Aeracoustic Analysis of a Vertical Axis Wind Turbine installed in the Wake of a Bluff Body

Submitted By

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NATIONAL UNIVERSITY OF SCIENCES AND TECHNOLOGY

ISLAMABAD

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A thesis submitted in partial fulfillment of the requirements for the degree of

MS Mechanical Engineering

Thesis Supervisor:

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Thesis Supervisor's Signature: ____

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MASTER THESIS WORK

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Acknowledgements

I am thankful to my Creator Allah Subhana-Watala to have guided me throughout this work at every step and for every new thought which You setup in my mind to improve it. Indeed I could have done nothing without Your priceless help and guidance. Whosoever helped me throughout the course of my thesis, whether my parents or any other individual was Your will, so indeed none be worthy of praise but You.

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Abstract

Wind energy is a clean and renewable energy resource. Main streaming of this form of energy has produced the possibility of making wind turbines a part of urban landscape. Aerodynamic performance of a Horizontal & Vertical Axis Wind Turbines has been significantly improved by efficient blade design and smart composite materials. This study primarily deals with the acoustic effects of a vertical axis wind turbine (VAWT) when deployed in an urban environment. A parametric study has been performed to investigate the acoustic response of a VAWT in the wake of a bluff body by varying bluff body's size, distance from turbine centre and the tip speed ratio. Study has shown that tip speed ratio has the most profound impact on the noise radiated by a VAWT because by changing the TSR from 2.3 to 0.4, noise can be reduced as much as 20 dBs. Increasing bluff body's size decreases the noise level but it also severely affects the power coefficient of the turbine whereas increasing body's distance from turbine centre reduces noise without severely affecting the torque coefficient.

Key Words: *Wind Energy, Aeroacoustics, CFD, FLUENT, ANSYS*

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CHAPTER 1: INTRODUCTION

Severe weather patterns are being observed throughout the world due to global warming. Rising sea levels, heat waves, changes in precipitation patterns, expansion of deserts and melting of glaciers are some of the effects of this phenomenon. To reverse these devastating effects or to minimize them, global community is striving to cut carbon emissions and to shift towards cleaner and renewable energy resources. Wind energy is a sustainable and renewable alternative to fossil fuels and it has minimal effects on environment [1]. Wind energy can be harnessed using horizontal or a vertical axis wind turbine. This study specifically deals with vertical axis wind turbines, and these turbines are of two type i.e. Drag based and lift based.

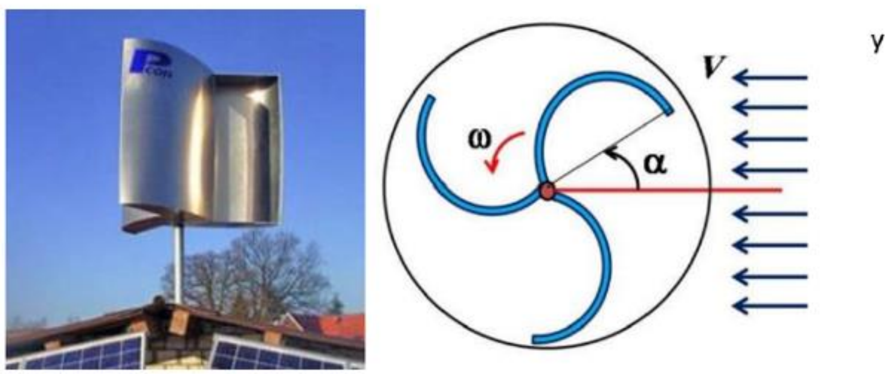

Figure 1 Drag Based Vertical Axis Wind Turbine

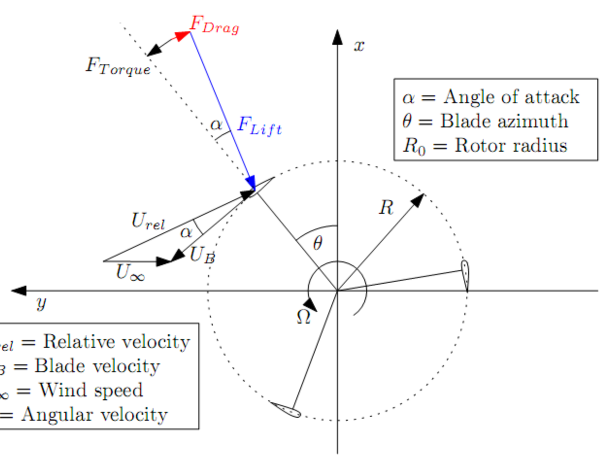

Figure 2 Lift Based Vertical Axis Wind Turbine

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Their working principle is shown in Figure 1 and Figure 2. As we can see in Figure 1, drag based savonius VAWTS are easy to manufacture because we simply cut a cylinder in two halves and then attach them back to back on a rotor. However its main drawback is that its efficiency is strongly affected and limited by the drag on one half of the cylinder moving upstream. To overcome this problem, lift based VAWTS have been designed, these rotors do not have problems like drag based VAWTs and their power coefficients can be as high as a Horizontal Axis wind Turbine. Added benefits of a vertical axis turbine is that whatever is the direction of wind, it works plus repair and maintenance is very easy because the generator and gear box are at ground level.

Wind energy has gained enormous attention of the energy researchers in the last decade. A lot of wind turbines have been installed in wind rich regions, and some of them are quite near private accommodations and work areas , however mainstreaming of wind energy has its own problems and one of the major problems is societal rejection of wind turbines installation in urban environments due to noise annoyance. Noise generated by a wind turbine has two components: mechanical noise and aerodynamic noise. Generator and gearbox are the sources of mechanical noise and this type of noise can be efficiently minimized using conventional mechanical techniques. However aerodynamic noise, which is caused by the interaction of turbine blade with incoming wind turbulence is quite difficult to overcome. [2] Traditionally, noise estimations were done using various empirical and semi empirical methods due to limited computational power. With the advancement of CFD (Computational Fluid Dynamics) techniques, it has become possible to perform aeroacoustic analysis/simulations and gather important data required for design purpose.

In this research, a reference vertical axis wind turbine (VAWT) will be modelled and aeroacoustic simulations will be performed by placing it in the downstream region of a bluff body, so that the effects of urban structures on turbine performance can be studied. This research also tends to suggest such suitable locations for installing VAWTS in urban environments to have minimum noise annoyance.

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CHAPTER 2: AEROACOUSTIC TEST CASE

Using CFD (Computational Fluid Dynamics) for simulation of vertical axis wind turbines has enabled a precise design tool to be available. But this use has been restricted to date due to the problems that traditional RANS (Reynolds-Averaged Navier Stokes) turbulence models have experienced in anticipating correctly the extremely unstable characteristics that are essential to wind turbine noise prediction [3].

This study uses Large Eddy Simulation model to obtain an instantaneous turbulent flow field. FW-H acoustic analogy formulations are then used to estimate noise generated by the turbine.

2.1 Governing Equations

2.1.1 LES Formulation

The LES method has solved the three-dimensional unstable incompressible equations of Navier Stokes. In the LES strategy, it is acknowledged that the big turbulent structures are usually much more vigorous than the tiny scales, thus resolving these big constructions, while the computational mesh scales are modeled on lower scales.

The equations are written in conservation form as:

$$\frac{\partial \overline{u}_i}{\partial x_i} = 0$$

$$\frac{\partial}{\partial t}\overline{u}_i + \frac{\partial}{\partial x_j}(\overline{u}_i\overline{u}_j) = -\frac{1}{\rho}\frac{\partial \overline{p}}{\partial x_i} + \frac{1}{\rho}\frac{\partial \tau_{ij}^R}{\partial x_j} + \nu\nabla^2\overline{u}_i$$

$$\frac{\partial \bar{u}i}{\partial xi} = 0$$

Where $\bar{u}_i$ and $\bar{p}$ show resolved velocity components and pressure respectively. The effect of unresolved scales is merged in the sub grid-stress tensor, which contains the residual stresses, τ_{ij}^R defined as:

$$\tau_{ij}^R = \rho\left(\overline{u_i u_j} - \overline{u}_i\overline{u}_j\right)$$

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The sub grid scale stresses resulting from the filtering operation are unidentified and require modeling. The dynamic sub grid model formulation of Lilly [4] is used to include the effects of unresolved turbulent scales.

2.1.2 Aeroacoustic Formulation

The Ffowcs-Williams and Hawkings (FW-H) method [5] is used in this study to model the noise issue. The FW-H equation is the most general form of the Lighthill [6] acoustic equivalence and is suitable for estimating noise generated by rigid bodies in arbitrary movement. This equation is inhomogeneous in nature and it can be obtained by combining the equation of continuity and the equations of Navier Stokes. It is possible to write the FW-H equation [7] [8] as:

$$\frac{1}{a_0^2}\frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = \frac{\partial^2}{\partial x_i \partial x_j}\{T_{ij}H(f)\} - \frac{\partial}{\partial x_i}\{[P_{ij}n_j + \rho u_i(u_n - v_n)]\delta(f)\} + \frac{\partial}{\partial t}\{[\rho_0 v_n + \rho(u_n - v_n)]\delta(f)\}$$

2.1.3 Turbine Geometry

Reference turbine selected for this research has these characteristics. It has naca 4518 aerofoil with 100 mm chord length. Turbine diameter is 600 mm and its height is 700 mm. It has three straight blades. Reason to choose this turbine is that experimental as well as theoretical results were available for it. These results were readily used for validation of this research.

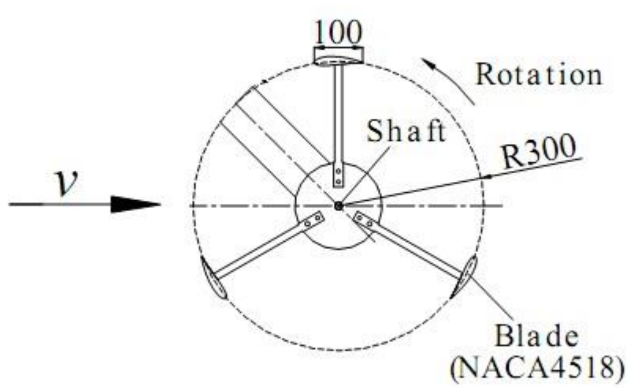

Figure 3 Turbine Geometry (Top View)

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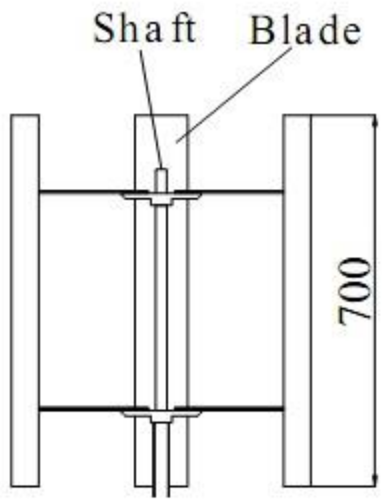

Figure 4 Turbine Geometry (Front View)

2.3 Boundary Conditions & Problem Description:

The left outer edge of the domain is modeled as the velocity inlet of 8 m / s and the right outer edge of the domain is modeled as the pressure outlet as shown in Figure 5. Turbine rotational speed was set to 300 rpm (Tip Speed Ratio = 1.2). This research simulates the unstable flow field and aerodynamic noise radiated from a VAWT. Ansys Fluent 15.0 is used to solve the Navier Stokes equations using the LES technique based on the finite volume method. Since the free stream mach number is quite low, the fluid is presumed to be incompressible. The pressure-based solver is selected because of the incompressibility of the stream, which is traditionally introduced to solve incompressible low-speed flows. Velocity and pressure equations are coupled using PISO algorithm.

