric and electromagnetic induction. It was shown that in the electromagnetic induction, the internal coil resistance affected the flutter speed and deteriorated the performance of the system. The parameter study showed that the combination of low dimensionless radius of gyration, low pitch-to-plunge frequency ratio and large dimensionless chord-wise offset of the elastic axis from the centroid enhanced the performance by increasing the output power as well as decreasing the cut-in wind speed.

Later, Dias et al. (2014) continued the study of the hybrid aeroelastic energy harvester with combined piezoelectric and inductive couplings based on a 3DOF airfoil, as shown in Figure 2.65. A control surface was introduced, bringing in a third displacement, i.e., the control surface displacement. Theoretical parametric study showed that increasing the dimensionless radius of gyration, dimensionless chordwise offset of the elastic axis from the centroid, and control surface pitch-to-plunge frequency ratio, and decreasing the pitch-to-plunge frequency ratio increased the power output while reduced the cut-in speed. It was concluded that the

3DOF configuration enhanced the performance of the harvester by offering a broader design space and set of parameters for system optimization. It was recommended to introduce electromechanical coupling into the control surface DOF to further enhance the performance.

Figure 2.65 Schematic of a 3-DOF hybrid piezoelectric-inductive aeroelastic energy harvester with a control surface introduced (Dias et al., 2014)

Earliest studies of airfoil-based aeroelastic energy harvesters also include the work of Abdelkefi et al. (2012a-e) and Abdelkefi and Nuhait (2013). Abdelkefi et al. (2012a) theoretically investigated the performance of an airfoil-based piezoaeroelastic energy harvester with the method of normal form, which was validated by the numerical integrations. The nonlinear aerodynamic lift and moment were modeled with a quasi-steady approximation incorporating a stall model (Section 2.2.3.3). It was found that the system's instability to the subcritical type depended significantly on the cubic nonlinearity of the torsional spring. In a subsequent study, Abdelkefi et al. (2012b) implemented two linear velocity feedback controllers, as shown in Figure 2.66, to reduce the flutter speed to any desired value and hence generate energy from limit cycle oscillations at any desired low wind speed. This was realized by introducing two vibration velocity dependent terms into the system governing equations. The velocities of the pitch and plunge were sensed and fed to circuits, of which the outputs were sent to the actuators that forced the pitch and plunge motions. It was found theoretically that the aerodynamic nonlinearity produced a supercritical bifurcation while the cubic stiffness nonlinearity produced supercritical or subcritical bifurcation depending on whether the stiffness nonlinearity was hard or soft. Moreover, it was shown that at high wind speeds far away from the flutter speed the oscillations became quasi-periodic or even chaotic, which could be postponed to higher speeds using the

introduced velocity feedback controllers.

Figure 2.66 Schematic of piezoaeroelastic energy harvester with linear velocity feedback controllers (Abdelkefi et al., 2012b)

Subsequently, Abdelkefi et al. (2012c) investigated the effects of the eccentricity between the gravity and elastic axes (Figure 2.11) on the performance of the piezoaeroelastic energy harvester. Theoretically, it was found that by properly choosing small eccentricities, the output power to an external load can be increased by an order of magnitude. The significant sensitivity of the harvested power to the eccentricity and the nonlinear coefficient of the torsional spring was also observed theoretically in Abdelkefi et al. (2012e).

Abdelkefi et al. (2012d) further introduced a third DOF, i.e., the control surface flap, into the piezoaeroelastic energy harvester and performed a nonlinear analysis with the aerodynamic loads formulated with two methods, i.e., the quasi-steady approximation and the unsteady representation based on the Duhamel formulation. Theoretical comparison based on normal form solutions showed that the two methods predicted comparable results when the pitch and plunge motions were periodic. However, the quasi-steady approximation failed to give accurate predictions at high wind speeds above 1.05 times of the flutter speed and when the motions became non-periodic or chaotic with subcritical instability. This was explained to be attributed to the failure of properly considering the dynamic wake effects of the flap of the quasi-steady approximation.

In a subsequent study, Abdelkefi and Nuhait (2013) considered an airfoil-based piezoaeroelastic energy harvester where the airfoil was not symmetric but with a

camber, as shown in Figure 2.67. The effects of the camber value and location on the flutter speed and the output power of the harvester were investigated theoretically. It was found that the flutter speed changed significantly with the increasing camber for small values of the camber location, yet was not sensitive to the camber for large values of the camber location. Increasing the camber was found to decrease the output power and delay the secondary Hopf bifurcation which was associated with chaotic motions; while increasing the camber location was found to increase the output power. It was recommended that symmetric airfoil should be used to enhance the output power of airfoil-based piezoaeroelastic energy harvesters.

Figure 2.67 Schematic of cambered piezoaeroelastic system (Abdelkefi and Nuhait, 2013)

Instead of harvesting energy only from flowing wind, Bibo and Daqaq (2013a) theoretically investigated the performance of an airfoil-based piezoaeroelastic energy harvester which concurrently harvested energy from ambient vibrations and wind, by introducing harmonic base excitation in the plunge direction. The nonlinear aerodynamic lift and moment were modeled with the quasi-steady approximation. Cubic nonlinearities were introduced for the plunge and pitch. Complex motions were predicted under the combined excitations by analytical solutions based on the normal form method, which were validated by numerical integrations. In a subsequent study, Bibo and Daqaq (2013b) performed experiment to demonstrate these complex motions by attaching the harvester prototype to a seismic shaker which provided the harmonic base excitation and putting the whole system in a wind tunnel which provided the aerodynamic loads, as shown in Figure 2.68(a). It was found both theoretically and experimentally that below the flutter speed, the flow amplified the output power from base excitations, with the harvester undergoing periodic response. Beyond the flutter speed, it was shown that when the

base excitation frequency was not close to the frequency of the self-sustained oscillations induced by the flutter instability, the harvester underwent two-period quasi-periodic oscillations with amplitude modulation in the time domain, with which the electric output approached that from the aeroelastic flutter only. When the excitation and flutter induced oscillation frequencies were close, synchronized periodic oscillations occurred with the same period as the excitation. The power was enhanced when the excitation frequency was right above resonance, while it dropped when the excitation frequency was slightly below resonance, as shown in Figure 2.68(b). It was concluded that the harvester under combined excitations was superior to that under one type of excitation. The output power was improved by over three times compared to that from an aeroelastic harvester and a vibration harvester together, as shown in Figure 2.68(b).

Figure 2.68 (a) Experiment setup of an airfoil-based piezoaeroelastic energy harvester under combined based excitation and aerodynamic loads (b) comparison of power from two harvesters under combined excitations (dashed) and combined power from an aeroelastic energy harvester and a vibration energy harvester. σ represents shift of excitation frequency from resonance.

Bae and Inman (2013) analyzed the performance of an airfoil-based piezoaeroelastic energy harvester with the root-locus method and time-integration method. It was concluded that the characteristics of responses depended on the pitch-to-plunge frequency ratio which should be properly designed to induce stable limit cycle oscillations (LCOs). It was found that energy can be harvested from stable LCOs when the frequency ratio was larger than 1.0 in a wide range of wind speeds below the flutter speed for free play nonlinearity and over the flutter speed for cubic hardening nonlinearity.

Wu et al. (2015) theoretically investigated the performance of an airfoil-based piezoaeroelastic energy harvester with free play nonlinearity and showed that the amplitudes of pitch and plunge motions as well as output power increased with the free play gap, with the power and the gap having an approximate linear relationship in particular. Moreover, it was found that discrete gusts in the incoming flows influenced the phase of the dynamic and electrical responses yet had no influence on the electrical output amplitude.

Different from the utilization of airfoils attached to cantilever tips, Kwon (2010) experimentally investigated the performance of a T-shaped piezoelectric cantilevered energy harvester shown in Figure 2.69. The T-shape resembled a half H-sectional shape of the Tacoma Narrow Bridge which collapsed in 1940 due to large amplitude aeroelastic flutter. The T-shape cantilever underwent coupled bending and torsional motions in wind flows. Using a prototype with $L \times B \times H = 100 \times 60 \times 30 \text{ mm}$, a 0.2 mm thick aluminum substrate and six attached PZT patches of $28 \times 14 \text{ mm}$ for each, a flutter speed of 4m/s and a maximum power of 4.0mW at a wind speed of 15m/s were measured with a load of $4 \text{ M}\Omega$. Annual output energy of 4.3Wh was calculated at an assumed mean wind speed of 5m/s. It was concluded that it had the potential to power a mobile electronic apparatus cost-effectively with several proposed harvesters of simple structure design consisting of only a bimorph cantilever.

Figure 2.69 Schematic of T-shaped piezoelectric cantilever for wind energy harvesting (Kwon, 2010)

Subsequently, Park et al. (2014) continued the study of T-shaped cantilever where

electromagnetic transduction was employed, as shown in Figure 2.70. It was found experimentally that the onset of the harvester occurred only when the load resistance surpassed a certain value, i.e., the flutter onset resistance. CFD simulations were performed to estimate the aerodynamic damping thus predict the flutter onset resistance, which agreed well with experiment. Using a prototype of $42 \times 30 \times 20 \text{ mm}$ with a 0.1016 mm thick cantilever substrate, a maximum power of around 1.1 mW was delivered to a $1 \text{ k}\Omega$ load at a wind speed of 8 m/s .

Figure 2.70 Schematic of T-shaped electromagnetic flutter energy harvester (Park et al., 2014)

Modal convergence flutter based energy harvester designs also include the work of Boragno et al. (2012). In their design, a wing was attached to a support by two elastomers with one for each side, which provided bending and torsional stiffness at the same time. Influences of parameters including the elastomer elastic constant, wing mass, position of mass center and elastic attachment point on oscillation responses were investigated. The self-sustained oscillations with properly adjusted parameters were concluded to be suitable for energy harvesting purpose though no specific transduction mechanism was proposed. A similar device called flutter mill has been demonstrated by Sharp (Flutter mill) with electromagnetic transduction.

2.3.4.2 Energy harvesters based on cross-flow flutter

Inspired by the natural flapping leaves in the tree subject to ambient flows, Li and Lipson (2009) proposed a device consisting of a PVDF stalk, a plastic hinge and a triangular polymer/plastic “leaf”, as shown in Figure 2.71. Two configurations were investigated with different direction arrangements of the stalk, i.e., the horizontal-stalk leaf and the vertical-stalk leaf. The horizontal-stalk underwent

bending motion while the leaf underwent coupled bending and torsional motions around the hinge, i.e., modal convergence flutter, similar to the airfoil-based piezoaeroelastic energy harvester. The vertical-stalk, where the long axes of the stalk was perpendicular to the incoming flow, underwent cross-flow flutter, which was demonstrated more clearly in a subsequent study of Li et al. (2011) (Figure 2.72) with more experiments performed. It was found that the cross flow configuration generated one order of magnitude more power than the parallel configuration. Performances of different leaf's shapes, different leaf's area, different stalk scales (short, long, narrow-short) and different PVDF layer configurations (single-layer, adhered double-layer, and air-spaced double-layer) were measured and compared. It was found that the circle, square, and equilateral triangle shapes of leaf had similar and the best performance, and the cut-in wind speed and peak power increased with leaf's area. Stalk scale and PVDF layer configuration affected the power with a non-monotonous, complex behavior. A peak power of $615\mu\text{W}$ was obtained with an adhered double-layer stalk of $72\times 16\times 0.41\text{mm}$ at 8m/s on a $5\text{M}\Omega$ load and the maximum power density of $2036\mu\text{W}/\text{m}^3$ was obtained with a unimorph narrow-short stalk of $41\times 8\times 0.205\text{mm}$ at 7m/s on a $30\text{M}\Omega$ load. It was concluded that although the proposed device had low power density compared to commercial wind turbines, it owned advantages of being robust, simple, miniature sized and able to blend in urban and natural environments.

Figure 2.71 Schematic of flapping-leaf energy harvester (a) horizontal-stalk leaf and (b) vertical-stalk leaf (Li and Lipson, 2009)

Figure 2.72 Motion sketch of (a) parallel-flow (horizontal) stalk-leaf harvester and (b) cross-flow (vertical) stalk-leaf harvester (Li et al., 2011).

Analyses of flapping-leaf energy harvesters similar to that in Figure 2.71(a) were also performed by McCarthy et al. (2013, 2014) with smoke-flow visualization and tandem harvester arrangements and coauthors Deivasigamani et al. (2014) with a parallel-flow asymmetric configuration proposed as shown in Figure 2.73, where the offset of the leaf axes from the stalk axes induced torsional motions of stalk around axis x . It is actually a modal convergence flutter based harvester, due to the similar constructions to those in Figure 2.71, we put its reviews in this section of cross-flow flutter. The PVDF partly operated in the d_{32} mode which had a low piezoelectric conversion coefficient, therefore, it was concluded that power from torsion (d_{32}) in parallel flow configuration acted only as a low-value peripheral supplement to that from bending. The output was similar to that of a parallel flow flapping-leaf harvester, much lower than that of a cross-flow counterpart harvester.

Figure 2.73 Schematic of parallel-flow flapping-leaf energy harvester with asymmetric configuration (Deivasigamani et al., 2014)

Studies on energy harvesting from cross-flow flutter were also conducted by De

Marqui et al. (2010, 2011), aiming at harvesting energy with the wings of unmanned air vehicles (UAVs). Using an electromechanically coupled finite element (FE) model, a preliminary study was performed by De Marqui et al. (2009) on a plate with embedded piezoceramics under base excitations with no air flows. Subsequently, aerodynamic loads were introduced to the plate to simulate the condition during flight by De Marqui et al. (2010) using coupled FE model and unsteady vortex-lattice model to predict the electric outputs. In a later study of De Marqui et al. (2011), segmented electrodes shown in Figure 2.74(a) were used to avoid cancelation of electrical output during typical coupled bending-torsion aeroelastic modes, as shown in Figure 2.74(b). It was found that the peak power from the segmented electrode was larger than that from the continuous electrode for all considered load resistance at a flutter speed of 40 m/s. Torsional motions of the coupled modes were found to become relatively significant for segmented electrodes, associated with improved broadband performance and increased flutter speed. Based on the doublet-lattice method, De Marqui et al. (2011) conducted further analysis with a frequency-domain piezoaeroelastic model that incorporated an electromechanical FE model and an unsteady aerodynamic model. At the flutter speed, the frequency for effective power output with coupled bending-torsion motions was clearly shown using the calculated FRFs.

Figure 2.74 (a) Schematic of a piezoelectric power generator wing under cross flow with segmented electrodes (De Marqui et al., 2010) and (b) coupled bending-torsion mode shape at flutter speed (De Marqui et al., 2011)

Cross-flow flutter based energy harvesters also include the patented windbelt that was produced by Humdinger Wind Energy, LLC (Humdinger Wind Energy, Windbelt Innovation), which extracts energy from wind using electromagnetic transduction with a properly tensioned flexible belt undergoing flutter motions

when subjected to flows.

Finally, a novel type of flutter termed dual cantilever flutter different from the above mentioned cases was reported and analyzed for energy harvesting purpose by Hobeck et al. (2014), which has been reviewed in Section 2.2.5.

2.3.5 Energy harvesters based on wake galloping

Wind energy extraction exploiting wake galloping phenomenon was studied by Jung and Lee (2011). They developed a device consisting of two paralleled cylinders to extract power from the leeward cylinder, which oscillated due to the wakes from the windward cylinder (Figure 2.75). Electromagnetic transduction was employed. It was found that with proper distance (4-5 times the cylinder diameter) between the parallel cylinders, the leeward cylinder could oscillate with considerable magnitude. With two cylinders of 5cm in diameter and 0.85m in length with a space of 25cm in between, an average output power of 50–370mW was measured under a wind speed of 2.5–4.5m/s with different coil and spring configurations. Piezoelectric energy harvesting could also utilize the wake galloping with proper arrangement of parallel cylinders based on these results.

Figure 2.75 (a) Prototype of electromagnetic generator and (b) elastic mounting system using springs (Jung and Lee, 2011)

Abdelkefi et al. (2013d) enhanced the performance of an energy harvester with the wake galloping phenomenon by placing a circular cylinder in the windward direction of a square galloping cylinder, both of which were attached with MFC sheets at the root. It was found experimentally that the range of wind speeds for effective energy

harvesting can be significantly widened. The level of enhancement significantly depended on the upstream cylinder size and spacing distance between the two cylinders. With an upstream cylinder of 27.15cm in length and 1.25cm in diameter, the power from the square cylinder was greatly enhanced when the spacing was larger than 16cm, especially from 1.8m/s to 3.51m/s where the single square cylinder generated no power without the introduced wake galloping effects.

2.3.6 Energy harvesters based on turbulence-induced vibration

A big problem of the above mentioned harvesters is that most of them oscillate and generate energy only in laminar flow conditions, which are usually not the case in natural environment where turbulence exists thus stabilizes the harvesters. Also, they require a cut-in speed as the minimum limit of wind speed, below which no power can be generated. Yet turbulence-induced vibrations (TIVs) never vanish even with very small average wind speed, which could be utilized by energy harvesters based on TIVs (Hobeck and Inman, 2012a).

Akaydin et al. (2010a) experimentally investigated the performance of a cantilevered harvester placed in turbulent boundary layer flow near the bottom wall of a wind tunnel, as shown in Figure 2.76. It was found that electrical output monotonically increased with the wind speed. With a boundary layer thickness of $\delta=115\text{mm}$, the global and local maxima of power were observed independent of wind speed at $h=40\text{ mm}$ and 75 mm , respectively, which were far away from the maximum turbulent kinetic energy location of around 8 mm . Time domain signals of voltage showed that the dominant frequency of 46 Hz was close to the resonance frequency of the beam; while the secondary dominant frequency was around 317 Hz near the turbulence frequency of 275Hz , leading to the conclusion that higher frequency excitations due to turbulence existed in TIVs.

Figure 2.76 Schematic of placing an elastic beam harvester into turbulent boundary

layer to induce vibration, with smoke visualization on left (Akaydin et al., 2010a)

Hobeck and Inman (2012a) proposed the concept of harvesting energy from highly turbulent flows with “piezoelectric grass”, as shown in Figure 2.77, consisting of an array of generating elements in the highly turbulent wake of a bluff body or in entirely turbulent fluid flows. A combination technique of electromechanical modeling for the structure and statistical modeling for the turbulent induced forces was proposed, which was the first documented experimentally validated TIV energy harvesting model. This model was further modified in a subsequent study of Hobeck and Inman (2014) as reviewed in Section 2.2.3.5. Experimentally, a peak power output of 1.0mW per cantilever was measured for the four-element harvester array fabricated with PZT ($101.60\text{mm} \times 25.40\text{mm} \times 101.60\mu\text{m}$ steel substrate attached with $45.97\text{mm} \times 20.57\text{mm} \times 152.40\mu\text{m}$ PZT) at a mean wind speed of 11.5m/s and a load of $49.2\text{k}\Omega$. A peak power of $1.2\mu\text{W}$ per cantilever was measured at 7m/s on a $4.70\text{M}\Omega$ load for the six-element array with PVDF ($72.60\text{mm} \times 16.20\text{mm} \times 178.00\mu\text{m}$ Mylar substrate attached with $62.00\text{mm} \times 12.00\text{mm} \times 30.00\mu\text{m}$). It was found that the performance depended on the wind speed and harvester location downstream of a bluff body, and the optimum condition occurred when the harvester’s natural frequency matched the primary bluff body’s vortex shedding frequency. The main advantage was concluded to be in its robustness and survivability due to its inherent redundancy since only minor reduction in total power happened if one element was damaged.

Figure 2.77 Schematic of “piezoelectric grass” arrays fabricated with (a) PVDF and (b) PZT (Hobeck and Inman, 2012a)

2.3.7 Other small-scale wind energy harvester designs

With designs different from the above mentioned cases, recent progress on energy harvesting from wind flows also includes a damped cantilever pipe carrying flowing fluid (Elvin and Elvin, 2009c), a harmonica-type aeroelastic micropower generator (Bibo *et al.*, 2012), a tensioned piezoelectric film facing laminar and/or turbulent incoming flows (Ovejas and Cuadras, 2011), a hinged-hinged piezoelectric beam facing turbulent airflows (Kwuimy *et al.*, 2012), and a micromachined piezoelectric airflow energy harvester inside a Helmholtz resonator (Matova *et al.*, 2011).

Bibo *et al.* (2012) proposed a harmonica-type micropower generator consisting of a cantilever embedded within a cavity, as shown in Figure 2.78. Pressure from the incoming flow caused deflection of the cantilever, generating a small gap which in turn reduced the flow pressure. The periodic fluctuations in the pressure induced the beam to undergo self-sustained oscillations. It was found that using optimal chamber volume and decreased aperture's width reduced the cut-in wind speed. The optimal load for maximum power did not vary considerably with the inflow rate. With a $58 \times 16.26 \times 0.38$ mm piezoelectric cantilever, an average power of around $50 \mu\text{W}$ was obtained at an average wind speed of 7.5 m/s , which was sufficient to power a health monitoring sensor interface chip.

Figure 2.78 Schematic of harmonica-type aeroelastic micropower generator (Bibo *et al.*, 2012).

Ovejas and Cuadras (2011) investigated the energy harvesting performances of a PVDF film generator with three different setup configurations, as shown in Figure

2.79. In the bluff body configuration of setup (a), two cylindrical bluff bodies of 0.5cm diameter are attached to the PVDF film in the windward direction, with the wind flowing parallel with the surface film; while in the other two configurations with one or two ends fixed, i.e., setup (b) and (c), the wind flows perpendicularly to the surface film. The generated energy was delivered to a storage capacitor after rectification through a diode bridge. Two types of flow conditions were considered: laminar flow generated by a wind tunnel and turbulent flow generated by a hairdryer. Experiment showed that the turbulent flow gave a higher voltage than the laminar flow did. It was explained that the rotational turbulent flow added to the vortex shedding from the two bluff bodies, thus enhancing power generation. Effects of the film geometrical parameters were investigated. It was found that the voltage in the storage capacitor increased nonlinearly with the film thickness. For setup (b) and (c), the voltage increased with the film area (which depended generally on the film length because the film width was not significantly varied), but for setup (a), voltage was independent with the film area. It was explained that the piezofilm was not effectively strained out of the effective length of the shedded vortex. This observation is quite similar to that of the abovementioned energy harvesting eel (Allen and Smits, 2001). With performance comparison, setup (a) outperformed the other two and was recommended for energy harvesting. A maximum power of $0.2\mu\text{W}$ was measured with a setup (a) film that was 15.6cm in length, 1.9cm in width, $40\mu\text{m}$ in thickness and 9.9nF in capacitance. Future studies were recommended to include the determination of the contribution of multi modes and optimization of the oscillation and resonant frequency coupling.

Figure 2.79 Configurations of PVDF film generator (a) bluff body configuration, (b) one fixed sided and (c) two fixed side (Ovejas and Cuadras, 2011).

2.4 Enhancement techniques involved in small-scale wind energy harvesting systems

2.4.1 Enhancement with modified structural configurations

In the literature, various methods have been proposed to improve the efficiency of power output for the vibration-based piezoelectric energy harvesters. The beam configuration can greatly influence the strain distribution throughout the harvester, resulting in significant difference in power generation. For example, a trapezoidal cantilever harvester can generate more than twice power than the rectangular one (Roundy *et al.*, 2005). The multi-modal techniques can enlarge the bandwidth of operating frequencies of the vibration-based energy harvesters, e.g., the piezoelectric energy harvester with a dynamic magnifier (Arafa *et al.*, 2011) and the 2DOF energy harvester with closed translational and rotational mode frequencies (Kim *et al.*, 2011). With frequency up-conversion techniques, the low-frequency ambient vibration can be transferred to high frequency vibrations, providing a frequency-robust energy harvesting solution for the low frequency oscillations (Wickenheiser and Garcia, 2010). Nonlinear energy harvesters (monostable or bistable) using magnetic interactions were also pervasively employed to broaden the operating bandwidth of vibration energy harvesters to adapt to environments with broadband or frequency-variant vibrations (Cottone *et al.*, 2009; Erturk and Inman, 2010; Tang and Yang, 2012).

The above mentioned techniques can be extended and modified to enhance the wind energy harvesting. For instance, Zhao *et al.* (2014a) used a 2DOF structure with magnetic interaction to enhance the output power of a galloping harvester in the low wind speed range, as shown in Figure 2.57. The low frequency oscillation of the outer frame excited the high resonance frequency of the inner beam via magnetic frequency up-conversion mechanism, improving the mechanical strain energy at low wind speeds.

Some other techniques have been proposed from the mechanical aspects to modify the structural designs so as to enhance the performance of aeroelastic energy

harvesters. These efforts include attaching the cylindrical bluff body to the beam instead of separating them to enhance the effects of vortex shedding (Akaydin et al., 2012), adding a movable mass to adjust the resonance frequency thus broadening the functional wind speed range of a harvester based on vortex-induced vibrations (Weinstein et al., 2012), integrating the base vibratory excitation with the aerodynamic forces for performance improvement of energy harvesters based on airfoil aeroelastic flutter (Bibo and Daqaq, 2013; Bibo et al., 2015), VIV (Dai et al., 2014) and galloping (Yan and Abdelkefi, 2014; Yan et al., 2014), reducing flutter speed and enhancing power output of airfoil-based piezoaeroelastic energy harvests by introducing free-play or cubic stiffness nonlinearities (Bibo et al., 2015), and adding a beam stiffener to amplify the electromechanical coupling thus increasing the power extraction efficiency of the aeroelastic energy harvesters (Zhao and Yang, 2015a).

2.4.2 Enhancement with sophisticated interface circuits

In the field of VPEH, many power conditioning circuit techniques have been developed to regulate and enhance power transfer from the piezoelectric materials to the terminal load or storage components, including the impedance adaptation (Ottman et al., 2002, 2003), synchronized switch harvesting on inductor (SSHI) (Guyomar et al., 2005; Lallart and Guyomar, 2008; Shu et al., 2007; Lien et al., 2010; Lefeuvre et al., 2006; Liang and Liao, 2009, 2012), synchronous charge extraction (SCE) (Lefeuvre et al., 2005, 2006, 2007; Tang and Yang, 2011), and energy storage circuits (Wickenheiser et al., 2010; Wu et al., 2009). However, very limited researches have been reported on the integration of advanced interfaces with aeroelastic energy harvesters to enhance their power output.

Enhancing power generation performance of wind energy harvesters with modified interface circuits has been considered by Taylor et al. (2001). They implemented a switched resonant-power converter, which was similar to a series SSHI yet without the full wave rectifier in an oscillating piezoelectric eel. Robbins et al. (2006) implemented a quasi-resonant rectifier in a flapping PVDF (similar to eel), and Bryant et al. (2013) employed an SSDCI circuit with a separate microcontroller

based peak detection system in a airfoil-based piezoelectric flutter energy harvester. De Marqui et al. (2011) used a resistive-inductive circuit to extract energy from a fluttering bimorph plate under cross flow condition, and showed that the power output was enhanced to about 20 times larger than the case with a pure resistor in the circuit at the short circuit flutter speed and short circuit flutter frequency.

Zhao et al. (2014b, 2015a) investigated the feasibility of employing a self-powered SCE interface to enhance the performance of a galloping-based piezoelectric energy harvester, as shown in Figure 2.80 depicting the equivalent circuit model (ECM) of harvester and the SCE diagram. Experimental and theoretical comparison of the performance of SCE with a standard circuit revealed three main advantages of SCE in galloping-based harvesters. Firstly, SCE eliminates the requirement of impedance matching and ensures the flexibility of adjusting the harvester for practical applications, since the output power from SCE is independent of electrical load. Secondly, 75% of piezoelectric materials can be saved by the SCE compared to the standard circuit. Thirdly, the SCE helps to alleviate the fatigue problem with a smaller transverse displacement during harvester operation.

Figure 2.80 Schematic of self-powered SCE circuit integrated with a galloping piezoelectric energy harvester (ECM diagram) (Zhao et al., 2015a)

Zhao and Yang (2015b) further proposed the analytical solutions of responses of a galloping-based piezoelectric energy harvester. Explicit expressions of power, voltage, displacement amplitude, optimal load and electromechanical coupling as well as cut-in wind speed for the simple AC, standard and SCE circuits were derived, which were validated with wind tunnel experiments and circuit simulation.

It was found that the three circuits generated the same maximum power but the SCE achieved it with the smallest coupling value. Moreover, the SCE was found to give the smallest displacement and highest cut-in wind speed, while the standard circuit was found to have the largest displacement and lowest cut-in speed. It was concluded that the SCE is suitable for the cases with small coupling and relative high wind speeds; while the AC and standard circuit are suitable for large coupling cases. With the AC and standard circuits, small loads are better for cases requiring high current, such as charging batteries, and large loads suits the conditions having high threshold voltages.

Subsequently, Zhao et al. (2015b) investigated the capability of enhancing power output of a galloping-based piezoelectric energy harvester with the SSHI interfaces. Both series SSHI (S-SSHI) and parallel SSHI (P-SSHI) were considered, as shown in Figure 2.81, of which the performances were compared with a standard circuit. Experimentally, it was found that the benefits of SSHI became more significant at high wind speeds. A power increase of 43.75% was obtained with the P-SSHI at 6m/s. However, the SSHI circuits gave lower output power than the standard circuit did in the low wind speed range. By comparing experimental measurements to theoretical predictions from ECM simulation, it was found that the S-SSHI suffered severe energy loss especially at high wind speeds.

Figure 2.81 Schematic of self-powered SSHI circuits (S-SSHI or P-SSHI depending on connection of S1) implemented to a galloping harvester (Zhao et al., 2015b)

2.5 Application of small-scale wind energy harvesting in self-powered wireless sensors

Many studies in vibration energy harvesting have investigated the feasibility of extracting energy from ambient vibrations to implement self-powered wireless sensors. Similarly, a main purpose of small-scale wind energy harvesting is to power the wireless sensors placed in an airflow-existing environment with the extracted flow power.

Flammini et al. (2010) demonstrated the viability of harvesting small-scale wind energy to power autonomous sensors in air ducts used for heating, ventilating, and air-conditioning (HVAC). To demonstrate the concept, a small-scale wind turbine with a commercial electromagnetic generator and commercial fan blades (six blades of 4cm) were employed as the wind energy harvester. The wind turbine was attached with the electronic circuit consisting of the autonomous sensor and the readout unit. Signals were transmitted through electromagnetic coupling at 125 kHz between the antenna of the transponder (U3280M) in the airflow-powered autonomous sensor and the transceiver (U2270B) in the readout unit. The airflow speed was measured by the autonomous sensor, by measuring the rotor frequency using the infrared Light Emitting Diode (LED) and an NPN silicon phototransistor (Optek OPB815-WZ), which were powered only during the measurement phase. It was shown experimentally that the system was able to work at wind speeds higher than 4 m/s, with comparable wind speed predictions to the readings from a reference flowmeter, confirming the feasibility of powering autonomous sensors for airspeed monitoring with airflow energy.

In a subsequent study, Sardini and Serpelloni (2011) extended the application of the integrated harvester and sensor to monitor the air temperature. A small-scale wind turbine consisting of a DC servomotor (1624T1,4G9 Faulhaber) of 32×32×22 mm and two blades of 65mm diameter, was attached with an autonomous sensing system consisting of a microcontroller, an integrated temperature sensor, and a radio-frequency transmitter, the sensing unit prototype, as shown in Figure 2.82. The system was able to transmit signal at wind speeds higher than 3m/s with a

433-MHz point-to-point communication to a receiver placed within 4~5m distance from the sensor with a time interval of 2s. It was concluded that the self-powered wireless sensor harvesting ambient airflow energy can be applied in environmental health monitoring applications.

Figure 2.82 Schematic of autonomous sensor integrated with an electromagnetic wind energy generator (Sardini and Serpelloni, 2011)

Along with the recently increasing desire on indoor microclimate control, Xiang et al. (2013) developed the first documented Trinity system, i.e., wind energy harvesting, synchronous duty-cycling and sensing, as a self-sustaining sensing system to monitor and control the wind speed at individual outlets of a HVAC system according to real-time population density. The schematic of the trinity is shown in Figure 2.83. The energy generated from a galloping-based energy harvester that consisted of a bimorph and a square sectioned bluff body was delivered to the power management module to power the sensor and charge the two thin-film batteries (if surplus energy was available). Low-power self calibration strategy and per-link synchronization were implemented for synchronous duty-cycling to ensure the receivers to wake up in time to receive data packets from the respective senders. The low duty-cycles ($<0.42\%$) were due to the fact that the energy harvested was not sufficient to continuously activate the sensing nodes. The wind speed was inferred by sampling the voltage of the harvester based on the measured relationship between voltage and wind speed, accomplished with an amplifier circuit consuming a low power more than $500\mu\text{W}$. The Trinity prototype successfully predicted agreed wind speeds with an anemometer from 3~6m/s at 16

HVAC outlets. It was concluded that the Trinity is a successful demonstration of a self-powered indoor sensing system given a carefully designed network operation mode.

Figure 2.83 Schematic of Trinity indoor sensing system (Xiang et al., 2013)

2.6 Chapter summary

This chapter reviews the fundamentals and the state-of-the-art techniques of energy harvesting, with a focus on small-scale piezoelectric energy harvesting from wind flows. With the advancement in electronic techniques, the generated power in milliwatt by harvesting energy from ambient wind flows is viable to power wireless sensor nodes. Typical wind flow speed in HVAC and outdoor natural wind in urban areas is 2~10m/s, which is the focused range for the reviewed small-scale wind energy harvesting techniques.

The modeling methods of the harvester system in the literature are generally classified into three categories: the mathematical modeling method, the equivalent circuit modeling method, and the computational fluid dynamics (CFD) method. Theoretical analyses with the mathematical models establish clear relationships between the multi-way coupling and final mechanical and electrical responses, enabling fast evaluation of the harvester's performance and efficient parameter optimization. Yet for the case with complex interface circuits for signal rectification and regulation, theoretical formulation is quite cumbersome due to the added complex coupling by nonlinear electronic components. In such a case, equivalent circuit modeling is greatly advantageous by representing the aerodynamic and

mechanical components with equivalent electronic components, and enabling system-level simulation with the overall system circuit model. Computational fluid dynamics gives visualized interactions between the flow domain and the harvester structure domain. However, the backward electromechanical coupling as well as the effects of external load or more complex interfaces on system responses are not able to be integrated and need to be considered separately. Future work on developing integrated multi-way coupling CFD model is desired.

Miniaturized windmills or wind turbines can generate a significant amount of power. Yet the biggest concern is that the rotary components are not desired for long-term use of such small sized devices. Besides the windmills and turbines, summaries of various energy harvesting devices based on VIV, galloping, flutter, wake galloping, TIV and other types reviewed in this paper are presented in Table 2.3~Table 2.9. Their merits or limitations and other information that the authors feel useful are also given in the tables. Power density per swept area/volume is calculated by dividing the maximum output power by the frontal area normal to the flow/device volume. Because the volume of some accessory components (e.g., the joints) and elements taking very small proportions of the whole volume (e.g., a short piezoelectric sheet attached to a long substrate cantilever) are ignored, the power density values should be considered the estimated upper bound for comparison purpose only. It should be noted that each technique investigated is suitable for a specific condition and has weakness in other conditions. One should choose a suitable technique or design according to the specific wind flow conditions, like whether the flow is smooth or turbulent, whether the flow speed is stable or frequently varies, and what the dominant wind speed range is, etc. This literature review is intended to provide some useful guidance for the researchers who are interested in small-scale wind energy harvesting. The development of integrated wind powered devices, like self-powered wireless sensors with generated power from small-scale wind energy harvesting, should be the focus of future research, leading the lab research on wind energy harvesting to real engineering applications like civil and infrastructure health monitoring, environmental monitoring or even an overall monitoring system in smart buildings and smart cities.

