Vibrations Analysis of Vertical Axis Wind Turbine

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in

Mechatronics

By

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Abstract

This is a thesis concerned with Vertical Axis Wind Turbines (VAWT) and researches in Vibration. The Vertical axis wind turbines (VAWT) compared with Horizontal axis wind turbines (HAWT) has a lower efficiency. However, the supporting structure of VAWT structure is relatively simple. It is suitable in poor wind conditions. Besides, VAWT also generates lower noise and vibrations.

In this study, Finite element method is used to calculate and obtain natural Frequencies of Mechanical vibration. A simple model of Vertical Axis Wind Turbines Natural Frequencies under three conditions was used, to determine and analyse the relationship between Vibration and the shape of the Wind Turbines. This study offers a solution, that assists in analysing the vibration of Vertical Axis Wind Turbines and also provides improvements, in order to help the design and development Vertical Axis Wind Turbines. A way to reduce the vibration of the VAWT is offered in order to increase the lifetime and efficiency of Vertical Axis Wind Turbines.

Acknowledgement

Studying toward to Master degree in Massey has been the most important and extraordinary experience in my life. This training progress has improved the way of my thinking and ability of solving problems.

I would like to thank my thesis adviser Dr. Jen-Yuan (James) Chang whose expertise, advice and continuous support during this course of the research. I consider myself to be extremely fortunate to learn from and work with him.

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Contents

Chapter 1

Introduction

1.1. The importance of energy sources

Nowadays, energy shortages are becoming an increasingly relevant question. It concerns the climate change of our planet, the survival and development of the human race, peace between countries; freedom and the quality of life of the individual. The world we live in today is one of global warming, deteriorating habitats, climate change, increasing oil prices, financial crises and wars. Much of this can be attributed to the depletion and lack of oil sources, and the methods countries are using to combat this. This is causing the world to furiously devote their sources in search of a solution to energy shortage; every developed country in the world is focusing their research on new and renewable energy sources, with many governments pouring funds into their renewable energy research.

1.2. The history and advantages of Wind Energy

1.2.1 The history of wind energy

1.2.1 .1 Mechanical power

Sailors have been using wind power for thousands of years, and architects have used wind as a part of natural ventilation in buildings since ancient times. The use of wind to provide mechanical power came much later. The wind wheel of the Greek engineer Heron of Alexandria in the 1st century AD is the earliest known instance of using a wind-driven wheel to power a machine.

The first practical windmills were in use in Iran at least by the 9th century and possibly as early as the 7th century. Windmills were used extensively in Northern Europe to grind flour from the 1180s, and wind pumps were used to drain land for agriculture and for building.

In the US, the development of the "water-pumping windmill" was the major factor in allowing the farming and ranching of vast areas otherwise devoid of readily accessible water. Wind pumps contributed to the expansion of rail transport systems throughout the world, by pumping water from water wells for steam trains.(1)(2)(3)(4)

1.2.1.2 Electrical power

In July 1887, a Scottish academic, Professor James Blyth, built a cloth-sailed wind turbine in the garden of his holiday cottage in Marykirk and used the electricity it produced to charge accumulators which he used to power the lights in his cottage. His experiments culminated in a UK patent in 1891. In the 1890s, the Danish scientist and inventor Poul la Cour constructed wind turbines to generate electricity, which was used to produce hydrogen and Oxygen by electrolysis and a mixture of the two gases was stored for use as a fuel. La Cour was the first to discover that fast rotating wind turbines with fewer rotor blades were the most efficient in generating electricity and in 1904 he founded the Society of Wind Electricians.

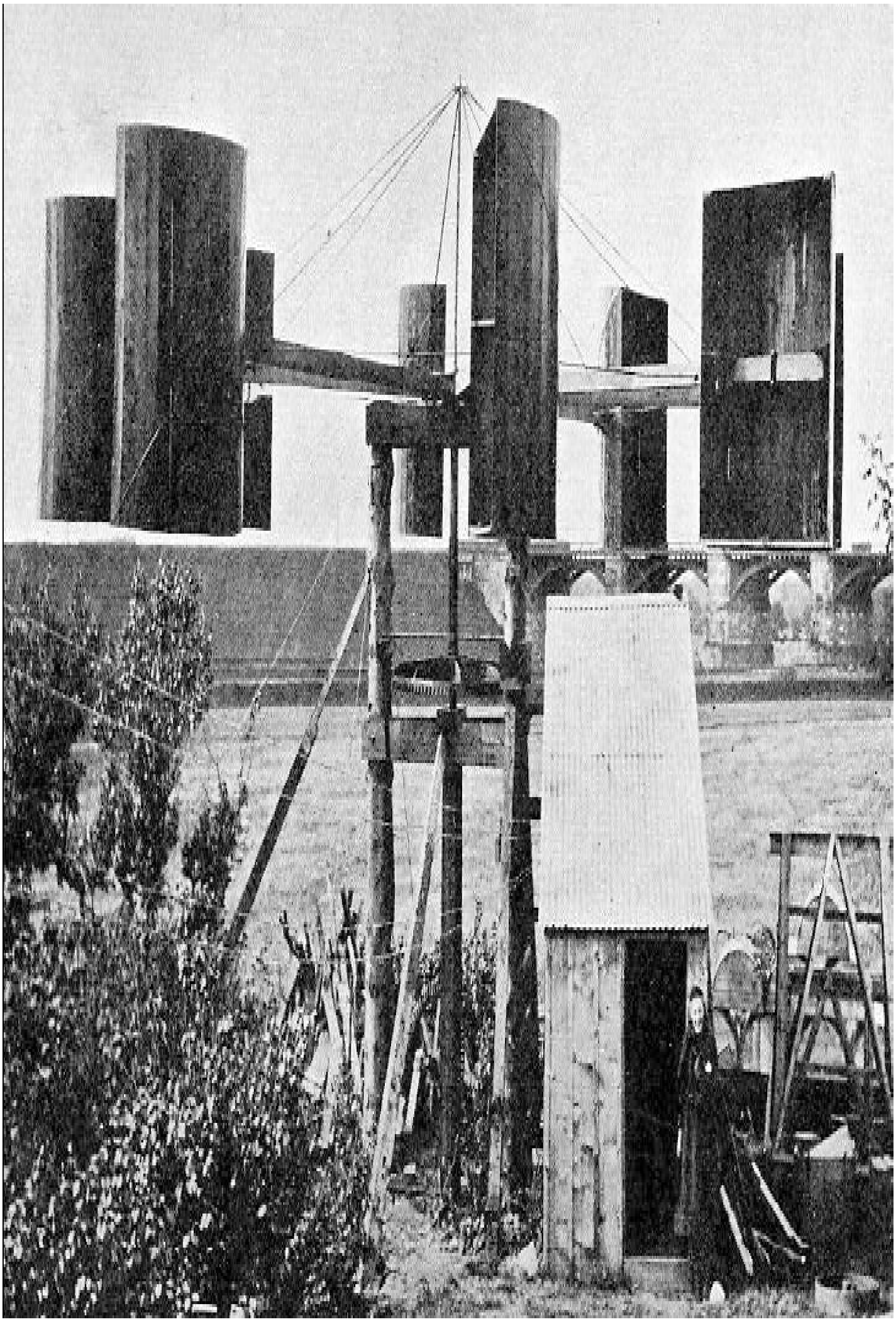


Fig-1-2-1Blyth's windmill at his cottage in Marykirk in 1891

By the mid-1920s, wind generators developed by companies found widespread use in the rural areas of the US but by the 1940s the demand for more power and the coming of the electrical grid throughout those areas made these small generators obsolete.

The Darrieus wind turbine is a type of vertical axis wind turbine (VAWT) used to generate electricity from the energy carried in the wind. The turbine consists of a number of aerofoil's usually—but not always—vertically mounted on a rotating shaft or framework. This design of wind turbine was patented by Georges Jean Marie Darrieus, a French aeronautical engineer in 1931. Fig-1-2-2-1

In 1975 the United States Department of Energy funded a project to develop utility-scale wind turbines. The NASA wind turbines project built thirteen experimental turbines which paved the way for much of the technology used today. Since then, turbines have increased greatly in size with the Enercon E-126 (Fig-1-2-2-2) capable of delivering up to 7 MW. Wind turbine production has expanded to many countries and wind power is expected to grow worldwide in the twenty-first century.(5)(6)(7)(8)

Fig-1-2-2-1 Darrieus VAWT

Fig-1-2-2-2 Enercon E-126 standard HAWT

1.2.2 The current situation and future prospects

1.2.2.1The Current Situation

Worldwide there are now many thousands of wind turbines operating. World wind generation capacity more than quadrupled between 2000 and 2006, doubling about every three years. The United States pioneered wind farms and led the world in installed capacity in the 1980s and into the 1990s. In 1997 German installed capacity surpassed the U.S. and led until once again overtaken by the U.S. in 2008. China has been rapidly expanding its wind energy in the late 2000s and passed the U.S. in 2010 to become the world leader.

Fig1-2-3 Wind power farm

Fig-1-2-4 Usage of small scale VAWT in modern life

Below is a graph of the wind power production of each country in 2011

Table 1-2-1 wind power production

Country	Wind power capacity (MW) provisional	% of total
China	62,733‡	26.3
United States	46,919	19.7
Germany	29,060	12.2
Spain	21,674	9.1
India	16,084	6.7
France	6,800‡	2.8
Italy	6,747	2.8
United Kingdom	6,540	2.7
Canada	5,265	2.2
Portugal	4,083	1.7
(rest of world)	(32,446)	(13.8)
World total	**238,351 MW**	**100%**

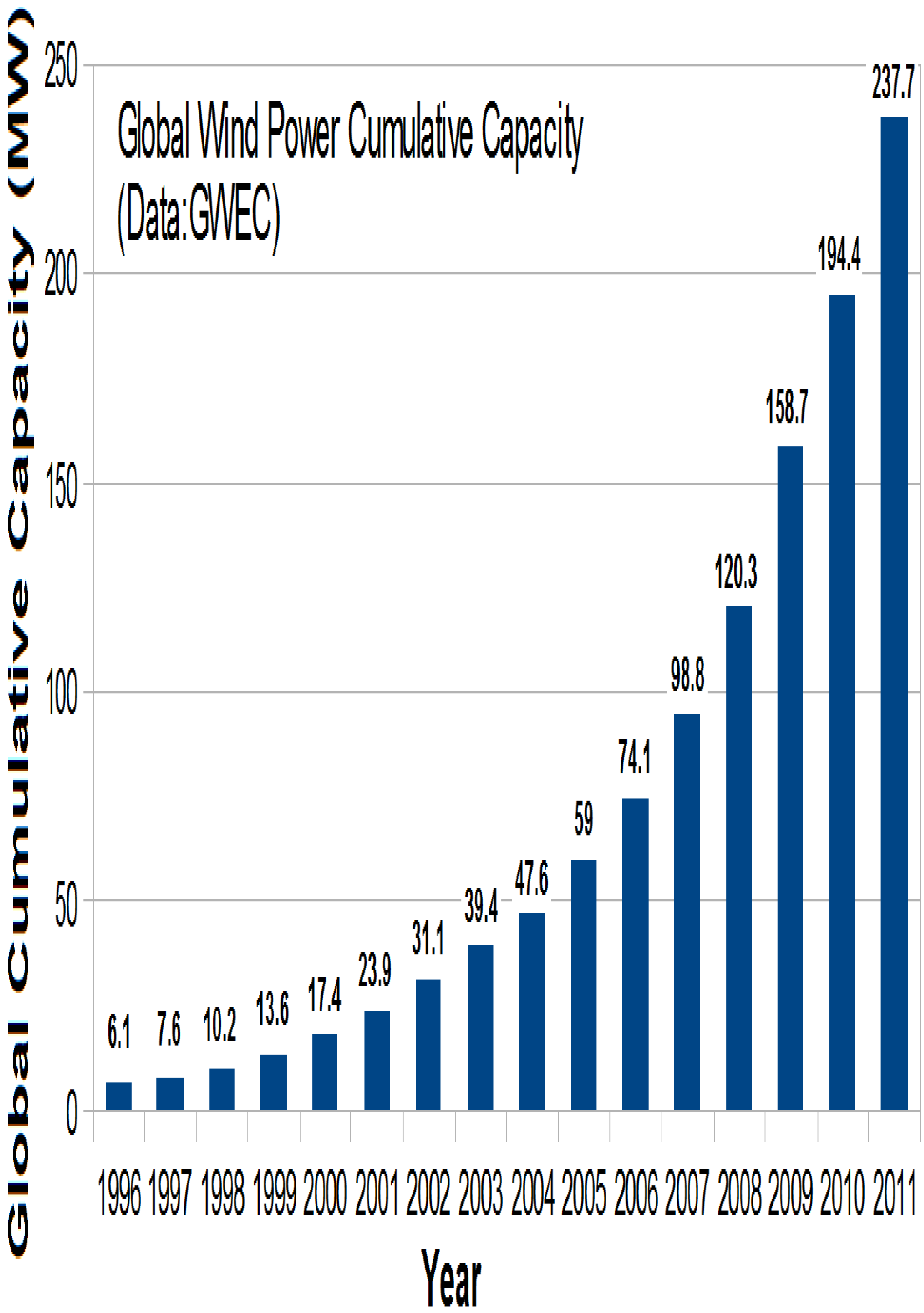


Fig-1-2-5 wind energy produced worldwide wind energy produced worldwide

This graph shows that the amount of wind energy produced worldwide has constantly been on the exponential increase from 1996 through to 2011. And the total amount of wide energy produce last year in 2011 was nearly ten times what it was 10 years ago.

Currently, 48 percent of total wind energy capacity is in Europe, with 24 percent in North America and 25 percent in Asia. Wind energy is one of the fastest growing forms of new electricity generation throughout the world, with capacity growing on an annual basis at around 30%.

The International Energy Agency expects global electricity demand to grow by 1.5 to 2.5 percent per year. Much of this increased demand will be met with wind energy as it is a cost effective and proven solution that is available today. Although the wind power industry was affected by the global financial crisis in 2009 and 2010, a BTM Consult five year forecast up to 2013 projects substantial growth. Over the past five years the average growth in new installations has been 27.6 percent each year. In the forecast to 2013 the expected average annual growth rate is 15.7 percent. More than 200 GW of new wind power capacity could come on line before the end of 2013. Wind power market penetration is expected to reach 3.35 percent by 2013 and 8 percent by 2018.

Table 1-2-2 Global installed wind power capacity

	Capacity installed in 2009 (MW)	Total capacity end 2009 (MW)
Asia	15,442	39,610
Europe	10,526	76,152
North America	10,946	38,383
Latin America	622	1274
Pacific region	577	2221
Africa and Middle East	230	865
Total	**38,343**	**158,505**

Growth trends

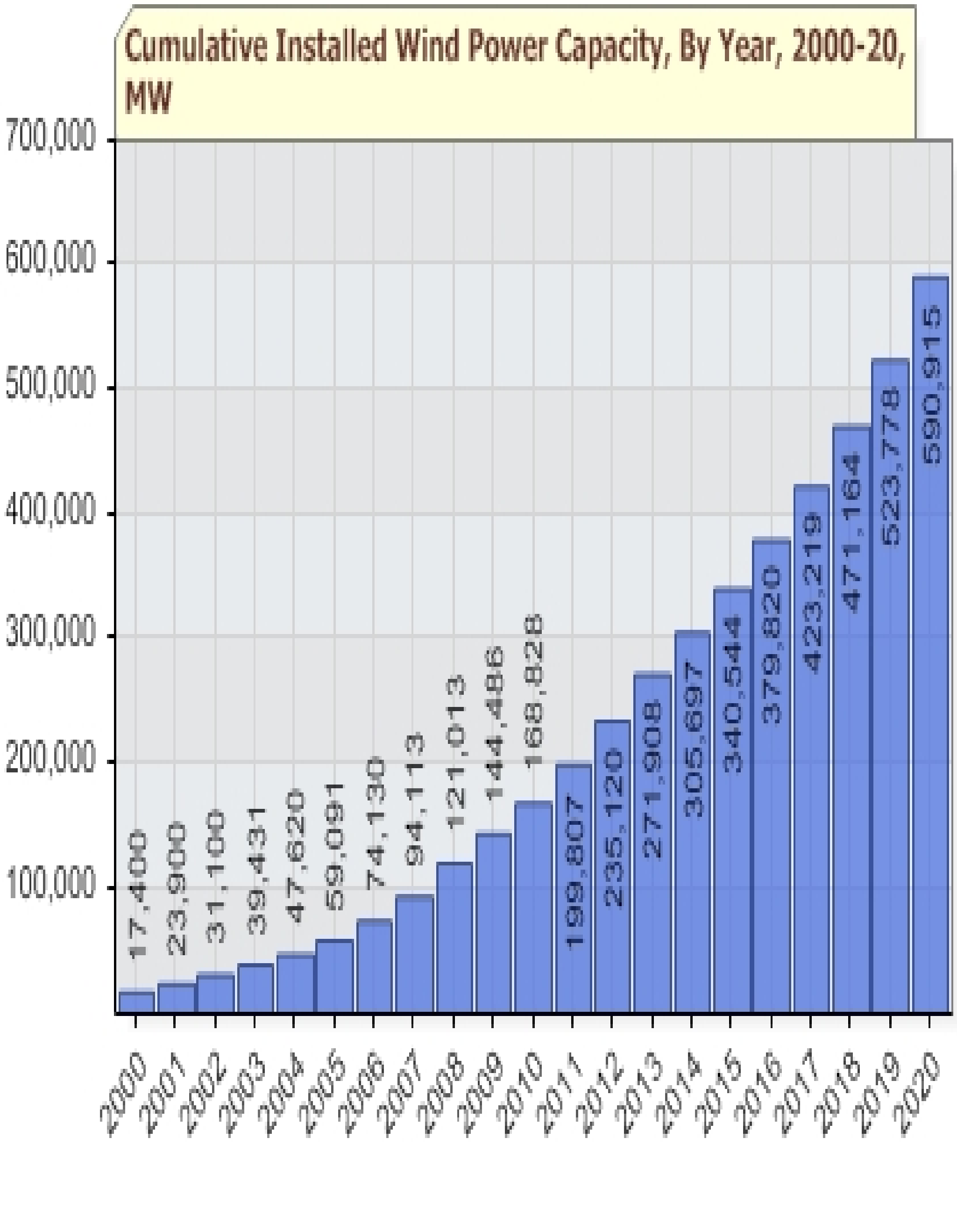


Fig-1-2-6
Worldwide installed capacity 2000–2020 [MW], developments and prognosis. Data source: WWEA

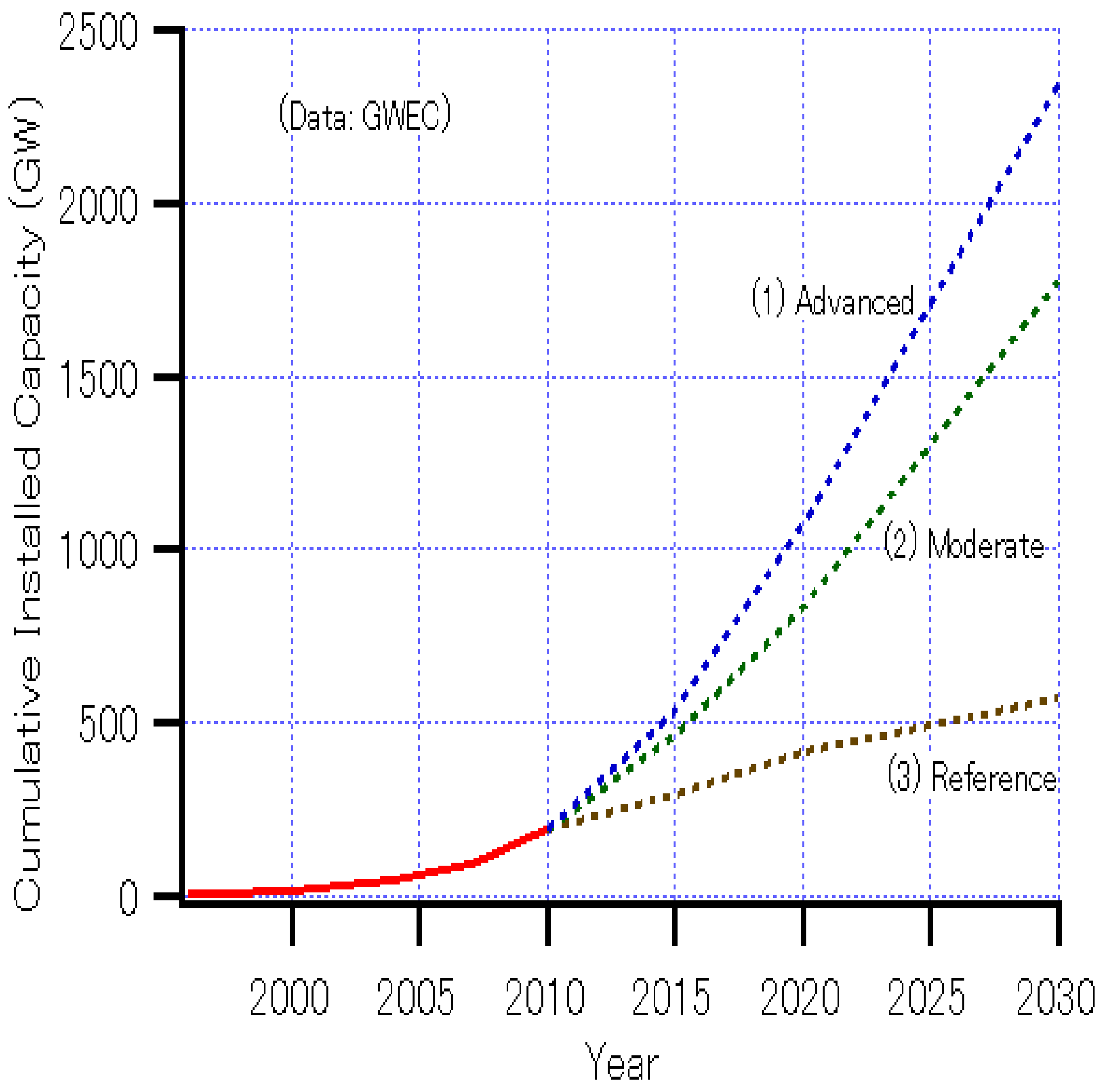


Fig-1-2-7

Worldwide installed wind power capacity forecast (Source: Global Wind Energy Council).