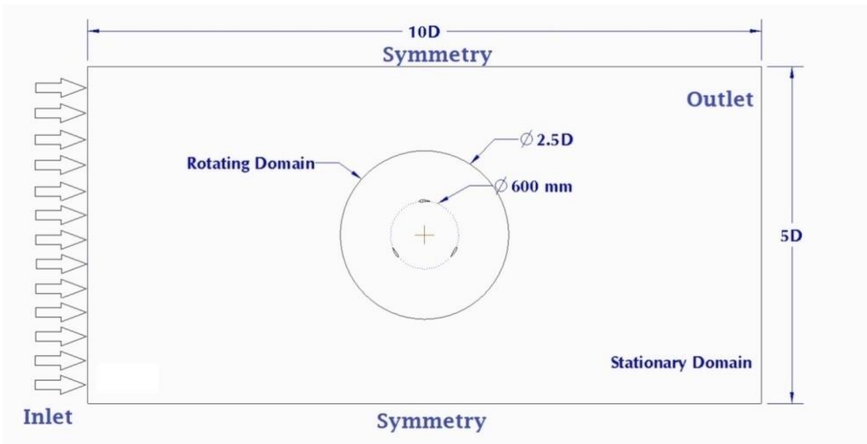

Figure 5 Boundary Conditions

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Turbulent flow field obtained as a result is then used as an input to FW-H acoustic analogy formulations to estimate real time noise data. Numerical time step used was $2.5*10^{-5}$ seconds to have low courant number and to ensure accurate temporal descritization. Simulation was run for 5 revolutions but data considered was for 3 revolutions only.

2.4 Grid Independence

Spatial discretization was accomplished through central differences of second order. Sliding mesh technique was used to model two domains, inner rotating domain and outer stationary domain. Cylindrical interface was used between these 2 domains. Special attention was paid to lessening mesh non orthogonality and skewness while building the mesh. The mesh used has 400000 cells. Height of cells along the boundary layer of airfoil was set to be 10^{-5} to have $y+<1$.

Grid independence study was done to find the optimum mesh size. A coarse mesh containing 25000 cells was initially generated; the number of cells were then doubled and quadrupled. When the cell count was increased from 0.2 million cells to 0.4 million cells, no significant change in torque coefficient was observed. Hence the simulation was done using the optimum mesh size of 0.4 million cells. Figure 6 & Figure 7 show the finalized mesh and the grid independence curve.

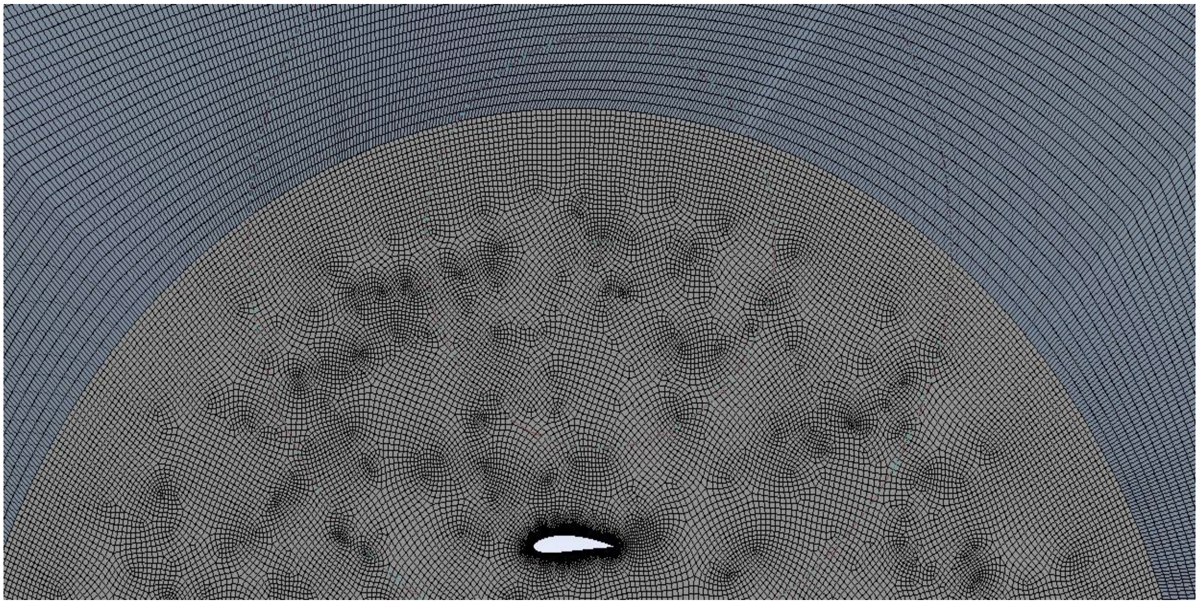

Figure 6 Optimized Mesh

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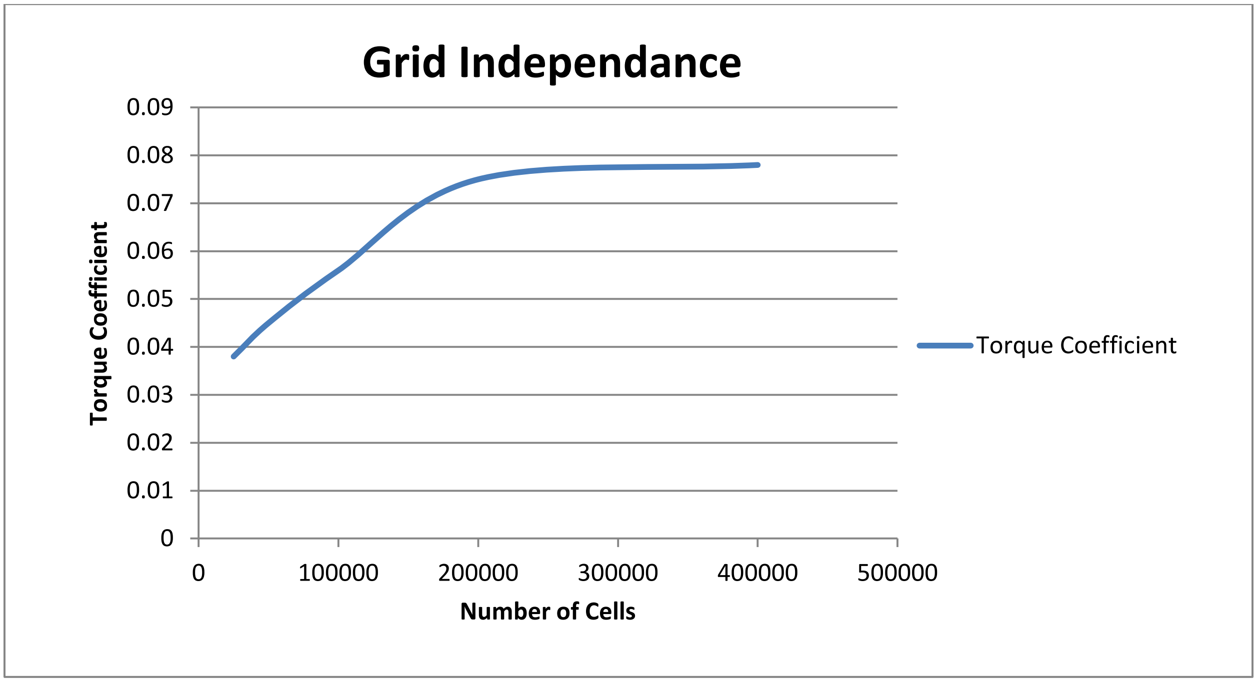

Figure 7 Grid Independence Curve

2.5 Test Case Results & Validation

2.5.1 Aerodynamic Results

Validation of aerodynamic results was carried out by comparing the measure torque coefficient with the experimental results of Takao et al. Measure mean torque coefficient obtained was 0.08. It is comparable with experimental results of the same turbine obtained by Takao et al. Both of these curves have been shown in Figure 8 & Figure 9.

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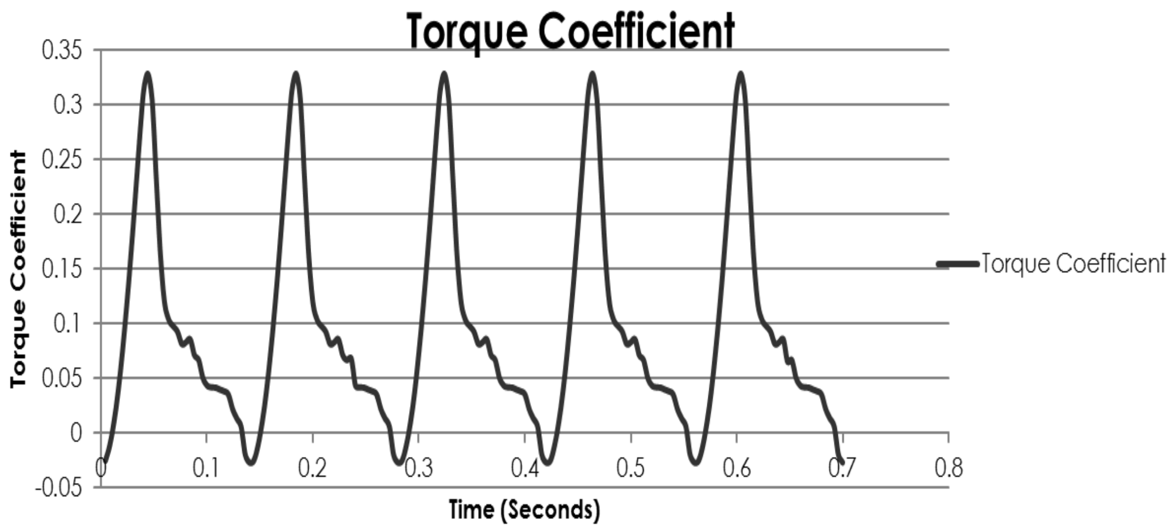

Figure 8 Torque Coefficient as obtained from Numerical Solver at γ=1.2

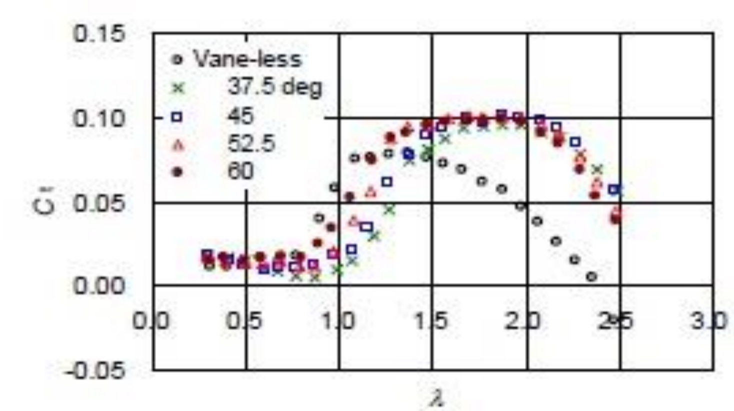

Figure 9 Torque Coefficient Obtained in Experimental Results of Takao et al

2.5.2 Acoustics Results

For validation of acoustics results, estimated noise levels were compared with the results obtained by Amir Nejat et al. Maximum noise level of 43 dBs & 61 dBs was estimated at 300 & 625 rpm respectively. It is comparable with aeroacoustic noise predictions of Amir Nejat et al. Sound pressure level curves have been shown in Figure 10, 11 & 12.