Table 2.3 Summary of various small-scale windmills and wind turbines investigated

Author	Transduction		Mechanical transfer	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per swept area (mW/cm ²)	Advantages/disadvantages
Rancourt et al. (2007)	Electromagnetic		-	-	-	130	11.8	4.2cm in dia.	9.38	High power generation efficiency at high wind speed; at low wind speed, efficiency decreased sharply due to the friction in the generator and the internal electric resistance
Howey et al. (2011)	Electromagnetic		-	3	-	4.3	10	3.2cm in dia.	0.535	Bearing loss and resistive generator loss limits the miniaturization of the turbine
Priya et al. (2005)	Piezoelectric		Contact via mechanical stopper	-	-	10.2	-	12 bimorphs in a circular array, each of 6×2×0.05cm ³	0.0902	Proves the feasibility of efficiently harvesting low speed wind energy using piezoelectric materials;
Priya (2005)	Piezoelectric		Contact via mechanical stopper	2.1	5.4	7.5	4.5	10 bimorphs in a circular array, each of 6×2×0.06cm ³	0.0663	Bimorphs are not vibrating in phase so the output has to be individually processed
Chen et al. (2006)	Piezoelectric		Contact via mechanical stopper	2.1	6.2	1.2	5.4	5.08×11.6×7.62cm ³	0.0134	Easy to fabricate; Space efficient with a rectangular-array arrangement of transducers; Combined circuit can be used because all the bimorphs are vibrating in phase; Power was much lower compared to the circular windmill.
Myers et al. (2007)	Piezoelectric		Contact via mechanical stopper	2.4	-	5	4.5	7.62×10.16×12.70 cm ³	0.0388	Captured wind energy is increased by employing three fan blades
Bressers et al. (2011)	Piezoelectric		Contact-less via magnetic interaction	0.9	-	1.2	4.0	16.51×16.51×22.8 6cm ³ ;	3.18×10 ⁻³	Minimizing the frictional loss by avoiding direct mechanical contact; Prolong the fatigue life of piezoelectric elements; Lowered down the cut-in wind speed
Karami et al. (2013)	Piezoelectric		Contact-less via magnetic interaction	2	-	4	10	8×8×17.5cm ³	0.0286	Introducing nonlinearity into the harvester; Utilizing both nonlinear parametric excitation and ordinary excitation, help to achieve low cut-in wind speed, high output power and large operational range of wind speed

Table 2.4 Summary of various VIV energy harvester devices investigated*¹

Author	Transduction	Bluff body shape	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (lock-in) (m/s)	Dimensions	Power density per volume (mW/cm ³)* ²	Advantages/disadvantages and other information
Allen and Smits (2001) (in water)	Piezoelectric	Plate	0.05 (water speed)	0.8 (water speed)	-	-	Bluff body, frontal dimension: 5.05cm & 3.81cm; Eel membrane, length: 45.7cm & 7.6cm		Investigated and confirmed the feasibility of harvesting fluid energy via VIV; Determines that optimal performance occurs at resonance condition
Taylor et al. (2001) (in water)	Piezoelectric	Plate	-	-	3V (peak voltage)	0.5 (water speed)	PVDF eel: 24cm×7.6cm×150 μm	1.10(V/cm ³)	
Robbins et al. (2006)	Piezoelectric	Cylinder	-	-	7.8	6.7	Flapping PVDF membrane: 25.4cm×17.78cm×457.2 μm	0.378	The use of windward bluff body and mass on the free end of the flapping piezoelement can enhance energy conversion; Experimentally proves the use of quasi-resonant rectifier can increase the efficiency by a factor of 2.3 compared to a standard full-wave rectifier; Theoretically confirms the use of AFC/MFC can increase the efficiency by a factor of 25 compared to PVDF
Pobering and Schwesinger (2008)	Piezoelectric	Polygon	4.5	45	0.108	45	Bluff body frontal dimension: 1.035cm Three identical cantilevers: 1.4×1.18×0.035cm ³	0.0817	The use of piezoelectric bimorph cantilever ensures only the first mode deformation to guarantee no charge cancellation on the surface; Theoretically proposes the optimal geometry of L/D=2.125; Adjacent cantilevers arrangement enhances output power
Pobering et al. (2009)	Piezoelectric	D-shape	15	-	1	40	Bluff body frontal dimension: 1.035cm Nine identical cantilevers (Series 1): 2.2×1.18×0.035cm ³	0.164	Experimentally validates the optimal geometry of L/D=2.125; Use of stapled piezoelectric layers enhance output power; Adjacent cantilevers arrangement can further lower the cut-in wind speed down to 8m/s
Akaydin et al. (2010a);	Piezoelectric	Cylinder	-	-	0.004	7.23	Bluff body: 3cm in dia., 1.2m in length	4.72×10 ⁻⁶	The driving mechanism of the beam's oscillation was discovered via CFD as the combined effect of the

Akaydin et al. (2010b)							Cantilever: $3 \times 1.6 \times 0.02 \text{ cm}^3$		overpressure resulted from the stagnation region and the suction of the core of another vortex on the opposite side; The optimal position of the upstream tip of the cantilever was found to along the centerline and at a distance of $x/D=2$; Non attachment of bluff body and cantilever results in very low output power
Akaydin et al. (2012)	Piezoelectric	Cylinder	-	-	0.1	1.192	Bluff body: 1.98cm in dia., 20.3cm in length Cantilever: $26.7 \times 3.25 \times 0.0635 \text{ cm}^3$	1.47×10^{-3}	Attachment of the cylinder on the cantilever tip and use of PZT instead of PVDF greatly enhanced the output power; Attachment of the cylinder on the cantilever tip reduces the resonance wind speed for maximum power.
Weinstein et al. (2012)	Piezoelectric	Cylinder	2	5 (5.5)	5	5.5	Bluff body: 2.5cm in dia., 11cm in length; Cantilever: $2.86 \times 0.63 \times 0.25 \text{ cm}^3$; Whole plane size: $22.5 \times 11 \text{ cm}^2$	0.0918	Broadening operational wind speed range, because the harvester resonance frequency thus the resonance wind speed can be tuned by adjusting positions of the added weight; Tuning mechanism is not automatic
Gao et al. (2013)	Piezoelectric	Cylinder	3.1	-	0.03	5 (turbulent flow speed)	Bluff body: 2.91cm in dia., 3.6cm in length Cantilever: $3.1 \times 1.0 \times 0.0202 \text{ cm}^3$	1.25×10^{-3}	Turbulent flow results in higher output power of harvester than laminar flow; Turbulence excitation is claimed to be the dominant driving mechanism of the harvester, vortex shedding excitation in the lock-in region gives add-on contribution; cantilever and cylinder are in parallel
Tam Nguyen et al. (2013)	Piezoelectric	Triangle			0.59×10^{-6}	20.7	Two identical bluff bodies: 0.425cm in base length, 0.218cm in altitude; PVDF film: $2.5 \times 1.3 \times 0.0205 \text{ cm}^3$	3.70×10^{-6}	Can be easily deployed in the pipelines, tire cavities or machinery by installing a diaphragm on the wall; Output power is relatively low compared to other devices; Methods of enhancing power are proposed, e.g., optimizing the blockage ratio, adjusting the diaphragm position, using material with high piezoelectric constants, etc.

*¹If more than one sized prototypes were investigated in the reference, the information of dimension and critical wind speeds listed in the table corresponds to the one giving maximum output power.

*²The device volume is approximated without considering the piezoelectric element volume, thus the power density calculated is the conservative estimates showing the upper bound.

Table 2.5 Summary of various galloping energy harvester devices investigated

Author	Transduction	Bluff body shape	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per volume (mW/cm ³)	Advantages/disadvantages and other information
Sirohi and Mahadik (2011)	Piezoelectric	Triangle	3.6	6.1	50	5.2	Bluff body: 4cm in side, 25.1cm in length Cantilever: 16.1×3.8×0.0635cm ³	0.281	High peak power is achieved, with high electromechanical coupling; Validates the feasibility of predicting galloping energy harvester's response with quasi--steady hypothesis; Abrupt decrease in output power at 6.1m/s is undesired.
Sirohi and Mahadik (2012)	Piezoelectric	D-shape	2.5	-	1.14	4.7	Bluff body: 3cm in dia., 23.5cm in length; Cantilever: 9×3.8×0.0635cm ³	0.0134	Output power continuously increases with wind speed, with no cut-out wind speed; Wide operational wind speed range; Requiring turbulent flow to function well, e.g., flow from a rotating fan, because D-shape cannot self-oscillate in laminar flow; Cantilever and bluff body prism are in parallel.
Zhao et al. (2012) ;Yang et al. (2013)	Piezoelectric	Square	2.5	-	8.4	8.0	Bluff body: 4×4×15cm ³ ; Cantilever: 15×3×0.06cm ³	0.0346	Experimentally determining for the first time that the square section is optimal for galloping energy harvesting compared to other shapes, giving the largest power and the lowest cut-in wind speed; High peak power is achieved, with high electromechanical coupling; Wide operational wind speed range.
Zhao et al. (2014a)	Piezoelectric	Square	1.0	-	4	5.0	Bluff body: 4×4×15cm ³ ; Cut-out cantilever: inner beam: 5.7×3×0.03cm ³ ; outer beam: 17.2×6.6×0.06cm ³ with cut out at the inner beam location	0.0162	Reducing the cut-in speed with the cut-out configuration; Enhancing output power in the low wind speed range; Output power is limited in high wind speeds
Zhao and Yang (2015a)	Piezoelectric	Square	2.0	-	12	8.0	Bluff body: 4×4×15cm ³ ; Cantilever: 27×3.4×0.06cm ³ ;	0.0455	Effective for all three types of harvesters based on galloping, vortex-induced vibration and flutter, with greatly enhanced output power; Displacement is not increased, thus not aggravating

							Stiffener: $8 \times 4.5 \times 0.5 \text{ cm}^3$		fatigue problem; Easy to implement; Cut-in wind speed is undesirably increased.
Zhao et al. (2014b, 2015a)	Piezoelectric	Square	3.5	-	1.14	5.0	Bluff body: $2 \times 2 \times 10 \text{ cm}^3$; Cantilever: $13 \times 2 \times 0.06 \text{ cm}^3$	0.0274	Employed synchronous charge extraction (SCE) eliminates the requirement of impedance matching and ensures the flexibility of adjusting the harvester for practical applications; Saving 75% of piezoelectric materials by the SCE compared to the standard circuit; Achieving smaller transverse displacement and alleviating fatigue problem with SCE.
Zhao et al. (2015b)	Piezoelectric	Square	3.0	-	3.25	7.0	Bluff body: $2 \times 2 \times 10 \text{ cm}^3$; Cantilever: $13 \times 2 \times 0.06 \text{ cm}^3$	0.0782	Parallel synchronized switching harvesting on inductor (SSHI) enhances output power compared to standard circuit; SSHI shows more significant benefits at high wind speeds, yet loses benefit at low wind speeds; Series SSHI endures unexpected energy losses in experiment.
Ewere and Wang (2014)	Piezoelectric	Square	2	-	13	8	Bluff body: $5 \times 5 \times 10 \text{ cm}^3$; Cantilever: $22.8 \times 4 \times 0.04 \text{ cm}^3$; Bump stop: gap size: 0.5cm, contact area: $1.27 \times 4 \text{ cm}^2$; location: 13cm along cantilever	0.0512	Incorporating an impact bump stop successfully relieves the fatigue problem; Achieving substantial 70% reduction in displacement amplitude with only 20% voltage reduction

Table 2.6 Summary of various flutter energy harvester devices investigated

Author	Transduction	Flutter instability	Flap at the tip	Cut-in wind speed (flutter speed) (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per volume (mW/cm ³)	Advantages/disadvantages and other information
Bryant and Garcia (2009); Bryant and Garcia (2011)	Piezoelectric	Modal convergence flutter	Airfoil, profile NACA0012	1.86	-	2.2	7.9	Airfoil: semichord 2.97cm, span 13.6cm; Cantilever: 25.4×2.54×0.0381cm ³	7.17×10 ⁻³	Semi-empirical model of the nonlinear electromechanical and aerodynamic system accurately predicted electrical and mechanical response; Successfully predicts the flutter boundary with one of the real parts of the first two eigenvalues turning positive and the two imaginary parts coalescing; Wide operational wind speed range; Subcritical Hopf bifurcation, a large initial disturbance is fundamental for system startup.
Bryant et al. (2012b)	Piezoelectric	Modal convergence flutter	Flat plate	Stiff host structure				Flat plate tip: chord 3cm, span 6cm, thickness 0.79mm; Cantilever: 7.6×2.5×0.0381cm ³	Stiff host structure 20.0 Compliant host structure 16.3	Compared to a stiff host structure, a compliant host structure reduces the cut-in wind speed, cut-in frequency and oscillation frequency; The peak power is shifted toward the lower wind speeds with the compliant host structure.
				17.3	29	43	26			
				Compliant host structure						
				15.2	29	35	25			
Bryant et al. (2014)	Piezoelectric	Modal convergence flutter	Flat plate	17.3	29	43	26	Flat plate tip: chord 3cm, span 6cm, thickness 0.79mm; Cantilever: 7.6×2.5×0.0381cm ³	20.0	Confirms the feasibility of using ambient flow energy harvesting to power aerodynamic control surfaces.
Erturk et al. (2010)	Piezoelectric	Modal convergence flutter	Airfoil	9.30	-	10.7	9.30	Airfoil: semichord 12.5cm, span 50cm; Cantilever: -	2.27×10 ⁻³	Effect of electromechanical coupling on flutter energy harvesting is analyzed; Found that the optimal load gave the maximum flutter speed due to the associated maximum shunt damping effect during power extraction

Sousa et al. (2011)	Piezoelectric	Modal convergence flutter	Airfoil	Linear configuration				Airfoil: semichord 12.5cm, span 50cm; Cantilever: -	Linear configuration	The free play nonlinearity reduces the cut-in wind speed and increased the output power; Theoretically determining that the hardening stiffness brings the response amplitude to acceptable levels and broadens the operational wind speed range.
				12.1		12	12.1		2.55×10^{-3}	
				With free play nonlinearity					With free play nonlinearity	
				10.0	en	27	10.0		5.73×10^{-3}	
Bibo and Daqaq (2013b)	Piezoelectric	Modal convergence flutter	Airfoil, profile NACA0012	2.3	-	0.138^{*1}	3 (With base acceleration 0.15m/s^2)	Airfoil: semichord 4.2cm, span 5.2cm; Cantilever: -	8.38×10^{-4}	Concurrent flow and base excitations enhances power generation performance; Concurrent excitations increases output power by 2.5 times below the flutter speed, and over 3 times above the flutter speed; Above the flutter speed, requiring careful adjustment because power is sensitive to base acceleration frequency,
Kwon (2010)	Piezoelectric	Modal convergence flutter	Flat plate tip; Whole device T-shape	4	-	4.0	15	Flat plate tip: $6.0 \times 3.0\text{cm}^2$; Cantilever: $10.0 \times 6.0 \times 0.02\text{cm}^3$	2.56	Simple T-shape structure, easy to fabricate; No rotating components; Wide operational wind speed range
Park et al. (2014)	Electromagnetic	Modal convergence flutter	Flat plate tip; Whole device T-shape	4	-	1.1	8	Flat plate tip: $3.0 \times 2.0\text{cm}^2$; Cantilever: $4.2 \times 3.0 \times 0.01016\text{cm}^3$	5.82	Determining that the onset of the harvester requires the load resistance to surpass the flutter onset resistance.
Li et al. (2011)	Piezoelectric	cross-flow flutter	-	4	-	0.615	8	adhered double-layer stalk: $7.2 \times 1.6 \times 0.041\text{cm}^3$	1.30	Cross flow configuration generated one order of magnitude more power than the parallel configuration; Having high power density per weight and per volume; Being robust, simple, miniature sized; Being easy to blend in urban and natural environments due to its “leaf” appearance
Deivasigamani et al. (2014)	Piezoelectric	cross-flow flutter	Triangle	-	-	0.0883	8	Isosceles triangle tip: 8cm in base, 8cm in height, 0.35mm in thickness;	0.0651	Determining that vertical stalk configuration is superior to the horizontal stalk with five times more output power.

								Stalk: 7.2×1.6×0.0205cm ³		
Humdinger	Electromagnetic	cross-flow flutter	-	2.7* ²	8.9* ³	≈9* ⁴	10* ⁵	Membrane: 12×0.7cm ² ; Casing: 13x3x2.5cm ³	0.923	Successfully powers wireless sensor nodes; Being compact and robust; Low cut-in wind speed and wide operational wind speed range.
Hobeck et al. (2014)	Piezoelectric	dual cantilever flutter	-	≈3	-	0.796	≈13	Two identical cantilevers: 14.6×2.54×0.0254 cm ³	0.422	Very wide operational wind speed range with efficient power generation. Generating a significant amount of power from 3 m/s to 15 m/s when gap is small.

*¹Calculated by $18\text{mw/g} \times (0.15\text{m/s}^2 / 9.8\text{m/s}^2) / 2$ from the information given by the authors of the reference.

*^{2,3}Transferred from 6mph and 20mph, respectively (http://www.humdingerwind.com/#/wi_micro/).

*^{4,5}Obtained from the figure in the datasheet of µmicroWindbelt (http://www.humdingerwind.com/pdf/microBelt_brief.pdf).

Table 2.7 Summary of various wake galloping energy harvester devices investigated

Author	Transduction	Prism shape	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per volume (mW/cm ³)	Advantages/disadvantages and other information
Jung and Lee (2011)	Electromagnetic	Circular cylinder	≈1.2	-	370.4	4.5	Two identical cylinders: 5cm in dia., 85cm in length; Spacing distance: 25cm	0.111	Determining the proper distance between the parallel cylinders for wake galloping energy harvesting, as 4-5 times the cylinder diameter; High output power; Wide operational wind speed range; Device volume is too big.
Abdelkefi et al. (2013d)	Piezoelectric	Windward: circular cylinder; Leeward: square cylinder	0.4	-	0.04~0.05	3.05	Circular cylinder: 1.25cm in dia., 27.15cm in length; Square cylinder: 1.28×1.28×26.67cm ³ ; Spacing distance: 24cm; Piezoelectric two identical cantilevers: 15.24×1.8×0.0305cm ³ .	5.72×10 ⁻⁴	Power from a galloping square cylinder was greatly enhanced by wake effects of an upstream circular cylinder; Operational wind speed range was widened by wake galloping; Diameter of the upstream cylinder and the spacing distance between two cylinders require careful adjustment.

Table 2.8 Summary of various TIV energy harvester devices investigated

Author	Transduction	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per volume (mW/cm ³)	Advantages/disadvantages and other information
Akaydin et al. (2010a)	Piezoelectric	$\approx 5^{*1}$	-	0.55×10^{-4}	11	Bluff body: 3cm in dia., 1.2m in length; Cantilever: $3 \times 1.6 \times 0.02 \text{ cm}^3$; Distance from the wall: 4cm	6.48×10^{-8}	Performance of harvester in turbulent boundary layer depends on the distance from the wall; dominant oscillation frequency was close to the beam resonance frequency.
Hobeck and Inman (2012a)	Piezoelectric	$9 \sim 10^{*2}$	-	4.0	11.5	Bluff body: $4.45 \times 4.45 \times 10.92 \text{ cm}^3$; Four identical cantilevers in an array: $101.60 \text{ mm} \times 25.40 \text{ mm} \times 101.60 \mu\text{m}$ steel substrate attached with $45.97 \text{ mm} \times 20.57 \text{ mm} \times 152.40 \mu\text{m}$ PZT	0.0184	The first TIV energy harvesting model with experimental validation; Being robust and survivable due to its inherent redundancy, with minor reduction in total power caused by one damaged element; Suitable for highly turbulent fluid flow environments like streams or ventilation systems

^{*1}Obtained from the information of Figure 11 of the reference.

^{*2}Obtained from the information of Figure 7(b) of the reference.

Table 2.9 Summary of other types of energy harvester devices investigated

Author	Transduction	Cut-in wind speed (m/s)	Cut-out wind speed (m/s)	Maximum power (mW)	Wind speed at max power (m/s)	Dimensions	Power density per volume (mW/cm ³)	Advantages/disadvantages and other information
Bibo <i>et al.</i> (2012)	Piezoelectric	$\approx 5.5^{*1}$	-	0.05	7.5	Cantilever: $5.8 \times 1.626 \times 0.038 \text{ cm}^3$; Chamber volume: 2300 cm^3	2.17×10^{-5}	Mimicking the basic physics of music-playing harmonicas; Using optimal chamber volume and decreasing aperture's width reduce the cut-in wind speed;
Ovejas and Cuadras (2011)	Piezoelectric	-	-	0.0002	12.3	Two identical bluff bodies: 0.5cm in dia., Piezoelectric film: $15.6 \text{ cm} \times 1.9 \text{ cm} \times 40 \mu\text{m}$	1.25×10^{-5}	Bluff body configuration outperformed the one fixed side and two fixed side configurations; Rotational turbulent flow from a dryer added to vortex shedding effects, giving higher electrical output than the laminar flow did.

^{*1}Obtained from the information of Figure 13(b) of the reference.

CHAPTER 3 COMPARATIVE STUDY OF TIP CROSS-SECTIONS FOR EFFICIENT GALLOPING ENERGY HARVESTING²

3.1 Introduction

Flow-induced vibrations due to vortex shedding, galloping, flutter and wake galloping can be harnessed using piezoelectric energy harvesters, as summarized in Section 2.3. Translational galloping is a self-excited aeroelastic phenomenon, giving rise to transverse oscillations normal to the direction of wind flows in structures with weak damping when wind velocity exceeds a critical value. It is a better choice to obtain structural vibrations for energy harvesting purpose compared to the vortex-induced vibrations and flutter, for its advantage of large vibration amplitude and the ability of oscillating in infinite range of wind velocities. Barrero-Gil *et al.* (2010) theoretically analyzed the potential use of transverse galloping to obtain energy using a one-degree-of-freedom (1DOF) model. Different galloping energy harvesters consisting of cantilevers with attached prisms of triangular and D-shape cross-sections have been reported by Sirohi and Mahadik (2011, 2012). Abdelkefi *et al.* (2013a) theoretically analyzed the influence of load resistance and bluff body cross-section geometry on the onset of galloping and harvested power. However, no experimental comparative study has been reported in the literature.

This chapter presents an experimental study on the influence of cross-section geometries (square, rectangles with distinct aspect ratios, equilateral triangle and D-section) of bluff bodies on galloping piezoelectric energy harvesting in the laminar flow condition. The flow condition is kept the same for all harvesters to ensure fair comparison. A lumped parameter 1DOF model integrating electromechanical and aerodynamic formulations is also established and validated.

² Content based on this chapter has been published as “Yang, Y., Zhao, L. and Tang, L. (2013). Comparative study of tip cross-sections for efficient galloping energy harvesting. *Applied Physics Letters*, 102(6), 064105.”

3.2 Proposed Galloping Energy Harvester Design

The proposed galloping energy harvester consists of a piezoelectric cantilever beam clamped at one end, and connected to a rigid prism with a specific cross section at the free end (Figure 3.1(a)). Translational galloping was first analyzed by Den Hartog (1956). The simplified 1DOF model comprises a bluff body M mounted on a spring K and a damping C as shown in Figure 3.1(b). Wind U flows from the right to the left, with the angle of attack α . The aerodynamic force F_z acting on the bluff body causes it to oscillate in the z direction. A criterion for galloping is identified as $\partial C_{F_z}/\partial \alpha > 0$, where C_{F_z} is the aerodynamic force coefficient (Païdoussis et al., 2011).

Figure 3.1 (a) Proposed galloping PEH and (b) Schematic of a bluff body subjected to wind flows

3.3 Experiment setup

Figure 3.2(a) shows the schematic of the proposed energy harvester. The fabricated prototype device and the experimental setup in the wind tunnel are shown in Figure 3.2(b). The joint of the connection between the beam and the tip is detachable. The dimensions of the aluminum cantilever are $L_b=150\text{mm}$, $B_b=30\text{mm}$, and $t_b=0.6\text{mm}$. Two piezoelectric sheets (DuraAct P-876.A12 from Physik Instrumente) are attached to the root of the cantilever beam on both sides, and connected in parallel. Each piezoelectric sheet has a length of 61mm, a width of 30mm, a thickness of 0.5mm, and a capacitance of 90nF. The cantilever and each piezoelectric sheet weigh 7.29g and 3.5g respectively. Considering different values of $\partial C_{F_z}/\partial \alpha$ of various cross sections of the tip body (Païdoussis et al., 2011), five cross sections

are chosen for comparison, including square, equilateral triangle, two rectangles with different aspect ratios, and D-section. The five tip bodies have equal length of 150mm and equal area of wind exposure as $40 \times 150 \text{ mm}^2$. Dimensions of the tip cross sections are shown in Table 3.1. For a fair comparison, the weights of the five tip bodies are all adjusted to 26.8g by attaching various steel masses onto their inner surfaces to achieve a constant fundamental frequency for the harvester with different tip bodies.

Figure 3.2 (a) Top and side views of proposed galloping energy harvester and (b) Experimental setup

Table 3.1 Dimensions of tip cross sections

Section shape
dimension h×d (mm)	40×40	40×60	40×26.7	40 (side)	40 (dia.)

3.4 Analytical Model

The analytical model of the proposed harvester should take into account the electromechanical coupling effect, as well as the interaction between the wind flow and the tip body. A coupled 1DOF model is developed to simulate the

electro-fluid-structural coupling behavior. The governing equations are written as

$$\begin{cases} M_{eff} \ddot{w} + C\dot{w} + Kw + \Theta V = F_Z \\ \frac{V}{R_L} + C_P \dot{V} - \Theta \dot{w} = 0 \end{cases} \quad (3.1)$$

where w , C and K are the tip displacement in the direction normal to the wind flow, the damping coefficient and the stiffness of the harvester, respectively; V and R_L are the generated voltage and applied load resistance, respectively; C_P is the total capacitance of the piezoelectric sheets in parallel connection; Θ is the electromechanical coupling coefficient; and M_{eff} is the effective mass expressed by $M_{eff} = (33/140)M_b + M_{tip}$ with M_b and M_{tip} being the mass of the cantilever and the tip body, respectively. The coupling coefficient Θ is obtained as

$$\Theta = \sqrt{(\omega_{noc}^2 - \omega_{nsc}^2) M_{eff} C_P} \quad (3.2)$$

where ω_{noc} and ω_{nsc} are the fundamental natural frequencies of the harvester for the open circuit and short circuit conditions, respectively.

The aerodynamic model established here is based on the quasi-steady hypothesis (Païdoussis et al., 2011). The aerodynamic force is given as

$$F_Z = \frac{1}{2} \rho_a h L_{tip} U^2 C_{F_z} \quad (3.3)$$

where ρ_a , h , L_{tip} and U are air density, frontal dimension facing the wind flow (Figure 3.2(a)), length of the tip body, and wind velocity, respectively. C_{F_z} can be expressed as a polynomial function of the angle of attack α :

$$\begin{cases} C_{F_z} = \sum_{i=1}^m A_i(\alpha)^i + \sum_{j=2}^k A_j(\alpha)^j \frac{\dot{w}}{|\dot{w}|} \\ \alpha = \frac{\dot{w}}{U} + w'(L_b) \end{cases} \quad (3.4)$$

where $\dot{w}$ is the vibration velocity of the tip body; A_i and A_j are the empirical coefficients; and $w'(L_b)$ denotes the rotation angle of the beam at the free end due to the translational motion. The first sum in the expression of C_{F_z} involves odd integers (i, m) and the second involves even integers (j, k).

3.5 Results and Discussion

3.5.1 Experimental Results

Figure 3.3 Experimental results for various cross sections: (a) Power versus load resistance for square-section tipped energy harvester; (b) Power versus load resistance at 6m/s; (c) Voltage versus wind velocity at 105kΩ; (d) Power versus wind velocity at 105kΩ.

The root of the beam is clamped using a metal support during the wind tunnel test. Wind velocity is measured using a pitot tube and a manometer. Voltage data is acquired through the NI 9229 DAQ module. The average output power is calculated by $P = V_{RMS}^2 / R_L$, where V_{RMS} is the Root Mean Square (RMS) voltage across R_L .

Figure 3.3(a) shows the output power measured over a range of load resistances at different wind velocities for the proposed harvester with a square-sectioned tip. The optimal load resistance is around 105kΩ and hardly changes with the wind velocity (in our testing range). Here, the optimal load resistance refers to the one when the output power is the maximum. For galloping energy harvesters with different tip bodies, the optimal R_L is obtained at a specific wind velocity and applied to all wind velocities. Figure 3.3(b) shows the measured output power versus load resistance at

a wind velocity of 6m/s for the five cross sections. It is observed that the optimal R_L for the five cross sections are almost the same, which are all around 105k Ω . This is reasonable. The oscillation frequency due to galloping is always consistent with the fundamental frequency of the harvester (Païdoussis et al., 2011), which is equal for the five harvesters since the tip masses are the same and the axial force on the beam due to wind is negligible. Hence, equal optimal R_L is achieved, which can be approximated by $R_L=1/\omega_n C_p$.

The measured voltage and output power at different wind velocities with optimal R_L are plotted in Figure 3.3(c) and 3.3(d), respectively. It is obvious that the square section generates the largest voltage and power, with a peak output power of 8.4mW measured at 8m/s, which is also higher than the peak power achieved in the literature (e.g. 4mW in Kwon (2010), 2.2mW in Bryant and Garcia (2011), 1.14mW in Sirohi and Mahadik (2012)). Moreover, the square section starts to oscillate at a wind velocity of 2.5m/s, which is lower than the other harvesters in our test and other previously reported ones (e.g. 4m/s in Kwon (2010), 8mph (3.576m/s) in Sirohi and Mahadik (2011)). This could be attributed to the largest value of $\partial C_{F_z}/\partial \alpha$ for the square section according to the criterion of galloping instability.²⁴ Power outputs for the rectangle (40×26.7mm²), D-section and equilateral triangle are quite low. The reason for the first two sections is the failure to meet the criterion of galloping instability in the laminar flow (which is the case for our test), as $\partial C_{F_z}/\partial \alpha$ equals to 0 for the rectangle (40×26.7mm²) and -0.1 for the D-section, respectively (Païdoussis et al., 2011). However, in the turbulent flow, the value for the D-section becomes positive (i.e. $\partial C_{F_z}/\partial \alpha=0.79$ for turbulence intensity Tu=11%), which enables the D-section to undergo self-excited oscillation as presented by Sirohi and Mahadik (2012). They blew the D-section with an axial fan that generated turbulent flow and obtained a stable output power around a wind velocity of 5.5mph. For equilateral triangle, the predicted cut-in wind velocity is around 8.5m/s, which is beyond our testing range. The minor peaks for the rectangle (40×26.7mm²) and equilateral triangle are caused by the resonance at the first torsional mode due to the vortex shedding (Figure 3(c) and 3(d)). Slight irregular torsional deflections were observed during the wind tunnel test when the wind velocity reached 5.5m/s. These

deflections increased to the maximum (resonance) at 6.5m/s and then disappeared over 7.5m/s, as the vortex shedding frequency increased with the wind velocity.

3.5.2 Simulation Results

The responses of the harvester with preferred square-sectioned tip can be predicted by simulation with the derived analytical model. The short circuit and open circuit natural frequencies for the proposed harvester are measured to be 6.84Hz and 6.89Hz under base excitations, respectively. The electromechanical coupling coefficient Θ is then determined to be 0.000373N/V by Equation (3.2). The damping ratio ζ , which is expressed as $\zeta = C/(2M_{eff}\omega_{nsc})$, is measured using the logarithmic decrement technique. The harvester is first excited at 6.84Hz at the unity RMS acceleration and then the shaker is suddenly shut down to obtain the attenuation curves of short circuit current, from which ζ is calculated to be 0.0148. The coefficients A_i and A_j for square cross section are shown in Table 3.2 (Païdoussis *et al.*, 2011).

Figure 3.4 Simulation results for square section: (a) Power versus wind velocity at 105kΩ; (b) Power versus load resistance at 6m/s; (c) Voltage versus wind velocity at 105kΩ; (d) Power versus wind velocity at 105kΩ. (a) uses initial ζ , (b~d) use modified ζ .

Table 3.2 Aerodynamic coefficients of polynomial expansion of C_{F_z}

Coef.	A ₁	A ₂	A ₃	A ₄	A ₅	A ₆	A ₇
Square	2.69	0	1.684×10^2	0	6.27×10^3	0	5.99×10^4

Using the above parameters, the output power versus wind velocity is calculated by numerical integration (Figure 3.4(a)). Obvious discrepancy between the calculated and measured power is observed. This should be attributed to the inaccurate damping ratio used in the simulation. In practice, the damping is amplitude-dependent, i.e., not constant. When the energy harvester is subjected to the low-speed wind flow in our test, the oscillation amplitude is much smaller than that in the logarithmic decrement test. For a more rational simulation, after a few trials, a modified damping ratio of 0.005 is used for the voltage and power prediction. The calculated output power versus load resistance agrees well with the measured results as shown in Figure 3.4(b). The calculated voltage and output power at 105k Ω are plotted in Figure 3.4(c) and 3.4(d), respectively. Again, the results match well with the measured data except for the hysteresis region between 2.8m/s and 3.8m/s. Theoretically, when wind velocity increases from zero to 3.8m/s, the power should increase along line 1, then jump to the upper line 2 increasing along it till 8m/s. When the wind velocity reduces from 8m/s to 2.8m/s, the power should decrease along line 2, and then drop to line 1 at 2.8m/s. However, the experiment does not capture this hysteresis phenomenon. Sources for these discrepancies may include the use of constant damping in simulation, the vortex shedding from the tip body and the difference between the operating condition such as Reynolds number in our test and that used in Païdoussis et al. (2011) to get the aerodynamic coefficients.

3.6 Chapter Summary

In summary, this chapter presents a comparative study of galloping energy harvester with different cross-section tips. Experimental results confirm the superiority of square cross-section tip over the others used in the literature. A peak output power

of 8.4mW is achieved which is sufficient to power small sensors. An analytical model is established and verified by experimental results. It is recommended that the tip of square section should be used for galloping-based small wind energy harvesting.

CHAPTER 4 COMPARISON OF MODELING METHODS AND PARAMETRIC STUDY FOR PIEZOELECTRIC WIND ENERGY HARVESTER³

4.1 Introduction

In order to comprehensively study the mutual coupling behavior and accurately predict the mechanical and electrical responses of a galloping piezoelectric energy harvester (GPEH), electromechanically coupled analytical modeling with the correct formulation of galloping-induced aerodynamic force is required. In the area of aerodynamics, many researchers have studied the effect of various parameters on the galloping stability of different structures, including the angle of attack, cross-section geometries, flow turbulence, Reynolds number, etc (Barrero-Gil et al., 2009; Alonso et al., 2007; Alonso and Meseguer, 2006; Luo et al., 1999; Païdoussis et al., 2011). For energy harvesting purpose, Barrero-Gil *et al.* (2010) theoretically analyzed the potential use of transverse galloping to harvest energy using a 1DOF model. No specific design of harvester device was proposed. Abdelkefi *et al.* (2012g) considered a GPEH with a square prism as the bluff body, and investigated the effects of Reynolds number and load resistance on the threshold of galloping and the level of harvested power. They further theoretically compared the performances of different cross-section geometries of the bluff body for GPEHs with linear and nonlinear analysis (Abdelkefi *et al.*, 2013a). A 1DOF model was employed for these two studies, with the value of electromechanical coupling manually specified. Yang *et al.* (2013) experimentally studied the influence of cross-section geometry on the performance of GPEHs, and validated the established 1DOF model with the experimental results. Sirohi and Mahadik (2011) studied a GPEH consisting of two piezoelectric cantilevers connected to a prism with equilateral triangular section. They also developed another GPEH using a composite

³ Content based on this chapter has been published as “Zhao, L., Tang, L. and Yang, Y. (2013). Comparison of modeling methods and parametric study for a piezoelectric wind energy harvester. *Smart Materials and Structures*, 22(12), 125003.”

piezoelectric cantilever connected in parallel to a prism with D-shaped section (Sirohi and Mahadik, 2012). For these two harvesters, an approximated distributed parameter model based on energy method (Rayleigh-Ritz type of discretization) was used. Abdelkefi *et al.* (2013b) employed the harvester design of Sirohi and Mahadik (2011) and further studied the influence of load resistance on the threshold of galloping and harvested power level with an Euler-Bernoulli distributed parameter model. Single mode (fundamental mode) was considered in their analysis. In all the analytical models mentioned above, the aerodynamic forces were formulated based on quasi-steady hypothesis (QSH). Each model method has its own merits, such as the simple form and ease of application of the 1DOF model, and the accuracy of the distributed parameter models. Although many studies have been conducted on the issues of the electromechanical models for vibration piezoelectric energy harvesters under base excitation (Erturk and Inman, 2008b, 2008c), more work need to done on the modeling and better designing of GPEH which incorporate the aerodynamic forces.

This chapter compares different modeling methods for GPEHs. The 1DOF model established in the preceding study (Yang et al., 2013), and the Euler-Bernoulli distributed parameter model using single mode and multi-modes are considered followed by experimental validation. The merits, disadvantages and applicability of these methods are discussed. Subsequently, employing the most effective model from the comparison, parametric study is performed to study the effects of load resistance, wind exposure area of the bluff body, mass of the bluff body and length of the piezoelectric sheets on the electroaeroelastic behavior (cut-in wind speed and power output level) of GPEHs. The results can be exploited for designing and optimizing GPEHs for better power output performance.