As shown on the graphs above, the total of installed wind energy farms are constantly on the increase, and by 2020, the total power output is expected to be 2000000 megawatts.

In the coming years, significant growth is expected in Asia, with Asia expected to overtake Europe as the region with largest installed capacity by 2014. Industry growth in Asia is fuelled by activity in China, which added a staggering 13.8 gig watts of new capacity in 2009.

In Europe, offshore wind farms are set to account for an increasing share of new capacity. (9)(10)(11)(12)(13)(14)(15)

Here in New Zealand, wind energy capacity will continue to increase in line with international trends. By the mid 2011 capacity will have increased to about 610 MW, up from 497 MW at the end of 2009.

Wind power, as an alternative to fossil fuels, is plentiful, renewable, widely distributed, clean, produces no greenhouse gas emissions during operation and uses little land. Any effects on the environment are generally less problematic than those from other power sources. As of 2010 wind energy production was over 2.5% of worldwide power, growing at more than 25% per annum. The overall cost per unit of energy produced is similar to the cost for new coal and natural gas installations.

Wind energy is one of the most renewable energy sources around, its source is seemingly endless and does little harm to the environment.

As an island New Zealand has perfect conditions to harness wind energy, and the government should take advantage of this to fully utilize our natural wind energy sources because it can lessen our reliance on other energy sources that may do harm to the environment. The study Mechanical Vibrations is vital to harnessing this energy and vital to the wellbeing of our nation.(16) Fig-1-2-8; Fig-1-2-9

The development of large HAWT units has received adequate funding and support form Governments and power companies. But the growth and development of small VAWT has been slow in comparison.

Fig-1-2-8 New Zealand’s oldest wind turbine in Wellington in 1993

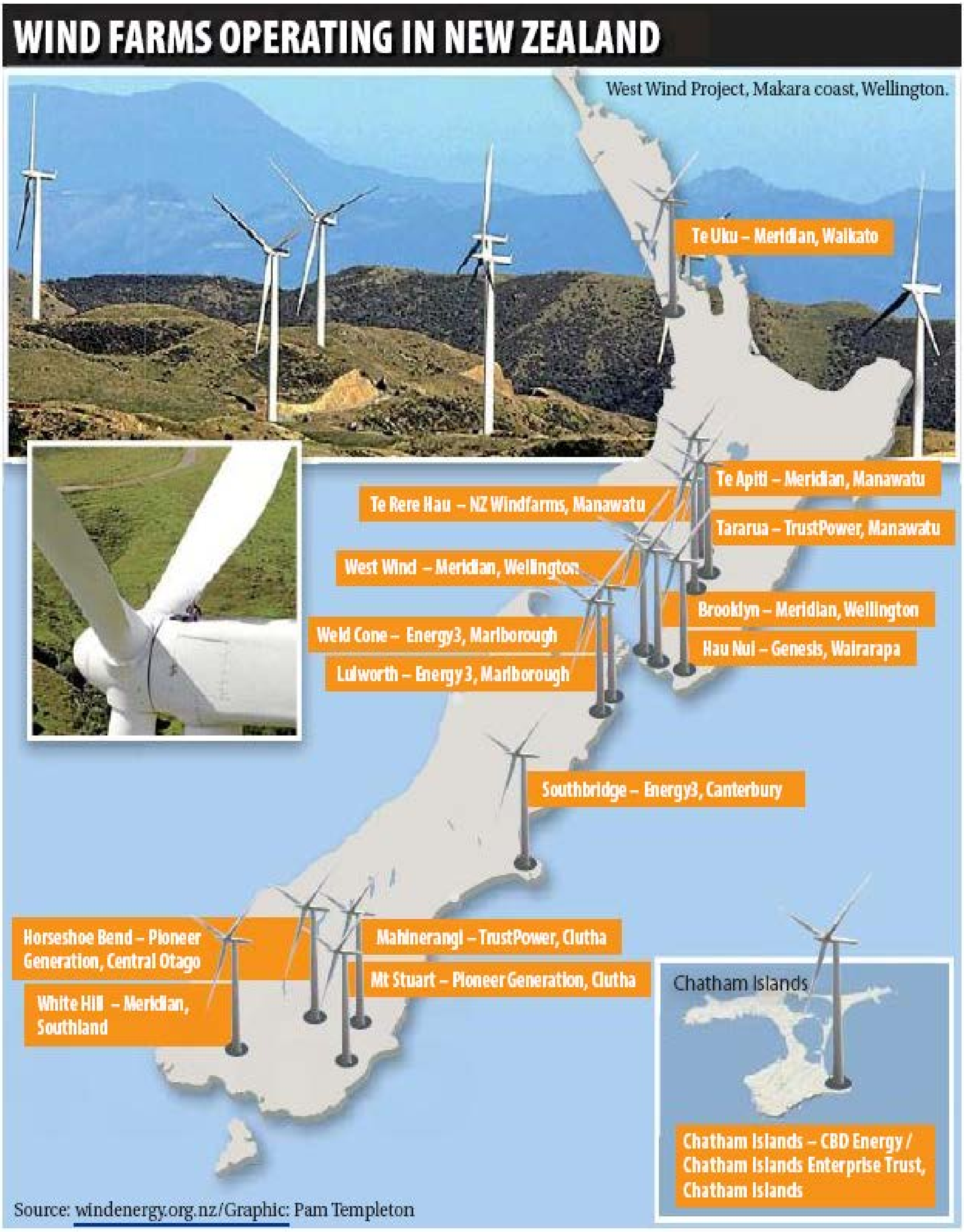


Fig-1-2-9 Various wind farm in New Zealand as of 2010

1.3 Comparisons between VAWT and HAWT

1.3.1 The advantages and disadvantages of VAWT

Wind turbines are classified as VAWT and HAWT

Advantages of vertical axis wind turbines (VAWT):

VAWTs offer a number of advantages over traditional horizontal-axis wind turbines (HAWTs). They can be packed closer together in wind farms, allowing more in a given space. This is not because they are smaller, but rather due to the slowing effect on the air that HAWTs have, forcing designers to separate them by ten times their width.

VAWTs are rugged, quiet, Omni-directional, and they do not create as much stress on the support structure. They do not require as much wind to generate power, thus allowing them to be closer to the ground. By being closer to the ground they are easily maintained and can be installed on chimneys and similar tall structures.

- The advantages of the VAWT configuration are that the generator and gearbox can be housed on the ground, and even some distance away from that turbine; and that a VAWT is unidirectional and requires no yaw mechanism.
- Relatively cost of production, installation and transport compared to horizontal axis turbines are less.
- The turbine doesn’t need to be pointed into the wind to be effective. This is an advantage on sites where the wind direction is highly variable. Hilltops, ridgelines and passes can have higher and more powerful winds near the ground than higher up because due to the speed up effect of winds moving up a slope. In these places, vertical axis turbines are suitable.
- The blades spin at slower speeds than the horizontal turbines, decreasing the risk of injuring birds.
- It is significantly quieter than the horizontal axis wind turbine. As a result, vertical axis wind turbines work well on rooftops, making them particularly useful in residential and urban environments.
- They may also be built in locations where taller structures are prohibited by law.
- They are particularly suitable for areas with extreme weather conditions, like in the mountains where they can supply electricity to mountain huts.

- 

Fig-1-3-1Examples of a VAWT

Fig-1-3-1Examples of a HAWT

Disadvantages of vertical axis wind turbines (VAWT):

The disadvantages of typical VAWT systems are that they usually operate near the ground where there's not much wind; they produce wavy (sinusoidal) power pulses to drive mechanism; they don't start themselves in a breeze; and repair of the main bearing usually means having to take the whole machine apart.

The blades of a VAWT are prone to fatigue as the blade spins around the central axis. The vertically oriented blades used in early models twisted and bent as they rotated in the wind. This caused the blades to flex and crack. Over time the blades broke apart and sometimes leading to catastrophic failure. Because of this problem, Vertical axis wind turbines have proven less reliable than horizontal-axis wind turbines (HAWTs).

Research programmers have sought to overcome the inefficiencies associated with VAWTs by reconfiguration of turbine placement within wind farms. It is thought that, despite the lower wind-speed environment at low elevations, "the scaling of the physical forces involved predicts that [VAWT] wind farms can be built using less expensive materials, manufacturing.(17)(18)(19)

1.4. Comparisons between H-rotor; Darrieus and HAWT three types of Wind energy

Fig-1-4-1 H-rotor

We wish to determine our research direction via the contrast and comparison of the respective characteristics of three types of Wind turbines.

Fig-1-4-1 H-rotor

Fig-1-4-1 H-rotor

Fig-1-4-2 Darrieus

Fig-1-4-3 HAWT

Table1.4.1 The differences between H-rotor 、Darrieus and HAWT (Eriksson et al.， 2008)

	H-rotor	Darrieus	HAWT
Blade profile	Simple	Complicated	Complicated
Yaw mechanism needed	No	No	Yes
Pitch mechanism possible	Yes	No	Yes
Tower	Yes	No	Yes
Guy wires	Optional	Yes	High
Noise	Low	Moderate	
Blade area	Moderate	Large	Small
Generator position	On ground	On ground	On top of tower
Blade load	Moderate	Low	High
Self starting	No	No	Yes
Tower interference	Small	Small	Large
Foundation	Moderate	Simple	Extensive
Overall structure	Simple	Simple	Complicated

Through the analysis of the three kinds of rotors, we found that the H—rotor had the simplest structure, was the most adaptable, the easiest to use and most suitable for a small to medium VAWT. It is also the most suitable to use in New Zealand and thus I have made it the subject of my thesis.

The effect mechanical vibrations on mechanical systems
Vibration is the motion of a particle or body or system of connected bodies displaced form a position of equilibrium. Most vibrations are undesirable in machines and structures, because they produce increased stresses, energy losses, cause added wear, increase bearing loads, induce fatigue, and absorb energy from the system. Rotating machine parts need careful balancing in order to prevent damage from vibrations. Resonance also has a major impact on mechanical structure and mechanics. (20)

1.5. Thesis Outline

This thesis presents structure response under different stable conditions, observing wind turbines natural frequencies vibrations, analysing the reasons and providing solution, and explain the current state of wind turbines.

The fundamentals of Wind energy and mechanical vibrations theory will be briefly discussed in Chapter 2

In chapter 3, the Solidworks tool is used to design the structure of the Wind turbine motor. The finite element method is employed to examine and characterize the frequencies and the mode shapes of the Wind turbine structure.

In chapter 4, the measured data from simulation results is employed to investigate and analyse through Fourier expansion methods. The deformation of protrusion of the wind turbine in different natural frequencies and mode shapes is shown to find ways of improvements through the tests.

Chapter 5 summarizes the result of the structural change of the wind turbine discusses paths for future research.

Chapter 2

Fundamental of Wind Energy and Mechanical Vibrations theory

2.1. VAWT General aerodynamics

The forces and the velocities acting in a Darrieus turbine are depicted in figure 1. The resultant velocity vector, $\vec{W}$, is the Victorian sum of the undisturbed upstream air velocity, $\vec{U}$, and the velocity vector of the advancing blade, .$-\vec{\omega} \times \vec{R}$

$$\vec{W} = \vec{U} + \left(-\vec{\omega} \times \vec{R}\right) \tag{1}$$

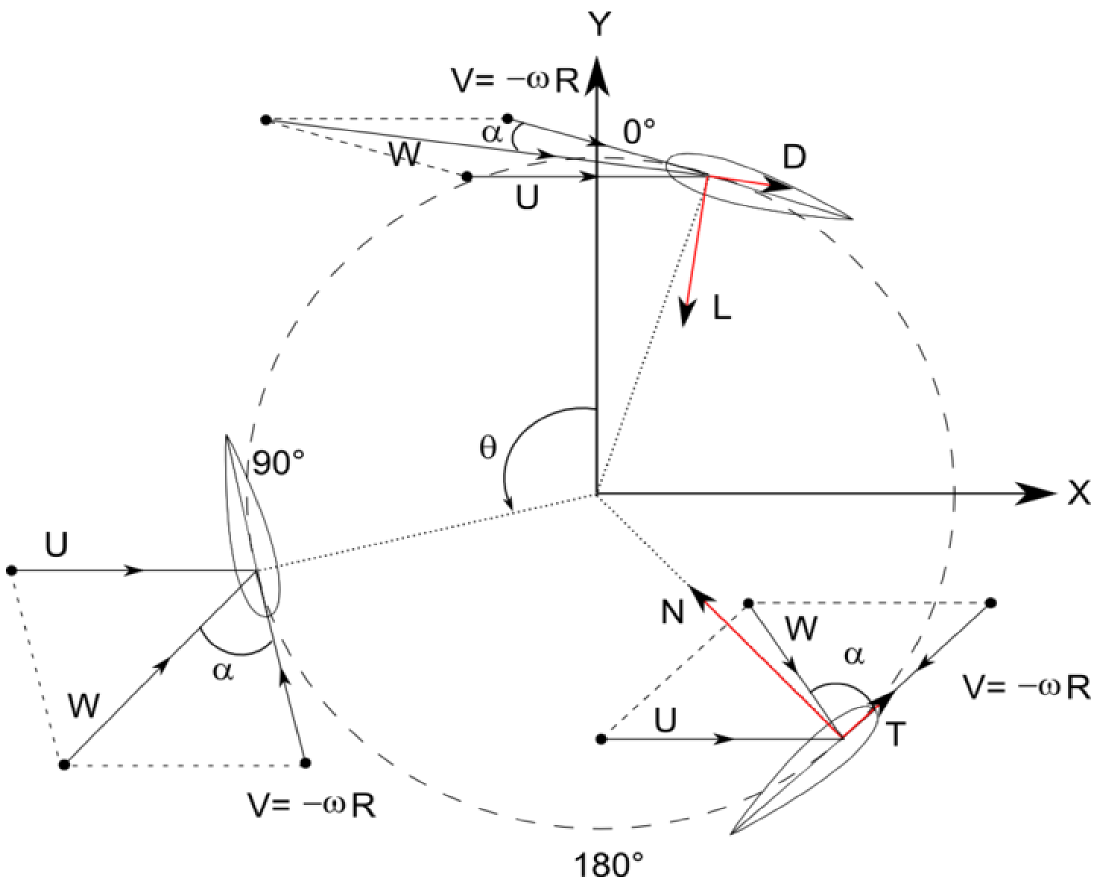


Fig 2-2-1: Forces and velocities acting in a Darrieus turbine for various azimuthal positions

Fig-2-1-2 vertical axis wind turbine

Thus, the oncoming fluid velocity varies, the maximum is found for $\theta = 0°$ and the minimum is found for $\theta = 180°$, where θ is the azimuthally or orbital blade position. The angle of attack, α, is the angle between the oncoming air speed, W, and the blade's chord. The resultant airflow creates a varying, positive angle of attack to the blade in the upstream zone of the machine, switching sign in the downstream zone of the machine.
From geometrical considerations, the resultant airspeed flow and the angle of attack are calculated as follows:

$$W = U\sqrt{1 + 2\alpha\cos\theta + \alpha^2} \tag{2}$$

$$\partial = \tan^{-1}\left(\frac{\sin\theta}{\cos\theta + a}\right) \tag{3}$$

where $\alpha = \frac{\omega R}{U}$ is t he tip speed ratio parameter.

The resultant aerodynamic force is decomposed either in lift (F_L) - drag (D) components or normal (N) - tangential (T) components. The forces are considered acting at 1/4 chord from the leading edge (by convention), the pitching moment is determined to resolve the aerodynamic forces. The aeronautical terms lift and drag are, strictly speaking, forces across and along the approaching net relative airflow respectively. The tangential force is acting along the blade's velocity and, thus, pulling the blade around and the normal force is acting radically, and, thus, is acting against the bearings. The lift and the drag force are useful when dealing with the aerodynamic behavior around each blade, i.e. dynamic stall, boundary layer, etc.; while when dealing with global performance, fatigue loads, etc., it is more convenient to have a normal-tangential frame. The lift and the drag coefficients are usually normalized by the dynamic pressure of the relative airflow, while the normal and the tangential coefficients are usually normalized by the dynamic pressure of undisturbed upstream fluid velocity.

$$C_L = \frac{F_L}{1/2\rho AW^2} \tag{4}$$

;

$$C_D = \frac{D}{1/2\rho AW^2} \ ; \tag{5}$$

$$C_T = \frac{T}{1/2\rho AU^2} \ ; \tag{6}$$

$$C_N = \frac{N}{1/2\rho AU^2} \tag{7}$$

A = Surface Area
The amount of power, P, that can be absorbed by a wind turbine.

$$P = \frac{1}{2} C_P \rho A V^3 \tag{8}$$

Where C_p is the power coefficient, ρ is the density of the air, A is the swept area of the turbine, and ν is the wind speed.(21)(22)

2.2. Fundamental of vibrations

Vibration occurs when a system is displaced from position of stable equilibrium. The system tends to return to this equilibrium position under the action of restoring forces. The system keeps moving back and forth across its position of equilibrium .A system is a combination of elements intended to act together to accomplish an objective. A static elements is one whose output at any given time depends only on the input at the time while a dynamic element is one whose present output depends on past input .A static system contains all elements while a dynamic system contains at least one dynamic element。(20)

2.2.1 *What causes the system to vibrate: from conservation of energy point of view*

Vibration motion could be understood in terms of conservation of energy. In the above example we have extended the spring by a value of and therefore have stored some potential energy yin the spring. Once we let go of the spring, the spring tries to return to its un-stretched state (which is the minimum potential energy state) and in the process accelerates the mass. At the point where the spring has reached its un-stretched state all the potential energy that we supplied by stretching it has been transformed into kinetic energy. The mass then begins to decelerate because it is now compressing the spring and in the process transferring the kinetic energy back to its potential. Thus oscillation of the spring amounts to the transferring back and forth of the kinetic energy into potential energy.