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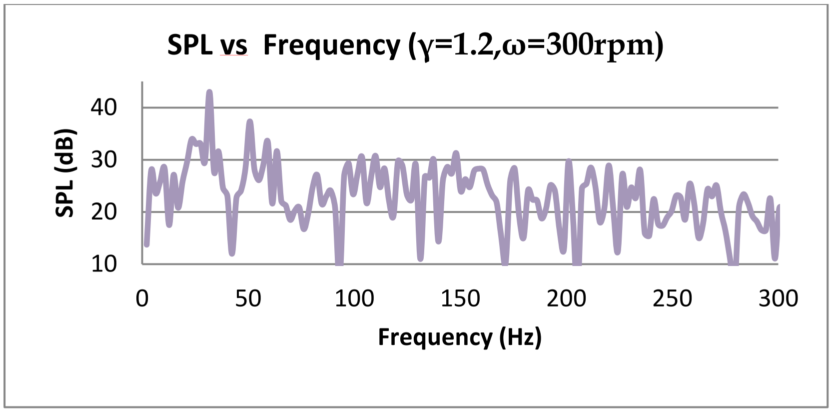

Figure 10 Sound Pressure Level at 300 rpm

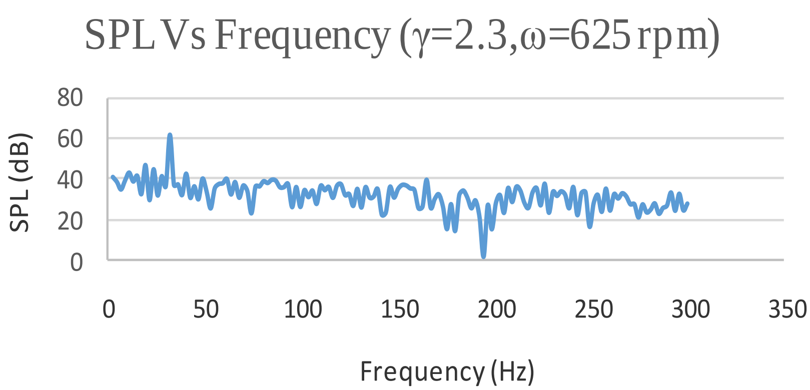

Figure 11 Sound Pressure Level at 625 rpm

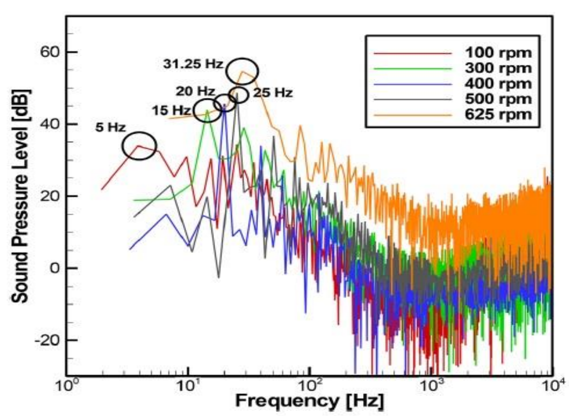

Figure 12 Sound Pressure Levels as Estimated by Amir Nejat et al.

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Chapter 3 Parametric Study

After validation of test case, 3 basic parameters and their 4 sublevels were identified, geometry for all the 10 cases was updated keeping in view urban environments, and simulations were done for all the 10 cases. These parameters have been shown in the table below:

Table 1 Parametric Study

Bluff Body Size	Bluff Body Distance	Tip Speed Ratio
0.5 D	2.0 D	2.3
1.0 D	4.0 D	1.6
1.5 D	6.0 D	1.2
2.0D	8.0 D	0.4

Assuming these parameters to be independent, 10 different combinations of these three parameters were made. These cases have been shown in the table-2:

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Table 2 Cases for Simulation

Case #	**TSR**	**Obstruction Size**	**Obstruction Distance**	**Parameter to be Varied**
1	2.3	0.5D	2.0D	Bluff Body Size
2	2.3	1.0D	2.0D	
3	2.3	1.5D	2.0D	
4	2.3	2.0D	2.0D	
5	2.3	0.5D	4.0 D	Bluff Body Distance
6	2.3	0.5D	6.0 D	
7	2.3	0.5D	8.0 D	
8	1.6	0.5D	2.0 D	Tip Speed Ratio
9	1.2	0.5D	2.0 D	
10	0.4	0.5D	2.0 D	

Keeping all the settings and basic methodology the same as was in the test case, simulations were done for all the 10 cases.

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Chapter 4 Results & Discussion

4.1 Effect of Bluff Body Size on Sound Pressure Level and Torque Coefficient

To study the effect of bluff body size, turbine was placed in downstream region of a bluff body. Square bluff body was selected and the length of each side of this square was varied from 0.5 times turbine diameter to 2.0 times the turbine diameter. Simulations were done by varying the bluff body size while other two parameters were fixed, bluff body distance at 2.0 D from turbine center and tip speed ratio at 2.3. For acoustic calculations, microphone was placed at 2 meters behind the turbine center for all the cases. Sound pressure levels and torques generated were measured in each case and following trends were observed.

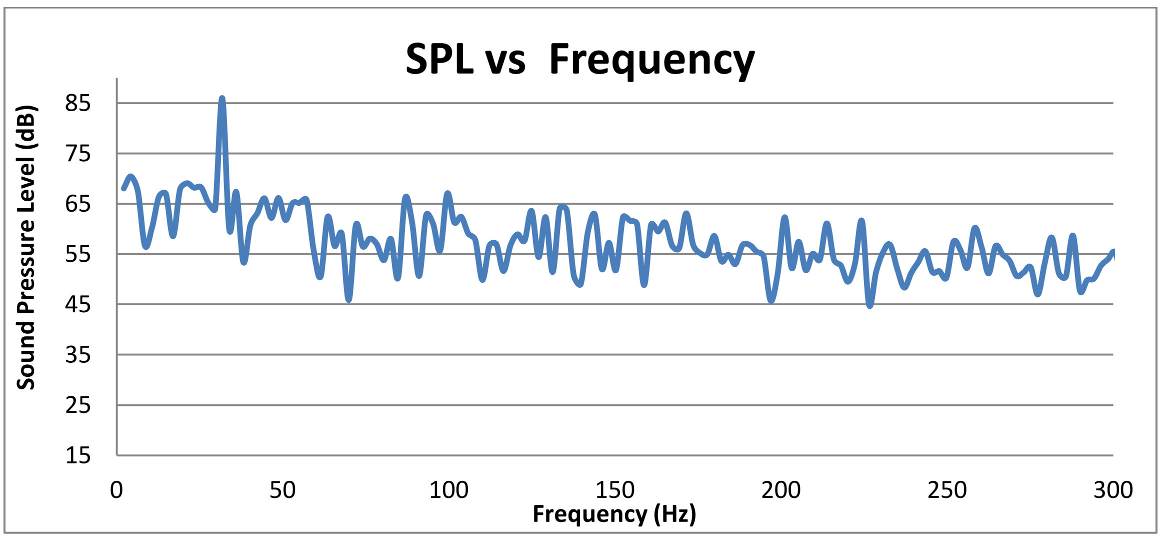

Figure 13 Sound Recorded with 0.5 D Bluff Body Size

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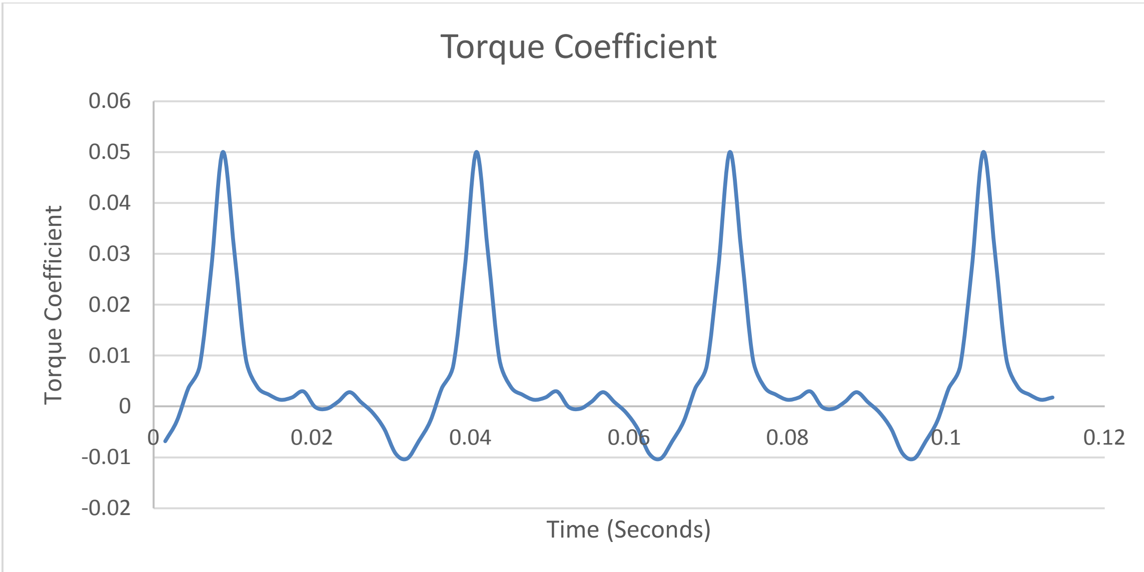

Figure 14 Torque Coefficient Recorded with 0.5 D Bluff Body Size

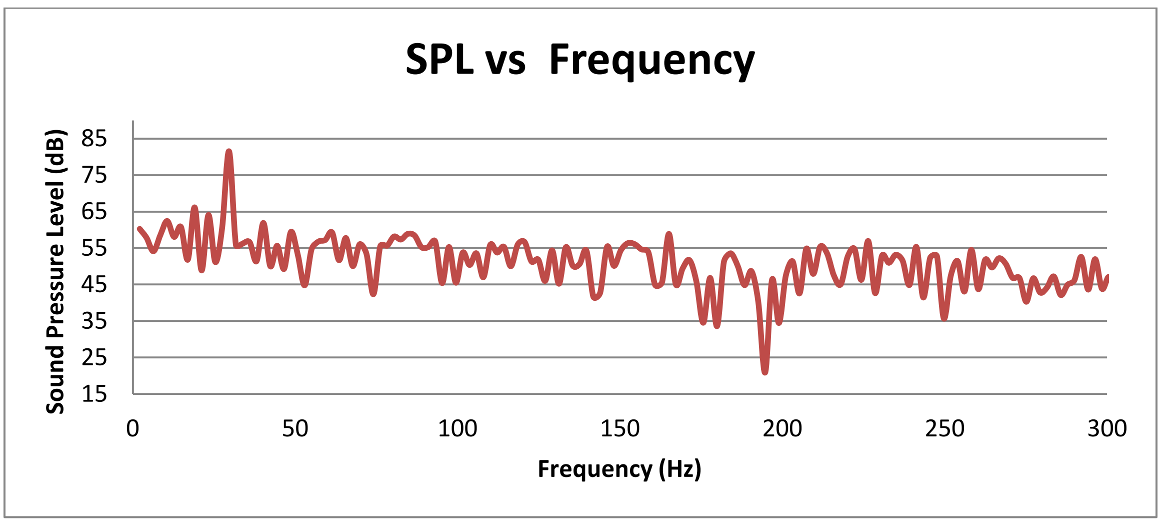

Figure 15 SPL at Bluff Body Size 1.0 D

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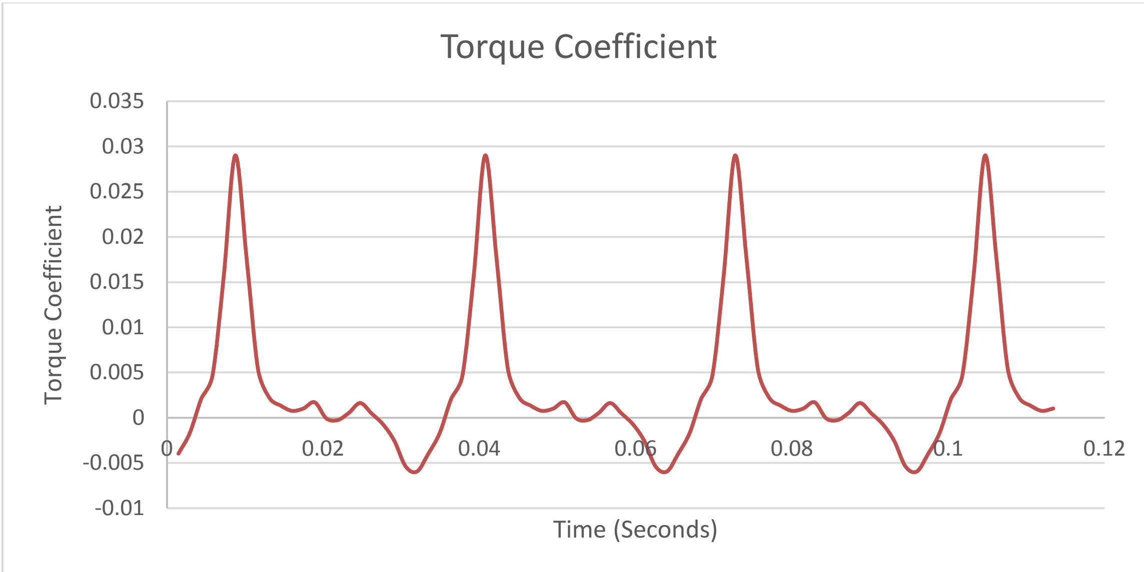