4.2 Mechanism of galloping

Translational galloping is a self-excited phenomenon giving rise to large amplitude oscillations of bluff body when subjected to wind flows. Assume that a bluff body is elastically mounted as shown in Figure 4.1. The governing equation of the galloping motion can be written as

$$M\ddot{w}(t) + C\dot{w}(t) + Kw(t) = F_z(t) \quad (4.1)$$

Figure 4.1 Schematic of a bluff body undergoing galloping

where w is the vertical position of the bluff body in z direction; M is the mass of the bluff body; C is the damping coefficient; K is the stiffness of the system and F_z is the aerodynamic force. The overdot denotes differentiation with respect to time t . F_z can be expressed as (Païdoussis et al., 2011)

$$F_z = \frac{1}{2} \rho_a S_{ip} U^2 C_{Fz} \quad (4.2)$$

where ρ_a is the air density; S_{ip} is the exposure area facing the flow; U is the wind speed and C_{Fz} is the total aerodynamic force coefficient in z direction. QSH is employed here, which considers that the aerodynamic force during galloping oscillation is equal to that when the bluff body is steady with the corresponding angle of attack. This hypothesis is applicable when the oscillation is slow enough, which requires that the characteristic timescale of the flow is much smaller than the characteristic timescale of the oscillation, corresponding to the character of galloping phenomenon. The larger the wind speed, the safer to use QSH. QSH has been confirmed to be able to successfully model galloping in many studies (Den Hartog, 1956; Barrero-Gil et al., 2009; Païdoussis et al., 2011). For a specific cross-section geometry, C_{Fz} is a function of the angle of attack α (Figure 4.1), and can be determined through experiments (Den Hartog, 1956; Barrero-Gil et al., 2009; Alonso et al., 2007; Alonso and Meseguer, 2006; Luo et al., 1999; Païdoussis et al., 2011). It is common to express C_{Fz} as a polynomial expansion as

$$C_{Fz} = \sum_{r=1,2,\dots} A_r (\alpha)^r \quad (4.3)$$

where A_r are empirical coefficients for the polynomial fitting (Barrero-Gil et al., 2010; Païdoussis et al., 2011). If the bluff body undergoes only translational oscillation without rotation, α can be expressed as

$$\alpha = \frac{\dot{w}}{U} \quad (4.4)$$

Substituting Equations (4.2-4.4) into Equation (4.1), and dividing both sides by M yields

$$\ddot{w}(t) + \left(2\zeta\omega_n - \frac{1}{2M}\rho_a S_{tip} U \left[\sum_{r=1,2,\dots} A_r \left(\frac{\dot{w}}{U} \right)^{r-1} \right] \right) \dot{w} + \omega_n^2 w = 0 \quad (4.5)$$

where ζ is the damping ratio and ω_n is the natural frequency of the system. The aerodynamic force can be considered as an effective nonlinear damping as shown in Equation (4.5), rendering the galloping “self-excited”. The criterion for galloping instability, which is identified by Den Hartog (1956), is expressed as

$$\frac{\partial C_{Fz}}{\partial \alpha} > 0 \quad (4.6)$$

The criterion requires $A_1 > 0$. For arbitrarily small $\dot{w}$, the system is controlled by the linear damping $2\zeta\omega_n - \frac{1}{2M}\rho_a S_{tip} U A_1$, which is positive, thus the oscillations will be damped to the zero equilibrium. When U increases and exceeds a certain value, the linear damping becomes negative, giving rise to galloping oscillations of the bluff body (Hopf bifurcation). When $\dot{w}$ is large enough due to the increasing oscillation amplitude, the nonlinear damping $\frac{1}{2M}\rho_a S_{tip} U \left[\sum_{r=2,\dots} A_r \left(\frac{\dot{w}}{U} \right)^{r-1} \right]$ should be taken into account which makes the overall damping non-negative. Limit cycle oscillation will occur when the damping reaches zero. Due to the self-excited and self-limiting characteristics of galloping, it is a prospective energy source for energy harvesting.

4.3 Comparison of modelling methods for galloping piezoelectric energy harvester

A typical GPEH is usually designed as a piezoelectric cantilever attached with a bluff body at the free end, as shown in Figure 4.2. The bluff body, which is with a specific cross section, oscillates in the direction normal to the incoming flow due to galloping. Two piezoelectric sheets are bonded to each side of the substrate beam, generating electricity from the mechanical strain which is developed due to the bluff

body oscillation. The analytical model of a GPEH should consider both the electromechanical coupling effect and the aerodynamic force acting on the bluff body. Commonly used models for GPEHs include the 1DOF model, approximate distributed parameter model with Rayleigh-Ritz type of discretization, and Euler-Bernoulli distributed parameter model with exact analytical mode shapes. The main difference between these models lies in the representation of the electromechanical coupling term. This section will compare the merits, disadvantages and applicabilities of the lumped parameter and distributed parameter models. The aerodynamic forces are all formulated based on quasi-steady hypothesis, although their representation formulas are various due to the different mechanical parameters employed in the corresponding electromechanical equations. The power storage technique is not considered in this chapter, so the simple electric circuit only consists of an external resistive load R_L .

Figure 4.2 Schematic of a typical GPEH

4.3.1 1DOF model

In the preceding study, a simple 1DOF model was established to simulate the electroaeroelastic behavior of a GPEH (Yang et al., 2013). In that model, the harvester was considered to oscillate close to the fundamental frequency, which was confirmed with visual observation during the experiment. The governing equations are given as

$$\begin{cases} M_{eff}\ddot{w}(L,t) + C_{eff}\dot{w}(L,t) + K_{eff}w(L,t) + \Theta V(t) = F_z(t) = \frac{1}{2}\rho_a h l_{tip} U^2 C_{Fz} \\ \frac{V(t)}{R_L} + C_p \dot{V}(t) - \Theta \dot{w}(L,t) = 0 \end{cases} \quad (4.7)$$

where $w(L,t)$ is the displacement of the bluff body in the direction normal to the wind flow; C_{eff} and K_{eff} are the effective damping and stiffness of the harvester; $V(t)$ is the generated voltage across R_L ; C_p is the total capacitance of two piezoelectric sheets with parallel connection; Θ is the electromechanical coupling term; h and l_{tip} are the frontal dimension and length of the bluff body, the product of which equals to S_{tip} ; and M_{eff} is the effective mass approximated as $M_{eff}=33/140M_b+M_{tip}$, where M_b and M_{tip} are the mass of the cantilever and the bluff body. Θ is determined easily though experiments by

$$\Theta = \sqrt{(\omega_{noc}^2 - \omega_{nsc}^2)M_{eff}C_p} \quad (4.8)$$

where ω_{noc} and ω_{nsc} are the open circuit and short circuit resonant frequencies of the harvester. C_{Fz} is expressed as in Equation (4.3), and the attack angle is modified to $\alpha = \dot{w}(L,t)/U + w'(L,t)$, where $w'(L,t)$ is the rotation angle at the free end due to the deflection of the beam, approximated as $w'(L,t) = 3w(L,t)/2L$.

By defining a state vector $\mathbf{X}$:

$$\mathbf{X} = \begin{Bmatrix} X_1 \\ X_2 \\ X_3 \end{Bmatrix} = \begin{Bmatrix} w(L,t) \\ \dot{w}(L,t) \\ V(t) \end{Bmatrix} \quad (4.9)$$

the governing equations can be written in the state space form as

$$\dot{\mathbf{X}} = \begin{Bmatrix} \dot{w}(L,t) \\ \ddot{w}(L,t) \\ \dot{V}(t) \end{Bmatrix} = \begin{Bmatrix} X_2 \\ -2\zeta\omega_n X_2 - \omega_n^2 X_1 - \frac{\Theta}{M_{eff}} X_3 + \frac{\rho_a h l_{tip} U^2}{2M_{eff}} \left[\sum_{r=1,2,\dots} A_r \left(\frac{X_2}{U} + \frac{3X_1}{2L} \right)^r \right] \\ \frac{\Theta}{C_p} X_2 - \frac{X_3}{R_L C_p} \end{Bmatrix} \quad (4.10)$$

where $2\zeta\omega_n = C_{eff}/M_{eff}$ and $\omega_n = \omega_{nsc}$. Equation (4.10) can then be numerically solved in MATLAB using the solver like ode45 to determine the vibration response of the beam, cut-in wind speed, and generated voltage across R_L . The average power P_{ave} is related with the root mean square (RMS) voltage V_{RMS} as $P_{ave} = V_{RMS}^2/R_L$.

4.3.2 Euler-Bernoulli distributed parameter model

Figure 4.3 (a) Top view of considered GPEH and (b) cross-section of composite beam for $x_1 < x < x_2$.

Distributed parameter electromechanical models based on Euler-Bernoulli beam theory have been studied a lot for vibration piezoelectric energy harvesters with base-excitations (Erturk and Inman, 2008a, 2008b, 2008c). For a GPEH, the electromechanical equations are developed based on similar assumptions: (a) The Euler-Bernoulli beam assumptions are applied to the composite beam; (b) The perfectly conductive electrodes fully cover the top and bottom surfaces of the piezoelectric sheet inducing uniform electric field through the thickness; (c) Only the z -direction vibration (transverse vibration) is considered; (d) The damping mechanisms including the internal strain rate damping and external air damping satisfy the proportional damping criterion. The schematics of the GPEH showing the coordinate directions and the cross-section of the composite beam are presented in Figure 4.3. We consider the electrically parallel connection for the two piezoelectric sheets. The coupled governing equations for the GPEH can be written as (Erturk and Inman, 2008a; Abdelkefi et al., 2013b)

$$\begin{cases} \frac{\partial^2}{\partial x^2} \left[YI(x) \frac{\partial^2 w(x,t)}{\partial x^2} \right] + c_s I(x) \frac{\partial^5 w(x,t)}{\partial x^4 \partial t} + c_a \frac{\partial w(x,t)}{\partial t} + m(x) \frac{\partial^2 w(x,t)}{\partial t^2} \\ + \theta V(t) \left[\frac{d\delta(x-x_1)}{dx} - \frac{d\delta(x-x_2)}{dx} \right] = F_z \delta(x-L) \\ \frac{V(t)}{R_L} + C_p \frac{dV(t)}{dt} - \theta \int_{x_1}^{x_2} \frac{\partial^3 w(x,t)}{\partial x^2 \partial t} dx = 0 \end{cases} \quad (4.11)$$

where $w(x, t)$ is the transverse deflection of the beam in z direction; $\delta(x)$ is the Dirac delta function; c_s and c_a are the coefficients of strain rate damping and air damping; $m(x)$ is the distributed mass of the beam, being expressed as $m(x) = \rho_s h_s b_s$ for $0 < x < x_1$ and $x_2 < x < L$, and $m(x) = \rho_s h_s b_s + 2\rho_p h_p b_p$ for $x_1 < x < x_2$, where ρ_s , h_s and b_s are the mass density, thickness and width of the substrate, respectively, ρ_p , h_p and b_p are the

corresponding terms of the piezoelectric sheet, and x_1 and x_2 are respectively the starting and ending positions of the piezoelectric sheets along the beam; $YI(x)$ is the bending stiffness of the composite beam expressed as $YI(x) = Y_s b_s h_s^3 / 12$ for $0 < x < x_1$ and $x_2 < x < L$, and $YI(x) = Y_s b_s h_s^3 / 12 + 2Y_p b_p [(h_p + h_s/2)^3 - h_s^3 / 8] / 3$ for $x_1 < x < x_2$, where Y_s and Y_p are the Young's modulus of the substrate and piezoelectric material, respectively; θ is the piezoelectric coupling term given by $\theta = -Y_p d_{31} b_p (h_p + h_s)$, where d_{31} is the piezoelectric constant; and C_p is the total capacitance as in the 1DOF model, being given by $C_p = 2\epsilon_{33}^s b_p (x_2 - x_1) / h_p$ ($(x_2 - x_1)$ equals to the length of the piezoelectric sheets L_p), where ϵ_{33}^s is the permittivity at constant strain. As in the 1DOF model, F_z is the aerodynamic force due to galloping.

The transverse deflection $w(x, t)$ can be represented as

$$w(x, t) = \sum_{r=1}^{\infty} \phi_r(x) \eta_r(t) \quad (4.12)$$

where $\phi_r(x)$ is the mass normalized mode shape for the r^{th} mode of the corresponding undamped free vibration problem; and $\eta_r(t)$ is the modal coordinate. Erturk and Inman (2008a) presented the procedure for the exact analytical solution of $\phi_r(x)$ for the cantilever fully and uniformly covered with piezoelectric materials. As for the cantilever beam which is partially covered by piezoelectric sheets (the case for our prototype in the experiment), Abdelkefi *et al.* (2013b) developed the Euler-Bernoulli distributed parameter model by deriving the exact analytical mode shape function to analyze the electroaeroelastic behavior of a GPEH. Yet such analytical solution for the segmented mode shapes is quite cumbersome. Single mode (fundamental mode only) was considered in their analysis. In this chapter, we obtain $\phi_r(x)$ and $\phi_r'(x)$ easily by finite element method in Matlab. Hermite cubic functions are employed as the interpolation functions for the beam element.

F_z is still in the form as in the 1DOF model (Equation (4.7)), but the attack angel is here expressed in modal coordinates:

$$\alpha = \frac{\sum_{r=1}^{\infty} \phi_r(L) \eta_r(t)}{U} + \sum_{r=1}^{\infty} \phi_r'(L) \eta_r(t) \quad (4.13)$$

Subsequently, introducing Equations (4.12) and (4.13) into Equation (4.11) yields the coupled governing equations in modal coordinates:

$$\begin{cases} \ddot{\eta}_r(t) + 2\zeta_r \omega_r \dot{\eta}_r(t) + \omega_r^2 \eta_r(t) + \chi_r V(t) = \phi_r(L) \times \frac{1}{2} \rho_a h l_{ip} U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\sum_{r=1}^{\infty} \phi_r(L) \dot{\eta}_r(t)}{U} + \sum_{r=1}^{\infty} \phi_r'(L) \eta_r(t) \right]^i \\ \frac{V(t)}{R_L} + C_p \dot{V}(t) - \sum_{r=1}^{\infty} \chi_r \dot{\eta}_r(t) = 0 \end{cases} \quad (4.14)$$

where ζ_r is the modal mechanical damping; ω_r is the undamped natural frequency of the r th vibration mode; and χ_r is the modal electromechanical coupling term written as $\chi_r = \theta[\phi_r'(x_2) - \phi_r'(x_1)]$. Before proceeding to solve Equation (4.14), a short explanation of how to evaluate ζ_r is presented here. As mentioned in assumption (d), the mechanical damping which includes the internal strain rate damping and external air damping is treated with the proportional damping (Rayleigh damping) assumption. For a uniform piezoelectric beam of which the flexural rigidity YI and mass per unit length m are both uniformly distributed (Erturk and Inman, 2008a, 2008b, 2008c), ζ_r can be simply expressed as $\zeta_r = c_s I \omega_r / 2YI + c_a / 2m \omega_r$. Once the damping ratios for two separate vibration modes are known, the two unknown constant damping coefficients c_s and c_a can be mathematically calculated. Yet for non-uniform piezoelectric beam which is partially covered by piezoelectric sheets as in our case, c_s and c_a are no longer constant along the beam length. Whereas, we can still assume that $c_s I(x)$ and $c_a(x)$ are stiffness proportional and mass proportional, respectively, and obtain two constant values of $(c_s I / YI)_{ave}$ and $(c_a / m)_{ave}$. However, to avoid the calculation of these coefficients as well as the inaccuracy caused by the proportional damping assumption, we can obtain ζ_r experimentally using the Frequency Response Function (FRF) obtained in base-vibration condition (Meirovitch, 2001) and proceed to numerical solution of Equation (4.14) for the mechanical and electric responses of the GPEH.

Assume the first n^{th} modes are considered. Again, we introduce the state vector:

$$\mathbf{X} = \begin{Bmatrix} X_1 \\ X_2 \\ \vdots \\ X_{2r-1} \\ X_{2r} \\ \vdots \\ X_{2n+1} \end{Bmatrix} = \begin{Bmatrix} \eta_1(t) \\ \dot{\eta}_1(t) \\ \vdots \\ \eta_r(t) \\ \dot{\eta}_r(t) \\ \vdots \\ V(t) \end{Bmatrix} \quad (r = 1, 2 \dots n) \quad (4.15)$$

The governing equations can then be written in the state space form as

$$\dot{\mathbf{X}} = \begin{Bmatrix} \dot{\eta}_1(t) \\ \ddot{\eta}_1(t) \\ \vdots \\ \dot{\eta}_r(t) \\ \ddot{\eta}_r(t) \\ \vdots \\ \dot{V}(t) \end{Bmatrix} = \begin{Bmatrix} -2\zeta_1\omega_1 X_2 - \omega_1^2 X_1 - \chi_1 X_{2n+1} + \phi_1(L) \times \frac{\rho_a h l_{tip} U^2}{2} \left[\sum_{i=1,2,\dots} A_i \left(\frac{\phi_1(L) X_2}{U} + \phi_1'(L) X_1 \right)^i \right] \\ \vdots \\ -2\zeta_r\omega_r X_{2r} - \omega_r^2 X_{2r-1} - \chi_r X_{2n+1} + \phi_r(L) \times \frac{\rho_a h l_{tip} U^2}{2} \left[\sum_{i=1,2,\dots} A_i \left(\frac{\phi_r(L) X_{2r}}{U} + \phi_r'(L) X_{2r-1} \right)^i \right] \\ \vdots \\ \sum_{r=1}^n \frac{\chi_r X_{2r}}{C_p} - \frac{X_{2n+1}}{R_L C_p} \end{Bmatrix} \quad (4.16)$$

Like the 1DOF model, Equation (4.16) can be solved in MATLAB using ode45. Two cases with different numbers of modes employed will be considered in the following model comparison section. Firstly, we only consider the fundamental mode by letting n equal to 1 in Equation (4.16), since a GPEH is oscillating near its fundamental frequency (Sirohi J and Mahadik, 2011, 2012). Secondly, in order to get a more accurate expression for the galloping aerodynamic force, we take into account the first three modes ($n=3$) to investigate how the higher modes influence the overall responses of a GPEH.

4.4 Model comparison based on experimental validation

4.4.1 Experimental setup

Figure 4.4 (a) Fabricated prototype and installation in wind tunnel. (b) Experimental setup

Table 4.1 Parameters of cantilever beam and piezoelectric sheets (Yang et al., 2013)

Properties	Beam	Piezoelectric sheets	Properties	Beam	Piezoelectric sheets
Length (mm)	150	61	Mass Density (kg m ⁻³)	2700	3825
Width (mm)	30	30	Capacitance (nF)	---	90
Thickness (mm)	0.6	0.5	Young's Modulus (GPa)	69	23.3
Material	Aluminum	DuraAct P-876.A12	Piezoelectric Constant d_{31} (pm V ⁻¹)	---	-174

Table 4.2 Parameters of bluff body for three configurations

Properties	Configuration 1	Configuration 2	Configuration 3
Cross-section	square	square	square
Section dimensions (mm ²)	40×40	40×40	40×40

Length l_{tip} (mm)	150	150	100
Mass M_{tip} (kg)	0.0268	0.0228	0.0268

A prototype device is fabricated and tested in the wind tunnel following the same experimental setup procedure with the preceding study (Yang et al., 2013), and the measured results are presented to validate the aforementioned 1DOF and distributed parameter models. The effectiveness of the models are compared based on the agreement between their predictions and experiments. The cantilever beam is the same with the one in Yang et al., (2013), which is composed of a aluminium substrate with two piezoelectric sheets (DuraAct P-876.A12 from Physik Instrumente) bonded to each side of the root, being connected in parallel with the total capacitance $C_p=180\text{nF}$. A metal support is employed to fix the root of the cantilever in the wind tunnel. A bluff body is attached to the free end. According to the preceding experimental study on the cross-section geometry of the bluff body (Yang et al., 2013), a square section is employed for its best performance over other geometries in the laminar flow with the lowest cut-in wind speed and the largest output power. Three configurations are tested in the wind tunnel with different bluff body properties for adequate model comparisons. The properties of the beam and piezoelectric sheets are listed in Table 4.1, while those of the bluff body are shown in Table 4.2. The prototype and the installation in the wind tunnel are shown in Figure 4.4(a). Prior to the wind tunnel test, some parameters like ζ , ω_{nsc} and ω_{noc} need to be measured under base excitations. The damping ratio ζ for the 1DOF model is the same as ζ_I in the distributed parameter model. For Configuration 1, $\zeta_I=0.005$, $\zeta_2=0.015$ and $\zeta_3=0.034$; for Configuration 2, $\zeta_I=0.008$, $\zeta_2=0.019$ and $\zeta_3=0.041$; for Configuration 3, $\zeta_I=0.004$, $\zeta_2=0.012$, $\zeta_3=0.031$. During the wind tunnel test, the wind speed is measured by a pitot tube anemometer, and the voltage across the load resistance is measured by the NI 9229 DAQ module. Overall experimental setup is presented in Figure 4.4(b).

4.4.2 Model comparison

Figure 4.5 (a) Measured and predicted average power P_{ave} versus load resistance R_L for configuration 1 at $U=4, 5$ and 6 m/s . (b) Measured and predicted average power P_{ave} versus wind speed U for three configurations at the respective optimum R_L .

The measured and predicted values of average power P_{ave} versus load resistance R_L are first compared in Figure 4.5(a). The data presented are obtained for configuration 1 at three different wind speeds. As shown in the picture, for $U=4, 5$ and 6 m/s , P_{ave} initially increases with R_L until reaches an optimum value, then decreases when R_L continues growing. The optimum R_L barely changes for the three wind speeds, which are all around $105\sim120\text{ k}\Omega$. All the three models can capture this trend well, yet with some discrepancies. Also, both the single mode distributed parameter model (shorted for Distri. (mode 1) here and hereafter) and multiple mode distributed parameter model (shorted for Distri. (mode 1-3) here and hereafter) predict a shallow valley around the optimum R_L for $U=4\text{ m/s}$, which is an important aspect for the influence of R_L on P_{ave} and will be addressed in the next section.

Moreover, no obvious difference is observed between the predictions of Distri. (mode 1) and Distri. (mode 1-3). Figure 4.5(b) shows the measured and predicted P_{ave} versus wind speed U for the three different GPEH configurations at $R_L=105\text{ k}\Omega$. It is noted that P_{ave} increases monotonically with U . Again, all the three models can predict consistent results with the experiments. As for the cut-in wind speed U_{cr} , the 1DOF model predicts better results for all the three configurations, while as for the harvested power level, no obvious superiority is observed for one model over the others since small discrepancies exist for all the three models. The main source of the error is probably the unavoidable small turbulence in the wind tunnel which disturbs the bluff body to start oscillating before the wind speed reaches the real cut-in speed. In general, the three models can successfully predict the electroaeroelastic behavior with good agreement with the experiments. The 1DOF model is advantageous for its simplicity and ease of obtaining the coupling coefficient for a fabricated prototype, while the distributed models own their merits in the better representation of the aerodynamic force and ease for parametric study. In the following study about the effects of different parameters on the output power, Distri. (mode 1) is employed to obtain the simulation results.

4.5 Parametric study using distributed parameter model (1st mode)

In this section, a parametric study is presented in order to better understand the electroaeroelastic behavior of the GPEH. The effects of load resistance R_L , wind exposure area S_{tip} , mass of the bluff body M_{tip} , and length of the piezoelectric sheets L_p on the cut-in wind speed as well as the output power level are investigated.

4.5.1 Effects of the load resistance R_L

Figure 4.6(a) shows the variation of the average output power P_{ave} with the wind speed U for different load resistance R_L . Here, Configuration 1 is considered and its properties are shown in Table 4.1 and Table 4.2. It can be seen that the growth rate of P_{ave} is greatly affected by R_L . The growth rate of P_{ave} firstly increases with R_L until $105\text{k}\Omega$, then decreases when R_L is further increasing. An enlarged view is provided to show this trend more clearly around the threshold of galloping. It is

noted that the value of R_L which owns the largest growth rate of P_{ave} also gives the largest cut-in wind speed U_{cr} . The variation of U_{cr} with R_L is presented in Figure 4.6(b). As can be seen from this curve, U_{cr} increases with R_L up to the maximum value then decreases with R_L . As for the harvested power level, the largest power is obtained with R_L around 105k Ω for $U > 4$ m/s. Features of the optimum R_L can be seen from Figure 4.6(c) which displays the variation of P_{ave} with R_L at different wind speeds. For $U = 2$ m/s, P_{ave} is zero between 20 and 800k Ω since this speed is lower than the respective U_{cr} for these R_L values (Figure 4.6(b)). When U is slightly larger than U_{cr} (i.e., $U = 3$ and 4m/s), a shallow valley exists around 105k Ω . When U continues growing (i.e., $U = 5 \sim 8$ m/s), there is a peak for P_{ave} between 100 and 120k Ω , where lies the optimum $R_L = 105$ k Ω . It is worth mentioning that R_L for the valley and peak of P_{ave} overlap with each other, corresponding to that with the largest U_{cr} . Here onwards, we regard R_L as the optimum one if the largest U_{cr} and power growth rate are achieved in the response of P_{ave} versus U (i.e., the curve for $R_L = 105$ k Ω in Figure 4.6(a)). At the optimum R_L , either a valley or a peak appears in the response of P_{ave} versus R_L at a specific U .

Figure 4.6 (a) Variation of average power P_{ave} with wind speed U for different load resistance. (b) Variation of average power P_{ave} with load resistance R_L for different wind speeds. (c) Variation of cut-in wind speed U_{cr} with load resistance.

4.5.2 Effects of the wind exposure area of the bluff body S_{tip}

Figure 4.7 shows the effects of the wind exposure area of the bluff body S_{tip} ($h \times l_{tip}$) on the electroaeroelastic behavior of the GPEH. Harvester properties are the same with those listed in Table 4.1 and Table 4.2 (Configuration 1) except the length of the bluff body l_{tip} . Figure 4.7(a) is obtained with $U = 6\text{m/s}$. It is determined that the optimum R_L for the various S_{tip} are equal to each other, which is reasonable since the natural frequencies are the same with equal M_{tip} . The variation of P_{ave} with U at the optimum R_L for different S_{tip} is shown in Figure 4.7(b). With increasing S_{tip} , the cut-in wind speed decreases while the harvested power increases, both beneficial for harvesting the wind power. This can be expected since the increasing wind exposure

area results in an increase in the aerodynamic force. Moreover, the growth rate of power is not much affected by S_{tip} .

Figure 4.7 Variation of (a) average power P_{ave} with load resistance R_L at 6m/s and (b) average power P_{ave} with wind speed U at optimum R_L for different wind exposure areas of bluff body S_{tip} .

4.5.3 Effects of the mass of the bluff body M_{tip} (or fundamental frequency of the GPEH)

Figure 4.8 shows the effects of the mass of the bluff body M_{tip} on the electroaeroelastic behavior of the GPEH. Harvester properties are the same with those list in Table 4.1 and Table 4.2 (Configuration 1) except M_{tip} . The most obvious change due to varying M_{tip} is in the fundamental frequency of the harvester as indicated in the legend. Figure 4.8(a) shows the variation of P_{ave} with R_L for $U=6m/s$, from which the respective optimum R_L can be obtained for each M_{tip} . It is noted that the optimum R_L varies with different M_{tip} (i.e., different fundamental frequency ω_1). With the respective optimum R_L , the variation of P_{ave} with U is displayed in Figure 4.8(b). As can be seen from this figure, the cut-in wind speed increases and the harvested power decreases with the increasing M_{tip} (or the decreasing of ω_1). Moreover, the growth rate of the harvested power also decreases with the increasing M_{tip} . It can be summarized that for the GPEH with a specific

piezoelectric cantilever, reducing M_{tip} can achieve a better performance on extracting power from the wind.

Figure 4.8 Variation of (a) average power P_{ave} with load resistance R_L at 6m/s and (b) average power P_{ave} with wind speed U at respective optimum R_L for different masses of bluff body M_{tip} .

4.5.4 Effects of the length of the piezoelectric sheets L_p

Figure 4.9 shows the effects of the lengths of piezoelectric sheets L_p on the electroaeroelastic behavior of the GPEH. Harvester properties are the same with those list in Table 4.1 and Table 4.2 (Configuration 1) except L_p and C_p since the latter one varies linearly with the former one when other parameters are kept unchanged. The respective optimum R_L is determined from Figure 4.9(a), which varies with L_p because of the change in the total stiffness of the composite beam. Figure 4.9(b) shows the variation of P_{ave} with U at the respective optimum R_L . The cut-in wind speed as well as the growth rate of the harvested power increases with the increasing L_p . The largest L_p can extract the highest power at relatively high wind speeds ($>12\text{m/s}$). A useful factor to evaluate the performance of the GPEH is the power density which is calculated as power per piezoelectric volume in this chapter. Figure 4.9(c) shows the power density versus the wind speed for different L_p . It is noted that the power density is not monotonically increasing with L_p . At high wind speeds the highest power density is obtained by a medium value of L_p , which is 100mm among the considered discrete values. A maximum power

density of 12.96mW/cm^3 is achieved at 14m/s with $L_p=100\text{mm}$. A comparison of the power density with other piezoelectric wind energy harvesters is shown in Figure 4.10. As can be seen from this figure, the present GPEH shows good performance in power density, especially at relatively high wind speeds. The present GPEH is less advantageous at lower wind speeds when compared to the harvesters in Bryant and Garcia (2011) and Sirohi and Mahadik (2011). However, the present GPEH owns the largest overall range of wind speed for effective power generation with considerable power density. Note that the data for the present GPEH in the figure is not optimized in the aspects of S_{tip} , M_{tip} , etc.

Figure 4.9 Variation of (a) average power P_{ave} with load resistance R_L at 10m/s , (b) average power P_{ave} with wind speed U at respective optimum R_L and (c) power density with wind speed U for different lengths of piezoelectric sheets L_p .

Figure 4.10 Comparison of power density with other piezoelectric wind energy harvesters

4.6 Chapter Summary

In this chapter, a comparison study is presented on the performance of the modelling methods for GPEH, including the 1DOF model, single mode and multi-mode Euler-Bernoulli distributed parameter models. A typical GPEH consisting of a piezoelectric cantilever attached to a square-sectioned bluff body is considered. Procedures for these modelling methods are presented, in which the aerodynamic forces are all formulated based on quasi-steady hypothesis. Subsequently, wind tunnel experiments for the fabricated prototypes are conducted to validate and evaluate these models. The results show that all these models can successfully predict the variation of the average power with the load resistance and the wind speed. Quite small difference is observed between the single-mode and multi-mode Euler-Bernoulli distributed parameter models. The distributed parameter model owns a more rational representation of the aerodynamic force, while the 1DOF model gives a better prediction on the cut-in wind speed. In general, the 1DOF can predict the electroaeroelastic behavior of a GPEH device accurately enough with its advantage of simplicity and ease of obtaining the electromechanical coupling term.

The influence of the load resistance, wind exposure area and mass of the bluff body, and length of the piezoelectric sheets on the cut-in wind speed as well as the output power level of the GPEH are investigated with a single-mode Euler-Bernoulli distributed parameter model. It is determined that U_{cr} and the growth rate of P_{ave} increases first with R_L to a certain value, then drops with R_L . When U is large enough the largest P_{ave} is obtained with the optimum R_L , which owns the largest U_{cr} as well. With the corresponding optimum R_L , increasing S_{tip} and decreasing M_{tip} can increase P_{ave} as well as reduce U_{cr} . For the case of L_p , larger L_p gives larger U_{cr} and larger growth rate of P_{ave} , yet not guarantees larger power density (power per piezoelectric volume), which is obtained at a medium L_p . Compared to other piezoelectric wind energy harvesters, our present GPEH gives higher power density at high wind speeds and owns the largest range of U with considerable power density. Future work involves improving the performance of the GPEH at low wind speeds.

CHAPTER 5 ENHANCED PIEZOELECTRIC GALLOPING ENERGY HARVESTIN USING TWO DEGREE-OF-FREEDOM CUT-OUT CANTILEVER WITH MAGNETIC INTERACTION⁴

5.1 Introduction

Most of the piezoelectric aeroelastic energy harvesters reported in the literature, as summarized in Section 2.3, extract wind power effectively only when the wind speed is high enough, above a critical value (cut-in wind speed) for galloping and flutter harvesters, or when the wind speed is around a specific value for vortex-shedding-based harvesters. The challenge to improve the efficiency of aeroelastic energy harvesters in the low wind speed range remains.

For vibration-based piezoelectric energy harvesting, numerous studies have been reported to improve the energy harvesting efficiency. The improvements mainly lie in two aspects: enhancing output power and broadening the bandwidth of operating frequency. For output power enhancing, mechanical (Rupp et al., 2009) and circuitry (Lefeuvre et al., 2005) optimization techniques have been investigated. For broadband energy harvesting, techniques include the multimodal configuration (Arafa et al., 2011; Li et al., 2013; Wu et al., 2012), resonance frequency tuning (Challa et al., 2008), exploitation of nonlinearities (Ramlan et al., 2010; Erturk et al., 2009; Tang and Yang, 2012), and frequency up-conversion (Wickenheiser and Garcia, 2010). To our best knowledge, the techniques mentioned above, such as the multi-modal design and magnetically induced nonlinearity, have yet been applied to small wind energy harvesters.

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This chapter presents a two degree-of-freedom (2DOF) galloping energy harvester with magnetic interaction in order to improve the power generation efficiency in the low wind speed range. A cut-out piezoelectric beam design is employed, with a square sectioned bluff body connected at the tip of the outer frame. The principle of galloping is employed due to its advantage mentioned above. Wind tunnel experiment demonstrates its capability of both reducing the cut-in wind speed and enhancing output power in the low wind speed range (lower than 5m/s) which is typical for HVAC systems' flow condition.

5.2 Proposed 2DOF piezoelectric galloping energy harvester design

The schematic of the proposed 2DOF piezoelectric galloping energy harvester is shown in Figure 5.1. The cantilever beam of the conventional 1DOF galloping harvester is cut to create an inner beam inside the outer frame, and two magnets are attached to exploit magnetic interaction between the bluff body and the free end of the inner beam. Piezoelectric elements are employed in both the outer and inner beams to convert strain energy to electricity. The root of the outer beam can be regarded as a dynamic magnifier (Arafa et al., 2011) for the inner beam oscillation, and the magnetic interaction introduces stiffness nonlinearity into the system, similar to the vibration-based energy harvesting condition (Erturk et al., 2009). However, the nonlinear behavior in this galloping energy harvester is much more complex than that in vibration energy harvesting due to the four way coupling involving fluid, structure, electronic interface circuit and magnetic interaction.

Figure 5.1 Schematic of novel 2DOF piezoelectric galloping energy harvester with magnetic interaction

Figure 5.2 (a) Experimental setup and (b) side view of proposed 2DOF harvester and 1DOF counterpart.

5.3 Experimental setup

A prototype of the proposed 2DOF energy harvester is fabricated and tested in the wind tunnel (Figure 5.2(a)). A 150mm long tip body with square section ($40 \times 40 \text{ mm}^2$) is connected to the free end of the outer beam. The mass of the tip body of the outer beam and the tip (proof) mass of the inner beam are 0.0401kg and 0.0082kg, respectively. Two piezoelectric sheets (DuraAct P-876.A12 from Physik Instrumente) with parallel connection are bonded to each side of the outer beam at the root, while one same piezoelectric sheet is attached to the root of the inner beam. These employed piezoelectric transducers consist of a centered layer of piezoceramic plate, which is sandwiched between two layers of polymers. To verify its superiority to the conventional 1DOF galloping energy harvester, a 1DOF counterpart prototype is also devised and tested for comparison (Figure 5.2(b)). For a fair comparison, the same piezoelectric sheets are bonded to the 1DOF counterpart at corresponding positions. The parameters of the proposed harvester are presented in Table 5.1. Two magnets (M1219-5 from AssemTech) are embedded in the plastic holders and attached to the tip body of the outer beam and the proof mass of inner beam with repulsive configuration. The gap between the magnets is set to be $\Delta = 13, 10, 8$ and 7 mm .

Table 5.1 Geometric and material properties of proposed 2DOF galloping energy harvester

Properties	Outer beam	Inner beam	Piezo sheets
Length, L_{b1}, L_{b2}, L_p (mm)	172	57	61 (50 is effective)
Length L_{b3}	70	--	--
Width, B_{b1}, B_{b2}, B_p (mm)	66	30	30
Thickness, t_{b1}, t_{b2}, t_p (mm)	0.6	0.3	0.5
Material	Aluminum	Aluminum	DuraAct P-876.A12
Density, ρ (kg m^{-3})	2700	2700	3825
Capacitance (nF)	---	--	90

5.4 Results and discussion

5.4.1 Influence of the gap between the magnets

Figure 5.3 shows the measured average power output for various Δ as well as the 1DOF counterpart over a range of load resistances R_L and wind velocities U . The average power P_{ave} dissipated in R_L is calculated using

$$P_{ave} = \frac{1}{T} \int_0^T \frac{V^2(t)}{R_L} dt \quad (5.1)$$

where T is sufficient length of time and $V(t)$ is the transient voltage output. In the wind tunnel experiment, for each condition with specific distance between magnets, specific wind speed and specific resistive load, we recorded the transient voltage values for 50s and calculated the average power output (it is observed that 50s is sufficiently long since the average power values do not change much with increasing time duration). The optimal R_L is only determined at 4m/s for various Δ (left column of Figure 5.3) and then this resistance is applied for other wind speeds. The average output power versus wind speed for each case is shown in the right column of Figure 5.3. As studied by the authors (Yang et al., 2013), the optimal resistance is constant for the whole wind speed range for the 1DOF harvester. For the proposed 2DOF harvester, the optimal resistance determined at 4m/s may not necessarily be the exact optimal values for other wind speeds. But even with the non-optimized resistance, the proposed 2DOF harvester presents obvious superiority to the 1DOF harvester in the low wind speed range (this will be detailed in the following part).

When the outer beam undergoes galloping, the magnet attached to the tip body regularly passes by the proof mass of the inner beam, inducing the vibrations of the inner beam via magnetic interaction. To investigate necessity and effect of magnetic interaction, we tested the same 2DOF structure with non-magnetic mass in the wind tunnel by replacing the magnet attached to the inner beam with an equal-weighted non-steel mass. It is found that without the magnetic interaction, the inner beam could hardly generate power, and the total generated power is lower than the conventional

1DOF harvester in all wind speed range. With magnetic interaction, the voltage responses in time domain at 4m/s for various Δ are shown in Figure 5.4. For the largest gap $\Delta=13\text{mm}$, the inner beam is excited with small oscillations, with a small voltage amplitude due to the weak magnetic interaction (Figure 5.4(b)), which is also reflected in the small output power shown in Figure 5.3(b) (right figure). The cut-in wind speed is beneficially reduced to 1m/s, much smaller than that for the 1DOF counterpart (around 2.5m/s). A drop in the output power for the inner beam is also observed at 3.5m/s. For the medium gap ($\Delta=10$ and 8mm), the inner beam is effectively excited to vibrate at higher frequencies. Large jumps occur when the outer beam tip body passes by the inner beam (Figure 5.4(c) and 5.4(d)). The output power of the inner beam is also greatly increased (Figure 5.3(c) and 5.3(d)). However, the voltage and output power of the outer beam decreases, due to the increased difficulty of passing by the inner beam caused by the increased repulsive magnetic force. When the gap is further reduced to $\Delta=7\text{mm}$, the increased repulsive magnetic force induces a decrease in the probability for the outer beam tip to pass by the inner beam. Thus, only a few large jumps are observed in Figure 5.4(e). It is also noted in Figure 5.3(e) that the output power of the outer and inner beam are both reduced for this case.

