Terms and Conditions of Use of Digitised Theses from Trinity College Library Dublin

Copyright statement

All material supplied by Trinity College Library is protected by copyright (under the Copyright and Related Rights Act, 2000 as amended) and other relevant Intellectual Property Rights. By accessing and using a Digitised Thesis from Trinity College Library you acknowledge that all Intellectual Property Rights in any Works supplied are the sole and exclusive property of the copyright and/or other IPR holder. Specific copyright holders may not be explicitly identified. Use of materials from other sources within a thesis should not be construed as a claim over them.

A non-exclusive, non-transferable licence is hereby granted to those using or reproducing, in whole or in part, the material for valid purposes, providing the copyright owners are acknowledged using the normal conventions. Where specific permission to use material is required, this is identified and such permission must be sought from the copyright holder or agency cited.

Liability statement

By using a Digitised Thesis, I accept that Trinity College Dublin bears no legal responsibility for the accuracy, legality or comprehensiveness of materials contained within the thesis, and that Trinity College Dublin accepts no liability for indirect, consequential, or incidental, damages or losses arising from use of the thesis for whatever reason. Information located in a thesis may be subject to specific use constraints, details of which may not be explicitly described. It is the responsibility of potential and actual users to be aware of such constraints and to abide by them. By making use of material from a digitised thesis, you accept these copyright and disclaimer provisions. Where it is brought to the attention of Trinity College Library that there may be a breach of copyright or other restraint, it is the policy to withdraw or take down access to a thesis while the issue is being resolved.

Access Agreement

By using a Digitised Thesis from Trinity College Library you are bound by the following Terms & Conditions. Please read them carefully.

I have read and I understand the following statement: All material supplied via a Digitised Thesis from Trinity College Library is protected by copyright and other intellectual property rights, and duplication or sale of all or part of any of a thesis is not permitted, except that material may be duplicated by you for your research use or for educational purposes in electronic or print form providing the copyright owners are acknowledged using the normal conventions. You must obtain permission for any other use. Electronic or print copies may not be offered, whether for sale or otherwise to anyone. This copy has been supplied on the understanding that it is copyright material and that no quotation from the thesis may be published without proper acknowledgement.
Vibration Control of Offshore Wind Turbines using Tuned Liquid Column Dampers
VIBRATION CONTROL OF OFFSHORE WIND TURBINES USING TUNED LIQUID COLUMN DAMPERS

by

Shane Colwell

Thesis submitted to the University of Dublin, Trinity College, for the Degree Doctor of Philosophy

July 2007
DECLARATION

The author hereby declares that this thesis, in whole or in part, has not been submitted to any other University as an exercise for a degree. Except where reference has been given in the text, it is entirely the author’s own work.

The author confirms that the Library may lend or copy this thesis upon request, for academic purposes.

[Signature]

Shane Colwell

July 2007
SUMMARY

This study investigates the vibration control of offshore wind turbines with passive dampers. Investigations into the Tuned Liquid Column Damper (TLCD) are carried out. The TLCD, along with additional passive dampers are implemented in an offshore wind turbine model with rotating blades subjected to a combined wind and wave loading and the results are presented.

Theoretical and experimental investigations into the TLCDs influence in reducing the vibration response of a wind turbine model are carried out. A closed form solution for the transfer function of the maximum displacement of the structure, modelled as a single-degree-of-freedom (SDOF) system, with applied harmonic base motion, is presented. It is observed that changes in the orifice size in the pipe have a direct impact on the damping characteristics of the damper. Theoretical and experimental investigations into the passive damping properties of various fluids, including magnetorheological (MR) fluid, in the TLCD are undertaken. The coefficient of head loss for different fluids used in TLCDs is investigated. By implementing both water and glycol in the TLCD, one achieves significant structural response reduction to that of the undamped structure. Glycol or glycol/water mixtures are viable for structures were freezing temperatures might abound. In terms of volumetric efficiency, MR fluid is almost as effective as water within the TLCD in passively damping structural responses on the top of a SDOF structure. It is observed both experimentally and theoretically that an appropriately low viscous MR fluid may be implemented in MR-TLCDs, thus making MR-TLCDs practical for applications in wind turbines.

The structural responses of offshore wind turbines with attached TLCD are simulated. Through modelling an offshore wind turbine as a MDOF system with rotating blades under wind and wave excitations, an accurate dynamic portrait of a complex slender structure is presented. The WTT assembly was discretized into ultimately an eighteen DOF system, with the tower and dampers incorporating nine DOFs and the three blades each represented by three DOFs. The wind excitation is represented by the Kaimal spectrum. Spatial correlation is accounted for by coherence function and
simulating the excitation in the time domain. Wave excitations were generated using the JONSWOP spectrum and then simulated in the time domain. A correlation between the wind and wave excitations using joint distribution is possible through integrating the mean wind speed with formation of the JONSWAP spectrum. When an offshore wind turbine is equipped with a TLCD and subjected to wind and wave forces, reductions of up to 55% in the peak response of the system without TLCD is achieved. The TLCD performs consistently well, in terms of reducing base bending moments and peak responses in the structure, across all excitations considered. The option open to design the wind turbine more efficiently with less steelwork and less foundation expenses is thus afforded when one implements TLCDs in offshore wind turbines. It was also observed through use of the rain-flow calculation method for fatigue, that implementation of a TLCD in an offshore wind turbine greatly increases the fatigue life of the wind tower assembly.

A hybrid damping system incorporating a base isolation (BI) system at the platform, a TMD at the BI level for increased lateral stiffness and a TLCD at the nacelle of the offshore wind turbine is investigated. The offshore wind turbine tower with hybrid damping scheme was subjected to a combined wind, wave and rotating blade excitation. A parametric investigation on the hybrid damping scheme was undertaken to both understand the interaction of the various dampers and for optimisation of the overall damping performance. In conjunction with the hybrid damping scheme, an additional method of energy generation derived from the vibrating TMD is proposed and explored. With offshore wind turbine fitted with the BI-TMD-TLCD damping system, substantial reductions in displacement and acceleration response are attained. In one case, when the WTT is subjected to fluctuating wind and wave loadings deriving from a mean wind speed of 30 m/s, the peak acceleration and displacement response are reduced from the undamped WTT by 72% and 68%, respectively. The mass ratio of the BI system has a large impact on the performance of the damping system. The mass of the BI system, and hence stiffness, must be such as to prevent substantial lateral displacement responses at the platform level. Additional energy may be generated from the vibration of the TMD, which ultimately derives from the wind and wave loadings exciting the offshore wind turbine. It was found that under operational conditions, an additional 13.1 kWhs of energy are produced from the WTT.
ACKNOWLEDGEMENTS

My work was carried out in the Department of Civil, Structural & Environmental Engineering at Trinity College, Dublin. This work was funded by the Irish Research Council for Science, Engineering and Technology.

This work could not have been completed without the help of the technical staff in the dynamics laboratory. In particular, thanks go to Gerard McGranaghan, Chris O'Donovan and Kevin Ryan for all their help during the experimental phases of the PhD.

My heartfelt thanks go to my research supervisor, Professor Biswajit Basu. Without his general guidance, intelligence and diligence this work would not have been possible. It is the measure of the man that I not only see him as a mentor, but also a friend.

I would also like to thank all the friends whom have helped me throughout my life. These friends come from a plethora of places that I have shaped me throughout my life. Friends from the places which have contributed most to my education: Kilnamanagh, Drimnagh Castle and Trinity. Friends from my time spent living in New York, Switzerland and Berlin. Friends I have met on my meandering travels across the world. Each one of them has contributed something to my development as a person that will never be forgotten.

Above all, this Doctorate is dedicated to my family, without whom anything I have achieved to date has been possible. From my Grandmother, who whispered in my ear when I was a very young boy that one day I would get a PhD from Trinity, to the incredible support of my brother Derek and finally to the incredible love emanating from my parents, Anne and Shay Colwell.

It was once said that “When a person really desires something, all the universe conspires to help that person to realize his dream.” Thus, I thank everybody who has helped me in realizing this Thesis.
TABLE OF CONTENTS

DECLARATION ........................................................................................................ ii
SUMMARY ............................................................................................................... iii
ACKNOWLEDGEMENTS ....................................................................................... vi
TABLE OF CONTENTS ......................................................................................... vii
LIST OF FIGURES .............................................................................................. xi
LIST OF TABLES ................................................................................................. xvi

CHAPTER 1 - INTRODUCTION AND LITERATURE REVIEW .......................... 1
  1.1 INTRODUCTION AND MOTIVATION ......................................................... 1
  1.2 REVIEW OF LITERATURE AND BACKGROUND ..................................... 3
    1.2.1 Wind Turbine Towers .......................................................................... 3
      1.2.1.1 The rotor ......................................................................................... 5
      1.2.1.2 The nacelle ..................................................................................... 5
      1.2.1.3 The Tower and foundations ........................................................... 6
      1.2.1.4 Turbine and foundation design for offshore .................................... 6
      1.2.1.5 Dynamics of Offshore Wind Turbines ............................................. 7
    1.2.2 Vibration Control .................................................................................. 8
      1.2.2.1 Tuned Mass Dampers .................................................................... 8
      1.2.2.2 Tuned Liquid Column Dampers ..................................................... 10
      1.2.2.3 Base Isolation Systems and combined damper systems ............... 15
      1.2.2.4 Magnetorheological Fluids ............................................................. 17
    1.2.3 Wind Loading ....................................................................................... 18
    1.2.4 Wave Loading ...................................................................................... 20
    1.2.5 Offshore Wind Turbine Reliability ...................................................... 22
  1.3 ORGANISATION OF THE THESIS ............................................................. 24

CHAPTER 2 - INVESTIGATIONS ON THE PERFORMANCE OF A TLCD WITH DIFFERENT ORIFICE DIAMETER RATIOS ................................................. 26
5.6.1 Wind and Wave Excitation ................................................................. 123
  5.6.1.1 Wind and Wave Excitation with mean wind speed = 30 m/s .......... 124
  5.6.1.2 Wind and Wave Excitation with mean wind speed = 18 m/s .......... 128
5.6.2 WTT response with BI system, TMD and TLCD ............................... 131
  5.6.2.1 WTT with BI system, TMD and TLCD under 30 m/s wind and wave
loading ....................................................................................................... 133
  5.6.2.2 WTT with BI system, TMD and TLCD under 18 m/s wind and wave
loading .......................................................................................................140
5.6.3 Parametric Investigation ................................................................. 146
5.6.4 Harnessing of the TMDs Energy ...................................................... 150
  5.6.4.1 Mass ratio of BI system = 10% ....................................................... 154
  5.6.4.2 Mass ratio of BI system = 5% ......................................................... 154
5.7 CONCLUSIONS .................................................................................... 156

CHAPTER 6 - INCREASED AVAILABILITY OF OFFSHORE WIND
TURBINES WITH TLCD DUE TO WIND AND WAVE INDUCED
ACCELERATIONS......................................................................................... 157
  6.1 INTRODUCTION................................................................................ 157
  6.2 SHUT DOWN CRITERIA .................................................................... 157
  6.3 WIND TURBINE FRAGILITY ANALYSIS ......................................... 158
  6.4 WIND TURBINE ECONOMICS ......................................................... 165
  6.5 ANALYSIS OF RESULTS .................................................................. 166
  6.6 ECONOMIC EVALUATION ............................................................... 172
  6.7 CONCLUSIONS ................................................................................ 172

CHAPTER 7 - EXECUTIVE SUMMARY AND CONCLUSIONS .............. 174
  7.1 EXECUTIVE SUMMARY ................................................................. 174
  7.2 CONCLUSIONS ................................................................................ 177
  7.3 RECOMMENDATIONS FOR FURTHER STUDY ............................... 180
LIST OF FIGURES

Figure 1.1  Major components of a horizontal axis wind turbine
Figure 1.2  TMD
Figure 1.3  One Wall Centre (Glotman Simpson Consultant Engineers, 2001)
Figure 2.1  Experimental Set-up
Figure 2.2  Schematic depicting the experimental set-up
Figure 2.3  The interaction between the TLCD and the Structure
Figure 2.4  Test for linearity with fitted line using least squares method
Figure 2.5  DAF versus frequency for base amplitude 3.75mm and orifice
diameter 1d
Figure 2.6  DAF versus frequency for base amplitude 3.75mm and orifice
diameter 0.75d
Figure 2.7  DAF versus frequency for base amplitude 3.75mm and orifice
diameter 0.5d
Figure 2.8  DAF versus frequency for base amplitude 3.75mm and orifice
diameter 0.25d
Figure 2.9  DAF versus frequency for base amplitude 2.5mm and orifice
diameter 1d
Figure 2.10  DAF versus frequency for base amplitude 2.5mm and orifice
diameter 0.75d
Figure 2.11  DAF versus frequency for base amplitude 2.5mm and orifice
diameter 0.5d
Figure 2.12  DAF versus frequency for base amplitude 2.5mm and orifice
diameter 0.25
Figure 2.13  Orifice influence on SDOF structure with Base Amplitude = 3.75mm
Figure 2.14  Orifice influence on SDOF structure with Base Amplitude = 2.5mm
Figure 2.15  Optimum orifice diameter for SDOF structure under different
excitation frequencies for varying amplitude
Figure 2.16  Undamped response with base amplitude = 3.75mm
Figure 2.17  Undamped response with base amplitude = 2.5mm
Figure 2.18  Cross sections of different horizontal pipes
Figure 3.14 Variation of $C_h$ against frequency ratio for the different test cases for structure S1
Figure 3.15 Displacement transfer function of fluid, $H_o(\omega)$ against frequency ratio for the different test cases for structure S1
Figure 3.16 Variation of $\xi$ against frequency ratio for water and MR fluid cases for structure S1
Figure 3.17 Non linear simulated transfer function of water-structure S1 with constant $\xi$ (0.124) and with optimised values for $\xi$
Figure 3.18 Non linear simulated transfer function of MR fluid-structure S1 with constant $\xi$ (0.124) and with optimised values for $\xi$
Figure 4.1 Structural Model
Figure 4.2 Time series for wind excitation at Node 7
Figure 4.3 PSDF for the wind excitation
Figure 4.4 Time series for the ‘moderate’ wave excitation
Figure 4.5 PSDF of ‘moderate’ wave elevation
Figure 4.6 Wave force at node 3 for ‘moderate’ wave excitation
Figure 4.7 Time series for the ‘strong’ wave excitation
Figure 4.8 PSDF of ‘strong’ wave elevation
Figure 4.9 Wave force at node 3 for ‘strong’ wave excitation
Figure 4.10 Mode shapes of MDOF system
Figure 4.11 MDOF time-history response under wind excitation with and without TLCD
Figure 4.12 MDOF time-history response under wind and wave excitation with and without TLCD
Figure 4.13 MDOF, with rotating blades, time-history response under ‘moderate’ wind and wave excitation with and without TLCD
Figure 4.14 MDOF, with rotating blades, time-history response under ‘strong’ wind and wave excitation with and without TLCD
Figure 4.15 Base moment of MDOF, with rotating blades, time-history response under ‘strong’ wind and wave excitation with and without TLCD
Figure 4.16 Rainflow Calculation Method
Figure 5.1 Schematic of the structural system
Figure 5.2 Time series for wind excitation at Node 7
Figure 5.3  Time series for wind excitation at Node 8
Figure 5.4  Time series for wind excitation at Node 9
Figure 5.5  Simulated time history of sea surface elevation
Figure 5.6  Wave force at node 2 for 'strong' wave excitation
Figure 5.7  Wave force at node 3 for 'strong' wave excitation
Figure 5.8  Time series for wind excitation at Node 7
Figure 5.9  Time series for wind excitation at Node 8
Figure 5.10 Time series for wind excitation at Node 9
Figure 5.11 Simulated time history of sea surface elevation
Figure 5.12 Wave force at node 2 for 'operational' wave excitation
Figure 5.13 Wave force at node 3 for 'operational' wave excitation
Figure 5.14 Time-history response of peak response of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.15 Time-history response of peak acceleration of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.16 Time-history response of the TMD
Figure 5.17 Time-history response of the TLCD liquid
Figure 5.18 Time-history response of peak response of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.19 Time-history response of peak acceleration of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.20 Time-history response of the TMD
Figure 5.21 Time-history response of the TLCD liquid
Figure 5.22 Time-history response of peak response of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.23 Time-history response of peak acceleration of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.24 Time-history response of the TMD
Figure 5.25 Time-history response of the TLCD liquid
Figure 5.26 Time-history response of peak response of MDOF, with rotating blades, under wind and wave excitation with and without TLCD, TMD and BI system (5%)
Figure 5.27  Time-history response of peak acceleration of MDOF, with rotating blades, under wind and wave excitation with and without TLCD, TMD and BI system (5%)
Figure 5.28  Time-history response of the TMD
Figure 5.29  Time-history response of the TLCD liquid
Figure 5.30  Displacement response of the TMD with $\mu_{\text{Bi}} = 10\%$
Figure 5.31  Velocity response of TMD with $\mu_{\text{Bi}} = 10\%$
Figure 5.32  Instantaneous Energy produced in TMD with $\mu_{\text{Bi}} = 10\%$
Figure 5.33  Cumulative Energy produced in TMD with $\mu_{\text{Bi}} = 10\%$
Figure 5.34  Displacement response of the TMD with $\mu_{\text{Bi}} = 5\%$
Figure 5.35  Velocity response of the TMD with $\mu_{\text{Bi}} = 5\%$
Figure 5.36  Cumulative Energy produced in TMD with $\mu_{\text{Bi}} = 5\%$
Figure 6.1  Map of wind speed extrapolated to 80 m and averaged over all days of the year for Europe (Archer and Jacobson, 2005)
Figure 6.2  Map of wind speed extrapolated to 80 m and averaged over all days of the year for North America (Archer and Jacobson, 2005)
Figure 6.3  Typical set of fragility curves of building structures (Leon and Atanasiu, 2007)
Figure 6.4  Acceleration demand at the nacelle of undamped wind turbine
Figure 6.5  Acceleration demand at the nacelle of wind turbine with TLCD
Figure 6.6  Displacement demand at the nacelle of undamped wind turbine
Figure 6.7  Displacement demand at the nacelle of wind turbine with TLCD
Figure 6.8  Wind-induced acceleration fragility curves for the WTT
Figure 6.9  Wind-induced acceleration fragility curves for the WTT with TLCD
LIST OF TABLES

Table 2.1  Equivalent damping ratios for the various test cases
Table 3.1  Summary of test cases and damping results for structure S1
Table 3.2  Summary of test cases and damping results for structure S2
Table 4.1  Degree of fatigue damage for wind turbine assembly with and without TLCD
Table 5.1  Variation of tuning of TMD and BI
Table 5.2  Variation of mass ratio of the TMD
Table 5.3  Variation of tuning of mass ratio of BI system (mean wind speed = 30m/s)
Table 5.4  Variation of tuning of mass ratio of BI system (mean wind speed = 18m/s)
Table 6.1  Distributions and parameters for modelling wind turbine response variability
Table 6.2  Mean wind speed acceleration thresholds
Table 6.3  Parameters in calculation of fragility curves
CHAPTER 1
INTRODUCTION AND LITERATURE REVIEW

1.1 INTRODUCTION AND MOTIVATION

In these uncertain times of oil supply, unrest in energy supplying nations and global
warming, the world is increasingly turning to renewable sources to feed its
overwhelming appetite for energy. Ireland is ranked 6th in the EU-25 in terms of oil
dependant electricity generation and as our natural oil deposits are practically zero,
we are one of the most imported-oil reliant nations on Earth, leaving us at the mercy
of fluctuating oil prices. As the economies of China and India expand, the magnitude
of Ireland’s unsustainable dependence on imported oil is increased even further. In
addition, the Kyoto summit held in 1997 decreed that global production of green
house gases produced by developed countries by reduced by approximately 5% below
1990 levels by 2008-2012. In the context of Ireland, the government has pledged to
limit the increase of greenhouse gas emissions to 13% above 1990 levels by 2008-
2012. In Britain’s case, to meet its targets of producing 20% of all energy from
renewable sources by 2020, a task force set up has acknowledged that offshore wind
energy will be the primary driver in meeting the target. Thus, alternative methods of
renewable energy production in order to feed our energy habits must be found.

Various methods of energy production exist, from nuclear power to solar power.
However, the case for offshore wind energy, above onshore wind energy and also
other types of energy production, appears to be the strongest. Many people stereotype
wind turbines as being destroyers of an area’s scenic beauty and generators of noise
pollution. Wind farms placed in the ocean can quickly overcome these factors as vast
farms placed in the sea are only eyesores to lonely fishermen and oil tanker captains.
The main environmental issues, such as visual impact on coastal landscapes, conflicts
with the shipping industry and adverse changes in the local marine life, are quite
minute compared with the ecological footprints left by such industries as nuclear
power and coal burning power plants. It has been found that fish stocks have actually
increased since the construction of Danish offshore farms.
In the sea, due to the relative flatness of the topography, wind speeds are greater than on dry land. The energy content of the wind is cubed with each metre per second increase in a wind's speed. The higher one goes in the atmosphere, the higher the wind speeds. Thus, by constructing extremely tall wind turbines in the ocean, of the order of 100m above sea level, vast amounts of wind energy can be harnessed. With immense tracts of sea available for wind turbine installation, vast wind farms can be constructed, creating greater economies of scale and more efficient use of invested money. Also, economically optimised turbines yield some 50% more energy at sea than at nearby land locations. Hence, with our vast coastline, Ireland is potentially sitting on a treasure trove of renewable energy.

The total European consumption of power today is 3,030 TWh/year (2006) (European Wind Energy Association, 2006). The projected potential of wind energy available in Europe, with turbine farms located within 30km of the coast, is 3,028 TWh/year. It has been estimated that the total European potential for currently feasible sites is some 960 TWh/year (BTM Consult ApS, 1999). The projected potential for offshore wind energy in Ireland is 54.8 TWh/year. With projected energy consumption in Ireland in 2010 to be 44TW/h/year, it is clear that utilisation of our vast coastal wind energy can have a great impact on Ireland's continued development as a nation. With nearly every European country setting minimum wind energy targets, it is logical to assume that the future market for wind turbine production is immense. To exploit this market, engineers must ensure that wind turbines are technologically optimised structurally, mechanically and electrically to provide an efficient means of converting the wind's energy to electrical energy.

The aim of this thesis is to investigate the applicability of dampers in offshore wind turbines for the mitigation of vibrations caused by wind and wave excitations. Tuned Liquid Column Dampers (TLCDs) are investigated, with particular emphasis on the orifice damping, through theoretical and experimental studies. The use of various liquids in the TLCD is investigated experimentally and theoretically. A model that excites an offshore structure through coupled wind and wave loading is developed. The implementation of a TLCD in an offshore wind turbine is theoretically investigated. A new wind turbine model incorporating base isolation, a Tuned Mass Damper (TMD) and a TLCD is proposed and investigated. Finally, the long term
reliability effects of implementing a TLCD in an offshore wind turbine is investigated through fragility curves.

1.2 REVIEW OF LITERATURE AND BACKGROUND

1.2.1 Wind Turbine Towers

A wind turbine is a machine that converts the kinetic energy of the wind into mechanical power. Wind turbines / windmills have existed in one form or another since ancient times, where the mechanical power was used to grind grain to produce flour. The first windmills on record were built by the Persians in approximately 900 AD. By the mid-1950’s, the advent of various technologies that convert fossil fuels to practical forms of energy seemed to have relegated wind turbines into history as a means of power generation. By the late 1960’s, however, the use of the wind turbine as a viable means of modern energy conversion was in the early throes of an exponential rise. Forward to 2007, and wind turbines represent a genuine alternative to fossil fuelled based methods of power generation.

To understand the rise of wind turbines, it is necessary to consider three main factors which were inherent in the rise of wind energy. The first of all is need. A need for alternative methods of energy production that are independent to the finiteness of the earths fossil fuel reserves and the unseemly environmental repercussions that these methods accrue over time. The need for wind energy encouraged the political will for the support of wind energy, which became tangible support in various forms such as tax incentives. Second, there was the potential of wind energy. Wind resources exist everywhere on earth, with some countries in particular enjoying abundant resources. Thirdly, there has been an explosion in research into wind turbines. Factors such as a deeper understanding of the wind, improvements in energy conversion systems and increased understanding of the structural dynamics of wind turbines have allowed wind turbine to become taller, require less materials, become more adaptable to wind conditions and overall, more efficient.
Thus, the capacity of wind turbines has increased greatly over the years, from 10 kW to turbine towers today that can generate up to 5MW. To develop wind turbines of this magnitude, the horizontal-axis wind turbine (HAWT) has evolved into a specific format. HAWT rotors are usually classified according to the rotor orientation (upwind or downwind of the tower), rotor control (pitch versus stall), number of blades (usually two or three), hub design (rigid or teetering), and how they are aligned with the wind (free yaw or active yaw). The principal components of a typical HAWT are shown in Figure 1.1.1.

Figure 1.1 Major components of a horizontal axis wind turbine
1.2.1.1 The rotor

The rotor consists of the hub and blades of the wind turbine. The predominant configuration used today is a turbine with upwind rotors and three blades. The main reason for using three blades is that when all three blades are undergoing vibration, there is theoretically a zero bending moment at the hub. The blades are long, slender members that extend radially from the central hub. Both the hub and the blades are generally made from glass fibre reinforced plastic (GRP), but sometimes wood/epoxy laminates are also used.

1.2.1.2 The nacelle

The nacelle houses all the electrical and mechanical equipment inside a protective casing. The drive train consists of the rotating parts of the wind turbine. This generally includes a low-speed shaft, a gearbox, and a high speed shaft. The rotational energy from the blades is transferred into the gearbox, where the initial rotational frequency (in the order of two to three hertz for a typical MW class turbine) is increased by a factor of approximately thirty to fifty. For a typical 1MW class generator, the blades will rotate in the range 20-30 revolutions per minute (rpm) (US Department of Energy, 2006). The generator then converts this mechanical energy into electrical energy. The generator, usually induction or synchronous generators, generally require a near-constant rotational speed. Variable speed generators are being increasingly used. The main advantages of variable speed generators are reduced wear and tear on the wind turbine and increased energy capture over a wider range of wind speeds.

The nacelle also contains a yaw orientation system, which is required to turn the wind turbine rotor against the wind. If the rotor system is not perpendicular to the wind, a yaw error is said to have arisen. When this happens, one part of the blade system is subject to a larger loading than the others as it closer to the source direction of the wind main wind input than the others blades. The blades will bend back and forth in a flapwise direction for each turn of the rotor, subjecting the rotor system to increased fatigue loads. When wind speeds become too high for the respective wind turbine
system, the yaw control system will deflect the rotor system away from the direction of the wind. This prevents excessive loads in the gear box and vibrations in the nacelle but also translates to a loss in operational efficiency.

1.2.1.3 The tower and foundations

The function of the tower is to place the rotor and nacelle systems in an elevated position subject to effectively high wind speeds for wind turbine operation. The tower transfers all net loads accrued from supporting the rotor and nacelle systems into the foundations. The principal types of tower design currently in use include lattice towers, concrete towers and more commonly, steel tube towers. A rule of thumb suggests that tower height is usually 1.5 times that of the rotor diameter. Typical foundation designs based on reinforced concrete are used for onshore wind turbines.

1.2.1.4 Turbine and foundation design for offshore

The case for offshore wind power is that the wind conditions generally are much better offshore than on land and also that there are no environmental constraints in terms of noise and visual pollution at play. The main conclusion of an indepth study into offshore wind (Danish Energy Agency and the IEA CADDET Renewable Energy Technologies Programme, 2000) is that the offshore wind technical potential in Europe is larger than the total energy consumption. As offshore turbines are subject to a harsher and more isolated environment than their onshore brethren, offshore turbines need to have higher reliability and lower maintenance requirements. Arrays of larger turbines can be placed offshore, which lowers the per MW infrastructure costs. Less concern about noise pollution allow higher tip speed ratio of the rotor, which is directly related to increases in noise pollution. A higher tip speed ratio reduces component sizes and costs, and hence improves turbine efficiencies.