Figure 16 Torque Coefficient Recorded with 1.0 D Bluff Body Size

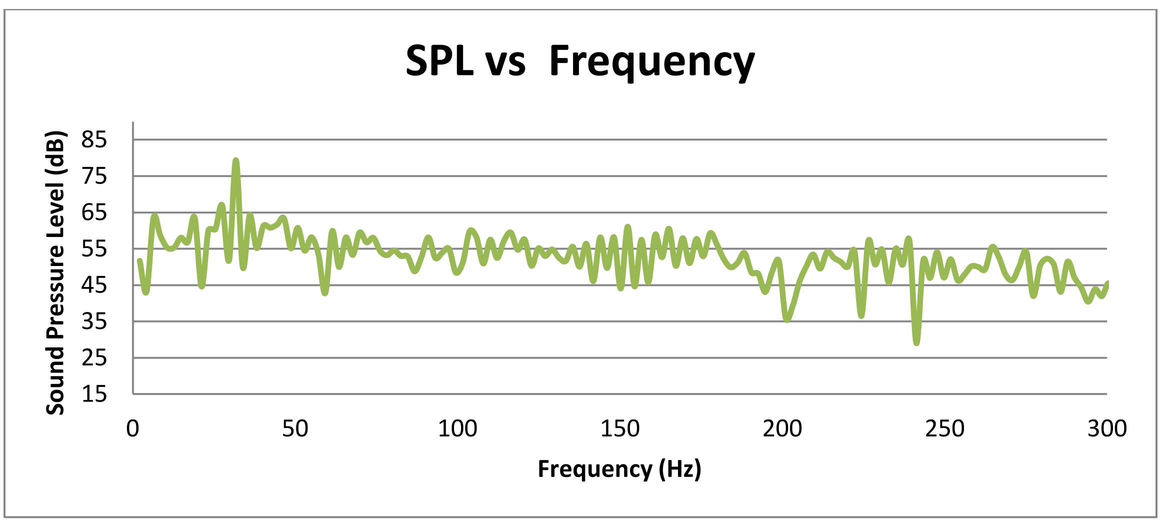

Figure 17 SPL at Bluff Body Size 1.5 D

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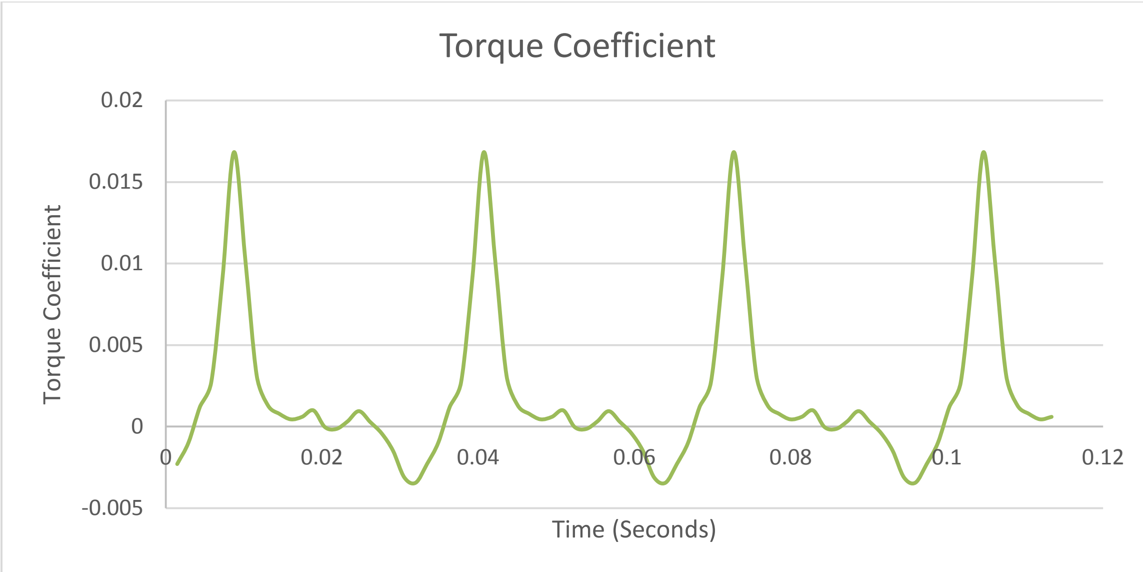

Figure 18 Torque Coefficient Recorded with 1.5 D Bluff Body Size

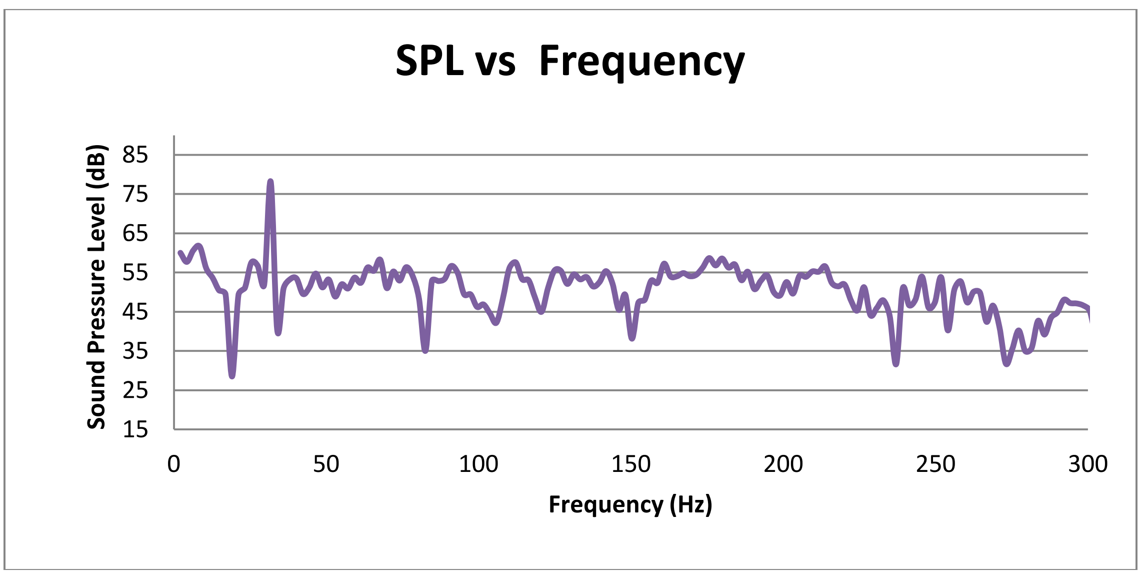

Figure 19 SPL at Bluff Body Size 2.0 D

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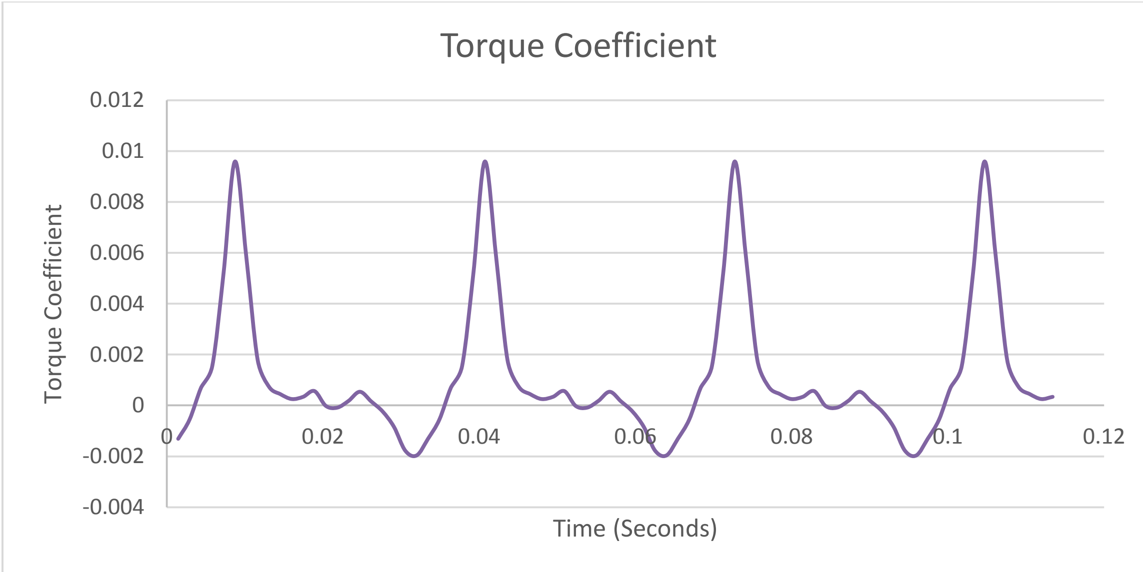

Figure 20 Torque Coefficient Recorded with 2.0 D Bluff Body Size

4.1.1 Overall Trend

It can be seen in the following graphs that by increasing the bluff body size, noise levels as well as torque coefficients rapidly decreases. Main reason behind this trend is that when bluff body size crosses the size turbine diameter, very limited air flow is available to turbine blades which results in decreased power output as well as reduced noise levels.