In our simple model the mass will continue to oscillate forever at the same magnitude, but in a real system there is always something called damping that dissipates the energy, eventually bringing it to rest.

2.2.2 Types of vibration**:**

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point. The oscillations may be periodic such as the motion of a pendulum or random such as the movement of a tire on a gravel road.

More often, vibration is undesirable, wasting energy and creating unwanted sound – noise. For example, the vibration motions of engines, electric motors, or any mechanical device in operation are typically unwanted. Such vibrations can be caused by imbalances in the rotating parts, uneven friction, the meshing of gear teeth, etc. Careful designs usually minimize unwanted vibrations.

The fundamentals of vibration analysis can be understood by studying the simple mass–spring–damper model. Indeed, even a complex structure such as an automobile body can be modeled as a "summation" of simple mass–spring–damper models. The mass–spring–damper model is an example of a simple harmonic oscillator. The mathematics used to describe its behavior is identical to other simple harmonic oscillators such as the RLC circuit.

2.2.3 Theory of free vibration:

Free vibration occurs when a mechanical system is set off with an initial input and then allowed to vibrate freely. Examples of this type of vibration are pulling a child back on a swing and then letting go or hitting a tuning fork and letting it ring. The mechanical system will then vibrate at one or more of its "natural frequency" and damp down to zero. Forced vibration is when an alternating force or motion is applied to a mechanical system. Examples of this type of vibration include a shaking washing machine due to an imbalance, transportation vibration (caused by truck engine, springs, road, etc.), or the vibration of a building during an earthquake. In forced vibration the frequency of the vibration is the frequency of the force or motion applied, with order of magnitude being dependent on the actual mechanical system.

When placed into motion, oscillation will take place at the natural frequency fn which is a property of the system. We now examine some of the basic concepts associated with the free vibration of systems with one degree of freedom.

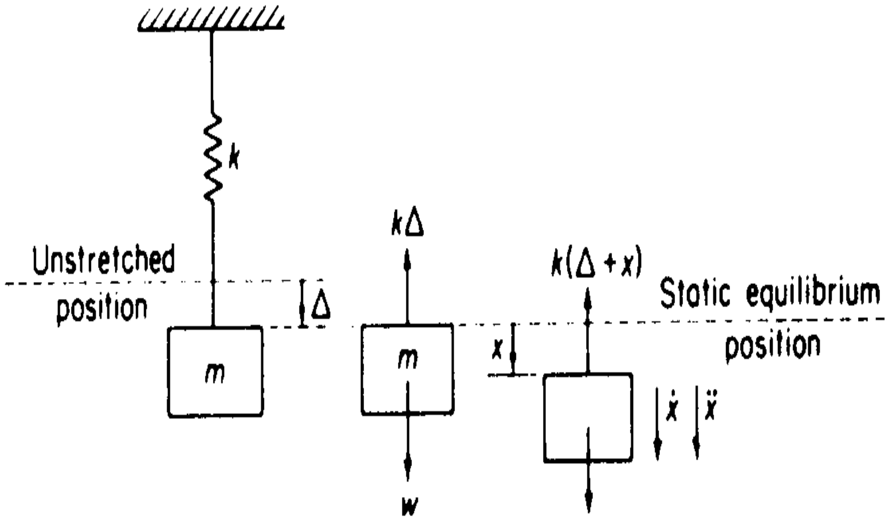


Fig-2-2-1 Spring-Mass System and Free-Body Diagram

Newton's second law is the first basis for examining the motion of the system. As shown in Fig. 2 the deformation of the spring in the static equilibrium position is D , and the spring force *kD* is equal to the gravitational force *w* acting on mass m

$$K\Delta = W = mg \tag{9}$$

By measuring the displacement *x* from the static equilibrium position, the forces acting on *m* are $k(\Delta+x)$ and *w*. With *x* chosen to be positive in the downward direction, all quantities - force, velocity, and acceleration are also positive in the downward direction.

We now apply Newton's second law of motion to the mass *m* :

$$mx = \sum F = w - k(\Delta + x) \tag{10}$$

and because *kD* = *w*, we obtain :

$$mx = -kx \tag{11}$$

It is evident that the choice of the static equilibrium position as reference for *x* has eliminated *w*, the force due to gravity, and the static spring force *kD* from the equation of motion, and the resultant force on *m* is simply the spring force due to the displacement *x.*

By defining the circular frequency w *n* by the equation

$$\omega_n^2 = \frac{k}{m} \tag{12}$$

Eq. 6 can be written as

$$x + \omega_n^2 x = 0 \tag{13}$$

and we conclude that the motion is harmonic. Equation (8), a homogeneous second order linear differential equation, has the following general solution :

$$x = A \sin \omega_n t + B \cos \omega_n t \tag{14}$$

where A and *B* are the two necessary constants. These constants are evaluated from initial conditions , $x(0) and x(0)$,and Eq. (9) can be shown to reduce to

$$x = \frac{x(0)}{\omega_n} \sin \omega_n t + x(0) \cos \omega_n t \tag{15}$$

The natural period of the oscillation is established from , $e_n \tau = 2\pi$ or

$$\gamma = 2\pi \sqrt{\frac{m}{k}} \tag{16}$$

and the natural frequency is

$$f_n = \frac{1}{\tau} = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \tag{17}$$

These quantities can be expressed in terms of the static deflection D by observing Eq. (5), $k\Delta = mg$ $k\Delta = mg$. Thus, Eq. (12) can be expressed in terms of the static deflection D as

$$f_n = \frac{1}{2\pi} \sqrt{\frac{g}{\Delta}} \tag{18}$$

Note that ,$\tau, f_n and\ \omega_n$, depend only on the mass and stiffness of the system, which are properties of the system.

2.2.4 Free vibration with damping

Mass Spring Damper Model

We now add a "viscous" damper to the model that outputs a force that is proportional to the velocity of the mass. The damping is called viscous because it models the effects of an object within a fluid. The proportionality constant *c* is called the damping coefficient and has velocity (lbf s/ in or N s/m).

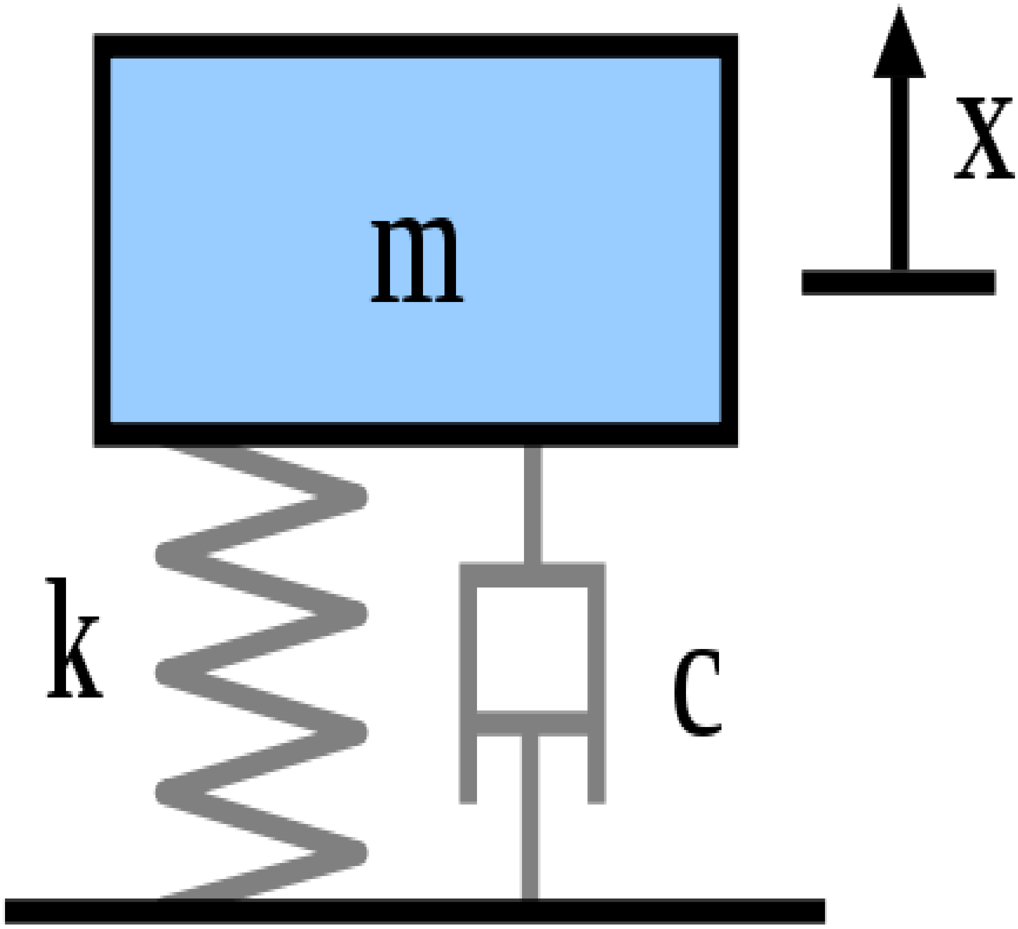


Fig-2-2-2

$$F_d = -cv = -c\dot{x} = -c\frac{dx}{dt} \qquad (19)$$

By summing the forces on the mass we get the following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = 0 \quad (20)$$

The solution to this equation depends on the amount of damping. If the damping is small enough the system will still vibrate, but eventually, over time, will stop vibrating. This case is called under damping – this case is of most interest in vibration analysis. If we increase the damping just to the point where the system no longer oscillates we reach the point of critical damping (if the damping is increased past critical damping the system is called over damped). The value that the damping coefficient needs to reach for critical damping in the mass spring damper model is:

$$C_c = 2\sqrt{km} \quad (21)$$

To characterize the amount of damping in a system a ratio called the damping ratio (also known as damping factor and % critical damping) is used. This damping ratio is just a ratio of the actual damping over the amount of damping required to reach critical damping. The formula for the damping ratio (ς), of the mass spring damper model is:

$$\varsigma = \frac{c}{2\sqrt{km}} \quad (22)$$

For example, metal structures (e.g. airplane fuselage, engine crankshaft) will have damping factors less than 0.05 while automotive suspensions in the range of 0.2–0.3.

The solution to the under damped system for the mass spring damper model is the following:

$$x(t) = X_e^{-\varsigma\omega_n t}\cos(\sqrt{1-\varsigma^2}\,\omega_n t - \emptyset), \quad \omega_n = 2\pi f_n \quad (23)$$

The value of *X*, the initial magnitude, and ϕ, the phase shift, are determined by the amount the spring is stretched. The formulas for these values can be found in the references.

2.2.5 Damped and undammed natural frequencies**:**

The major points to note from the solution are the exponential term and the cosine function. The exponential term defines how quickly the system “damps” down – the larger the damping ratio, the quicker it damps to zero. The cosine function is the oscillating portion of the solution, but the frequency of the oscillations is different from the undammed case.

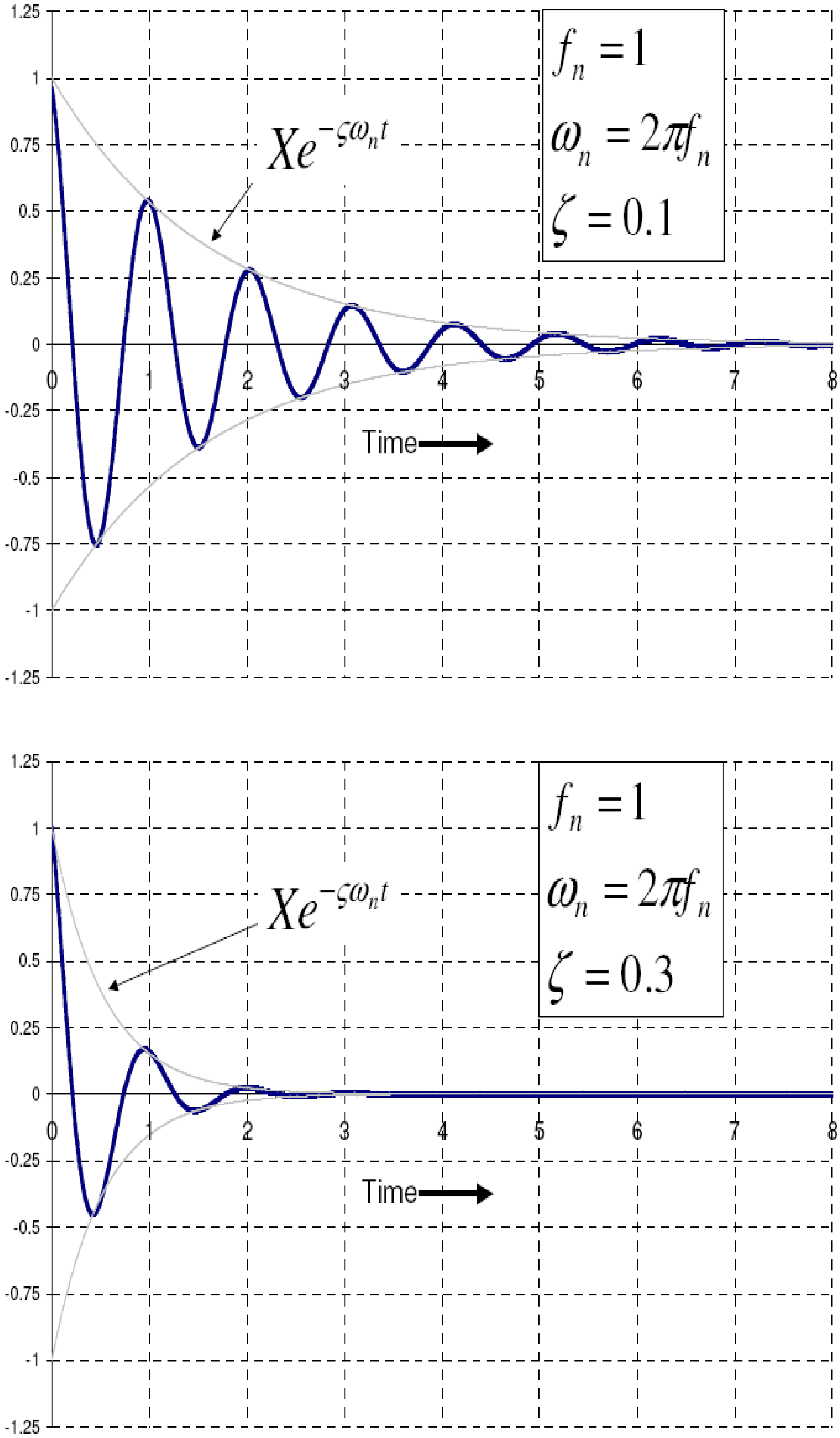

$f_n = 1$
$\omega_n = 2\pi f_n$
$\zeta = 0.1$
$Xe^{-\varsigma \omega_n t}$
Time
$f_n = 1$
$\omega_n = 2\pi f_n$
$\zeta = 0.3$
$Xe^{-\varsigma \omega_n t}$
Time

Fig-2-2-3

The frequency in this case is called the "damped natural frequency", f_d, and is related to the undammed natural frequency by the following formula:

$$f_d = f_n\sqrt{1 - \varsigma^2} \tag{24}$$

The damped natural frequency is less than the undammed natural frequency, but for many practical cases the damping ratio is relatively small and hence the difference is negligible. Therefore the damped and undammed description are often dropped when stating the natural frequency (e.g. with 0.1 damping ratio, the damped natural frequency is only 1% less than the undammed).

The plots to the side present how 0.1 and 0.3 damping ratios effect how the system will “ring” down over time. What is often done in practice is to experimentally measure the free vibration after an impact (for example by a hammer) and then determine the natural frequency of the system by measuring the rate of oscillation as well as the damping ratio by measuring the rate of decay. The natural frequency and damping ratio are not only important in free vibration, but also characterize how a system will behave under forced vibration.

2.2.6 Forced vibration with damping:

In this section we will see the behavior of the spring mass damper model when we add a harmonic force in the form below. A force of this type could, for example, be generated by a rotating imbalance.

$$F = F_0 \cos(2\pi f t) \ . \tag{25}$$

If we again sum the forces on the mass we get the following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(2\pi f t) \tag{26}$$

The steady state solution of this problem can be written as:

$$x(t) = X \cos(2\pi f t - \emptyset). \tag{27}$$

The result states that the mass will oscillate at the same frequency, f, of the applied force, but with a phase shift ϕ.

The amplitude of the vibration “X” is defined by the following formula.

$$X = \frac{F_0}{k}\frac{1}{\sqrt{(1+r^2)^2+(2\varsigma r)^2}} \tag{28}$$

Where "r" is defined as the ratio of the harmonic force frequency over the undammed natural frequency of the mass–spring–damper model.

$$r = \frac{f}{f_n} \tag{29}$$

The phase shift, ϕ, is defined by the following formula.

$$\phi = arc\tan\left(\frac{2\varsigma r}{1-r^2}\right) \tag{30}$$

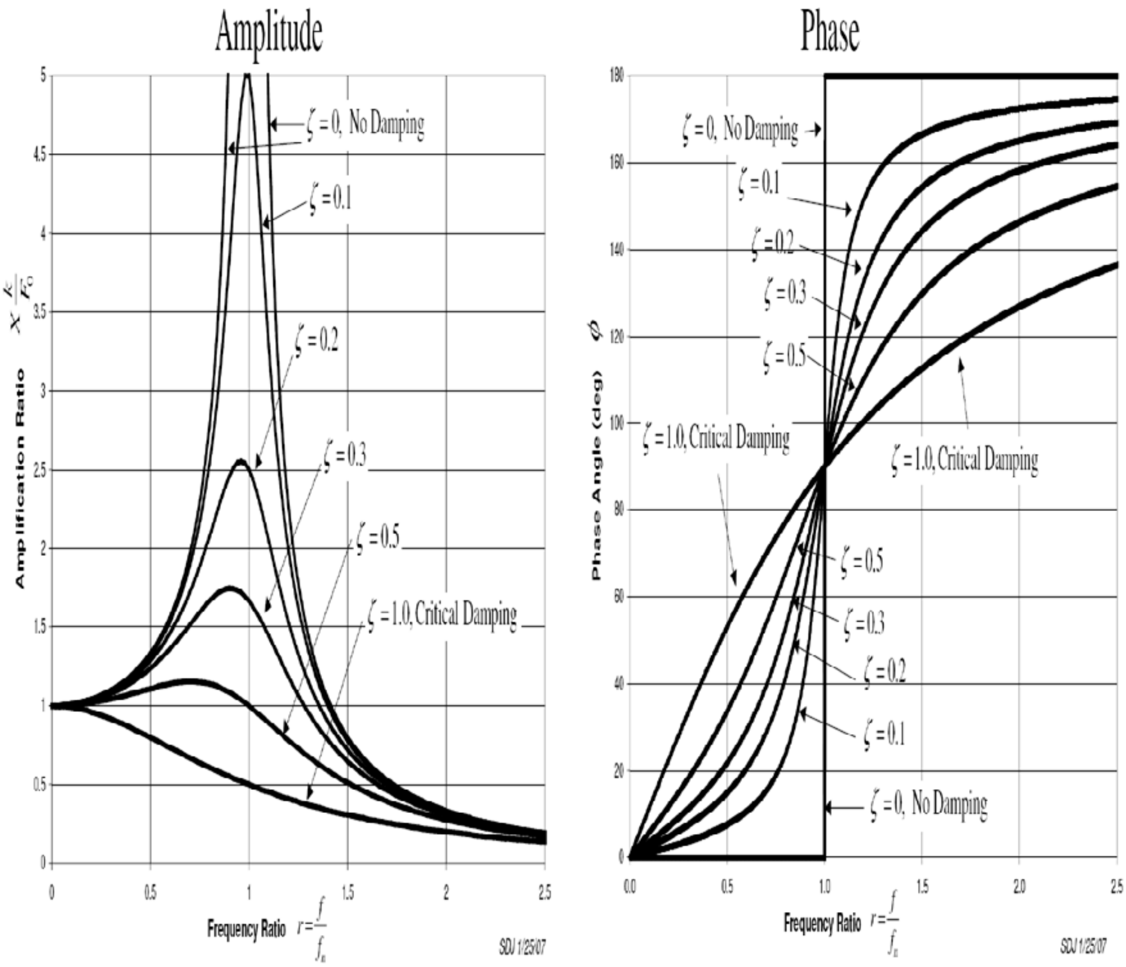


Fig-2-2-4

The plot of these functions, called "the frequency response of the system", presents one of the most important features in forced vibration. In a lightly damped system when the forcing frequency nears the natural frequency ($r \approx 1$) the amplitude of the vibration can get extremely high. This phenomenon is called resonance (subsequently the natural frequency of a system is often referred to as the resonant frequency). In rotor bearing systems any rotational speed that excites a resonant frequency is referred to as a critical speed.