Concrete foundation costs are approximately proportional to the water depth squared and tend to be unusable beyond water depths of 10m. For depths beyond 10m, steel
monopole foundations are preferred as when heavy-duty piling equipment is used to
drive them 10-20m into the sea bed, no seabed preparation is needed. Tripod
foundations are anchored to three steel piles driven 10-20m into the seabed. For
deeper waters, floating wind turbines using spar buoys have been proposed
(Heronemus, 1972). More complex floating pontoons that support multiple wind
turbines have also been proposed (Halfpenny et al., 1995). A floating design that
incorporates multiple, interconnected moorings from a large concrete shelled buoy to
the sea bed was investigated and found to be physically viable (Tong, 1998), albeit
relatively expensive.

1.2.1.5 Dynamics of Offshore Wind Turbines

When moving offshore, several factors must be taken into account when designing the
offshore turbine and the associated structures including foundations. Wind-excited
vibrations of structures induce fluctuating stresses around the mean deformation states
that lead to fatigue damage accumulation and can determine structural failure without
exceeding design wind actions (Repetto and Solari, 2001). These wind excited
vibrations must also be accounted for when designing the foundations of the turbine.
Reduction of vibration is also a good measure for a successful design of blade and
structure. The reduction of vibration will thus foster two very important design goals,
low cost and high stability, while maximizing energy production. A recent study
(European Wind Energy Association, 1998) concluded that if the natural frequency of
the wind turbine is low, dynamic magnification due to wave loading may be
important. Traditional ways of reducing vibration within the turbines have been to
separate the natural frequencies of the structure from wind gusts and also the
harmonics of the rotor speed. Negam and Maalawi (2000) considered an optimisation
strategy of maximizing the system natural frequencies and opted for higher natural
frequencies for reducing both the steady state and transient responses of the structure.
Other ways in which designers have designed, as recommended by codes such as
Eurocode 3, against the consequences of structural vibrations have been to use more
steel and stiffeners around the shell (Lavassas et al., 2003) and also to employ deeper
foundations. Among the conclusions drawn up by Halliday (2001) in a study on off-
shore wind turbines in Ireland, it was concluded that for fatigue loading calculation
the linear frequency domain approach is likely to be an appropriate method of analysis of combined wind and wave loading. This is because linear wave theory is adequate for the dominant part of the wave load spectrum. The wind and wave induced behaviour of off-shore structures has been investigated with a parametric study by Bisht and Jain (1998) and a first order reliability analysis by Ditlevsen (2002), using a Rosenblatt translation. However the use of dampers in controlling dynamically induced vibrations in offshore wind turbines from both wind and waves has not been investigated in detail by researchers so far.

1.2.2 Vibration Control

As research and investment forces improvements in high strength material technology and structural efficiency, structures are becoming lighter, more slender and more daring architecturally. These advances eliminate the traditional way in which engineers dealt with vibrations incurred within their structures: by using a combination of mass and stiffness. Consequently, the structural vibrations caused by wind, earthquakes and even people are more likely to lead to occupant discomfort, fatigue in the structure or structural failure. In addition to structural problems caused by vibrations, induced accelerations in acceleration sensitive equipment may cause the equipment to cease functioning. In the case of a wind turbine, this translates to ‘downtime’ in the availability of the acceleration sensitive equipment, which ultimately means reduced energy production. This leaves the engineer with the option of damping to suppress vibrations that are induced within the structural system.

1.2.2.1 Tuned Mass Dampers

A Tuned Mass Damper (TMD) is a type of vibration absorber, where an auxiliary mass, M, is connected to the main structure through a spring, K and a dashpot, C (Figure 1.2), such that it oscillates at the same frequency as the structure but with a phase shift. TMDs are tuned to a specific mode, usually the fundamental mode, for optimal vibration suppression.
Figure 1.2 TMD

TMDs have been widely used for vibration control in mechanical systems (Den Hartog, 1956), (Crandall and Marks, 1963). The optimum tuning ratio for a TMD was determined as that which minimises the maxima of the peaks of the transfer function (where the transfer function is defined as the ratio of the input excitation to that of the output response).

In recent years, TMD theory has been adopted to reduce vibrations of tall buildings and other civil engineering structures. Kwok and Samali (1995) concluded that a tall structure with TMD has additional damping of 3% to 4% of critical damping and 40% to 50% reduction in the wind induced response. Rana and Soong (1998) examined a simplified procedure for TMD design and performed a parametric study to enhance the understanding of the TMD. Another study on the optimal design of TMDs for building vibration control (Chang, 1999) subjected a coupled TMD~SDOF system to a broadband white noise excitation and found the TMD performed well. In addition to response reduction from earthquake motions, TMDs were also found to provide additional damage protection by increasing the yield strength of a structure subjected to a random excitation by an equivalent of 20% (Pinkaew et al., 2003). It has been found that it is prudent to take soil-structure interaction into account in the tuning of the TMD to the structure-foundation frequency when the underlying soil is susceptible to lessening stiffness over time (Ghosh and Basu, 2004). Such changes in the primary structures stiffness/natural frequency can render a TLCD or TMD
ineffective due to loss of tuning. A smart TMD or semi-active TMD would be needed in such cases to prevent loss of tuning. (Nagarajaiah and Nadathur, 2005). The control of wind excited building with semi-active TMD (STMD) was investigated by Nagarajaiah and Nadathur (2005). The variable stiffness TMD was found to be both robust in terms of stiffness uncertainty and effective in terms of reducing the response of the structure under wind excitation. Nagarajaiah and Sonmez (2007) have studied the response of STMD and multiple STMD under harmonic, stationary and non-stationary excitations. The effectiveness of the STMD in reducing the structural response when the primary structures stiffness changes was shown.

1.2.2.2 Tuned Liquid Column Dampers

Tuned liquid dampers (TLD), of which Sakai et al. (1989) proposed the first TLD as a means of suppressing vibrations within structures, are dampers whose damping effects depend on the liquid residing in the damper and which are specifically tuned to the natural frequency of a structure. A TLCD is a U-shaped tube, which contains a liquid, usually water. Tests by Sakai et al. (1989) demonstrated the effectiveness of Tuned Liquid Column Dampers (TLCDs) at damping vibrations in civil engineering structures. The tuning ratio, which is the ratio of the natural frequency of the TLCD to that of the structure, is optimised in order to ensure an efficient transfer of shear force from the TLCD to the structure.

As an earthquake or the wind induces a lateral vibration in a structure, the vibrational energy is transmitted from the structure into the rigid TLCD container, which in turn transfers the excitation into the TLCD liquid. Since the motion of the TLCD is essentially out of phase with the motion of the structure, a gravitational restoring force acting on the displaced TLCD liquid suppresses the vibrations of the structure. Energy is dissipated by the viscous interaction of the TLCD liquid and the rigid TLCD container, as well as head loss due to orifices installed inside the TLCD container (Hitchcock et al., 1997).

The utilization of TLCDs over TMDs as a means of suppressing vibration energy within structures is being accelerated due to factors such as: they can dissipate very
low amplitude excitations, they are consistent over a wide range of excitation levels and they are self-contained passive damping device, with no auxiliary equipment, personnel or power required to operate and maintain it and are easy to install. The TLCD also enjoys the advantage of reducing both internal shear forces and deflections in the structure (Taylor Devices Inc., 2001). Compared to TLDs, TLCDs prove more efficient in respect to volumetric efficiency, TLCDs introduce extra damping effects and variable damping due to the orifice, and the damping effect of TLCDs are easier to quantify.

A TLCD has several significant advantages compared to other types of energy dissipaters, such as hysteric (friction), visco-elastic (rubber), tuned masses, and elasto-plastic (yielding metal) types.

These advantages may be summarised as:

1. The output of a fluid damper is essentially out of phase with primary bending and shearing stresses in a structure. This indicates that the damper can be used to reduce both internal shear forces and deflections in the structure.
2. A fluid damper is self-contained, no auxiliary equipment, personnel or power is required to operate and maintain it.
3. They can dissipate very low amplitude excitations.
4. Fluid dampers are generally less expensive to purchase, install, and maintain than other types.
5. In a case of emergency, the water in the LCD may be used for fire fighting purposes (Balendra et al., 2001).

In the design of a TLCD, there are many factors taken into account to optimise the parameters of the damper to maximize the suppression of undesirable vibrations. These parameters include its mass ratio (i.e., the ratio of the mass of the damper to the mass of the structure), its tuning ratio (i.e., the ratio of the damper frequency to the natural frequency of the structure) and the head loss coefficient. The mass ratio remains fixed with the implementation of the TLCD and the aspects for which the tuning ratio depends upon (i.e., the length of the liquid column, the ratio of the horizontal to the vertical parts of the tube and also the ratio of the cross sectional areas of the horizontal and vertical parts of the tube) may also remain fixed, however there
is an inherent ability in the orifice for its characteristics to be changed after installation of the TLCD. This aspect also has ramifications in the derivation of a closed form solution for the system as the TLCD, due to the presence of the orifice is an inherently non-linear system. Sakai et al. (1989), Watkins and Hitchcock (1992) and Gao et al. (1997) have all studied various aspects of the TLCD. It was observed that it is possible to install a TLCD in a flexible structure by increasing the area ratio when the required length of liquid column is too long. Sakai et al. (1989) defined the relationship between the coefficient of head loss and the liquid damping, whilst Gao et al. (1999) studied the optimization of TLCDs with regards to various aspects in the design of TLCD. Balendra et al. (1995) studied the effectiveness of TLCDs for vibration control of a tower under stochastic wind excitations. Balendra et al. (1995) also determined the optimum parameters of the TLCD for maximum suppression of wind induced accelerations over a wide range of tower dimensions. They proved that TLCDs are effective in reducing wind-induced vibrations in towers and suggested that by adjusting the orifice opening one could optimise the performance of the TLCD. Yalla and Kareem (2003) studied the use of a semi-active TLCD which had the ability to change the orifice opening with an electropneumatic valve. They concluded that the semi-active orifice system can improve the performance of a fixed orifice. Ghosh and Basu (2005) showed how soil properties can have a direct influence on the dynamic properties of a structure while designing the TLCD for seismic vibration control. As soil properties are likely to change over time, the ability to change the damping properties of the TLCD through manipulation of the orifice opening represent an advantage of the TLCD over some of its damping rivals.

Samali et al. (1992) investigated the application of TLCDs to tall buildings and concluded that they may be successfully used to damp vibrations in such buildings. Balendra et al. (1998) studied the performance of the TLCD on vibration control of structures with varying degree of taper. They concluded that flexural buildings experienced greater acceleration and displacement reductions than shear buildings. They also observed that the height at which the maximum reduction in structural acceleration is achieved decreases with the degree of taper. Gao et al. (1999) have studied multiple TLCDs in suppressing structural vibrations for both across-wind and along wind directions. To control both the along-wind and across-wind motions simultaneously of a structure subjected to wind excitation, Zhang and Zhang (1993)
proposed a TLCD in the form of crossed dampers, which consequently reduced the across-wind resonance response of a multi-degree of freedom system by 50%. TLCDs have been implemented in Hotel Cosima, Hyatt Hotel and Ichida Building in Osaka (Shimizu and Teramura, 1994) and also in One Wall Centre in Vancouver (Glotman Simpson Consultant Engineers, 2001). One wall centre is presented in Figure 1.3. Changing soil properties over time may alter the natural frequencies of the structure and although the TLCD is usually tuned to the natural frequency of the structure, it will also dampen the shifted frequencies.

Sun et al. (1992) utilized the shallow water wave theory with consideration of water residing in a tank to study the TLDs damping effects and experimentally confirmed that their model accounts for the effect of breaking waves. Tamura et al. (1995) investigated the effectiveness of TLDs under wind excitation and proved that the implementation of TLDs in four structures decreased wind induced accelerations between 1/2 to 1/3 times the undamped values. Koh et al. (1995) investigated the combined use of liquid dampers which are tuned to different vibration frequencies of a multi-degree-of-freedom structure. Tamura et al. (1996) investigated the efficiency of the TCLD in a 77.6m Tokyo international airport tower. TLCDs have been found to be effective in reducing vibrations in towers when analysed in the time domain (Wang et al., 1996). Reed et al. (1998) investigated TLDs through shaking table tests and numerical modelling for large amplitude excitation. They concluded that the tank behaves as a hardening spring system due to the liquid sloshing motion, and this trend is enhanced as excitation amplitude increases. Examples of TLD-controlled structures include Nagasaki airport tower, Yokohoma marine tower, Sakitama bridge, Shin Yokohoma Prince Hotel and Tokyo international airport tower. Figure 1.3 shows One Wall Centre (Glotman Simpson Consultant Engineers, 2001), which has two 125,000 litre TLCDs installed to reduce lateral displacement caused by vortices trailing the leeward corner of the building.

In general, the energy-loss mechanisms in additionally damped systems are much more complicated than a simply viscous damping force. One may instead determine the amount of supplemental damping that is obtained due to the incorporation of the damper (the equivalent viscous damping ratio). The equivalent viscous damping ratio gives an appropriate measure of the performance of the damper. Igusa and Xu (1994)
formulated the additional equivalent damping of a structure with multiple Tuned Mass Dampers (TMDs). Tamura et al. (1996) studied the damping increase of the 77.6m high Tokyo International Airport Tower using a tuned liquid damper system. Ghosh and Basu (2004b) studied the equivalent viscous damping of a structure-spring connected TLCD system subjected to seismic excitation by modifying the conventional TLCD to be suited for short period structures.

Figure 1.3 One Wall Centre (Glotman Simpson Consultant Engineers, 2001)
Fujino et al. (1992) experimentally tested high viscosity fluids in liquid dampers and concluded that high viscosity fluids do not necessarily provide better performance of liquid dampers. Hitchcock et al. (1997) showed that by increasing the region of viscous interaction the TLCD liquid damping ratio can be increased. They achieved this by mixing and testing various TLCD liquids consisting of different proportions of fresh water and methylated spirits. Etylene glycol is a hydrocarbon which is often stored in offshore structures for anti-freeze purposes. Vandiver and Mitone (1978) investigated the effects of liquid storage tanks containing glycol on the dynamic response of offshore structures and concluded that prudent selection of the geometry of the storage tanks would dampen the response of the platform of the offshore structure. By using a combination of potential flow theory and an equivalent mass spring system for the case of zero damping, the equivalent masses of the standing waves in an oscillating tank were determined. Additional studies into TLCDs have been conducted by Shum and Xu (2002), Xue et al. (2000), Wu and Hsieh (2002), Colwell and Basu (2006a) and Colwell and Basu (2007a).

Offshore wind turbines are slender structures which are subjected to various excitation forces. By implementing TLCDs in offshore wind turbines, the vibrations and bending moments are expected to be reduced and hence there will be an increase in the fatigue life of the structure. Also the downtime of acceleration sensitive equipment is expected to be reduced, which decreases the cost of producing energy from offshore wind turbines.

1.2.2.3 Base Isolation Systems and combined damper systems

Base isolation (BI) is thought of as an aseismic design approach in which the building is protected from the hazards of earthquake forces. The base isolating mechanism reduces the transmission of horizontal acceleration into the structure so that any damaging earthquake motion cannot be transmitted into the building. The main concept in base isolation is to reduce the fundamental frequency of structural vibration to a value lower than the predominant energy containing frequencies of earthquake ground motions. The other purpose of an isolation system is to provide means of energy dissipation thereby reducing the transmitted acceleration into the
superstructure. Accordingly, by traditionally using base isolation devices in the foundations, the structure is essentially uncoupled from the ground motion during earthquakes. Reviews of earlier work and recent works on base isolation system are provided by Kelly (1986) and Buckle and Mayes (1990). An analysis of a BI system taking into account nonlinear behaviour of single and multiple building base isolated structures was investigated by Nagarajaiah et al. (1991) and Tsopelas et al. (1994).

The most commonly used isolation system utilizes laminated rubber bearings (Kelly, 1987). Laminated rubber bearings can significantly reduce the acceleration response of a structure however their low stiffness in the horizontal directions may potentially cause unacceptably large lateral displacements. Thus, the sensitivity of a building with base isolation system subjected to wind loads is increased as base isolation generally increases the horizontal flexibility of a structure (Henderson and Novak, 2006), (Chen and Ahmadi, 1992). Liang et al. (2002) concluded that wind loads can cause excessive accelerations in base isolated structures. The base isolation system under wind loading was found to dampen the acceleration response of the structure, but did little for the displacement response reduction of the superstructure.

It is necessary to reduce the excessive displacement response of a base isolation system subjected to wind excitation by utilizing an energy-dissipation mechanism. Previous systems have included high damping rubber bearings (Derham and Kelly, 1985), lead-filled rubber bearings (Robinson, 1982) or oil dampers (Kuroda et al., 1989).

Tsai (1995) investigated the effect of TMD on the response of seismically excited structures with base isolation. The response on base-isolated structures equipped with the tuned-mass damper is quite dependent on the input earthquake motions. Adding the tuned-mass damper to the structure enhances the structural response if the input frequency is lower than the natural frequency of the structure. This is not serious for base isolation because the natural frequency of base-isolated structures is generally lower than the dominant frequency of real earthquakes. In terms of wind and wave excitation, care must be taken that the natural frequency of the structure is below that of the frequencies of highest energy content. Kareem (1997) investigated base-isolated buildings with TMDs at both the top of a structure and at the base isolation
level under wind loadings. He concluded that the lower TMD, placed at the base isolator, is more effective in mitigating wind induced response. A review of cases where buildings have been equipped with seismic base isolation and passive dampers in Japan is given by Kitagaway and Midorikawaz (1998). A ‘smart’ base isolation benchmark building was developed by Nagarajaiah and Narasimhan (2006).

As offshore wind turbines are subjected to a combined wind and wave loading, whose correlation may potentially cause excessive structural vibrations, isolating the wave forces from the structure is expected to prove effective in reducing the induced vibrations in the structure.

1.2.2.4 Magnetorheological Fluids

Magnetorheological (MR) fluids are ‘smart’ materials and belong to the class of controllable fluids whose rheological properties are altered with an applied field. The application of MR fluids in civil engineering structures have been conducted by Dyke et al. (1996), Spencer et al. (1997), Spencer et al. (2003), Yoshioka et al. (2002), Yang et al. (2002) and Saharabudhe and Nagarajaiah (2005). Recent studies have shown that semi-active and hybrid dampers, such as those dampers containing MR Fluids, are as successful at damping wind or seismic induced vibrations as fully active dampers. Nagarajaiah et al. (2006) investigated a multiple hybrid damping scheme through the seismic response of ‘smart’ isolated structures with semi-active MR dampers and variable stiffness ‘SAIVS’ system.

Deviating from conventional applications of MR fluids in dampers is the potential use of these fluids in TLCDs. Ying et al. (2005) developed a semi-active optimal control method for non-linear multi-degree-of-freedom systems with MR-TLCD under wind excitation, which combines the benefits of active and passive control methods. Ni et al. (2004) numerically investigated a semi-active MR-TLCD installed at the top floor of a 50-storey building driven by a proposed optimal control strategy. Significant response reduction in terms of displacement, interstory drift and acceleration, in comparison with that obtained by using a passive TLCD were recorded. However, not much research has been carried out on this application, with no experimental results
available in the literature. For the applicability of MR fluids in semi-active TLCDs, investigations on the passive damping properties (head loss due to flow) of the fluids are necessary as the intention in semi-active MR-TLCDs would be to change the head loss by applying a magnetic field. Colwell and Jakob (2006) have shown experimentally that an applied field may be used effectively to alter the damping characteristics of an MR-TLCD in reducing vibrations in a slender structure.

Real life implemented MR fluids tend to possess a very high viscosity in the ‘off’ state. If the MR fluids possess a viscosity too high then it may be difficult to implement in a TLCD as the damping in a TLCD predominantly depends on movement of the liquid induced by the gravitational restoring force. The MR-TLCDs investigated theoretically by Ying et al. (2005) and Ni et al. (2004) both utilize the passive damping properties of the residing MR fluid and also assume a continuous power supply to allow the operation of the semi-active control methods developed. If the power supply ceases to operate for a technical reason, the MR-TLCD relies exclusively on the passive damping properties of the residing liquid. MR fluids are currently, in relation to water, very viscous fluids with a relatively high mass density for use in MR-TLCDs. Thus, it is imperative that the passive damping properties of the MR fluid within MR-TLCDs are understood.

As TLCDs, which are passive damping devices, are expected to provide an effective method of vibration control in offshore wind turbines, it is expected that an MR-TLCD (which is a semi-active damper) with proven effective passive damping properties can improve upon the damping performance of a TLCD installed in an offshore wind turbine.

1.2.3 Wind Loading

Wind is the movement of air caused by uneven heating of the Earth's surface. The wind velocity is greatest at the top of the atmospheric boundary layer, where the depth of this layer can range from between a few hundred meters to several kilometres (Simiu and Scanlan, 1978). The reduction in wind velocity at the earth’s surface is due to frictional drag between the air flow and any elements that protrude into the
wind flow, such as trees and structures. Hence, wind velocities are generally higher out at sea where there are few, if any, protrusions into the wind's flow. For normal winds the atmospheric extends over a height of 1-2 kilometres. It is within the atmospheric boundary that turbulence develops. In wind engineering, turbulence is characterized by the presence of swirling vortices of rotating air of various sizes that act against a structure. It is these eddies that mix together in three dimensional space, causing gusts which vary in both space and time to develop.

Davenport (1961) proposed an expression for the velocity spectrum for the distribution of energy within turbulent flow, at a height related to the size of gusts at that height. However this spectrum was independent of height and was also seen to overestimate the energy in the higher frequency range. Harris (1971) proposed a velocity spectrum which was independent of height that guaranteed a non-zero integral length scale of turbulence. Kaimal et al. (1972) developed the first expression for the variation of spectral energy with height, which includes eddy currents of varying size acting between the structural nodes. Harris (1990) subsequently provided a wind spectrum based on a modified version of the Von-Kármán spectrum which included the variation of spectral energy with height.

The wind velocity spectrum experienced by rotating blades is somewhat different to that experienced by a fixed base tower. As the wind as seen by the rotor system is constantly changing as the rotor rotates, the general flow turbulence may bring wind speed changes on a time scale of about 5 seconds (Manwell et al., 2002). If the blades are rotating once a second, the blades 'sample' different parts of the flow field at a much faster rate than the general changes in the wind itself, causing rapidly changing flow at the blade. Connell (1980) studied the rotationally sampled spectrum on a rotating blade and reported that as the spectral density is altered, the variance is shifted from the lower frequencies to peaks located at integer multiples of the rotational frequency. Kristensen and Frandsen (1982) developed a model to predict the power spectrum associated with a rotating blade, which differed from that when the rotation was not taken into account. Madsen and Frandsen (1984) used the rotationally sampled spectrum to obtain the structural response of rotating blades. Sorensen et al. (2002) presented a wind model for the dynamic interaction between wind farms and the power system to which they are connected. The model also
included the effects of rotationally sampled turbulence on blades. Murtagh et al. (2005) proposed an approach to study the along-wind response of a wind turbine tower coupled with a rotationally sampled rotor system in the time domain. A reduction in the amount of modes needed to adequately examine the coupled tower-rotor system was recommended. It was also found that the tip-structural response of the tower decreased with increased rotational speed of the rotor system.

1.2.4 Wave Loading

The turbulence from offshore wind produces small capillary waves at the sea surface with similarly small wavelengths in the range of centimeters. The wind acts on the tiny walls that these ripples create, causing them to become larger. Wind blowing over the wave produces pressure differences along the wave profile causing the wave to grow. The process is unstable because, as the wave gets bigger, the pressure differences get bigger, and the wave grows exponentially. Finally, the waves begin to interact among themselves to produce longer waves (Hasselmann, 1980). This interaction transfers wave energy from short waves generated by Miles mechanism to waves with frequencies slightly lower than the frequency of waves at the peak of the spectrum. Eventually, this leads to waves going faster than the wind (Pierson and Moskowitz, 1964).

Despite the irregular random nature of wind inducing seemingly random wave heights and wave periods in a typical offshore setting, any associated changes that occur with the state of the sea are slow enough to allow the characterization of a sea state. A sea state is defined (Ditlevsen, 2002) as a wave situation that is approximately constant during some time interval of appropriate duration in respect to the significant wave height $H_s$ and the characteristic zero crossing point $T_z$. $H_s$ is the average height (trough to crest) of the one-third highest waves valid for the indicated 12 hour period. The sea surface, which is made up of a superposition of sinusoidal waves with random heights and periods, will oscillate vertically in an aperiodic manner around a zero level reference point. The time intervals between each meeting of the sea surface and the zero level vary around an average value, $T_z$. Hence, the process is narrow-banded and can be related with the mean wind speed.
Pierson and Moskowitz (1964) assumed that if the wind blew for a long time over a large area, the waves would come into equilibrium with the wind and low frequency waves are developed in a fully developed sea. A large area being defined as 5,000 wave lengths on a side and a long time being defined as 10,000 wave periods (Stewart, 2005). This is the concept of a fully developed sea. To obtain a spectrum of a fully developed sea, they used accelerometers to measure wave data on ships in the North Atlantic. Hasselmann et al. (1980) measured wave data in the North Sea and discovered that a sea never actually becomes fully developed. They developed their own spectrum (JONSWAP) which takes into account a higher peak than the Pierson-Moskowitz spectrum.

The expression for the in-line force per unit length along a cylinder was first investigated over 50 years ago (Morison et al., 1950). The mitigation of vibrations of an offshore structural system with additional damper when subjected to in-line random wave forces was investigated by Lee (1997). In the stochastic analysis performed, a random type of wave forces derived from the Morison equation for small bodies was applied. It was observed that in terms of the power spectral density, the effect of the vibration mitigation and the dynamic performance of the offshore structural system were greatly improved when the new damping devices were applied to the offshore structural system. By considering the environmental loading on a structure with a TMD to be a long-term nonstationary stochastic process characterized by a probabilistic power spectral density function (Li and Hu, 2002), it was found that the use of a TMD could significantly reduce the fatigue damage of an offshore structure. A method of determining the ‘generalized’ wave force was based on an analytical approximation of the mode shape function, together with the physical wave loading being calculated from linearized Morison equation (Li et al., 2003). The effectiveness of $H_2$ active control is numerically demonstrated, with a reduction in the standard deviation of the deck motion for the passive TMD case of 50% for the active TMD case with $H_2$ active control. A jacket-type offshore platform modelled as a SDOF structure with attached active mass damper (AMD) was recently simulated by using the linearized Morison equation is used to estimate the wave load (Ma, 2006). An active control strategy (Ahmad and Ahmad, 1999) was used to determine the control force magnitude for serviceability and survival of tension leg platforms (TLPs) subjected to wind and wave loadings. Significant reduction in low-frequency
hull motion of the TLP was observed. Thus, although several investigations have been carried out for control of offshore structures, not much attention has been focused on the vibration control of flexible offshore structures subjected to joint wind and wave loading with additional damping system.

1.2.5 Offshore Wind Turbine Reliability

In general, reliability is the ability of a system to perform and maintain its functions in routine circumstances, as well as hostile or unexpected circumstances. Wind turbine reliability is a critical factor in the success of a wind energy project. Poor reliability directly affects both the project’s revenue stream through increased operation and maintenance (O&M) costs and reduced availability to generate power due to turbine downtime. Walford (2006) summarized the various aspects of quantifying the true reliability of wind turbines. The key project evaluation metric used to evaluate the overall effect of reliability on a wind turbines cost is the Cost of Energy (COE). This metric takes into account initial capital costs (ICC), the fixed charge rate per year (FCR), the levelized replacement cost (LRL), O&M costs and annual energy production (AEP). It is evident that an increase in reliability increases the AEP and reduces the O&M costs, thus decreasing the COE.