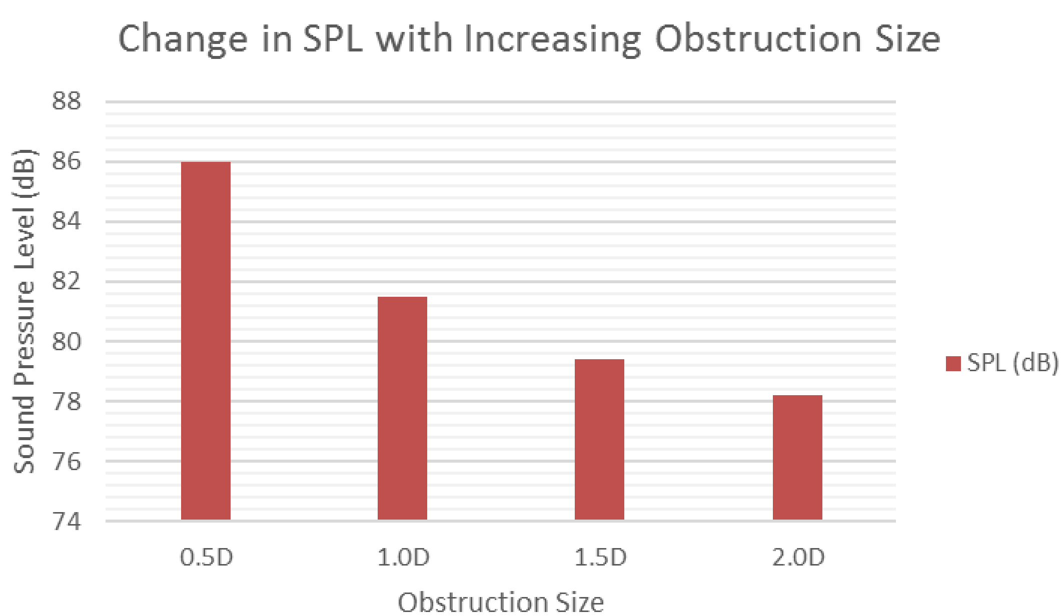

Figure 21 Effect of Bluff Body Size on SPL

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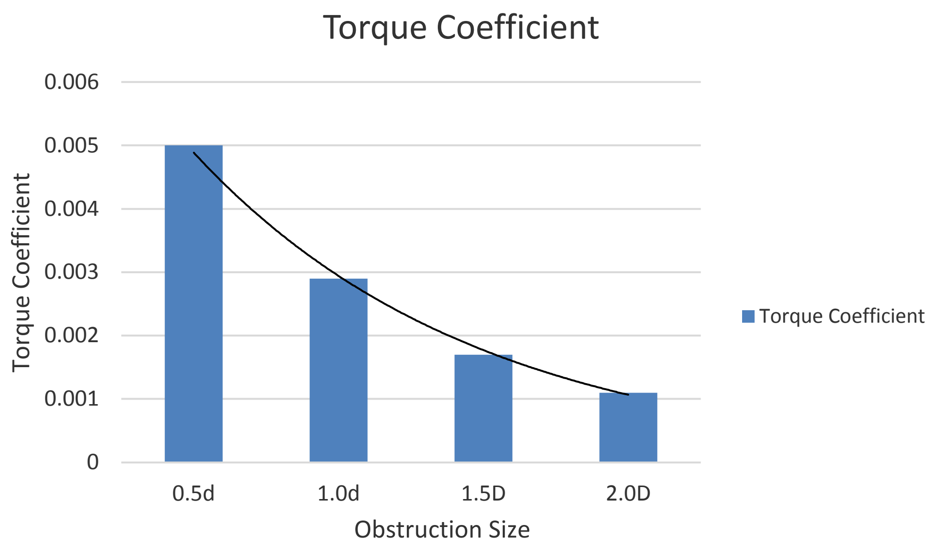

Figure 22 Effect of Bluff Body Size on Torque Coefficient

4.2 Effect of Bluff Body Distance from Turbine Center on Sound Pressure Level and Torque Coefficient

To have minimum noise annoyance for human communities, this parameter plays a vital role. Varying distance of turbine from residential buildings and finding a suitable location is very important for societal acceptance of wind turbines. Special attention was given in settings sublevels of this parameter Bluff body distance from turbine center was varied from 2 times of turbine diameter to 8 times of turbine diameter. Extra noise generated due to inclusion of a bluff body in front of turbine diminished when the distance of bluff body from turbine center reached 8 times the turbine diameter. Estimated noise and torque coefficient curves are presented below.

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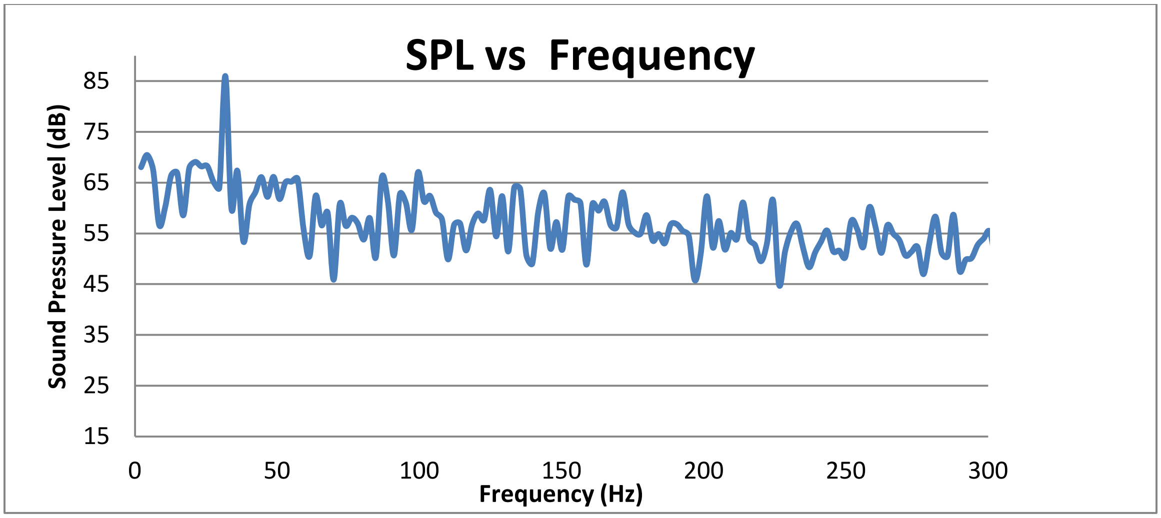

Figure 23 SPL at Bluff Body Distance 2.0 D

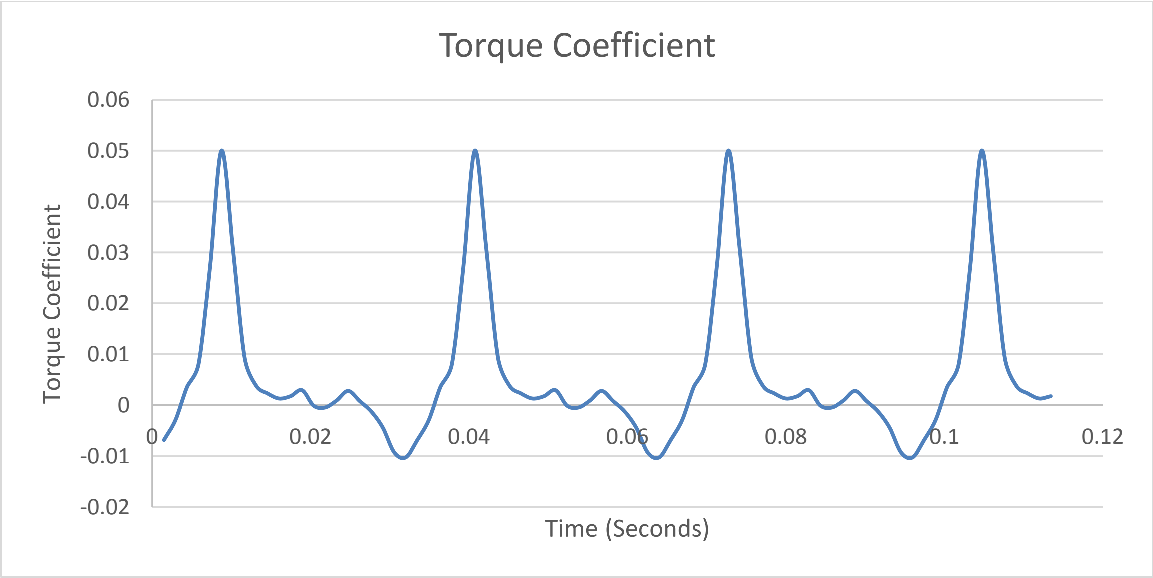

Figure 24 Torque Coefficient at Bluff Body Distance 2.0 D

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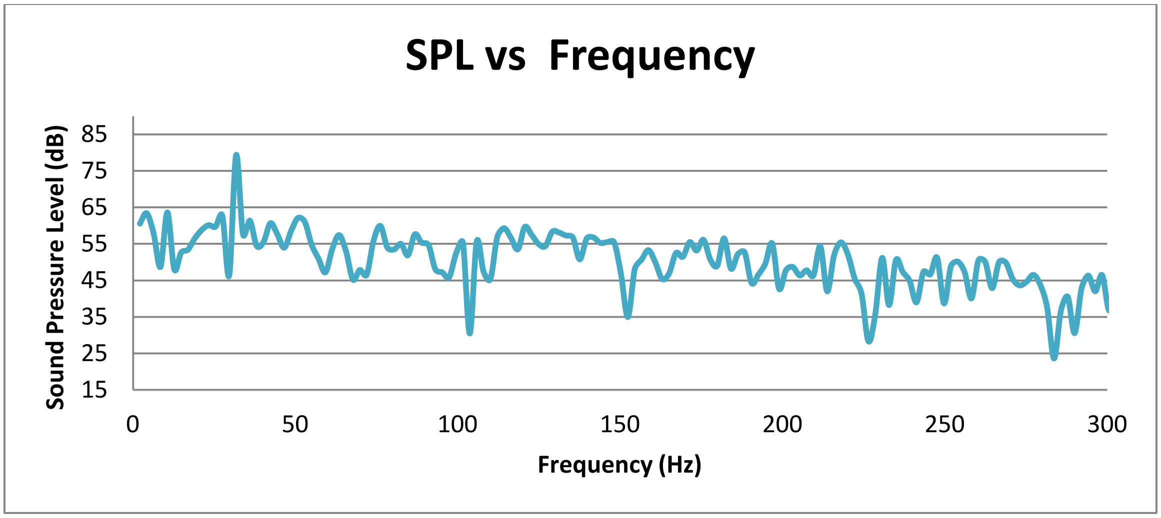