If resonance occurs in a mechanical system it can be very harmful – leading to eventual failure of the system. Consequently, one of the major reasons for vibration analysis is to predict when this type of resonance may occur and then to determine what steps to take to prevent it from occurring. As the amplitude plot shows, adding damping can significantly reduce the magnitude of the vibration. Also, the magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted (for example, changing the speed of the machine generating the force).

The following are some other points in regards to the forced vibration shown in the frequency response plots.

At a given frequency ratio, the amplitude of the vibration, *X*, is directly proportional to the amplitude of the force F_0(e.g. if you double the force, the vibration doubles)

With little or no damping, the vibration is in phase with the forcing frequency when the frequency ratio $r < 1$ and 180 degrees out of phase when the frequency ratio $r > 1$

When $r \ll 1$ the amplitude is just the deflection of the spring under the static force F_0 This deflection is called the static deflection δ_{st}. Hence, when $r \ll 1$ the effects of the damper and the mass are minimal.

When $r \gg 1$ the amplitude of the vibration is actually less than the static deflection δ_{st}. In this region the force generated by the mass ($F = ma$) is dominating because the acceleration seen by the mass increases with the frequency. Since the deflection seen in the spring, *X*, is reduced in this region, the force transmitted by the spring ($F = kx$) to the base is reduced. Therefore the mass–spring–damper system is isolating the harmonic force from the mounting base – referred to as vibration isolation. Interestingly, more damping actually reduces the effects of vibration isolation when $r \gg 1$ because the damping force ($F = cv$) is also transmitted to the base. (23)(24)

2.2.7 What causes resonance?

A resonance occurs when a structure or material naturally oscillates at a high amplitude at a specific frequency. This frequency is known as a structural resonant frequency. Typically a structure will have many resonant frequencies.

A dictionary definition of resonance gives us -

“the state of a system in which an abnormally large vibration is produced in response to an external stimulus, occurring when the frequency of the stimulus is the same, or nearly the same, as the natural vibration frequency of the system.

When the damping in a structure is small, the resonant frequencies are approximately equal to the natural frequencies of the structure, which are the frequencies of free vibrations of the molecules of the material itself.

The damper, instead of storing energy, dissipates energy. Since the damping force is proportional to the velocity, the more the motion, the more the damper dissipates the energy. Therefore a point will come when the energy dissipated by the damper will equal the energy being fed in by the force. At this point, the system has reached its maximum amplitude and will continue to vibrate at this level as long as the force applied stays the same. If no damping exists, there is nothing to dissipate the energy and therefore theoretically the motion will continue to grow on into infinity.

Furthermore, an individual resonance is the condition when a natural frequency of a structure or material and the frequency at which it is being excited are equal or very nearly equal. This results in the structure or material vibrating strongly and is the classical resonance state. This resonance state can often lead to unexpected behavior of the structure or material.

Multiple degrees of freedom systems and mode shapes:

The simple mass–spring damper model is the foundation of vibration analysis, but what about more complex systems? The mass–spring–damper model described above is called a single degree freedom (SDOF) model since we have assumed the mass only moves up and down. In the case of more complex systems we need to discredited the system into more masses and allow them to move in more than one direction – adding degrees of freedom. The major concepts of multiple degrees of freedom (MDOF) can be understood matrix symmetric matrices.

Any complex system with multiple degrees of freedom can be analyzed using simple analytical techniques.(25)

2.3. Natural frequency and mode shapes

Natural frequency is the frequency at which a system naturally vibrates once it has been set into motion of free vibration of a system .for a multiple degree-of-freedom system; the natural frequencies are the frequencies of the normal modes of vibration.

Normal mode of vibration is a mode of vibration that is uncoupled from other modes of vibration of a system .When vibration of the system is defined as an Eigen value problem, the normal modes are the eigenvectors and the normal mode frequencies are the Eigen values .The term “classical normal mode ’’ is sometimes applied to the normal modes of a vibrating system characterized by vibration of each element of the system at the same frequency and phase .In general classical normal modes exist only in system having no damping or having particular types of damping .
The frequencies of the normal modes of a system are known as its natural frequencies or resonant frequencies normal mode of an oscillating system is a pattern of motion in which all parts of the system moves sinusoid ally with the same frequency and with a fixed phase relation. The motion described by the normal modes is called resonance. The frequencies of the normal modes of a system are known as its natural frequencies or resonant frequencies. A physical object, such as a building, bridge or molecule, has a set of normal modes that depend on its structure, materials and boundary conditions.(20)

Each mode is entirely independent of all other modes. Thus all modes have different frequencies (with lower modes having lower frequencies) and different mode shapes.

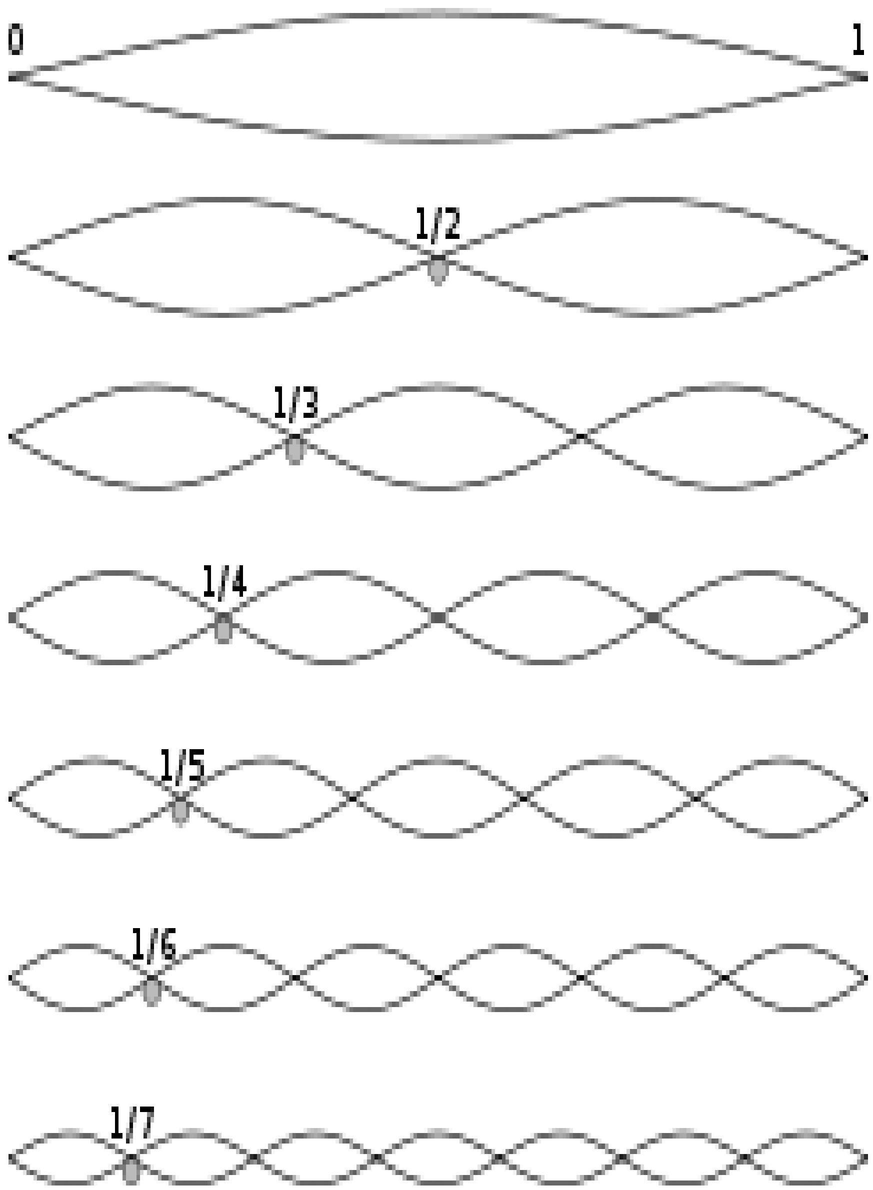


Fig-2-3 Different models and fixed systems have their own natural frequencies.

Chapter 3 The construction of normal H-rotor VAWT models

3.1. Chapter Overview

This chapter will introduce the Solidworks software，and explain the use of Solidworks modelling to draw VAWT and to create finite element webs.

A structure of the normal VAWT mode will be introduced and also the geometry parameter and material properties will be specified in this chapter. Furthermore, the VAWT MODE will be made by Solidworks examined through finite element simulations.

3.2.Introduction to Solidworks

SolidWorks is a 3D mechanical CAD (computer-aided design) program that runs on Microsoft Windows and is being developed by Dassault Systèmes SolidWorks Corp., a subsidiary of Dassault Systèmes, S. A. (Vélizy, France). SolidWorks is currently used by over 1.3 million engineers and designers at more than 130,000 companies worldwide. FY2009 revenue for SolidWorks was 366 million dollars.

SolidWorks is a Para solid-based solid modeller, and utilizes a parametric feature-based approach to create models and assemblies.

3.3 Use Solidworks to design a normal H-rotor VAWT model

3.3.1Use the website (http://www.ae.illinois.edu/m-selig/ads/afplots/naca2415.gif) to find NACA2414AIRFOIL, (Fig-3-3-1), which was imported into SOILDWORKS and design a NORMAL three blade H-ROTOR VAWT (Fig-3-3-2)

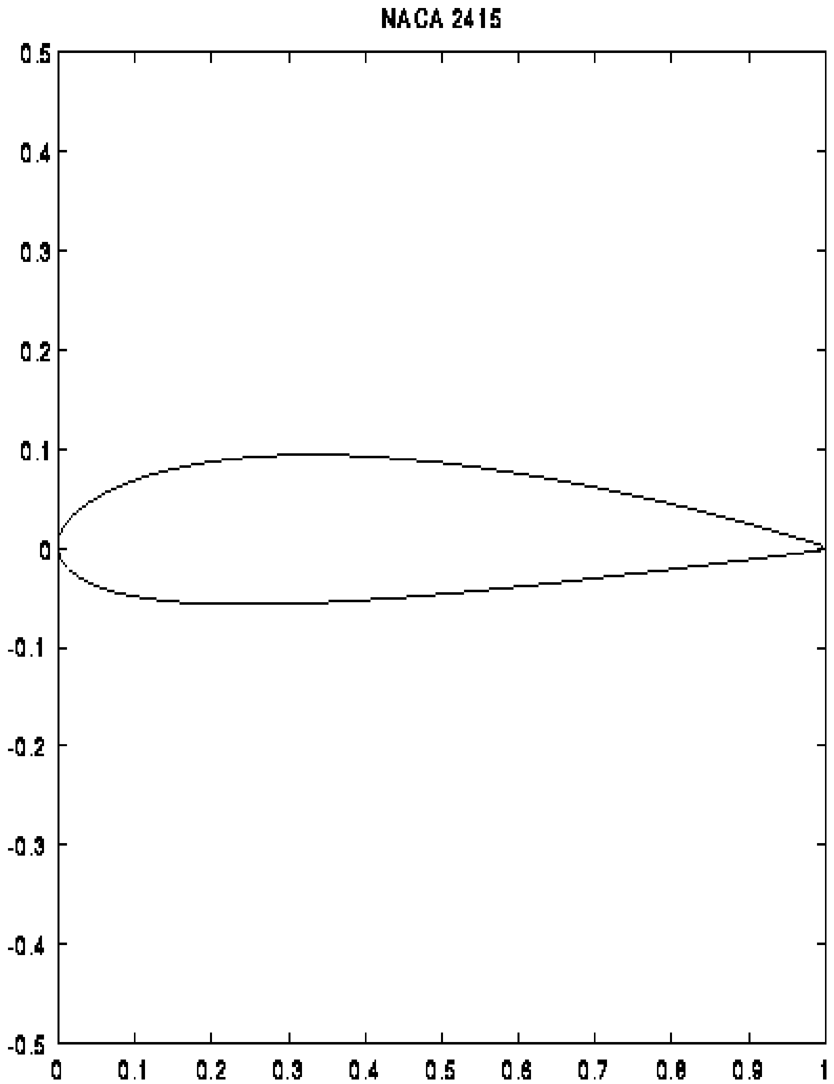


Figure- 3-3-1 NACA 2415

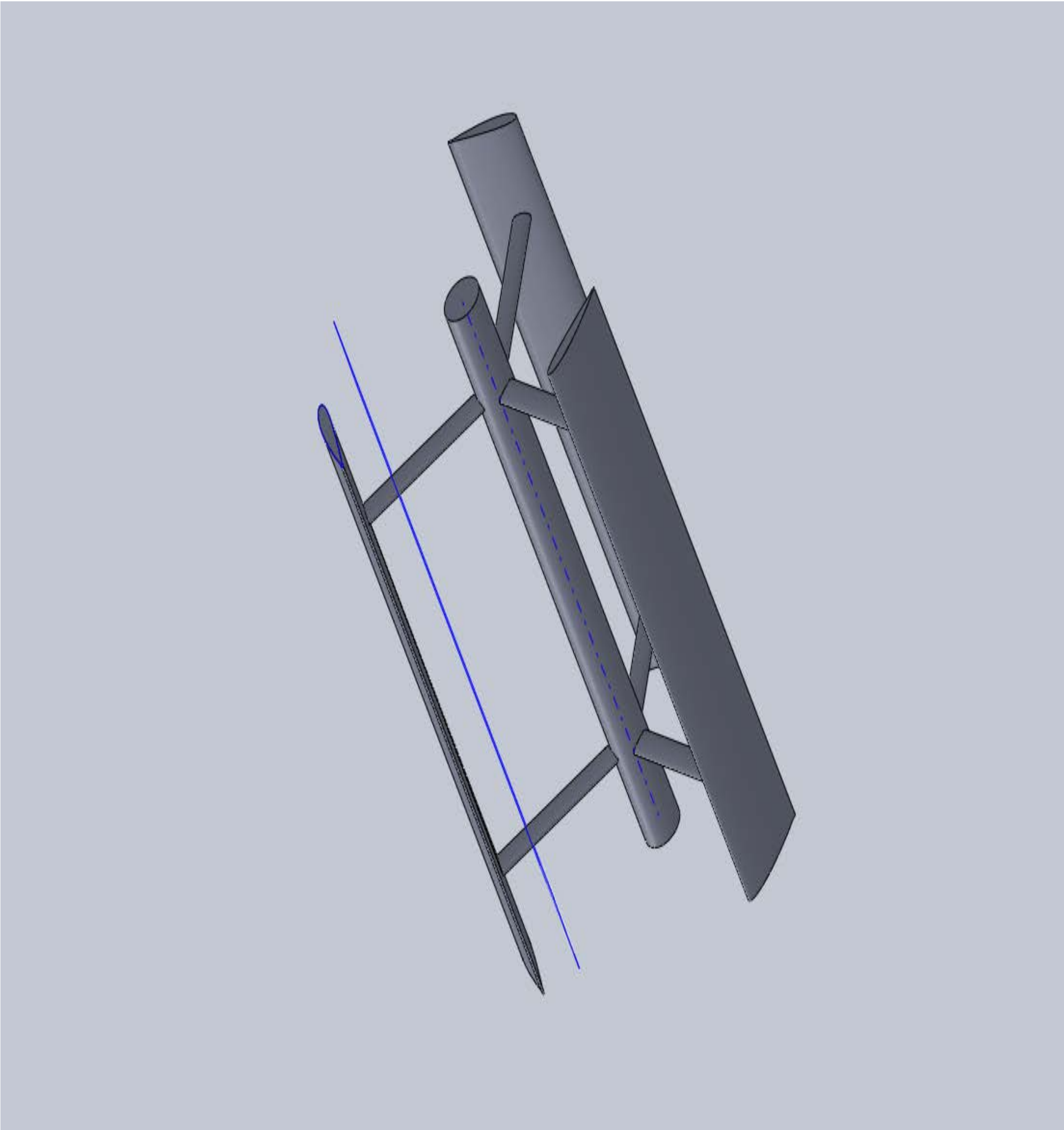


Fig-3-3-2 normal H-rotor VAWT

Size: Diameter of the centre shaft is 50mm, and the rod is 1000mm long. The connecting shafts are oval shaped and has a vertical diameter of 30mm with a horizontal diameter of 20mm. The blades of the turbine are 100mm wide and 1000mm long. The diameter of the whole turbine is 600mm. The turbine is made of Aluminium Alloy 1060-H18 ， the properties of that alloy are displayed below.（Fig-3-3-3）

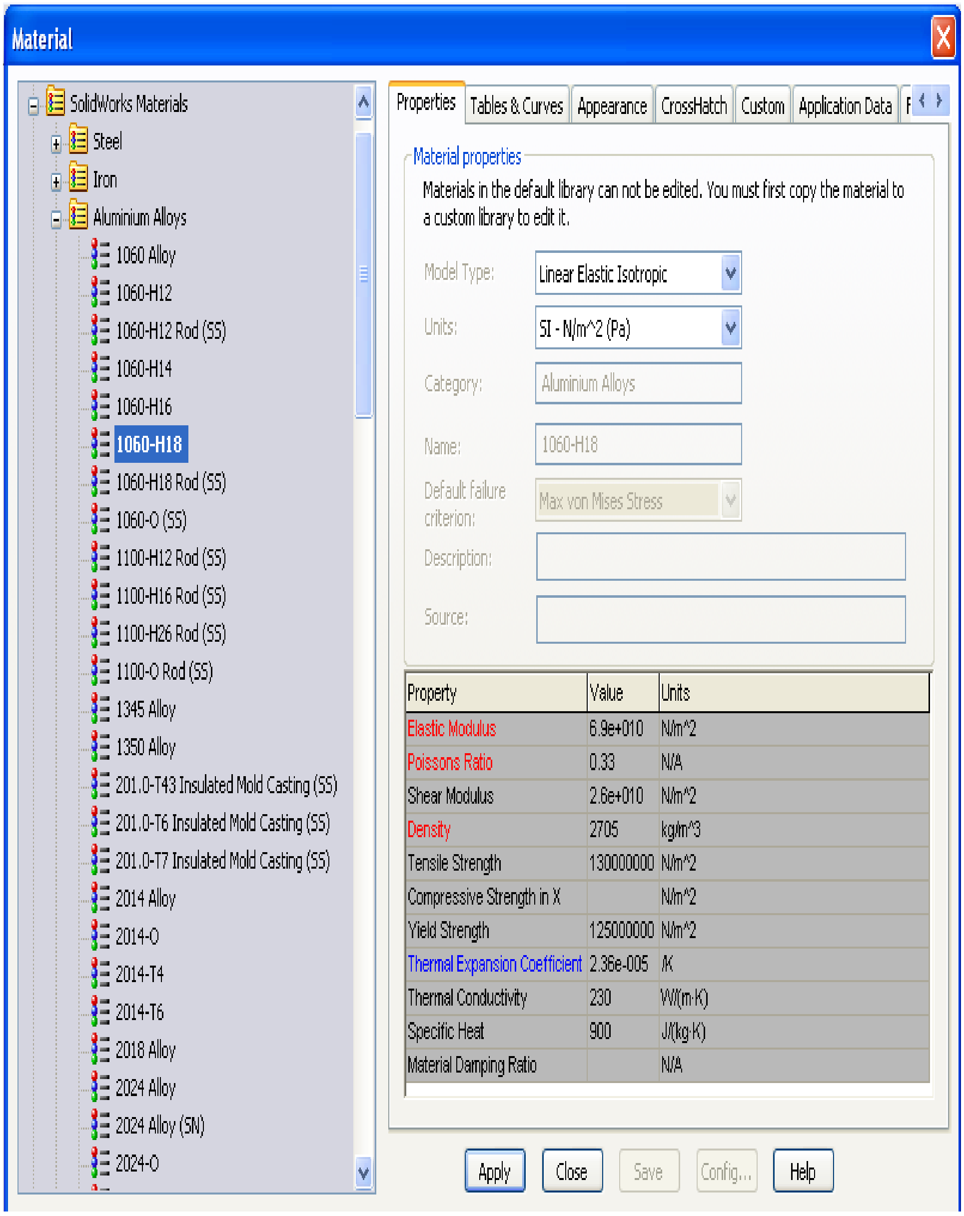


Fig-3-3-3

3.4. Finite Element Analysis

A finite element analysis is very important for designing a good VAWT.