In current wind turbines, when the wind speed reaches a certain level, established through anemometers attached to the nacelle, a protective measure is triggered which shuts down the operation of the wind turbine. The protective measure assumes a direct relationship between mean wind speed and structural acceleration response. This protective approach ignores the structural response of the system, and unacceptable levels of acceleration or displacement can be attained by the towers even at wind speeds below the threshold for shutdown. A recent study has shown that the effects of wind turbulence, and not only wind speed, can induce large structural responses in coupled wind tower/rotor-blade systems (Murtagh and Basu, 2006). Changes in wind direction can also introduce unwanted response effects, but their nature is transient and modern control technologies, such as yaw control, are sufficient for variant wind angles of attack (Farret et al., 2000).
Recently, several attempts have been made to correlate wind turbine performance with wind speeds (Saranyasootorn and Manuel, 2004) (Knut O. Ronold et al., 1999) (Tavner et al., 2006). The studies aim to correlate blade flapping, structural fatigue and component failure rates with wind speeds deriving from stochastic models based on recorded data. However, without taking into account mechanics-based models of wind turbines, an accurate picture of the effects of wind turbine turbulence on the effects of turbine availability are not quantified.

In their study, Dueñas-Osorio and Basu (2007) explicitly model the structural response of wind turbines subjected to variable wind speeds. The acceleration response of the turbine system was coupled with a probabilistic description of the dynamic properties of the wind turbine to produce an annual distribution of the wind hazard. Wind induced fragility curves are derived from an annual distribution of the wind hazard to obtain a distribution of the acceleration response for various levels of wind speed.

The dynamic model was embedded into a probabilistic simulation routine to explore the variability of the acceleration response given various levels of wind speed and dynamic characteristics of the system. Fragility curves that relate the probability of exceeding predefined acceleration levels with annual wind speeds were constructed from the simulation data.

By implementing dampers in offshore wind turbines, the vibrations and bending moments are reduced and hence there will be an increase in the fatigue life of the structure. The DNV offshore standards (DNV, 2004) calculate the fatigue life under the assumption of linearly cumulative damage. A case study of fatigue analysis of an offshore wind power generation facility subjected to wind and wave loading was recently carried out (Shiraishi et al., 2006). The wave spectrum was modelled by the Pierson-Moskowitz spectrum and the fluctuating wind speed time history was prepared using the Ochi-Shin spectrum. They used the rain-flow calculation method and the zero-upcrossing calculation method to estimate the fatigue life at various points of an offshore wind turbine assembly subjected to wind and wave loading. In a
case such as an offshore wind turbine tower where the stress fluctuations caused by wind and wave excitations are superimposed, it can be used as a suitable calculation method.

1.3 ORGANISATION OF THE THESIS

The body of the work is segmented into seven chapters.

Chapter 2 investigates the properties of the TLCD itself by presenting a parametric investigation on the TLCD. The undamped structure is modelled as a uniform beam having a lumped circular mass at the top to replicate a wind turbine tower with the nacelle and the blades at the top. The undamped structure is next coupled with the TLCD and transfer functions in the frequency domain for the response of the system are presented. Particular emphasis is placed on the coefficient of head loss, which is parameter than can be easily changed throughout the lifetime of the TLCD. The TLCD-structure system, with varying coefficient of head loss, is investigated experimentally with a TLCD-structure system constructed in the Civil Engineering Laboratory, in Trinity College. The experimental results are validated with respect to the theoretical formulation.

Water has traditionally been used in TLCDs although semi-active control and additional functional requirements (anti-freezing) of TLCDs can be achieved with MR fluids and glycol as resident TLCD liquids, respectively. Thus, Chapter 3 presents an experimental investigation into the passive damping properties of various fluids, including MR fluid, in the TLCD. The effects of the various liquids are investigated through the coefficient of head loss, which has ramifications for the structural and liquid response. The experimental results are validated through the theoretical formulation presented.

Chapter 4 examines the excitation of an offshore wind turbine modelled as a MDOF system under wind and wave loadings. The chapter presents a model for calculating the time-history response of an offshore wind turbine, with coupled rotating blade
system, equipped with a TLCD and subjected to wind and wave forces. A correlation between the wind and wave excitations using joint distribution is presented. The benefits of installing TLCDs in offshore wind turbines, in terms of increased fatigue life and decreased construction costs, are examined.

**Chapter 5** presents a hybrid BI-TMD-TLCD damping system in offshore wind turbines. The system utilizes the properties of each damper to both isolate and damp the structure from the wind, wave and blade excitations. A parametric investigation into the various factors influencing the performance of the offshore wind turbine- BI-TMD-TLCD is presented. An additional method of energy generation, involving the TMD, deriving from the harnessing of nature's forces is also proposed and investigated.

**Chapter 6** examines the long term benefits of installing TLCDs in offshore wind turbines using observed wind data in two locations. A long term probabilistic model of the wind turbine response and the risk of peak accelerations in offshore wind turbines exceeding predefined cut-off limits are presented by means of fragility curves.

**Chapter 7** summarizes and concludes upon the material investigated and presented upon in the preceding chapters. Offshore wind turbines and damping systems, represent an ever-evolving avenue of research with numerous possibilities. Thus, some recommendations for future research are also offered.
Chapter 2

INVESTIGATIONS ON THE PERFORMANCE OF A TLCD WITH DIFFERENT ORIFICE DIAMETER RATIOS

2.1 INTRODUCTION

Offshore wind turbines are slender structures which are subjected to various excitation forces. By implementing TLCDs in offshore wind turbines, the vibrations and bending moments are expected to be reduced and hence there will be an increase in the fatigue life of the structure. Also the downtime of acceleration sensitive equipment is expected to be reduced, which decreases the cost of producing energy from offshore wind turbines. To optimise the performance of the TLCD, various parameters may be investigated. One of the most important parameters on the performance of the TLCD is the coefficient of head loss, which is related to the orifice diameter ratio.

An experimental investigation into the influence of the orifice in a TLCD in changing the vibration suppression characteristics of the fundamental mode of a wind turbine is carried out in this chapter. A method to find the natural frequencies of a SDOF system modelled as a cantilevered beam is presented in Section 2.2. A closed form solution for the transfer function of the maximum displacement of the structure, modelled as a single-degree-of-freedom (SDOF) system, with applied harmonic base motion, has been presented in Section 2.3. A standard equivalent linearization technique has been employed to cater for the non-linear damping of the orifice in the TLCD. The performance of the orifice has been evaluated through the experimental testing of various base displacement amplitudes and frequencies applied to the TLCD-structural system in Section 2.4.
An SDOF system was built on a platform capable of undergoing sinusoidal excitation to pre-defined input frequencies and displacements. A TLCD was also built with the ability to change its orifice size. Both theoretical and experimental results are plotted to examine and compare the performance of TLCD in Section 2.5.

### 2.2 Parameters of the SDOF System

The structure is modelled as a uniform beam having a lumped circular mass at the top to replicate a wind turbine tower with the nacelle and the blades at the top. To obtain the modal properties of the structure, it considered as an elementary beam under flexural deformation idealized to have uniform distribution of mass and stiffness along its length with a lumped mass at the top. The free vibration equation of motion for the system is (Clough and Penzien, 1976)

\[
EI \frac{\delta^4 v(z,t)}{\delta z^4} + m \frac{\delta^2 v(z,t)}{\delta t^2} = 0
\]  

(2.1)

where, the flexural rigidity and the mass per unit length are denoted by \(EI(z)\) and \(m(z)\), respectively. In equation (2.1), \(v(z,t)\) is the transverse displacement response of the structure at a distance \(z\) from the ground at an instant of time \(t\). After separation of variables, equation (2.1) may be converted to a set of two ordinary differential equations in time and in space with the variables \(\ddot{Y}(t)\) and \(\phi(z)\) denoting the generalised displacement amplitude and the mode shape, respectively.

\[
\ddot{Y}(t) + \omega^2 Y(t) = 0
\]  

(2.2)

\[
\phi'''(x) - a^4 \phi(x) = 0
\]  

(2.3)
Solving the mode shape equation by introducing an exponential function leads to the form of the mode shape as

\[ \phi(z) = A_1 \cos az + A_2 \sin az + A_3 \cosh az + A_4 \sinh az \]  

(2.4)

where, \( A_1, A_2, A_3 \) and \( A_4 \) are real constants to be evaluated by using the boundary conditions and \( a^4 = \frac{\omega_i^2 \bar{m}}{EI} \), with \( a \) being a parameter related to the \( i^{th} \) mode considered, \( \omega_i \) is the \( i^{th} \) natural frequency and \( \bar{m} \) the mass per unit length of the beam. The four boundary conditions to be satisfied for the lumped mass cantilever are

\[ \begin{align*}
\phi(0) &= 0 \\
\phi''(0) &= 0 \\
\phi''(L) &= \frac{\omega_i^2 \phi'(L) j_1}{EI} \\
\omega_o &= \frac{2 \pi}{T} \sqrt{1 - \zeta^2}
\end{align*} \]  

(2.5)

where, \( j_1 \) is the corresponding rotary mass moment of inertia. The prime denotes differentiation with respect to space, \( z \) and \( \phi''(L) \) and \( \phi'''(L) \) represent the moment and shear at the free end of the cantilever where the mass is concentrated. Incorporating the boundary conditions into equation (2.1), four equations in terms of the constants \( A_1, A_2, A_3 \) and \( A_4 \) are produced. Arranging in matrix form and evaluating the determinant of the resulting 4x4 matrix to zero, the resulting equation that yields the eigenvalues of the system becomes (Murtagh, 2005)

\[ \begin{align*}
\left[ a^4 - \frac{\omega_i^4 j_1 \bar{m}}{(EI)^2} \right] \cosh(aL) \cos(aL) &- \frac{\omega_i^2 ma + \omega_i^2 j_1 a^3}{EI} \cosh(aL) \sin(aL) + \\
\frac{\omega_i^2 ma - \omega_i^2 j_1 a^3}{EI} \cos(aL) \sinh(aL) &- a^4 \frac{\omega_i^2 j_1 \bar{m}}{(EI)^2} \\
\end{align*} \]  

(2.6)
where, \( m \) is the mass of the free end body and \( j_1 \) is the rotary mass moment of inertia. An iterative scheme is used to solve equation (2.6) for the natural frequency of the structure. Since only the fundamental natural frequency is of interest here equation (2.6) has been solved for the lowest value of \( a \).

To obtain a value for the damping of the structural system, the logarithmic decrement method based on free vibration response is used. This method is used to determine the damping ratio of an undercritically damped structure subjected to an initial displacement by measuring the reduction in amplitude of successive peaks of the displacement and the damping ratio is given by

\[
\zeta = \frac{1}{2\pi} \ln \left( \frac{u_p}{u_{p+1}} \right) \quad (2.7)
\]

In equation (2.7), \( u_p \) and \( u_{p+1} \) are the amplitudes of the \( p \) and \( (p+1) \)th peaks respectively. A similar expression holds good for the estimation of damping using the acceleration response of the structure which is used in this study and is given by

\[
\zeta = \frac{1}{2\pi} \ln \left( \frac{a_p}{a_{p+1}} \right) \quad (2.8)
\]

where, \( a_p \) and \( a_{p+1} \) are the accelerations of the \( p \) and \( (p+1) \)th peaks respectively. The damped natural frequency, \( \omega_D \), of the structure is given by

\[
\omega_D = \frac{2\pi}{T} \sqrt{1 - \zeta^2} \quad (2.9)
\]

where the natural period of the structure is given by
When the fundamental mode is only of interest, the SDOF structure is represented by a simple single degree of freedom system with natural frequency $\omega_n$ and damping ratio $\zeta$. The equation of motion for a SDOF structure subjected to a time varying force, $F(t)$, is represented by

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = F(t)$$

(2.11)

where, $x$ is the relative displacement of degree of freedom of interest.

### 2.3 TLCD-STRUCTURAL SYSTEM

The equation of motion of the liquid column damper for the TLCD-Structure system subjected to a base acceleration $\ddot{z}(t)$, as developed by Sakai et al. (1989), is given as

$$\rho A L \ddot{u}(t) + \frac{1}{2} \rho A \zeta |u(t)| \dot{u}(t) + 2 \rho A g u(t) = -\rho A B \{ \ddot{x}(t) + \ddot{z}(t) \}$$

(2.12)

where $u(t)$ is the change in elevation of the liquid column, $x(t)$ is the horizontal displacement of the SDOF system with TLCD, relative to the ground, $A$ is the cross sectional area of the TLCD, $B$ is the horizontal dimension and $L$ is the total length of the tube. The notation $\zeta$, $\rho$, and $g$ represent coefficient of head loss controlled by the opening ratio of the orifice, liquid mass density and the acceleration due to gravity respectively. By adopting an equivalent linearization procedure (Iwan and Yang, 1971), the linearized form of equation (2.12) may be written as
\[ \rho AL\ddot{u} + \rho AC_p \dot{u} + 2\rho Agu = -\rho AB\{\ddot{x} + \ddot{z}\} \quad (2.13) \]

where \( C_p \) represents the equivalent linearized coefficient of head loss. Normalizing equation (2.13) with respect to the mass of the liquid, \( \rho AL \), yields

\[ \ddot{u} + \frac{C_p}{L} \dot{u} + \omega_i^2 u = -\alpha \{\ddot{x}(t) + \ddot{z}\} \quad (2.14) \]

where,

\[ \omega_i = \sqrt{\frac{2g}{L}} \quad (2.15) \]

represents the natural frequency of the TLCD and the ratio of the horizontal section of the TLCD tube to its total length is given by

\[ \alpha = \frac{B}{L} \quad (2.16) \]

The expression for the equivalent linearized coefficient, \( C_p \), for harmonic motion of the liquid in the U-tube, assuming an in-compressible liquid, is obtained by minimising the mean square value of the error between equations (2.12) and (2.14) and is given as

\[ C_p = \frac{4\omega U(\omega)\xi}{3\pi} \quad (2.17) \]

where, \( U(\omega) \) is the magnitude of the amplitude of the liquid displacement in the TLCD at a frequency \( \omega \). Introducing the shear force transmitted from the TLCD to the SDOF structure, \( [\rho AL\{\ddot{x} + \ddot{z}\} + \rho AB\ddot{u}] \), into the system equation yields the equation of motion for the mass, \( m \), of the structural model with attached TLCD, as
\[ m(\ddot{x}(t) + \dot{z}(t)) + c\dot{x}(t) + kx(t) = -\rho AL \{\ddot{x}(t) + \dot{z}(t)\} - \rho AB\ddot{u}(t) \]  

(2.18)

where, the right hand side of equation (2.18) describes the shear force from the TLCD induced into the SDOF structure. If equation (5) is normalized with respect to \( m \), one obtains

\[(1 + \mu)\ddot{x}(t) + 2\zeta\omega_n\dot{x}(t) + \omega_n^2x(t) = -\mu\omega_n\ddot{u}(t) - (1 + \mu)\ddot{z}(t) \]  

(2.19)

where, \( \mu \) is the ratio of the mass of the liquid in the damper to that of the structure, \( \zeta \) is the damping ratio of the structure (without TLCD) and \( \omega_n \) represents the natural frequency of the SDOF system.

On Fourier transforming equations (2.14) and (2.19), the following equations are obtained (Colwell and Basu, 2005)

\[ U(\omega) = \frac{\alpha}{\omega^2 - \omega^2 + \frac{C_p}{L}\omega_n} \left\{ \omega^2X(\omega) - \dot{Z}(\omega) \right\} \]  

(2.20)

\[ X(\omega) = \frac{1}{\omega_n^2 - \omega^2(1 + \mu) + 2i\omega_n\zeta\omega} \left\{ \omega^2\alpha\mu U(\omega) - (1 + \mu)\dot{Z}(\omega) \right\} \]  

(2.21)

In equations (2.20) and (2.21), \( \tilde{X}(\omega), U(\omega) \) and \( \tilde{Z}(\omega) \) are the Fourier transforms of the corresponding time-dependent variables, \( x(t), u(t) \) and \( z(t) \) respectively and \( i = \sqrt{-1} \).
2.4 EXPERIMENTAL INVESTIGATION

A four-columned structure with a mass at its top was used to model a shear beam and was built on a platform capable of applying sinusoidal excitation. The SDOF structure (representative of the fundamental mode of a wind turbine) had a mass of 8kg and columns of length 200mm, possessing a moment of inertia of $6.67 \times 10^{-11} \text{m}^4$ and a Young's modulus of elasticity equal to $7.5 \times 10^9 \text{N/m}^2$. The natural frequency of the structure was 13.869 radians per second.

A frequency generator, which could generate frequencies accurately at an interval of 0.001Hz, was used as an input to the magnetron at the base of the structure in order to apply sinusoidal base excitations. Linear variable differential transformers (LVDTs) were used to obtain both the applied base displacement and the tip displacement at the tip of the two models. A data scanner, which provides fast simultaneous data acquisition and digitisation of multiple channels of various analog inputs, was used to log and graph the values obtained from the LVDTs. The scanner has input shielding, which has the ability to reduce noise and protect the integrity of the bridge signal from Electro-Static Discharge (ESD) and Radio Frequency (RF) noise. Figure 2.1 shows the experimental set-up of the SDOF structure. The two LVDT’s and the shaking table are indicated on the figure. Further information regarding the experimental equipment used in the testing in this section may be found in the Appendix A. A U-shaped TLCD consisting of upright, rectangular columns coupled with a horizontal cylinder with variable orifice, filled with water to a height, H, at two ends was located at the top of the structure. The coefficient of head loss, $\xi$, was controlled by the opening ratio of the orifice. The TLCD was clamped to the structure and the walls of the TLCD were assumed to be of uniform thickness while the liquid was considered to be incompressible, inviscid and free at its top surface. The values of B, L for the TLCD are 64mm and 125.5mm, respectively. The tuning ratio is 91% of the undamped structure. A schematic of the SDOF-TLCD experimental set-up is shown in Figure 2.2. A picture of the TLCD attached to the structure can be seen in Figure 2.3.
Figure 2.1 Experimental Set-up

Figure 2.2 Schematic depicting the experimental set-up
Figure 2.3 The interaction between the TLCD and the Structure

Figure 2.4 Test for linearity with fitted line using least squares method
In order to test that the undamped structure behaves linearly, a test for linearity was carried out. With the frequency of the base excitation held constant, the base amplitude was increased in steps through the resonant range. An example of tip amplitude versus base amplitude plot for the SDOF structure subjected to a sinusoidal base excitation is shown in Figure 2.4. Figure 2.4 was taken at a frequency close to resonance (i.e., 2.08Hz) and as can be seen, even with very large amplitudes present, the response of the structure is almost linear.

2.4.1 Harmonic Forced Vibration response

For a specific value of circular orifice opening ratio and amplitude of excitation, the relationship between the dynamic amplification factor (DAF, which is the tip displacement divided by the base displacement) and the excitation frequency is investigated by means of an applied base sinusoidal motion of constant amplitude at a frequency and varying the frequency over a range. The TLCD was constructed with variable orifice capability. The mass ratio, $\mu$ for the TLCD was 4.63 % and the value of $\alpha$ was 0.51 for the tests formed for this study. The orifice ratios used in the tests were 1.0d, 0.75d, 0.5d and 0.25d, where d is the diameter of the horizontal pipe. At each individual frequency, the structure was allowed to reach a steady state, from which the corresponding tip displacements and base amplitudes were obtained. Several tests, each for different structural cases (damped and undamped) under different amplitudes and frequencies, were carried out and analysed.

2.4.2 SDOF-LCD System: Optimum orifice diameter

The TLCD was mounted on the top of the SDOF structure. Figures 2.5 to 2.8 show DAF versus frequency curves obtained from the testing of the damped SDOF structure for the different orifice values, from an orifice size of 1d (Figure 2.5) down to an orifice size of 0.25d (Figure 2.8), at base amplitude of 3.75mm, contrasted against the results obtained from the testing of the undamped SDOF structure.
Figure 2.5 DAF versus frequency for base amplitude 3.75mm and orifice diameter 1d

Figure 2.6 DAF versus frequency for base amplitude 3.75mm and orifice diameter 0.75d
Figure 2.7 DAF versus frequency for base amplitude 3.75mm and orifice diameter 0.5d

Figure 2.8 DAF versus frequency for base amplitude 3.75mm and orifice diameter 0.25d
It can be seen from Figure 2.8 that the least relative reduction in amplitude during resonance was observed. This may be explained by the fact that when an orifice plate of 0.25d is used, the water is not permitted to flow freely between the two tanks, which reduces the damping effect that the gravitational restoring force acting on the liquid may impart on the structure. Also, it was seen that there was less viscous interaction between the liquid and the rigid container, resulting in less energy loss. The combination of these two effects would have overshadowed the increased head loss due to the relatively small orifice plate size, resulting in a reduced suppression of the structural vibrations.

Figures 2.9 to 2.12 show DAF versus frequency graphs derived from the testing of the damped SDOF structure from an orifice size of 1d (Figure 2.9) down to an orifice size of 0.25d (Figure 2.12), at base amplitude of 2.5mm.

![Figure 2.9 DAF versus frequency for base amplitude 2.5mm and orifice diameter 1d](image_url)
Figure 2.10 DAF versus frequency for base amplitude 2.5mm and orifice diameter 0.75d

Figure 2.11 DAF versus frequency for base amplitude 2.5mm and orifice diameter 0.5d
Plots 2.9 to 2.12 mirror the results obtained for the induced amplitude of 3.75mm in Figures 2.5 to 2.8. This would suggest that for a SDOF structure modelled as a shear beam the contribution of the orifice is independent of the amplitude of the lateral base excitation, however the two base amplitudes tested are not dissimilar enough to prove this.

Figures 2.13 and 2.14 show the influence of the orifice in terms of reducing the overall displacements over a range of frequencies for base displacements of 2.5mm and 3.75mm respectively. The reduction in the displacement of the system is given as a percentage of the displacements recorded at the same frequencies for the undamped SDOF system. Figures 2.13 and 2.14 indicate that there is a clear correlation between orifice size and damper performance.
Figure 2.13 Orifice influence on SDOF structure with Base Amplitude = 3.75mm

Figure 2.14 Orifice influence on SDOF structure with Base Amplitude = 2.5mm
At high frequencies the orifice plate imparts the optimal damping on the system. This may be due to the fact that at high frequencies, the structure is moving too quickly to allow adequate movement of the water column. Thus the effects of the gravitational restoring force and the head loss due to the orifice are minimised. Thus the sloshing–slamming of the liquid becomes the predominant mode of energy dissipation. In a sense, at high frequencies, the damper effectively acts as an additional mass on the structure, which decreases the stiffness of the structure and results in a higher amplitude response.

Figure 2.15 gives a graphical representation of the orifice plate sizes which produce the optimal damping on the system for each particular frequency. It can be seen that the two curves are quite similar, with the only exception between the two being the optimal orifice sizes at 1.5Hz and 1.9Hz, where the displacement differences were quite small.

Figure 2.15 Optimum orifice diameter for SDOF structure under different excitation frequencies for varying amplitude
2.5 SYSTEM VALIDATION

This section involves the implementation of the theoretical analysis that was developed to verify how good the theoretical analysis was to represent the system. To obtain the natural frequency and damping of the structure from the undamped forced vibration tests, the method of least squares to fit the response amplitude curve was implemented. A natural frequency value of 13.84 rad/s and damping ratio of 3% for the undamped structure were obtained.

The theoretically identified response is validated with the experimental response of the SDOF structure in terms of the DAF without TLCD for base excitations of 3.75mm and 2.5mm in Figures 2.16 and 2.17, respectively. The damped theoretical response calculations involve the instigation of the equations developed for $X(\omega)$, $U(\omega)$ and $C_p$ which give the peak displacement of the structure.

![Figure 2.16 Undamped response with base amplitude = 3.75mm](image-url)
To calculate the damped response of the SDOF structure, the values that were used for $\xi$ were taken from a table presented by Balendra et al. (1995). In contrast to the rectangular orifice used by Balendra et al. (1995), a circular orifice is used in this investigation. The rectangular and circular orifice cross sections are shown in Figures 2.18(A) and 2.18(B).
Implementing the equations formulated in section 2.3 for the case of a 2.5mm base excitation with the rigidly attached TLCD for the five orifice cases yields Figures 2.19 to 2.23.

Figure 2.19 DAF versus frequency for base amplitude 2.5mm and orifice diameter 0.75d

Figure 2.20 DAF versus frequency for base amplitude 3.75mm and orifice diameter 0.75d
Figure 2.21 DAF versus frequency for base amplitude 3.75mm and orifice diameter 0.5d

Figure 2.22 DAF versus frequency for base amplitude 2.5 mm and orifice diameter 0.5d
It can be seen that the theoretically predicted displacements match up well to the experimentally obtained displacements at the resonant range of frequencies. Thus, the theoretical formulae, taking into account the various orifice sizes utilised, adequately model the response of the damped structure. However, the theoretical predictions and experimentally obtained values differ at relatively low and high frequency ranges when the experimental undamped and damped displacement results are quite similar. The difference may be attributed to the value of $\zeta$ that was used as well as well as that the derivation for the formulae in section 2.3 assumed a lightly damped structure, whereas the structure used in the experimental investigation had relatively high structural damping. The fact that the model uses an equivalent viscous damping value, where matching is done at resonance, also contributes to non-coincidence away from resonance. The linearization of the effects of the orifice may also have resulted in the difference.
2.6 CONCLUSIONS

It has been shown from this chapter that the tuning of the natural frequency of the TLCD with the fundamental mode of the wind turbine reduces to a considerable extent the vibration response of the turbine subjected to a base excitation simulating the wave loading. The effectiveness of the TLCD in suppressing harmonic excitations at resonance for SDOF systems has been experimentally demonstrated. It is observed that changes in the orifice size in the pipe had a direct impact on the damping characteristics of the damper. Orifice sizes to optimise the damping of the TLCD based on the experimental results were proposed for the SDOF system. The experimental results and the equivalent damping ratio values obtained suggest that TLCD performance is dependant on excitation amplitude. In specific cases, due to an optimal coefficient of damping not being provided, the structure-TLCD response may increase for narrow banded (harmonic in this case) excitations. It was found that orifices sizes between 0.5d and 1d produced optimal performances of the TLCD at various frequencies. Thus, even though the TLCD was specifically designed to operate at a certain frequency that induces resonance in the structure, by controlling the opening ratio one could still obtain a satisfactory performance from the TLCD at non-resonant frequencies. An expression for the response of structure-TLCD system was formulated and the theoretical damped response were compared with the experimental results.
3.1 INTRODUCTION

In Chapter 2, parameters that optimize the performance of a TLCD, most notably the coefficient of head loss, in a slender structure were investigated. In this chapter, an experimental investigation into the passive damping properties of various fluids, including magnetorheological (MR) fluid, in the TLCD is undertaken. The coefficient of head loss for different fluids used in TLCDs to reduce structural responses in a wind turbine modelled as a single-degree-of-freedom (SDOF) system subjected to base excitation is investigated.

Water has traditionally been used in TLCDs although semi-active control and additional functional requirements (anti-freezing) of TLCDs can be achieved with MR fluids and glycol as resident TLCD liquids, respectively. The semi-active MR-TLCD works by utilizing the ability to change the damping properties (i.e., head loss) of the MR fluid by applying a magnetic field within the TLCD. However, the effectiveness of the MR-TLCD relies upon an adequate movement of the MR fluid within the TLCD (for both tuning and damping through head loss). Hence an investigation into the damping properties of an MR fluid as the residing liquid within the TLCD is imperative for semi-active MR-TLCDs to be realizable. The performances of water, glycol and an MR fluid are compared and the merits of each of the fluids in providing adequate passive damping to the turbine structure are discussed. The equivalent viscous damping in the structure provided by the TLCDs using each of the fluids is also obtained for both harmonic and broad banded excitations. The theoretical analysis also aims to find if the existing TLCD theory may be implemented to accurately describe the passive damping performance of the MR-TLCD.
Equations for calculating the coupled TLCD-structure response in the time domain are presented in Section 3.2. The Kanai-Tajimi type base excitation that is applied to the structure in the experimental tests is presented in Section 3.2.1. An experimental investigation into the effects of various liquids in the TLCD is presented in Section 3.3. Experimental results are used to calculate the non-linear coefficient of head loss based on a theoretical formulation in Section 3.3.3. The numerical simulations of the responses of the structure-TLCD with various fluids used in TLCDs are validated with the experimental results in Section 3.4.