Figure 5.3 Power versus load resistance (left column) and power versus wind speed (right column) with different distances Δ between repulsive magnets.

Figure 5.4 Voltage responses at 4m/s for different configurations with various distances between magnets.

5.4.2 Comparison of output power

The power outputs from the outer and inner beam for different Δ are further compared in Figure 5.5. It is observed in Figure 5.5(a) that the outer beam for the four cases with magnetic interaction have smaller cut-in wind velocities and higher power output up to 3m/s, as compared to the 1DOF configuration. The output power decreases when the gap between the magnets reduces. Figure 5.5(b) shows that for the inner beam, the medium gap ($\Delta=10$ and 8mm) provides much larger output power than the 1DOF harvester up to 5m/s. The total power from the outer and inner beams of the 2DOF harvester is compared with that of 1DOF harvester in Figure 5.6. The 2DOF harvester shows obvious superiority to its 1DOF counterpart in terms of the lower cut-in wind speed of 1m/s and the larger output power up to 4.5m/s. At 2.5m/s, the output power of the 2DOF harvester with $\Delta=8$ mm is nearly four times of that of the 1DOF counterpart. In the scenarios where only ultra small-scale wind is available, such as ventilation systems, or the highly urbanized areas with limited open space, the proposed 2DOF energy harvester has good potential for application.

Figure 5.5 Comparison of output power for different configurations from (a) outer beam (root area for 1DOF) and (b) inner beam (middle area for 1DOF).

Figure 5.6 Comparison of total output power for different configurations.

5.5 Chapter Summary

In summary, the proposed 2DOF piezoelectric galloping energy harvester with a cut-out beam design and magnetic interaction can improve the power generation efficiency in the low wind speed range with both enhanced output power and reduced cut-in wind speed. Maximum improvements occur with proper distance between the magnets (i.e. $\Delta=8\text{mm}$), when the inner beam can be excited to vibrate at high

frequencies through magnetic interaction. A cut-in speed of 1m/s and nearly four time increase in power at 2.5m/s is obtained for the 2DOF harvester with $\Delta=8\text{mm}$, as compared to the 1DOF harvester. Total output power is enhanced until wind speed goes up to 4.5m/s. Experimental results confirm that the proposed energy harvester is a viable power supplier for wireless sensing nodes in indoor monitoring systems or highly urbanized areas where only small-scale wind flows are available.

CHAPTER 6 ENHANCED AEROELASTIC ENERGY HARVESTING WITH A BEAM STIFFENER⁵

6.1 Introduction

A conventional aeroelastic piezoelectric energy harvester (APEH) consists of a piezoelectric composite cantilever connected with a bluff body at its free end. Unlike the target of broadening the working frequency bandwidth in the hot area of vibration-based energy harvesting (Aladwani et al., 2012; Zhou et al., 2014; Tang and Yang, 2012), the performance enhancement of APEHs aims at reducing the cut-in wind speed, enlarging the effective wind speed range and increasing the harvested power at the target wind speed. Efforts have been devoted into this through structural modification (Zhao et al., 2014a) and circuit optimization (Bryant et al., 2013; Zhao et al., 2014b).

Besides the modified 2DOF structural design for a GPEH as proposed in Chapter 5, in this chapter, a convenient method is presented to boost the output power level as well as to enlarge the effective wind speed range, simply by adding a beam stiffener to the substrate beside the transducer, as shown in Figure 6.1(a). The principle is to amplify the electromechanical coupling by magnifying the slope of the mode shape of the harvester at the edge of the transducer, which is achieved by stiffening the substrate to the right of the transducer (Figure 6.1(b)). This is inspired by the fact that the electromechanical coupling is proportional to the difference of mode shape slopes at the starting and ending locations of the transducer (Abdelkefi et al., 2013b; Zhao et al., 2013). Magnified electromechanical coupling results in increased capability of energy conversion thus increased power extracted from the flow. Or from a more intuitive perspective, stiffening the beam section causes redistribution of the strain energy

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induced by aerodynamic force. The strain energy in the substrate beside the transducer is redistributed to the transducer when it is stiffened, thus boosting the available transferred energy. To investigate the response of the proposed APEH with a beam stiffener and to validate its superior performance over the conventional one, we consider three types of APEHs, which are based on the aeroelasticity principles of galloping (Figure 6.1(a)), vortex-induced vibration (VIV) (Figure 6.2(a)), and flutter (Figure 6.2(b)). Wind tunnel experiment results of a galloping-based APEH with a beam stiffener are used to prove the concept and to validate the theoretical model. While previous research works have focused on a single aeroelasticity phenomenon, in this chapter, a comprehensive comparison of power generation efficiency of the three types of APEHs is presented, serving as a choice guideline for optimal energy harvesting for the given wind situation where the WSNs are located.

(a)

(b)

Figure 6.1 (a) Schematic of proposed GPEH with a beam stiffener (b) comparison of mode shapes

Figure 6.2 Top view of modified (a) VIV-based APEH and (b) flutter-based APEH

6.2 Analytical model

Figure 6.3 Cross sectional views of each segment

6.2.1 Modal analysis for undamped free vibration

The three types of APEHs as mentioned above consist of identical cantilevers which are sandwiched by two piezoelectric transducers at the root, and a specific bluff body (or airfoil) connected at the free end. With the beam stiffener, the whole cantilever is divided into three segments. To study its electro-aero-mechanical responses, we consider a distributed parameter model based on the Euler-Bernoulli beam theory. The case of undamped free vibration without the aerodynamic force is considered first. The equation of motion for each segment is

$$EI_i \frac{\partial^4 w_i(x_i, t)}{\partial x_i^4} + m_i \frac{\partial^2 w_i(x_i, t)}{\partial t^2} = 0, \quad i=1,2,3 \quad (6.1)$$

where $0 < x_1 < L_1$, $0 < x_2 < L_2$, $0 < x_3 < L_3$; $w_i(x_i, t)$ is the cantilever's transverse deflection; and EI_i and m_i are the bending stiffness and distributed mass of each segment, respectively. The cross sectional views of each segment are shown in Figure 6.3. The expressions of EI_i and m_i are given as, respectively, $EI_1 = E_s b_s h_s^3/12 + 2E_p b_p [(h_p + h_s/2)^3 - h_s^3/8]/3$ and $m_1 = \rho_s h_s b_s + 2\rho_p h_p b_p$ for $0 < x_1 < L_1$, $EI_2 = E_s b_s h_s^3/12 + 2E_{bs} b_{bs} [(h_{bs} + h_s/2)^3 - h_s^3/8]/3$ and $m_2 = \rho_s h_s b_s + 2\rho_{bs} h_{bs} b_{bs}$ for $0 < x_2 < L_2$, $EI_3 = E_s b_s h_s^3/12$ and $m_3 = \rho_s h_s b_s$ for $0 < x_3 < L_3$, where E , ρ , h and b are the Young's modulus, mass density, thickness and width, respectively, with the subscript s , p , and bs denoting the substrate, piezoelectric patch (at open-circuit condition) and the beam stiffener, respectively. To solve the mode shape and natural frequency problems, the boundary conditions at $x_1=0$ and $x_3=L_3$ are determined as

$$\begin{aligned} w_1(0, t) = 0, \quad \frac{\partial w_1(x_1, t)}{\partial t} \Big|_{x_1=0} = 0, \quad EI_3 \frac{\partial^2 w_3(x_3, t)}{\partial x_3^2} \Big|_{x_3=L_3} = -J_t \frac{\partial^3 w_3(x_3, t)}{\partial x_3 \partial t^2} \Big|_{x_3=L_3}, \\ EI_3 \frac{\partial^3 w_3(x_3, t)}{\partial x_3^3} \Big|_{x_3=L_3} = M_t \frac{\partial^2 w_3(x_3, t)}{\partial t^2} \Big|_{x_3=L_3} \end{aligned} \quad (6.2)$$

where M_t and J_t are, respectively, the mass and rotary inertia of the bluff body. The transition conditions at $x_1=L_1$ and $x_2=L_2$ are

$$\begin{aligned} w_1(L_1, t) = w_2(0, t), \quad \frac{\partial w_1(x_1, t)}{\partial x_1} \Big|_{x_1=L_1} = \frac{\partial w_2(x_2, t)}{\partial x_2} \Big|_{x_2=0}, \\ EI_1 \frac{\partial^2 w_1(x_1, t)}{\partial x_1^2} \Big|_{x_1=L_1} = EI_2 \frac{\partial^2 w_2(x_2, t)}{\partial x_2^2} \Big|_{x_2=0}, \quad EI_1 \frac{\partial^3 w_1(x_1, t)}{\partial x_1^3} \Big|_{x_1=L_1} = EI_2 \frac{\partial^3 w_2(x_2, t)}{\partial x_2^3} \Big|_{x_2=0} \quad (6.3) \\ w_2(L_2, t) = w_3(0, t), \quad \frac{\partial w_2(x_2, t)}{\partial x_2} \Big|_{x_2=L_2} = \frac{\partial w_3(x_3, t)}{\partial x_3} \Big|_{x_3=0}, \\ EI_2 \frac{\partial^2 w_2(x_2, t)}{\partial x_2^2} \Big|_{x_2=L_2} = EI_3 \frac{\partial^2 w_3(x_3, t)}{\partial x_3^2} \Big|_{x_3=0}, \end{aligned}$$

$$EI_2 \frac{\partial^3 w_2(x_2, t)}{\partial x_2^3} \Big|_{x_2=L_2} = EI_3 \frac{\partial^3 w_3(x_3, t)}{\partial x_3^3} \Big|_{x_3=0} \quad (6.4)$$

In order to solve the eigenvalue problem for undamped free vibration, we let $w_i(x_i, t) = \phi_i(x_i)e^{i\omega t}$ where $\phi_i(x_i)$ are the mode shapes for each segment and ω is the natural frequency. In such a way, the equation of motion in Equation (6.1) and the boundary and transition conditions in Equations (6.2) to (6.4) can be rewritten in terms of $\phi_i(x_i)$ and ω . Therefore, Equation (6.1) becomes a differential equation

$$EI_i \frac{d^4 \phi_i(x_i)}{dx_i^4} - \omega^2 m_i \phi_i(x_i) = 0, \quad i = 1, 2, 3 \quad (6.5)$$

of which the solution, i.e., the series of $\phi_{ir}(x_i)$ can be expressed as

$$\phi_{ir}(x_i) = A_{ir} \sin(\beta_{ir} x_i) + B_{ir} \cos(\beta_{ir} x_i) + C_{ir} \sinh(\beta_{ir} x_i) + D_{ir} \cosh(\beta_{ir} x_i), \quad i = 1, 2, 3 \quad (6.6)$$

where $\beta_{ir}^4 = \omega_r^2 m_i / EI_i$, and the subscript r denotes the r th mode. Introducing the mode shapes in Equation (6.6) into the twelve boundary and transition conditions results in an eigenvalue problem consisting a 12×12 coefficient matrix of the vector $a = [A_1, B_1, C_1, D_1, A_2, B_2, C_2, D_2, A_3, B_3, C_3, D_3]$. The natural frequencies ω_r are obtained as a common practice by forcing the 12×12 matrix to be singular, and the coefficients A_{ir} , B_{ir} , C_{ir} , and D_{ir} will subsequently be obtained from the corresponding eigenvector a . After the mode shapes are obtained, a mass normalization process for $\phi_{ir}(x_i)$ are employed as

$$\sum_{i=1}^3 \int_0^{L_i} m_i \phi_{ir}(x_i) \phi_{is}(x_i) dx_i + M_t \phi_{3r}(L_3) \phi_{3s}(L_3) + J_t \frac{d\phi_{3r}(x_3)}{dx_3} \Big|_{x_3=L_3} \frac{d\phi_{3s}(x_3)}{dx_3} \Big|_{x_3=L_3} = \delta_{rs} \quad (6.7)$$

where δ_{rs} is the Kronecker delta.

6.2.2 Electromechanical model

To obtain the mechanical equation with electromechanical coupling, the structural and air damping, the piezoelectric effect and the aerodynamic force are introduced into the mechanical equation. Because in Segment 1 with $0 < x_1 < L_1$, piezoelectric patches are included, the generated voltage has additional effects on the bending moment as

$$M_1(x_1, t) = EI_1 \frac{\partial^2 w_1(x_1, t)}{\partial x_1^2} + \theta V(t) [H(x_1) - H(x_1 - L_1)] \quad (6.8)$$

Where θ is the coupling term expressed as $\theta = -E_p d_{31} b_p (h_p + h_s)$ with d_{31} being the piezoelectric constant, and $H(x)$ is the Heaviside function. The equation of motion in Equation (6.1) is modified to

$$EI_i \frac{\partial^4 w_i(x_i, t)}{\partial x_i^4} + c_{si} I_i \frac{\partial^5 w_i(x_i, t)}{\partial x_i^4 \partial t} + c_{ai} \frac{\partial w_i(x_i, t)}{\partial t} + m_i \frac{\partial^2 w_i(x_i, t)}{\partial t^2} + \delta_{li} \theta V(t) \left[\frac{d\delta(x_1)}{dx} - \frac{d\delta(x_1 - L_1)}{dx} \right] = F_{air}(t) \delta(x_3 - L_3) \quad (6.9)$$

where F_{air} is the aerodynamic acting on the bluff body; c_{si} and c_{ai} are the employed to account for the effects of strain rate damping and air viscous damping, which will be addressed later.

The circuit equation with electromechanical coupling is established based on the piezoelectric constitutive relation (IEEE Standard on Piezoelectricity, 1987). The vector of the electric displacement for a single piezoelectric patch has non-zero term only in direction 3, i.e., in the direction of transverse displacement along the z axis, which can be expressed in terms of the strain at the center of the piezoelectric patch (Figure 6.3) and the generated voltage (Erturk and Inman, 2008a) as

$$D_3(x_1, t) = -d_{31} E_p \frac{h_p + h_s}{2} \frac{\partial^2 w_1(x_1, t)}{\partial x_1^2} - \epsilon_{33}^s \frac{V(t)}{h_p} \quad (6.10)$$

where ϵ_{33}^s is the piezoelectric permittivity component at constant strain. Applying Gauss law (IEEE Standard on Piezoelectricity, 1987), and letting the two piezoelectric patches be connected in parallel, the current output is

$$I(t) = \frac{d}{dt} \left[\int_0^{L_1} 2b_p D_3(x_1, t) dx \right] = \theta \int_0^{L_1} \frac{\partial^3 w_1(x_1, t)}{\partial x_1^2 \partial t} dx - C_p \frac{dV(t)}{dt} \quad (6.11)$$

$$C_p = \frac{2\epsilon_{33}^s b_p L_1}{h_p}$$

where C_p is the total capacitance of the two piezoelectric patches. Because only a load resistor R is considered in the circuit, the current is simply equal to $V(t)/R$. Therefore, the coupled circuit equation is written straightforwardly as

$$\frac{V(t)}{R} + C_p \frac{dV(t)}{dt} - \theta \int_0^{L_1} \frac{\partial^3 w_1(x_1, t)}{\partial x_1^2 \partial t} dx = 0 \quad (6.12)$$

With the obtained mode shapes in Equation (6.6), the displacement can be expanded in terms of modal coordinates $\eta_r(t)$ as $w_i(x_i, t) = \sum \phi_{ir}(x_i) \eta_r(t)$. Up to this end, introducing the expanded $w_i(x_i, t)$ into Equations (6.9) and (6.12), applying the orthogonality conditions in Equation (6.7), and considering only the fundamental mode yield the coupled modal electromechanical equations as

$$\begin{aligned} \ddot{\eta}(t) + 2\zeta\omega_n\dot{\eta}(t) + \omega_n^2\eta(t) + \chi V(t) &= \phi_3(L_3)F_{air}(t) = f_{air}(t) \\ \frac{V(t)}{R} + C_p\dot{V}(t) - \chi\dot{\eta}(t) &= 0 \end{aligned} \quad (6.13)$$

where ζ is the modal mechanical damping which can be experimentally measured; ω_n is the fundamental frequency; f_{air} is modal aerodynamic force of which the formulation is dependent on the specific aeroelastic instability phenomenon, and is introduced in the following section ; and χ is the modal electromechanical coupling term written as $\chi = \theta[\phi_1'(L_1) - \phi_1'(0)]$. It is clear that once the geometric and material parameters regarding the piezoelectric cantilever are fixed, θ is constant and χ depends purely on (and is proportional to) the value of $\phi_1'(L_1) - \phi_1'(0)$. Considering that $\phi_1'(0)$ is kept to be zero with a fixed boundary condition, by varying L_2 , $\phi_1'(L_1)$ is altered, influencing electromechanical coupling as well as the total power generation performance.

6.2.3 Aerodynamic model

The galloping-based APEH (GPEH) in Figure 6.1(a) consists of the above introduced piezoelectric cantilever and a square-sectioned bluff body fixed at its free end. The modal galloping force $f_{galloping}$ is calculated based on the quasi-steady hypothesis (Barrero-Gil et al., 2010), which is applicable to slow oscillations when the characteristic timescale of the flow is much smaller than the characteristic timescale of the oscillation. This is the case of the galloping phenomenon. $f_{galloping}$ is calculated by

$$f_{galloping}(t) = \phi_3(L_3) \times \frac{1}{2} \rho h l U^2 \sum_{j=1}^3 A_j \left[\frac{\phi_3(L_3)\dot{\eta}(t)}{U} + \phi_3'(L_3)\eta(t) \right]^j \quad (6.14)$$

where ρ , h , l , U and A_j are the air density, frontal dimension and length of the bluff body, wind speed, and empirical aerodynamic coefficients, respectively. The final governing electro-aero-mechanical equations for the present GPEH are obtained by replacing f_{air} in Equation (6.13) with $f_{galloping}$.

The VIV-based APEH (VIVPEH) (Figure 6.2(a)) consists of the same piezoelectric cantilever and a cylindrical bluff body fixed at its free end. The modal circuit equation remains the same as the second equation of (6.13). For the motion of equation, along with the structural and air viscous damping, the fluid-added damping is required to be considered. Here we consider that the fluid-added damping $F_{damping}$ is acting on the bluff body together with the aerodynamic force due to vortices F_{VIV} . Therefore, Equation (6.9) needs to be modified to

$$EI_i \frac{\partial^4 w_i(x_i, t)}{\partial x_i^4} + c_{si} I_i \frac{\partial^5 w_i(x_i, t)}{\partial x_i^4 \partial t} + c_{ai} \frac{\partial w_i(x_i, t)}{\partial t} + F_{damping} \delta(x_3 - L_3) + m_i \frac{\partial^2 w_i(x_i, t)}{\partial t^2} + \delta_{il} \theta V(t) \left[\frac{d\delta(x_1)}{dx} - \frac{d\delta(x_1 - L_1)}{dx} \right] = F_{VIV}(t) \delta(x_3 - L_3) \quad (6.15)$$

where $F_{damping}$ and F_{VIV} are calculated by Facchinetti et al. (2004)

$$F_{damping} = \frac{1}{2} C_D \rho h l U \frac{dw_3(L_3, t)}{dt}$$

$$F_{VIV} = \frac{1}{4} C_{L0} \rho h l U^2 q(t) \quad (6.16)$$

where C_D is the mean sectional drag coefficient, C_{L0} is the steady lift coefficient (i.e., the lift coefficient for a fixed cylinder subjected to vortex shedding), and q is a dimensionless wake variable that is employed to describe the fluctuating behavior of the vortex lift coefficient (i.e., the lift coefficient for a flexibly supported cylinder that moves due to vortex shedding). It should be noted that both $F_{damping}$ and F_{VIV} are in the direction normal to the wind flow. Again, applying the expanded $w_i(x_i, t)$ and the orthogonality conditions, the coupled equation of motion is rewritten in modal coordinate as

$$\ddot{\eta}(t) + 2\zeta\omega_n\dot{\eta}(t) + \frac{1}{2} C_D \rho h l U \phi_3^2(L_3) \dot{\eta}(t) + \omega_n^2 \eta(t) + \chi V(t) = f_{VIV}(t),$$

$$f_{VIV}(t) = \frac{1}{4} \phi_3(L_3) C_{L0} \rho h l U^2 q(t) \quad (6.17)$$

where f_{viv} is the modal VIV force. As in the commonly used phenomenological models of VIV, the wake variable q that represents the fluctuating nature of the vortex shedding is modeled in the wake oscillator. As summarized by Facchinetti et al. (2004), q can be modeled to be related either to the weighted average of the transverse flow component, or to the mean transverse displacement of the near wake layer relative to the mean streamwise axis, or to the transverse velocity of a near wake fluid mass. Here we employ the phenomenological model used by Facchinetti et al. (2004) in which the q is modeled in a wake oscillator satisfying the van der Pol equation, as given in Equation (6.18). ε and A are constants obtained from experiments and ω_f is the vortex-shedding frequency. It is common to express ω_f as $\omega_f = 2\pi StU/h$, with St being the Strouhal number. In such a way, q as well as f_{viv} at a constant wind speed U will undergo limit cycle oscillations with a constant frequency equal to the vortex shedding frequency corresponding to that U . F_q denotes the coupling effects of the structural movement on the wake dynamics, which is modeled here as a linear term about the acceleration of the transverse vibration of the cylinder (Facchinetti et al., 2004)

$$\ddot{q}(t) + \varepsilon \omega_f [q^2(t) - 1] \dot{q}(t) + \omega_f^2 q(t) = F_q$$

$$F_q = \frac{A}{h} \frac{d^2 w_3(L_3, t)}{dt^2} = \frac{A}{h} \phi_3(L_3) \ddot{\eta}(t) \quad (6.18)$$

Finally, for the flutter-based APEH (FPEH) in Figure 6.2(b), the piezoelectric cantilever is the same with the former two cases, while the bluff body is changed to a NACA0012 airfoil connected to the free end using a revolute joint. The electromechanically coupled equation of motion is given in Equation (6.19), where we consider that the piezoelectric coupling ΘV is introduced to the plunge degree of freedom only. The lumped effective mass M_T , transverse damping C_w , transverse stiffness K_w , and the electromechanical coupling Θ are linked to the distributed parameter model by $M_T = 1/\phi_3^2(L_3)$, $C_w = 2\zeta\omega_n/\phi_3^2(L_3)$, $K_w = \omega_n^2/\phi_3^2(L_3)$ and $\Theta = \chi/\phi_3(L_3)$. I_a , M_w , C_a , K_a and X_G are the rotary moment and mass of the airfoil, the damping and rotary stiffness of the revolute joint and the distance from the airfoil mass center to the revolute joint, respectively. w_t and α are the transverse and rotational displacement of the airfoil at the

joint, respectively. The flutter lift force $L_{flutter}$ and moment $M_{flutter}$ are modeled using a quasi-steady approximation, as depicted in Equations (6.20) (Bibo and Daqaq, 2013b; Strganac et al., 2000). The model is established based on the concept of effective angle of attack $\alpha_{eff} = \alpha + w_t/U + (0.5b-a)\dot{\alpha}/U$ as in the Theodorsen's unsteady thin-airfoil theory (Theodorsen, 1934). b , S , a , and C_L are the half chord and span of the airfoil, distance from the mid chord to the revolute joint and lift coefficient, respectively. But here we ignore the noncirculatory effects due to the apparent-mass effects in the lift and moment. The nonlinear term of α_{eff} is introduced in order to predict the amplitude of the limit cycle oscillations beyond the flutter boundary. It should be noted that the quasi-steady approximation is only applicable for slow harmonic oscillations when the reduced frequency $k = b\omega(\text{Hz})/U$ is restricted to be very small (Hodges and Pierce, 2002). As for the FPEH in the present study, it can be shown later that k is always less than 0.1, making it suitable to apply the quasi-steady approximation. This quasi-steady model has been experimentally validated in the work of Bibo and Daqaq (2013b) regarding the flutter speed and the steady state amplitude beyond the flutter boundary. As for the coupled circuit equation, it should be noted that in the second equation of (6.13), χ is replaced by Θ for the model simulation of FPEH.

$$\begin{aligned} M_T \ddot{w}_t(t) + M_w X_G \ddot{\alpha}(t) + C_w \dot{w}_t(t) + K_w w_t(t) + \Theta V(t) &= -L_{flutter}(t), \\ M_w X_G \ddot{w}_t(t) + I_\alpha \ddot{\alpha}(t) + C_\alpha \dot{\alpha}(t) + K_\alpha \alpha(t) &= M_{flutter}(t) \end{aligned} \quad (6.19)$$

$$\begin{aligned} L_{flutter}(t) &= \rho b S U^2 C_L [\alpha_{eff}(t) - c_3 \alpha_{eff}^3(t)], \\ M_{flutter}(t) &= \left(\frac{b}{2} + a \right) L_{flutter}(t) \end{aligned} \quad (6.20)$$

6.3 Experimental verification

Figure 6.4(a) shows the experimental setup of a GPEH prototype added with the beam stiffener, which is installed in the wind tunnel to investigate its aeroelastic and power response. The cantilever of the prototype is fabricated with an aluminum substrate

sandwiched between two piezoelectric transducers (DuraAct P-876.A12 from Physik Instrumente). Two sheets of acrylic glass are bonded besides the transducers as the beam stiffener. The wind speed is measured by a hotwire anemometer, and the voltage across the load is measured by the NI 9229 DAQ module. The parameters of the aluminum substrate, piezoelectric transducer and acrylic glass beam stiffener are listed in Table 6.1. The square-sectioned bluff body has a length of $l=150\text{mm}$, a side length of the cross section $h=40\text{mm}$, and a total mass of $M_t=15.1\text{g}$. The rotary inertia J_t is ignored. During the test, three values of L_2 , 20mm, 50mm, and 80mm are considered, i.e., three beam stiffeners with different length are tested. The two transducers are connected in parallel, with a total capacitance of 180nF. The average power is calculated with $P=V_{RMS}^2/R$ where the RMS voltage $V_{RMS}=V/\sqrt{2}$.

Table 6.1 Parameters of aluminum substrate, piezoelectric transducer and acrylic glass beam stiffener

Parameter	Aluminum substrate	Piezoelectric transducer	Acrylic glass beam stiffener
Length, $L_1+L_2+L_3$, L_1, L_2 (mm)	270	61	20, 50, 80, 110, 140, 209
Width, b_s , b_p , b_{bs} (mm)	34	34	45
Thickness, h_s , h_p , h_{bs} (mm)	0.6	0.5	5
Mass density, ρ_s , ρ_p , ρ_{bs} (kg m^{-3})	2700	3825	1180
Young's modulus, E_s , E_p , E_{bs} (GPa)	69	23.3	3.2
Capacitance, C_p (nF)	--	90	--
Piezoelectric constant, d_{31} ($\text{pm}\cdot\text{V}^{-1}$)	--	-174	--

Figure 6.4 (a) Experimental setup (b) experimental results

Figure 6.4(b) shows the experimental result along with the theoretical prediction (aerodynamic coefficients in Equation (6.14) are taken as $A_1=2.3$, $A_2=0$, $A_3=-18$ (Barrero-Gil et al., 2010)). Both results clearly indicate the superiority of the GPEH with beam stiffener, generating significantly increased power compared to the conventional counterpart. All the considered cases of L_2 successfully give enhanced output power, with the highest power achieved with $L_2=80\text{mm}$. This is as expected since $\phi_1'(L_1)$ is larger for 80mm than for 20mm and 50mm, as reflected in the mode shapes shown in Figure 6.5(a). In the aspect of cut-in wind speed U_{cr} , while for $L_2=20$ and 50mm no obvious increase in U_{cr} is observed, the U_{cr} associated with $L_2=80\text{mm}$ is apparently increased due to the hardened stiffness of the system introduced by the stiffener, which is also reflected in the change of fundamental frequencies shown in Figure 6.5(b). Yet the variation of U_{cr} is not captured in the experiment due to the inevitable turbulence in the wind tunnel.

6.4 Variation of the mode shapes and fundamental frequency

Subsequently, with the theoretical models, further calculations of responses of power P , displacement of the bluff body w_t and efficiency η^* (which denotes the ratio of the converted electrical power to the available mechanical power in the flow) are

conducted for $L_2=20, 50, 80, 110, 140$ and 209mm . Before proceeding to these results, we should take a further look at the variation of $\phi(x)$ and ω_n , both of which have great effects on the responses. The effective mass of the harvester system M_{eff} can be derived based on the equivalent expressions of kinetic energy, given by

$$\frac{1}{2} M_{eff} [\dot{\phi}_3(L_3) \dot{\eta}(t)]^2 = \frac{1}{2} M_t [\dot{\phi}_3(L_3) \dot{\eta}(t)]^2 + \frac{1}{2} \left\{ \int_0^{L_1} m_1 [\dot{\phi}_1(x_1) \dot{\eta}(t)]^2 dx_1 + \int_{L_1}^{L_2} m_2 [\dot{\phi}_2(x_2) \dot{\eta}(t)]^2 dx_2 + \int_{L_2}^{L_3} m_3 [\dot{\phi}_3(x_3) \dot{\eta}(t)]^2 dx_3 \right\} \quad (6.21)$$

$$M_{eff} = \frac{M_t \phi_3^2(L_3) + \int_0^{L_1} m_1 \phi_1^2(x_1) dx_1 + \int_{L_1}^{L_2} m_2 \phi_2^2(x_2) dx_2 + \int_{L_2}^{L_3} m_3 \phi_3^2(x_3) dx_3}{\phi_3^2(L_3)} \quad (6.22)$$

Table 6.2 Influence of beam stiffener on effective mass of harvester

Length of the beam stiffener L_2	Effective mass of the harvester M_{eff}	Increasing percentage of M_{eff}
0	18.1g	--
20mm	17.9g	-1.10%
50mm	17.8g	-1.66%
80mm	18.6g	2.76%
110mm	22.4g	23.76%
140mm	31.7g	75.14%
209mm	61.8g	241.44%

Straightforwardly, the effective mass can be calculated as in Equation (6.22). For the conventional harvester without the beam stiffener, the M_{eff} is calculated to be 18.1g. Adding the beam stiffener with a varying length of $L_2=20, 50, 80, 110, 140$ and 209mm , the respective M_{eff} is listed in Table 6.2, along with the increasing percentage compared to the original M_{eff} of the conventional harvester. It is found that the change of M_{eff} is not significant for small and medium of L_2 of 20, 50 and 80mm; while when a long beam stiffener is used with a L_2 of 110, 140 and 209mm, M_{eff} is significantly increased, up to 241.44% for 209mm, especially. Note that the counter-intuitively slight decrease of M_{eff} at $L_2=20$ and 50mm (i.e., with a decrease of 1.10% and 1.16%, respectively,

shown in Table 6.2) is due to the decreased $\phi_3(x_3)$ as shown in Figure 6.5(a), causing a smaller contribution of the beam mass to the total effective mass which is considered to be at the free end. Associated with the variation M_{eff} with L_2 , it is shown from Figure 6.5(a) that with L_2 increases from 0 to 140mm, $\phi_1'(L_1)$ gradually increases, leading to increasing electromechanical coupling. Yet when L_2 further increases to 209mm, $\phi_1'(L_1)$ decreases. This is because that the added mass by the acrylic stiffener becomes prevalent and induces greatly increased effective mass M_{eff} , which is associated with the obviously decreased $\phi_3(L_3)$. Similarly, ω_n gradually increases due to the extra bending stiffness induced by the beam stiffener until 110mm, then decreases when L_2 further increases because the effect of the increased bending stiffness is outweighed by the greatly increased M_{eff} .

In addition, two more things need to be clarified. According to a previous parametric study of a galloping based energy harvester carried out by Zhao et al. (2013), with the same stiffness of the harvesting system, increasing the effective mass results in a reduced power output. This is different from the case of a vibration energy harvester with base excitation, whose output power is enlarged by an increased effective mass, because the direct excitation source of the harvester is its own inertia, i.e., the larger the effective mass, the more the energy input. Nevertheless, the direct energy input of an aeroelastic energy harvester is due to the aerodynamic force, which is purely dependent on the aerodynamic properties of the bluff body yet independent on the mass. In contrast, a larger effective mass induces more difficulty in exciting the harvester. This is why we choose the material of acrylic glass which has a low mass density for the beam stiffener to limit the added mass.

Also, in order to make sure the beam stiffener is “rigid” enough, a large thickness of 5mm is chosen for each sheet of acrylic glass, with which the total bending stiffness EL_2 is calculated to be $1433.31 \times 10^7 \text{ N} \cdot \text{m}^2$. This corresponds to over 300 times increase compared to the original bending stiffness of the substrate which is $4.35 \times 10^7 \text{ N} \cdot \text{m}^2$. Therefore, the beam segment with the introduced stiffener is regarded to be “rigid”. It

should be noted that other materials with higher Young's modulus can be a better candidate for the beam stiffener, which can achieve a significant increase of the bending stiffness EI_2 with a smaller thickness h_{bs} than we used here (i.e., 5mm). But the mass density of the chosen material cannot be too high, in order to limit the added mass of the stiffener, therefore, metal materials are not recommended.

(a)

(b)

Figure 6.5 (a) Variation of fundamental mode shapes and (b) fundamental frequencies with length of beam stiffener

6.5 Parametric study and discussion

6.5.1 Effects of the wind speed and length of beam stiffener on the harvester's response

In the following theoretical simulations for the GPEH, VIVPEH and FPEH, parameters regarding the piezoelectric cantilevers and masses of the bluff bodies (for the flutter, it is the sum of the masses of the airfoil and revolute joint) are identical, such that the $\phi(x)$ and ω_n for all harvesters are the same as those shown in Figure 6.5. The variations of P and w_t at optimal load R_{opt} (which gives the maximum P over the whole R range) with U for the three types of APEHs are plotted in Figure 6.6. Figure 6.6(a) clearly indicates the dramatic enhancement of power output for the GPEH with the beam stiffener, especially for larger values of L_2 at high U . The growing rate of P (i.e. the slope of the curve of P versus U) increases with L_2 . At a small wind speed $U=3\text{m/s}$, the conventional GPEH generates a power of $P=0.60\text{mW}$, while the GPEH with a stiffener of $L_2=80\text{mm}$ generates $P=1.49\text{mW}$; at $U=5\text{m/s}$, P from the conventional GPEH is 1.42mW , while it is increased to 7.57mW with a beam stiffener of $L_2=110\text{mm}$; at a large $U=15\text{m/s}$, the conventional GPEH provides $P=3.10\text{mW}$, while the stiffened GPEH with $L_2=209\text{mm}$ achieves $P=139.74\text{mW}$. These correspond to 1.5, 4.3 and 44.0 times increase of power output at the respective wind speed. The dashed curve depicts the envelope of achievable power by the GPEH with beam stiffener. Nevertheless, the U_{cr} of GPEH with beam stiffener also increases with L_2 . This drawback can be overcome by a pre-measurement of the wind speed at the target location of harvesters or WSNs, employing the optimal L_2 to achieve the best power generating performance according to Figure 6.6(a). The variation of w_t with U is presented in Figure 6.6(b), which clearly shows that the stiffened GPEHs have smaller displacement compared to the conventional counterparts, except for $L_2=209\text{mm}$. This further confirms the benefit of the beam stiffener of fatigue mitigation and durability enhancement.

Figure 6.6 Responses of (a) (b) GPEH (c) (d) VIVPEH and (e) (f) FPEH. (a) (c) (e) P versus U , (b) (f) w_t versus U and (d) w_t versus R . Dashed curves represents envelope of attainable P .

For the VIVPEH, high amplitude oscillations occur in a lock-in region where the vortex shedding frequency gets close to the structure's natural frequency. As shown in Figure 6.6(c) (with the aerodynamic parameters $C_D=2.0$, $C_{L0}=0.3$, $\varepsilon=0.3$, $A=12$ (Facchinetti et al., 2004), and $St=0.105$ (Akaydin et al., 2012)), there exist the cut-in and cut-out wind speeds for the VIVPEH. As expected, the lock-in region shifts to higher U with increasing L_2 until 110mm, then shifts to smaller U when L_2 further increases. This agrees to the variation of ω_n in Figure 6.5(b). The maximum P of 0.09mW is achieved by the conventional VIVPEH at $U=1.94\text{m/s}$, while the maximum P of 1.26mW is obtained by the stiffened VIVPEH at $U=3.50\text{m/s}$ with $L_2=110\text{mm}$, corresponding to 13.1 times increase. Besides the enhancement of power, the beam stiffener brings an broadened lock-in region as well, i.e., from 1.20m/s to 1.96m/s for the conventional VIVPEH, and from 2.50m/s to 3.62m/s for the stiffened one with $L_2=110\text{mm}$. The dashed curve gives the attainable power envelope of the stiffened VIVPEH, which can be achieved by appropriately adjusting L_2 according to the pre-tested target U . Figure 6.6(d) clearly indicates that enhanced power is reached with comparable or even smaller displacement.