Figure 25 SPL at Bluff Body Distance 4.0 D

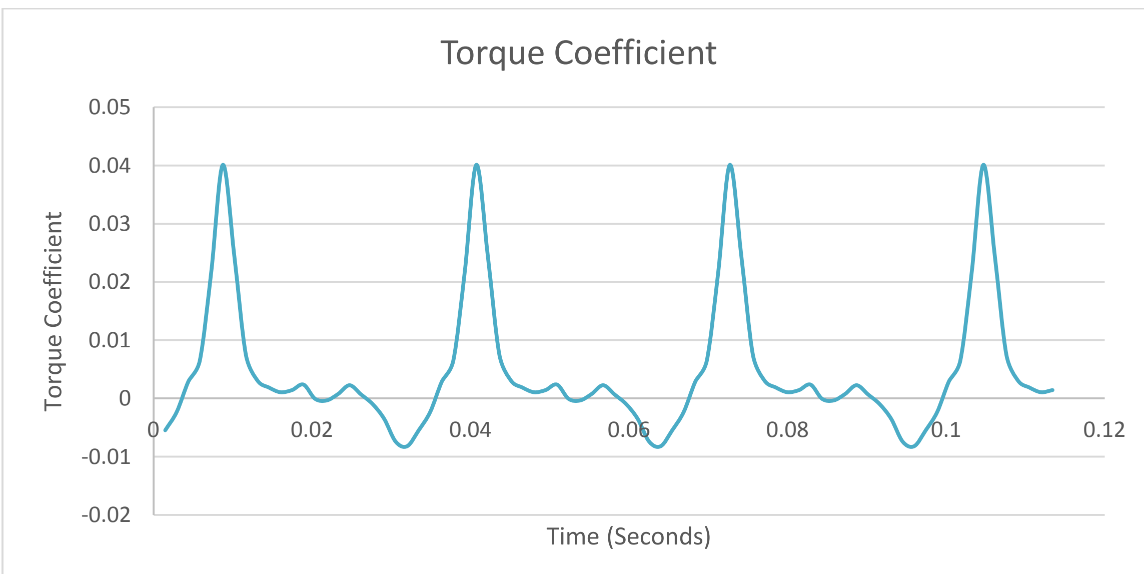

Figure 26 Torque Coefficient at Bluff Body Distance 4.0 D

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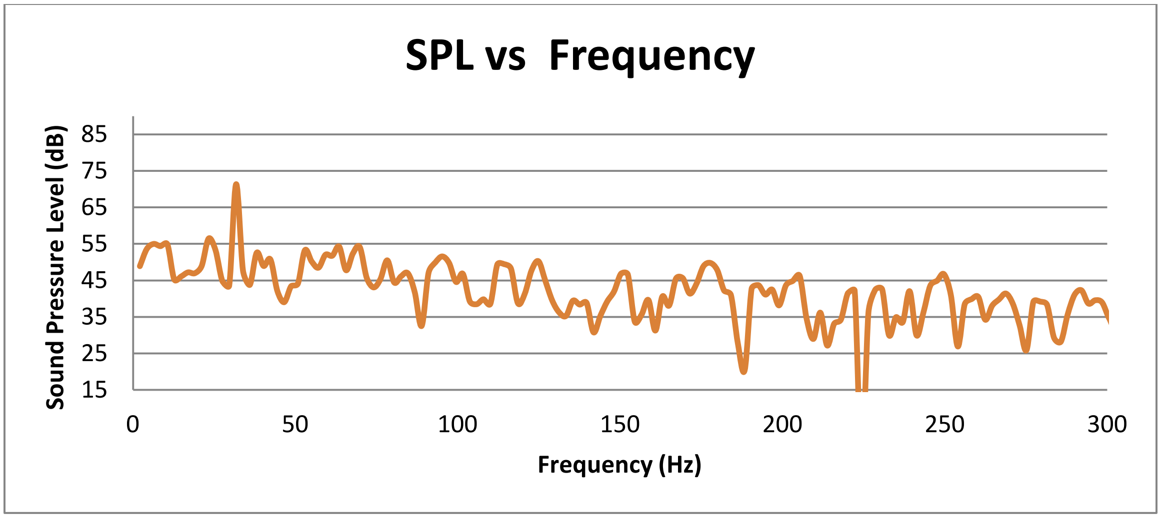

Figure 27 SPL at Bluff Body Distance 6.0 D

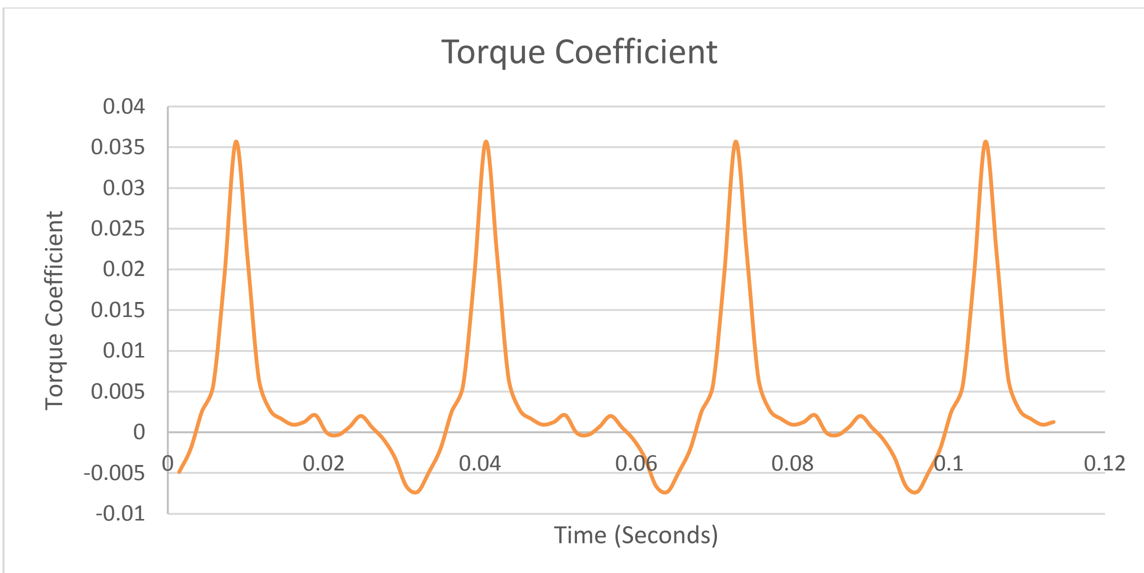

Figure 28 Torque Coefficient at Bluff Body Distance 6.0 D

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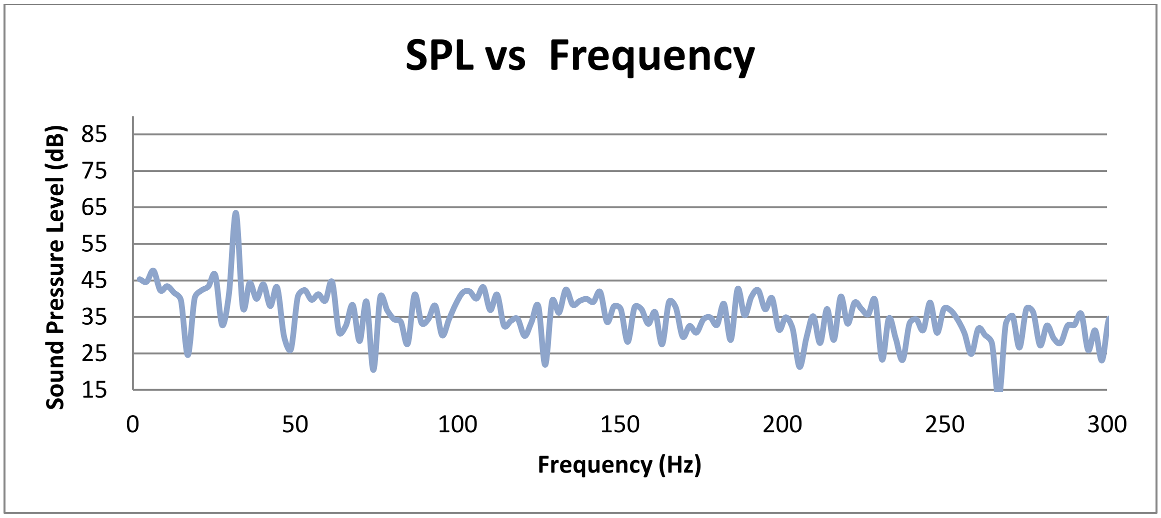

Figure 29 SPL at Bluff Body Distance 8.0 D

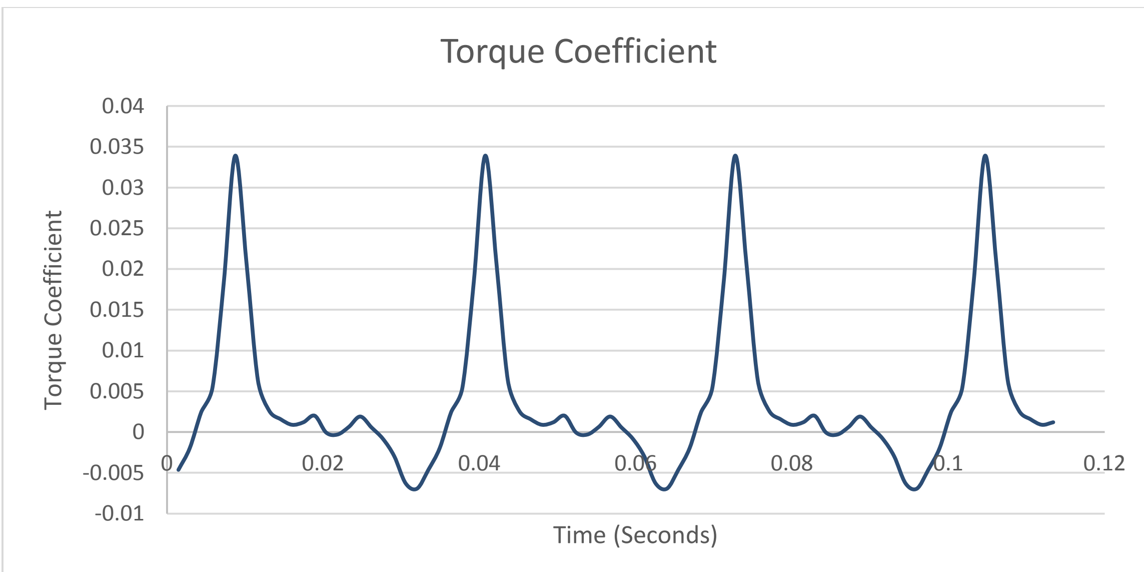

Figure 30 Torque Coefficient at Bluff Body Distance 8.0 D

4.2.1 Overall Trend

From the graphs shown below, as the bluff body distance from turbine center reached 8 times the turbine diameter, noise levels became the same as they were when placed in a free stream as shown in Figure 11. Furthermore, since air flow was available to turbine, no significant decrease in torque coefficient was observed. This was one of the most important finding of this research so far.

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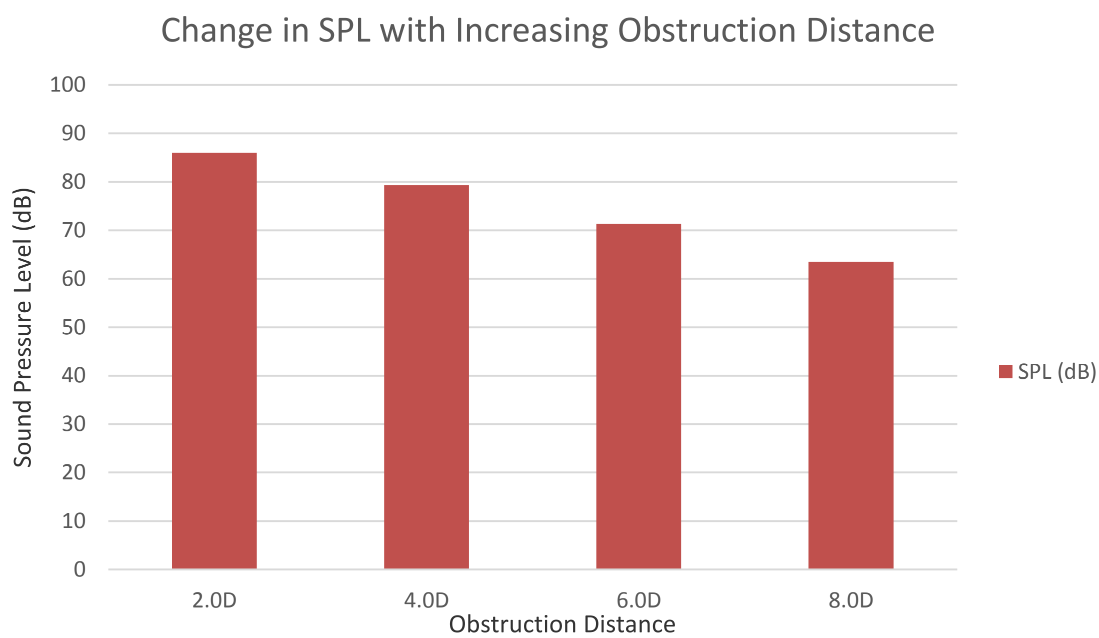