3.2 STRUCTURE-TLCD SYSTEM

The transfer matrix is defined as the matrix that algebraically relates a system’s output to its input. Through combining Equations (2.14) and (2.19), a different representation of the transfer function relating the tip displacement of the SDOF oscillator to the base acceleration applied to the structure may be derived

\[
H_s(\omega) = \frac{-\alpha^2 \mu \omega^2 - (1 + \mu) \left[ \omega_t^2 - \omega^2 + \frac{2C_p}{\omega_L} \right]}{\left[ \omega_n^2 - \omega^2 (1 + \mu) + 2i\zeta \omega \omega_n \right] \left[ \omega_t^2 - \omega^2 + \frac{2C_p}{\omega_L} \right] - \alpha^2 \mu \omega^4} \tag{3.1}
\]

where, \( i \) is the complex quantity \((i = \sqrt{-1})\). The equation that relates the base acceleration of the structure to the displacement of the liquid within the U-tube is given as

\[
H_u(\omega) = \frac{-\alpha \left( \omega_n^2 + 2i\zeta \omega \omega_n \right)}{\left[ \omega_n^2 - \omega^2 (1 + \mu) + 2i\zeta \omega \omega_n \right] \left[ \omega_t^2 - \omega^2 + \frac{2C_p}{\omega_L} \right] - \alpha^2 \mu \omega^4} \tag{3.2}
\]

The equivalent viscous damping to a structure, \( \zeta_s \), obtained by the addition of the TLCD to the structure can be found by considering the structural displacement
transfer function of the SDOF structure, with natural frequency \( \omega_n \) and damping \( \zeta \), related to the base acceleration given by

\[
H^*_{\omega}(\omega) = \frac{1}{\left(\omega_n^2 - \omega^2\right) + 2i\zeta \omega \omega_n} \tag{3.3}
\]

By equating the appropriate response and the values (e.g., root-mean-square (r.m.s) or peak amplitude) obtained using equations (3.1) and (3.3), and a particular excitation (e.g., random white noise or filtered white noise or a harmonic), the equivalent viscous damping of the structure-TLCD can be calculated.

To numerically simulate the coupled structure-TLCD interaction when subjected to random excitation in the time domain, one can combine equations (2.14) and (2.19) to form the following equation

\[
M_{\text{mod},1}\ddot{Z}_1 + C_{\text{mod},1}\dot{Z}_1 + K_{\text{mod},1}Z_1 = F_{T_1}(t) \tag{3.4}
\]

in which the \((n+1)\)-dimensional displacement vector \(Z_1\), the \((n+1)\)-dimensional generalized mass matrix \(M_{\text{mod},1}\), damping \(C_{\text{mod},1}\) and stiffness matrix \(K_{\text{mod},1}\), and the \((n+1)\) dimensional generalized total input force vector \(F_{T_1}(t)\), are given by (Colwell and Basu, 2004)

\[
Z_1 = \begin{bmatrix} \chi \\ \eta \end{bmatrix}, \tag{3.5}
\]

\[
M_{\text{mod},1} = \begin{bmatrix} 1 + \mu & \mu \alpha \\ \alpha & 1 \end{bmatrix}, \tag{3.6}
\]

\[
C_{\text{mod},1} = \begin{bmatrix} 2\zeta \omega_n & 0 \\ 0 & \frac{\zeta}{2L} \|\eta\| \end{bmatrix}, \tag{3.7}
\]
where, $I_n$ is the n-dimensional identity matrix. The terms $\zeta_i$ and $\omega_i$ represent the damping ratio and natural frequency of the fundamental mode, respectively. $F$ represents the force applied to the structural system.

### 3.2.1 Base Excitation

A random variable may be seen as a number which characterizes an event, which belongs to a complete set of events. Every time the ‘experiment’ is performed (i.e., the specified set of conditions is realized), a sample value of the random variable is obtained. The sample function obtained will be different every time the ‘experiment’ is performed, and there is thus an infinite set of possible sample functions. Covariance, or autocorrelation, is the measure of how much two variables vary together. That is to say, the covariance becomes more positive for each pair of values which differ from their mean in the same direction, and becomes more negative with each pair of values which differ from their mean in opposite directions. In this way, the more often they differ in the same direction, the more positive the covariance, and the more often they differ in opposite directions, the more negative the covariance. Autocovariance functions are the covariance functions that act along the same axis only. When the autocovariance functions are normalised by their respective variances, they become what is known as autocorrelation functions. A Fourier Transform of an autocorrelation function yields a magnitude of the random excitation, or turbulence, known as the power spectral density function (PSDF). The PSDF of a waveform will give the power carried by the wave, per unit frequency.
Offshore wind turbine structures are subjected not just to wind and wave excitations but also sometimes to earthquake excitations at sea bed level and are to be designed for safety against such loadings. The application of base excitation will also form a general idea on the behaviour of turbine structures with a damper when vibrating. The base acceleration process applied to the structure is represented by its PSDF, $S_\beta(\beta)$. In this case, the Kanai-Tajimi spectrum is used as a base excitation applied to the structural system. The Kanai-Tajimi spectrum is given as follows

$$S_\beta(\beta) = \phi_0 \frac{1 + 4\zeta_\phi^2 \left(\omega/\omega_\phi\right)^2}{\left[1 - \left(\omega/\omega_\phi\right)^2\right]^2 + 4\zeta_\phi^2 \left(\omega/\omega_\phi\right)^2} \tag{3.10}$$

The parameters $\zeta_\phi$ and $\omega_\phi$ are the damping ratio and natural frequency of the ground respectively, which may be varied to suit specific local soil conditions. Values of 0.5 and $5\pi$ are chosen here to replicate previous recorded ground motion data (Collins, 2006). For the experimental study considered in section 3.3.2.3, $\phi_0=1cm^2/s^3$ is assumed in equation (3.10) without any loss of generality.

### 3.3 Experimental Investigation

Two tubular test structures (S1 and S2) made of slender annular pipe were used to model flexible structures representative of wind turbine towers. A pre-determined steel weight was placed at the top of each of the tubes in order to achieve a desired fundamental natural frequency. The tubes were made from polyvinyl chloride (PVC). A schematic diagram of the structural model with attached TLCD is shown in Figure 3.1.
A frequency generator was used to input signals to the magnetron at the base of the structure in order to excite the structure. A linear variable differential transformer (LVDT) was used to obtain the applied base displacements to the structure and an accelerometer was attached at the top of the structure in order to obtain the acceleration response at the top of the model. A U-shaped TLCD consisting of upright, cylindrical columns coupled with a horizontal cylinder, filled with fluid to a certain height of the vertical columns was located at the top of the structure. Two horizontal tubes were constructed for the TLCD. The first was a simple acrylic pipe, whereas the second one had a horizontal section with an orifice in the centre of it, thus enabling the coefficient of head loss, $\xi$, to be varied. The circular orifice used was half the diameter of the tube for the study in this paper as the optimal damping is known to be produced for the cases of the orifice diameters between 0.5 to 1.0 times the diameter of the horizontal tube (Balendra et al., 1995).

It may be noted that the experimental results in this paper are derived from a laboratory scaled model of the structure with the TLCD, which may have some scale effects. To minimise the scale effects certain principles of similitude have been
adopted. To ensure the preservation of the dynamic effects, the natural frequency of
the structure and that of the TLCD have been considered to be the same as that of a
prototype structure. For a hydraulic similitude to account for the surface friction loss,
Reynolds’ number has to be preserved. It is a difficult task to control this for the flow
in the tube in the experimental set-up and hence has been violated. However, since the
flow in the tube of the TLCD will be almost laminar the violation of this criterion will
not have significant effect on the results as it may have for other types of liquid
dampers.

3.3.1 Test liquids

In order to investigate and ascertain the influence of different fluids which have
various respective benefits for use in TLCDs; water, glycol (which is also used as an
anti-freeze compound) and MR fluid as passive damping liquids were tested as the
residing liquids within the TLCD. With an applied field in an MR-TLCD, one may
manipulate the behaviour of the fluid for use in a semi-active context, making the
passive damping properties of the liquid important. Another possible advantage of
low viscosity MR fluid used in TLCDs is the fact that the relatively large particles
within the liquid, which may form “floating” particles, can be used to reduce the beat
phenomenon that has been observed in certain real life applications of TLCDs. The
beat phenomenon, attributed to the back and forth transfer of energy between the
structure and the sloshing liquid, can cause increases in the response of a structure-
TLCD system in the free vibration state. The beat phenomenon also introduces
difficulties in calculating the system damping from the free vibration response time
histories. Tamura et al. (1995) found that the addition of floating particles within the
TLCDs in the Tokyo International Airport Tower caused the beat phenomenon to
greatly diminish. Another fluid of interest is glycol. When mixed with water, glycol
significantly lowers the freezing point of water.

The densities of water, glycol and the low viscosity MR fluid are 1 (g/ml), 1.04 (g/ml)
and 1.65 (g/ml), respectively. A number of viscosity tests using the DIN4 (ISO 9002,
1994) standard viscosity cup were performed in order to obtain the viscosities of the
respective liquids. The viscosities of the water, glycol and MR fluid were measured as
0.89 centipoise, 16.2 centipoise and 45.6 centipoise, respectively. The mass ratio, tuning ratio, and α value for each test case are given in Table 3.1 (Section 3.3.2.1) and Table 3.2 (Section 3.3.2.2) for structures S1 and S2, respectively.

It may be noted that the mass ratios for the MR fluid cases are relatively higher due to the higher density of the fluid as compared to water and glycol. As MR fluids are used more widely in civil engineering applications, it is likely that the formulation of such fluids will gear towards lower density MR fluids which may be practically implemented in MR-TLCDs without sacrificing much on the desired MR properties. The iron particulates in the MR fluid prepared for the experiments in this study remained in suspension for a sufficient amount of time as to be consistent for the duration of the testing. Over a longer time, settling was observed in the MR fluid, however with adequate mixing, the MR fluid became homogenous again.

3.3.2 Vibrational response

In order to characterise the vibrational response of structural systems S1 and S2 with and without the TLCDs, both free and forced vibration tests were carried out. Five log decrement free vibration tests were undertaken for each test case and the average values for the damping and the natural frequency were obtained. For the harmonic forced vibration tests, the structural models were excited with a base sinusoidal motion of constant amplitude over a range of frequencies. At each individual frequency, the structure was allowed to reach a steady state, from which the corresponding tip accelerations at the top of the structure and the base amplitudes were obtained. Free vibration tests were carried out on the undamped structure in order to calculate the damped natural frequency and the natural damping of the structure. The experimental set up is seen in Figure 3.2.
The mass of the undamped structure S1 is 1.36kg, the outside diameter of the pipe is 21.29mm, the thickness of the pipe is 1.188mm, the second moment of area, \( I = 3.80 \times 10^{-9} \text{m}^4 \), the length of the beam is 1 m, the elastic modulus of the pipe was established from applying a unit force and measuring the displacement of the beam and was \( 3.45 \times 10^9 \text{Nm}^{-2} \), the cross sectional area of the pipe is \( 7.5 \times 10^{-5} \text{m}^2 \) and the mass per unit length is \( 0.13 \text{kgm}^{-1} \).

The mass of the undamped structure S2 is 1.75kg, the outside diameter of the pipe is 21.29mm, the thickness of the pipe is 1.188mm, the second moment of area, \( I = 3.80 \times 10^{-9} \text{m}^4 \), the length of the beam is 0.6m, the elastic modulus of the pipe was established from applying a unit force and measuring the displacement of the beam.
and was $2.65 \times 10^9 \text{Nm}^{-2}$, the cross sectional area of the pipe is $7.5 \times 10^{-5} \text{m}^2$ and the mass per unit length is $0.11 \text{kgm}^{-1}$.

An ‘Acceleration versus Time’ log decrement plot for the undamped structure S1, given an initial displacement of 90mm, can be seen in Figure 3.3. Using the test results, average values of the time periods and acceleration peaks were taken, leading to the free vibration damped natural frequency and the period of the structure.

![Log decrement graph for undamped structure S1](image)

**Figure 3.3 Example of log decrement graph for undamped structure S1**

With the natural frequency and structural damping obtained from the theoretical calculations obtained from equation (2.6), the theoretical relative tip displacement of the structure can be established given a known input. The chosen response representation is the transfer function, $H(\omega)$, which is the ratio of the base acceleration of the structure to the relative displacement at the structures. A plot of $H(\omega)$ versus frequency for the undamped structure, both experimentally and theoretically, is seen in Figure 3.4. As expected, the plot for the system has a single peak occurring at the natural frequency of the structure.
3.3.2.1 Structure S1 with TLCD

The damped natural frequency and damping ratio of structure S1 were obtained from experimental tests on the structure without TLCD as 11.04 rad/sec and 0.94% respectively and the values were very close as obtained both from the free and the forced vibration cases. For the experimental tests undertaken on structure S1, the TLCD was tuned to the natural frequency of the structure subject to the condition that the displacement of the liquid should be less than (L-B)/2. This condition ensures that the free surface of the liquid does not enter the horizontal pipe. Gao et al. (1997) studied the optimisation of U-shaped liquid column dampers and produced tables for the optimisation of the performance of the TLCD in terms of the tuning ratio. By extrapolating the work done by Gao et al. (1997), a tuning value of 0.96 was selected for the experimental investigation. As the tuning ratio is related to the length of the liquid column, a tuning ratio of 0.96 also ensures that the precondition of (L-B)/2 being less than the displacement of the liquid is adhered to. The value of $\alpha$ for structural system S1 was 0.35. A TLCD with a horizontal tube length of 60mm and two vertical column lengths of 40mm was selected. The mass ratio ($\mu$) for the cases of the water, glycol and MR fluid in the TLCD were 4%, 4.06% and 6.5% respectively.
The TLCD was rigidly attached to the free end of the structure. The amplitude of the transfer functions ($|H_x(\omega)|$) which relates the relative displacement at the top of the structure to the applied base acceleration were then plotted against the corresponding excitation frequencies. The results from the five test cases can be seen in Figures 3.5 - 3.7. The five cases are TLCD containing MR fluid, glycol, water with no orifice and TLCD with orifice containing glycol and water. The equivalent viscous damping ratio values obtained experimentally using the free vibration decay method (averaged over the results from five trials for each case) and the half power (band-width) method using forced vibration results for all the test cases are presented in Table 3.1.

![Figure 3.5](image)

**Figure 3.5** Experimental displacement transfer function of structure S1 and structure-TLCD with MR fluid against frequency ratio
Figure 3.6 Experimental transfer function of structure S1, structure-TLCD with glycol and structure-TLCD with glycol and orifice against frequency ratio

Figure 3.7 Experimental transfer function of structure S1, structure-TLCD with water and structure-TLCD with water and orifice against frequency ratio
Table 3.1 Summary of test cases and damping results for structure S1

<table>
<thead>
<tr>
<th>TLCD liquid</th>
<th>Mass ratio (%)</th>
<th>Tuning ratio</th>
<th>$\alpha$</th>
<th>Equivalent viscous damping using half-power (bandwidth) method</th>
<th>Equivalent viscous damping free vibration decay method</th>
<th>Theoretical equivalent viscous damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.96%</td>
<td>0.94%</td>
<td>0.98%</td>
</tr>
<tr>
<td>Water</td>
<td>4</td>
<td>0.96</td>
<td>0.35</td>
<td>2.21%</td>
<td>2.19%</td>
<td>2.25%</td>
</tr>
<tr>
<td>Water with orifice</td>
<td>4</td>
<td>0.96</td>
<td>0.35</td>
<td>2.15%</td>
<td>2.27%</td>
<td>2.22%</td>
</tr>
<tr>
<td>Glycol</td>
<td>4.05</td>
<td>0.96</td>
<td>0.35</td>
<td>2.97%</td>
<td>3.01%</td>
<td>3.11%</td>
</tr>
<tr>
<td>Glycol with orifice</td>
<td>4.05</td>
<td>0.96</td>
<td>0.35</td>
<td>3.01%</td>
<td>3.16%</td>
<td>3.04%</td>
</tr>
<tr>
<td>MR fluid</td>
<td>6.5</td>
<td>0.96</td>
<td>0.35</td>
<td>3.05%</td>
<td>3.04%</td>
<td>3.12%</td>
</tr>
</tbody>
</table>

The cases of water and glycol without any orifice produce significant reductions in the transfer function as compared to the cases with orifice. The TLCD case with water and no orifice reduces the response of the structure at resonance by 44%. The TLCD case with glycol and no orifice shows a 42% reduction in the response of the structure at resonance compared to the undamped case. The reduced effects of orifice for the cases with glycol and water in damping the structure are supported by (Balendra et al., 1995), where it is shown that the influence of the orifice is greater at higher values of $\alpha$.

The MR fluid produced much the same performance as the glycol and water cases without orifice. In the experimental tests, though the MR fluid moved inside the MR-TLCD, a marked reduction in the amplitude of the movement of the MR fluid was
observed. The MR fluid showed a 42% reduction in the tip response of the structure at resonance. The experimental test with the MR fluid suggests that in times of failure of the semi-active MR-TLCD control system, with the MR-TLCD relying exclusively on the passive damping capability of the MR fluid, the MR fluid can be relied upon to act as the resident liquid in a conventional TLCD.

There was a perceptible difference between the fluid behaviour of glycol and water in the TLCD during testing. The difference in the velocity and amplitude height between water and MR fluid was clearly visible to the observer. There were no large inclines of the liquid surface at any time as the surface moved up and down with a horizontal straight surface. The liquid surface never came close to the top of the vertical tubes of the TLCD for the test cases in this study, however, there may exist cases where the maximum amplitude of the liquid may pose a threat in exceeding the vertical dimensions of the TLCD.

3.3.2.2 Structure S2 with TLCD

The damped natural frequency and damping of structure S2 are obtained from experimental tests on the structure S2 as 8.90 radians per second and 0.92% respectively using similar tests carried out for the structure S1. The natural frequency of the structure S2 allowed a wider scope in parametric study of the TLCD in terms of the variation of the tuning ratio, $\alpha$ values and mass ratio of the TLCD. For each of the liquid cases, three different liquid column lengths, and hence three different cases of mass ratio, $\alpha$ and tuning ratio, were investigated. Table 3.2 shows the different test cases investigated for structure S2.

Figures 3.8 to 3.10 show the amplitude of the transfer functions, $|H_\nu(\omega)|$ for the cases with water, glycol and MR fluid as detailed in Table 3.2. The test cases where the TLCD was tuned (Case B for all fluids) to the natural frequency of the structure produce the greatest structural response reduction. Case A, despite an increased $\alpha$ value, imparts the lowest damping to the structure for the cases with glycol and water as the effect is offset by decreased mass ratio. MR fluid cases A and C produce similar response reductions, which show that the MR fluid in a TLCD will have a
slightly different interaction effect due to the effects of the $\alpha$ value and the mass ratio. Thus, the viscosity of the MR fluid within the TLCD imparts some effects on the system that are not consistent with those observed for the water and glycol cases. Water produces the highest equivalent viscous damping ratios, followed by glycol and then MR fluid. This would indicate that for the test cases studied, the lower the viscosity of a fluid, the greater is its performance within a TLCD.

Figure 3.8 Experimental transfer function of structure S2 and structure-TLCD with water cases A, B, C against frequency ratio
Figure 3.9 Experimental transfer function of structure S2, structure-TLCD with glycol cases A, B, C against frequency ratio

Figure 3.10 Experimental displacement transfer function of structure S2, structure-TLCD with MR fluid cases A, B, C against frequency ratio
<table>
<thead>
<tr>
<th>TLCD liquid</th>
<th>Test case</th>
<th>Mass ratio (%)</th>
<th>Tuning ratio</th>
<th>$\alpha$</th>
<th>RMS values (m/s²)</th>
<th>Theoretical equivalent viscous damping values from Kanai-Tajimi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.3355</td>
<td>1.2%</td>
</tr>
<tr>
<td>Water</td>
<td>A</td>
<td>3.05</td>
<td>1.09</td>
<td>0.35</td>
<td>0.1966</td>
<td>6.67%</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>3.77</td>
<td>1.00</td>
<td>0.29</td>
<td>0.1954</td>
<td>6.70%</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>4.30</td>
<td>0.94</td>
<td>0.25</td>
<td>0.2120</td>
<td>6.15%</td>
</tr>
<tr>
<td>Glycol</td>
<td>A</td>
<td>3.20</td>
<td>1.09</td>
<td>0.35</td>
<td>0.2095</td>
<td>6.24%</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>3.95</td>
<td>1.00</td>
<td>0.29</td>
<td>0.2113</td>
<td>6.18%</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>4.52</td>
<td>0.94</td>
<td>0.25</td>
<td>0.2104</td>
<td>6.21%</td>
</tr>
<tr>
<td>MR Fluid</td>
<td>A</td>
<td>4.96</td>
<td>1.09</td>
<td>0.35</td>
<td>0.2158</td>
<td>6.05%</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>6.13</td>
<td>1.00</td>
<td>0.29</td>
<td>0.2178</td>
<td>6.00%</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>6.99</td>
<td>0.94</td>
<td>0.25</td>
<td>0.2139</td>
<td>6.11%</td>
</tr>
</tbody>
</table>

Table 3.2 Summary of test cases and damping results for structure S2

3.3.2.3 Structure S2 with TLCD: Random excitation

The structure S2 was next excited at the base with an excitation simulated from a broad-banded Kanai-Tajimi type of power spectral density function (PSDF) (Clough and Penzien (1976)), for which firm ground conditions were assumed with the parameters $\omega_x$, $\xi_x$, $S_o$ taken as 20 rad/sec, 0.6 and $10^{-2}$ cm²/sec³ respectively. Figures 3.11 to 3.13 show the amplitude of the Fast Fourier transform (FFT) of the acceleration response at the top of the structure for three representative cases from Table 3.2 and compare those with the structural response without TLCD. The r.m.s. values calculated from the recorded acceleration time histories for the various test cases are presented in Table 3.2.
Figure 3.11 Amplitude of FFT of acceleration response for structure S2 and structure-TLCD water case B against frequency ratio

Figure 3.12 Amplitude of FFT of acceleration response for structure S2 and structure-TLCD glycol case C against frequency ratio
Significant reductions in the r.m.s. acceleration responses for all cases with TLCDs were observed as compared to the one without any TLCD. There is no significant difference between r.m.s. values for cases A, B and C for each of the respective test liquids. The equivalent viscous damping in the structure for the cases considered in Table 3.2 are calculated by equating the experimentally measured r.m.s. values of the acceleration responses to those obtained theoretically by using the Kanai-Tajimi input PSDF and the appropriate transfer function of an equivalent SDOF system derived from equation (3.3). The equivalent viscous damping values are presented in Table 3.2. By comparing the supplemental damping obtained from the cases with narrow banded excitations (in Table 3.1), it may be observed that the TLCD is effective in providing significantly more damping to the structure subjected to broad banded excitations (as shown in Table 3.2). The reason for higher damping imparted in the case of structures subjected to broad banded excitations is possibly due to the non-linear interaction occurring at different frequencies with the non-linear damping in the TLCD, which is absent in narrow banded responses. The effects on the reduction in
the response of the structure with the MR-TLCD damper with different parameters are similar to those observed with water and glycol, even though there is a significant difference in the viscosity and the densities of the fluids.

3.3.3 Discussion on experimental values of $C_h$

The values of equivalent linearized coefficient of head loss, $C_h$, are calculated using the proposed linearized theory using equation (3.1) and experimentally obtained transfer function amplitudes, $|H_x(\omega)|$. Plots of the variation of the equivalent linearized coefficient of head loss, $C_h$, with frequency ratio ($=\omega/\omega_0$) for each of the respective test cases for structural system S1 are shown in Figure 3.14. Close to resonance $C_h$ approaches a low, consistent value for all the liquids, which suggests that a constant value of the non-linear coefficient of head loss, $\xi$, can be used to describe the system behaviour near resonance. At resonance, the MR fluid case has a slightly higher value of $C_h$ than the other test cases, i.e. exhibiting a higher level of linearized damping in the fluid motion at resonance as compared to the other fluids. As the coefficient of head loss is related to the equivalent linearized coefficient of head loss, this suggests that due to the particles within the fluid, additional damping is induced upon the structure. The application of a magnetic field will change the damping characteristics of the liquid, a property of the MR fluid which may be used efficiently in a semi-active MR-TLCD.

The cases of glycol and water follow in close agreement in terms of the equivalent linearized coefficient of head loss, indicating that the slight difference in viscosity between the two liquids does not transmit into a significant difference in their performance in the TLCD. The water and glycol cases with orifice also show similar performances in the TLCD.
Figure 3.14 Variation of $C_h$ against frequency ratio for the different test cases for structure S1

3.3.4 Variation of the amplitude of the various liquid responses in the TLCD

Plots of $H_0(\omega)$ for structural system S1, which is the transfer function relating the magnitude of the displacement of the liquid within the TLCD, to the input base acceleration, for the various test cases are given in Figure 3.15. Figure 3.15 shows how the motion of the MR fluid is lower than the rest of the other liquids. This attribute of the MR fluid may prove useful when external excitations may cause another liquid (i.e. water) to move beyond the limit set by $(L-B)/2$.

It is quite interesting to note the double peaked nature of the liquid response plots. Usually the structural response is of a double peaked nature, which indicates an efficient interaction between fluid and structure. In the structure-TLCD systems examined in this paper, it has been shown that there may be a single peaked structure-
TLCD response due to various combined factors (mass ratio, damping ratio, $\alpha$ and the tuning ratio), as is the case here. The double peaked nature is also expected in the response of the TLCD resident liquid and has been recorded by Gao et al. (1997) in their study on V-shaped TLCDs. An optimally tuned damped-response has two peaks with the natural frequency of the structure at the trough between the two peaks. In this case, non-linearities of the structure, non-linearities of the motion of the liquid and a non-optimally tuned TLCD have contributed to a slight shift to the left in the frequency domain of the damped response.

![Figure 3.15 Displacement transfer function of fluid, $H_u(\omega)$ against frequency ratio for the different test cases for structure S1](image)

3.4 SIMULATION RESULTS AND VALIDATION

For the purpose of validation of the formulation presented in this paper, the theoretical acceleration response time histories of the structures with TLCD have been computed.
By simulating a free vibration test numerically, based on the governing differential equations (using the value of $C_h$ corresponding to the natural frequency of vibration which is the predominant frequency of the frequency spectra in free vibration), the values for the equivalent damping, $\zeta_e$ were obtained for the different test cases for structure $S1$ and compared with experimental values in Table 3.1. As can be seen, the theoretical values are in close accordance with the experimental values in all the cases. In addition to water and glycol, the theoretical model accurately describes the free vibration damping characteristic of the MR fluid.

3.4.1 Investigation into non-linear behaviour of the structure-TLCD system

On using the linearized theory, the values of $C_h$ obtained (from the experimentally obtained transfer function, $|H_x(\omega)|$) subsequently give the transfer function for the liquid displacement, $|H_u(\omega)|$ from equation (3.2). This transfer function multiplied with the base acceleration amplitude gives the amplitude of the liquid displacement, $U(\omega)$, which substituted in equation (2.17) yields the value of the non-linear coefficient of damping, $\xi$. The values of $\xi$ for the TLCD with water and MR fluid are calculated at about resonance to be 0.124 and 0.46 respectively. With the value of $\xi$ near resonance as the initial estimate, the response of the structure with TLCD is calculated using simulation of the non-linear coupled equations (2.14) and (2.19) for a range of frequencies around resonance. By numerically optimizing the difference between the theoretical and the experimental transfer function, the values of $\xi$ are obtained for each excitation frequency and the values for the TLCD with water and MR fluid are plotted in Figure 3.16.