Table 6.3 Physical and mechanical parameters for FPEH

Parameters	Value
$I_a (\text{g} \cdot \text{cm}^2)$	24.3
$M_w (\text{g})$	7.84
$C_a (\text{N} \cdot \text{m} \cdot \text{s})$	0.0000163
$K_a (\text{N} \cdot \text{m}/\text{rad})$	0.0019
$X_G (\text{mm})$	3
$b (\text{mm})$	42
$S (\text{mm})$	52
$a (\text{mm})$	-21

The physical and mechanical parameters involved in the simulation of the FPEH are listed in Table 6.3. With the aerodynamic parameters $C_L=6.283$ and $c_3=6.5$, subcritical Hopf bifurcation is observed, as shown in Figure 6.6(e) and 6.6(f). Moreover, there

exist cut-out wind speeds when U grows to the higher range. Enhanced power output is also achieved (Figure 6.6(e)), e.g., at $U=5\text{m/s}$, P is increased from 0.096mW for the conventional FPEH to 1.28mW , i.e., 12.3 times increase for the stiffened one with $L_2=50\text{mm}$. More obvious enhancement is observed at high U range, e.g., at $U=17\text{m/s}$, $P=26.52\text{mW}$ is obtained with $L_2=140\text{mm}$. Inspecting the responses of w_t shown in Figure 6.6(f), it is seen that the involvement of the beam stiffener induces almost doubled displacement for FPEH. The maximum w_t is around 30mm , still acceptable for the 270mm cantilever, but the fatigue problem requires attention when employing the stiffened FPEH.

Figure 6.7 Comparison of influences of electromechanical coupling and frequency on output power (a) GPEH (b) VIVPEH and (c) FPEH.

Further calculations are carried out to determine the dominant contribution to the enhanced power generation performance, i.e., to weigh the contributions of (i) the increased electromechanical coupling induced by the increased $\phi_I'(L_1)$ and (ii) the increased fundamental frequency ω_n . $\phi_I'(L_1)$ as well as the electromechanical coupling are kept the same with those of the conventional harvester, but ω_n is changed to the increased values shown in Figure 6.5(b). The results of power generation are plotted in Figure 6.7. It is shown that the increased fundamental frequency does contribute to the enhanced power output, but much less significantly than $\phi_I'(L_1)$ does. For example, at $U=10\text{m/s}$, with the increased frequency of $L_2=50\text{mm}$, the output power is increased from 2.76mW of the conventional harvester to 3.07mW , corresponding to a 11.23% increase, while with the increased electromechanical coupling (or $\phi_I'(L_1)$) it is increased

to 7.58mW, corresponding to a 174.64% increase. Therefore, it is clear that the major contributing factor induced by the beam stiffener is the increased electromechanical coupling, not the extra bending stiffness that is brought along with it.

6.5.2 Efficiency comparison

Figure 6.8 Efficiency of power conversion of (a) GPEH (b) VIVPEH and (d) FPEH.

The efficiency of power conversion for the three types of harvesters is calculated using $\eta^* = P/P_w$, where P_w is the total input wind energy, expressed as, for GPEH and VIVPEH, $P_w = \rho U^3 l(h/2 + w_t)$, and for FPEH, $P_w = \max\{\rho U^3 S(w_t + \alpha(b+a)), \rho U^3 S(w_t - \alpha(b-a))\}$. Figure 6.8(a) indicates that the maximum η^* for GPEHs increases with L_2 until 110mm, then decreases. The peak η^* of 0.89% is achieved with $L_2=110$ mm at $U=5.9$ m/s. For all the GPEHs, it is observed that highest η^* is reached right after the cut-in wind speed. As for the VIVPEH shown in Figure 6.8(b), the highest η^* appears at the middle of the respective lock-in region, which similarly increases with L_2 first, then decreases. The peak η^* is 0.68% at $U=3.2$ m/s with $L_2=110$ mm. For the FPEH (Figure 6.8(c)), the peak η^* is attained to be 5.58% at $U=0.9$ m/s with $L_2=20$ mm. The highest η^* is also reached around the flutter boundary. In a word, all three types of stiffened harvesters achieve the enhanced efficiency. Taking a general comparison, the FPEH owns the highest power extraction efficiency, while the VIVPEH has the lowest efficiency. However, the VIVPEH performs best at small wind speed range, especially when U stays stable, such as the situation near the ventilation outlets in the buildings. The GPEH and FPEH

perform better at larger U . Since the FPEH can be tuned to have subcritical Hopf bifurcation, the power output is more stable than the GPEH when U varies, which is the common situation in the outdoor environment. Nevertheless, it owns the drawback of larger displacement when compared to the GPEH, leading to the requirement of trade-off for appropriate choice.

6.6 Chapter summary

This chapter demonstrates that with the easy method of adding a beam stiffener, power generation capabilities of the galloping-based, VIV-based and flutter-based aeroelastic energy harvesters are significantly increased. The principle is to increase the electromechanical coupling by modifying the fundamental mode shape of the harvester with the introduced beam stiffener. The main merit of this method is its simplicity and effectiveness. Piezoelectric transducer is employed to transfer the flow-induced vibration energy into electrical energy. Theoretical models are established for the three types of aeroelastic energy harvesters, including the electromechanical model based on the Euler-Bernoulli beam theory and Kirchhoff laws which represents the coupling effects between the structural and electrical components, as well as the aerodynamic model for the calculations of the aerodynamic force due to the specific aeroelastic phenomenon.

Wind tunnel experimental verification is performed with a prototype of galloping-based harvester at different lengths of the beam stiffener and different wind speeds. Nonlinear numerical analysis is conducted with the identified parameters to investigate the effects of the beam stiffener length and wind speed on the harvested power, transverse displacement and efficiency of flow power conversion. It shows that the maximum power can be boosted by more than a dozen times, with comparable or even smaller displacement. A maximum power of 139.74mW is achieved by the galloping-based harvester at $U=15\text{m/s}$, which is significantly higher than most of the harvesters' output reported in the literature and sufficient to power most commercial

wireless sensors currently used in real applications. For the VIV-based harvester, the lock-in region is also broadened. The flutter-based harvester is found to have the highest efficiency, and is suggested to operate in large range of wind speeds; the VIV-based harvester is recommended for environments with fixed and small wind speed, while the galloping-based harvester has much higher power output than the former two at high wind speeds. It is concluded that the addition of a beam stiffener is an effective and practical method for the enhancement of aeroelastic energy harvesting devices.

CHAPTER 7 SYNCHRONIZED CHARGE EXTRACTION IN GALLOPING PIEZOELECTRIC ENERGY HARVESTING⁶

7.1 Introduction

Aeroelasticity phenomena are the basic operating principles of the wind energy harvesters. The power generation process incorporates a series of mutual coupling behaviors, i.e., the aeroelectromechanical coupling. The nonlinear aerodynamic force induces mechanical vibrations through the fluid-structure coupling, then the mechanical strain energy is converted into electricity through the direct piezoelectric effects along with the electromechanical coupling. Moreover, due to the converse piezoelectric effects, mutual interaction between the fluid and the piezoelectric aspects is induced through the mechanical strain, although this interaction is insignificant. In the area of aeroelastic energy harvesting, the complex nonlinear aeroelastic aspect alone brings great challenges in the analysis of power conversion ability. In the literature, researches of the wind energy harvesters have mostly focused on the conversion of wind energy into electrical energy, e.g., how the wind speed and turbulence in the flow influence the oscillation frequency, amplitude and generated power level, yet have not considered the power regulation, nonlinear power extraction technique or storage of the electricity. The electric circuit is mostly simplified to include a pure resistor only as an energy dissipater, so that the complex coupling behavior is simplified in the aspect of interface circuit. In such a case, AC output is provided. However, in general, the harvested energy is supposed to be utilized in WSNs and other electronic devices (Paradiso and Starner, 2005), or to be stored in electrical energy storage systems. In either scenario, DC signal is required. Therefore, an interface circuit for AC-DC rectification and regulation needs to be implemented for wind energy harvesters before they can be

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employed in real applications. Nevertheless, involving the advanced nonlinear power extraction interface circuit further complicates the mutual coupling behaviors between the fluid, structure and electric circuit. On the other hand, for VPEH, which focus on tapping existing mechanical vibration energy, considerable efforts have been devoted to optimizing the interface circuit to boost the power output, including impedance adaption (Ottman et al., 2002, 2003), synchronized switching harvesting on inductor (SSHI) (Guyomar et al., 2005; Lallart and Guyomar, 2008; Shu et al., 2007; Lien et al., 2010; Lefeuvre et al., 2006; Liang and Liao, 2009, 2012), and synchronized charge extraction (SCE) (Lefeuvre et al., 2005, 2006, 2007; Tang and Yang, 2011, Tang et al., 2011). Interface circuits for energy storage have also been analyzed in the literature (Wickenheiser et al., 2010; Wu et al., 2009). The impedance adaption utilizes a rectifier bridge and certain DC-DC converters to enhance power generation (Ottman et al., 2002). The SSHI and SCE circuits involve a controller that switches synchronously with the harvester's vibration (Lallart and Guyomar, 2008; Liang and Liao, 2012; Lefeuvre et al., 2006). Lefeuvre et al. (2005) showed that the SCE interface increases the harvested power by 400% as compared with the impedance adaption method, and this performance is beneficially independent of load resistance. Nevertheless, the integration and analysis of these interface circuits to enhance the power output of aeroelastic energy harvesters has received far less attention. To fill this gap, this chapter focuses on the feasibility of employing the SCE interface in piezoelectric aeroelastic energy harvesters. Galloping instability is exploited here due to its aforementioned merits.

To conduct system level evaluation of the whole system, which incorporates mechanical structure of the harvester, piezoelectric transducer, the aerodynamic forces and the complex interface circuit, one should take into account all the complex coupling behaviors between the above elements. Hence, the analytical or numerical formulations will be quite cumbersome. On the other hand, the equivalent circuit model (ECM) of VPEH has been reported in the literature (Elvin and Elvin, 2009a; Yang and Tang, 2009). The very recent work by the authors (Tang et al., 2014) proposed a new

ECM for GPEH, with the aerodynamic force represented by a user-defined component (non-standard). This approach provides the possibility to evaluate the GPEH with complex interface circuits by system level circuit simulation.

This chapter investigates the feasibility and applicable region of SCE interface in the GPEH system based on the system level circuit simulation with a derived circuit representation of the whole system. The responses of the harvester are studied in detail with an emphasis on the influence of resistive load, electromechanical coupling and wind speed on the output power and the transverse displacement. Section 7.2 presents the working principle of the GPEH and SCE interface. Section 7.3 presents the experimentally validated ECM for the harvester as well as the schematic of the whole system incorporating complex interface circuits for system-level simulation. Section 7.4 provides the circuit simulation results and discussions. As a point of comparison, the performance of SCE circuit is compared with a standard circuit. It is found that the advantage of SCE circuit in the GPEH system lies in three points: enhanced power output with weak electromechanical coupling thus reducing the use of piezoelectric material, power output independent of load resistance thus avoiding impedance matching challenge, and smaller transverse displacement thus having less fatigue problem and enhancing the durability of the harvester. Moreover, a theoretical expression for the applicable region of SCE is provided. Finally, summary and recommendations are given in Section 7.5.

7.2 Galloping Piezoelectric Energy Harvester

7.2.1 Electromechanical and aerodynamic model

Typical configuration of a GPEH is given in Figure 7.1(a). A square-sectioned bluff body is connected to the free end of a cantilever, which is covered with piezoelectric laminates (Yang et al., 2013). When the wind speed is low, the harvester system is controlled by a positive linear damping and is always damped to the stationary status. With the increase of the wind speed, this linear damping decreases, and suddenly

becomes negative at a critical wind speed, which is the so-called cut in wind speed U_{cr} .

Figure 7.1 Configuration of a typical GPEH (b) schematic of GPEH geometry and cross section of composite beam for $x_1 < x < x_2$.

The vibration amplitude increases due to the negative damping, but another nonlinear damping term will limit the amplitude simultaneously and induce the final “limit cycle oscillation” (Zhao et al., 2013). Beyond the cut-in wind speed U_{cr} , the periodic galloping oscillation of the bluff body in the direction normal to the flow results in alternating strain at the root area, hence generating AC voltage in the piezoelectric elements. The connected energy harvesting interface further rectifies and regulates this AC voltage into DC voltage for practical applications like charging batteries or powering electronic devices. Yet as mentioned earlier, the interface is mostly simplified

as a pure resistor R_L in the literature for simplicity of formulation (Figure 7.1(b)), which is not realistic in application. Instead, this chapter considers more practical interface circuits. We investigate the SCE circuit and compare its performance with the standard circuit for galloping-based energy harvesting. The operating principle will be introduced in Section 7.2.2.

With the simplified circuit comprising only an R_L , several analytical models have been reported to simulate the coupling behavior between the mechanical structure, piezoelectric transducer, and the aerodynamic force. The whole electro-aero-mechanical model of the GPEH includes two parts: the electromechanical model and the aerodynamic model. The electromechanical models generally fall into two categories: the lumped parameter model (Yang et al., 2013; Abdelkefi et al., 2012) and the distributed parameter model (Abdelkefi et al., 2013b; Sirohi and Mahadik, 2011). The aerodynamic models are all established based on quasi-steady hypothesis. A comparison of the performance of these analytical models is conducted by Zhao et al. (2013) and the merits and disadvantages of respective models are discussed. Basically, both the lumped parameter model and the distributed parameter model can successfully predict the electroaeroelastic behavior of the GPEH. The main advantage of the lumped model lies in its simplicity and the ease of obtaining the coupling coefficient via test, while the benefit of the distributed model is its better representation of the aerodynamic force and ease of parametric study. We present the governing equations of the GPEH system using a single-mode distributed parameter model (Zhao et al. 2013), which is actually able to degrade into a lumped parameter model for the derivation of ECM in Section 7.3.

$$\begin{cases} \ddot{\eta}(t) + 2\zeta\omega_n\dot{\eta}(t) + \omega_n^2\eta(t) + \chi V(t) = f_a(t) \\ I(t) + C_p\dot{V}(t) - \chi\dot{\eta}(t) = 0 \end{cases} \quad (7.1)$$

$$f_a(t) = \phi(L_t) \times \frac{1}{2} \rho_a h l_{ip} U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\phi(L_t)\dot{\eta}(t)}{U} + \phi'(L_t)\eta(t) \right] \quad (7.2)$$

In Equation (7.1), $\eta(t)$ is the modal coordinate; ζ is the mechanical damping of the first vibration mode; ω_n is the undamped fundamental frequency; C_p is the total capacitance

of the piezoelectric sheet; $V(t)$ is the generated voltage across the piezoelectric sheet; $I(t)$ is the current flow into the interface circuit; and χ is the modal electromechanical coupling term written as $\chi = \theta[\phi'(x_2) - \phi'(x_1)]$ with x_1 and x_2 being respectively the starting and ending positions of the piezoelectric sheets along the beam (Figure 7.1(b)), $\phi(x)$ being the mass normalized fundamental mode shape and θ the piezoelectric coupling term determined by the geometric and material parameters of composite beam (Abdelkefi et al., 2013b; Zhao et al. 2013); $f_a(t)$ is the modal aerodynamic force formulated as Equation (7.2), where U is wind speed; A_i are empirical coefficients for the aerodynamic force calculation; and ρ_a , h and l_{tip} are air density, the frontal dimension and length of the bluff body, respectively. L_t denotes the position of the centre of the bluff body. The overdot denotes the derivative with respect to time t .

7.2.2 Interface circuits

7.2.2.1 Standard DC interface

Figure 7.2 (a) Standard interface circuit and (b) corresponding waveforms of tip displacement and voltage

A standard DC interface (Figure 7.2(a)) includes a full-wave rectifier, a filtering capacitor C_f and a resistor R_L (Lefeuvre et al., 2005). The DC voltage V_c can be regarded as constant if $C_f R_L$ is much larger than the oscillating period. When the voltage across the harvester V is less than V_c , the rectifier is blocked and outgoing current I is null; when V reaches V_c , the rectifier conducts and transfers the electrostatic

energy from the piezoelectric transducer to C_f and R_L . The displacement and voltage waveforms with a standard interface in steady state are shown in Figure 7.2(b).

7.2.2.2 SCE interface

The schematic of the SCE interface is given in Figure 7.3(a), which includes a full-wave rectifier, an inductor L_{SCE} , a diode D_{SCE} , a switch S , a filter capacitor C_f and a resistor R_L . The switch is usually controlled by an external displacement sensor and a controller to transfer energy only at the time instants when the harvester reaches its maximum (and minimum) deflection (Figure 7.3(b)). When the switch is open, the piezoelectric transducer experiences open-circuit condition and energy accumulates on its own piezoelectric capacitor. Once the structural displacement reaches its peak amplitude, the switch is closed for a short time interval and the accumulated electrostatic energy is transferred quickly from the piezoelectric transducer to L_{SCE} . The energy on the inductor L_{SCE} is further released to filter capacitor C_f and the load resistance R_L . C_f is usually set to be large to smooth the voltage signal across R_L . The waveforms of structural displacement and voltage are shown in Figure 7.3(b).

Figure 7.3 (a) SCE interface circuit and (b) corresponding waveforms of tip displacement and voltage

7.2.2.3 Self-powered SCE interface

Figure 7.4 Schematic of self-powered SCE interface circuit

To implement the SCE circuit, additional power supplier is required to power the external displacement sensor and a switch controller. In order to avoid the additional power supplier for genuine autonomy, self-powered interfaces have been enthusiastically pursued, such as the self-powered SSHI interface (Liang and Liao, 2009, 2012; Lallart and Guyomar, 2008) and self-powered SCE interface (Zhu et al., 2012; Wu et al., 2014). The key component in the self-powered interfaces is the electronic breaker proposed by Richard et al. (2007). With the electronic breaker, the integrated interface can automatically detect the displacement peak and switch on and off to extract energy from the piezoelectric transducer. The schematic of the self-powered SCE interface is shown in Figure 7.4, which consists of a full-wave rectifier, an electronic breaker and a flyback converter. Most of the time the transistor Q_1 is blocked and electrical charge is accumulated on the piezoelectric transducer. Once the peak displacement is detected, Q_1 is conducted, triggering on Q_2 and transferring the energy from the piezoelectric transducer to the inductor L_1 in the flyback converter, which temporarily stores the energy. Then the transistors are blocked again and the piezoelectric transducer accumulates energy in open circuit condition. Taking advantage of this autonomous nonlinear interface, the remaining parts of this chapter study its feasibility of boosting power extraction from wind flows by integrating it with a GPEH.

7.3 Equivalent circuit model for GPEH

7.3.1 Principle of equivalent circuit model for GPEH

For a GPEH system integrated with the complex interface circuits that have much more complex behaviors than a pure resistor, analytical formulations are quite cumbersome due to the multi-way coupling effects. On the other hand, circuit simulators such as SPICE and SIMetrix are powerful tools for system-level simulation to evaluate the coupling behavior of the whole system if the ECM of the GPEH is derived. Some studies have reported the ECM of VPEH based on the analogy between the mechanical and electrical domain parameters (Table 7.1) (Elvin and Elvin, 2009a; Yang and Tang, 2009). The vibratory excitation force can be easily represented as a standard voltage source. While in aeroelastic energy harvesting, a non-standard component should be constructed to represent the nonlinear aerodynamic force in the ECM. We employ the method proposed in our previous work of Tang et al. (2014). The analogy of parameters in the lumped parameter model is the same with those listed in Table 7.1. To link this model with the single-mode distributed parameter model presented in Section 7.2.1, we should first derive the degraded lumped parameter model. By employing $M_{eff}=1/\phi^2(L_t)$, $C_{eff}=2\zeta\omega_n/\phi^2(L_t)$, $K_{eff}=\omega_n^2/\phi^2(L_t)$, $\Theta=\chi/\phi(L_t)$, Equation (7.3) is obtained from Equations (7.1) and (7.2), where $w(L_t, t)$ is the displacement of the bluff body expressed as $w(L_t, t)=\phi(L_t)\eta(t)$; M_{eff} , C_{eff} , and K_{eff} are the effective lumped mass, damping and stiffness, respectively; and Θ is the electromechanical coupling.

$$\begin{cases} M_{eff}\ddot{w}(L_t, t) + C_{eff}\dot{w}(L_t, t) + K_{eff}w(L_t, t) + \Theta V(t) = F_a(t) = \frac{1}{2}\rho_a h l_{up} U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\dot{w}(L_t, t)}{U} + w'(L_t, t) \right]^i \\ I(t) + C_p \dot{V}(t) - \Theta \dot{w}(L_t, t) = 0 \end{cases} \quad (7.3)$$

Introducing the analogies between mechanical and electrical domain parameters, Equation (7.3) becomes

$$\begin{cases} L\ddot{q}(t) + R\dot{q}(t) + \frac{1}{C}q(t) + NV(t) = \frac{1}{2}\rho_a h l_{up} U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\dot{q}(t)}{U} + \frac{\phi'(L_t)}{\phi(L_t)}q(t) \right]^i \\ I(t) + C_p \dot{V}(t) - N\dot{q}(t) = 0 \end{cases} \quad (7.4)$$

Table 7.1 Analogy between mechanical and electrical domain parameters

Electrical parameters	Mechanical parameters
Charge $q(t)$	Tip Displacement $w(L_t, t) = \phi(L_t)\eta(t)$
Current $I(t) = \dot{q}(t)$	Tip Velocity: $\dot{w}(L_t, t) = \phi(L_t)\dot{\eta}(t)$
Inductance L	Effective mass $M_{eff} = 1/\phi^2(L_t)$,
Resistance R	Effective damping $C_{eff} = 2\zeta\omega_n/\phi^2(L_t)$
Capacitance C	Reciprocal of effective stiffness $1/K_{eff} = \phi^2(L_t)/\omega_n^2$
Ideal transformer turn ratio N	Electromechanical coupling $\Theta = \chi/\phi(L_t)$

 Figure 7.5 Equivalent circuit model for GPEH with a simple resistive load R_L

The schematic of circuit system is shown in Figure 7.5. The nonlinear aerodynamic force is represented by an user-defined arbitrary voltage source $\tilde{V}(t)$ depending on the current $\dot{q}(t)$ and charge $q(t)$ flowing in the left circuit loop. Since charge $q(t)$ cannot be the input variable for $\tilde{V}(t)$, we use $V_C(t)$ (the voltage across C) alternatively and $\tilde{V}(t)$ is then defined as $\tilde{V}(t) = -\frac{1}{2}\rho_a h l_{up} U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\dot{q}(t)}{U} + \frac{\phi'(L_t)}{\phi(L_t)} C V_C(t) \right]$, where the product of C and $V_C(t)$ equals to $q(t)$.

7.3.2 Schematic for system-level circuit simulation

Figure 7.6 Overall circuit diagram of GPEH system with standard interface

Figure 7.7 Overall circuit diagram of GPEH system with self-powered SCE interface

The overall diagram of the equivalent circuit representation of GPEH integrated with the standard interface is shown in Figure 7.6, which is obtained by replacing the pure resistor R_L in Figure 7.5 with the standard interface in Figure 7.2(a). The filter capacitor C_f is set to be $100\mu\text{F}$. Similarly, replacing R_L in Figure 7.5 with the self-powered SCE interface in Figure 7.4 yields the entire schematic of GPEH-self-powered-SCE system, as shown in Figure 7.7. When the electronic breaker detects the peak displacement amplitude, Q_1 and Q_2 quickly conduct. The accumulated electrical charge accumulated on C_p is instantly transferred to L_1 through D_3 and Q_2 . When Q_2 conducts, C_p and L_1

constitutes an LC circuit, with the resonant period T_0 given as

$$T_0 = 2\pi\sqrt{L_1 C_p} \quad (7.5)$$

The energy transferring duration, or the conduction duration of Q_2 should be close to a quarter of this resonant period. During simulation, L_1 is set to be 47mH. With C_p in the magnitude of nF, it is ensured that $T_0 \ll T$, where T is fundamental oscillation period of the harvester, thus the energy transferring process happens instantly. C_f is chosen to be 10 μ F which is large enough to smooth the voltage across R_L . The power consumption in the electronic breaker can be lowered down to <5% of the harvested power (Liang and Liao, 2009; Lallart and Guyomar, 2008; Richard et al., 2007), thus this power discount is not considered in this chapter. The entire GPEH system with the two interfaces is simulated in the circuit simulator SIMetrix. During the circuit simulation, a differential voltage probe provided in the simulator is employed to obtain the voltage across R_L and to evaluate the power.

7.3.3 Experimental setup

Before conducting the system-level simulation with the ECM, wind tunnel test is conducted to validate the ECM model with different interfaces, using a fabricated GPEH prototype shown in Figure 7.8(a). A square-sectioned bluff body made from polystyrene foam is connected to an aluminum cantilever, which is bonded with a piece of piezoelectric sheet (MFC M2814-P2, Smart Materials Corp.) at the root. The root of the cantilever is fixed to a rigid metal support. The mass of the bluff body is 1.8g, and the dimensions are 20mm $\times$ 20mm $\times$ 100mm. With the experiment temperature being 20 $^{\circ}$ C, the air density is set to be 1.204kg/m 3 . ζ is determined to be 0.011 using the logarithmic decrement technique (Yang et al., 2013). Short circuit fundamental frequency is determined to be 16.67Hz. Other properties of the GPEH prototype are listed in Table 7.2. The entire wind tunnel experiment setup is shown in Figure 7.8(b). The piezoelectric electrodes are connected to the SCE harvesting circuit, with a resistive load R_L connected in the terminal. Standard circuit and the simple AC circuit are easily obtained by removing the relative electronic components from the electric

board. The voltage signal across the piezoelectric sheet and R_L is monitored by the NI 9229 data acquisition module (National Instruments). The wind speed is measured with a hotwire anemometer.

Table 7.2 Properties of GPEH prototype

Properties	Beam	Piezoelectric sheet
Length L_s, L_p (mm)	130	28
Width b_s, b_p (mm)	20	14
Thickness h_s, h_p (mm)	0.6	0.3
Material	Aluminum	MFC M2814-P2
Mass Density (kg m^{-3})	2700	5440
Capacitance C_p (nF)	--	25.7
Young's Modulus (GPa)	69	30.336

Figure 7.8 Close-up view of fabricated GPEH prototype in wind tunnel experiment (a)
Entire experiment setup used for validation of equivalent circuit model.

7.4 Results and Discussion

7.4.1 Experimental results

Figure 7.9 (a) Measured and predicted P_{ave} versus R_L at $U=4$ m/s (b) Measured and predicted P_{ave} versus U at optimal R_L .

The wind tunnel experiment is first carried out with the simple AC circuit. During the test, the average power is calculated by $P_{ave} = (V_{RMS})^2 / R_L$, where V_{RMS} is the root-mean-square voltage across R_L . Circuit simulation is conducted using the ECM with parameters indicated in Figure 7.5, which are obtained with the analogy in Table 7.1. The empirical aerodynamic coefficients A_i ($i=1,2,3$) are taken as $A_1=2.3$, $A_2=0$, and $A_3=-18$ (Barrero-Gil et al., 2010). The measured and predicted variations of average power as a function of load resistance are plotted in Figure 7.9(a). The circuit simulation successfully predicts the trend and determines accurately the optimal load resistance of 333 k Ω where the maximum power is achieved. It should be noted that once a unique peak power is detected at a specific wind speed (i.e. 4 m/s), the corresponding optimal R_L is safe to be employed for other wind speeds, as the value of optimal R_L for a galloping-based energy harvester with simple AC circuit remains constant with various wind speeds (Zhao et al., 2013). The measured and predicted

average power outputs over the range of wind speed are compared in Figure 7.9(b), where good correlation is also observed.

Experiment and circuit simulation are further carried out with the standard and SCE circuits. Figures 7.10(a) and 7.10(b) show the steady state response of voltage across the piezoelectric sheet as well as the terminal resistive load R_L in the SCE circuit, which are obtained from circuit simulation and experiment measurement, respectively, with a wind speed of 4m/s and a terminal resistive load of 665 k Ω . The circuit simulation gives a stable amplitude of steady state voltage across the piezoelectric sheet, while for the experimentally measured responses, there appears a fluctuation component in the amplitude which is due to the inevitable flow turbulence in the wind tunnel. A stable DC voltage across R_L is measured from both simulation and experiment. The amplitudes of the piezoelectric voltage and DC voltage from experiment are a bit smaller compared to those from simulation. Enlarged views of the voltage profile across the piezoelectric sheet are given at the bottom of Figure 7.10(a) and 7.10(b), showing that both simulation and experiment obtain the same voltage pattern as that expected in Figure 7.3(b). Results of the standard circuit are depicted in Figure 7.10(c) and 7.10(d), with good correlations observed in the voltage amplitudes and patterns.

Figure 7.10 Steady state responses of piezoelectric voltage and terminal DC voltage for (a)(b) SCE circuit and (c)(d) standard circuit. (a)(c) Circuit simulation; (b)(d) wind tunnel experiment.

Figures 7.11(a) and 7.11(b) give the measured and predicted power responses for the SCE and standard circuit, at two different wind speeds of 4m/s and 5m/s, respectively. “ $P_{\text{piezo, SP-SCE}}$ ” represents the total extracted power from the piezoelectric sheet in the employed self-powered SCE circuit, calculated by $P_{\text{piezo, SP-SCE}} = C_p(V_m)^2/T$, where “ V_m ” is the amplitude of steady state voltage across the piezoelectric sheet; “ $P_{RL, \text{SP-SCE}}$ ” and “ $P_{RL, \text{Standard}}$ ” represent the consumed power on the terminal resistive load R_L in the self-powered SCE circuit and standard circuit, respectively, where P_{RL} is calculated with the output DC voltage V_c as $P_{RL} = (V_c)^2/R_L$. In Figure 7.11(a), at $U=4\text{m/s}$, both simulation and experiment show that a constant extracted power of 0.41mW from the piezoelectric sheet is obtained with varying load resistance for the self-powered SCE circuit, whereas for the standard circuit, the power is measured to reach the maximum of 0.34mW at the optimal R_L of 411k Ω . Although the extracted power from the piezoelectric sheet with the self-powered SCE circuit is slightly higher than the maximum power of the standard circuit, the output power (consumed power on R_L) is lower, measured to be around 0.27mW. The energy loss is due to the power consumption of the electronic breaker, components’ voltage conversion loss and other imperfections of the circuit. When the wind speed is increased to 5m/s, the predicted

and measured extracted power remains constant, but the measurement gives a lower power of 1.14mW than the predicted result of 1.49mW. This should be attributed to the internal energy losses on the piezoelectric leakage resistance (Lefeuvre et al., 2007). It can be seen that when the aerodynamic force is increased, this piezoelectric energy loss becomes severe. The measured output power P_{RL} is decreased especially at low and high range of load resistance restricted by the imperfections of circuit components, but within the range of 50~300k Ω the measured output power is generally stable at around 0.9mW, which is higher than the maximum power from the standard circuit, with a practical efficiency of around 80% compared to the total extracted power. It should be noted that although in circuit simulation the energy loss due to device imperfections are neglected, the electronic breaker still consumes energy to function, causing a power difference between P_{piezo} , SP-SCE and P_{RL} , SP-SCE. Yet this loss is limited to a small portion of P_{piezo} , SP-SCE when the excitation is high enough, as reflected in Figure 7.11(b), maintaining the advantage of electronic breaker with its auto-detection and auto-control capabilities. The amplitudes of steady state voltage across the piezoelectric sheet from the two circuits are compared in Figures 7.11(c) and 7.11(d). The self-powered SCE circuit generates a constant voltage magnitude with various values of R_L , while the voltage magnitude from the standard circuit increases with R_L . The voltage magnitude from the self-powered SCE circuit is always higher than that of the standard circuit. In general, the circuit simulation successfully captures the trend and magnitude of the responses of power and voltage, providing a quick and convenient method for parametric study and performance analysis under various conditions. In the following sections, the powers of the self-powered SCE circuit and the standard circuit both refer to the final output power consumed on R_L .

Figure 7.11 Average power versus load resistance: (a) 4m/s (b) 5m/s; Peak piezoelectric voltage versus load resistance (c) 4m/s (d) 5m/s.

7.4.2 Comparison of power output

In this section, we investigate the feasibility of the self-powered SCE interface in GPEH via circuit simulation, by comparing its performance to the standard interface. The parameter k_e^2/ζ is employed to indicate the strength of electromechanical coupling (Shu and Lien, 2006; Tang and Yang, 2011), where k_e is the dimensionless electromechanical coupling coefficient, defined as $k_e^2 = \chi^2 / \omega_n^2 C_p$. k_e^2/ζ is proven to be the critical parameter to indicate the applicable region of SCE interface for VPEH (Tang and Yang, 2011). However, for GPEH, it is not sufficient to determine the applicability of SCE circuit as the wind speed has a great influence on power generation, which is reflected in Figure 7.11. A criteria formula will be proposed to identify the applicability of SCE in the GPEH system.

With the corresponding parameters of the prototype in Section 7.3, k_e^2/ζ is obtained as 1.0326. During circuit simulation, k_e^2/ζ is varied by simultaneously magnifying C_p and N (or Θ) (i.e., magnifying both two parameters by n times with respect to their original values). ζ is fixed to be 0.011. Thus k_e^2/ζ will be altered to be n times of 1.0326 ($n=0\sim10$). A larger value of k_e^2/ζ represents a stronger electromechanical coupling. With different values of U , R_L and k_e^2/ζ , simulation is conducted to obtain the steady state response and the average power can be calculated with the measured voltage from the differential voltage probe.

(a)

(b)

(c)

Figure 7.12 Variations of average power output with load resistance and electromechanical coupling (a) $U=5$ m/s (b) $U=7$ m/s (c) $U=9$ m/s

The responses of average power over the range of load resistance and electromechanical coupling for the self-powered SCE interface and standard interface are compared in Figure 7.12. Three different wind speeds are considered. First of all, with the increase of U , the peak power values achieved by both circuits increased.

Secondly, along the axis of R_L , the power output from self-powered SCE interface is generally uniform at each k_e^2/ζ value. On the other hand, one optimal load resistance R_{Lopt}^1 exists for the standard interface when k_e^2/ζ is below the optimal value $(k_e^2/\zeta)_{STD}$. Beyond $(k_e^2/\zeta)_{STD}$, R_{Lopt}^1 corresponds to the minimum output power instead, and two different optimal load resistances R_{Lopt}^2 and R_{Lopt}^3 branch out. The value of R_{Lopt}^2 gradually decreases with k_e^2/ζ , while the value of R_{Lopt}^3 increases with k_e^2/ζ . In addition, R_{Lopt}^2 and R_{Lopt}^3 generate an equal value of maximum output power. It is noted that the value of R_{Lopt}^1 slightly shifts to the left, i.e., becomes smaller, since the oscillating frequency of the harvester is affected by the coupling. The behavior that power output with self-powered SCE interface is independent of R_L leads to the advantage that no additional impedance matching module is required, thus making much convenient the installation and adjustment of the harvesting system. Moreover, along the axis of k_e^2/ζ , two different behaviors are observed for the two circuits. The power from self-powered SCE interface first increases with k_e^2/ζ up to the optimal value $(k_e^2/\zeta)_{SCE}$, then decreases when k_e^2/ζ further increases. Nevertheless, the optimal output power (power at the optimal load resistance) for the standard interface continuously grows with k_e^2/ζ until the optimal coupling $(k_e^2/\zeta)_{STD}$, then enters the saturation region. Beyond $(k_e^2/\zeta)_{STD}$, in a counterintuitive manner, power remains unchanged even though k_e^2/ζ increase. It is noted that for each wind speed, the attainable maximum values of output power from the self-powered SCE and standard interface are equal. Observing the three figures with different wind speeds, it is also noted that $(k_e^2/\zeta)_{SCE}$ and $(k_e^2/\zeta)_{STD}$ are not constant, but increase with U . In other words, the coupling in GPEH increases with the wind speed. This is an important and different phenomenon from the case for VPEH, where the two optimal coupling values $(k_e^2/\zeta)_{SCE}$ and $(k_e^2/\zeta)_{STD}$ stay unchanged when the base excitation acceleration varies. It should be noted that in Figure 7.12(c), when k_e^2/ζ further increases, R_{Lopt}^2 and R_{Lopt}^3 will also appear.

Figure 7.13 Variation of maximum output power with k_e^2/ζ for self-powered SCE and standard circuit at different wind speeds.