Figure 31 Overall Effect of Bluff Body Distance on SPL

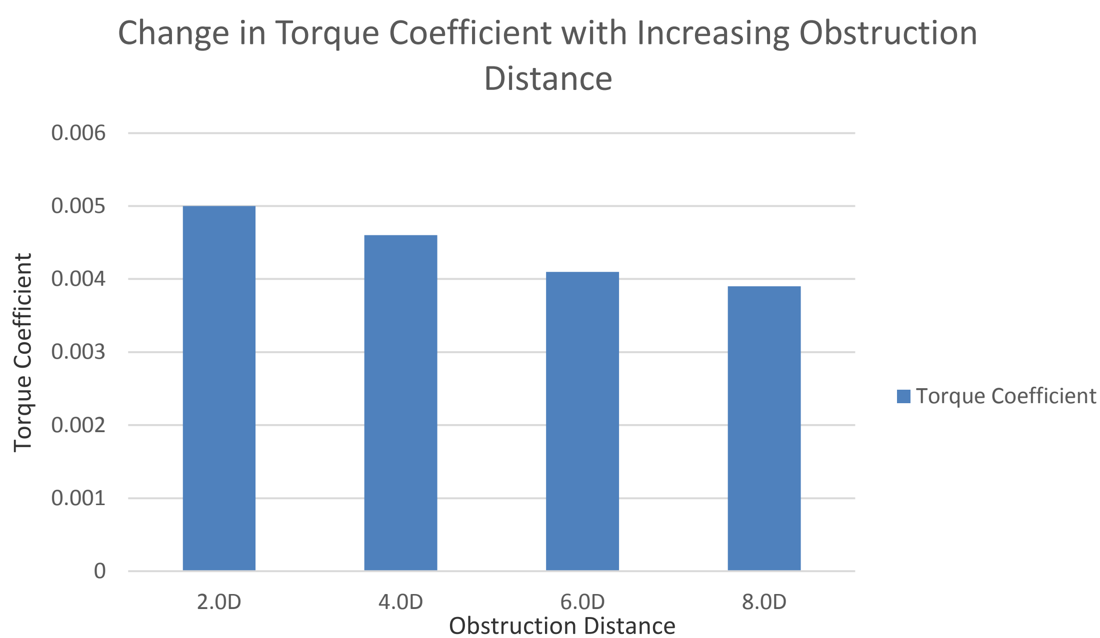

Figure 32 Overall Effect of Bluff Body Distance on Torque Coefficient

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4.3 Effect of Tip Speed Ratio on Sound Pressure Level and Torque Coefficient

Based on wind speeds and turbine designs, various rotational speeds (tip speed ratios) of VAWTs can be selected, therefore, 3rd parameter chosen for the parametric study was tip speed ratio. It had 4 levels i.e. 2.3, 1.6, 1.2, & 0.4. These tip speed ratios corresponded to rotational speeds of 625 rpm, 400 rpm, 300 rpm and 100 rpm respectively. Effect of these variations on turbine power output and noise generated are shown in the graphs below.

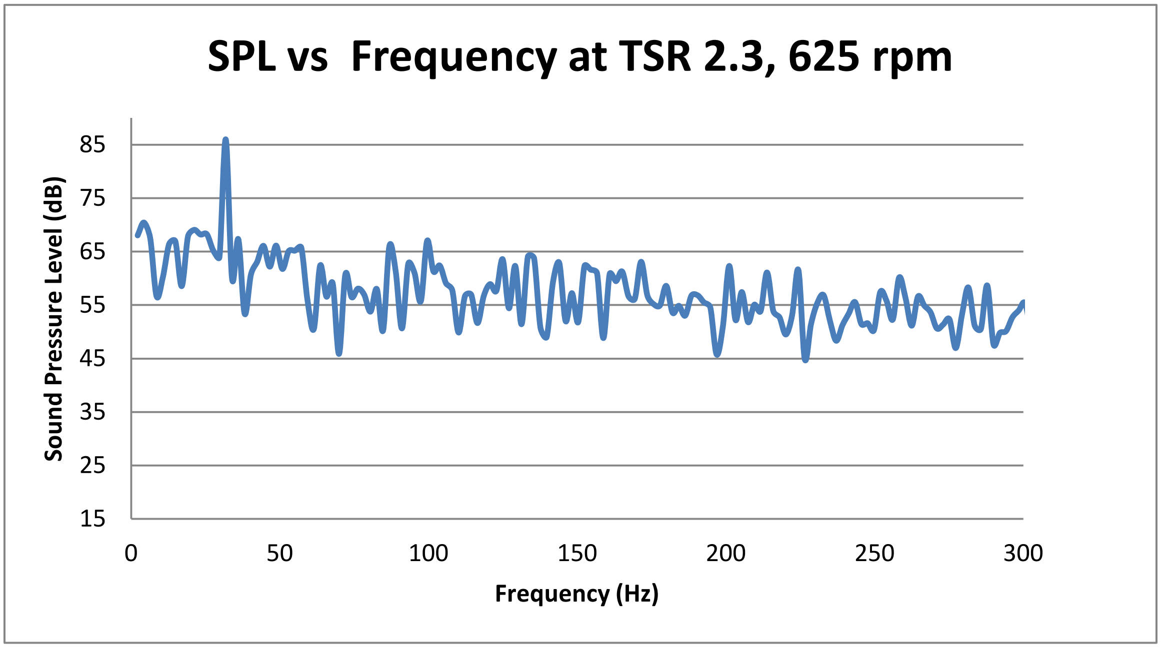

Figure 33 SPL at TSR 2.3

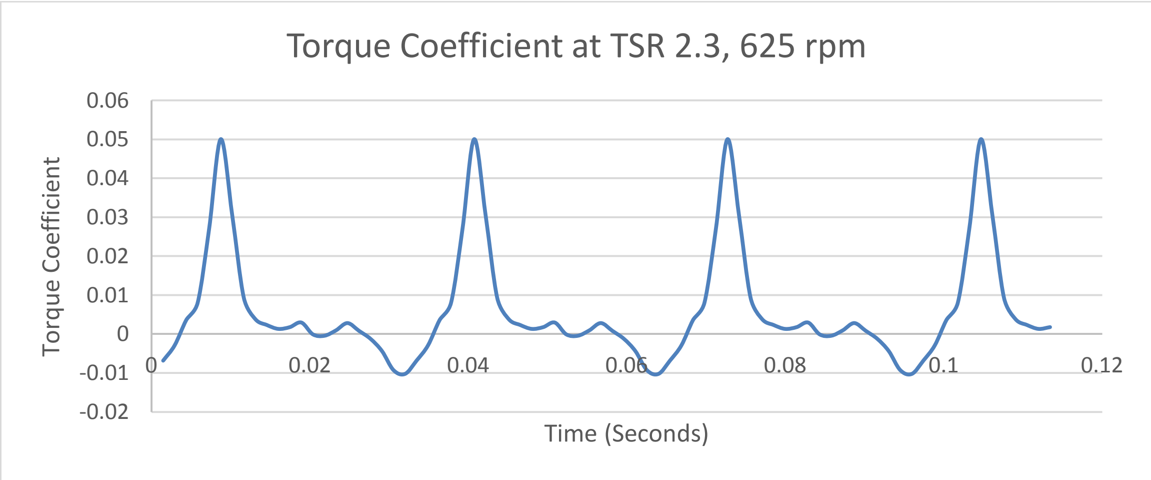

Figure 34 Torque Coefficient at TSR 2.3

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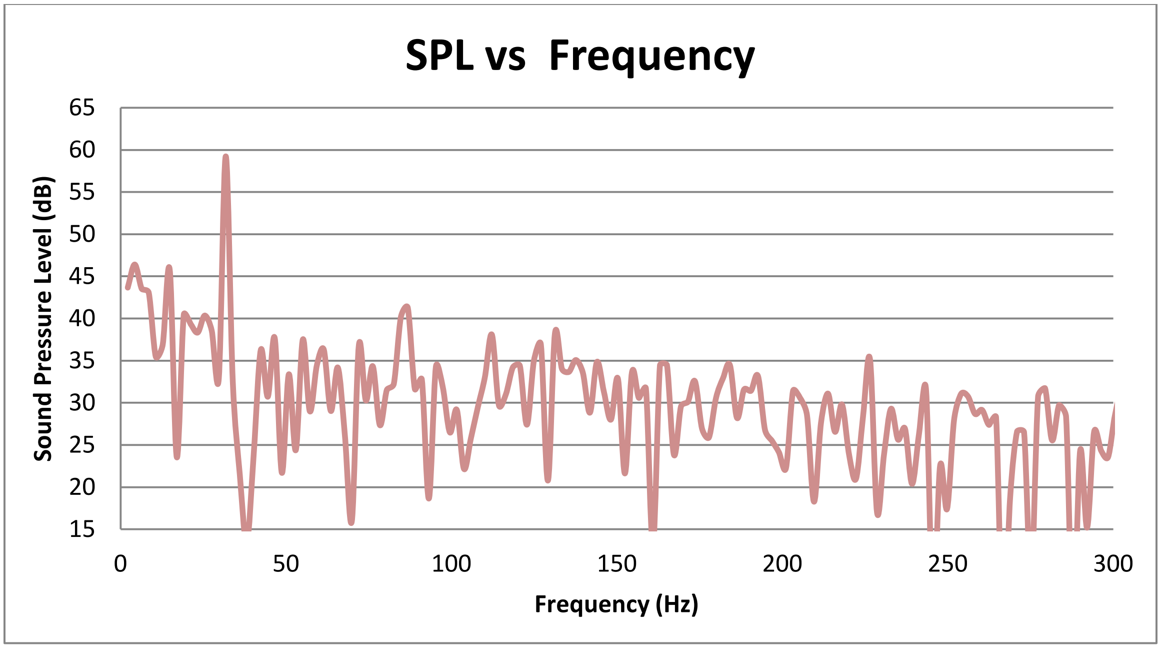