It is seen that the values of $\xi$ calculated from numerical optimization are close to the values obtained about resonance and are fairly constant in the range shown. This confirms the validity of the non-linear TLCD model in this range for the different fluids considered. Beyond this range, the values start deviating from the constant value and equation (2.12) with constant value of $\xi$ may not accurately describe the response of the structure-TLCD system. When the excitation frequencies are away from the range of frequencies around resonance, the liquid motion in the TLCD is not
significant (also evident from the amplitude of liquid motion in Figure 3.15) and the non-linear damping is not that predominant.

Figure 3.16 Variation of $\xi$ against frequency ratio for water and MR fluid cases for structure S1

Figure 3.17 and Figure 3.18 present the theoretical non-linear response (both for the cases with numerically optimized and constant values of $\xi$) of the structure-TLCD system compared with the corresponding experimental results for the cases of water and MR fluid respectively. It can be seen that the constant values of $\xi$ (0.124 and 0.46 for the TLCD with water and MR fluid respectively) gives a good match with the experimental results across the range considered around resonance. This suggests that in a simulation for the response of a TLCD-system, a constant value of $\xi$ accurately represent the coefficient of head loss.
Figure 3.17 Non linear simulated transfer function of water-structure S1 with constant $\zeta$ (0.124) and with optimised values for $\zeta$.

Figure 3.18 Non linear simulated transfer function of MR fluid-structure S1 with constant $\zeta$ (0.124) and with optimised values for $\zeta$. 
3.5 CONCLUSIONS

TLCDs are highly efficient in reducing response in wind turbine structures. Reductions up to 47% in the response of the structure due to the TLCDs were recorded experimentally in this study. The study indicates that TLCD with water and glycol can achieve significant structural response reduction. The same can be said regarding the MR fluid, which also achieved substantial reductions in the structural response.

The non-linear quadratic damping theory accurately models the behaviour of a TLCD in the resonant range of excitation frequencies where there is significant liquid motion leading to increased damping through head loss in TLCDs in addition to the effect of tuning.

The difference in viscosity between water and glycol does not transmit to a significant difference in the passive damping characteristics of the system. Hence, glycol or glycol/water mixtures are viable for structures were freezing temperatures might abound.

In terms of volumetric efficiency, MR fluid is almost as effective as water within the TLCD in passively damping structural responses on the top of a SDOF structure. This is promising for the potential use of MR fluids in a semi-active damping context within the TLCD where the damping due to head loss is controlled. In addition, under certain circumstances, such as low excitation forces or in a disruption of the electrical supply to the semi-active control system, the passive damping properties of the MR-TLCD liquid will be exclusively relied upon.

The displacement response of the MR fluid within the TLCD was significantly lower than the other liquids, indicating its potential use in situations where space for the vertical columns in the TLCD is limited.

This section has shown experimentally and theoretically that an appropriately low viscous MR fluid may be implemented in MR-TLCDs, thus making MR-TLCDs
practical for applications in wind turbines. An MR fluid with relatively low viscosity may be relied upon in an MR-TLCD, although the designer will have to make adequate arrangements to cater for the increased mass ratio and ensure that the benefits of the semi-active MR-TLCD outweigh those of the passive TLCD. Appropriate MR fluids that perform efficiently in the MR-TLCD must be formulated.
Chapter 4

TLCDS IN OFFSHORE WIND TURBINES FOR STRUCTURAL CONTROL WITH ECONOMIC IMPLICATIONS

4.1 INTRODUCTION

With offshore wind turbines becoming larger, being moved out further at sea and subjected to ever greater wind and wave forces, it is necessary to analyse the dynamics and minimise the responses of these structures. In Chapter 2, parameters that optimize the performance of a TLCD, most notably the coefficient of head loss, in a slender structure were investigated. In Chapter 3, an experimental investigation into the passive damping properties of various fluids, including magnetorheological (MR) fluid, in the TLCD was undertaken. In this chapter, the structural response of offshore wind turbines are simulated with attached TLCD for controlling the vibrations induced within the structure. This requires a realistic simulation of the forces that these tall, flexible and slender structures are subjected to. The benefits of implementing a damper to control the resulting undesirable vibrations that are induced within the structure are investigated.

In Section 4.3, the Kaimal spectrum, which takes into account eddy currents of varying size acting between the structural nodes, is presented. The effect of the wind excitation without this variation of spectral energy with height is presented for the case of rotating blades in Section 4.3.1. In Section 4.4, a methodology based on the JONSWAP spectrum to apply a force from a wave excitation is presented. Since sea waves are caused by wind blowing for a sufficiently long time, the state of the sea is related to wind parameters and there exists the possibility of correlating wind and wave loading conditions on structures. The Kaimal spectrum for wind loading is combined with the JONSWAP wave spectrum to formulate correlated wind and wave loadings in Section 4.5. The offshore turbine tower is modelled as a Multi-Degree-of-Freedom (MDOF) structure. A fatigue analysis, based on the rainflow algorithm, to
investigate the impact of a TLCD on the fatigue life of an offshore wind turbine, is presented in Section 4.6.

Cases of the blades lumped at the nacelle along with rotating blades are investigated. Cases for flat sea conditions under ‘moderate’ wind loading exciting the lumped mass structure, with which parallels to onshore wind turbines may be drawn, are first simulated in Section 4.7.2. In Section 4.7.3, the lumped mass structure is excited under ‘strong’ wind and wave loading. The reduction in bending moments and structural displacement response with TLCDs for each case are examined in Section 4.7.5. A fatigue analysis is carried out and the implementation of TLCDs is seen to enhance the fatigue life of the structure.

An economic analysis on the benefits of installing a TLCD in an offshore wind turbine, with regards to the extended fatigue life and reduced bending moments at the base of the structure, is presented in Section 4.9.

4.2 MDOF OFFSHORE TURBINE COUPLED WITH TLCD

The TLCD considered is composed of a U-shaped pipe of circular cross section with orifice installed in it. If one re-writes equation (2.12) to account for a force applied nodally and not through seismic excitation, one gets

\[ \rho A L \ddot{u} + \frac{1}{2} \rho A \xi \dddot{u} + 2 \rho A g u = -\rho A B \dddot{x}_n \]  

(4.1)

where, \( x_n \) is the horizontal displacement at the top of the structure where the TLCD is installed. The overdot represents differentiation with respect to time. Normalizing equation (4.1) with respect to the mass of the liquid yields

\[ \ddot{u} + \frac{\xi}{2L} \dddot{u} + \omega^2 u = -\alpha \dddot{x}_n \]  

(4.2)

where \( \alpha = \frac{-B}{L} \) is the ratio of the horizontal portion of the TLCD tube to its total length and \( \omega_L = \sqrt{2g/L} \) is the natural frequency of the TLCD.

80
4.2.1 MDOF offshore turbine with blades lumped at the nacelle

The case where the mass of the blades is concentrated into the nacelle is first examined. The equation of motion for the structural model, which is represented as an n-degree of freedom structure (Figure 4.1), with attached TLCD is given by (Colwell and Basu, 2006c)

\[
[M][\dddot{X}] + [C]\dot{\ddot{X}} + [K]X = \{F(t)\} - \rho AB\ddot{u} \{R_1\} - \rho A\ddot{x}_n \{R_1\}
\]  

(4.3)

where \(\{X\} = \{x_1, x_2, ..., x_n\}\) denotes the n-dimensional horizontal displacement vector of the structure and \(\ddot{x}_n\) represents the horizontal acceleration at top of the tower (base of the TLCD container); \([M]\), \([C]\) and \([K]\) are the n-dimensional mass, damping and stiffness matrices of the structure, respectively; \(\{F(t)\}\) is the n-dimensional total force loading and \(\{R_1\} = \{1, 0, ..., 0\}\) is an n-dimensional constant vector. The last two terms on the right of equation (4.3) represent the force transmitted from the TLCD into the structure. In Figure 4.1 (Wijnink and Hengeveld, 2006), the virtual mass of the underwater components are taken into account through the equations (Wilson, 2003)

\[
m_{n,4} = m_i + 0.375m_{v,3},
\]

(4.4)

\[
m_{n,3} = m_{vh,3} + 0.375m_{v,3} + 0.375m_{v,4},
\]

(4.5)

\[
m_{n,2} = m_{vh,2} + 0.375m_{v,2} + 0.375m_{v,3} + 0.375m_{v,4},
\]

(4.6)

\[
m_{n,1} = m_{vh,1} + 0.375m_{v,1} + 0.375m_{v,2} + 0.375m_{v,3} + 0.375m_{v,4}
\]

(4.7)

\[
m_{vi} = C_A\rho u_i
\]

(4.8)

and

\[
m_{vh,i} = m_i + C_A\rho u_i
\]

(4.9)
where $m_i$ and $m_{n,i}$ are the actual mass at node $i$ and the overall mass (including virtual mass) at node $i$, respectively. $C_A$, $\rho$, and $v_i$ are the added mass coefficient (which can be approximated as unity), the mass density of water and the volume of water displaced by node $i$, respectively.

Figure 4.1 Structural Model
The structural displacement response can be represented as a product of the mode shape matrix, \( \Phi \), normalized with respect to the structural mass and the \( n \)-dimensional modal coordinate vector, \( \eta \). By inserting \( X = \Phi \eta \) into equation (4.3), the governing equation of the system becomes

\[
([I] + [\mu_n]) [\dot{\eta}] + [\Theta] [\dot{\eta}] + [\Gamma] \{\eta\} = [\phi]^T \{F(t)\} - \mu \alpha [\phi]^T \ddot{u} \{R_i\} \tag{4.10}
\]

where the \( n \)-dimensional matrices \([\Theta] = [\phi]^T [C][\phi] = \text{diag}(2\zeta_1\omega_1, 2\zeta_2\omega_2, \ldots, 2\zeta_n\omega_n)\) and \([\Gamma] = [\phi]^T [K][\phi] = \text{diag}(\omega_1^2, \omega_2^2, \ldots, \omega_n^2)\). The terms \( \zeta_i \) and \( \omega_i \) represent the damping ratio and natural frequency of the \( i \)-th mode respectively. The ratio of the mass of the liquid in the damper to that of the structure is given by \( \mu = \rho AL / \text{trace}(M_m) \) and the corresponding \( n \)-dimensional is given by

\[
[\mu_n] = \begin{bmatrix}
0 & \cdots & 0 \\
\vdots & \ddots & \vdots \\
0 & \cdots & \mu
\end{bmatrix} \tag{4.11}
\]

The \( n \)-dimensional modal coordinate vector at the point where the structure is connected to the TLCD is given by \( \eta_n \) and \( n \) is the amount of degrees of freedom considered.

By combining equations (4.2) and (4.10), the following equation for the wind turbine idealized as a MDOF with a lumped mass at the free end incorporating TLCD is obtained as

\[
[M_{\text{mod}}] \{\ddot{Z}\} + [C_{\text{mod}}] \{\dot{Z}\} + [K_{\text{mod}}] \{Z\} = \{F_T(t)\} \tag{4.12}
\]

in which the \((n+1)\)-dimensional displacement vector \( \{Z\} \), the \((n+1)\)-dimensional generalized mass matrix \([M_{\text{mod}}]\), damping matrix \([C_{\text{mod}}]\) and stiffness matrix \([K_{\text{mod}}]\), and the \((n+1)\) dimensional generalized total input force vector \(\{F_T\}\), are given by (Colwell and Basu, 2007)
where, $I_m$ is the n-dimensional identity matrix and the apostrophe.

### 4.3 WIND EXCITATION

The wind loading on any structural member is decomposed into a constant mean wind load and a fluctuating wind component. The total wind force on any structural member is a summation of the mean and the fluctuating components. For wind excitations, the PSDF matrix, $S_{\nu}(\omega)$ is obtained from the along wind excitation applied to the structure and the turbulence between two points for a N degree of freedom (DOF) structure. The spectrum used in this paper to simulate the fluctuating along wind velocity spectrum was proposed by (Kaimal et al., 1972)

$$S_{\nu}(z, \omega) = \frac{v_i^2}{2\pi \omega} \frac{200n_m}{(1 + 50n_m)^{5/3}}$$

(4.18)
in which

\[ n_m = \frac{2\pi \omega z}{\bar{v}(z)} \]  

(4.19)

and

\[ \bar{v}(z) = \frac{1}{k_{vk}} \nu_s \ln \frac{z}{z_o} \]  

(4.20)

with \( z \) being the vertical coordinate, \( \omega \) is the frequency (rad/sec), \( \nu_s \) is the friction velocity (m/s), \( n_m \) is the Monin coordinate, \( k_{vk} \) is the Von-kàrmàn’s constant and \( z_o \) is the roughness length. When a continuous structure is discretized into a MDOF system, the modal fluctuating drag force power spectrum, which includes spatial correlation information, is expressed as

\[
S_{D,M_0}(\omega) = (C_D A_T \rho)^2 \sum_{k=1}^{N} \sum_{l=1}^{N} S_{v_{vk}v_{vl}}(\omega) \bar{v}_k \bar{v}_l \phi_{T,j}(k)\phi_{T,j}(l)
\]  

(4.21)

where, \( A_T \) is the total area of the structure exposed to the wind, ‘k’ and ‘l’ are spatial nodes, \( S_{v_{vk}v_{vl}}(\omega) \) is the velocity auto PSDF when \( k=l \) and the cross PSD function when \( k \neq l \), \( \bar{v}_k \) and \( \bar{v}_l \) are the mean wind velocities at nodes \( k \) and \( l \) respectively, and \( \phi_{T,j}(k) \) and \( \phi_{T,j}(l) \) are the nodal \( k \) and \( l \) components of the \( j^{th} \) mode shape. The auto and cross PSD terms is given as

\[
S_{v_{vk}v_{vl}}(\omega) = \sqrt{S_{v_{vk}v_{vk}}(\omega)S_{v_{vl}v_{vl}}(\omega) \co h(k,l;\omega)}
\]  

(4.22)

where, \( S_{v_{vk}v_{vk}}(\omega) \) and \( S_{v_{vl}v_{vl}}(\omega) \) are the velocity auto PSDFs at nodes \( k \) and \( l \) respectively. The spatial coherence function between nodes \( k \) and \( l \) is given by (Simiu and Scanlan, 1978)
where, \( |k - l| \) is the spatial separation and \( L_s \) is the length scale given by

\[
L_s = \frac{\bar{v}}{2\pi \omega D}
\]

In equation (4.24) \( \bar{v} \) is the average mean wind velocity between nodes \( k \) and \( l \) and \( D \) is a decay constant. Any arbitrary fluctuating drag force time-history, \( f_{D,Mj}(t) \), with zero mean, may be represented by a discrete Fourier transform with a discretized version of a continuous frequency content as

\[
f_{D,Mj}(t) = \sum_{k=1}^{\infty} a_k \cos(\omega_k t) + \sum_{k=1}^{\infty} b_k \sin(\omega_k t)
\]

where \( a_k \) and \( b_k \) are the Fourier coefficients, \( \omega_k \) is the discretized frequency and \( t \) is the time instant. Assuming the fundamental mode contributes significantly to the response of the tower (with \( j=1 \)), the total drag force experienced is made up of a mean component and a fluctuating component

\[
F(t) = \bar{f}_{\text{mean},i} + f_{D,Mj}(t)
\]

The mean drag force at node \( \text{\textquoteleft}i\text{\textquoteleft} \) can be represented as

\[
\bar{f}_{\text{mean},i} = 0.5C_D A_i \rho \bar{v}_i^2
\]

where, \( C_D \), \( A_i \), \( \rho \) and \( \bar{v}_i \) are the drag coefficient, area associated with node \( \text{\textquoteleft}i\text{\textquoteleft} \), air density and mean wind velocity at node \( i \), respectively. The total drag forces can be applied at different nodes on the tower.
4.3.1 Wind excitation on rotating blades and coupled blade/tower system

The rotating wind turbine blades are modelled as discretized versions of prismatic cantilevered beams of rectangular hollow cross section. The variation of the mean velocity with rotating blades is expressed as

\[
\bar{v}_i = \bar{v}_i(H) + \bar{v}_{i,\text{ref}} \cos(\Omega t)
\]  

(4.28)

where, \( \bar{v}_i(H) \) and \( \bar{v}_{i,\text{ref}} \) are the mean wind velocity at the respective node height and the varying component of the wind velocity, respectively. Using equation (4.28) in equation (4.27), the mean nodal drag force is calculated. In the case of the rotating blades, the fluctuating nodal drag force is given by

\[
f_i(t) = C_D A_i \rho \bar{v}_i v_i'(t)
\]  

(4.29)

where, \( v_i'(t) \) is the fluctuating velocity component described by equation (4.18). The total force at node \( i \) is obtained from the summation of the mean and fluctuating nodal drag forces. The effective shear force transmitted into the nacelle of the tower due to blade vibration may be given by (Murtagh et al., 2005)

\[
V_{\text{bl}}(t) = V_{\text{bl}}(t) + \ddot{x}_n(t) M_{\text{bl},f}
\]  

(4.30)

where \( M_{\text{bl},f} \) is the mass of the three blades and \( \ddot{x}_n(t) \) is the absolute acceleration at the nacelle of the tower. The total base shear, \( V_{\text{bl}}(t) \), exerted by a rotating blade is equal to the summation of the inertial forces over the entire length of the blade

\[
V_{\text{bl}}(t) = \bar{m}_b \int_0^{L_b} \{\ddot{u}_b(t)\} \, dx
\]  

(4.31)

where \( \bar{m}_b \) is the mass per unit length of the blade, \( L_b \) is the length of the blade and acceleration of the blade, \( \ddot{u}_b(t) \), relative to the nacelle. To account for the effect of
rotating blades on the response of the tower, a sub-structure approach is followed (Murtagh et al., 2005). The response of the rotating blades subjected to wind loading is calculated at first. Then, the coupling of the blades and the tower is accounted for by assuming the effect of the motion at the top of the tower and transfer of shear from the blades to the top of the tower (nacelle).

4.4 WAVE EXCITATION

Data was collected (Hasselmann et al., 1973) and analyzed during the Joint North Sea Wave Observation Project (JONSWAP) and it was found that the wave spectrum continues to develop through non-linear, wave-wave interactions even for very long times and distances compared to the Pierson-Moskowitz spectrum. The JONSWAP spectrum takes into account the higher peak of the spectrum in a storm for the same total energy as compared with Pierson-Moskowitz and also the occurrence of frequency shift of the spectra maximum. The spectrum takes the form

$$S_{\eta\eta}(\omega) = \frac{\alpha g^2}{\omega^5} \exp\left\{-\frac{5}{4}\left(\frac{\omega_m}{\omega}\right)^4\right\} \frac{\exp\left[\frac{(\omega-\omega_m)^2}{2\sigma_w^2}\right]}{\gamma}$$

(4.32)

where, $\eta$ is the function of water surface elevation. Equation (4.32) defines a stationary Gaussian process of standard deviation equal to 1. In equation (4.32), $\gamma$ is the peak enhancement factor (3.3 for the North sea), $g$ is the acceleration of gravity and $\omega$ is the circular wave frequency. The wave data from the JONSWAP project was used to calculate the values of the constants in equation (4.32) as follows

$$\alpha = 0.076 \left(\frac{U_{10}^2}{Fg}\right)^{0.22},$$

(4.33)
\[ \omega_m = 22 \left( \frac{g^2}{U_{10} F} \right)^{\frac{1}{3}}, \]  

(4.34)

and

\[
\sigma = \begin{cases} 
0.07 & \omega \leq \omega_m \\
0.09 & \omega > \omega_m 
\end{cases} 
\]  

(4.35)

where \( U_{10} \) is the mean wind speed 10m from the sea surface, \( F \) (fetch) is the distance from the shoreline of the wave field under consideration. The fetch varies in its non-dimensional form as follows (Ditlevsen, 2002)

\[ 10^{-1} < \frac{gF}{U_{10}^2} < 10^4 \]  

(4.36)

The total wave force acting on the offshore wind turbine is

\[ f_z(t) = \int_0^d p(z,t) \phi_n(z) dz \]  

(4.37)

where, \( \phi_n(z) \) is the shape function of the offshore structure exposed to the wave loading, \( d \) is the depth of the water surface, \( z \) is the vertical coordinate axis, \( p(z,t) \) is the wave force acting on the column that can be calculated by the linearized Morison equation (Sarpkaya and Isaacson, 1981)
\[ p(z,t) = K_d \sqrt{\frac{8}{\pi}} \sigma_v v(z,t) + K_m a(z,t) \]  

(4.38)

with,

\[ K_d = (1/2) C_d \rho d_e \]  

(4.39)

\[ K_m = (1/4) C_m \rho \pi d_e^2 \]  

(4.40)

and \(C_d\) is the drag coefficient, \(C_m\) is the inertia coefficient, \(d_e\) is the equivalent characteristic diameter of the turbine monopole and \(\rho\) is the fluid density. In computing the structural loading associated with the wave, the following assumptions are made: (1) the motion of the structure is \(\ll\) the motion of the wave, so that Morison’s equation (equation 4.2) applies and (2) the flow is predominantly in the inertia regime so that the structural loading term of Morison’s equation that involves \(C_M\) dominates the fluid drag term that involves \(C_d\). Following the linear wave theory, the horizontal velocity \(v(z,t)\) and acceleration \(a(z,t)\) of the water particle are both functions of the wave elevation (Li, 2003)

\[ v(z,t) = \omega \frac{\cosh(kz)}{\sinh(kd)} \eta(t) \]  

(4.41)

\[ a(z,t) = -j \omega^2 \frac{\cosh(kz)}{\sinh(kd)} \eta(t) \]  

(4.42)

and
\[ \omega^2 = gk \tanh(kd) \] (4.43)

where \( k \) is the wave number that can be determined from the linear dispersion relationship. The standard deviation of the velocity at location \( z \) can be obtained as

\[ \sigma_v = \left[ \int_0^\omega |T_v|^2 S_{\eta\eta}(\omega) d\omega \right]^{1/2} \] (4.44)

in which

\[ T_v = \omega \frac{\cosh(kz)}{\sinh(kd)} \] (4.45)

Substituting the appropriate terms into equation (4.37) yields the total wave force acting on the structure,

\[ f_w(t) = \int_0^d \left[ K_d \frac{8}{\pi} \sigma_v T_v + K_m T_a \right] \phi_w(z) dz \eta(t) \] (4.46)

where,

\[ T_a = -j \omega^2 \frac{\cosh(kz)}{\sinh(kd)} \] (4.47)
As the surface elevation of the sea rises and falls, the force is lumped proportionally into the 3 nodes at the low, mean and high water levels, which can be seen in Figure 4.1.

4.5 JOINT DISTRIBUTION OF WIND AND WAVES

The JONSWAP spectrum defined in Section 4.4 is a stationary Gaussian process and can be mapped into the process of the sea state defined by \((H_s, T_z)\) by letting the dimensionless time be \(t/T_z\) and the dimensionless process be \(X/\sqrt{\lambda_0} = 4X/H_s\) (Ditlevsen, 2002). The wind speed at 10 metres, \(U_{10}\), and the significant wave height from the JONSWAP spectrum, \(H_s\), can be related through the integral of equation (4.32)

\[
\sigma^2 = \int_0^{\infty} S_{\eta \eta}(\omega) d\omega = \lambda_0
\]

(4.48)

where, \(\sigma\) is the standard deviation of surface displacement. If a sea contains a narrow range of wave frequencies, \(H_s\) is related to the standard deviation of the sea surface displacement (Hoffman, 1975)

\[
H_s = 4\sigma
\]

(4.49)

The sample functions used for analysis in the joint distribution of wave period and height are approximated by the trigonometric polynomial (Colwell and Basu, 2006b)

\[
X(t) = \sum_{k=1}^{n} \sqrt{\left(S_{\eta \eta}(\omega_k) d\omega\right)} (X_{1k} \cos \omega_k t + X_{2k} \sin \omega_k t)
\]

(4.50)

where \(X_{1k}\) and \(X_{2k}\) for \(k=1, \ldots, n\) are mutually independent standard normal variables and \(\omega_{k+1} = \omega_k + d\omega\), with \(d\omega\) being an infinitesimal frequency step.
4.6 Fatigue Analysis

The method used to calculate the degree of fatigue wear is the rainflow-counting algorithm (Matsuiski and Endo, 1969). The rain-flow method is used in the analysis of fatigue data in order to reduce a spectrum of varying stress into a set of simple stress reversals. Its importance is that it allows the application of Miner’s Rule (equation (4.52)) in order to assess the fatigue life of a structure subject to complex loading. The fatigue evaluation applies Miner’s law considering the plate thickness effect and environmental conditions in accordance with (DNV, 2004). The fatigue curve (S-N chart) that was applied (Shiraishi et al., 2006) is given by

$$\log_{10} N = \log_{10} a - m \log_{10} \Delta \sigma \left( \frac{t}{t_{ref}} \right)^k \quad (4.51)$$

where, N is the allowed cycles of the stress range $\Delta \sigma$ (MPa), $\log_{10} a$ is the intercept on the $\log_{10} N$ axis for the S-N curve, m is the coefficient that represents the inclination, $t_{ref}$ is the reference plate thickness, t is the plate thickness, and k is an index parameter.

The degree of fatigue damage is given by

$$D_D = \sum_{i=1}^{I} \frac{n_{c,i}}{N_{D,i}} \quad (4.52)$$

and the fatigue life is given by

$$Y = \frac{1}{D_D} \quad (4.53)$$

where, $n_{c,i}$ is the number of stress cycles in the ith stress block and $N_{D,i}$ is the number of cycles to failure at the design stress range of the ith stress block interpreted from the characteristic long-term distribution of stress ranges.
4.7 NUMERICAL EXAMPLES

The time histories and power spectral densities of the wind and wave loadings are presented first. Next, the MDOF offshore structure with and without TLCD is excited by wind loading, followed by combined wind and wave loading. Finally, the MDOF model with rotating blades and TLCD is subjected to a combined wind and wave loading. Figure 4.1 shows the structural model under consideration with the corresponding idealized lumped mass MDOF system, with each node separated by massless elements having finite stiffness (Rogers, 1998). The foundation is assumed to be fully fixed into the ground with no rotation allowed. The base of the tower, \( d_e \), is 4.3m in diameter, tapering to 3.5m at the nacelle. The steel thickness at the bottom and the top of the structure is 18mm and 10mm, respectively. Steel with an elastic modulus equal to \( 210 \times 10^9 \) N/m² was assumed to be used in the construction of the turbine. The structural damping ratio is assumed to be 0.01.

4.7.1 Wind and Wave excitation

Sample time histories and the power spectral densities (PSD) for both the wind and the wave excitations are presented in Figure 4.2 to 4.9. The mean wind velocities at the top of the tower in Figure 4.2 and Figure 4.6 were taken as 18m/s and 30m/s, respectively. The density of air, the coefficient of drag and the roughness length for the water surface were taken as 1.225 kg/m³, 1.2 and 0.002, respectively. The wind spectrum covered a frequency range of 0-600Hz. Time histories of length 50 sec are presented. Figure 4.2 shows the time history of the drag force for the wind excitation at node 7. Figure 4.3 shows the target and simulated PSDs for the wind spectrum. It can be seen that in both the wind and the wave excitation cases, the target power spectral densities show close agreement with the simulated PSDs (Figure 4.3 and Figure 4.5)
Figure 4.2 Time series for wind excitation at Node 7

Figure 4.3 PSDF for the wind excitation
Two types of wave spectra were used in the analysis, a ‘moderate’ wave time history and a ‘strong’ wave time history. The following parameter values were used in the ‘moderate’ wave spectra simulation: \( k_0 = 1, l_s = 1200 \text{m}, C_d = 1.2, \rho_0 = 1.2 \text{kg/m}^3, z = 13 \text{m}, \alpha_p = 0.23, U_{10} = 18 \text{ (m/s)}, g = 9.81 \text{ m/s/s}, F = 85000 \text{m} \) and \( \gamma = 3.3 \). The resulting time history of the surface elevation of the sea for the ‘moderate’ wave loading can be seen in Figure 4.4. The PSDF for the moderate wave excitation can be seen in Figure 4.5. The nodal drag force for the ‘moderate’ wave excitation at node 3 is presented in Figure 4.6.