Figure 7.13 shows the effect of k_e^2/ζ on the responses of maximum output power from the self-powered SCE for different wind speeds, compared to those from the standard circuit. The power value for standard circuit refers to the power at the optimal load R_{Lopt}^1 or R_{Lopt}^2 (R_{Lopt}^3) as denoted in Figure 7.12. It is clear that power from SCE first increases with k_e^2/ζ , reaches the maximum at $(k_e^2/\zeta)_{SCE}$ (indicated with black circles), then decreases with k_e^2/ζ . Yet for the standard circuit, the optimal power increases first, then saturates beyond $(k_e^2/\zeta)_{STD}$ (indicated with black boxes). Both $(k_e^2/\zeta)_{SCE}$ and $(k_e^2/\zeta)_{STD}$ increase with wind speed U , in consistent with Figure 7.12. Also, it is seen that the self-powered SCE and standard circuits provide the same amount of maximum power, but the self-powered SCE always achieves this maximum with only one quarter of k_e^2/ζ compared to the standard circuit for all wind speeds, that is, $(k_e^2/\zeta)_{SCE} = 1/4 * (k_e^2/\zeta)_{STD}$ for all the considered U . For example, with $U=7\text{m/s}$, the maximum power is achieved at $k_e^2/\zeta=1.5495$ for the self-powered SCE, while the same maximum is obtained at $k_e^2/\zeta=6.1956$ for the standard circuit. This reveals another

important merit of SCE. Introducing the expression of χ and θ into k_e^2 , it becomes

$$k_e^2 = \frac{\left\{ -Y_p d_{31} (h_p + h_s) [\phi'(x_2) - \phi'(x_1)] \right\}^2}{\omega_n^2} \times \frac{b_p^2}{C_p} \quad (7.6)$$

with the coupling term θ expressed as $\theta = -Y_p d_{31} b_p (h_p + h_s)$, where Y_p , d_{31} , b_p , h_p and h_s are the piezoelectric Young's modulus, piezoelectric constant, width of the piezoelectric sheet, thickness of the piezoelectric sheet and thickness of the substrate, respectively. If the mechanical structure remains unchanged, k_e^2 is proportional to b_p^2/C_p , i.e., if n pieces of piezoelectric sheets are connected in parallel simultaneously, b_p and C_p are increased by a factor of n , increasing k_e^2 by a factor of n , too. Therefore, k_e^2 is proportional to the amount of piezoelectric materials (about the arrangement of attached piezoelectric sheets, readers can refer to the work of Lefeuvre et al., (2005)). Thus, with the SCE circuit, the use of piezoelectric materials can be saved by 75%. Furthermore, k_e^2/ζ at the intersection point of the curves, $(k_e^2/\zeta)_{\text{INT}}$ (the bold dashed loci in Figure 7.13), is the critical k_e^2/ζ indicating the boundary of applicability of SCE. Below $(k_e^2/\zeta)_{\text{INT}}$, SCE outperforms standard circuit with higher attainable power. To find the relations between $(k_e^2/\zeta)_{\text{INT}}$ and U and specify the exact applicable region of SCE, the power responses against k_e^2/ζ are simulated for more values of U .

Figure 7.14 Variation of $(k_e^2/\zeta)_{\text{INT}}$ as a function of $\rho_a h l_{tip} U / \omega_n M_{eff} \zeta$.

A linear relationship is observed between $(k_e^2/\zeta)_{INT}$ and the dimensionless parameter $\rho_a h l_{tip} U / \omega_n M_{eff} \zeta$, as plotted in Figure 7.14. In the area below the straight line (shaded area), the self-powered SCE outperforms the standard circuit with higher output power; while in the area above the straight line, the self-powered SCE underperforms the standard circuit. According to Figure 7.14, the following linear expression of $(k_e^2/\zeta)_{INT}$ can be proposed to estimate the applicability of SCE for GPEH.

$$\left(\frac{k_e^2}{\zeta} \right)_{INT} = \frac{2\pi}{5} \left(\frac{1}{4} A_1 \frac{\rho_a h l_{tip} U}{\omega_n M_{eff} \zeta} - 1 \right) \quad (7.7)$$

For $\frac{k_e^2}{\zeta \left(\frac{1}{4} A_1 \frac{\rho_a h l_{tip} U}{\omega_n M_{eff} \zeta} - 1 \right)} < 1.2566$, the SCE circuit outperforms standard circuit; and

for $\frac{k_e^2}{\zeta \left(\frac{1}{4} A_1 \frac{\rho_a h l_{tip} U}{\omega_n M_{eff} \zeta} - 1 \right)} \geq 1.2566$, SCE underperforms the standard circuit.

Alternatively, we can define an overall equivalent damping ratio as $\zeta_{eq} = \zeta \left(\frac{1}{4} A_1 \frac{\rho_a h l_{tip} U}{\omega_n M_{eff} \zeta} - 1 \right)$. In such a way, the applicable criterion of SCE in GPEH

can be written in a similar form with the case in VPEH, as that for $\frac{k_e^2}{\zeta_{eq}} < 1.2566$, SCE

outperforms the standard circuit, while for $\frac{k_e^2}{\zeta_{eq}} \geq 1.2566$, SCE underperforms the standard circuit.

7.4.3 Comparison of transverse displacement

(a)

(b)

(c)

Figure 7.15 Variations of tip displacement with load resistance and electromechanical coupling (a) $U=5\text{m/s}$ (b) $U=7\text{m/s}$ (c) $U=9\text{m/s}$

Figure 7.16 Variation of transverse displacement amplitude with k_e^2/ζ for self-powered SCE and standard circuit (achieved at optimal R_L) at different wind speeds.

Figure 7.15 compares the responses of tip displacement amplitude $w(L_t)$ over the range of load resistance and electromechanical coupling for the self-powered SCE and standard interfaces. First of all, as expected, $w(L_t)$ increases with U for both interfaces since higher wind speed leads to larger aerodynamic force. Secondly, along the axis of R_L , the self-powered SCE interface results in a generally uniform displacement amplitude when k_e^2/ζ changes, similar to the response of output power. As for the standard interface, the displacement amplitude exhibits a single minimum with an optimal load resistance, which always corresponds to R_{Lopt}^1 for the considered k_e^2/ζ region. Figure 7.16 presents the variation of $w(L_t)$ as a function of k_e^2/ζ at optimal R_L . It is worth noting that $w(L_t)$ obtained with the self-powered SCE is always smaller than that with the standard interface, especially when $k_e^2/\zeta > 6$. This reveals the stronger electrical damping of self-power SCE than the standard interface. In practice, smaller displacement amplitude is preferred as it results in smaller mechanical strain endured by the piezoelectric material and the substrate. Therefore, given the same power output, the fatigue problem can be alleviated with the SCE circuit as it has smaller $w(L_t)$ compared to the standard circuit. The durability of the harvesting system can thus be enhanced.

7.4.4 Comparison of cut-in wind speed

Figure 7.17 Variation of average power output with wind speed for different electromechanical coupling cases

Figure 7.18 Variation of cut-in wind speed with k_e^2/ζ for self-powered SCE and standard circuit

Figure 7.17 shows the variation of average power as a function of wind speed for four different values of k_e^2/ζ . It is observed that for various k_e^2/ζ , the self-powered SCE interface owns higher cut-in wind speed U_{cr} than the standard interface. For both interfaces, stronger coupling results in higher cut-in wind speed as U_{cr} increases with k_e^2/ζ . Also, the slope of the power response curve, which indicates the growing rate of output power, increases with k_e^2/ζ . The slope of the self-powered SCE interface is always larger than that of the standard, which implies that higher U is more beneficial to the self-powered SCE circuit. The variation of U_{cr} with k_e^2/ζ in Figure 7.18 confirms the increasing trend of U_{cr} with respect to k_e^2/ζ for both circuits. If k_e^2/ζ is maintained within the applicability region proposed in Section 7.4.2, the augment of U_{cr} is acceptable. For example, with $k_e^2/\zeta=1.50$, U_{cr} is 4.05m/s for the SCE and 2.95m/s for the standard, but the power is 5.99mW for SCE and 2.45mW for standard at $U=7$ m/s, respectively, that is, the SCE increases the power by 144.5%. At $U=9$ m/s, the power is 11.45mW for the SCE and 3.84mW for the standard, i.e., the SCE increases the power by 198.2%.

7.5 Chapter summary

This chapter investigates the feasibility of SCE interface in enhancing the performance of GPEH system. The basic working principles of the GPEH structure and several frequently used interfaces are illustrated with a focus on the self-powered SCE interface, where the key part is the self-detecting and self-control electronic breaker. Subsequently, the experimentally validated ECM for GPEH is employed and the whole circuit diagram of GPEH integrated with the self-powered SCE interface is derived to conduct the system level circuit simulation. The influences of resistive load, electromechanical coupling and wind speed on the output power and the transverse displacement are studied in detail. The performance of SCE circuit is compared with a standard circuit. The main findings can be concluded as follows.

- Output power: over the considered range of R_L , the output power generated by the self-powered SCE interface is uniform and independent of the value of R_L ; while for standard circuit the power achieves a single maximum at one optimal load resistance R_{Lopt}^1 for small k_e^2/ζ below $(k_e^2/\zeta)_{STD}$, and branches out beyond $(k_e^2/\zeta)_{STD}$ at R_{Lopt}^2 and R_{Lopt}^3 which produce the same amount of optimal power, with R_{Lopt}^1 becoming a minimum. R_{Lopt}^1 , R_{Lopt}^2 and R_{Lopt}^3 varies depending on k_e^2/ζ .
- Transverse displacement and cut-in wind speed: the tip displacement amplitude with the self-powered SCE interface is also uniform and independent of R_L at any value of k_e^2/ζ , similar to the output power; while the displacement amplitude with the standard circuit exhibits a single minimum at R_{Lopt}^1 even beyond $(k_e^2/\zeta)_{STD}$. For both circuits, increasing k_e^2/ζ reduces the displacement amplitude. The self-powered SCE is always associated with a smaller displacement and higher cut-in wind speed than the standard circuit, which is due to the stronger electromechanical damping induced by the nonlinear SCE interface.
- Attainable maximum power: The maximum power from the SCE first increases with k_e^2/ζ , then decreases with a peak achieved at $(k_e^2/\zeta)_{SCE}$; while for the standard circuit, the maximum power increases with k_e^2/ζ until $(k_e^2/\zeta)_{STD}$ then saturates. Both $(k_e^2/\zeta)_{SCE}$ and $(k_e^2/\zeta)_{STD}$ increases with wind speed U . At a specific U , the values of attainable maximum power are equal for the two circuits at their corresponding optimal k_e^2/ζ , but the self-powered SCE has a smaller optimal coupling as $(k_e^2/\zeta)_{SCE} = 1/4 * (k_e^2/\zeta)_{STD}$ for all considered values of U .
- Applicable region of SCE in GPEH: A criterion is proposed to determine the applicability of SCE based on the plots of the intersection coupling-damping ratio $(k_e^2/\zeta)_{INT}$ from the maximum power response curves. For

$$\frac{k_e^2}{\zeta \left(\frac{1}{4} A_1 \frac{\rho_a h l_{tip} U}{\omega M_{eff} \zeta} - 1 \right)} > 1.2566, \text{ the SCE circuit is capable of enhancing output power}$$

compared to the standard circuit; otherwise, the standard circuit is a better choice.

- Advantage of SCE in GEPH: It is summarized in this study that the advantages of SCE in GPEH lie in three main points. First, the output power from SCE is independent

of R_L thus no additional impedance matching process is required; second, the SCE circuit saves piezoelectric materials by 75% compared to the standard circuit since it achieves the equal maximum power at a much smaller k_e^2/ζ ; third, the smaller displacement amplitude with SCE alleviates the fatigue problem and enhances the durability of the harvesting system.

CHAPTER 8 ANALYTICAL SOLUTIONS FOR GALLOPING-BASED PIEZOELECTRIC ENERGY HARVESTERS WITH VARIOUS INTERFACING CIRCUITS⁷

Nomenclature

A_i	=	Empirical aerodynamic coefficients
C	=	Effective lumped damping
C_0	=	Capacitance of the capacitor in the equivalent circuit model
C_f	=	Filtering capacitor
C_p	=	Piezoelectric capacitance
D_{SCE}	=	Diode in the SCE circuit
f_a	=	Modal aerodynamic force
F_a	=	Aerodynamic force
h	=	Frontal height of the bluff body
I	=	Current across the piezoelectric sheet
K	=	Effective lumped stiffness
k_e^2	=	Alternative electromechanical coupling coefficient
$(k_e^2)^{opt}$	=	Optimal electromechanical coupling to achieve maximum normalized average power
L	=	Length of the bluff body
L_0	=	Inductance of the inductor in the equivalent circuit model
L_{SCE}	=	Inductor in the SCE circuit
L_t	=	Distance from the centre of the bluff body to the fixed end of the cantilever
M	=	Effective lumped mass
$\hat{M}$	=	Dimensionless mass ratio
N	=	Turn ratio of the ideal transformer in the equivalent circuit model
P_{ave}	=	Average power
$\hat{P}_{ave}$	=	Normalized average power
$\hat{P}_{ave}^{max}$	=	Maximum normalized average power at optimal load resistance
$\hat{P}_{ave}^{opt}$	=	Normalized average power at optimal load resistance
q	=	Charge in the equivalent circuit model
$\dot{q}$	=	Current in the equivalent circuit model
R	=	Load resistance
R_0	=	Load resistance in the equivalent circuit model
r	=	Normalized load resistance
r^{opt}	=	Optimal load resistance
S	=	Switch
U	=	Wind speed

⁷ Content based on this chapter is accepted for publication as “Zhao, L. and Yang, Y. (2015) Analytical solutions for galloping-based piezoelectric energy harvesters with various interfacing circuits, Smart Materials and Structures (accepted)”.

$\hat{U}$	=	Reduced wind speed
U_{cr}	=	Cut-in wind speed
$\hat{U}_{cr}$	=	Normalized cut-in wind speed
u	=	Displacement of the bluff body
$\dot{u}$	=	Vibration velocity of the bluff body
u_m	=	Magnitude of displacement of the bluff body
$\hat{u}_m$	=	Normalized magnitude of displacement of the bluff body
V	=	Voltage across the piezoelectric sheet
$\tilde{V}$	=	User-defined arbitrary voltage source in the equivalent circuit model
V_c	=	Output DC voltage
$\hat{V}_c$	=	Normalized output DC voltage
$\hat{V}_c^{opt}$	=	Normalized output DC voltage at optimal load resistance
V_{C_0}	=	Voltage across the capacitor C_0
V_m	=	Magnitude of voltage across the piezoelectric sheet

Greek symbols

β	=	Dimensionless coefficient, as $\phi'(L_t)/\phi(L_t)$
ζ	=	First mode damping ratio
η	=	Modal coordinate
Θ	=	Effective lumped electromechanical coupling
θ	=	Electromechanical coupling term
ρ	=	Air density
ϕ	=	Phase difference between voltage across the piezoelectric sheet and displacement of the bluff body
$\phi(x)$	=	Mass normalized fundamental mode shape
χ	=	Modal electromechanical coupling term
Ω	=	Normalized oscillating frequency
ω	=	Oscillating frequency
ω_n	=	Undamped fundamental frequency

Subscripts

sc	=	Short circuit condition
oc	=	Open circuit condition
AC	=	AC circuit
STD	=	Standard circuit
SCE	=	SCE circuit

8.1 Introduction

As explained in Section 7.1, bringing in the sophisticated nonlinear power extraction interface circuit would induce greatly complex mutual coupling behaviors. Chapter 7 has shown that the employment of the SCE circuit can eliminate the requirement of impedance matching, help to save piezoelectric materials by 75% compared to the

standard circuit and alleviate the fatigue problem with decreased transverse displacement. Moreover, the preliminary experimental study of Zhao et al. (2015b) showed that the parallel SSHI interface increased the output power of a GPEH prototype by 43.75% as compared to the standard circuit, boosting the output power from 1.6mW to 2.3mW at 6m/s. This chapter will further derive the analytical solutions of mechanical and electrical responses of GPEH with an SCE circuit along with the simple AC and standard circuit cases.

To accurately evaluate the electrical performance of energy harvesters and to efficiently enhance the efficiency with parameter optimizations, it is of great importance to obtain the accurate analytical expressions of the power, voltage and mechanical displacement responses. For VPEHs, there exist considerable published results for the analytical solutions of outputs with different circuits. For VPEH with the AC circuit, the lumped parameter model (Dutoit et al., 2005), Rayleigh-Ritz discretization model (Elvin and Elvin, 2009a) and distributed parameter model (Erturk and Inman, 2008a) are readily available. For VPEH with complex interface circuits, researchers have proposed some assumptions to simplify the formulation. For example, Ottman et al. (2002) provided the analytical formula for the response of impedance adaptation circuit under the uncoupled condition, which assumes that the mechanical vibration is not affected by the backward coupling force. This assumption is exclusively valid when the electromechanical coupling is weak. The work of Guyomar et al. (2005), Lefeuvre et al. (2005, 2006, 2007) and Wickenheiser et al. (2010) evaluated the performance of SSHI, SCE and energy storage circuit with the in-phase assumption, where the periodic excitation force and the velocity of the proof mass are in phase. Predictions based on the in-phase assumption are accurate for VPEHs operating at resonance. However, this assumption is no longer valid when the driving frequency deviates from the resonance. Lallart et al. (2010) developed a new modeling approach considering the conduction angle of voltage inversion in the standard and SSHI circuits, which gave accurate predictions for both resonance and out-of-resonance conditions for an acoustic piezoelectric energy harvester. Taking into account the global behavior of the entire

VPEH system, the accurate analytical solutions have been derived to evaluate the output responses of the standard circuit (Shu and Lien, 2006), the SSHI circuit (Shu et al., 2007) as well as the SCE circuit (Tang and Yang, 2011). Nevertheless, the accurate analytical solutions for aeroelastic energy harvesters are far less developed. The power responses of the above mentioned aeroelastic energy harvesters have been mostly obtained via numerical simulations. Only a few studies provided the explicit analytical solutions. Among these, Barrero-Gil et al. (2010) presented the analytic formula of galloping amplitude and energy conversion factor based on the Krylov-Bogoliuvov method without considering the electromechanical coupling; Bibo and Daqaq (2014) derived the analytical solutions for a GPEH with the backward coupling effect considered as viscous damping; Abdelkefi et al. (2013a) derived a nonlinear form of Hopf bifurcation for a GPEH. All these solutions are applicable only for the AC circuit with a pure resistor. To our best knowledge, no analytical solutions are available for GPEH with DC output (standard, SSHI or SCE) circuits.

In order to accurately evaluate the electric performance of GPEHs with both AC and DC output circuits, this work studies the steady-state response of GPEH connected to three circuits: the standard AC circuit (shorted as AC), the standard DC circuit (shorted as standard), and the SCE circuit. Theoretical solutions of output responses including power, voltage and mechanical displacement are derived based on the energy balance method in Section 8.2, taking into account the mutual coupling behaviors between the mechanical, piezoelectric, aerodynamic and electrical components. The output responses are explicitly dependent on parameters including mass, damping, stiffness, aerodynamic coefficients, wind speed, etc. Subsequently, the criterion of optimal coupling and load are derived for optimal power generation in Section 8.3 and the cut-in wind speeds for different circuits are formulated in Section 8.4. In Section 8.5, experiment and circuit simulation are conducted to validate the analytical solutions. Based on the validated analysis, comparisons of performances of the considered circuits are provided and discussed. Moreover, several guidelines for applications are recommended. Finally, conclusions are given in Section 8.6.

8.2 Analytical Solutions

Figure 8.1 Configuration of a typical GPEH system

A typical GPEH consists of a cantilever with a certain sectioned bluff body connected to its free end and piezoelectric laminates bonded to its root (Yang et al., 2013). When subjected to wind flows, galloping oscillation occurs, causing alternating strain in the piezoelectric sheet and producing AC voltage via direct piezoelectric effect. An electrical interface circuit is connected to the piezoelectric sheets to transfer, regulate or store the electrical charge. The single-mode distributed parameter model of a GPEH is depicted as Equations (8.1) and (8.2) (Abdelkefi et al., 2013b; Zhao et al., 2013):

$$\ddot{\eta} + 2\zeta\omega_n\dot{\eta} + \omega_n^2\eta + \chi V = f_a \quad (8.1)$$

$$I + C_p\dot{V} - \chi\dot{\eta} = 0 \quad (8.2)$$

$$f_a = \phi(L_t) \times \frac{1}{2} \rho h L U^2 \sum_{i=1,2,\dots} A_i \left[\frac{\phi(L_t)\dot{\eta}}{U} + \phi'(L_t)\eta \right]^i \quad (8.3)$$

where η , ζ , ω_n , C_p , and χ are the modal coordinate, first mode damping, undamped fundamental frequency, piezoelectric capacitance and modal electromechanical coupling term, respectively; V and I are the voltage and current across the piezoelectric sheet; f_a is the modal aerodynamic force expressed as Equation (8.3), where U , A_i , ρ , h ,

L and L_t are the wind speed, empirical aerodynamic coefficients, air density, frontal height and length of the bluff body, and the distance from the centre of the bluff body to the fixed end, respectively; and $\phi(x)$ is the mass normalized fundamental mode shape. In addition, χ can be easily obtained once $\phi(x)$ is calculated, depicted as $\chi = \theta[\phi'(x_2) - \phi'(x_1)]$, where the coupling term θ is determined by the geometric and material parameters of composite beam (Abdelkefi et al., 2013b; Zhao et al. 2013). The overdot represents the derivative with respect to time t . The whole geometry of the GPEH structure is shown in Figure 8.2(a).

Figure 8.2 (a) Schematic of geometry and (b) equivalent lumped parameter model of GPEH

Since the GPEH is determined to be always operating close to the first vibration mode (Zhao et al. 2013; Sirohi and Mahadik, 2011), the single-mode distributed parameter model can be degraded to a similar form of lumped parameter model as

$$M\ddot{u} + C\dot{u} + Ku + \Theta V = F_a = \frac{1}{2} \rho h L U^2 \left[A_1 \left(\frac{\dot{u}}{U} + \beta u \right) + A_3 \left(\frac{\dot{u}}{U} + \beta u \right)^3 \right] \quad (8.4)$$

$$I + C_p \dot{V} - \Theta \dot{u} = 0 \quad (8.5)$$

where the lumped parameters are linked to those in Equations (8.1) to (8.3) as

$$M = \frac{1}{[\phi(L_t)]^2} \quad (8.6)$$

$$C = \frac{2\zeta\omega_n}{[\phi(L_t)]^2} \quad (8.7)$$

$$K = \frac{\omega_n^2}{[\phi(L_t)]^2} \quad (8.8)$$

$$\Theta = \frac{\chi}{\phi(L_t)} \quad (8.9)$$

$$u = \eta \phi(L_t) \quad (8.10)$$

where M , C , K , Θ and u denote the effective lumped mass, damping, stiffness, coupling and displacement of the bluff body, respectively. β is a dimensionless coefficient expressed as $\beta = \phi'(L_t)/\phi(L_t)$. The first three aerodynamic coefficients are sufficient to represent the aerodynamic force with $A_2=0$ for the commonly employed bluff bodies (Barrero-Gil et al., 2010). This lumped parameter model has equal effectiveness with the distributed parameter model since it is degraded from the distributed parameters. For convenience of formulation and comparison, the following analysis is conducted based on Equations (8.4) and (8.5).

An AC circuit consists of a pure resistive load R to dissipate the produced electrical energy. It is a convenient circuit for preliminary evaluation of the mechanical and electrical responses, thus is quite often employed in the literature. A standard DC circuit includes a full-wave rectifier, a filtering capacitor C_f and a resistor R (Lefevre et al., 2005;), as shown in Figure 8.3(a). If $C_f R$ is much larger than the oscillating period, the output DC voltage V_c is nearly constant. In steady state, when $|V| < V_c$ with V

being the generated voltage in the piezoelectric sheets, the rectifier is blocked, otherwise it is conducted when $|V|$ reaches V_c and then $|V|$ is kept equal to V_c . The rectifier is regarded as ideal in the following analysis. The displacement and voltage waveforms for a standard circuit are given in Figure 8.3(b).

Figure 8.3 (a) Standard interface circuit and (b) corresponding voltage and displacement wave form

An SCE interface is composed of a full-wave rectifier, an inductor L_{SCE} , a diode D_{SCE} , a switch S , a filter capacitor C_f and a resistor R . S is closed only at the time instants when the harvester reaches maximum (and minimum) deflection; otherwise it keeps open with piezoelectric transducer experiencing the open-circuit condition. The electrostatic energy accumulates in the piezoelectric transducer during this open-circuit period, and is transferred instantly to L_{SCE} once S is closed. C_f is set to be large to smooth the voltage across R . Figure 8.4 shows the SCE configuration and the corresponding voltage and displacement wave forms. The controlling principle of S is set to be self-powered with a key electrical component called electric breaker (will be shown in Section 8.5.2), the power consumption of which is lowered down to $<5\%$ of the total harvested power (Liang and Liao, 2009; Lallart and Guyomar, 2008; Richard et al., 2007). Thus, we do not consider this energy loss in the analysis.

Figure 8.4 (a) SCE interface circuit and (b) corresponding wave forms of voltage and tip displacement

8.2.1 AC circuit analysis

To solve Equations (8.4) and (8.5) with a simple AC circuit, we first derive the relationship between the piezoelectric voltage V and the displacement u . Before the derivation, we introduce the following non-dimensional parameters to ease the analysis:

$$\hat{M} = \frac{M}{\rho h^2 L}, \hat{U} = \frac{U}{\omega_n h}, \omega_n = \sqrt{\frac{K}{M}}, k_e^2 = \frac{\Theta^2}{C_p K}, \zeta = \frac{C}{2\sqrt{KM}}, \Omega = \frac{\omega}{\omega_n}, r = C_p R \omega_n \quad (8.11)$$

where $\hat{M}$ is the dimensionless mass ratio (i.e., the ratio of the effective mass of the GPEH to the total mass of the surrounding flow); $\hat{U}$ is the reduced wind speed; k_e^2 is the alternative electromechanical coupling coefficient; and Ω and r are the normalized frequency and resistive load, respectively. During steady state vibration, we assume V and u to be of the following forms:

$$V(t) = -V_m \cos(\omega t + \varphi) \quad (8.12)$$

$$u(t) = -u_m \cos(\omega t) \quad (8.13)$$

where V_m and u_m are the magnitude of V and u , respectively; ω is the oscillating frequency (which is near but not exactly the same with ω_n ; φ is the phase difference between V and u . With only a resistor R in the circuit, the current is simply expressed as

$$I(t) = \frac{V(t)}{R} \quad (8.14)$$

Introducing Equations (8.12) to (8.14) into (8.5) yields

$$\frac{-V_m \cos(\omega t + \varphi)}{R} + C_p V_m \omega \sin(\omega t + \varphi) = \Theta u_m \omega \sin(\omega t) \quad (8.15)$$

Letting $t=0$, Equation (8.15) becomes

$$\frac{-V_m \cos \varphi}{R} + C_p V_m \omega \sin \varphi = 0 \quad (8.16)$$

from which the phase difference is determined to take the following form:

$$\tan \varphi = \frac{1}{C_p R \omega} = \frac{1}{r \Omega} \quad (8.17)$$

Therefore, we have

$$\sin \varphi = \frac{1}{\sqrt{r^2 \Omega^2 + 1}}, \cos \varphi = \frac{r \Omega}{\sqrt{r^2 \Omega^2 + 1}} \quad (8.18)$$

Integrating both sides of Equation (8.5) from time instant t_1 to t_2 gives

$$\int_{t_1}^{t_2} \frac{V(t)}{R} dt + \int_{t_1}^{t_2} C_p \dot{V}(t) dt = \int_{t_1}^{t_2} \Theta \dot{u}(t) dt \quad (8.19)$$

If we set $t_1=0$ and $t_2=T/2$, Equation (8.13) reaches the minimum $-u_m$ and maximum u_m , respectively. Now considering Equations (8.12), (8.13) and (8.18), from (8.19) we obtain the relationship between V_m and u_m as

$$V_m = \frac{\Theta}{\frac{1}{R\omega} \sin \varphi + C_p \cos \varphi} u_m = \frac{K}{\Theta} k_e^2 \frac{r \Omega}{\sqrt{1 + r^2 \Omega^2}} u_m \quad (8.20)$$

Now we can make use of the energy balance of the harvesting system to derive the explicit expressions of V_m and u_m . Multiplying Equation (8.4) by $\dot{u}$ and Equation (8.5) by V and integrating these equations with respect to time from instant t_1 to t_2 give the equations of energy balance as

$$\int_{t_1}^{t_2} F_a \dot{u} dt = \frac{1}{2} M \dot{u}^2 \Big|_{t_1}^{t_2} + \frac{1}{2} K u^2 \Big|_{t_1}^{t_2} + \int_{t_1}^{t_2} C \dot{u}^2 dt + \int_{t_1}^{t_2} \Theta V \dot{u} dt \quad (8.21)$$

$$\int_{t_1}^{t_2} I V dt + \frac{1}{2} C_p V^2 \Big|_{t_1}^{t_2} = \int_{t_1}^{t_2} \Theta V \dot{u} dt \quad (8.22)$$

Again, we consider the time duration as half the period with $t_1=0$ and $t_2=T/2$. In steady-state operation, there is no variation in kinetic energy and elastic energy during this half-period time, i.e., $\frac{1}{2}M\dot{u}^2 \Big|_0^{\frac{T}{2}}=0$, $\frac{1}{2}Ku^2 \Big|_0^{\frac{T}{2}}=0$, and we have the same absolute values of V from Equation (8.12), thus $\frac{1}{2}C_p V^2 \Big|_0^{\frac{T}{2}}=0$. Therefore, the equation of energy balance is reduced to

$$\int_0^{\frac{T}{2}} F_a \dot{u} dt = \int_0^{\frac{T}{2}} C \dot{u}^2 dt + \int_0^{\frac{T}{2}} IV dt \quad (8.23)$$

where the energy consumed by the load resistance for steady state is expressed in terms of V_m using Equation (8.14) as

$$\int_0^{\frac{T}{2}} IV dt = \frac{V_m^2 \pi}{2R\omega} \quad (8.24)$$

Next, considering Equation (8.20), substituting Equations (8.13) and (8.24) into Equation (8.23) and integrating over the half-period of vibration $T/2$ yield

$$\frac{1}{2} \rho h L U^2 \left[\frac{A_1 \omega \pi}{2U} + \frac{3A_3 \omega \pi}{8U} \left(\frac{\omega^2}{U^2} + \beta^2 \right) u_m^2 \right] = \frac{1}{2} C \omega \pi + \frac{\Theta^2 \pi R \omega}{2(1 + C_p^2 R^2 \omega^2)} \quad (8.25)$$

from which we can obtain the explicit expression of the displacement amplitude as

$$u_m = \sqrt{\frac{\frac{1}{2} C \omega \pi + \frac{\Theta^2 \pi R \omega}{2(1 + r^2 \Omega^2)} - \frac{1}{4} \rho h L U A_1 \omega \pi}{\left(\frac{\omega^2}{U^2} + \beta^2 \right) \frac{3}{16} \rho h L U A_3 \omega \pi}} \quad (8.26)$$

Introducing the dimensionless parameters in Equation (8.11) and rewriting Equation (8.26) into a dimensionless form give

$$\hat{u}_m = \frac{u_m}{h} = \frac{1}{h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta + \frac{k_e^2 r}{2(1 + r^2 \Omega^2)} - \frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}}}{\frac{3}{16} A_3 \frac{\hat{U}}{\hat{M}}}} \quad (8.27)$$

Here and hereafter, in order to shorten the expression for convenience, we introduce the

following symbols: $G_1 = h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}$, $G_2 = \sqrt{-\frac{3}{16} A_3 \Omega \frac{\hat{U}}{\hat{M}}}$, and $G_3 = \sqrt{-\zeta \Omega - \frac{k_e^2 r \Omega}{2(1+r^2 \Omega^2)} + \frac{1}{4} A_1 \Omega \frac{\hat{U}}{\hat{M}}}$. Now using Equation (8.20), we can express the magnitude of piezoelectric voltage V_m explicitly as

$$V_m = \frac{M \omega_n^2 h}{\Theta} k_e^2 \frac{r \Omega}{\sqrt{1+r^2 \Omega^2}} \frac{G_3}{G_1 G_2} \quad (8.28)$$

The average power at steady state can be calculated straightforwardly as

$$P_{ave} = \frac{V_m^2}{2R} = M \omega_n^3 h^2 \frac{k_e^2 r \Omega^2}{2(1+r^2 \Omega^2)} \frac{G_3^2}{G_1^2 G_2^2} \quad (8.29)$$

Also, the expressions of voltage and power can be rewritten in the non-dimensional forms as

$$\hat{V}_m = \frac{V_m}{\left(\frac{M \omega_n^2 h}{\Theta} \right)} = k_e^2 \frac{r \Omega}{\sqrt{1+r^2 \Omega^2}} \frac{G_3}{G_1 G_2} \quad (8.30)$$

$$\hat{P}_{ave} = \frac{P_{ave}}{M \omega_n^3 h^2} = \frac{k_e^2 r \Omega^2}{2(1+r^2 \Omega^2)} \frac{G_3^2}{G_1^2 G_2^2} \quad (8.31)$$

Next, we need to find the explicit expression of the normalized frequency Ω . It is well known that for VPEH, there are two resonances for the harvester at the two extreme conditions: short circuit condition and open circuit condition. Similarly, for a GPEH, when subjected to no wind flows, i.e., $U=0$, we have

$$\Omega_{sc}=1, \quad \Omega_{oc} = \sqrt{1+k_e^2} \quad (8.32)$$

Yet with $U \neq 0$, the expressions of the two resonances will be changed to incorporate the influence of U . Employing Equations (8.12) and (8.13) back into (8.4), and expressing V_m in terms of u_m as in Equation (8.20), we have

$$\begin{aligned}
 & M\omega^2 u_m \cos(\omega t) + C\omega u_m \sin(\omega t) - Ku_m \cos(\omega t) - Kk_e^2 \frac{r\Omega}{\sqrt{1+r^2\Omega^2}} u_m \cos(\omega t + \varphi) \\
 &= \frac{1}{2} \rho h L U^2 \left\{ A_1 \left[\frac{\omega u_m \sin(\omega t)}{U} + \beta(-u_m \cos(\omega t)) \right] + A_3 \left[\frac{\omega u_m \sin(\omega t)}{U} + \beta(-u_m \cos(\omega t)) \right]^3 \right\}
 \end{aligned} \tag{8.33}$$

Again we set $t=0$, and from Equation (8.33) we have

$$\Omega^2 - 1 - k_e^2 \frac{(r\Omega)^2}{1+r^2\Omega^2} = \frac{1}{2K} \rho h L U^2 (-A_1\beta - A_3\beta^3 u_m^2) \tag{8.34}$$

where the term $A_3\beta^3 u_m^2$ can be easily expanded using the expression of u_m as

$$A_3\beta^3 u_m^2 = A_1\beta \left\{ \frac{1}{\frac{\Omega^2 \omega_n^2}{\beta^2 U^2} + 1} \left[\frac{\zeta + \frac{k_e^2 r}{2(1+r^2\Omega^2)}}{\frac{3}{16} \frac{\rho h L U}{\omega_n M} A_1} - \frac{4}{3} \right] \right\} \tag{8.35}$$

To simplify the derivation process of Ω from Equation (34), we introduce a coefficient f with which the right side of (34) is approximated by a parameter regarding r and U , as shown in Equation (37). Since the harvester under transverse galloping operates at its first bending mode, it is reasonable to assume that $\Omega \approx 1$ thus Ω is dropped in the coefficient f . This simplifies the calculations and gives a simpler explicit solution of Ω which has negligible discrepancies as compared to the numerical iteration results.

$$f = \frac{1}{\frac{\omega_n^2}{\beta^2 U^2} + 1} \left[\frac{\zeta + \frac{k_e^2 r}{2(1+r^2)}}{\frac{3}{16} \frac{\rho h L U}{\omega_n M} A_1} - \frac{4}{3} \right] \tag{8.36}$$

Now Equation (8.34) is rewritten as

$$\Omega^2 - 1 - k_e^2 \frac{(r\Omega)^2}{1+r^2\Omega^2} = -(1+f) \frac{1}{2K} \rho h L U^2 A_1\beta \tag{8.37}$$

from which the explicit expression of the normalized galloping frequency for a GPEH

is obtained as

$$\Omega = \sqrt{\frac{\left(1 + k_e^2 - \frac{1+f}{2} \frac{\hat{U}^2}{\hat{M}} A_1 \beta h\right) r^2 - 1 + \sqrt{\left[\left(1 + k_e^2 - \frac{1+f}{2} \frac{\hat{U}^2}{\hat{M}} A_1 \beta h\right) r^2 - 1\right]^2 - 4r^2 \left(\frac{1+f}{2} \frac{\hat{U}^2}{\hat{M}} A_1 \beta h - 1\right)}}{2r^2}} \quad (8.38)$$

Inspecting Equation (8.38), it is confirmed that without wind flows, i.e. $\hat{U}=0$, $\Omega_{sc}=1$ with $r=0$, $\Omega_{oc}=\sqrt{1+k_e^2}$ with $r \Rightarrow \infty$. Under a wind speed $\hat{U} \neq 0$, it suggests that

$$\begin{cases} \Omega_{sc} = \sqrt{1 - \frac{1+f}{2} \frac{\hat{U}^2}{\hat{M}} A_1 \beta h} \\ \Omega_{oc} = \sqrt{1 + k_e^2 - \frac{1+f}{2} \frac{\hat{U}^2}{\hat{M}} A_1 \beta h} \end{cases} \quad (8.39)$$

The coefficient f is guaranteed to be positive by inspecting Equation (8.35), thus it is clear from Equation (8.39) that when the GPEH is subjected to a incoming flow, the vibration frequency is decreased compared to the free vibration case. The larger the wind speed, the smaller the vibration frequency becomes.

8.2.2 Standard circuit analysis

The derivation for the output responses of a standard DC circuit is based on the schematic shown in Figure 8.3(a). During steady-state vibration, the full-wave rectifier is kept block when the absolute value of piezoelectric voltage V is lower than the constant-valued rectified DC voltage V_c , thus the piezoelectric transducer experiences open-circuit condition with $I=0$. Once the absolute value of V reaches V_c , the rectifier conducts, and V is maintained equal to V_c . Then the rectifier blocks again when V starts to decrease simultaneously with displacement u , e.g., at time instant t_2 as shown in Figure 8.3(b). Similar to the case of AC circuit, we first determine the relationship between voltage V_c and displacement amplitude u_m .