Figure 35 SPL at TSR 1.6

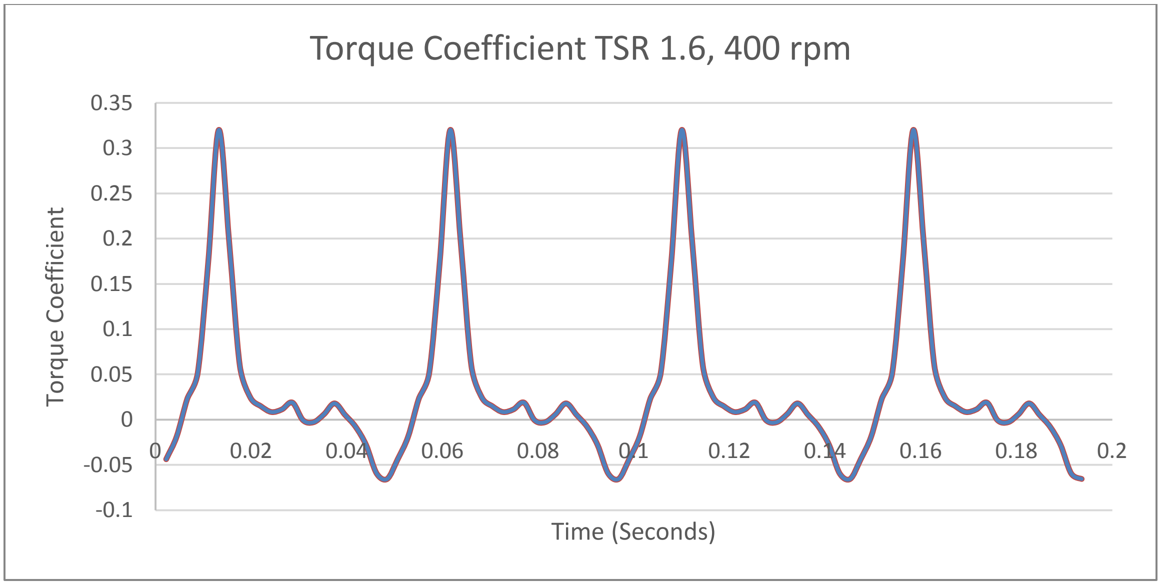

Figure 36 Torque Coefficient at TSR 1.6

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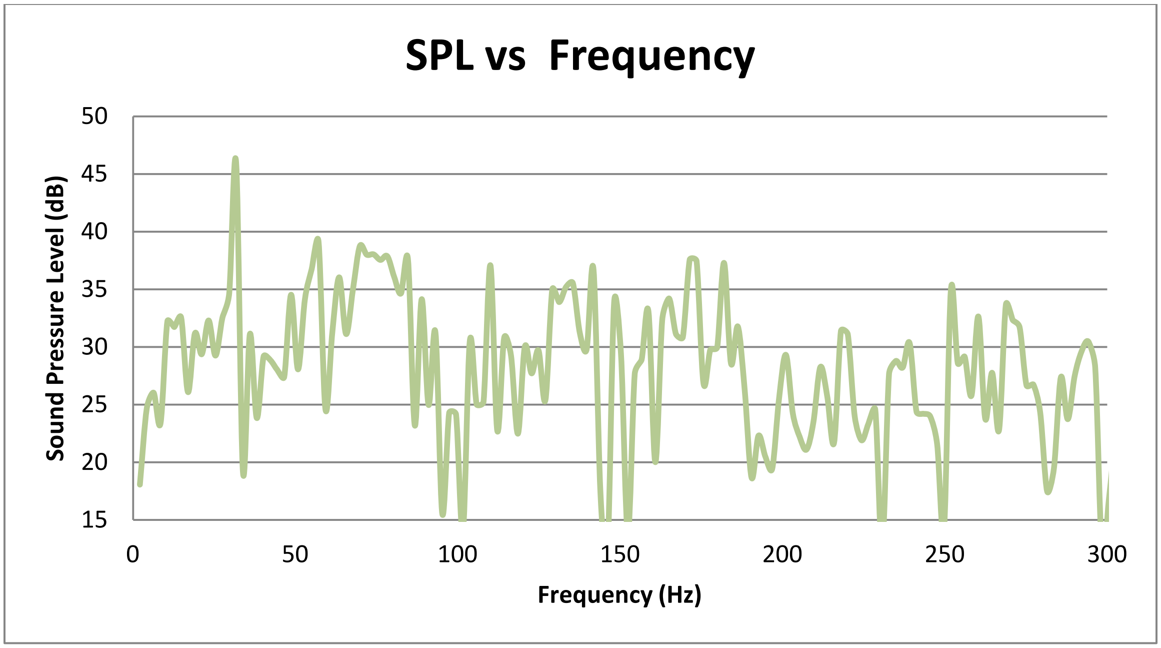

Figure 37 SPL at TSR 1.2

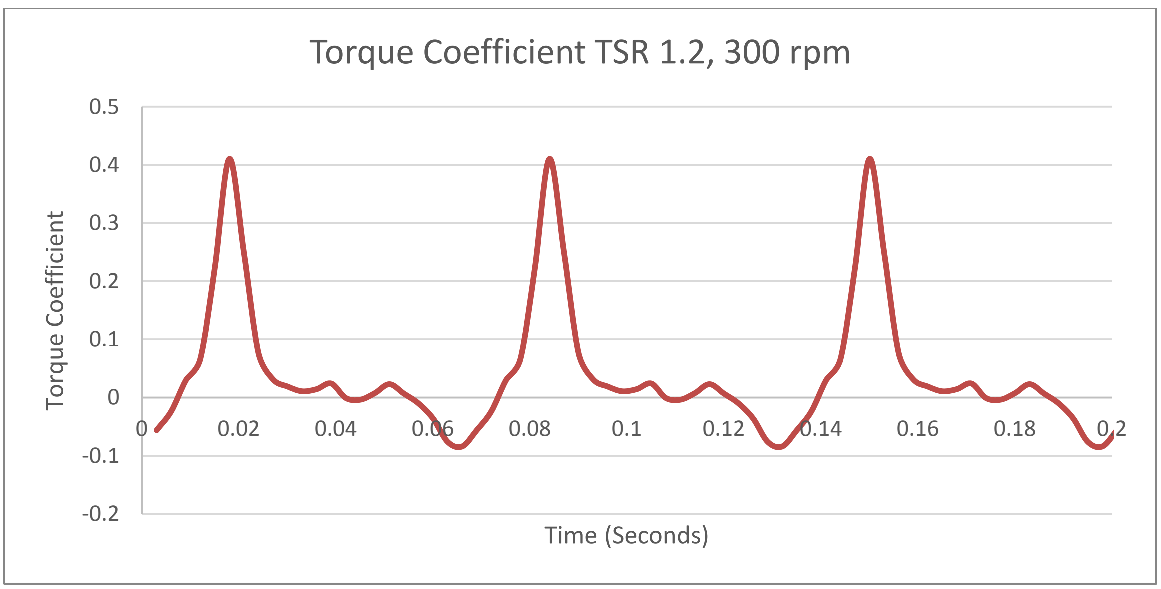

Figure 38 Torque Coefficient at TSR 1.2

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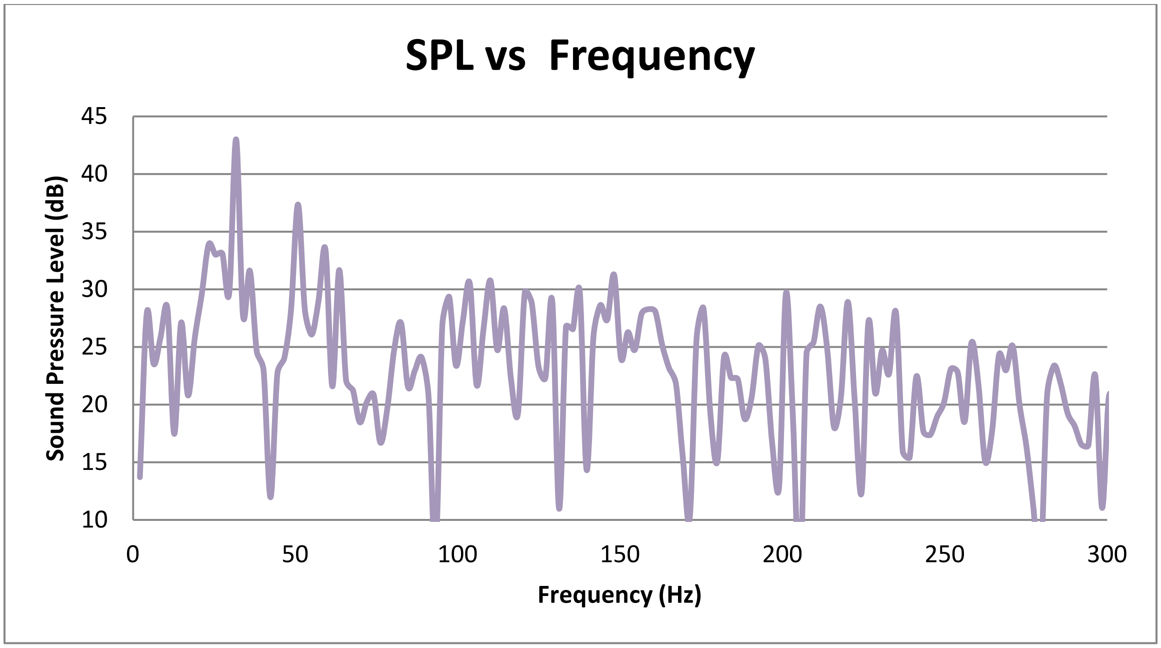

Figure 39 SPL at TSR 0.4

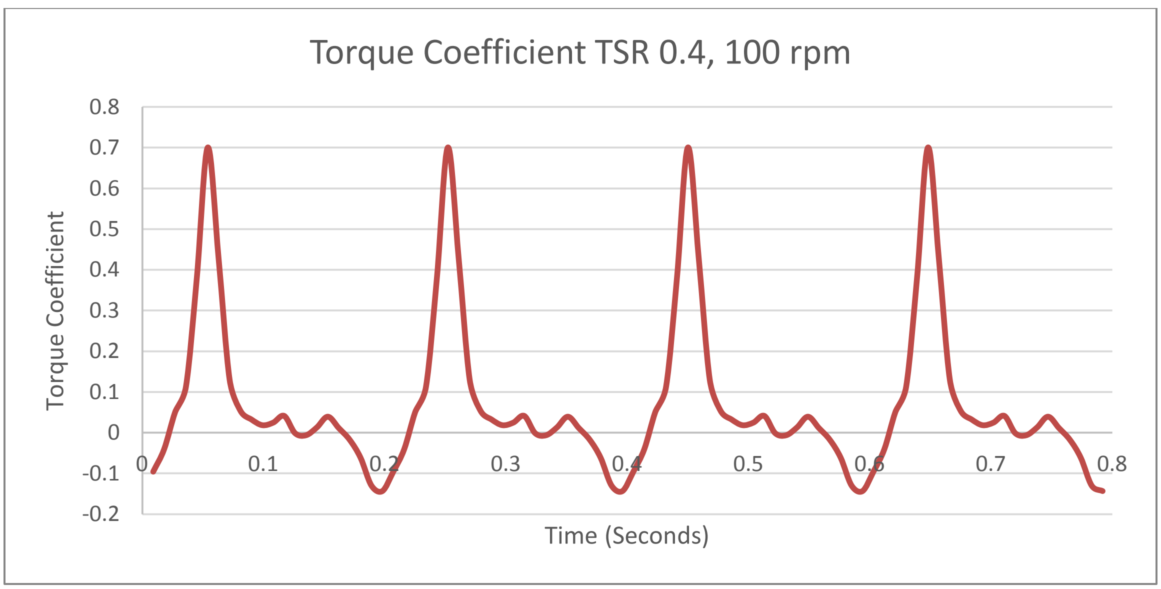

Figure 40 Torque Coefficient at TSR 0.4

4.3.1 Overall Trend

Noise levels are directly proportional to rotational speed of the turbine because with increased rotational speed, instantaneous pressure fluctuations, responsible for noise generation, increase. However, since at lower speed, flow separation is quite low, so better torque coefficients are achieved at lower rpms. So it’s always a tradeoff to choose between lower noise levels and

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higher power output.

Chapter 5 Conclusions

This study was designed to learn the performance behavior of a VAWT, considering both the aspects of aerodynamic as well as acoustics. A test case was first solved validated using published literature and experimental work. Afterwards, total 10 different scenarios or test cases were planned in which bluff body size, bluff body distance from turbine center, and rotating speed of VAWT was varied. These 3 parameters were selected to have an idea about turbine performance when installed in some urban environment.

When bluff body size was varied from 0.5 times the turbine diameter to 2 times the turbine diameter, noise level lowered by approximately 9% but torque coefficient of the turbine decreased by almost 5 times. In this case, noise reduction was achieved but at a very high cost of available torque. Similarly, by varying bluff body distance from turbine center from 2 times the turbine diameter to 8 times the turbine diameter, 26% reduction in noise and 18% reduction in torque coefficient was observed. It was also noted that when the distance between the bluff body and turbine center reached 8 times the turbine diameter, noise levels reached the value as if the turbine was placed in free stream. In other words, at that position of turbine, the acoustic effects of bluff body were nullified.

In the end, rotational speed of the turbine was varied from 625 rpm to 100 rpm and it was found that noise generated was directly proportional to rotational speed of the turbine whereas torque generated by the turbine at lower rpms was greater than the torque generated at higher rpms.

To sum it up, for a turbine, to have optimum aerodynamic as well as aeroacoustic performance, bluff body distance from turbine center should be at least 8 times the turbine diameter, bluff body size should not be increased to more than 4 times to that of turbine diameter, and rotational speed to kept as low as possible.

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