![Figure 4.4 Time series for the ‘moderate’ wave excitation](image-url)
Figure 4.5 PSDF of ‘moderate’ wave elevation

Figure 4.6 Wave force at node 3 for ‘moderate’ wave excitation
The following parameter values were used in the 'strong' wave spectra simulation: $k_0 = 1$, $l_x = 1200$ m, $C_d = 1.2$, $\rho_0 = 1.2$ kg/m$^3$, $z = 40$ m, $\alpha_p = 0.23$, $U_{j0} = 30$ (m/s), $g = 9.81$ m/s/s, $F = 85000$ m and $\gamma = 3.3$. The resulting time history of the surface elevation of the sea for the 'strong' wave loading can be seen in Figure 4.7. The PSDF for the moderate wave excitation can be seen in Figure 4.8. The nodal drag force for the 'moderate' wave excitation at node 3 is presented in Figure 4.9.

Figure 4.7 Time series for the 'strong' wave excitation
Figure 4.8 PSDF of 'strong' wave elevation

Figure 4.9 Wave force at node 3 for 'strong' wave excitation
4.7.2 MDOF Tower with Blades lumped at Nacelle under ‘Moderate’ Wind Excitation

The structure adopted in the numerical study is an offshore wind turbine of total height 100m. The turbine tower with the blades lumped at the nacelle is first considered in order to investigate the performance of a TLCD on a cantilevered structure subject to wind excitation. The average wind speed used in this section is 30m/s and the wave force applied at node 7 can be seen in Figure 4.2. The fundamental natural frequency of the system, to which most of the tip displacement may be attributed to, for the structure without TLCD is 1.1 rad/s.

The parameters of the TLCD were optimised to impart the highest damping on the structural system (Gao et al., 1997). The mass of the TLCD was 1% of the total mass of the structure. The natural frequency of the TLCD was tuned to 99.2% of that of the fundamental frequency of the MDOF structure. The total length of the liquid column in the TLCD was 15.2m. The breath, \( B \), which is the horizontal part of the TLCD, is taken as 2.5m, leaving vertical columns of 6.35m. In terms of accommodating the TLCD in an offshore wind turbine, the vertical columns could be located in the tubular tower below the nacelle or within the tubular tower and through the nacelle. Another option would be to locate the TLCD either on top of or at the side of the nacelle. The values of \( \xi \) and \( \alpha \) were 30 and 0.2, respectively. The larger the value of \( \alpha \), which is the ratio of the breath of the TLCD to that of the length of the liquid column, the more effective the TLCD. However, in the case of a TLCD in a wind turbine, horizontal space is at a premium so one must use a lower value of \( \alpha \) than is ideal in order to tune the TLCD to the natural frequency of the wind turbine.

Figure 4.10 shows the mode shapes of the structure. Figure 4.11 displays a segment of the time history responses for both the MDOF structure without TLCD and MDOF structure with TLCD in the case of a fluctuating wind of mean 30m/s. A reduction from the tip displacement of approximately 35.5% is obtained by installing the TLCD in the offshore wind turbine subjected to wind loading and no wave loading. For the theoretical case under consideration, the tip design bending moment at the base from...
the contribution of the vibrating tower is reduced from $3.1234 \times 10^5$ kNm to $1.7312 \times 10^5$ kNm.

Figure 4.10 Mode shapes of MDOF system
In this section, the MDOF offshore wind turbine with and without TLCD is excited by a combined wind and wave loading. The wave loading in Figure 4.9 is the input wave excitation in this case. As the surface elevation increases and decreases, the total force from the sea is lumped into the appropriate nodes. Figure 4.12 presents the time-history response of the MDOF system subjected to a combined wind and wave loading. It can be seen that the interaction of the wind and wave forces increases the tip response of the structure without TLCD by approximately 55% at its tip. In the time segment presented, the influence of the TLCD is evident in reducing the tip response by 38% at approximately 1.1 seconds. The peak bending moment for the turbine without damper is $4.1356 \times 10^5$ KNm as opposed to $2.49 \times 10^5$ KNm for the turbine with TLCD.
4.7.4 MDOF Tower with Blades lumped at Nacelle under ‘Strong’ Wind and Wave Excitation

The MDOF structure (with and without TLCD) with rotating blades was subjected to a wind and wave loading. The wind velocity taken at a height of 10m was 18m/s. The blades were 60m in length and individually weighed 9.5 tonnes. The altered fundamental frequency for the tower – blade system is 0.93 rad/s. Thus, the length of the liquid column is 22.6m, and the value of $\alpha$ is 0.11. The time history response of the aforementioned system is given in Figure 4.13.

It can be seen that in times of moderate wind and wave loading, the blades impart the largest loading on the structural tower. The response of the tower with TLCD shows a marked reduction when a TLCD is employed, with reductions of approximately 60% in the response evident. The maximum design bending moment for the theoretical...
simulation at the base of the structure from is reduced from $6.2607 \times 10^4$ kNm to $4.0101 \times 10^4$ kNm.

![Graph showing time-history response with and without TLCD](image)

**Figure 4.13** MDOF, with rotating blades, time-history response under 'moderate' wind and wave excitation with and without TLCD

### 4.7.5 MDOF Tower with Rotating Blades under ‘Strong’ Wind and Wave Excitation

Next, the MDOF (with and without TLCD) structure with rotating blades was subjected to a stronger wind and wave loading. The applied wind and wave loadings in this case can be seen in Figure 4.2 and Figure 4.6, respectively. The blades rotated at a frequency equal to the fundamental natural frequency of the MDOF system. The blades were 60m in length and individually weighed 9.5 tonnes. The altered fundamental frequency for the tower – blade system is 0.93 rad/s. Thus, the length of
the liquid column is 22.6m, and the value of $\alpha$ is 0.11. The time history response of the aforementioned system is given in Figure 4.14.

Figure 4.14 MDOF, with rotating blades, time-history response under 'strong' wind and wave excitation with and without TLCD

For the MDOF system with optimised TLCD, a maximum reduction of 55% is recorded as compared to that of the MDOF system without TLCD. The maximum design bending moment for the theoretical simulation at the base of the structure from is reduced from $5.35 \times 10^5$ kNm to $2.99 \times 10^5$ kNm. The variation of the base moment for the aforementioned numerical case is presented in Figure 4.15.
Figure 4.15 Base moment of MDOF, with rotating blades, time-history response under ‘strong’ wind and wave excitation with and without TLCD

4.8 FATIGUE LIFE

This study calculates the fatigue life of the turbine tower with and without the TLCD using the rain-flow calculation method. The tower locations chosen for the fatigue analysis are the butt weld connection connecting the tower to the mono-pile foundation (which in this case is taken as 1 metre below the platform mass) and the connection of the mono-pile foundation with the sea-bed (which is taken as 1m above the point of fixity). The following specific set values are taken from (Shiraishi et al., 2006)

a) Conditions in the atmosphere
   - $\log_{10} a = 12.164$, $m = 3$, $k = 0.20$ ($N<10^7$)

b) Undersea conditions (with electrolytic protection)
   - $\log_{10} a = 11.764$, $m = 3 \gamma$, $k = 0.20$, $\gamma$ (material coefficient) = 1.25 ($N<10^7$)
An example of the methodology of the rainflow calculation method can be seen in Figure 4.16. The axis is presented as such to portray the rainflow method intuitively. Table 4.1 presents the cumulative fatigue damage rate and fatigue life obtained by the rain-flow calculation method. The results of the fatigue life calculations indicate that the life span of a wind turbine assembly with TLCD is substantially longer to that of a wind turbine assembly without TLCD.

<table>
<thead>
<tr>
<th>Location</th>
<th>Case</th>
<th>Cumulative fatigue damage rate (year)</th>
<th>Fatigue life (years)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mono-pile tower</td>
<td>Tower without TLCD</td>
<td>0.00117</td>
<td>895</td>
</tr>
<tr>
<td>Tower with TLCD</td>
<td>0.00014</td>
<td>3192</td>
<td></td>
</tr>
<tr>
<td>Mono-pile seabed</td>
<td>Tower without TLCD</td>
<td>0.012192</td>
<td>82</td>
</tr>
<tr>
<td>Tower with TLCD</td>
<td>0.000906</td>
<td>390</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.1 Degree of fatigue damage for wind turbine assembly with and without TLCD
Figure 4.16 Rainflow Calculation Method
4.9 Economic Implications with Regards to Reduced Deflections and Bending Moments

The main parameters governing wind power economics have been found to be (EWEA, 1998)

1) Initial investment costs (including foundations, wind turbine tower, grid-connection etc.)
2) Operation & Maintenance costs
3) Electricity / average wind speed
4) Turbine lifetime
5) Discount rate

The implementation of a TLCD in an offshore wind turbine directly influences factors 1 and 4, in regards to decreased foundation and turbine tower costs and longer turbine life, respectively.

In the average cost of a wind turbine project, foundations costs represent 16% of the total cost and wind turbines 51%. In the Vindeby (DEA, 2000) offshore wind farm, the foundations cost 2.3 million Euro, including installation, out of a total of 10.26 million Euro.

One may approximate the savings of monopole construction by implementing TLCDs by benchmarking moment capacity of turbines through a report by DOWEC (Hendriks, 2004). By decreasing the maximum bending moment by 50%, the diameter of the concrete monopile is reduced from 6m to 5m, with the wall thickness being reduced from 60mm to 50mm.

To quantify the savings in tower steelwork, one may simulate the turbine, without TLCD, with rotating blades under wind and wave loading, to find the tower structure needed to produce a response similar to that of the turbine with TLCD. It was found that a tower with base diameter 5.5m of thickness 22mm, and diameter of 4m connecting to the nacelle, with thickness 12mm, produces the same response of the previous tower arrangement with TLCD (4.3m and 18mm diameter and thickness at
base, respectively, and 3.5m diameter and 10mm thickness at the tip, respectively). Apart from saving money in the fabrication of the turbine tower, the constructability of the giant towers is eased. Transportation of members by road is limited by the size of the road network. Thus, reducing the diameter of the turbine tower will make larger wind turbines realizable.

4.10 CONCLUSIONS

This chapter examined the excitation of an offshore wind turbine modelled as a MDOF system under wind and wave loadings. The Kaimal spectrum was chosen to represent the wind excitation and JONSWAP spectrum was used for the wave excitation. Cases of the blades lumped at the nacelle and of rotating blades were examined. A correlation between the wind and wave excitations using joint distribution was presented.

It was found that when an offshore wind turbine is equipped with a TLCD and subjected to wind and wave forces, reductions of up to 55% in the tip response of the same system without TLCD may be achieved. The TLCD performs consistently well, in terms of reducing base bending moments and tip responses in the structure, across all excitations considered. The option open to design the wind turbine more efficiently with less steelwork and less foundation expenses is thus afforded when one implements TLCDs in offshore wind turbines.

It was also observed that through use of the rain-flow calculation method for fatigue, that implementation of a TLCD in an offshore wind turbine greatly increases the fatigue life of the wind tower assembly.
Chapter 5

A HYBRID BI-TMD-TLCD SYSTEM IN OFFSHORE WIND TURBINES

5.1 INTRODUCTION

In Chapters 2 and 3, investigations into parameters that optimize the performance of a TLCD and the passive damping properties of various fluids, including magnetorheological (MR) fluid, in the TLCD were undertaken. Following on, in Chapter 4, the structural response of offshore wind turbines was simulated with attached damper (TLCD) for controlling the vibrations induced within the structure.

The thrust of this chapter is an investigation into a novel design for an offshore wind turbine that incorporates a number of passive dampers. Base isolation (BI) has traditionally been used in an aseismic design approach in which the building is protected from the hazards of earthquake forces by a mechanism which reduces the transmission of horizontal acceleration into the structure. The main purpose of the BI system is to reduce the natural frequency of structural vibration to a level below the predominant energy-containing frequencies of the excitation. A secondary purpose of the BI system is to provide an additional means of energy dissipation, thereby reducing the transmitted acceleration into the system. The use of the BI system in reducing vibrations in an offshore wind turbine is investigated in this chapter.

The sensitivity of a structure with base-isolation system to wind loads is increased as base-isolation generally increases the horizontal flexibility of a structure. Thus, the addition of TMDs into the system is both necessary to reduce the lateral displacement of the structure and advantageous in reducing the effect of wind and wave excitations. In conjunction with the BI system and TMD, TLCDs are used to reduce the vibration within the offshore wind turbine.

Finally, an innovative method of acquiring additional energy from the overall structural system is proposed.
5.2 **BASE ISOLATION**

Base Isolation systems that adopt the use of elastomeric bearings, with the elastomer usually made of either natural rubber or neoprene, have been predominant in recent years. In this approach traditionally used for seismic protection, the structure is decoupled from the horizontal components of the earthquake ground motion by interposing a layer with low horizontal stiffness between the structure and the foundation. This layer gives the structure a fundamental frequency that is much lower than its fixed-base frequency and also much lower than the predominant frequencies of the ground motion.

The first dynamic mode of the isolated structure involves deformation only in the isolation system, the structure above being to all intents and purposes rigid. The higher modes that will produce deformation in the structure are orthogonal to the first mode and consequently also to the ground motion. These higher modes do not participate in the motion, so that if there is high energy in the ground motion at these higher frequencies, this energy cannot be transmitted into the structure. The isolation system does not absorb the earthquake energy, but rather deflects it through the dynamics of the system.

As the natural frequency of offshore wind turbines may be close to the frequency where the wave excitation’s energy is concentrated, the applicability of the base isolation isolating the structure primarily from the wave loadings is investigated.

The shear rigidity and the number of bearings to be used in the base isolation system is determined from the desired isolation period (Kelly, 1986)

\[ T_b = 2\pi \left( \frac{W_b}{K_b g} \right)^{1/2} \]  

(5.1)

where \( W_b \) is the static load. The stiffness of the rubber bearings may be represented by
where \( G_b A_{sb} \) and \( L_b \) represent shear rigidity and length of an individual bearing, respectively.

5.3 TUNED MASS DAMPERS

The TMD works through the addition of a small mass (<5% of total mass of structure) connected to a structure with a spring and dashpot. The net effect of adding the small mass on the structure, aside from a slight decrease in natural frequency and a slight increase in external force, is the addition of a 'force term'. The TMD is optimised when this 'force term' lags that of the structural system by a phase angle of 90°. The resultant effect is energy dissipation through the vibration of the TMD.

An investigation has concluded that previous studies performed ((Warburton, 1981) and (Kareem, 1983)) suggest the same optimal tuning parameters (tuning ratio and damping) for a TMD in a structure with BI system agree with those for a SDOF structure with fixed base. However, for a base isolated structure with TMDs, the results deviated from conventional values used for sizing dampers (Kareem, 1997). The introduction of base isolation into a system fundamentally changes the distribution of structural parameters, resulting in non-uniform values of tuning ratio and damping for differing structure heights. As a consequence, the optimization of TMD parameters for each base-isolated structural system must be performed individually (Kareem, 1995).

For comparison, the optimum tuning frequency and damping ratio of the TMD which can minimize the steady-state response of the damped primary structure have been derived as (Tsai, 1995)
\[
\chi_{\text{TMD}} = \left( \frac{\sqrt{1-0.5\mu_{\text{TMD}}} + \sqrt{1-2\zeta^2}}{1+\mu_{\text{TMD}}} \right) - \left( 2.375 - 1.034\sqrt{\mu_{\text{TMD}}} - 0.426\sqrt{\mu_{\text{TMD}}} \right) \sqrt{\mu_{\text{TMD}}} \zeta \\
- \left( 3.730 - 16.903\sqrt{\mu_{\text{TMD}}} - 20.496\sqrt{\mu_{\text{TMD}}} \right) \sqrt{\mu_{\text{TMD}}} \zeta^2
\]

and

\[
\zeta_{\text{TMD}} = \sqrt{\frac{3\mu_{\text{TMD}}}{8(1+\mu_{\text{TMD}})(1-0.5\mu_{\text{TMD}})}} + (0.151\zeta - 0.170\zeta^2) \\\n+ (0.163\zeta - 4.980\zeta^2) \mu_{\text{TMD}}
\]

where \( \mu_{\text{TMD}} \) is the mass ratio of the TMD to the structure.

**5.4 MODELLING OF OFFSHORE WIND TURBINE WITH MULTIPLE DAMPERS**

Section 5.4 models an offshore wind turbine with a TMD, TLCD and a BI system. The BI system connects the platform, which is above sea level, to the tubular structural tower. The TMD is connected to the BI system for added lateral stiffness and displacement reduction at the platform. The TLCD is contained within the nacelle displacement degrees of freedom of SS1 and SS2 are relative to the sea bed. Each individual blade, which is 60m long, is discretized into a 3DOF system. The blades are decoupled from the tower, with the interaction between the tower and the blades occurring in the time domain through a transmitted shear force (equation (4.30)). In simulating the coupling of the tower and the blades, the base shear from the MDOF blade system is first calculated, then coupled with the vibrating tower to update the overall shear force transmitted into the structure.
Figure 5.1 Schematic of the structural system
In terms of the labelling of the global system nodes, the node numbers are organized into an ascending order from the base of SS1. Thus, SS1 comprises of nodes 1-4, the BI system is node 5, the TMD is node 6, SS2 is comprised of nodes 7-9, and the TLCD represents node 10. When labelling the local nodes of SS1 and SS2, the nodes are also labelled in ascending order. For example, the node connecting SS1 with the BI system, and the node connecting SS2 with the TLCD, are labelled as SS1,4 and SS2,3, respectively. A schematic of the system is presented in Figure 5.1.

5.4.1 Construction of System Representation

The structure SS1 is connected to the base isolation system. The force opposing the movement of SS1 is the relative motion between the BI system and SS1. Thus, the first four equations of motion for the system are

\[
\{\ddot{x}_{SSI}(t)\} + \left(2\left[\zeta_{n,SSI}\right] + \left[\Gamma_{BI}\right]\right)\{\ddot{x}_{SSI}(t)\} \\
-\left[\Gamma_{BI}\right]\ddot{x}_{BI}(t) + \left(\left[\omega_{n,SSI}\right]^2 + \left[\Gamma_{BI}\right]\right)\{\dot{x}_{SSI}(t)\} \\
-\left[\Gamma_{BI}\right]x_{BI} = \left[M_{n,SSI}\right]^{-1}\{F_{in,SSI}(t)\}
\]  

where

\[
\left[\Gamma_{BI}\right] = \text{trace}(M_{4,SSI})^{-1}C_{BI}R_{BI,SSI},
\]

\[
\left[\Gamma_{BI}\right] = \text{trace}(M_{4,SSI})^{-1}K_{BI}R_{BI,SSI},
\]

and

\[
R_{BI,SSI} = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]
where \([K_{SS1}], K_{BI}, C_{BI}, \{X_{SS1}\}\) and \(X_{BI}\) represent the stiffness matrix of SS1, the stiffness of the BI system, the damping of the BI system, the displacement vector of SS1 and the displacement of the BI system, respectively. The natural frequency and damping of SS1 is represented by \(\omega_{n,SS1}\) and \(\zeta_{n,SS1}\), respectively. \(\{F_{m,SS1}(t)\}\) represents the overall force acting on SS1.

The BI system is connected to SS1, SS2 and the TMD. The base motion of the BI system is assumed to be the same as the top node of SS1 (node: SS1,4). The relative motion of SS2 and the TMD are assumed to oppose the direction of movement to that of the BI system. The equation of motion for the sub-system is given as

\[
\ddot{x}_{BI}(t) + \left( C_{BI} + C_{SS2,1} + C_{TMD} \right) M_{BI}^{-1} \dot{x}_{BI}(t) - M_{BI}^{-1} C_{TMD} \dot{x}_{TMD} \\
- C_{TMD} M_{BI}^{-1} C_{SS2,1} x_{SS2,1} - 2 \zeta_{BI} \omega_{BI} x_{SS1,4} \\
+ \left( K_{BI} + K_{SS2,1} + K_{TMD} \right) M_{BI}^{-1} x_{BI}(t) - M_{BI}^{-1} K_{TMD} x_{TMD} \\
- M_{BI}^{-1} K_{SS2,1} x_{SS2,1} - \omega_{BI}^2 x_{SS1,4} = 0
\]  

(5.9)

where \([K_{SS2,1}], K_{TMD}, C_{TMD}, \{X_{SS2,1}\}, \{X_{SS1,4}\}\) and \(X_{TMD}\) represent the nodal stiffness of SS2,1, the stiffness of the TMD, the damping of the TMD system, the displacement vector of node SS2,1, the displacement vector of node SS1,4 and the displacement vector of the TMD, respectively. The natural frequency and damping of the BI system are given by \(\omega_{BI}\) and \(\zeta_{BI}\), respectively. Structure SS2 is connected to the BI system and the TLCD. For the sub-system SS2, three equations of motion are given as

\[
\begin{align*}
\left( \begin{bmatrix} I_{m,SS2} & \mu_{SS2} \end{bmatrix} \right) \{\ddot{x}_{SS2}(t)\} + 2 \zeta_{n,SS2} \omega_{n,SS2} \{\dot{x}_{SS2}(t)\} - \dot{x}_{BI}(t) \{R_{BI}\} \\
+ \omega_{n,SS2}^2 \{\dot{x}_{SS2}(t)\} - x_{BI}(t) \{R_{BI}\} = [M_{n,SS2}]^{-1} \{F_{m,SS2}(t)\} - \mu \alpha \ddot{u} \{R_{TLCD}\}
\end{align*}
\]  

(5.10)
where

\[
\mu_{SS2} = \begin{bmatrix}
0 & 0 & 0 \\
0 & 0 & 0 \\
0 & 0 & \mu
\end{bmatrix}, \quad (5.11)
\]

\[
R_{BI} = \begin{bmatrix}
1 \\
0 \\
0
\end{bmatrix}, \quad (5.12)
\]

\[
R_{TLCD} = \begin{bmatrix}
0 \\
0 \\
1
\end{bmatrix} \quad (5.13)
\]

and

\[
[I_{m,SS2}] = \begin{bmatrix}
1 & 0 & 0 \\
0 & 1 & 0 \\
0 & 0 & 1
\end{bmatrix} \quad (5.14)
\]

The natural frequency and damping of SS2 are given by \( \omega_{n,SS2} \) and \( \zeta_{n,SS2} \), respectively. \( \{F_{in,SS2}(t)\} \) represents the overall force acting on SS1. In addition to damping excitations in the system, the TMD serves as a stiffener for the BI system, to which it is connected

\[
\ddot{x}_{TMD}(t) + 2\zeta_{TMD}\omega_{TMD}\dot{x}_{TMD}(t) + \omega_{TMD}^2 x_{TMD}(t) - M_{TMD}^{-1} C_{BI} \dot{x}_{BI}(t) - M_{TMD}^{-1} K_{BI} x_{BI}(t) = 0 \quad (5.15)
\]

The natural frequency and damping of the TMD are given by \( \omega_{TMD} \) and \( \zeta_{TMD} \), respectively.
The equation of motion for the TLCD system is

$$\ddot{u} + \frac{\xi}{2L} |\dot{u}| \dot{u} + \omega_n^2 u = -\alpha \left\{ \ddot{x}_{SS2,3} \right\}$$  \hspace{1cm} (5.16)$$

where $\ddot{x}_{SS2,3}$ and $u$ represent the absolute acceleration of the nacelle to SS and the displacement response of the TLCD liquid, respectively. The TLCD parameters $\xi$, $L$, $\omega_n$ and $\alpha$ represent the coefficient of head loss, the length of the liquid column, the natural frequency of the TLCD and the ratio of the breath of the liquid column to the overall length of the liquid column. As in Section 4.2, $X=\Phi \eta$ is inserted into the system equations so that the structural displacement response is represented as a product of the mode shape matrix, $\Phi$, normalized with respect to the structural mass and the $n$-dimensional modal coordinate vector, $\eta$. When one combines Equations (5.5) to (5.15), the overall system equation becomes

$$M_{\text{TOTAL}} \ddot{X}_{\text{TOTAL}} + C_{\text{TOTAL}} \dot{X}_{\text{TOTAL}} + K_{\text{TOTAL}} X_{\text{TOTAL}} = F_{\text{TOTAL}}$$ \hspace{1cm} (5.17)$$

The global mass matrix is composed as

$$M_{\text{TOTAL}} = \begin{bmatrix} I_{m,SS1} & 0 & \cdots & 0 \\ 0 & 1 & \cdots & \vdots \\ \vdots & \ddots & \ddots & \vdots \\ 0 & \cdots & 0 & I_{m,SS2} + \mu_{SS2} \end{bmatrix}$$ \hspace{1cm} (5.18)$$

where

$$I_{m,SS1} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ \hspace{1cm} (5.19)$$

and

$$I_{m,SS2} + \mu_{SS2}$$
The global damping matrix is

\[
C_{\text{TOTAL}} = \begin{bmatrix}
[\Theta_{SS1}] + [C_{BI,SSI}] & 0 & -C_{BI,V} & 0 \\
2\zeta_{TMD}\omega_{TMD} & -2\zeta_{TMD}\omega_{TMD} & C_{BI,INT} & 0 \\
-2\zeta_{BI}\omega_{BI} R_{SS1}^T & -M_{BI}^1 C_{TMD} & -M_{BI}^1 C_{SS2,1} R_{BI}^T & \xi \\
0 & -\Theta_{SS2,1} & \frac{\xi}{2L} & [\Theta_{SS2}]
\end{bmatrix}
\]

(5.20)

where

\[
[\Theta_{SS1}] = 2[\zeta_{n,SS1}] [\omega_{n,SSI}],
\]

(5.21)

\[
[\Theta_{SS2}] = 2[\zeta_{n,SS2}] [\omega_{n,SSI}],
\]

(5.22)

\[
\Theta_{SS2,1} = 2\zeta_{SS2,1}\omega_{SS2,1} R_{BI}^T,
\]

(5.23)

\[
R_{SS1} = \begin{bmatrix}
0 \\
0 \\
0 \\
1
\end{bmatrix},
\]

(5.24)

\[
[C_{BI,SSI}] = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & \text{trace}([M_{SS1,4}])^{-1} C_{BI}
\end{bmatrix},
\]

(5.25)

\[
C_{BI,V} = \text{trace}([M_{SS1,4}])^{-1} C_{BI} R_{SSI}
\]

(5.26)

and

\[
C_{BI,INT} = M_{BI}^{-1} (C_{BI} + C_{SS2,1} + C_{TMD})
\]

(5.27)
In equation (5.23), $\zeta_{n,SS2}$ and $\omega_{n,SS2}$ represent the damping and natural frequency of the first node in SS2 (global mode 7). The global stiffness matrix is

$$K_{\text{TOTAL}} = \begin{bmatrix}
\omega^2_{n,SSI} + \omega^2_{BI,SSI} & 0 & -K_{BI,V} & 0 \\
-\omega^2_{BI,SSI} & \omega^2_{TMD} & -\omega^2_{TMD} & 0 \\
-M^T_{BI}K_{TMD} & K_{BI,INT} & -M^{-1}_{BI}K_{SS2, INT}R^T_{BI} & 0 \\
0 & -\omega^2_{SS2, INT}R^T_{BI} & \left[\omega^2_{n,SSI}\right] & \omega^2_L
\end{bmatrix} \quad (5.28)$$

where

$$\left[\omega^2_{BI,SSI}\right] = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & \text{trace}(\left[M_{SSI,4}\right])^{-1}K_{BI}
\end{bmatrix} \quad (5.29)$$

$$K_{BI,V} = -\text{trace}(\left[M_{SSI,4}\right])^{-1}K_{BI}R_{SSI}, \quad (5.30)$$

$$K_{BI,INT} = M^{-1}_{BI}(K_{BI} + K_{SS2, INT} + K_{TMD}), \quad (5.31)$$

$$X_{\text{TOTAL}} = \begin{bmatrix}
\eta_{SSI} \\
x_{TMD} \\
x_{BI} \\
\eta_{SS2} \\
u
\end{bmatrix} \quad (5.32)$$

$$F_{\text{TOTAL}} = \begin{bmatrix}
F_{m,SSI}(t) \\
0 \\
0 \\
F_{m,SS2}(t) \\
0
\end{bmatrix} \quad (5.33)$$
In equation (5.32), \( \eta_{s1} \) and \( \eta_{s2} \) represent the modal coordinate vectors of SS1 and SS2, respectively. The \( F_{\text{TOTAL}} \) vector is the input force vector in the time domain which represents the wind excitation and wave excitation. The coupled effect of the blade-tower interaction is also represented in the \( F_{\text{TOTAL}} \) vector. In simulating the coupling of the tower and the blades, the base shear from the MDOF blade system is first calculated, then coupled with the vibrating tower to ascertain the overall shear force transmitted into the structure.