During a particular half vibration period time for steady-state operation, for instance, from t_1 to t_2 , the average current flowing through C_f is zero, thus the electrical charge

generated by the transducer equals to that across R , written as

$$\int_t^{t_2} I(t) dt = \frac{V_c}{R} \frac{T}{2} \quad (8.40)$$

Integrating Equation (8.5) with respect to time t from t_1 to t_2 gives

$$\frac{V_c}{R} \cdot \frac{T}{2} + \int_t^{t_2} C_p \dot{V}(t) dt = \int_t^{t_2} \Theta \dot{u}(t) dt \quad (8.41)$$

Based on the waveforms of V and u shown in Figure 8.3(b), the relationship between V_c and u_m is obtained as

$$V_c = \frac{2\Theta R \omega}{\pi + 2C_p R \omega} u_m \quad (8.42)$$

which is identical to that derived by Lefeuvre et al. (2005) and Shu and Lien (2006). Similar to the AC circuit, the equations of energy balance are obtained by multiplying Equation (8.4) by $\dot{u}$ and Equation (8.5) by V , and integrating from t_1 to t_2 , as shown in Equations (8.21) and (8.22). With the waveforms in Figure 8.3(b), over the half vibration period ($t_1=0$ and $t_2=T/2$), we have $\frac{1}{2} M \dot{u}^2 \Big|_0^{\frac{T}{2}} = 0$, $\frac{1}{2} K u^2 \Big|_0^{\frac{T}{2}} = 0$, and $\frac{1}{2} C_p V^2 \Big|_0^{\frac{T}{2}} = 0$, and the transferred electrical energy by the piezoelectric transducer equals to that consumed by the load R , written as

$$\int_{t_1}^{t_2} IV dt = \frac{V_c^2}{R} \frac{T}{2} \quad (8.43)$$

Hence, the equation of energy balance is simplified as

$$\int_0^{\frac{T}{2}} F_a \dot{u} dt = \int_0^{\frac{T}{2}} C \dot{u}^2 dt + \frac{V_c^2}{R} \frac{T}{2} \quad (8.44)$$

Assuming that u takes the form of Equation (8.13), employing Equation (8.42) in Equation (8.44) results in

$$\frac{1}{2} \rho h L U^2 \left[\frac{A_1 \omega \pi}{2U} + \frac{3A_3 \omega \pi}{8U} \left(\frac{\omega^2}{U^2} + \beta^2 \right) u_m^2 \right] = \frac{1}{2} C \omega \pi + \frac{4\Theta^2 R \omega \pi}{(\pi + 2C_p R \omega)^2} \quad (8.45)$$

The magnitude of displacement is then calculated as

$$u_m = \sqrt{\frac{\frac{1}{2}C\omega\pi + \frac{4\Theta^2 R\omega\pi}{(\pi + 2C_p R\omega)^2} - \frac{1}{4}\rho h L U A_1 \omega\pi}{\left(\frac{\omega^2}{U^2} + \beta^2\right) \frac{3}{16}\rho h L U A_3 \omega\pi}} \quad (8.46)$$

Introducing the dimensionless parameters and normalizing u_m into a dimensionless form, we have

$$\hat{u}_m = \frac{1}{h\sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta\Omega + \frac{4k_e^2 r\Omega}{(\pi + 2r\Omega)^2} - \frac{1}{4}A_1\Omega \frac{\hat{U}}{\hat{M}}}{\frac{3}{16}A_3\Omega \frac{\hat{U}}{\hat{M}}}} \quad (8.47)$$

With Equation (8.42), and introducing $G_4 = \sqrt{-\zeta\Omega - \frac{4k_e^2 r\Omega}{(\pi + 2r\Omega)^2} + \frac{1}{4}A_1\Omega \frac{\hat{U}}{\hat{M}}}$, the magnitude of rectified voltage V_c can be explicitly expressed as

$$V_c = \frac{M\omega_n^2 h}{\Theta} \frac{2k_e^2 r\Omega}{\pi + 2r\Omega} \frac{G_4}{G_1 G_2} \quad (8.48)$$

And the average power is given as

$$P_{ave} = \frac{V_c^2}{R} = M\omega_n^3 h^2 \frac{4k_e^2 r\Omega^2}{(\pi + 2r\Omega)^2} \frac{G_4^2}{G_1^2 G_2^2} \quad (8.49)$$

The dimensionless forms of V_c and P_{ave} are

$$\hat{V}_c = \frac{V_c}{\left(\frac{M\omega_n^2 h}{\Theta}\right)} = \frac{2k_e^2 r\Omega}{\pi + 2r\Omega} \frac{G_4}{G_1 G_2} \quad (8.50)$$

$$\hat{P}_{ave} = \frac{P_{ave}}{M\omega_n^3 h^2} = \frac{4k_e^2 r\Omega^2}{(\pi + 2r\Omega)^2} \frac{G_4^2}{G_1^2 G_2^2} \quad (8.51)$$

The normalized frequency for the standard circuit is regarded as the open-circuit resonance Ω_{oc} given in Equation (8.39) in the later validating calculations, considering that it experiences open-circuit condition when the rectifier is blocked.

8.2.3 SCE circuit analysis

In steady-state operation of an SCE circuit, the piezoelectric transducer experiences the open-circuit condition with the switch S being open for most of the time except at the instants of extreme displacement, i.e. minimum u at t_1 and maximum u at t_2 as shown in Figure 8.4(b). Therefore, the outgoing I is zero from t_1 to t_2 . The relationship between the piezoelectric voltage amplitude V_m and the displacement amplitude u_m is obtained as follows by integrating Equation (8.5) from t_1 to t_2 with $\int_{t_1}^{t_2} IVdt = 0$:

$$V_m = \frac{2\Theta}{C_p} u_m \quad (8.52)$$

Equation (8.52) is identical to the result derived by Lefeuvre et al. (2005). In the equations of energy balance (8.21) and (8.22), with $t_l=0$ and $t_2=T/2$, we have $\frac{1}{2} M \dot{u}^2 \Big|_0^{\frac{T}{2}} = 0$, $\frac{1}{2} K u^2 \Big|_0^{\frac{T}{2}} = 0$, and $\frac{1}{2} C_p V^2 \Big|_0^{\frac{T}{2}} = \frac{1}{2} C_p V_m^2$, thus the reduced energy balance equation is

$$\int_0^{\frac{T}{2}} F_a \dot{u} dt = \int_0^{\frac{T}{2}} C \dot{u}^2 dt + \frac{1}{2} C_p V_m^2 \quad (8.53)$$

With u in the form of Equation (8.13), Equation (8.53) is expanded as

$$\frac{1}{2} \rho h L U^2 \left[\frac{A_1 \omega \pi}{2U} + \frac{3A_3 \omega \pi}{8U} \left(\frac{\omega^2}{U^2} + \beta^2 \right) u_m^2 \right] = \frac{1}{2} C \omega \pi + \frac{2\Theta^2}{C_p} \quad (8.54)$$

The magnitude of displacement and the corresponding non-dimensional form are obtained as

$$u_m = \sqrt{\frac{\frac{1}{2} C \omega \pi + \frac{2\Theta^2}{C_p} - \frac{1}{4} \rho h L U A_1 \omega \pi}{\left(\frac{\omega^2}{U^2} + \beta^2 \right) \frac{3}{16} \rho h L U A_3 \omega \pi}} \quad (8.55)$$

$$\hat{u}_m = \frac{u_m}{h} = \frac{1}{h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta \Omega + \frac{2k_e^2}{\pi} - \frac{1}{4} A_1 \Omega \frac{\hat{U}}{\hat{M}}}{\frac{3}{16} A_3 \Omega \frac{\hat{U}}{\hat{M}}}} \quad (8.56)$$

Finally, with $G_5 = \sqrt{-\zeta\Omega - \frac{2k_e^2}{\pi} + \frac{1}{4}A_1\Omega\frac{\hat{U}}{\hat{M}}}$, the voltage amplitude and the average power together with their corresponding non-dimensional forms are expressed as

$$V_m = \frac{M\omega_n^2 h}{\Theta} 2k_e^2 \frac{G_5}{G_1 G_2} \quad (8.57)$$

$$P_{ave} = \frac{\frac{1}{2}C_p V_m^2}{\frac{T}{2}} = M\omega_n^3 h^2 \frac{2k_e^2 \Omega}{\pi} \frac{G_5^2}{G_1^2 G_2^2} \quad (8.58)$$

$$\hat{V}_m = \frac{V_m}{\left(\frac{M\omega_n^2 h}{\Theta}\right)} = 2k_e^2 \frac{G_5}{G_1 G_2} \quad (8.59)$$

$$\hat{P}_{ave} = \frac{P_{ave}}{M\omega_n^3 h^2} = \frac{2k_e^2 \Omega}{\pi} \frac{G_5^2}{G_1^2 G_2^2} \quad (8.60)$$

The vibration frequency Ω of the GPEH with SCE circuit equals to Ω_{oc} as shown in Equation (8.39).

8.3 Optimal Power

Inspecting the expressions of average power, Equations (8.31), (8.51) and (8.60), with all the geometrical and material, aerodynamic parameters $\hat{M}$, h , ζ , β , $\hat{U}$, and A_1 to A_3 fixed, it is noted that for the AC and standard circuits, the power depends on both the load resistance r and the coupling k_e^2 , while for the SCE circuit, the power is independent of r and dependent on k_e^2 only. Thus in the following derivations, the maximum average power at steady state for the AC and standard circuit is obtained with k_e^2 fixed along with the other parameters, and by adjusting r according to

$$\frac{\partial \hat{P}_{ave}(r, \Omega, k_e, \zeta, \hat{U}, \hat{M}, \beta, h, A_1, A_3)}{\partial r} = 0 \quad (8.61)$$

The roots of Equation (8.61), indicated as r^{opt} , denotes the optimal load resistance

under the specific k_e^2 condition. $\hat{V}_c^{opt}$ and $\hat{P}_{ave}^{opt}$ denote the normalized voltage and power at the optimal load r^{opt} , respectively. Moreover, with the variation of k_e^2 , there exists the maximum $\hat{P}_{ave}^{opt}$, which is indicated as $\hat{P}_{ave}^{max}$, with the k_e^2 to achieve this maximum indicated as $(k_e^2)^{opt}$.

The maximum average power $\hat{P}_{ave}^{max}$ for SCE circuit is obtained by calculating the optimal coupling $(k_e^2)^{opt}$ according to

$$\frac{\partial \hat{P}_{ave}(\Omega, k_e^2, \zeta, \hat{U}, \hat{M}, \beta, h, A_1, A_3)}{\partial k_e^2} = 0 \quad (8.62)$$

8.3.1 AC circuit analysis

Introducing Equation (8.31) into Equation (8.61), it is found that the number of positive real roots depends on the value of k_e^2 . If Equation (8.63) is satisfied, only one root is physically realistic, as expressed in Equation (8.64). With this optimal load r_1^{opt} , the average power achieves the maximum for the given k_e^2 . The corresponding expressions of the non-dimensional displacement, voltage and average power at r_1^{opt} are given

by Equations (8.65) to (8.67), with $G_6 = \sqrt{-\zeta\Omega + \frac{1}{4}A_1\Omega\frac{\hat{U}}{\hat{M}} - \frac{k_e^2}{4}}$. It should be noted that

since Ω varies with r , it needs a few steps of iterations to pinpoint the value of r_1^{opt} . On the other hand, if k_e^2 is large enough to satisfy Equation (8.68), there will be two additional practical roots r_2^{opt} and r_3^{opt} , expressed as in Equation (8.69). It is observed that r_2^{opt} and r_3^{opt} vary with k_e^2 . The average power reaches the same maximum value at r_2^{opt} and r_3^{opt} , while it becomes minimum at r_1^{opt} . The non-dimensional displacement, voltage and average power at r_2^{opt} and r_3^{opt} are

expressed in Equations (8.70) to (8.72), with $G_7 = \sqrt{\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta}$ and $G_8 = \sqrt{-\frac{3}{8} A_3 \frac{\hat{U}}{\hat{M}}}$. It is noted from Equation (8.72) that the maximum power remains constant with large k_e^2 , unlike the case for small k_e^2 where it increases with k_e^2 as seen from Equation (8.67). We indicate the value of the critical k_e^2 , $2\Omega \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right)$, as shown in Equation (8.68), as $(k_e^2)_{AC}^{opt}$ and the constant maximum power as $(\hat{P}_{ave}^{max})_{AC}$. We will discuss about this value later.

$$k_e^2 \leq (k_e^2)_{AC}^{opt} = 2\Omega \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \quad (8.63)$$

$$r_1^{opt} = \frac{1}{\Omega} \quad (8.64)$$

$$\hat{u}_m^{opt} = \frac{u_m^{opt}}{h} = \frac{1}{h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta \Omega - \frac{1}{4} A_1 \Omega \frac{\hat{U}}{\hat{M}} + \frac{k_e^2}{4}}{\frac{3}{16} A_3 \Omega \frac{\hat{U}}{\hat{M}}}} = \frac{G_6}{G_1 G_2} \quad (8.65)$$

$$\hat{V}_c^{opt} = \frac{V_c^{opt}}{\left(\frac{M \omega_n^2 h}{\Theta} \right)} = \frac{k_e^2}{\sqrt{2}} \frac{G_6}{G_1 G_2} \quad (8.66)$$

$$\hat{P}_{ave}^{opt} = \frac{P_{ave}^{opt}}{M \omega_n^3 h^2} = \frac{k_e^2 \Omega}{4} \frac{G_6^2}{G_1^2 G_2^2} \quad (8.67)$$

$$k_e^2 > (k_e^2)_{AC}^{opt} = 2\Omega \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \quad (8.68)$$

$$r_{2,3}^{opt} = \frac{k_e^2 \pm \sqrt{k_e^4 - 8 \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right)^2 \Omega^2}}{2\Omega^2 \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right)} \quad (8.69)$$

$$\hat{u}_m^{opt} \Big|_{r=r_2^{opt}} = \frac{u_m^{opt} \Big|_{r=r_2^{opt}}}{h} = \hat{u}_m^{opt} \Big|_{r=r_3^{opt}} = \frac{u_m^{opt} \Big|_{r=r_3^{opt}}}{h} = \frac{1}{h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta - \frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}}}{\frac{3}{8} A_3 \frac{\hat{U}}{\hat{M}}}} = \frac{G_7}{G_1 G_8} \quad (8.70)$$

$$\hat{V}_{c2,3}^{opt} = \frac{k_e^2 \left(k_e^2 \pm \sqrt{k_e^4 - 8G_7^4 \Omega^2} \right)}{\sqrt{4G_7^4 \Omega^2 + \left(k_e^2 \pm \sqrt{k_e^4 - 8G_7^4 \Omega^2} \right)^2}} \frac{G_7}{G_1 G_8} \quad (8.71)$$

$$\left(\hat{P}_{ave}^{max} \right)_{AC} = \hat{P}_{ave}^{opt} \Big|_{r=r_2^{opt}} = \frac{P_{ave}^{opt} \Big|_{r=r_2^{opt}}}{M \omega_n^3 h^2} = \hat{P}_{ave}^{opt} \Big|_{r=r_3^{opt}} = \frac{P_{ave}^{opt} \Big|_{r=r_3^{opt}}}{M \omega_n^3 h^2} = \frac{\Omega^2}{2} \frac{G_7^4}{G_1^2 G_8^2} \quad (8.72)$$

8.3.2 Standard circuit analysis

Similar to the AC circuit, the number of roots for the standard circuit depends on k_e^2 . Introducing Equation (8.51) into Equation (8.61) gives the expression of the only practical r_1^{opt} when k_e^2 satisfies Equation (8.73). The expressions of the non-dimensional displacement, voltage and the maximum average power at r_1^{opt} are given by Equations (8.75) to (8.77), with $G_9 = \sqrt{-\zeta \Omega + \frac{1}{4} A_1 \Omega \frac{\hat{U}}{\hat{M}} - \frac{k_e^2}{2\pi}}$. If k_e^2 satisfies Equation (8.78), the maximum becomes minimum at r_1^{opt} and two constant maximum branch out at r_2^{opt} and r_3^{opt} given by Equation (8.79) with the corresponding non-dimensional displacement, voltage and the maximum average power given by Equations (8.80) to (8.82). Comparing the optimal coupling $\left(k_e^2 \right)_{STD}^{opt}$ for the standard circuit shown in Equation (8.78) with $\left(k_e^2 \right)_{AC}^{opt}$ in Equation (8.68), and comparing $\left(\hat{P}_{ave}^{max} \right)_{STD}$ in Equation (8.82) with $\left(\hat{P}_{ave}^{max} \right)_{AC}$ in Equation (8.72), it is seen that the AC circuit achieves the same amount of maximum power

$(\hat{P}_{ave}^{\max})_{AC} = (\hat{P}_{ave}^{\max})_{STD}$) with the standard circuit but with a smaller coupling with approximately $(k_e^2)^{opt}_{AC} = \frac{2}{\pi} (k_e^2)^{opt}_{STD}$.

$$k_e^2 \leq (k_e^2)^{opt}_{STD} = \pi \Omega \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \quad (8.73)$$

$$r_1^{opt} = \frac{\pi}{2\Omega} \quad (8.74)$$

$$\hat{u}_m^{opt} = \frac{u_m^{opt}}{h} = \frac{1}{h \sqrt{\frac{\Omega^2}{\hat{U}^2 h^2} + \beta^2}} \sqrt{\frac{\zeta \Omega - \frac{1}{4} A_1 \Omega \frac{\hat{U}}{\hat{M}} + \frac{k_e^2}{2\pi}}{\frac{3}{16} A_3 \Omega \frac{\hat{U}}{\hat{M}}}} = \frac{G_9}{G_1 G_2} \quad (8.75)$$

$$\hat{V}_c^{opt} = \frac{V_c^{opt}}{\left(\frac{M \omega_n^2 h}{\Theta} \right)} = \frac{k_e^2}{2} \frac{G_9}{G_1 G_2} \quad (8.76)$$

$$\hat{P}_{ave}^{opt} = \frac{P_{ave}^{opt}}{M \omega_n^3 h^2} = \frac{k_e^2 \Omega}{2\pi} \frac{G_9^2}{G_1^2 G_2^2} \quad (8.77)$$

$$k_e^2 > (k_e^2)^{opt}_{STD} = \pi \Omega \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \quad (8.78)$$

$$r_{2,3}^{opt} = \frac{k_e^2 - \frac{\pi \Omega}{2} \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \pm \sqrt{\left[k_e^2 - \frac{\pi \Omega}{2} \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right) \right]^2 - \frac{\pi^2 \Omega^2}{4} \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right)^2}}{\Omega^2 \left(\frac{1}{4} A_1 \frac{\hat{U}}{\hat{M}} - \zeta \right)} \quad (8.79)$$

$$\hat{u}_m^{opt} \big|_{r=r_2^{opt}} = \frac{u_m^{opt} \big|_{r=r_2^{opt}}}{h} = \hat{u}_m^{opt} \big|_{r=r_3^{opt}} = \frac{u_m^{opt} \big|_{r=r_3^{opt}}}{h} = \frac{G_7}{G_1 G_8} \quad (8.80)$$

$$\hat{V}_{c2,3}^{opt} = \frac{V_{c2,3}^{opt}}{\left(\frac{M \omega_n^2 h}{\Theta} \right)} = \frac{2k_e^2 \Omega \left[k_e^2 - \frac{\pi \Omega}{2} G_7^2 \pm \sqrt{\left[k_e^2 - \frac{\pi \Omega}{2} G_7^2 \right]^2 - \frac{\pi^2 \Omega^2}{4} G_7^4} \right]}{\pi \Omega^2 G_7^2 + 2\Omega \left[k_e^2 - \frac{\pi \Omega}{2} G_7^2 \pm \sqrt{\left[k_e^2 - \frac{\pi \Omega}{2} G_7^2 \right]^2 - \frac{\pi^2 \Omega^2}{4} G_7^4} \right]} \frac{G_7}{G_1 G_8} \quad (8.81)$$

$$\left(\hat{P}_{ave}^{max}\right)_{STD} = \hat{P}_{ave}^{opt} \big|_{r=r_2^{opt}} = \frac{P_{ave}^{opt} \big|_{r=r_2^{opt}}}{M \omega_n^3 h^2} = \hat{P}_{ave}^{opt} \big|_{r=r_3^{opt}} = \frac{P_{ave}^{opt} \big|_{r=r_3^{opt}}}{M \omega_n^3 h^2} = \frac{\Omega^2}{2} \frac{G_7^4}{G_1^2 G_8^2} \quad (8.82)$$

8.3.3 SCE circuit analysis

Since the power from the SCE circuit does not vary with r , but is dependent on the coupling k_e^2 only, we need to determine the critical value of k_e^2 at which the power achieves a maximum. Introducing Equation (8.60) into Equation (8.62), the expression of $\left(k_e^2\right)_{SCE}^{opt}$ is obtained as

$$\left(k_e^2\right)_{SCE}^{opt} = \frac{\pi \Omega}{4} \left(A_1 \frac{\hat{U}}{4 \hat{M}} - \zeta \right) \quad (8.83)$$

$$\hat{u}_m^{opt} = \frac{u_m^{opt}}{h} = \frac{G_7}{G_1 G_8} \quad (8.84)$$

$$\hat{V}_m^{opt} = \frac{V_m^{opt}}{\left(\frac{M \omega_n^2 h}{\Theta} \right)} = \frac{\pi \Omega}{2} \frac{G_7^3}{G_1 G_8} \quad (8.85)$$

$$\left(\hat{P}_{ave}^{max}\right)_{SCE} = \frac{P_{ave}^{max}}{M \omega_n^3 h^2} = \frac{\Omega^2}{2} \frac{G_7^4}{G_1^2 G_8^2} \quad (8.86)$$

At $\left(k_e^2\right)_{SCE}^{opt}$, the expressions of the non-dimensional displacement, voltage and the maximum average power at are given by Equations (8.84) to (8.86). Comparing Equations (8.72), (8.82) and (8.86), it reveals that the AC, standard and SCE circuits generate the same amount of maximum power, i.e.

$$\left(\hat{P}_{ave}^{max}\right)_{AC} = \left(\hat{P}_{ave}^{max}\right)_{STD} = \left(\hat{P}_{ave}^{max}\right)_{SCE} \quad (8.87)$$

Meanwhile, inspecting Equations (8.68), (8.78) and (8.83), the critical values of coupling k_e^2 , at which the power achieves or starts to enter the maximum state, can be ranked in an ascending order as $\left(k_e^2\right)_{SCE}^{opt} < \left(k_e^2\right)_{AC}^{opt} < \left(k_e^2\right)_{STD}^{opt}$, with the quantitative relations given as

$$(k_e^2)^{opt}_{SCE} = \frac{\pi}{8}(k_e^2)^{opt}_{AC} = \frac{1}{4}(k_e^2)^{opt}_{STD} \quad (8.88)$$

The above relations suggest that the SCE circuit is able to achieve the same maximum power with a much smaller coupling as compared to the AC and standard circuits. The coupling k_e^2 is related to the amount of piezoelectric materials. Smaller k_e^2 corresponds to less piezoelectric material for the same prototype. This implies that the SCE circuit owns superiority over the other two for its ability to achieve the same maximum power by saving piezoelectric materials. This is a main advantage of the SCE circuit.

8.4 Cut-in wind speed

With the derived explicit expressions of displacement for GPEH, the cut-in wind speed can be easily calculated. The principle of calculating the cut-in wind speed for different circuits is to determine the lower limit of U to guarantee the square root in the expression of u_m to be a practical real number. The derivation procedure is straightforward as follows.

8.4.1 AC circuit analysis

Inspecting Equation (8.26), for the commonly employed cross sections of the bluff body, A_3 is a negative number. To ensure that the root is not a complex number, it is required that

$$\frac{1}{2}C\omega\pi + \frac{\Theta^2\pi R\omega}{2(1+r^2\Omega^2)} - \frac{1}{4}\rho hLU A_1\omega\pi < 0 \quad (8.89)$$

from which the cut-in wind speed is calculated as

$$(U_{cr})_{AC} = \frac{4\omega_n M}{A_1 \rho h L} \left[\zeta + \frac{k_e^2 r}{2(1+r^2\Omega^2)} \right] \quad (8.90)$$

written in a non-dimensional form as

$$\left(\hat{U}_{cr}\right)_{AC} = \frac{4\hat{M}}{A_1} \left[\zeta + \frac{k_e^2 r}{2(1+r^2\Omega^2)} \right] \quad (8.91)$$

With the physical and aerodynamic parameters fixed, it is seen that the cut-in wind speed of the AC circuit depends on both the load resistance r and the electromechanical coupling k_e^2 . If r_1^{opt} in Equation (8.64) is employed for all the whole coupling range regardless of how much k_e^2 is, the following equation can be used for approximate evaluation of the onset wind speed of GPEH.

$$\left(\hat{U}_{cr}\right)_{AC}^{r_1^{opt}} = \frac{4\hat{M}}{A_1} \left(\zeta + \frac{k_e^2}{4\Omega} \right) \quad (8.92)$$

8.4.2 Standard circuit analysis

Similarly, in Equation (8.46), the numerator should be negative, yielding

$$\left(U_{cr}\right)_{STD} = \frac{4\omega_n M}{A_1 \rho h L} \left[\zeta + \frac{4k_e^2 r}{(\pi + 2r\Omega)^2} \right] \quad (8.93)$$

which can be written in the non-dimensional form as

$$\left(\hat{U}_{cr}\right)_{STD} = \frac{4\hat{M}}{A_1} \left[\zeta + \frac{4k_e^2 r}{(\pi + 2r\Omega)^2} \right] \quad (8.94)$$

Again, if r_1^{opt} in Equation (8.74) is employed, the cut-in speed of GPEH with a standard circuit can be approximately evaluated by

$$\left(\hat{U}_{cr}\right)_{STD}^{r_1^{opt}} = \frac{4\hat{M}}{A_1} \left[\zeta + \frac{k_e^2}{2\pi\Omega} \right] \quad (8.95)$$

8.4.3 SCE circuit analysis

From Equation (8.55), the cut-in wind speed for the SCE circuit is obtained as

$$\left(U_{cr}\right)_{SCE} = \frac{4\omega_n M}{A_1 \rho h L \Omega} \left[\zeta \Omega + \frac{2k_e^2}{\pi} \right] \quad (8.96)$$

which is normalized into

$$\left(\hat{U}_{cr}\right)_{SCE} = \frac{4\hat{M}}{A_1} \left[\zeta + \frac{2k_e^2}{\pi\Omega} \right] \quad (8.97)$$

Similar to the case of average power, the cut-in wind speed of the SCE circuit depends on the electromechanical coupling k_e^2 only.

Taking an approximate comparison of cut-in wind speeds between the three circuits, it can be seen that with the same coupling, the standard circuit is able to start oscillating with the smallest cut-in wind speed, while the SCE circuit requires the largest cut-in speed, summarized as

$$\left(\hat{U}_{cr}\right)_{SCE} > \left(\hat{U}_{cr}\right)_{AC}^{r_1^{opt}} > \left(\hat{U}_{cr}\right)_{STD}^{r_1^{opt}} \quad (8.98)$$

8.5 Validations

8.5.1 Experimental setup

Wind tunnel experiment is conducted with a fabricated prototype of GPEH to validate the analytical solutions of the AC circuit. The experimental results are compared with the analytical solutions, as well as the simulation results from an equivalent circuit model, which is introduced next in Section 8.5.2.

The wind tunnel experiment setup is shown in Figure 8.5. A piece of piezoelectric transducer (MFC M2814-P2, Smart Materials Corp.) is bonded to the root area of the substrate aluminum cantilever using epoxy. A bluff body with a square cross section is connected to the free end of the cantilever. The bluff body weights 1.8g, with dimensions of 20mm×20mm×100mm (i.e., $h=20\text{mm}$, $L=100\text{mm}$). The aluminum substrate is of 130mm in length, 20mm in width and 0.6mm in thickness, while the piezoelectric transducer is of 28mm in length, 14mm in width and 0.3mm in thickness. The Young's moduli for the substrate and the transducer are 69GPa and 30.336GPa,

respectively. The capacitance of the transducer is 25.7nF. The air density is regarded as 1.204kg/m^3 as the temperature in the lab is around 20°C . ζ is tested to be 0.011 and the short circuit fundamental frequency is 16.67Hz. The piezoelectric transducer is connected to a resistor R to construct the simple AC circuit. The voltage across R is measured as function of load resistance and wind speed, and the average power is calculated with $P_{ave}=(V_{RMS})^2/R$ with $V_{RMS}=V_m/\sqrt{2}$. The NI 9229 DAQ module (National Instruments) as well as Labview is employed to measure the electrical data and a hotwire anemometer is used to measure the wind speed.

Figure 8.5 Wind tunnel experiment setup of GPEH prototype

8.5.2 Equivalent circuit models

Equivalent circuit modeling of piezoelectric energy harvesters has been frequently pursued in the literature due to its advantageous capability of carrying out system-level simulation for complex interface circuits. In this section, we briefly introduce the equivalent circuit model of the GPEH. With this equivalent circuit model, the multi-way coupling behaviors between the structure, piezoelectric element, aerodynamic force and interface circuit can be incorporated simultaneously. System-level circuit simulation is first carried out with an AC circuit. The simulated responses are compared to the experimental results together with the analytical solutions. Subsequently, with the validated equivalent circuit model, circuit simulation is conducted to further analyze the behaviors of different circuits and validate the

analytical solutions for the standard and SCE circuits.

The equivalent circuit model for GPEH is based on the analogy between the mechanical and electrical parameters, similar to the case of VPEH (Elvin and Elvin, 2009a; Yang and Tang, 2009). Analogizing the displacement u as the charge q , the vibration velocity $\dot{u}$ as the current $\dot{q}$, the effective mass M , damping C and stiffness K as the inductor L_0 , resistance R_0 , and reciprocal of capacitor C_0 , respectively, and the electromechanical coupling Θ as the turn ratio N of the ideal transformer, the governing equations (8.4) and (8.5) are transformed into

$$L_0\ddot{q} + R_0\dot{q} + \frac{1}{C_0}q + NV = \frac{1}{2}\rho hLU^2 \left[A_1 \left(\frac{\dot{q}}{U} + \beta q \right) + A_3 \left(\frac{\dot{q}}{U} + \beta q \right)^3 \right] \quad (8.99)$$

$$I + C_p\dot{V} - N\dot{q} = 0 \quad (8.100)$$

The equivalent circuit model in Equations (8.99) and (8.100) is represented in the circuit diagram shown in Figure 8.6. The aerodynamic force is simulated with a user-defined arbitrary voltage source $\tilde{V}(t)$, with the method proposed in our previous work (Tang et al., 2014). $\tilde{V}(t)$ is expressed as

$$\tilde{V}(t) = -\frac{1}{2}\rho hLU^2 \left\{ A_1 \left[\frac{\dot{q}}{U} + \frac{\phi'(L_t)}{\phi(L_t)} CV_{C_0} \right] + A_3 \left[\frac{\dot{q}}{U} + \frac{\phi'(L_t)}{\phi(L_t)} CV_{C_0} \right]^3 \right\} \quad (8.101)$$

where V_{C_0} is the voltage across C_0 .

Figure 8.6 Equivalent circuit model of GPEH with a simple AC circuit

Replacing R in Figure 8.6 with the according circuit layouts shown in Figure 8.3(a) and Figure 4(a), the overall diagrams of circuit model for the standard and SCE circuits are constructed in Figure 8.7 and Figure 8.8, respectively, which are employed in the system-level circuit simulation to validate the analysis (Zhao et al., 2015). Note that electric breaker in Figure 8.8 is the self-powered detecting and controlling circuit to control the energy transfer from the harvester to the temporary energy storage inductor L_1 (shown in the bottom right). System-level circuit simulation carried out based on these two circuit models have been validated via wind tunnel experiments in Zhao et al. (2015).

Figure 8.7 Overall circuit diagram of GPEH system with standard interface

Figure 8.8 Overall circuit diagram of GPEH system with self-powered SCE interface

8.5.3 Results and discussion

8.5.3.1 Validation of AC circuit

Figure 8.9 AC circuit: comparison of power obtained from experiment, circuit simulation and analytical solution: (a) P_{ave} versus R at $U=4\text{m/s}$ (b) P_{ave} versus U at optimal R .

In the circuit simulation and analytical calculation, the empirical aerodynamic coefficients A_1 and A_3 are set to be 2.3 and -18, respectively (Barrero-Gil et al., 2010). The results of average power as a function of load resistance and wind speed from the experiment, circuit simulation and analytical solution are compared in Figure 8.9. In Figure 8.9(a), the average power is plotted against the load resistance with a wind

speed of 4m/s. With the parameters given in Section 8.5.1, k_e^2 is calculated to be 0.01136, smaller than $\left(k_e^2\right)_{AC}^{opt}$ which is calculated using Equation (8.68) to be 0.01537 for $U=4\text{m/s}$. As expected, there exists one optimal R for the maximum power generation. It can be observed that both the analytical solution and circuit simulation successfully predict this behavior with good agreement with the wind tunnel experiment results. With the optimal load (333k Ω), the power is measured and predicted as function of wind speed, shown in Figure 8.9(b). Again, both the cut-in wind speed and the trend of power increasing with wind speed are accurately predicted by the circuit simulation and analytical solution. Subsequently, with the validated circuit model, circuit simulation is further conducted to validate the analytical solution in terms of the normalized frequency and output responses at different coupling conditions.

Figure 8.10 AC circuit: comparison of normalized frequency obtained from circuit simulation and analytical solution: (a) variation of Ω with R at $U=4\text{m/s}$ (b) variation of Ω with U at optimal R .

Figure 8.10(a) shows the variation of normalized frequency from analytical prediction and circuit simulation as a function of load resistance at $U=4\text{m/s}$. As expected, the normalized oscillation frequency of GPEH increases with R , from the short circuit resonance frequency to the open circuit resonance frequency due to the

electromechanical damping effect of energy harvesting. The analytical solution agrees well with the circuit simulation especially at the medium R range, despite some discrepancies at the extremely small and large R . In Figure 8.10(b), the analytical and simulated Ω are plotted against U with $R=333\text{k}\Omega$. It is observed that with increasing U , Ω slightly decreases. Inspecting Equation (8.4), this can be attributed to the influence of the wind speed on the overall stiffness of the harvester. It can be deduced from these results that with increasing U , the effect of aerodynamic force on the stiffness reduction is also increased. Again, good correlation is observed between the analytical solution and the circuit simulation.

Figure 8.11 AC circuit: comparison of output responses obtained from circuit simulation and analytical solution at small coupling $k_e^2 = 0.01136$: (a) P_{ave} versus R (b) V_{RMS} versus R (c) u_m versus R

Figure 8.12 AC circuit: comparison of output responses obtained from circuit simulation and analytical solution at large coupling $k_e^2 = 0.09087$: (a) P_{ave} versus R (b) V_{RMS} versus R (c) u_m versus R

With the same small coupling $k_e^2 = 0.01136$, analytical calculation and circuit simulation are further carried out at three different wind speeds of 5, 7 and 9 m/s. The responses of average power, RMS voltage and displacement amplitude versus the load resistance are plotted in Figure 8.11. It is observed that at each U , the optimal R for maximum P_{ave} in Figure 8.11(a) corresponds to a minimum displacement amplitude in Figure 8.11(c). This phenomenon reveals that the optimal R brings the strongest electromechanical damping effect as well as the largest energy conversion capability to the GPEH. Good correlation is observed between the analytical solution and the circuit

simulation. Moreover, although it is not obvious in the plots, it should be noted that the optimal loads in Figure 8.11(a) are slightly shifted left for increasing U , which can be deduced from Equation (8.64) since Ω is varying with U .

Subsequently, we increase the coupling k_e^2 to 0.09087, achieved by increasing Θ and C_p by eight times simultaneously. The responses of average power, RMS voltage and displacement amplitude versus load resistance from analytical calculation and circuit simulation are given in Figure 8.12. It is noted from Figure 8.12(a) that at this large k_e^2 , two maximum powers with equal magnitude appear at R_2^{opt} and R_3^{opt} at each U . It is clear that the harvester has entered the region described in Equations (8.68) to (8.72). In between, there exists a valley of power at R_1^{opt} , as expected in Section 8.3.1. Also, the valley of power in Figure 8.12(a) corresponds to the valley of displacement amplitude, shown in Figure 8.12(c). Therefore, similar to the case with small coupling, R_1^{opt} still causes the strongest electromechanical damping effect, yet this damping under the large coupling condition becomes so strong that the displacement is reduced greatly, resulting in low strain energy in the transducer. At R_2^{opt} and R_3^{opt} , however, the displacement is not reduced so much, thus the power at these two loads is higher. Moreover, Figure 8.12 shows that the analytical solution agrees well with the circuit simulation.