Finite Element Analysis is based on the premise that an approximate solution to any complex engineering problem can be reached by subdividing the problem into smaller, more manageable (finite) elements. Using finite elements, complex partial differential equations that describe the behavior of structures can be reduced to a set of linear equations that can easily be solved using the standard techniques of matrix algebra.

Finite Element Analysis is one of several numerical methods that can be used to solve complex problems and is the dominant method used today. As the name implies, it takes a complex problem and breaks it down into a finite number of simple problems. A continuous structure theoretically has an infinite number of simple problems, but finite element analysis approximates the behavior of a continuous structure by analyzing a finite number of simple problems. Each element in a finite element analysis is one of these simple problems. Each element in a finite element model will have a fixed number of nodes that define the element boundaries to which loads and boundary conditions can be applied. The finer the mesh, the closer we can approximate the geometry of the structure, the load application, as well as the stress and strain gradients. However, there is a tradeoff: the finer the mesh, the more computational power is needed to solve the complex problem. The strategy of optimizing the mesh size can greatly reduce an analyst's time without compromising on the quality of analysis results.

Finite element analysis consists of a computer model of a material or design that is stressed and analyzed for specific results. It is used in new product design, and existing product refinement. In case of structural failure, FEA may be used to help determine the design modifications to meet the new condition.

There are generally two types of analysis that are used in industry: 2-D modeling, and 3-D modeling. While 2-D modeling conserves simplicity and allows the analysis to be run on a relatively normal computer, it tends to yield less accurate results. 3-D modeling, however, produces more accurate results while sacrificing the ability to run on all but the fastest computers effectively. Within each of these modeling schemes, the programmer can insert numerous algorithms (functions) which may make the system behave linearly or non-linearly. Linear systems are far less complex and generally do not take into account plastic deformation. Non-linear systems do account for plastic deformation, and many also are capable of testing a material all the way to fracture. (20)(26)(27)(28)(29)(30)

3.5 Using Solidworks modelling to design VAWT and simulate a finite element mesh

Structured Meshes

A structured mesh is characterized by regular connectivity that can be expressed as a two or three dimensional array. This restricts the element choices to quadrilaterals in 2D or hexahedra in 3D. The above example mesh is a structured mesh, as we could store the mesh connectivity in a 40 by 12 array. The regularity of the connectivity allows us to conserve space since neighborhood relationships are defined by the storage arrangement. Additional classification can be made upon whether the mesh is conformal or not.

Unstructured Meshes

An unstructured mesh is characterized by irregular connectivity is not readily expressed as a two or three dimensional array in computer memory. This allows for any possible element that a solver might be able to use. Compared to structured meshes, the storage requirements for an unstructured mesh can be substantially larger since the neighborhood connectivity must be explicitly stored.

Hybrid Meshes

A hybrid mesh is a mesh that contains structured portions and unstructured portions. Note that this definition requires knowledge of how the mesh is stored (and used). There is disagreement as to the correct application of the terms "hybrid" and "mixed." The term "mixed" is usually applied to meshes that contain elements associated with structured meshes and elements associated with unstructured meshes (presumably stored in an unstructured fashion).

The mesh that I have used in this modelling is a structural mesh.

By SolidWorks simulation mash the normal H-rotor VAWT model show

Fig-3-5-1

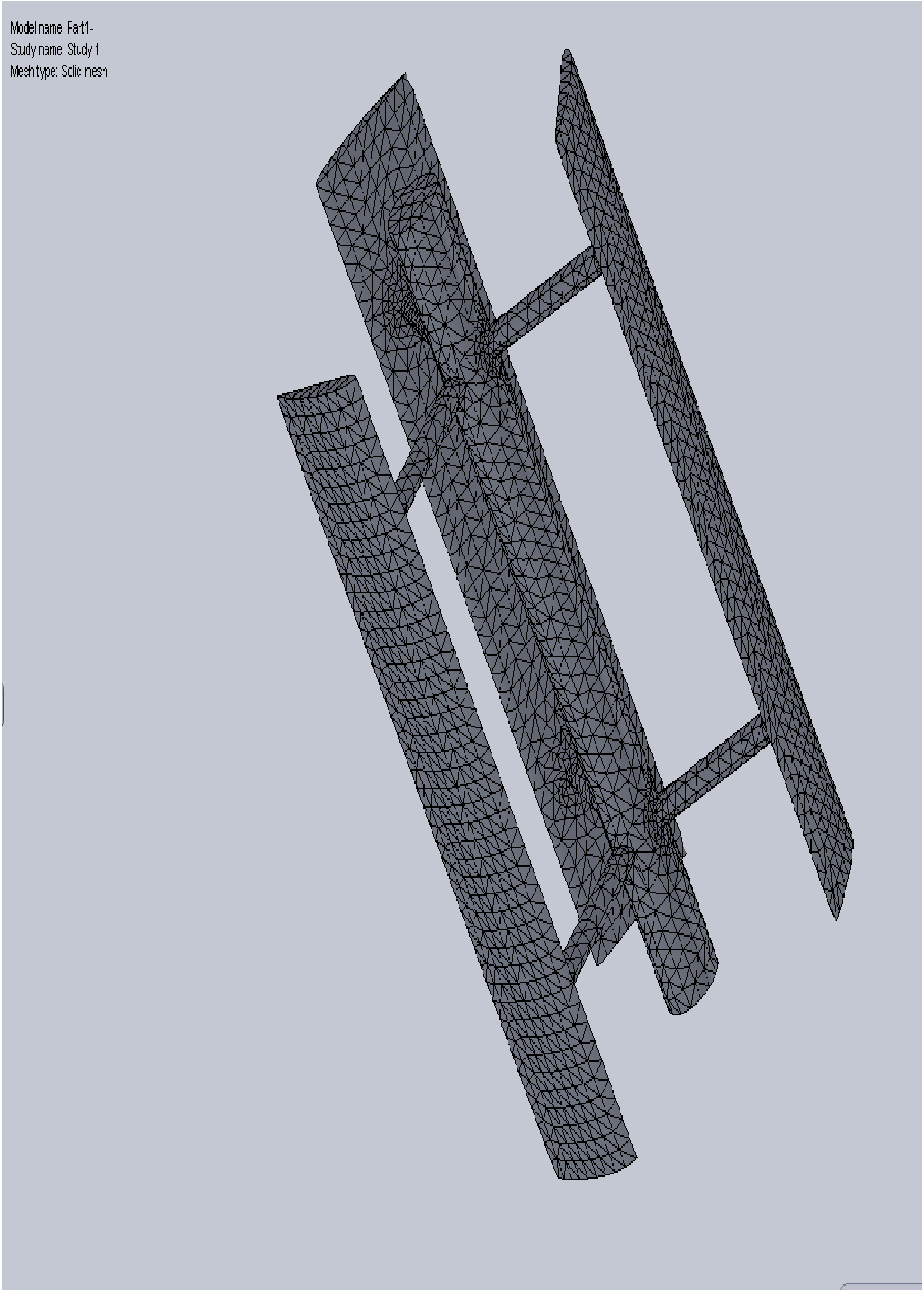


Fig-3-5-1 The normal H-rotor VAWT model mesh by Solidworks

3.6. Chapter summary

This chapter has presented a normal design of the VAWT, by using Solidworks and the finite elements model I have designed a normal H-rotor VAWT and paved the path for the analysis of the structural mesh of the VAWT. Therefore the Solidworks programme has played a large part in preparing results and data for analysis.

Chapter 4 Conclusion of simulation results

4.1. Chapter Overview

Through the use of Solidworks simulation I have tested the normal H-rotor VAWT model in the previous chapter at three fixed systems natural frequencies. I have analyzed their vibration and frequency, and found that there is the possibility of resonance, through the theory of the relationship between wind energy and natural frequency and shape, I have designed a new and improved VAWT, and through three types of fixed systems simulation at natural frequencies， I have observed and analysed the vibration and frequencies of the VAWT.

4.2. The simulation of the normal H-rotor VAWT model at three fixed system natural frequencies.

We used Solidworks to simulate a normal H-rotor wind turbine at three fixed system natural frequencies in order to observe the range; frequency and shape of the vibration to find a method for improvement.

4.2.1 Bearing fixtures normal H-rotor VAWT of fixed systems natural frequency simulation by Solidworks, We get simulation results shown (Fig-4-1) and a list of resonant frequencies and a list of mass participation

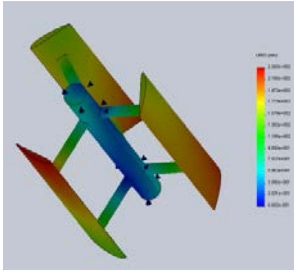
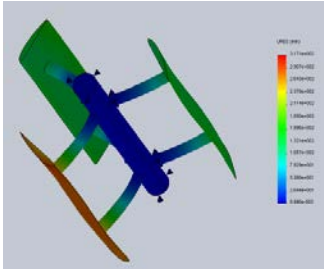
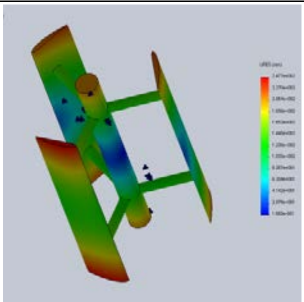
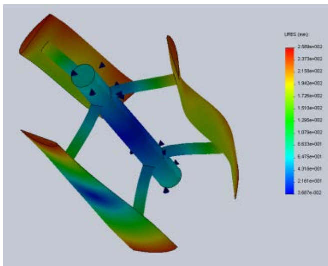
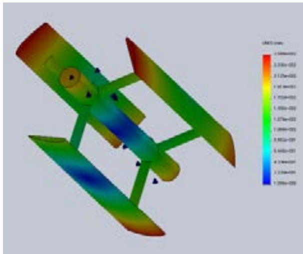
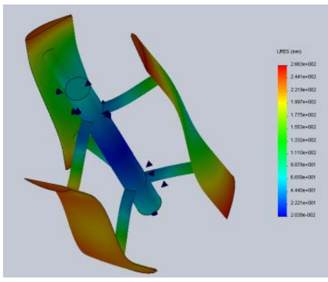
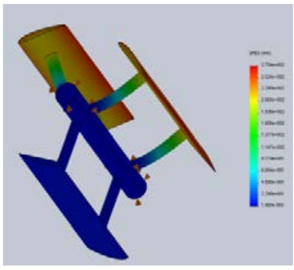
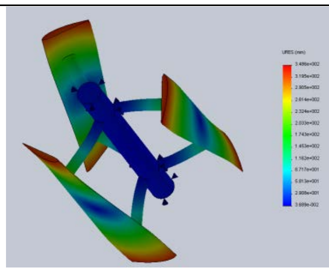
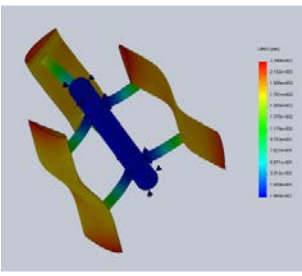
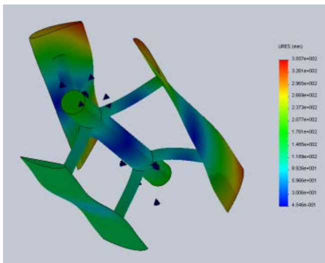
Number	Model shapes	Number	Model Shapes
1		6	
2		7	
3		8	
4		9	
5		10	

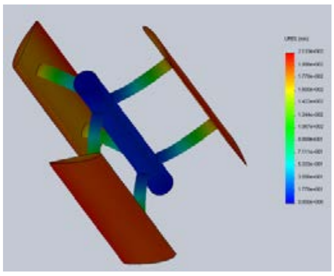
Fig-4-1 Bearing fixtures VAWT of natural frequency simulation by Solidworks

List of resonant frequencies			
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	0	0	1e+032
2	0.0099037	0.0015762	634.43
3	0.053061	0.008445	118.41
4	655.76	104.37	0.0095816
5	662.65	105.46	0.0094819
6	663.14	105.54	0.0094749
7	730.42	116.25	0.0086022
8	732.52	116.58	0.0085775
9	862.08	137.2	0.0072884
10	1076.1	171.26	0.005839

List of mass participation				
Mode No.	Freq (Hertz)	X direction	Y direction	Z direction
1	0	1.50E-12	1.45E-10	3.66E-12
2	0.001576	2.32E-15	1.43E-09	2.65E-11
3	0.008445	1.49E-09	7.32E-12	6.61E-11
4	104.37	1.75E-05	0.2554	2.88E-05
5	105.46	0.012256	2.11E-05	0.4939
6	105.54	0.24272	1.09E-05	0.024933
7	116.25	1.02E-10	3.52E-07	1.71E-07
8	116.58	3.70E-08	7.84E-07	0.000687
9	137.2	7.03E-08	1.06E-07	0.000457
10	171.26	9.01E-08	1.32E-08	7.09E-06
		Sum X = 0.25499	Sum Y = 0.25543	Sum Z = 0.52001

4.2.2 Two face fixed geometry VAWT of fixed systems natural frequency simulation by Solidworks. We get simulation results shown (Fig-4-2) and list resonant frequencies, list mass participation

1



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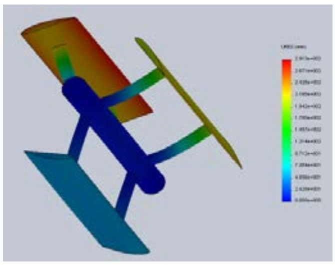

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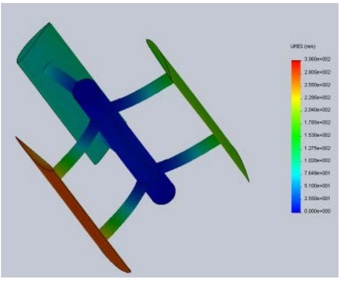

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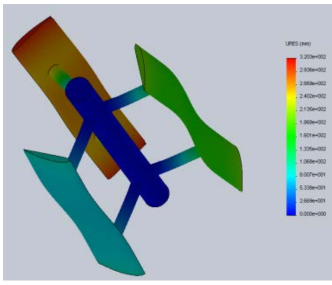

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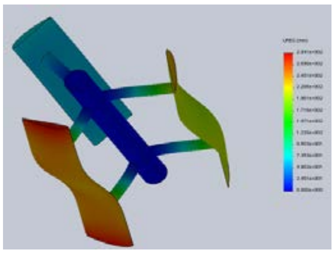

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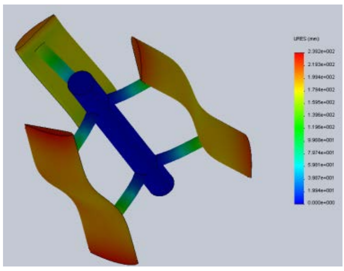

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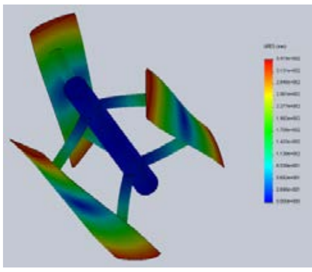

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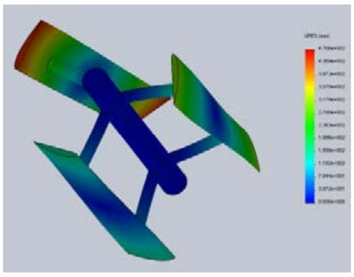

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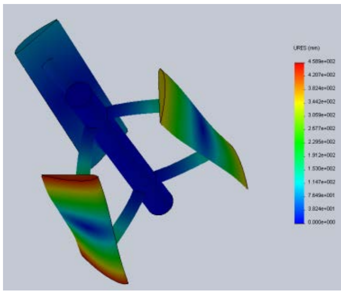

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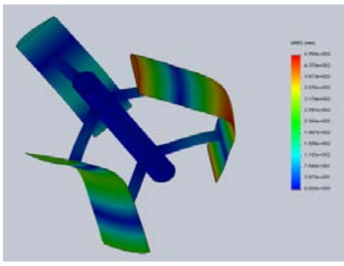


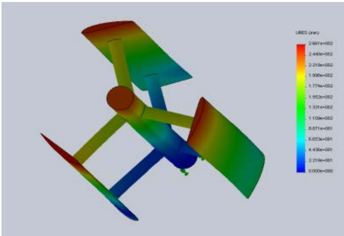
Fig-4-2 Two face fixed geometry VAWT de natural frequency simulation by Solidworks

List of resonant frequencies			
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	534.2	85.021	0.011762
2	623.06	99.163	0.010084
3	623.16	99.179	0.010083
4	638.97	101.7	0.009833
5	639.21	101.73	0.00983
6	642.41	102.24	0.009781
7	757.21	120.51	0.008298
8	821.51	130.75	0.007648
9	821.64	130.77	0.007647
10	1122.5	178.65	0.005598

List of mass participation				
Mode No.	Freq (Hertz)	X direction	Y direction	Z direction
1	85.021	1.52E-07	2.47E-08	2.40E-10
2	99.163	0.011271	0.27357	4.92E-08
3	99.179	0.27357	0.01127	7.83E-07
4	101.7	4.31E-08	2.29E-07	0.000842
5	101.73	3.85E-07	1.25E-08	0.000567
6	102.24	4.62E-07	1.06E-07	0.52265
7	120.51	5.69E-08	5.37E-09	0.001425
8	130.75	1.12E-08	2.20E-08	1.22E-08
9	130.77	2.76E-09	4.30E-09	9.85E-10
10	178.65	0.010236	0.00654	3.91E-10
		Sum X = 0.29508	Sum Y = 0.29138	Sum Z = 0.52549

4.2.3 One face fixed geometry VAWT of fixed systems natural frequency simulation by Solidworks. We get simulation results shown (Fig-4-3) and list resonant frequencies, list mass participation

1



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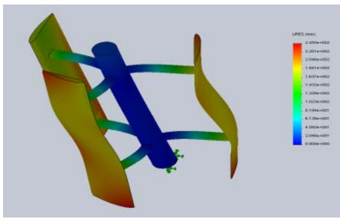

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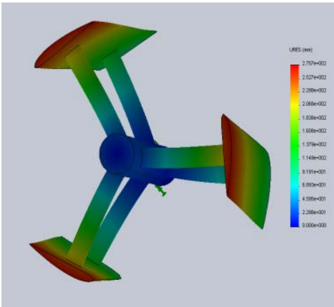