The wind excitation is generated as in Section 4.3. However for the present simulation, spatial correlation is accounted for by coherence function and simulating the excitation in the time domain. As the input PSDF has cross drag terms, spectral decomposition theory and random processes with orthogonal increments are used to generate spatially correlated time histories and apply these at respective nodes in SS1. The wave excitation is applied as in Section 4.4.

5.5 ADDITIONAL ENERGY GENERATION

The flow of air over the blades and through the rotor area makes a wind turbine function. Fundamentally, the wind turbine extracts energy by slowing the wind down. If the blades were 100% efficient, a wind turbine would not work because the air, having given up all its energy, would entirely stop. The Betz limit of energy conversion dictates that the maximum efficiency of the conversion of wind energy in a wind turbine is 59.3%. Typical energy conversion efficiencies for modern day wind turbines lie in the range 60-70%. Including all the further processes in converting the wind energy to electrical energy, overall energy conversion efficiencies lie in the range 30-40%. Thus, there is a perennial limit on the energy conversion efficiency of
turning wind into energy in wind turbines. The proposed design of the damping system for the wind turbine in this chapter creates additional avenues for utilizing the characteristics of the wind turbine to produce and transmit additional energy from the wind turbine.

When wind and wave forces excite the structure, the vibrational energy in the tower is dissipated in a number of ways, one of which is the translational movement of the TMD. As the TMD provides the main lateral stiffness for the BI system, it is reasonable to assume that an optimally tuned TMD will absorb a substantial amount of energy over the lifetime of the offshore wind turbine.

The instantaneous energy of the TMD system is given by

\[
\text{Energy}_{\text{TMD}} = \frac{1}{2} K_{\text{TMD}} x_{\text{TMD}}^2 + \frac{1}{2} C_{\text{TMD}} \dot{x}_{\text{TMD}}^2
\]  

(5.35)

By integrating the instantaneous energy of the TMD over an hour, one may find the total power (in W/hr) that the TMD absorbs from the wind and wave excitations.

### 5.6 NUMERICAL EXAMPLES

In Section 5.6, numerical results for are presented for the WTT-BI-TMD-TLCD system subjected to excitations from the rotating blades, wind force acting on the structure and wave force acting on the structure.

#### 5.6.1 Wind and Wave Excitation

In Section 5.6.1, the time histories and power spectral densities of the wind and wave loadings are presented. Two wind speeds, which relate the wind and wave excitations through Equation (4.33) are investigated. The first case in Section 5.6.1.1 simulates a wind loading of mean speed 30 m/s, which in preliminary calculations was shown to induce an acceleration response in the region of 25 ms\(^{-2}\) in the WTT nacelle. The
second case in 5.6.1.2 simulates a wind loading of mean speed 18m/s, which is a typical value for the operational wind speed of a WTT.

The Kaimal spectrum (Section 4.3) is used as the input spectrum. In this case, the wind force is applied nodally as outlined in Section 5.4. The JONSWAP spectrum (Section 4.4) is used as the input wave spectrum. Two wind speed cases are examined in Section 5.6.1, that of a ‘high’ wind speed of 30 m/s and that of an operational wind speed of 18 m/s. The estimated area of the wind turbine that the wind force interacts with is slightly reduced from that used in Section 4.7.

5.6.1.1 Wind and Wave Excitation with mean wind speed = 30 m/s

In Section 5.6.1.1, the mean wind velocity, which represents a correlation between the wind and wave excitations, at the top of the tower is taken as 30ms⁻¹. Simulated drag force time histories for the wind excitations applied to the respective nodes for a wind speed of 30 m/s are given from Figure 5.2 to Figure 5.4. The density of air, the coefficient of drag and the roughness length for the water surface were taken as 1.225 kgm⁻¹, 1.2 and 0.002, respectively. The wind spectrum cycled through a range of 0-600rad/sec. Time histories of length 50 sec are presented.

The JONSWAP spectrum was used to simulate the sea surface elevation spectrum. The following parameter values were used in the wave spectra simulation: \( k_0 = 1 \), \( l_x = 1200\text{m} \), \( C_d = 1.2 \), \( \rho_0 = 1.2 \text{ kg/m}^3 \), \( z = 21\text{m} \), \( \alpha_p = 0.23 \), \( U_{10} = 18 \text{ (m/s)} \), \( g = 9.81 \text{ m/s/s} \), \( F = 10000\text{m} \) and \( \gamma = 3.3 \). The resulting time history of the surface elevation of the sea for the ‘moderate’ wave loading is presented in Figure 5.5. The nodal drag force time histories for the wave excitation, deriving from a mean wind speed of 30m/s, at nodes 2 and 3 are presented in Figure 5.6 and Figure 5.7, respectively.
Figure 5.2 Time series for wind excitation at Node 7

Figure 5.3 Time series for wind excitation at Node 8
Figure 5.4 Time series for wind excitation at Node 9

Figure 5.5 Simulated time history of sea surface elevation
Figure 5.6 Wave force at node 2 for 'strong' wave excitation

Figure 5.7 Wave force at node 3 for 'strong' wave excitation
5.6.1.2 Wind and Wave Excitation with mean wind speed = 18 m/s

In Section 5.6.1.2, the mean wind velocity, which represents a correlation between the wind and wave excitations, at the top of the tower is taken as 18\(\text{ms}^{-1}\). Simulated drag force time histories for the wind excitations applied to the respective nodes for a wind speed of 18 m/s are given from Figure 5.8 to Figure 5.10.

As in Section 5.6.1.1, the following parameter values were used in the JONSWAP wave spectra simulation: \(k_0 = 1, l_x = 1200\text{m}, C_a = 1.2, \rho_o = 1.2 \text{ kg/m}^3, z = 21\text{m}, \alpha_p = 0.23, U_{l0} = 18 \text{ (m/s)}, g = 9.81 \text{ m/s/s}, F = 10000\text{m} \text{ and } \gamma = 3.3\). The simulated time history of the surface elevation of the sea deriving from a mean wind speed of 18 m/s is presented in Figure 5.11. The nodal drag force time histories for the wave excitation, deriving from a mean wind speed of 30 m/s, at nodes 2 and 3 are presented in Figure 5.12 and Figure 5.13, respectively.

![Figure 5.8 Time series for wind excitation at Node 7](image-url)
Figure 5.9 Time series for wind excitation at Node 8

Figure 5.10 Time series for wind excitation at Node 9
Figure 5.11 Simulated time history of sea surface elevation

Figure 5.12 Wave force at node 2 for 'operational' wave excitation
5.6.2 WTT response with BI system, TMD and TLCD

In Section 5.6.2, the WTT with BI system, TMD and TLCD is subjected to a combined wind and wave loading. The wind loadings of mean speed 30m/s and 18m/s from Section 5.6.1 are applied to the WTT system. The performance of the WTT with BI system, TMD and TLCD is compared with the WTT system without any damper (undamped case). Time histories responses for the WTT with the BI system having a mass ratio ($\mu_{BI}$) of 15% and 5% in relation to the SS2 are presented for each wind excitation case. Comparisons of displacement and acceleration response between the two cases are presented. The response of the TMD and the liquid in the TLCD are also presented.

The structure adopted in the numerical study is an offshore wind turbine tower (WTT) of nacelle height 130m above the sea floor. The WTT rises 100m above the platform level where the BI-TMD is located. For the purposes of investigation, three fundamental natural frequencies for the structure exist. The first is the overall natural

---

**Figure 5.13 Wave force at node 3 for 'operational' wave excitation**
frequency for the entire structure, $\omega_{n,SS}$, which is equal to 0.716 rad/sec. The second case is when the natural frequency of SS1, which is taken as a separate structure completely separated from SS2, is taken. In this case $\omega_{n,SS1}$ is equal to 6.414 rad/sec. The third case is when the natural frequency of SS2, which is taken as a separate structure completely separated from SS1, is taken. In this case $\omega_{n,SS2}$ is equal to 1.166 rad/sec. In each case, the respective the damper tuning ratio for the TLCD and TMD are taken from Gao et al. (1997) and Tsai (1995), respectively.

The mass ratio of the BI system ($\mu_{BI}$) is defined as the ratio of the mass of the BI system to that of the SS2. Time history responses for the WTT damper system with two different values of $\mu_{BI}$, 5% and 15%, are presented. As the BI system lies atop of the monopole foundation, a higher weight tolerance, as compared to that of the TLCD in the nacelle, is structurally viable. The main purpose of the BI system is to reduce the natural frequency of structural vibration to a level below the predominant energy-containing frequencies of the wave excitation. A secondary purpose of the BI system is to provide an additional means of energy dissipation, thereby reducing the transmitted acceleration into SS2. The natural frequency of the BI system is taken as a function of $\omega_{n,SS}$, unless otherwise stated. When the natural frequency of the BI system is taken as a function of $\omega_{n,SS}$, $\omega_{BI}$ is obtained from Equation (5.1) as 0.2159 rad/sec.

The mass ratio of the TMD ($\mu_{TMD}$) is defined as the ratio of the mass of the TMD to that of the SS2. The mass ratio of the TMD is taken as 1%, unless otherwise stated. The TMD is tuned to the natural frequency of the BI system using Equation (5.3). In the case of $\mu_{TMD} = 1\%$, $\chi_{TMD} = 98.75\%$.

The mass of the TLCD was 2% of the total mass of the SS2. Unless stated otherwise, the natural frequency of the TLCD was tuned to 99.2% of that of the fundamental frequency of the SS2 structure ($\omega_{n,SS2}$). The total length of the liquid column in the TLCD was 14.7m. The breath, $B$, which is the horizontal part of the TLCD, is taken as 2.5m, leaving vertical columns of 6.1m. In terms of accommodating the TLCD in
an offshore wind turbine, the vertical columns could be located in the tubular tower below the nacelle or within the tubular tower and through the nacelle. Another option would be to locate the TLCD either on top of or at the side of the nacelle. The values of $\xi$ and $\alpha$ were 5 and 0.17, respectively. The larger the value of $\alpha$, which is the ratio of the breath of the TLCD to that of the length of the liquid column, the more effective the TLCD is generally. However, in the case of a TLCD in a wind turbine, horizontal space is at a premium so one must use a lower value of $\alpha$ than is ideal in order to tune the TLCD to the natural frequency of the wind turbine.

The foundation is assumed to be fully fixed into the ground with no rotation allowed. The base of the tower, $d_0$, is 5m in diameter, tapering to 4m at the nacelle. The steel thickness at the bottom and the top of the structure is 20mm and 12mm, respectively. Steel with an elastic modulus equal to $210 \times 10^9$ N/m$^2$ was assumed to be used in the construction of the turbine. The structural damping ratio is assumed to be 0.01.

5.6.2.1 WTT with BI system, TMD and TLCD under 30m/s wind and wave loading

In this Section, the WTT (with and without the BI-TMD-TLCD damping system) with rotating blades is excited by wind and wave loadings deriving from a mean wind speed of 30 m/s (Section 5.6.1.1). Comparisons of displacement and acceleration response between the WTT-BI-TMD-TLCD system and the undamped WTT are presented in each response case. The response of the TMD and the liquid in the TLCD are also presented for each case. All the aforementioned response simulations are presented for $\mu_{BI} = 5\%$ and $\mu_{BI} = 15\%$. The blades rotated at a frequency equal to the fundamental natural frequency of the MDOF system. The blades are 60m in length and individually weighed 9.5 tonnes.

Figures 5.14 to 5.17 present response simulations for the case where $\mu_{BI} = 15\%$. Figure 5.14 presents the displacement response comparison for node 9 between the undamped WTT and the WTT with multiple damper system (BI system, TMD and TLCD). Figure 5.15 presents the acceleration response comparison for node 9 between the undamped WTT and the WTT with multiple damper system.
Values for the maximum displacement response, maximum acceleration response, root-mean-square (RMS) displacement response and root-mean-square acceleration response for various cases are tabulated and presented in Table 5.3 and Table 5.4.

The maximum response and the RMS of the response of node 9 for the undamped WTT are 0.1287m and 0.0414m, respectively. The nacelle displacement response (node 9) and the RMS of the response of node 9 for the WTT with multiple damper system are 0.0412m and 0.0180m, respectively. This represents a 68% reduction in displacement response and a 56% reduction in the RMS displacement response.

![Figure 5.14 Time-history response of tip response of the WTT under wind and wave excitation with and without multiple damper system](image)

**Figure 5.14 Time-history response of tip response of the WTT under wind and wave excitation with and without multiple damper system**
The maximum acceleration response and the RMS of the acceleration response of node 9 for the undamped WTT are 22.425 ms$^{-2}$ and 7.1580 ms$^{-2}$, respectively. The nacelle acceleration response and the RMS of the acceleration response of node 9 for the WTT with multiple damper system are 7.3492 ms$^{-2}$ and 2.0701 ms$^{-2}$, respectively. This represents a 72% reduction in acceleration response and a 71% reduction in the RMS acceleration response. The reduction both the acceleration response and the RMS acceleration response suggest that the base isolation system is successful in isolating the structure from induced accelerations deriving from the wind and wave excitations.

Figure 5.16 presents the displacement response of the TMD and Figure 5.17 presents the displacement response of the liquid in the TLCD.
Figure 5.16 Time-history displacement response of the TMD

Figure 5.17 Time-history displacement response of the TLCD liquid
Figures 5.18 to 5.21 present response simulations for the WTT with multiple damper system where $\mu_{\text{bl}} = 5\%$. Figure 5.18 presents the displacement response comparison for node 9 between the undamped WTT system and the WTT with multiple damper system. The nacelle displacement response (node 9) and the RMS of the response of node 9 for the WTT with multiple damper system are 0.0634m and 0.0301m, respectively. This represents a 50.8% reduction in displacement response and a 28% reduction in the RMS displacement response in comparison to the undamped WTT.

The nacelle acceleration response and the RMS of the acceleration response of node 9 for the WTT with multiple damper system are 7.5292 ms$^{-2}$ and 2.0719 ms$^{-2}$, respectively. This represents a 71% reduction in acceleration response and a 71% reduction in the RMS acceleration response.

In comparison to the WTT with multiple damper system where $\mu_{\text{bl}} = 15\%$, the displacement response reductions are not quite as high, however the reduction in acceleration response is similar.

![Figure 5.18 Time-history response of tip response of the WTT under wind and wave excitation with and without multiple damper system](image)

Figure 5.18 Time-history response of tip response of the WTT under wind and wave excitation with and without multiple damper system
Figure 5.19 Time-history response of tip acceleration of the WTT under wind and wave excitation with and without multiple damper system

Figure 5.20 presents the displacement response of the TMD and Figure 5.21 presents the displacement response of the liquid in the TLCD. In comparison to the TMD response when $\mu_{\text{till}} = 15\%$, it can be seen that the displacement response of the TMD is substantially larger when $\mu_{\text{till}} = 5\%$. This would indicate that a larger value of $\mu_{\text{till}}$ negates the influence of the TMD as the BI is possesses a large enough stiffness to counter the applied excitations.

The response of the liquid in the TLCD is also increased, indicating that the large value of $\mu_{\text{till}}$ was inhibiting the full potential of the damping properties of the TLCD. Further simulations of various damper combinations could be performed in order to isolate which damper is contributing to the overall reduction in the system response.
Figure 5.20 Time-history displacement response of the TMD

Figure 5.21 Time-history displacement response of the TLCD liquid
5.6.2.2 WTT with BI system, TMD and TLCD under 18 m/s wind and wave loading

In this section, the MDOF (with and without TLCD) structure with rotating blades is the wind and wave excitation deriving from a mean wind speed of 18 m/s (Section 5.6.1.2). Comparisons of displacement and acceleration response between the WTT with BI system, TMD and TLCD case and the undamped WTT are presented in each response case. The response of the TMD and the liquid in the TLCD are also presented for each case. All the aforementioned response simulations are presented for $\mu_{bi} = 5\%$ and $\mu_{bi} = 15\%$. The blades rotated at a frequency equal to the fundamental natural frequency of the MDOF system. The blades are 60m in length and individually weighed 9.5 tonnes.

Figure 5.22 to Figure 5.25 present response simulations for the case where $\mu_{bi} = 15\%$. Figure 5.22 presents the displacement response comparison for node 9 between the undamped WTT and the WTT with multiple damper system (BI system, TMD and TLCD). Figure 5.23 presents the acceleration response comparison for node 9 between the undamped WTT and the WTT with multiple damper system.

Figure 5.22 Time-history response of tip response of the WTT under wind and wave excitation with and without multiple damper system
The maximum response and the RMS of the response of node 9 for the undamped WTT are 0.092m and 0.0287m, respectively. The nacelle displacement response (node 9) and the RMS of the response of node 9 for the WTT with multiple damper system are 0.0371m and 0.0141m, respectively. This represents a 60% reduction in displacement response and a 50.8% reduction in the RMS displacement response.

The maximum acceleration response (Steady State) and the RMS of the acceleration response of node 9 for the undamped WTT are 15.3068 ms$^{-2}$ and 5.738 ms$^{-2}$, respectively. The nacelle acceleration response and the RMS of the acceleration response of node 9 for the WTT with multiple damper system are 7.3913 ms$^{-2}$ and 2.0814 ms$^{-2}$, respectively. This represents a 51.7% reduction in acceleration response and a 64% reduction in RMS response.

Figure 5.24 presents the displacement response of the TMD and Figure 5.25 presents the displacement response of the liquid in the TLCD.
Figure 5.24 Time-history response of the TMD

Figure 5.25 Time-history response of the TLCD liquid
Figures 5.26 to 5.29 present response simulations for the WTT with multiple damper system subjected to excitations deriving from a mean wind speed of 18 m/s where $\mu_{\text{Bi}} = 5\%$. Figure 5.26 presents the displacement response comparison for node 9 between the undamped WTT system and the WTT with multiple damper system.

The nacelle displacement response (node 9) and the RMS of the response of node 9 for the WTT with multiple damper system are 0.0549 m and 2.0217 m, respectively. This represents a 40% reduction in displacement response and a 30% reduction in the RMS displacement response in comparison to the undamped WTT. The nacelle acceleration response and the RMS of the acceleration response of node 9 for the WTT with multiple damper system are 7.5023 ms$^{-2}$ and 2.0827 ms$^{-2}$, respectively. This represents a 51% reduction in acceleration response and a 64% reduction in RMS acceleration response. In comparison to the WTT with multiple damper system where $\mu_{\text{Bi}} = 15\%$, the displacement response reductions are not quite as high, however the reduction in acceleration response is similar.

Figure 5.26 Time-history response of tip response of MDOF under wind and wave excitation with and without TLCD, TMD and BI system (5%)
Figure 5.27 Time-history response of tip acceleration of MDOF, with rotating blades, under wind and wave excitation with and without TLCD, TMD and BI system (5\%)

Figure 5.28 presents the displacement response of the TMD and Figure 5.29 presents the displacement response of the liquid in the TLCD.

In comparison to the TMD response when $\mu_{bl} = 15\%$, it can be seen that the displacement response of the TMD is larger when $\mu_{bl} = 5\%$. This would reaffirm the indication that a larger value of $\mu_{bl}$ negates the influence of the TMD.

The response of the liquid in the TLCD is also increased, reaffirming the indication that the large value of $\mu_{bl}$ was inhibiting the full potential of the damping properties of the TLCD.
Figure 5.28 Time-history displacement response of the TMD

Figure 5.29 Time-history displacement response of the TLCD liquid
5.6.3 Parametric Investigation

In this section, a parametric investigation on parameters relating to the performance of the WTT-TMD-BI-TLCD system is undertaken. As numerous forces are interacting in the WTT-BI-TMD-TLCD system, it is uncertain to what extent various damper variables can influence the whole system. The influence of the tuning of the BI system is investigated. As the primary aim of the TMD is to add stiffness to the BI system, the TMDs tuning is kept close to that of the BI system. The influence of the mass ratio of the TMD ($\mu_{\text{tmd}}$) to the SS2 on the performance of the overall damping system is investigated. The influence of the mass ratio of the BI system ($\mu_{\text{BI}}$) on the displacement and acceleration responses of the WTT system is presented.

Table 5.1 presents the response values when the tuning of the BI system and TMD system are broadly varied. As the TMDs main purpose is to increase the lateral stiffness of the BI system, the TMD is always tuned to the natural frequency of the BI system. The BI system is designed with respect to the natural frequency of the structure through equation (5.1), thus to aide comprehension, the variables presented on the left hand side of Table 5.2 are functions of different frequencies. These frequencies are the natural frequency of the overall structure ($\omega_{n,SS}$), the natural frequency of SS1 ($\omega_{n,SS1}$) and the natural frequency of SS2 ($\omega_{n,SS2}$). For all cases, every other variable was kept at the constant value used in the time history simulations.

It is seen from Table 5.1 that there is no great difference on the performance of the BI-TMD-TLCD damping system in regards to the variation of BI frequency between the values presented. When the BI system is tuned to SS2, there is a maximum reduction in displacement response in terms of the three cases considered. When the BI system is tuned to the SS, there is a maximum reduction in acceleration response. It is clear that the BI-TMD system is generally successful in preventing the transmission of predominant-energy containing frequencies into the overall super structure.
Table 5.1 Variation of tuning of TMD and BI

Table 5.2 presents the influence of the mass ratio of the TMD ($\mu_{md}$) on the overall response of the SS. Again, one can see that there is no great effect on the tip response of the SS by varying the mass of the TMD. The lack of influence of the TMD may be attributed to the greater mass ratio afforded the BI system and the damping primarily of structural modes other than the fundamental mode.
Table 5.2 Variation of mass ratio, $\mu_{TMD}$, of the TMD

Table 5.3 presents the influence of varying the tuning mass ratio of the BI system on the overall response of the SS for 30 m/s wind cases. The variation of $\mu_{BI}$ primarily affects the displacement response of the SS. In all cases, large reductions in RMS acceleration response and maximum acceleration response were found, with little variation. In terms of displacement response, as the mass of the BI system is related to its stiffness, this has a direct impact on the displacement response at the top of the WTT. Thus, at 15%, $\mu_{BI}$ provides the largest displacement response. As the $\mu_{BI}$ is decreased, it eventually reaches a point (somewhere between 3.5 and 2%), where the displacement response becomes larger than the undamped system. This could possibly indicate lateral failure of the BI system and would obviously have to be avoided at all costs.

Thus, with regards to acceleration reduction, displacement reduction and overall mass, the optimum level of $\mu_{BI}$, also taking into account construction costs, would appear to be around 10%. Increasing $\mu_{BI}$ over 10% gives one a disproportional reduction in displacement response.
<table>
<thead>
<tr>
<th>Parameters</th>
<th>Maximum Response (m)</th>
<th>Maximum Acceleration Response (m/s/s)</th>
<th>RMS of Response (m)</th>
<th>RMS of Acceleration Response (m/s/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped WTT</td>
<td>0.1287</td>
<td>26.4857</td>
<td>0.0414</td>
<td>7.1177</td>
</tr>
<tr>
<td>$\mu_{Bi} = 2%$</td>
<td>0.2060</td>
<td>7.2958</td>
<td>0.1107</td>
<td>2.0833</td>
</tr>
<tr>
<td>$\mu_{Bi} = 3.5%$</td>
<td>0.0981</td>
<td>7.5644</td>
<td>0.0473</td>
<td>2.0760</td>
</tr>
<tr>
<td>$\mu_{Bi} = 5%$</td>
<td>0.0634</td>
<td>7.5292</td>
<td>0.0301</td>
<td>2.0719</td>
</tr>
<tr>
<td>$\mu_{Bi} = 10%$</td>
<td>0.0448</td>
<td>7.3509</td>
<td>0.0197</td>
<td>2.0701</td>
</tr>
<tr>
<td>$\mu_{Bi} = 15%$</td>
<td>0.0412</td>
<td>7.3492</td>
<td>0.0180</td>
<td>2.0701</td>
</tr>
</tbody>
</table>

Table 5.3 Variation of tuning of mass ratio, $\mu_{Bi}$, of BI system (mean wind speed = 30m/s)

Table 5.4 presents the influence of varying the tuning mass ratio of the BI system on the overall response of the SS for 30m/s wind cases. Table 5.4 exhibits close similarities with Table 5.3. Again, $\mu_{Bi} = 15\%$, provides the largest displacement response. As $\mu_{Bi}$ is decreased, it eventually reaches a point (somewhere between 3.5 and 2%), where the displacement response becomes larger than that of the undamped system. This would suggest that at a certain value of $\mu_{Bi}$, the lateral stiffness provided by the BI system with TMD proves insufficiently stiff against the wind and wave excitation.
<table>
<thead>
<tr>
<th>Parameters</th>
<th>Maximum Response (m)</th>
<th>Maximum Acceleration Response (m/s/s)</th>
<th>RMS of Response (m)</th>
<th>RMS of Acceleration Response (m/s/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped WTT</td>
<td>0.0920</td>
<td>15.3068</td>
<td>0.0287</td>
<td>5.7379</td>
</tr>
<tr>
<td>$\mu_{BI} = 2%$</td>
<td>0.1375</td>
<td>7.3107</td>
<td>0.0728</td>
<td>2.0846</td>
</tr>
<tr>
<td>$\mu_{BI} = 3.5%$</td>
<td>0.0724</td>
<td>7.5100</td>
<td>0.0333</td>
<td>2.0838</td>
</tr>
<tr>
<td>$\mu_{BI} = 5%$</td>
<td>0.0549</td>
<td>7.5023</td>
<td>2.0817</td>
<td>2.0827</td>
</tr>
<tr>
<td>$\mu_{BI} = 10%$</td>
<td>0.0423</td>
<td>7.3850</td>
<td>0.0156</td>
<td>2.0826</td>
</tr>
<tr>
<td>$\mu_{BI} = 15%$</td>
<td>0.0371</td>
<td>7.3913</td>
<td>0.0141</td>
<td>2.0814</td>
</tr>
</tbody>
</table>

Table 5.4 Variation of tuning of mass ratio of BI system (mean wind speed = 18m/s)

5.6.4 Harnessing of the TMDs Energy

In Section 5.6.4, the mechanical energy generated from the movement of the TMD is investigated. The energy, in kWhs, produced from the TMD when the mass ratio of the BI system is 10% is investigated in Section 5.6.4.1. The vibrational energy produced from the TMD when the mass ratio of the BI system is 5% is investigated in Section 5.6.4.2.