Figure 8.13 compares the cut-in wind speed U_{cr} calculated by the analytical solutions with that predicted by the circuit simulation. In Figure 8.13(a), U_{cr} is calculated and simulated as a function of R at both small and large k_e^2 (i.e., 0.01136 and 0.09087). It is obvious that larger k_e^2 corresponds to larger U_{cr} since larger coupling leads to stronger electromechanical damping. For both k_e^2 , the trends of U_{cr} versus R are similar, increasing first up to the respective maximum value, then decreasing. U_{cr} is more sensitive to R at larger coupling. It is worth noting that for both k_e^2 , the critical R causing the maximum U_{cr} is around the region where the minimum u_m appears, by

comparing Figure 8.13(a) with Figures 8.11(c) and 8.12(c). This is not surprising, as the strongest electromechanical damping makes galloping the most difficult to occur. Figure 8.13(b) displays the relationship between U_{cr} and k_e^2 with r_1^{opt} in Equation (8.64) employed for each k_e^2 . Again, good correlation is observed between the analytical solution and circuit simulation, with both showing that U_{cr} increases linearly with k_e^2 as expected from Equation (8.92).

Figure 8.13 AC circuit: comparison of cut-in wind speeds obtained from circuit simulation and analytical solution: (a) U_{cr} versus R at different k_e^2 (b) U_{cr} versus k_e^2

8.5.3.2 Validation of standard circuit

With the circuit diagram shown in Figure 8.7, circuit simulation is carried out to validate the analysis for the standard circuit. Figure 8.14 shows the responses of average power, RMS voltage and displacement amplitude versus load resistance from analytical calculation and circuit simulation with small coupling $k_e^2 = 0.01136$, and Figure 8.15 gives the responses for large coupling $k_e^2 = 0.09087$. Again, good correlations are observed between the simulation and analytical results. Similar to the case of AC circuit, with small coupling, r_1^{opt} brings at the same time the maximum power and the minimum displacement (Figures 8.14(a) and 8.14(c)). Inspecting Figure 8.15(a) and 8.15(c), it is seen that with large coupling, at $U=5$ m/s and 7 m/s, the

maximum power is obtained at r_2^{opt} and r_3^{opt} , while r_1^{opt} corresponds to a valley of power and the minimum of displacement amplitude. Yet for $U=9\text{m/s}$, r_1^{opt} still gives the peak power, which implies that $k_e^2 = 0.09087$ has not reached $(k_e^2)^{opt}_{STD}$ yet, which is larger than $(k_e^2)^{opt}_{AC}$ as stated in Section 8.3.3.

Figure 8.14 Standard circuit: comparison of output responses obtained from circuit simulation and analytical solution at small coupling $k_e^2 = 0.01136$: (a) P_{ave} versus R (b) V_{RMS} versus R (c) u_m versus R

Figure 8.15 Standard circuit: comparison of output responses obtained from circuit simulation and analytical solution at large coupling $k_e^2 = 0.09087$: (a) P_{ave} versus R (b) V_{RMS} versus R (c) u_m versus R

The results of cut-in wind speed from analytical solutions and circuit simulation for the standard circuit are compared in Figure 8.16. The two predictions agree well with each other. Also similar to the case of AC circuit, the highest U_{cr} corresponds to the load region with the minimum u_m , shown in Figure 8.16(a), Figures 8.14(c) and 8.15(c). With r_1^{opt} , U_{cr} increases linearly with k_e^2 shown in Figure 8.16(b), as expected in Equation (8.95). Next, the attainable maximum power with optimal loads given in Equations (8.77) and (8.82), and the displacement amplitude in Equations (8.75) and (8.80) are compared to those from the circuit simulation in Figure 8.17. It is obvious

that the analytical predictions agree well with the simulation. It is also observed that beyond the loci of the critical coupling $(k_e^2)^{opt}_{STD}$, both power and displacement from the analytical solutions and the simulation enter the saturation stage, which validates the theoretical expression of this critical coupling value in Equation (8.78).

Figure 8.16. Standard circuit: comparison of cut-in wind speeds obtained from circuit simulation and analytical solution: (a) U_{cr} versus R at different k_e^2 (b) U_{cr} versus k_e^2

Figure 8.17 Standard circuit: variation of (a) power and (b) displacement amplitude at optimal load as a function of coupling obtained from circuit simulation and analytical solution.

8.5.3.3 Validation of SCE circuit

Figure 8.18 SCE circuit: comparison of output responses obtained from circuit simulation and analytical solution at small coupling $k_e^2 = 0.01136$: (a) P_{ave} versus R (b) V_{RMS} versus R (c) u_m versus R

Figure 8.19 SCE circuit: comparison of cut-in wind speed U_{cr} versus coupling k_e^2 obtained from circuit simulation and analytical solution

Figure 8.20 SCE circuit: variation of (a) power and (b) displacement amplitude at optimal load as a function of coupling obtained from circuit simulation and analytical solution.

The analytical solution of SCE circuit is validated with the circuit simulation with the circuit diagram shown in Figure 8.8. The results of average power, RMS voltage and displacement amplitude versus load resistance are shown in Figure 8.18, with a small coupling $k_e^2 = 0.01136$. It is observed that the power, voltage and displacement are independent of load, as predicted in Section 8.2.3. The results for the large coupling $k_e^2 = 0.09087$ are not given since both analysis and simulation predict zero output as can

be reflected from Figure 8.20. Figure 8.19 displays the analytical and simulation results of cut-in wind speed versus coupling, which shows the linear relationship between U_{cr} and k_e^2 as depicted in Equation (8.97). Then the analytical and simulation results of power against coupling for different U are plotted in Figure 8.20(a), which confirms the expressions of optimal coupling in Equation (8.83) and the maximum power in Equation (8.86). The loci of the optimal coupling $(k_e^2)^{opt}_{SCE}$ is indicated in figure. The results of u_m versus k_e^2 are compared in Figure 8.20(b). In all the above figures, the analysis agrees well with the circuit simulation.

8.5.4 Comparison between different interfacing circuits

With the validated analytical solutions, comparisons of normalized power and displacement amplitude between the AC, standard and SCE circuits are shown in Figure 8.21 with the above mentioned small and large couplings $k_e^2 = 0.01136$ and 0.09087 . U is kept at 7m/s. It is noted in Figure 8.21(a) that for the small coupling, the AC circuit achieves the maximum normalized power $\hat{P}_{ave}^{opt}$ at $r_1^{opt} = 1.0085$ with $\Omega = 0.9916$, and $\hat{P}_{ave}^{opt} = 2.030E-3$ is obtained using Equation (8.67); the standard circuit reaches the maximum at $r_1^{opt} = 1.5797$ with $\Omega_{xc} = 0.9944$, where $\hat{P}_{ave}^{opt} = 1.361E-3$ is obtained with Equation (8.77); as depicted in Equation (8.60), SCE circuit generates a constant power of $\hat{P}_{ave} = 3.965E-3$ invariant with r , with $\Omega_{xc} = 0.9944$. In this small coupling condition, the SCE generates the highest power, while the standard circuit gives the lowest. The scenario changes when large coupling is involved, as given in Figure 8.21(b). Two peaks of power appear for both the AC and standard circuits. The peaks have the same magnitude of $(\hat{P}_{ave}^{max})_{AC} = (\hat{P}_{ave}^{max})_{STD} = 4.474E-3$, as expressed in Equation (8.72) and (8.82). Yet for the SCE circuit, no power output is obtained because with this large coupling, the cut-in speed U_{cr} is larger than 7m/s, which is reflected in Figure 8.19.

Figure 8.21 Comparison of output responses between AC, standard and SCE circuits: (a)(b) normalized power versus normalized resistance (c)(d) normalized displacement versus normalized resistance for two different coupling cases (a)(c) $k_e^2 = 0.01136$ (b)(d) $k_e^2 = 0.09087$

The normalized displacement amplitudes with the three circuits for small and large couplings are compared in Figure 8.21(c) and 8.21(d). It is noted that for the AC and standard circuits, the minimum $\hat{u}_m$ is always achieved at the respective r_1^{opt} for either small or large coupling, while for the SCE circuit, $\hat{u}_m$ does not change with r . Also, $\hat{u}_m$ for the SCE circuit is the smallest and for the standard circuit is the largest. This reveals that the SCE circuit experiences the strongest electromechanical damping

due to the nonlinear power extraction process, and the standard circuit experiences the weakest one. The fact that the electromechanical damping of AC circuit is stronger than that of standard circuit is because the power extraction from the piezoelectric transducer in the AC circuit is continuous, while it is discrete in the standard circuit since the rectifier is blocked when $V < V_c$.

Figure 8.22 compares the cut-in wind speed of the three interfacing circuits. It is observed that for each value of coupling, the SCE circuit always has the highest cut-in wind speed while the standard circuit owns the lowest one. This corresponds to the strongest electromechanical damping in the SCE circuit and the weakest in the standard circuit, reflected in the comparison of u_m in Figure 8.21(c) and 8.21(d). Considering Figure 8.21(a), 8.21(b) and Figure 8.22, in the aspect of harvested power, the SCE circuit is most suitable for GPEH with small coupling and relative high wind speeds. At large coupling conditions, both the AC and standard circuits are productive. The behaviors of two optimal loads for the AC and standard circuits with large coupling (Figure 8.21(b)) suggest that a small load is suitable for the situations where high current is required, like charging batteries; while a large load is associated with high voltage, suitable for high voltage requirements. On the other hand, in the aspect of displacement amplitude, the SCE circuit is always preferable for the scenarios with limited space in the harvester design in view of its smallest u_m .

Figure 8.22 Comparison of cut-in wind speed between AC, standard and SCE circuits

8.6 Chapter summary

In this chapter, the analytical solutions of a GPEH with different interfacing circuits are derived for the first time, based on the energy balance method. The standard AC, standard DC and SCE circuits are studied. Incorporating the multi-way coupling behaviors between the mechanical, piezoelectric, aerodynamic and electrical components, the explicit expressions of power, voltage and mechanical displacement are presented, which depend on a number of parameters including the effective mass, effective stiffness, damping, wind speed, load resistance and electromechanical coupling. The galloping frequency is also formulated and shown to be dependent on all the above parameters especially the load and wind speed. Based on the formulations of power given in Equation (8.31) for the AC circuit, Equation (8.51) for the standard circuit and Equation (8.60) for the SCE circuit, the optimal load and coupling are calculated for maximum power generation. The cut-in wind speeds for these circuits are also formulated.

A prototype of GPEH with a square sectioned bluff body is fabricated and tested with an AC circuit in the wind tunnel. Circuit simulations are conducted for the AC, standard and SCE circuits, based on the introduced equivalent circuit model. Subsequently, the results from experiment and simulation are employed to validate the analytical solutions, with good agreement observed for all three interfacing conditions. The behaviors of the output responses are summarized as follows.

- The galloping frequency for the AC circuit increases from the short circuit resonance frequency to the open circuit one with load, and decreases with wind speed.
- With the AC and standard circuits, the harvested power depends on both the load and coupling. There exists one optimal load for maximum power output for small coupling, and two optimal loads with equal maximum power for large

coupling. Both circuits have their respective critical coupling values, beyond which the power enters a saturation region.

- With the SCE circuit, the power is invariant with the load, but depends on the coupling only. There exists one optimal coupling value with maximum power output.
- The three circuits generate the same maximum power, with the relationship of the optimal coupling values achieving the maxima given in Equation (8.88).
- In the aspect of vibration displacement amplitude, the SCE circuit gives the smallest displacement amplitude, while standard circuit gives the largest. At the same time, the SCE is associated with the highest cut-in wind speed and the standard circuit owns the lowest one.
- The behaviors of both displacement and cut-in wind speed reveal that energy harvesting induced electromechanical damping is the strongest for the SCE while the weakest for the standard circuit.

Finally, we would like to have a thorough look at the mutual coupling behaviors of the aeroelastic energy harvesting process. They are caused by the piezoelectromechanical coupling between the three aspects of the harvesting system, i.e., the mechanical aspect, the electrical aspect and the flow field. The fluid-structure interactions cause the mechanical vibration of the elastic harvester structure, and the alternating stress field induced by the vibration results in the electrical charges in the piezoelectric element via the direct piezoelectric effect, which is a forward electromechanical coupling behavior. If nonlinear interfaces are connected, the electrical outputs are then extracted by certain regulating process. Conversely, backward electromechanical coupling plays its role in the fact that electrical charges cause mechanical deformation on the piezoelectric element via the converse piezoelectric effect, thus affecting the mechanical vibration of the harvester and further influencing the flow field. Coupling behaviors between the flow field and the piezoelectric aspect exist during the whole energy harvesting process through the mechanical strain in the harvester's composite beam. The electromechanical coupling is clearly reflected in the influences of k_e^2 and r on the

transverse displacement amplitude u_m , as have been discussed with Figure 11(c), 12(c), 14(c), 15(c), 17(b), 18(c), and 20(b), as well as their influences on the vibration frequency Ω , as shown in Figure 10(a). The aero-elastic coupling is found in the influences of wind speed U on Ω , as has been discussed for Figure 10(b), and on u_m , which is also reflected in Figure 11(c), 12(c), 14(c), 15(c), 17(b), 18(c), and 20(b). As for the aero-electrical coupling, the influence of wind speed on the generated voltage, output power as well as optimal load resistance has been explicitly reflected in the derivation process in Section 2 and Section 3. The backward influence of mechanical vibration on the flow field is revealed by the harvester vibration altering the flow pressure distribution along the harvester's composite beam. However, during calculation, the aerodynamic force is considered concentrated on the bluff body where the dominant fluid-structure interaction occurs, thus the distributed flow pressure along the beam is ignored. Moreover, unlike the VIV-based harvesters, where the aerodynamic force amplitude and vortex shedding frequency change significantly due to the mechanical motions, galloping based harvesters have no such problem. Lastly, the backward electrical-aero coupling is ignored as the influence of electrical charges on the flow field is too insignificant to be observed during our experiment.

Besides the above findings, some guidelines for applications in different scenarios are suggested, considering the different behaviors of the three circuits. SCE circuit is recommended for GPEH with small coupling and relative high wind speeds; while AC and standard circuit are suitable when large coupling is available, with small loads being better for cases requiring high current and large loads for those requiring high voltage. The theoretical solutions derived in this chapter bring significant convenience for fast and accurate evaluation of the effectiveness of GPEHs. The scheme of normalization of the output responses can also be easily used for a fair comparison between devices with various parameters.

CHAPTER 9 CONCLUSIONS AND FUTURE WORK

9.1 Conclusions

This research consists of theoretical, numerical and experimental investigations of small-scale wind energy harvesting using piezoelectric materials. Better designs, modeling and efficiency enhancement techniques have been presented and discussed. The work carried out so far can be summarized as follows.

(1) Chapter 2 delivers a thorough review on the state-of-the-art techniques of small-scale piezoelectric energy harvesting from wind flows. Mathematical modeling, equivalent circuit modeling as well as models based on computational fluid dynamics simulation are summarized and discussed in detail for energy harvesters based on various aeroelastic instabilities. Vortex-induced vibrations, galloping, aeroelastic flutter, wake-induced oscillations and turbulence-induced vibrations are considered. Subsequently, available harvester structure designs as well as efficiency enhancing techniques reported in the literature are reviewed in detail. Several preliminary research works of applying small-scale wind energy harvesters in self-powered wireless sensing systems are also summarized. The author recommends that more attentions in the future work on wind energy harvesting should be paid on integrating the harvesters with practical devices thus implementing self-powered electronic systems in real engineering applications.

(2) Chapter 3 presents a comparative study of galloping energy harvester with different cross-section tips. Experimental results confirm the superiority of square cross-section tip over the others used in the literature. A peak output power of 8.4mW is achieved which is sufficient to power small sensors. An analytical model is established and verified by experimental results. It is recommended that the tip of square section should be used for galloping-based small wind energy harvesting.

(3) The comparison study on analytical models of GPEHs presented in Chapter 4 shows that the 1DOF model, single mode and multi-mode Euler-Bernoulli distributed parameter models can successfully predict the variation of average power with the load resistance and the wind speed. Quasi-steady hypothesis is shown to be applicable for galloping force formulation in GPEHs. Single-mode is sufficient to represent the dynamics as the GPEH oscillates constantly at its fundamental mode. The distributed parameter model owns a more rational representation of the aerodynamic force, while the 1DOF model gives a better prediction on the cut-in wind speed. In general, the 1DOF can predict the electroaeroelastic behavior of a GPEH device accurately enough with its advantage of simplicity and ease of obtaining the electromechanical coupling term. Parametric study shows that U_{cr} and the growth rate of P_{ave} increases first with R_L to a certain value, then drops with R_L . When U is large enough the largest P_{ave} is obtained with the optimum R_L , which owns the largest U_{cr} as well. With the corresponding optimum R_L , increasing S_{tip} and decreasing M_{tip} can increase P_{ave} as well as reduce U_{cr} . For the case of L_p , larger L_p gives larger U_{cr} and larger growth rate of P_{ave} , yet not guarantees larger power density (power per piezoelectric volume), which is obtained at a medium L_p .

(4) Chapter 5 presents a novel 2DOF piezoelectric galloping energy harvester with a cut-out beam design and magnetic interaction, which is proved to improve the power generation efficiency in the low wind speed range with both enhanced output power and reduced cut-in wind speed. Maximum improvements occur with proper distance between the magnets (i.e. $\Delta=8\text{mm}$), when the inner beam can be excited to vibrate at high frequencies through magnetic interaction. A cut-in speed of 1m/s and nearly four time increase in power at 2.5m/s is obtained for the 2DOF harvester with $\Delta=8\text{mm}$, as compared to the 1DOF harvester. Total output power is enhanced until wind speed goes up to 4.5m/s. Experimental results confirm that the proposed energy harvester is a viable power supplier for wireless sensing nodes in indoor monitoring systems or highly urbanized areas where only small-scale wind flows are available.

(5) Chapter 6 demonstrates that with the easy method of adding a beam stiffener, power generation capabilities of the galloping-based, VIV-based and flutter-based aeroelastic energy harvesters are significantly increased. The principle is to increase the electromechanical coupling by modifying the fundamental mode shape of the harvester with the introduced beam stiffener. Theoretical models are established for the three types of aeroelastic energy harvesters, including the electromechanical model based on the Euler-Bernoulli beam theory and Kirchhoff laws which represents the coupling effects between the structural and electrical components, as well as the aerodynamic model for the calculations of the aerodynamic force due to the specific aeroelastic phenomenon. Wind tunnel experiment is conducted to confirm the effectiveness of the propose method in a galloping harvester. Subsequently, nonlinear theoretical analysis shows that the maximum power can be boosted by more than a dozen times, with comparable or even smaller displacement. A maximum power of 139.74mW is achieved by the galloping-based harvester at $U=15\text{m/s}$, which is significantly higher than most of the harvesters' output reported in the literature and sufficient to power most commercial wireless sensors currently used in real applications. For the VIV-based harvester, the lock-in region is also broadened. The flutter-based harvester is found to have the highest efficiency, and is suggested to operate in large range of wind speeds; the VIV-based harvester is recommended for environments with fixed and small wind speed, while the galloping-based harvester has much higher power output than the former two at high wind speeds. It is concluded that the addition of a beam stiffener is an easy, effective and practical method for the enhancement of aeroelastic energy harvesting devices.

(6) Chapter 7 investigates the feasibility of SCE interface in enhancing the performance of GPEH system. It is determined that over the considered range of R_L , the output power generated by the self-powered SCE interface is uniform and independent of the value of R_L ; while the power achieves a single maximum at one optimal load resistance R_{Lopt}^1 for small k_e^2/ζ below $(k_e^2/\zeta)_{STD}$, and branches out beyond $(k_e^2/\zeta)_{STD}$ at R_{Lopt}^2 and

R_{Lopt}^3 which produce the same amount of optimal power, with R_{Lopt}^1 becoming a minimum. R_{Lopt}^1 , R_{Lopt}^2 and R_{Lopt}^3 varies depending on k_e^2/ζ . The tip displacement amplitude with the self-powered SCE interface is also uniform and independent of R_L at any value of k_e^2/ζ , similar to the output power; while the displacement amplitude with the standard circuit exhibits a single minimum at R_{Lopt}^1 even beyond $(k_e^2/\zeta)_{STD}$. At a specific U , the values of attainable maximum power are equal for the two circuits at their corresponding optimal k_e^2/ζ , but the self-powered SCE has a smaller optimal coupling as $(k_e^2/\zeta)_{SCE}=1/4*(k_e^2/\zeta)_{STD}$ for all considered values of U . Three main advantages of SCE in GPEH are summarized as: first, the output power from SCE is independent of R_L thus no additional impedance matching process is required; second, the SCE circuit saves piezoelectric materials by 75% compared to the standard circuit since it achieves the equal maximum power at a much smaller k_e^2/ζ ; third, the smaller displacement amplitude with SCE alleviates the fatigue problem and enhances the durability of the harvesting system.

(7) In Chapter 8, the analytical solutions of a GPEH with different interfacing circuits are derived for the first time based on the energy balance method. The explicit expressions of power, voltage and mechanical displacement are presented, which depend on a number of parameters including the effective mass, effective stiffness, damping, wind speed, load resistance and electromechanical coupling. The galloping frequency is also formulated and shown to be dependent on all the above parameters especially the load and wind speed. Based on the formulations of power given in Equation (8.31) for the AC circuit, Equation (8.51) for the standard circuit and Equation (8.60) for the SCE circuit, the optimal load and coupling are calculated for maximum power generation. The cut-in wind speeds for these circuits are also formulated. Validated by wind tunnel experiment and/or circuit simulation, the analytical solutions provide significant insights into the behaviors of the dynamic responses, and bring significant convenience for fast and accurate evaluation of the effectiveness of GPEHs. The scheme of normalization of the output responses can also be easily used for a fair comparison between devices with various parameters.

9.2 Future work

This section presents a preliminary work that has been done, which deserves further detailed studies, as well as some recommendations for future work.

9.2.1 Enhancement of Galloping-based Wind Energy Harvesting by Synchronized Switching Interface Circuits⁸

Using a self-powered S-SSHI circuit, Liang and Liao (2012) have experimentally and analytically proved that the power of a conventional vibration energy harvester can be boosted up to 200% more than that with the standard circuit. In this section, the energy harvesting capability of a GPEH using the SSHI circuits is investigated. The performances of S-SSHI and P-SSHI are compared with that of a standard circuit, with a focus on the power output at various wind speeds and the cut-in wind speed. Wind tunnel experiment is conducted to verify the equivalent circuit simulation results. Energy losses in the employed SSHI circuit are also discussed.

9.2.1.1 SSHI Interface circuit

Figure 9.1 (a) S-SSHI interface circuit and (b) waveforms of voltage and tip displacement

⁸ Content based on this section has been published as “Liya Zhao, Junrui Liang, Lihua Tang, Yaowen Yang, Haili Liu (2015) Enhancement of Galloping-based Wind Energy Harvesting by Synchronized Switching Interface Circuits, Proceedings of SPIE Smart Structures and Materials+ Nondestructive Evaluation and Health Monitoring, 9431: 943113”.

Figure 9.2 (a) P-SSHI interface circuit and (b) waveforms of voltage and tip displacement

The S-SSHI interface shown in Figure 9.1 and the P-SSHI interface shown in Figure 9.2 are considered. A drawback of the simple standard circuit is that during a certain interval in each cycle of vibration, the energy is returned back from the electrical part to the mechanical part, restraining the efficiency of power extraction. This issue can be overcome in the SSHI circuit by adding an inductive switch path to enhance the nonlinear power extraction process. The inductive switch path consisting of a switch S and an inductor L_i is connected in series to the bridge rectifier in S-SSHI (Figure 9.1(a)) and in parallel in P-SSHI (Figure 9.2(a)). Regardless of the series or parallel connection, S is triggered on once the transverse displacement of the bluff body $w(L_t)$ reaches its extreme values to form an S - L_i - C_p loop with the piezoelectric element. Then S is triggered off after half of the oscillating period of the loop to complete a natural inversion of the piezoelectric voltage V_p (Liang and Liao, 2012). Typical waveforms of the two interface circuits are given in Figures 9.1(b) and 9.2(b).

9.2.1.2 Wind tunnel experiment with prototypes of GPEH and circuit hardware

Prototype of GPEH and circuit hardware

Wind tunnel experiment with a fabricated GPEH prototype and a built up circuit hardware is carried out to evaluate the wind power extraction performance of the GPEH integrated with SSHI. The piezoelectric patch used is MFC M2814-P2 (Smart Materials Corp.). The air density ρ , damping ratio ζ and short circuit fundamental

frequency ω_n is 1.204kg/m³, 0.011 and 16.67Hz, respectively. Other properties of the GPEH prototype are listed in Table 9.1.

Table 9.1 Properties of composite cantilever and connected bluff body

Properties	Cantilever	
	Substrate beam	Piezoelectric sheet
Length L_t, L_p (mm)	130	28
Width b_s, b_p (mm)	20	14
Thickness h_s, h_p (mm)	0.6	0.3
Material	Aluminum	MFC M2814-P2
Mass Density (kg m ⁻³)	2700	5440
Capacitance C_p (nF)	--	25.7
Young's Modulus E_s, E_p (GPa)	69	30.336
	Bluff body	
Mass M_{tip} (kg)	0.0018	
Length l (mm)	100	
Cross section $h \times h$ (mm)	20×20	
Aerodynamic coefficients A_1, A_2, A_3	2.3, 0, -18

Figure 9.3 Schematic of built-up self-powered S-SSHI and P-SSHI circuit hardware

Figure 9.4 Hardware configuration for (a) standard circuit (b) self-powered S-SSHI circuit and (c) self-powered P-SSHI circuit.

Table 9.2 Circuit component values

	Bluff body
R_1 and R_2 (k Ω)	200
C_1 and C_2 (pF)	680
Diodes (D_1 to D_{12})	1N4004
PNP transistors (T_1 and T_4)	TIP32C
NNP transistors (T_2 and T_3)	TIP31C
C_f (μ F)	20
L_i (mH)	47
S-SSHI, r (Ω)	252.4
P-SSHI, r (Ω)	283.0

A circuit hardware is built up and connected to the GPEH prototype for the wind tunnel test. The schematic for the circuit hardware is shown in Figure 9.3. By changing the connection with the DIP switch, it can work as a self-powered S-SSHI, or a self-powered P-SSHI. Removing the inductor L_i in P-SSHI configuration, it can work as the standard circuit as well. In the self-powered SSHI configurations, electronic breakers composed of the envelope detector, comparator and switch are used to automatically detect the displacement extremes and perform switching actions, avoiding the requirement of an external displacement sensor and a switch controller. Both the maximum and minimum electronic breakers are included. The improved design of electronic breaker with complementary transistors topology is employed here, which is proposed by Liang and Liao (2012). With this design, all the isolating resistors in the breaker in Richard et al. (2007). and Lallart and Guyomar (2008) that consume some harvested energy can be removed. The detailed working principles and analysis of the employed self-powered SSHI circuit are provided in Liang and Liao (2012). In Table 9.2, the load resistances r represent the total equivalent series resistance of the non-ideal switching path, of which the values for S-SSHI and P-SSHI are obtained by measuring the respective reference voltages at the instant of the first voltage inversion.

Overall Experiment Setup in Wind Tunnel

Figure 9.5 shows the entire wind tunnel experiment setup. The voltage signal across the piezoelectric patch and the terminal load R_L is measured by the NI 9229 data acquisition module (National Instruments). The wind speed is measured with a hotwire anemometer.

Figure 9.5 Overall experiment setup in wind tunnel

9.2.1.3 Circuit Simulation

By making sure the fundamental oscillation period of the harvester is much larger than the resonant period of the LC circuit, the voltage inversion can be regarded to happen instantly. The entire GPEH system with the integrated interfaces is simulated in the circuit simulator SIMetrix. A differential voltage probe is used in the simulation to obtain the voltage V_{rect} across R_L . The output power is calculated by $P=(V_{rect})^2/R_L$.

9.2.1.4 Results and discussion

Power at Various Wind Speeds

(c)

Figure 9.6 Measured and predicted output power versus terminal load at 4m/s, 5m/s and 6m/s. (a) Standard circuit (b) S-SSHI circuit (c) P-SSHI circuit.

Under three different wind speeds of $U=4\text{m/s}$, 5m/s and 6m/s , the measured and predicted variations of average power as a function of load resistance for the standard, S-SSHI and P-SSHI interfaces are compared in Figure 9.6. In general, the simulation results agree well with the wind tunnel test measurements. Discrepancies get more significant at 6m/s for the standard and S-SSHI circuits. For all three circuits, the power increases with the wind speed. Comparing the amplitudes of the power from the three circuits, it is observed that P-SSHI gives the highest output power, with great enhancement compared to that with the standard circuit especially when the wind speed is high, e.g., at 6m/s the power is increased from 1.6mW with the standard circuit into 2.3mW with P-SSHI in the experiment, corresponding to a 43.75% increase. Enhancement is also predicted for S-SSHI by circuit simulation, yet is not reflected by the experimental results. There exist an optimal resistance R_{optimal} for maximum output power for all three circuits. With the variation of the wind speed, the value of R_{optimal} keeps constant for the standard circuit (actually R_{optimal} should be very slightly increased with the wind speed since the vibration frequency is slightly decreased when U is increasing according to the theoretical response solutions). However, R_{optimal} varies greatly for S-SSHI and P-SSHI at different wind speeds, which is more obviously observed from the simulation. When wind speed increases within the considered range

of $U=4\text{m/s}\sim 6\text{m/s}$, the predicted R_{optimal} decreases from $600\text{k}\Omega$ to $300\text{k}\Omega$ for S-SSHI, while R_{optimal} for P-SSHI increases from $400\text{k}\Omega$ to $800\text{k}\Omega$.

Figure 9.7 Measured and predicted power versus wind speed, with $R_L=412\text{k}\Omega$ for standard circuit, $R_L=375\text{k}\Omega$ for S-SSHI and $R_L=445\text{k}\Omega$ for P-SSHI.

Based on the measured responses, the respective R_L that gives relative high output power for all three wind speeds ($4\text{m/s}\sim 6\text{m/s}$) is chosen for each circuit, and then is used to investigate the response of power versus wind speed, as shown in Figure 9.7. Values of R_L used for each circuit are: $R_L=412\text{k}\Omega$ for the standard circuit, $R_L=375\text{k}\Omega$ for S-SSHI and $R_L=445\text{k}\Omega$ for P-SSHI. For P-SSHI, both experiment and simulation results show enhancement of output power compared to the standard circuit, with the degree of enhancement gets larger at higher wind speeds; while for S-SSHI, only circuit simulation captures the power enhancement, yet the experimental results are more or less comparative with those for standard circuit. This implies that a great energy loss occurs in the hardware S-SSHI which has not been successfully taken into account.

Inspecting the simulation responses, a general trend of power from the three circuits are observed at relatively high wind speeds, i.e., $\text{S-SSHI} > \text{P-SSHI} > \text{Standard}$. Moreover, the benefits of SSHI become more significant when the wind speed gets higher. At small wind speeds like below 3.8m/s , although not captured by experiment, the SSHI circuit

loses its benefit compared to standard circuit. These two issues are similar to the vibration based energy harvester as concluded in Liang and Liao (2012). Moreover, it is observed during simulation that the output power of self-powered S-SSHI is sensitive to the values of the capacitors C_1 and C_2 (Figure 9.3). The superiority of P-SSHI over S-SSHI and the great energy loss in S-SSHI deserve further investigations.

Discussions on the energy loss in self-powered SSHI

The discrepancy of power responses between experiment and simulation becomes more obvious at higher wind speed. Errors are partially due to the representation of the aerodynamic galloping force, since it is established with the assumption of small angle of attack. The large angle of attack at high wind speed may surpass the limit. Also, the energy loss becomes larger, especially for the S-SSHI. The energy loss during power extraction with SSHI interfaces can be due to the imperfect voltage inversions, voltage drops in the electronic components like the diodes and transistors, and internal loss within the piezoelectric element (Liang and Liao, 2012). The equivalent series resistance of the switching path r is obtained at $U=6\text{m/s}$. The value may not accurately represent the switching resistance at higher wind speed.

9.2.1.5 Section summary

This section investigates the energy harvesting capability of a galloping-based wind energy harvester using the SSHI interfaces and compares the performance of S-SSHI and P-SSHI with that of a standard interface. Wind tunnel experiment is carried out with the prototyped GPEH and a built up SSHI circuit hardware. System-level circuit simulation is conducted based on the equivalent circuit model of the GPEH. The benefits of SSHI compared to standard circuit become more significant at a high wind speed. In the low wind speed range, SSHI circuits lose its advantage. A power increase of 43.75% is obtained with P-SSHI compared to the standard circuit, boosting the power output from 1.6mW to 2.3mW. In addition, higher cut-in wind speed with the S-SSHI and P-SSHI with optimal resistances reveals the higher induced electrical

damping and higher power extraction ability than the standard circuit. Energy loss becomes more severe at high wind speeds, especially for the S-SSHI, which deserves careful investigation in the future. Moreover, accurate analytical solutions of mechanical and electrical responses deserve future efforts in order to provide a clear design criterion as well as ensure an efficient performance optimization process.

9.2.2 Theoretical modeling of wind energy harvesters with complex structural configuration

In Chapter 5, a 2DOF GPEH is proposed for power efficiency enhancement at low wind speed range, with a cut-out beam configuration and nonlinear stiffness induced by the magnetic interactions. The efficiency enhancing abilities is investigated by experimental measurement. Future efforts are demanded to derive a theoretical model to predict the mechanical and electrical responses of such harvesters with complex structural configurations thus to facilitate the parameter optimization for developing such structurally enhanced energy harvesters.

9.2.3 Integrated CFD-FEM- Circuit simulation models

In order to better simulate the electro-fluid-structural coupling behavior of small-scale wind piezoelectric energy harvesting systems, especially for the case with complex structural configurations and sophisticated interface circuits, future efforts can be devoted to establishing an integrated modeling approach, where CFD, FEM and circuit simulation are related and integrated into a linked model stream, with CFD predicting the aerodynamic forces and FEM predicting the mechanical responses which is further electromechanically coupled with the circuit model simulating the nonlinear power extraction process.

9.2.4 Small-scale wind energy harvesters installed on vehicles or traffic route sides

Vehicles like airplanes, cars, trucks or even bicycles are subjected to nearly

constant-speed wind flows during travelling. Telegraph poles, or advertising boards near the railways, bridges and highways will experience high-speed gust wind flows when travelling vehicles pass by them. At the same time, both of the two cases mentioned above undergo wide band random base vibrations. Future work can be conducted to develop energy harvesters that operate efficiently in smooth/gust flows with random base vibrations.

9.2.5 Miniature portable power generator

Human motions like running and waking will induce gentle breezes in the skin. If a miniature power generator (or MEMS) is attached onto a human body, the flow energy of the gentle breezes can be beneficially harnessed. A specially designed air inlet will enlarge the accessing wind speed. At the same time, the miniature power generator should have the frequency up-conversion ability to account for the low frequency vibrations (running/walking pace) during human motions.

9.2.6 Small-scale wind energy harvesters based on hybrid aeroelasticities

As shown in Chapter 2 and Chapter 6, the VIV-based energy harvesters can be adjusted to give high output power at a target low wind speed, yet the drawback is the wind speed bandwidth of effective power generation is quite narrow. Galloping/flutter-based harvesters have wide effective wind speed range, yet their output power near the cut-in speed is quite low. Experimental work is being carried out to develop a harvester with hybrid VIV and divergent aeroelasticity (galloping or flutter) in order to obtain both wide effective wind speed range and high output power at a low wind speed (near the cut-in speed).

9.2.7 Multi-directional wind energy harvester

Most of the reported wind energy harvesters, including those presented in this thesis, require an exact wind direction to operate efficiently, e.g., the wind must flow in

parallel with the cantilever for the APEH in Chapter 6. By properly designing the bluff bodies, like combining different bluff bodies together with various orientations, the harvester will be able to operate when the flow direction changes, which is much likely to appear in outdoor environments.

9.2.8 Large arrays of piezoelectric energy harvesters for practical deployments

In order to supply sufficient power for an electronic system, wind energy harvesters will be required to be arranged in large arrays for power accumulation. Future efforts are required to study the interactions between the harvester units and develop proper circuit to efficiently extract power from the harvesters which are likely working out of phase with each other.

9.2.9 Fatigue test and durability enhancement studies

When applying the wind energy harvesters in real engineering applications, their work duration should not be shorter than that of a normal electrochemical battery. But most of the studies in this area are at the stage of structural and circuit configuration designs. Fatigue test of these small-scale wind energy harvesters needs to be carried out. Future efforts should be devoted to enhancing the durability of these harvesters.

9.2.10 Energy storage technique

Finally, since the energy generated by harvesters are usually small to directly power most electronics, the harvesting system should be integrated with efficient energy storage medium to ensure the harvested energy to be usable as the power supply for wireless sensors and other electronic devices. Future efforts should also be devoted to the development and improvement of energy storage techniques.

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APPENDIX: PUBLICATIONS

Journal Papers

- **Zhao, L.** and Yang, Y. (2015) Small-scale wind energy harvesting: modeling, design, efficiency enhancement and applications in self-powered wireless sensing systems, *Journal of Intelligent Material Systems and Structures* (in review).
- **Zhao, L.** and Yang, Y. (2015) Analytical solutions for galloping-based piezoelectric energy harvesters with various interfacing circuits, *Smart Materials and Structures*, 24(7), 075023.
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Conference Papers

- **Zhao L.**, Liang J., Tang L., Yang Y. and Liu H. (2015) Enhancement of Galloping-based Wind Energy Harvesting by Synchronized Switching Interface Circuits, In: *Proceedings of SPIE Smart Structures and Materials+ Nondestructive Evaluation and Health Monitoring*, 9431: 943113.
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