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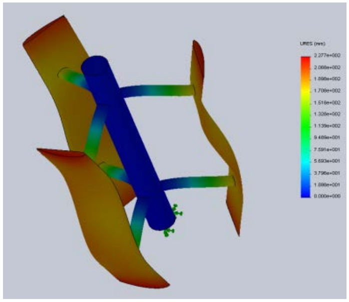

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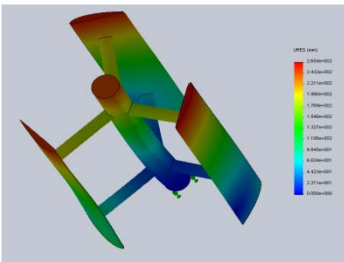

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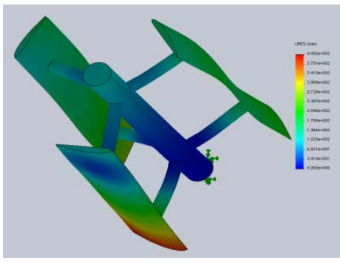

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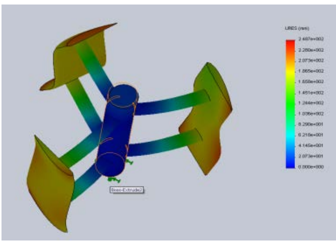

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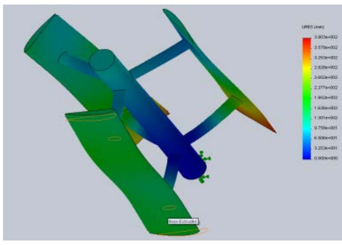

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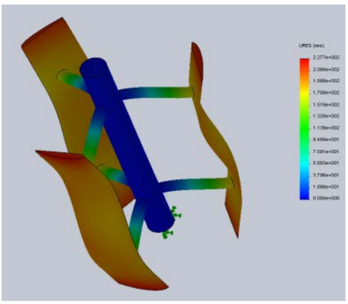

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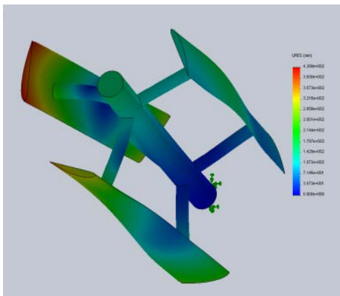


Fig-4-3 One face fixed geometry VAWT de natural frequency simulation by Solidworks

List of resonant frequencies			
Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	264.52	42.1	0.023753
2	264.52	42.1	0.023753
3	401.01	63.823	0.015668
4	629.02	100.11	0.009989
5	629.12	100.13	0.009987
6	640.63	101.96	0.009808
7	712.48	113.39	0.008819
8	712.52	113.4	0.008818
9	722.67	115.02	0.008694
10	919.69	146.37	0.006832

List of mass participation				
Mode No.	Freq (Hertz)	X direction	Y direction	Z direction
1	42.1	0.01479	0.52374	2.73E-09
2	42.1	0.52369	0.014802	1.57E-09
3	63.823	2.67E-09	5.08E-10	2.94E-05
4	100.11	0.0836	0.040731	3.33E-05
5	100.13	0.0407	0.083551	9.42E-06
6	101.96	1.90E-06	7.86E-06	0.52767
7	113.39	2.30E-05	0.00115	1.39E-06
8	113.4	0.001179	2.05E-05	1.12E-06
9	115.02	2.00E-07	1.63E-10	0.002716
10	146.37	0.04987	0.034338	1.06E-09
		Sum X = 0.71385	Sum Y = 0.69834	Sum Z = 0.53046

Through the analysis of the simulation of normal H-rotor VAWT model at fixed systems natural frequency we found:

1. Bearing fixtures normal H-rotor VAWT model of natural frequency simulation of natural frequency the range was 0---171.25Hz. The degree of change of the vibration was X axis towards 0.25499,Y axis towards 0.25543,Z axis towards 0.52001
2. For the two faced fixed geometry VAWT model of natural frequency simulation of natural frequency the range was 85.021----178.65Hz. The degree of change of the vibration was X axis towards 0.29508,Y axis towards 0.29138,Z axis towards 0.52549
3. For the one faced fixed geometry VAWT model of natural frequency simulation of natural frequency the range was 42.1----146.37Hz. The degree of change of the vibration was X axis towards 0.71385,Y axis towards 0.69834,Z axis towards 0.53046
4. Bearing fixtures has a lower vibration change compare to Two face fixed geometry or One face fixed geometry. The change of vibration One face fixed geometry was the largest.
5. The size of the central shaft had a large impact on the vibration of the VAWT.
6. The change in vibration of the central shaft was the largest，it is an important component to consider in the design of theVAWT.
7. When the vibration axis of the components is the same as the wind direction, there is resonance present. Show Fig-4-1 4,6,9,10. Fig-4-2 2,3,7,9. Fig-4-3 7,9,10.

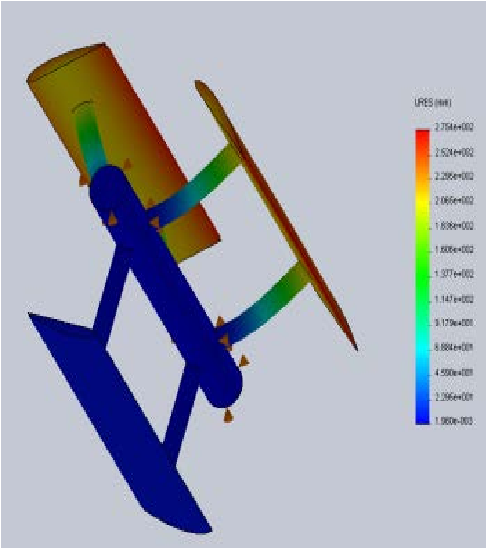
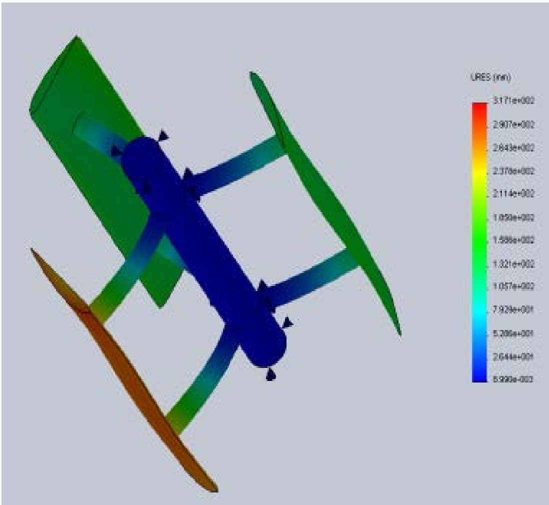
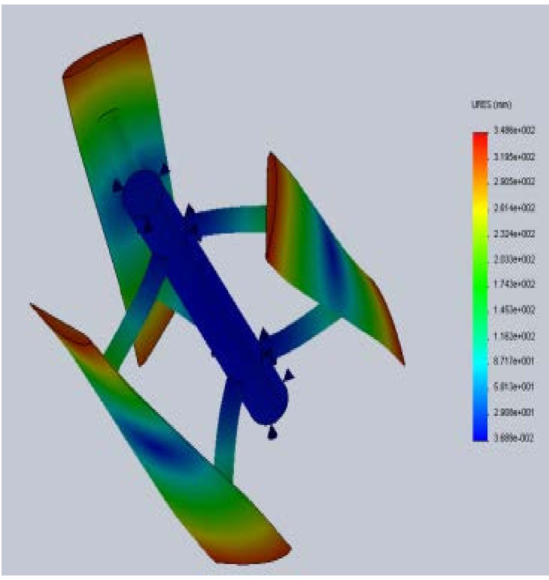
Fig-4-1　4	
Fig-4-1　6	
Fig-4-1　9	

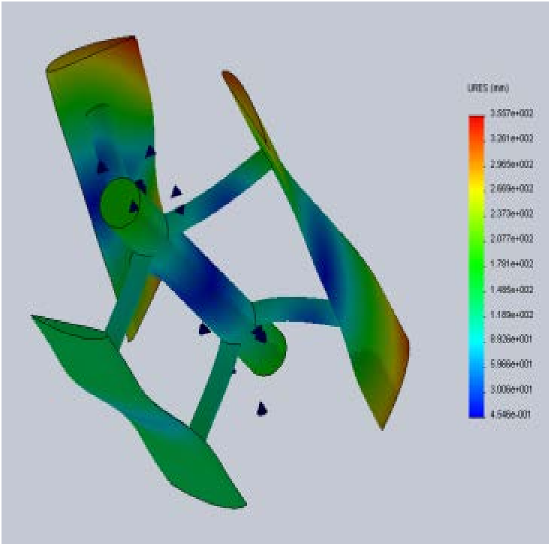
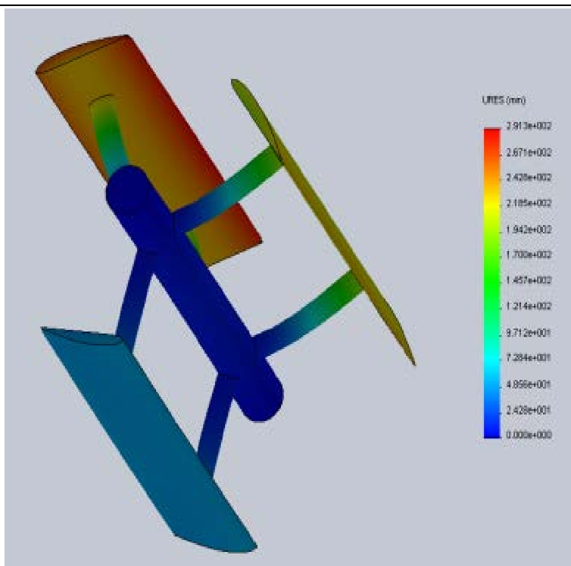
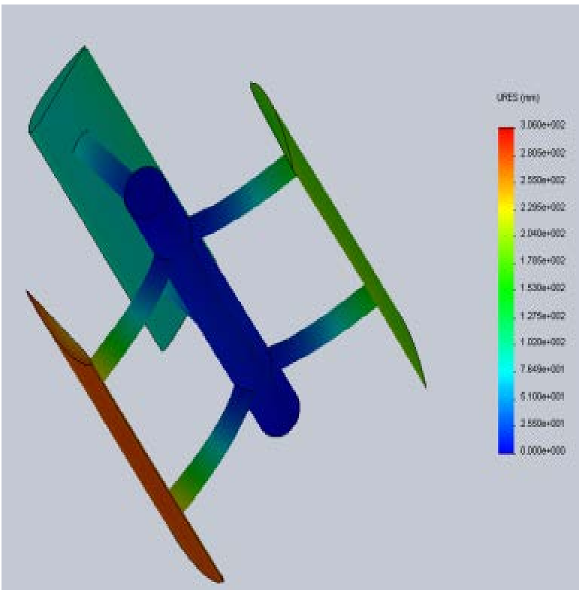
Fig-4-1 10	
Fig-4-2 2	
Fig-4-2 3	

Fig-4-2　7

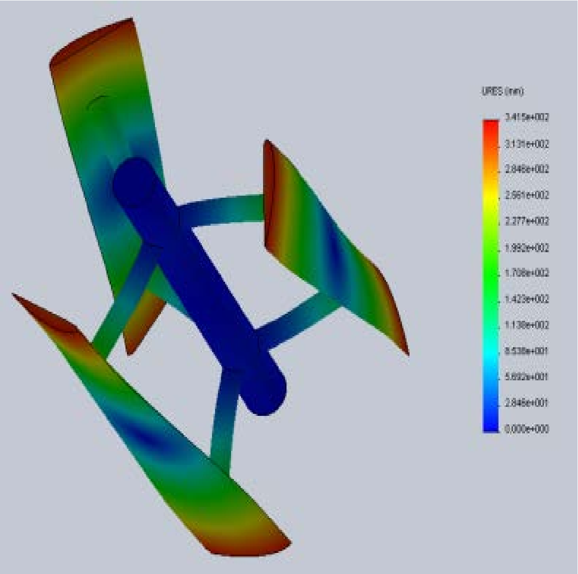


Fig-4-2　9

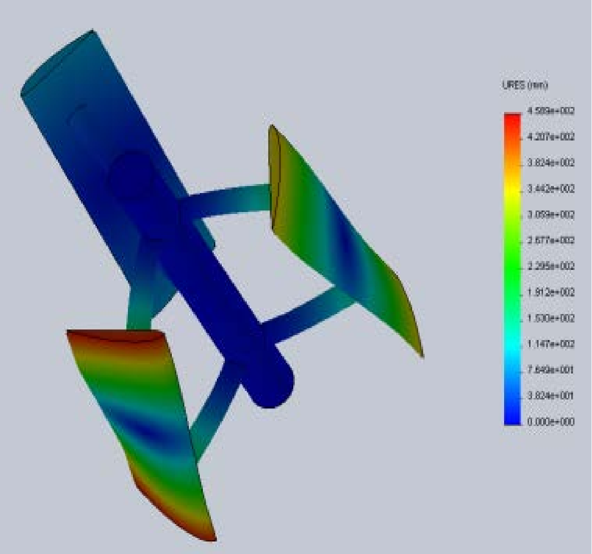


Fig-4-3　3

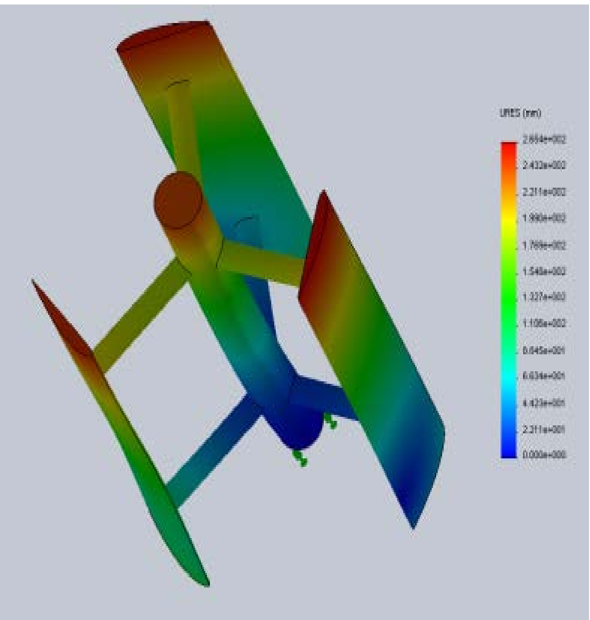
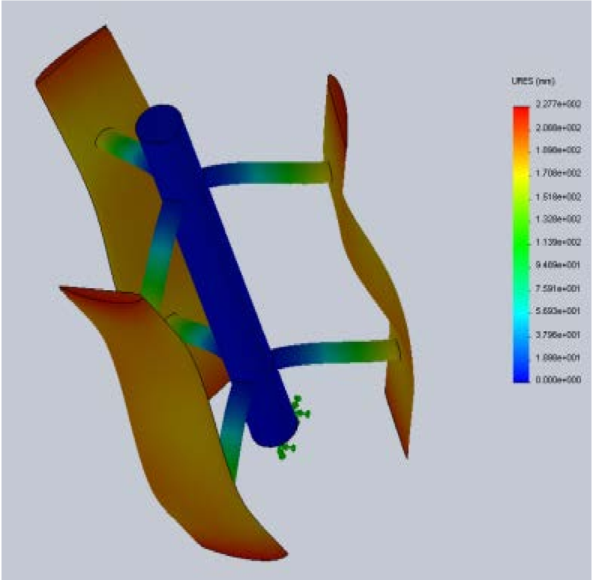
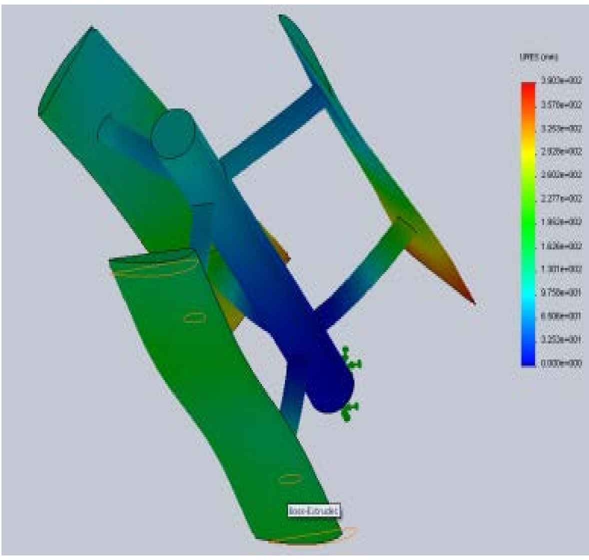
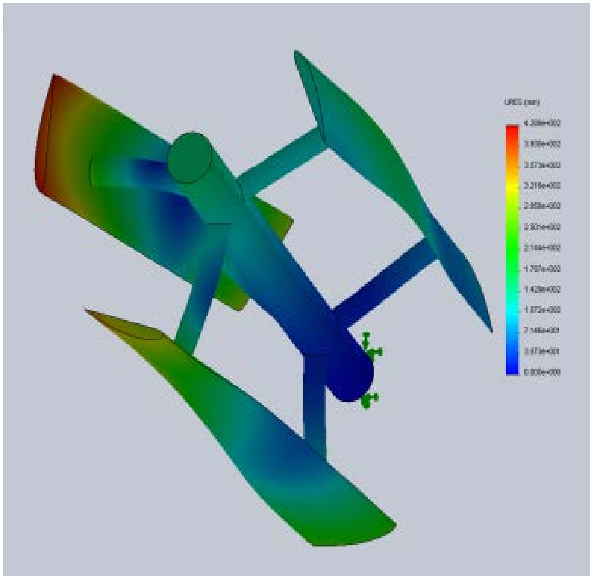


Fig-4-3 7	
Fig-4-3 9,	
Fig-4-3 10.	

4.3 New VAWT model design:

Following the vibration simulation normal H-rotor VAWT model, we analysed the results and found that by adding to the diameter of the central shaft; we could reduce the length and increase the stability of the VAWT unit and also the frequency of vibration and the degree of vibration. By thickening the length of the connecting shafts, we can increase the toughness of the VAWT, and therefore decrease the vibration frequency and the extent of the vibration, reducing the deformation of the VAWT. According to the principles of wind energy，increasing the length of the VAWT and diameter of the blades can increase the efficiency of the VAWT.

4.3.1Used SOLIDWORKS to draw a MODEL of the New VAWT:

Fig-4-3-1

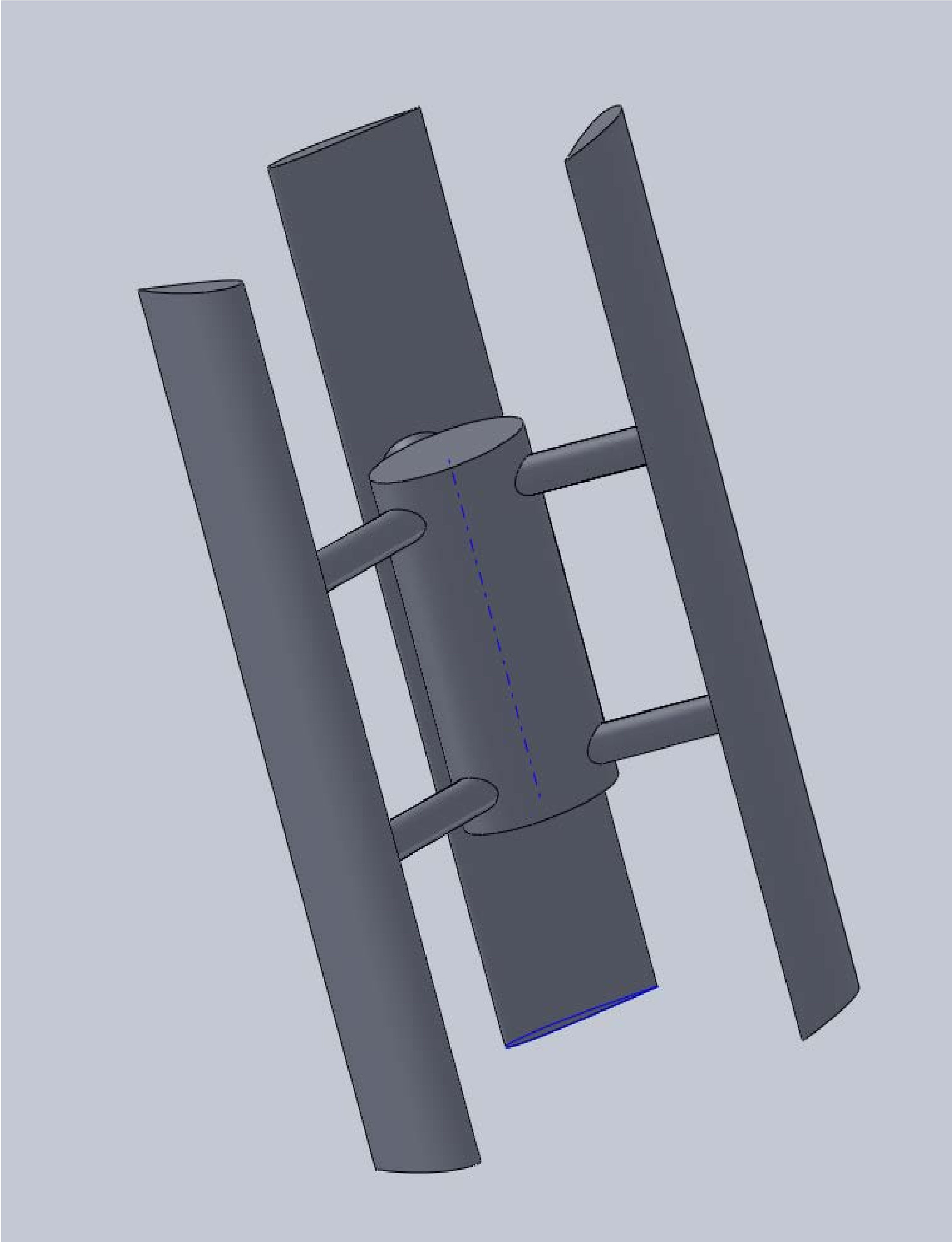


Fig-4-3-1 New H-rotor VAWT model

VAWT dimensions:
Diameter：1000MM, Height：1000MM
Central Shaft:
Diameter: 200MM,length：600MM
Connecting shafts：
Dimensions: Oval shaped 50MM x 30MM
Turbine blades:
Width: 200MM, model NACA2415

The turbine will be built with Aluminium Alloy 1060-H18 ，a diagram of its properties is given below（Fig-4-3-2）

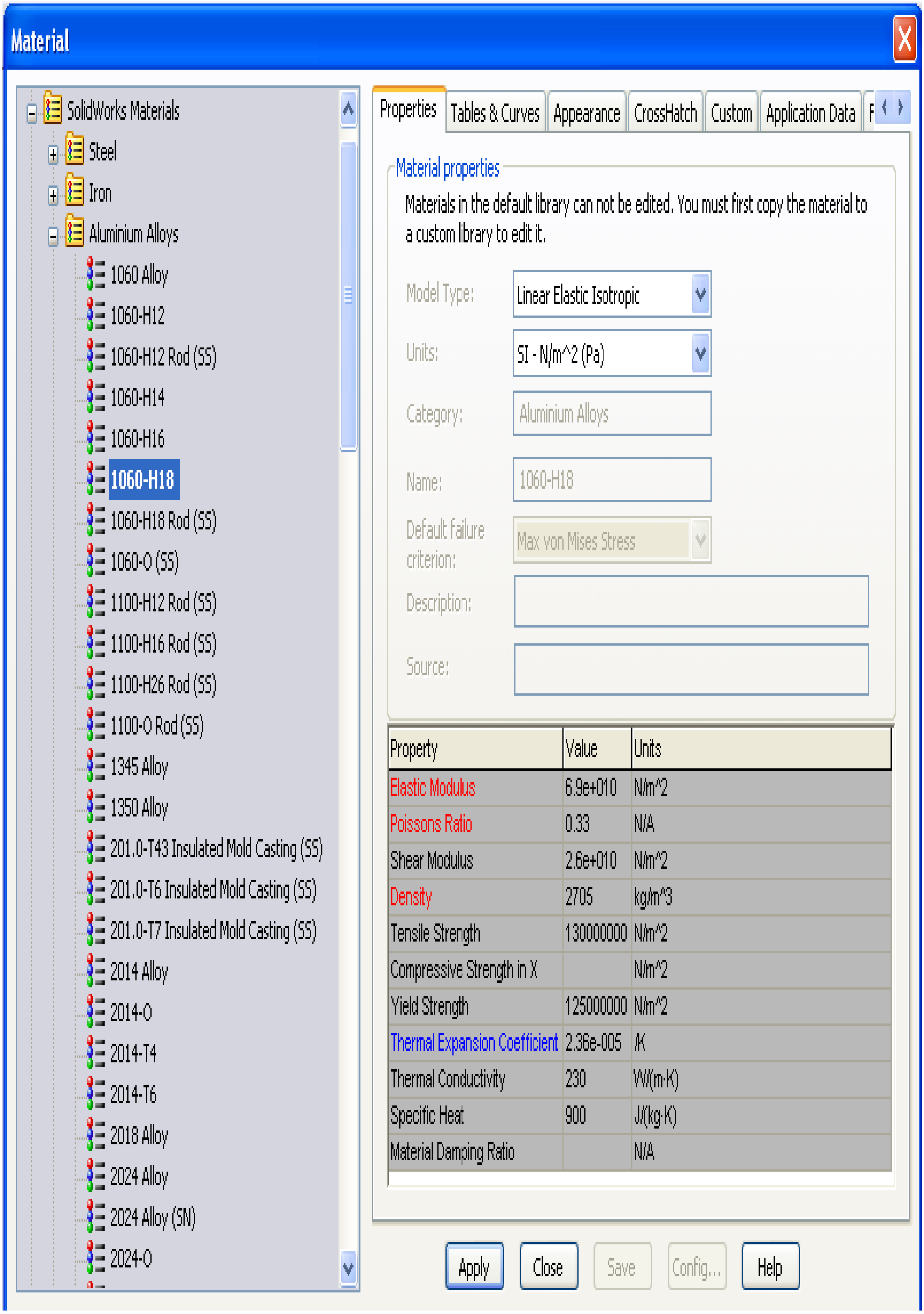

Material
SolidWorks Materials
Steel
Iron
Aluminium Alloys
1060 Alloy
1060-H12
1060-H12 Rod (SS)
1060-H14
1060-H16
1060-H18
1060-H18 Rod (SS)
1060-O (SS)
1100-H12 Rod (SS)
1100-H16 Rod (SS)
1100-H26 Rod (SS)
1100-O Rod (SS)
1345 Alloy
1350 Alloy
201.0-T43 Insulated Mold Casting (SS)
201.0-T6 Insulated Mold Casting (SS)
201.0-T7 Insulated Mold Casting (SS)
2014 Alloy
2014-O
2014-T4
2014-T6
2018 Alloy
2024 Alloy
2024 Alloy (SN)
2024-O
Properties
Tables & Curves
Appearance
CrossHatch
Custom
Application Data
Material properties
Materials in the default library can not be edited. You must first copy the material to a custom library to edit it.
Model Type:
Linear Elastic Isotropic
Units:
SI - N/m^2 (Pa)
Category:
Aluminium Alloys
Name:
1060-H18
Default failure criterion:
Max von Mises Stress
Description:
Source:
Property
Value
Units
Elastic Modulus
6.9e+010
N/m^2
Poissons Ratio
0.33
N/A
Shear Modulus
2.6e+010
N/m^2
Density
2705
kg/m^3
Tensile Strength
130000000
N/m^2
Compressive Strength in X
N/m^2
Yield Strength
125000000
N/m^2
Thermal Expansion Coefficient
2.36e-005
/K
Thermal Conductivity
230
W/(m·K)
Specific Heat
900
J/(kg·K)
Material Damping Ratio
N/A
Apply
Close
Save
Config...
Help

Fig-4-3-2

4.3.2 Solidworks simulation mesh shown Fig-4-3-3

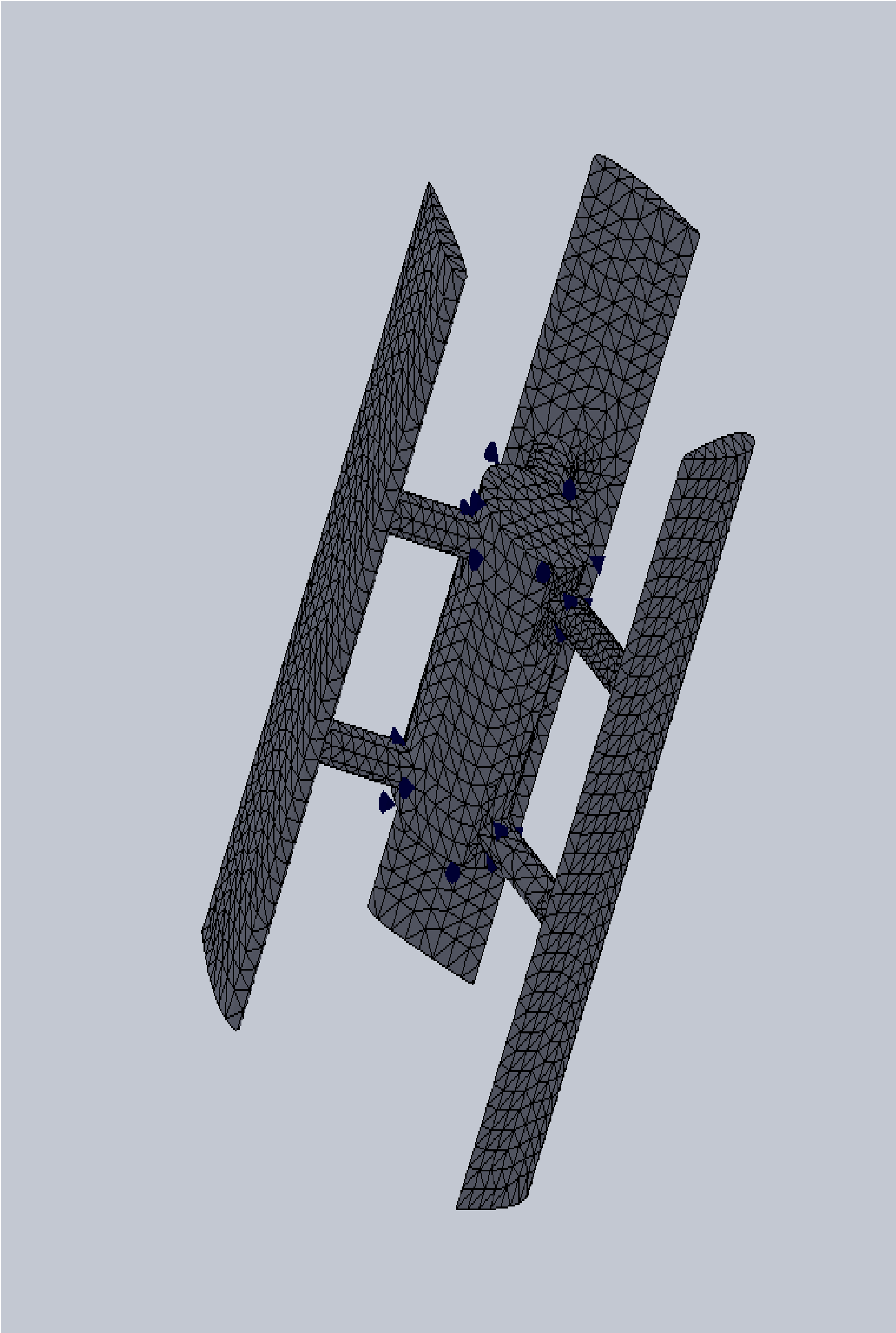


Fig-4-3-3 mash

4.4. Model of VAWT model at three types of fixed d by Solidworks compared with old model

We used Solidworks to simulate a new H -rotor model at three fixed system natural frequencies in order to find the differences in the newer and older models and conclude whether our improvements have had a positive effect.

4.4.1 Bearing fixtures new VAWT model of fixed systems natural frequency simulation by Solidworks We get simulation results show(Fig-4-4-1) and list resonant frequencies, list mass participation

.

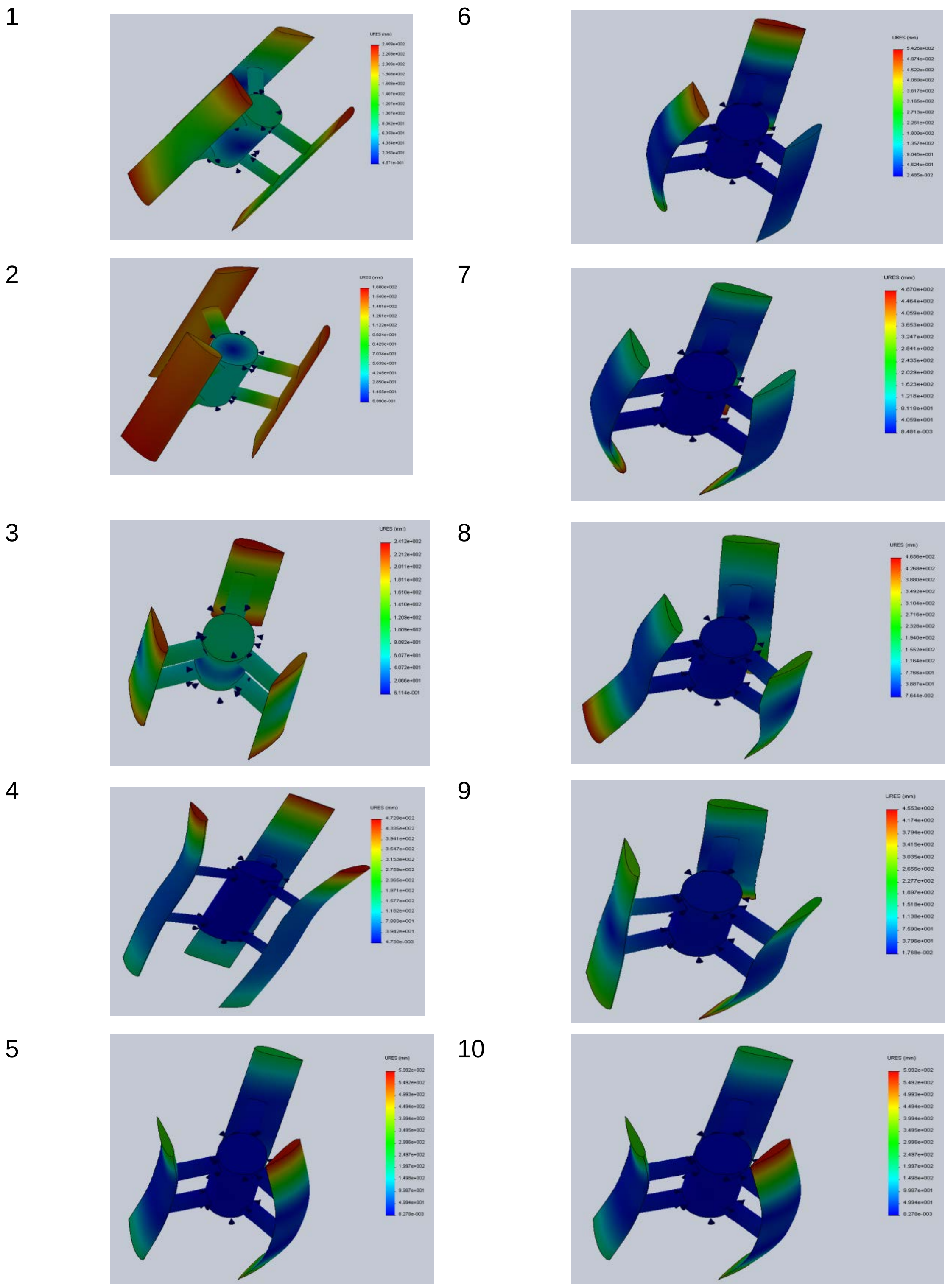


Fig-4-4-1 Bearing fixtures new VAWT model de natural frequency simulation

List of resonant frequencies

mode No	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	0.020042	0.0031897	313.51
2	0.0053275	0.00084789	1179.4
3	0.012894	0.0020521	487.31
4	1027.1	163.47	0.0061174
5	1054.	167.76	0.005961
6	1054.9	167.88	0.0059565
7	1054.9	169.3	0.0059066
8	1100.1	175.09	0.0057112
9	1100.8	175.2	0.0057078
10	1156.1	184	0.0054348

.

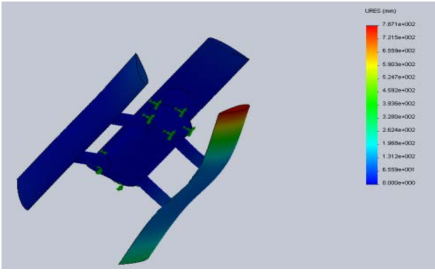
List of mass participation show

mode No.	Freq (Hertz)	X direction	Y direction	Z direction
1	0.0031897	7.8568e-007	3.3785e-011	1.2531e-009
2	0.0008478	4.1677e-009	1.6033e-011	7.5928e-012
3	0.0020521	7.4388e-011	7.8883e-007	1.9272e-009
4	163.47	1.1914e-005	1.0964e-006	0.048694
5	167.76	0.078372	0.015453	8.2607e-005
6	167.88	0.0154	0.078417	1.5501e-006
7	169.3	0.00010873	7.2581e-005	0.01208
8	175.09	0.0044381	0.0022722	4.3648e-011
9	175.2	0.0021354	0.0042054	2.6239e-007
10	184	1.7838e-007	4.7525e-007	6.7656e-005
		Sum X = 0.10047	Sum Y = 0.10042	Sum Z = 0.060926

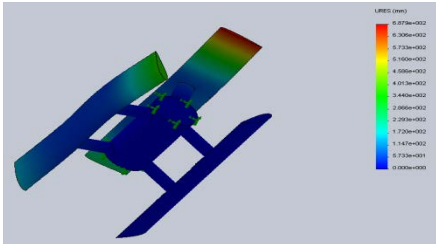
.

4.4.2 Two face fixed geometry New VAWT fixed systems model of natural frequency simulation by Solidworks We get simulation results shown (Fig-4-4-2) and list resonant frequencies, list mass participation

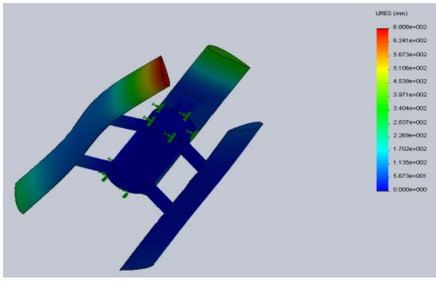
1



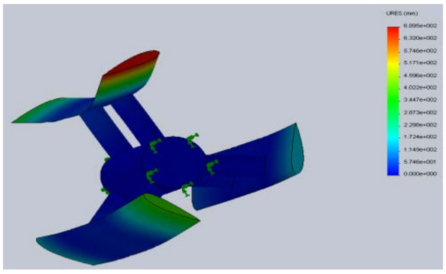
2



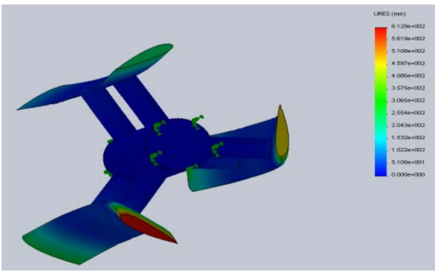
3



4



5



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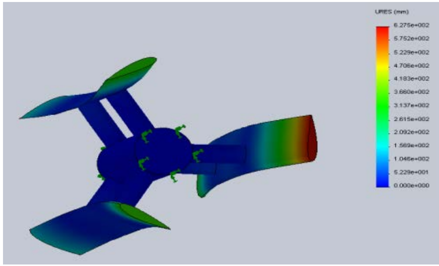

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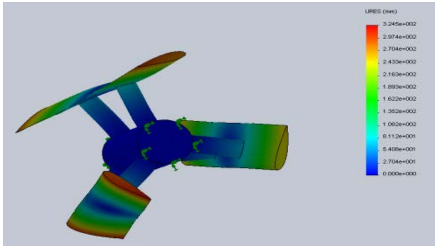
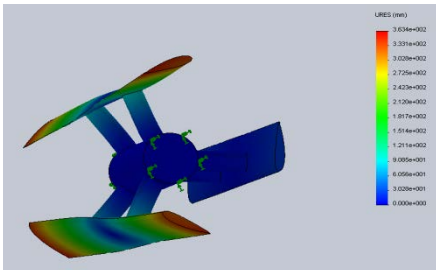
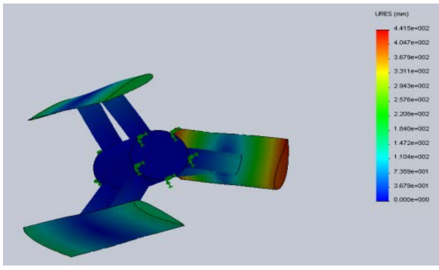


8



9



10

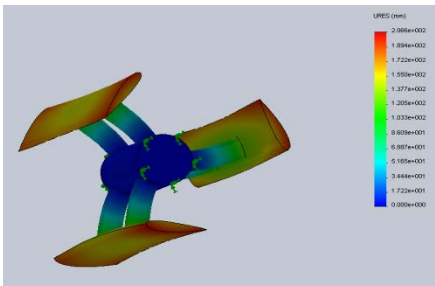


Fig-4-4-2Two face fixed geometry New VAWT model de natural frequency simulation

List of resonant frequencies

Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	978.59	155.75	0.0064206
2	979.12	155.83	0.0064172
3	1018	162.03	0.0061718
4	1036.1	164.91	0.0060641
5	1056.3	168.11	0.0059484
6	1057.4	168.29	0.0059421
7	1068.7	170.09	0.0058793
8	1084.6	172.61	0.0057933
9	1085.2	172.71	0.0057901
10	1363.1	216.95	0.0046094

.

List of mass participation

Mode No	Freq (Hertz)	X direction	Y direction	Z direction
1	155.75	0.0012806	0.0032585	1.7982e-007
2	155.83	0.0034306	0.0011987	2.6908e-006
3	162.03	1.1287e-009	1.6882e-005	0.032483
4	164.91	9.0074e-006	9.9523e-005	0.027213
5	168.11	0.003651	0.10419	0.00012347
6	168.29	0.10422	0.0037342	8.1267e-006
7	170.09	1.4996e-005	0.0001535	0.011573
8	172.61	0.0038651	0.0040447	1.2807e-008
9	172.71	0.0039056	0.0034134	1.6012e-008
10	216.95	3.4765e-006	4.6982e-005	3.1029e-005
		Sum X = 0.12038	Sum Y = 0.12016	Sum Z = 0.071434

.

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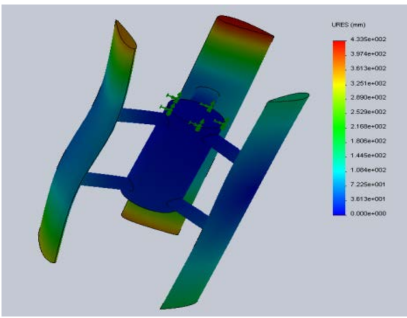
4.4.3 One face fixed geometry VAWT of fixed systems natural frequency simulation by Solidworks. We get simulation results shown (Fig-4-4-3) and list resonant frequencies, list resonant frequencies.

1



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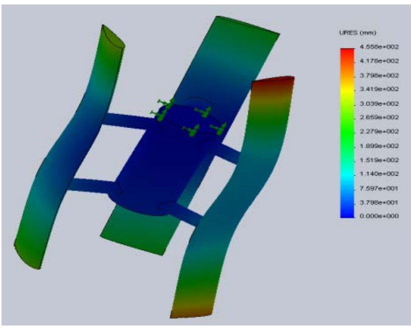
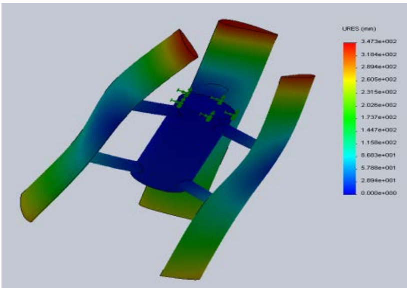


3



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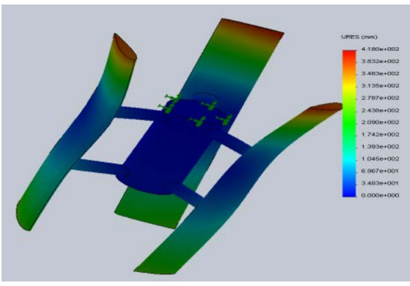

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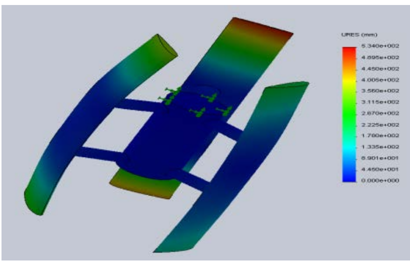

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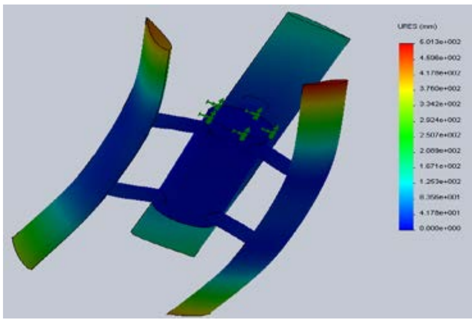

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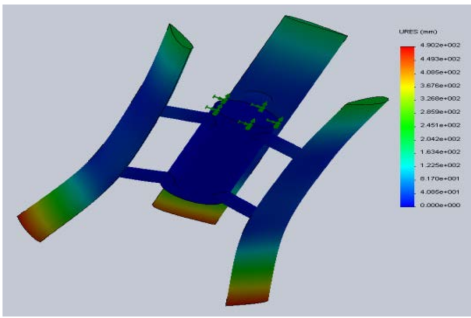

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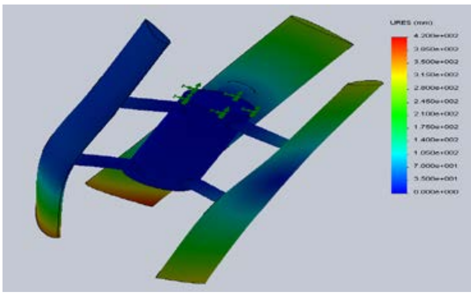

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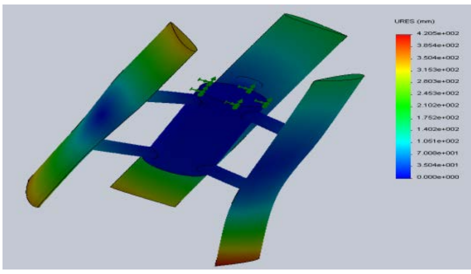

10

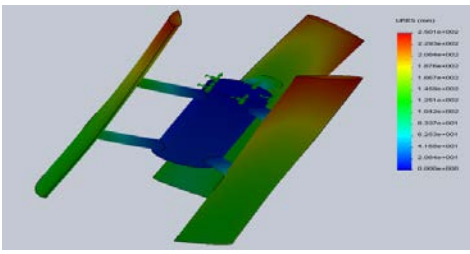

Fig-4-4-3 One face fixed geometry VAWT de natural frequency simulation by Solidworks

List of resonant frequencies

Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	1018.7	162.13	0.0061677
2	1020.1	162.36	0.0061591
3	1020.4	162.4	0.0061577
4	1056.3	168.12	0.0059482
5	1056.7	168.18	0.005946
6	1058.6	168.47	0.0059356
7	1085.2	172.72	0.0057898
8	1091.8	173.77	0.0057549
9	1093	173.96	0.0057486
10	1459.4	232.27	0.0043054

.

List of resonant frequencies

Mode No	Freq (Hertz)	X direction	Y direction	Z direction
1	162.13	0.010048	0.0027773	0.014961
2	162.36	0.0016308	0.013193	0.0030046
3	162.4	0.0048588	7.8319e-008	0.043211
4	168.12	0.0046061	0.080028	0.00010272
5	168.18	0.075725	0.0035628	0.00093935
6	168.47	0.0051265	0.0022506	0.011088
7	172.72	1.5081e-005	2.5784e-006	0.00043055
8	173.77	7.8298e-005	0.00036135	1.4868e-005
9	173.96	0.00031778	1.0028e-006	1.3588e-005
10	232.27	5.4156e-006	1.1361e-006	5.5783e-006
		Sum X = 0.10241	Sum Y = 0.10218	Sum Z = 0.073771

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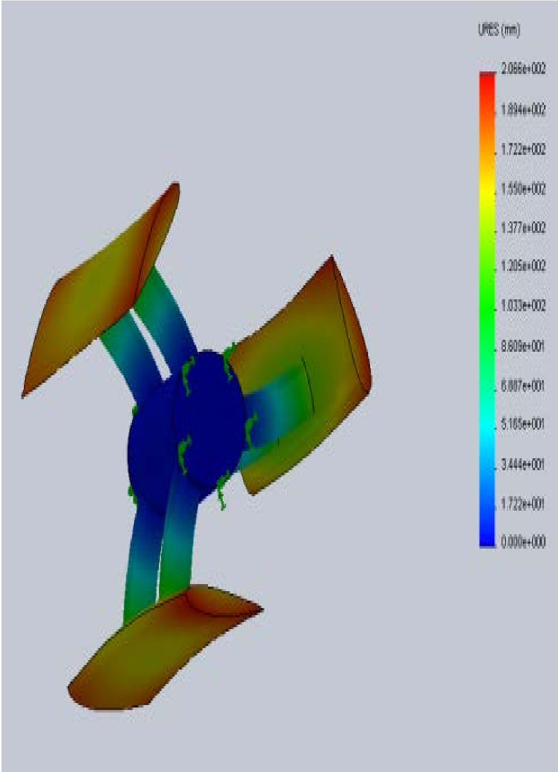
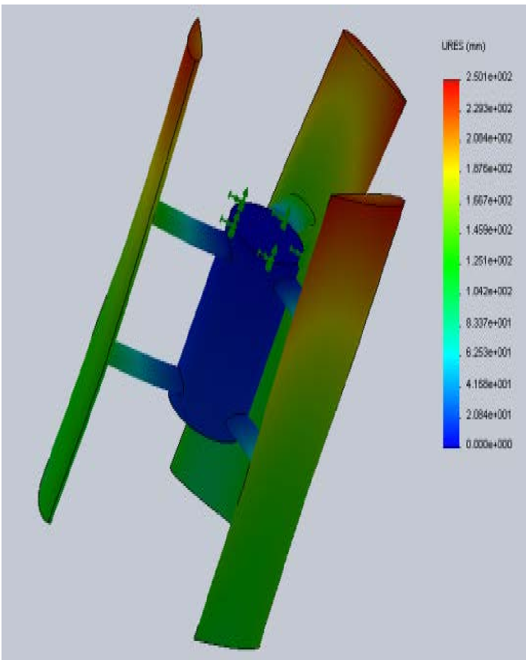
4.5 Chapter Summary

The properties of the new H-rotor VAWT model:

1. Bearing fixtures normal H-rotor VAWT model of natural frequency simulation of natural frequency the range was 0---184Hz. The degree of change of the vibration was X axis towards 0.10047,Y axis towards 0.10042,Z axis towards 0.060926
2. For the two faced fixed geometry VAWT model of natural frequency simulation of natural frequency the range was 155.75----172.61Hz. The degree of change of the vibration was X axis towards 0.12038,Y axis towards 0.12016,Z axis towards 0.071434
3. For the one faced fixed geometry VAWT model of natural frequency simulation of natural frequency the range was 162.13----232.23Hz. The degree of change of the vibration was X axis towards 0.10241,Y axis towards 0.102184,Z axis towards 0.073771
4. New VAWT model simulation shows that our previous analysis was correct because we improved the toughness and efficiency of the VAWT.
5. New model was larger than the old model by a scale of almost 2:1 （600mm---1000mm）, but the vibration frequency and extent of vibration did not increase，the deformation of the VAWT did not increase.
6. The design of the new model is effective，it is identical to the estimated results.
7. For the new model and the wind direction and vibration has little change, this really reduces the possibility of resonance. show Fig-4-4-2-10; Fig-4-4-3-10.

This thesis through the results of simulation of two selected VAWT units we have reached an expected result. And we have found a way which has some use in designing and constructing a VAWT. But this result is only valid in an ideal setting and does not consider external factors such as wind speed, temperature, material of the VAWT etc. This is an for our future research.

Fig-4-4-2-10	
Fig-4-4-3-10	

Chapter 5 Conclusion and Future Prospects

This chapter contains the conclusion and future prospects for research. This chapter will also summarize the vibration simulations and analysis, including the process of designing the normal VAWT and turning it into a New VAWT and a comparison between the two designs. I will also provide advice for future designers and vibration analyzers of VAWTs.

5.1 Conclusion

1. The natural frequencies of the VAWT and the extent of vibration of the VAWT have a large relationship with the structure and design of the VAWT.
2. Through the finite element analysis of Solidworks, it is found that the natural frequencies of the VAWTs could be controlled.
3. Through the analysis and simulation of the New VAWT, it is found that there was a decrease of Natural frequencies, vibration frequencies, vibration range compared to the previous normal VAWT model. I also found that the efficiency and sturdiness of the VAWT had increased due to the improvements that were made. This means that the chance of resonance is greatly decreased and also the improvement in design can also increase the power generated as well as increase the lifetime of the VAWT because the unit is less prone to resonance.
4. From history, humans are always discovering and utilizing wind energy and have had great success with it. The development of wind energy will growth aid the technological development of the human race, it has a great future.
5. VAWT is the newest future in the development of wind energy, but it is also the area that requires the most progress. It is the direction of future developments in wind energy, through the wisdom and research of scientists; it is possible to improve the current capacities of VAWT by correcting its weaknesses.

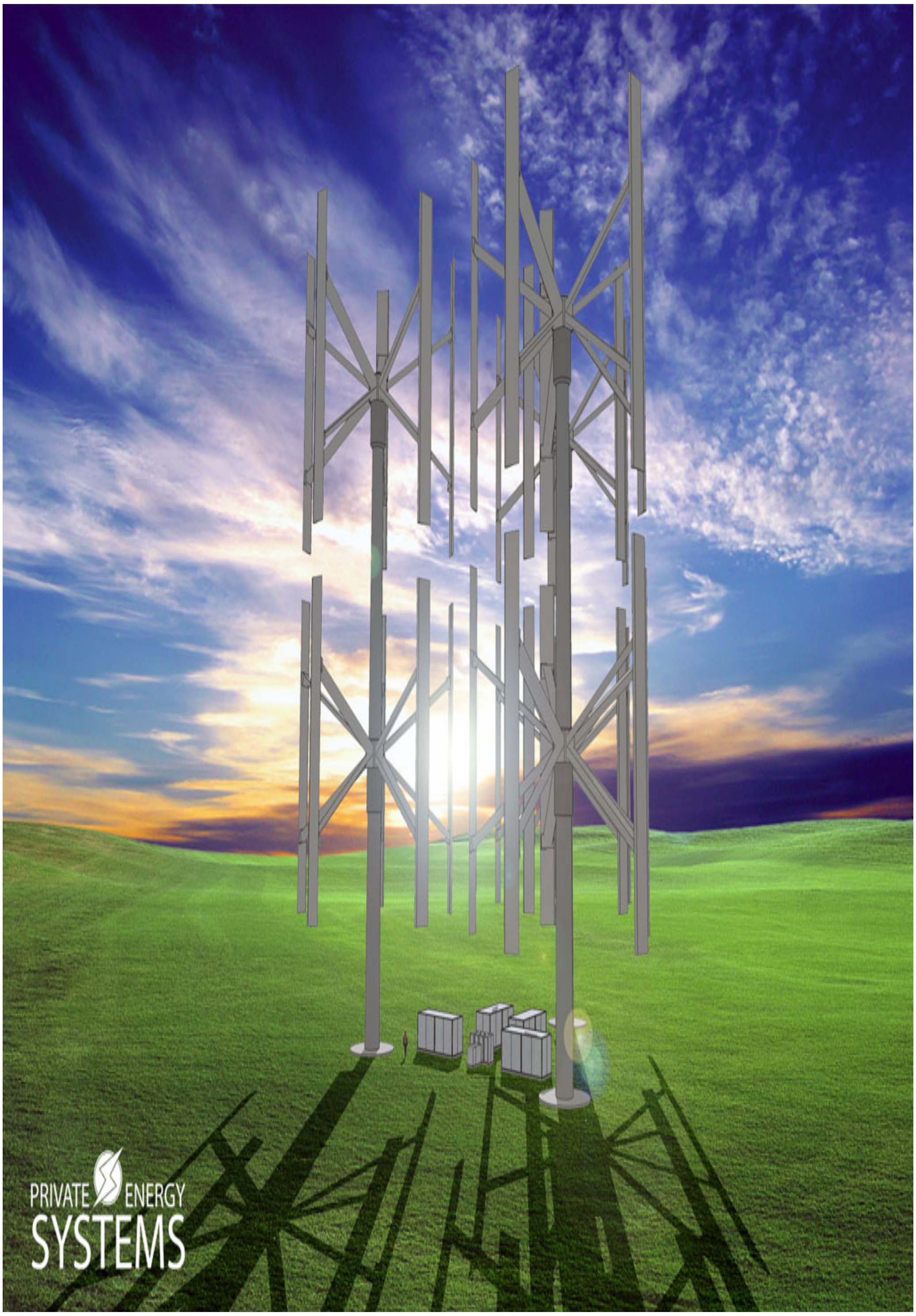


Fig-5-1　Future VAWT

5.2 Future Prospects

Wind energy is a renewable energy source that is limitlessly offered by the planet we live in, there are incredible future possibilities that have not yet been utilized. But as of present, there are still many technical problems that need to be overcome before its true potential can be harnessed. With the VAWT wind energy machines, the need to minimize resonance and to achieve a higher yield of energy is an important issue that needs to be considered. The following points can be used as a guide for possible future areas of research.

1. This thesis simulated the natural frequencies under controlled conditions with controlled variables. Therefore it is unknown whether these results apply in the real world. In the future, VAWTs could be tested under natural uncontrolled circumstances where there are many variables present which could impact on the vibration frequencies of the VAWT. This would make the conclusion more ecologically valid in the sense that it can be applied in the real world.

2. The geometrical structure of the VAWT has a large impact on the natural frequencies of the turbines. This thesis only provided one alternate model of the VAWT, which altered the dimensions of the central shaft, blades and connecting shaft of the unit, but did not change the overall geometric structure. In truth, changes could be made to the outer overall shape of the VAWT to help it reach a state where there is less resonance, increased lifetime and power output. So I recommend that the overall geometric structure of the VAWT is looked at to see if any over improvements can be made.

3. The material which was used to construct the VAWT also has an effect on its natural frequency and resonance. This thesis only tested a single type of material in the simulations that I did, so many more materials could have been used to help decrease the chance of resonance and also increase the efficiency and lifetime of the turbine, while also decreasing its vibration frequencies.

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