Equation (5.35) provides a means of quantifying the instantaneous energy of the moving TMD. When the cumulative energy deriving form the TMD in motion is added over time, it becomes axiomatic that a real, additional method of energy production deriving from the wind and wave excitation exists. Section 5.6.4 simulates the motion of the TMD for a period of time, and calculates the cumulative energy deriving from it. In terms of extracting energy from the TMD, as energy is being taken from the TMD, the dynamic response of the TMD and hence the SS itself will be somewhat altered, but this represents a small variation in the overall simulation of the system. As power lines are already in place from an offshore wind turbine, it is a
trivial matter to transmit the energy to an appropriate source. The electrical energy derived from the TMD in times of excitation could also be used to power the control systems etc., in the WTT itself.

5.6.4.1 Mass ratio of BI system = 10%

In Section 5.6.4.1, the case of an operational wind loading of mean velocity 18m/s and a BI mass ratio, $\mu_{\text{BI}}$, of 10% are considered. The TMD is tuned to the BI system, which is tuned to the overall natural frequency of the SS ($\omega_{n,SS}$). Figure 5.30 presents the displacement response of the TMD during a 50 second period of wind and wave excitation. A mass ratio ($\mu_{\text{tmd}}$) of 1% is assumed for the TMD.

![Response of TMD](image)

Figure 5.30 Displacement response of the TMD with $\mu_{\text{BI}} = 10\%$
Figure 5.31 presents the velocity response of the TMD subjected to wind and wave excitation. Figure 5.32 presents the instantaneous energy (Joules) produced in the TMD (Equation (5.35)) resulting from the movement of the TMD. Figure 5.33 presents the cumulative energy from the TMD over a 20 second period when the WTT is subjected to an operational mean wind speed of 18 m/s.

**Figure 5.31 Velocity response of TMD with $\mu_{bl} = 10\%$**
Figure 5.32 Instantaneous Energy produced in TMD with $\mu_{Bl} = 10\%$

Figure 5.33 Cumulative Energy produced in TMD with $\mu_{Bl} = 10\%$
If one assumes the WTT is subjected to a consistent wind and wave excitation deriving from a 18 m/s mean wind speed for a 24hr period, extrapolation of the cumulative energy produced in 20 seconds yields 32400 Js produced in a day. If the TMD was constantly moving under the same forces for a 1 year period, 11,826 KJ of energy would be produced. As 1kWh equals $3.6 \times 10^6$ Joules, this figure converts to 3.3 kWh.

5.6.4.2 Mass ratio of BI system = 5%

In Section 5.6.4.2, the case of an operational wind loading of mean velocity 18m/s with a BI mass ratio, $\mu_{bl}$, of 5% is investigated. A mass ratio ($\mu_{trm}$) of 1% is assumed for the TMD. Figure 5.36 presents the time history displacement response of the TMD during a 50 second period of wind and wave excitation. Figure 5.35 presents a simulation of the velocity response of the TMD.

![Response of TMD](image)

**Figure 5.34 Displacement response of the TMD with $\mu_{bl} = 5\%$**
Figure 5.35 Velocity response of the TMD with $\mu_{bl} = 5\%$

Figure 5.36 Cumulative Energy produced in TMD with $\mu_{bl} = 5\%$
Figure 5.36 presents the cumulative energy produced from the TMD for 20 seconds under an operational wind loading of mean speed 18 m/s. When one extrapolates the energy produced for the present case over the duration of a day, 129,600 Js of energy are produced. Converting this figure into a yearly kWh value yields 13.1 kWh.

5.7 CONCLUSIONS

Chapter 5 examined a novel design for an offshore wind turbine that incorporates a BI system, a TMD and a TLCD. Wind loadings were generated using the JONSWAP spectrum and wave excitations were generated using the Kaimal spectrum. The WTT assembly was discretized into ultimately an eighteen DOF system, the tower with dampers incorporating nine DOFs and the three blades each represented by three DOFs.

When the WTT assembly is fitted with the BI-TMD-TLCD damping system, substantial reductions in displacement and acceleration response are attained. In one case, when the WTT is subjected to fluctuating wind and wave loadings deriving from a mean wind speed of 30 m/s, the tip acceleration and displacement response are reduced from the undamped WTT by 72% and 68%, respectively. Thus, the BI system is successful in preventing the predominant energy-containing frequencies of the excitation from entering the SS2 and hence drastically reduces in the acceleration and displacement response of the WTT nacelle.

It was found in Section 5.6.3 that the mass ratio of the BI system has a large impact on the performance of the damping system. The mass, and hence stiffness, must be such as to prevent substantial lateral displacement responses at the platform level. It was found that the mass ratio of the BI system to the SS2 should be greater than 3.5%, with 10% fulfilling the operational performance of the BI system.

In Section 5.6.4, an additional method of energy generation deriving from the harnessing of nature's forces was proposed and investigated. It was found that under operational conditions, an additional 13.1 kWhs of energy are produced from the WTT.
Chapter 6

INCREASED AVAILABILITY OF OFFSHORE WIND TURBINES WITH TLCD SUBJECTED TO WIND AND WAVE INDUCED ACCELERATIONS

6.1 INTRODUCTION

In previous chapters, the structural response of offshore wind turbines was simulated with attached TLCD and other dampers for controlling the vibrations induced within the structure. This chapter presents a long term probabilistic model of the wind turbine response and investigates the risk of peak accelerations in offshore wind turbines exceeding predefined cut-off limits. The chapter quantifies the increased reliability of operation of offshore wind turbines with installed TLCDs over offshore wind turbines without installed TLCDs.

6.2 SHUT DOWN CRITERIA

In current wind turbines, when mean wind speed reaches a certain level, established through anemometers attached to the nacelle, a protective measure is triggered which shuts down the operation of the wind turbine. The protective measure assumes a direct relationship between mean wind speed and structural acceleration response. This protective approach ignores the structural response of the system, and unacceptable levels of acceleration or displacement can be attained by the towers even at wind speeds below the threshold for shutdown. Several wind turbine subcomponents are sensitive to these levels of accelerations. Some examples, in decreasing order of acceleration susceptibility, include generators, brakes, yaw systems, inverters, mechanical controls, electrical controls, hydraulic systems, gearboxes, shafts, and physical/electrical connections to the power grid. Previous studies have concluded that in wind farms under consideration, over half of turbine units are likely to experience abnormal operation due to demands on acceleration sensitive equipment.
The wind turbine, with rotating blades, is modelled as a Multi-Degree-of-Freedom (MDOF) structure and is dynamically excited by a time-varying wind excitation. The wind turbine system is coupled with a TLCD. The acceleration response of the turbine system with and without TLCD is coupled with a probabilistic description of the dynamic properties of the wind turbine to produce an annual distribution of the wind hazard (Colwell et al., 2007).

The wind turbine acceleration response distribution obtained from the simulation routine is used to construct fragility curves that relate the probability of exceeding predefined acceleration levels with a given set of annual wind speeds. These fragility curves provide the likelihood of failure of acceleration-sensitive equipment, and can approximate the fraction of wind turbines in a wind farm that undergo failure when subjected to a particular wind speed. The overall effect of reducing unavailability in wind turbines by implementing TLCDs is obtained through use of the fragility curves.

In addition, reductions in the tip acceleration response, and hence bending moments, of a wind turbine system with tuned liquid column damper (TLCD) employed over that of the system without TLCD, allow a more economic design of the structural tower and foundations of the system. With reductions in Initial Capital Cost (ICC) and increases in Annual Energy Production (AEP) through instalment of a TLCD, the Cost of Energy (COE) produced from wind turbines is reduced. Chapter 6 also quantifies the monetary effect of implementing dampers through the COE metric.

6.3 WIND TURBINE FRAGILITY ANALYSIS

The acceleration response at the wind turbine nacelle is sensitive not only to variations in wind speed and wind-induced turbulence but also to variability in the parameters used for dynamic characterization of wind turbines. Uncertainty in damping ratios, structural stiffness, and mass of wind turbine systems—tower and rotor blades—contribute significantly to the variation in their acceleration response.

A uniform random variable is used to model the damping ratios of wind turbine towers and rotor blades. These damping ratios are assigned to the fundamental mode
of the wind turbine tower, and the first three modes of the rotor blades. For the same modes considered in the assignment of damping ratios, additional uniform random variables are used to modify the calculated undamped natural frequencies of the tower and the rotor blade subsystems. Without loss of generality, these uniform random variables provide the means to capture inherent variability of the material, aging effects, vibration levels, and operation and maintenance conditions.

Table 6.1 summarizes the properties of the random variables used to simulate the acceleration-response of wind turbines. The proposed range for the damping ratio reflects the stress conditions of the steel tower, which under normal operation regimes are less than $\frac{1}{2} \times f_y$ (Chopra, 1995), where $f_y$ represents the yield stress of the steel. The variability in the natural frequencies corresponds to an arbitrary selection of $\pm 10\%$ their actual value. As the Weibull distribution is often used in the field of life data analysis due to its flexibility (it can take on the characteristics of other types of distributions), it is used herein to describe the annual distributions of wind speeds for different geographical locations. Table 6.1 includes the parameters of Weibull distributions that describe the annual variability of wind speeds at the two geographical locations chosen for the present study, Texas and Ireland. Figure 6.1 presents a map of wind speed at a height of 80m averaged over a year for Europe (in 2000).

<table>
<thead>
<tr>
<th>Variable</th>
<th>Distribution</th>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damping Ratio, $\zeta_{SS}$</td>
<td>Uniform</td>
<td>Lower and upper bounds: $a_{\zeta_{SS}}, b_{\zeta_{SS}}$</td>
<td>0.01, 0.03</td>
</tr>
<tr>
<td>Multiplier of undamped natural frequency, $\alpha_{SS}$</td>
<td>Uniform</td>
<td>Lower and upper bounds: $a_{\alpha_{SS}}, b_{\alpha_{SS}}$</td>
<td>0.90, 1.10</td>
</tr>
<tr>
<td>Annual wind speed, $v_w$ (m/s)</td>
<td>Weibull</td>
<td>Scale factor and mean wind speed: $\lambda_{\text{Texas}} (\lambda_{\text{Ireland}}), \bar{v}<em>{w, \text{Texas}} (\bar{v}</em>{w, \text{Ireland}})$</td>
<td>2.0 (2.0), 7.0 (9.0)</td>
</tr>
</tbody>
</table>

Table 6.1 Distributions and parameters for modelling wind turbine response variability
Figure 6.1 Map of wind speed extrapolated to 80 m and averaged over all days of the year for Europe (Archer and Jacobson, 2005)
The effects of the input random variables on the accelerations at the nacelle are explored with a Latin-Hypercube sampling technique. In the context of statistical sampling, a square grid containing sample positions is a Latin Square if (and only if) there is only one sample in each row and each column. A Latin-Hypercube is the generalisation of this concept to an arbitrary number of dimensions, whereby each sample is the only one in each axis-aligned hyperplane containing it. The Latin-Hypercube sampling scheme is chosen due to the independence of the variability and the fact that random samples can be accurately logged as they can be taken one at a time.

The distribution of each random variable is partitioned into 200 areas of equal probability. An equal number of wind-tower/rotor-blade systems are generated by assigning, without replacement, realizations of the input random variables to the dynamic model (from Section 4.2) that calculates acceleration response at the nacelle.
The response is monitored for mean wind speeds between a range from 5 m/s to 30 m/s. The typical cut-out mean wind speed in the wind energy industry is 25 m/s. A power model is also fit to the simulation data to provide an estimate of the median wind-induced acceleration. The power model has the following functional form

\[ \hat{a}_w = b \times (v_w)^c \]  

(6.1)

where \( \hat{a}_w \) is the median wind-induced acceleration at the top of the wind turbine in \( \text{m/s}^2 \), \( v_w \) is the wind speed in \( \text{m/s} \), and \( b \) and \( c \) are constants that can be obtained from linear regression in a logarithmic space.

The wind-induced acceleration relationship of equation (6.1) can be used within a probabilistic wind demand model to estimate the likelihood of exceeding acceleration thresholds as a function of increasing wind speed. In general, probabilistic demand models have a lognormal distribution to represent conditional probabilities of failure. In the present investigation the conditional probability of failure, or probability of exceeding a predefined acceleration limit, is denoted by

\[ p[a_w > a_{LS} | v_w] = 1 - \Phi \left( \frac{\ln(a_{LS}) - \ln(b \times (v_w)^c)}{\beta_{aw}} \right) \]  

(6.2)

where \( a_w \) is the wind-induced acceleration at the turbine top, \( a_{LS} \) represents acceleration thresholds that induce changes in performance state (e.g., safe or failed), and \( \beta_{aw|v_w} \) is the dispersion of the acceleration response as a function of wind speed. In this study, the values of \( \beta_{aw|v_w} \) are assumed to be independent of \( v_w \), and \( \beta_{aw|v_w} = \beta_{aw} \). The magnitude of \( \beta_{aw} \) represents the standard deviation of the sample acceleration response.

The values of \( a_{LS} \) correspond to acceleration levels at which acceleration-sensitive equipment can experience malfunctioning. Four acceleration thresholds are explored to cover a wide range of operating conditions: \( a_{LS} = \{5, 10, 15, 20\} \) in \( \text{m/s}^2 \). Research from the field of earthquake engineering has concluded that the functionality of industrial equipment can be impaired by moderate to high accelerations (Porter et al., 162).
As an example, the functionality of generators can be affected in the range of 5-10 m/s², inverters in the range of 10-15 m/s², and electrical controls in the range of 15-20 m/s². Modern wind turbines house all of the aforementioned equipment. Thus, if the aforementioned wind-induced acceleration levels are developed at the nacelle, then acceleration-sensitive equipment in the WTT will be susceptible to acceleration-induced failure. Additional wind turbine equipment components which are sensitive to acceleration include: yaw systems, mechanical control and brakes, hydraulic systems, gearboxes, and connections to the power grid.

The wind demand model of equation (6.2) can then be used to evaluate the probabilities of exceeding each of the $a_{LS}$ thresholds within a predefined range of wind speeds. The graphical representation of these relationships constitutes a set of fragility curves. Fragility curves, also known as damage functions (Reinhorn et al., 2001) are used to approximate damage from natural hazards (particularly earthquake excitations) and are an increasingly popular way of characterizing the probabilistic nature of the phenomena concerned. The term fragility means the probability of attaining a limit state, conditioned on a particular value of random demand (Ellingwood, 2001). In essence, fragility is a measure of vulnerability or an estimate of the overall risk a particular system faces over time. Figure 6.3 (adapted from Mander 1999) presents a typical set of fragility curves for a structure under earthquake excitation. The damage states include minor, moderate, extensive, and complete damage. It can be seen that with smaller levels of earthquake excitations, the probability of suffering minor damage is far greater than that of structural collapse/failure.
Figure 6.3 Typical set of fragility curves of building structures (Leon and Atanasiu, 2007)

Fragility curves have only recently been used in the wind turbine industry (Dueñas-Osorio and Basu, 2007), and may become useful tools to estimate the unavailability of typical wind turbines for a given wind speed at any geographical location. As such, they provide an ideal means to investigate and present tangible long term effects of TLCDs in offshore wind turbines.

The information contained in the fragility curves in the present study is conditioned by specific levels of mean wind velocity. However, these curves do not account for the likelihood of observing different levels of wind speed in a given year. Without weighting the probability of exceeding an acceleration threshold by the likelihood of wind speed realizations, it is impossible to estimate unconditional annual wind turbine unavailability due to acceleration-sensitive equipment failure.
6.4 WIND TURBINE ECONOMICS

In quantifying the overall cost of producing energy from a WTT, various factors are taken into account. The accepted Cost of Energy calculation for a wind turbine system is as follows (Cohen et al., 1989)

\[
\text{COE} = \frac{\text{ICC} \times \text{FCR} + \text{LRC}}{\text{AEP}_{\text{NET}}} + \text{O} \& \text{M} \quad (6.3)
\]

\[
\text{AEP}_{\text{NET}} = \text{AEP}_{\text{GROSS}} \times \text{Availability} \times (1 - \text{Loss}) \quad (6.4)
\]

where,

- COE = Cost of Energy ($/kWh),
- ICC = Initial Capital Cost ($),
- FCR = Fixed Charge Rate (%/year),
- LRC = Levelized Replacement Cost ($/year),
- O&M = Operations and Maintenance Costs ($/kWh)
- AEP = Annual Energy Production (kWh/year).

Equation (6.3) a reasonable approximation of the COE that would be estimated by a potential investor and takes equipment reliability into account when determining the AEP, O&M, and LRC terms. AEP is affected by equipment reliability through turbine downtime associated with both scheduled and unscheduled maintenance. O&M consists of both scheduled (preventive) and unscheduled (repair) maintenance costs, including expenditures for replacement parts, consumables, manpower and equipment. LRC costs are associated with major overhauls and component replacements over the life of a wind turbine. Usually this category includes only major components and is based on components whose expected life is less than the wind turbine's design life.
The total assumed cost is of replacement parts, deriving from wear and tear or acceleration induced damage, is spread over the machine lifetime. Equipment reliability directly affects the LRC in that the LRC figure is only as accurate as the component life estimates. Wind turbines are commonly designed so that the major component design lives are equal to the turbine’s design life. It is clear that increases in availability decreases the COE of a wind turbine.

6.5 ANALYSIS OF RESULTS

The WTT model used in the analysis of wind turbine fragility is the same as that used in Section 5.4, with the difference being that there is no BI system and TMD. Thus, the structure adopted in the numerical study is an offshore wind turbine tower (WTT) of nacelle height 130m above the sea floor. The WTT rises 100m above the platform level where the BI-TMD is located. The natural frequency for the entire structure, \( \omega_{n.55} \), which is 0.716 rad/sec. The WTT tapers from 4.3m at the base to 3.5m at the tip, with the respective steel thicknesses varying from 18mm to 10mm. The mass of each blade is taken as 5 tonnes. The values of mass ratio \( (\mu) \) and coefficient of head loss \( (\xi) \) for the TLCD are taken as 2.5% and 1%, respectively. The TLCD was tuned to 99.2% of the fundamental natural frequency of the WTT. The distributions and parameters for modelling the wind turbine response variability are the same as those presented in Table 6.1. In the construction of the fragility curves, each individual mean wind speed variable is inputted into the simulation of wave excitation and wind excitation outlined in Sections 4.4 and 4.3, respectively. The wave excitation is related to the mean wind speed through equations (4.33), (4.34) and (4.36).

The acceleration demand at the nacelle of the undamped wind turbine is presented in Figure 6.4. The acceleration demand at the nacelle of the wind turbine with TLCD is presented in Figure 6.5. The acceleration response is generally seen to be higher for the undamped WTT than the damped WTT. In terms of the power model, at a mean wind speed of 15m/s, the response of the undamped WTT is 6.5ms\(^{-2}\) whereas the acceleration response for the damped WTT is 2.5ms\(^{-2}\). At a mean wind speed of 35m/s, the response of the undamped WTT is 34ms\(^{-2}\) whereas the acceleration
response for the damped WTT system is $18.5 \text{ms}^{-2}$. The simulation results deviate from the power model plot more for the damped structure than the undamped structure as the TLCD is tuned to the original natural frequency, which imparts a greater variance in the simulation results as compared to the power model. It is observed that despite varying structural parameters over time, the TLCD at its original tuning setting continues to damp the structure to a satisfactory degree. Table 6.2 summarizes the parameters that characterize the set of typical wind turbine fragility curves.

Figure 6.4 Acceleration demand at the nacelle of undamped wind turbine
Figure 6.5 Acceleration demand at the nacelle of wind turbine with TLCD

<table>
<thead>
<tr>
<th>Acceleration Threshold, $a_{LS}$ (m/s²)</th>
<th>Undamped WTT Median Wind Speed, $v_{aLS}$ (m/s)</th>
<th>WTT with TLCD Median Wind Speed, $v_{aLS}$ (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>13</td>
<td>20</td>
</tr>
<tr>
<td>10</td>
<td>19</td>
<td>26</td>
</tr>
<tr>
<td>15</td>
<td>23.5</td>
<td>32</td>
</tr>
<tr>
<td>20</td>
<td>27</td>
<td>36</td>
</tr>
</tbody>
</table>

Table 6.2 Mean wind speed acceleration thresholds

The displacement demand at the nacelle of the undamped wind turbine is presented in Figure 6.6. The displacement demand at the nacelle of the wind turbine with TLCD is presented in Figure 6.7. As in the case of the acceleration demand curves there is a significant reduction in response across all mean wind speeds when the WTT is equipped with a TLCD. In terms of the power model, at a mean wind speed of 15m/s, the response of the undamped WTT is 0.017m whereas the acceleration response for the damped WTT is 0.0086m. At a mean wind speed of 35m/s, the response of the
undamped WTT is $34 \text{ms}^{-2}$ whereas the acceleration response for the damped WTT is $18.5 \text{ms}^{-2}$.

![Figure 6.6 Displacement demand at the nacelle of undamped wind turbine](image)

![Figure 6.7 Displacement demand at the nacelle of wind turbine with TLCD](image)
The complete set of fragility curves for wind turbine acceleration given mean wind speed for the WTT with and without the TLCD are presented in Figure 6.8 and Figure 6.9, respectively. Table 6.3 summarizes the parameters used in calculating the annual probability of exceeding four acceleration thresholds $a_{LS}$ at two distinct geographical locations Texas and Ireland.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Western Texas</th>
<th>Northern Ireland</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>5</td>
<td>10</td>
</tr>
<tr>
<td>$v_{aLS}$</td>
<td>15</td>
<td>21</td>
</tr>
<tr>
<td>$b_{sw}$</td>
<td>1.02</td>
<td>1.02</td>
</tr>
<tr>
<td>$v_{w^*}$</td>
<td>5.4</td>
<td>7.6</td>
</tr>
<tr>
<td>$g(v_{w^*})$</td>
<td>0.401</td>
<td>0.280</td>
</tr>
<tr>
<td>$g(v_{aLS})$</td>
<td>0.078</td>
<td>0.042</td>
</tr>
</tbody>
</table>

Table 6.3 Parameters in calculation of fragility curves

![Figure 6.8: Wind-induced acceleration fragility curves for the WTT without TLCD](image)
It is clear that high levels of acceleration at the wind turbine nacelle have significant probabilities of being exceeded during normal wind speed ranges. At the onset of protective shutdown wind speeds, $v_w \sim 25 \text{ m/s}$, the probabilities of exceeding each of the $a_{LS}$ thresholds for the WTT without TLCD are 86%, 66%, 51%, and 40%. These values highlight the considerable likelihood of observing malfunctioning of wind turbine subcomponents, and the potential to contribute significantly to the overall annual failure rates of wind turbines. In contrast to the undamped WTT, the probabilities of exceeding each of the $a_{LS}$ thresholds for the WTT with TLCD are 66%, 44%, 30%, and 23%. Thus, TLCDs in WTTs significantly reduce the probability of wind turbine subcomponents malfunctioning.
6.6 ECONOMIC EVALUATION

It is seen that the probability of exceeding given acceleration thresholds for the WTT are significantly reduced. Thus the $AEP_{\text{NET}}$ metric is increased, which decreases the overall COE. Any percentage increase in $AEP_{\text{NET}}$ is met with a strongly correlated decrease in the COE of a WTT, assuming that O&M constitutes a small percentage of the overall cost of producing energy in an offshore wind turbine. Thus, if the availability is increased by 10% a year in an area of strong wind, such as Texas or Ireland, this corresponds to slightly less than a 10% saving in producing energy. By introducing TLCDs in WTTs, the ICC metric is also decreased through savings in steelwork and foundations. LRC and O&M costs are also reduced throughout the lifetime of the WTT as components are subjected to smaller bending moments and acceleration levels. Decreases in the LRC and O&M cost metrics further decrease the overall COE of producing energy from offshore wind turbines.

6.7 CONCLUSIONS

The wind-induced acceleration fragility curves introduced in this study indicate that wind turbine turbines are prone to failure from acceleration-sensitive components. High acceleration levels are predicted for the nacelle at the onset of current shutdown wind speed criteria ($v_w \sim 25$ m/s). For instance, for acceleration thresholds $a_{1,5}$ of 5, 10, 15, and 20 m/s$^2$, the probabilities of being exceeded for the undamped WTT are 86%, 66%, 51%, and 40%, respectively. These levels of acceleration can clearly induce malfunctioning in critical wind turbine components such as generators, inverters, mechanical controls, electrical controls, and any movable part. When a TLCD is implemented in the WTT, for acceleration thresholds $a_{1,5}$ of 5, 10, 15, and 20 m/s$^2$, the probabilities of being exceeded for the undamped WTT are 67%, 44%, 30%, and 23%, respectively.

Improving the reliability of wind turbines from acceleration-sensitive equipment will reduce the overall failure rates of turbines per year. These reductions could constitute incentives for policy strategies that encourage wind power generation and integration with the power grid.
By introducing TLCDs in offshore wind turbines, the availability of the WTT is increased, the initial capital costs are decreased and overall maintenance and replacement costs are decreased. Hence the cost of producing energy from offshore wind turbines is decreased and offshore wind turbines become more economically viable and competitive.
Chapter 7

EXECUTIVE SUMMARY AND CONCLUSIONS

7.1 EXECUTIVE SUMMARY

The aim of this thesis is to investigate the applicability of dampers in offshore wind turbines for the mitigation of vibrations caused by wind and wave excitations. As offshore wind turbines are becoming larger and being placed further out at sea, the excitations that these flexible structures are subjected to become ever greater. The result is an increase in construction costs, a reduction in the fatigue life of the structure, reduced availability of the power generating mechanism time reduce and an increase in the cost of energy (COE). The COE is the ultimate thrust behind the establishment of offshore wind turbines as all breakeven analysis for offshore wind turbine developments and competitive analysis against other forms of energy derive from it. To analysis an offshore wind turbine with TLCD, three broad factors had to be considered. First, the damper itself and its’ interaction with a structural system had to be investigated. Secondly, the forces that excite an offshore wind turbine had to be investigated and simulated. Next, a reduced order model of a coupled tower and blades structure had to be realized. All the preceding stages were then combined to investigate the effect of various passive dampers an offshore wind turbine in the time domain. Following this, benefits such as increased fatigue life and reduced construction costs due to the installation of a TLCD in an offshore wind turbine were investigated. Finally, the long term reliability effects of installing a TLCD in an offshore wind turbine using measured wind data were examined.

Tuned Liquid Column Dampers (TLCDs) were first investigated, with particular emphasis on the orifice damping, through theoretical and experimental studies. A closed form solution for the transfer function of the maximum displacement of the structure, modelled as a single-degree-of-freedom (SDOF) system, with applied harmonic base motion, has been presented. A standard equivalent linearization technique was employed to cater for the non-linear damping of the orifice in the TLCD. The performance of the orifice was evaluated through the experimental testing
of various base displacement amplitudes and frequencies applied to an experimental TLCD-structural system in the Civil Engineering Laboratory, Trinity College Dublin. The SDOF system was built on a platform capable of undergoing sinusoidal excitation to pre-defined input frequencies and displacements. A TLCD was also built with the ability to change its orifice size. Both theoretical and experimental results were compared through various plots to examine and compare the performance of TLCD.

Next, an experimental investigation into the passive damping properties of various fluids, including magnetorheological (MR) fluid, in the TLCD was undertaken. The coefficient of head loss for different fluids used in TLCDs to reduce structural responses in a wind turbine modelled as a SDOF system subjected to base excitation was investigated. By using different liquids in a TLCD other than water, additional performance improvements may be achieved in regards to the TLCD. As the effectiveness of the semi-active MR-TLCD relies upon an adequate movement of the MR fluid within the TLCD and a TLCD in an offshore context requires a freeze free liquid, MR fluids and glycol were investigated as liquids in the TLCD, respectively. The performances of water, glycol and an MR fluid are compared both theoretically and experimentally and the merits of each of the fluids in providing adequate passive damping to the turbine structure are discussed. Both a Kanai-Tajimi type base excitation and harmonic excitation were applied to the structure in the experimental tests. The equivalent viscous damping in the structure provided by the TLCDs using each of the fluids was obtained for both harmonic and broad banded excitations. Equations for calculating the coupled TLCD-structure response in the time domain were presented. Experimental results were used to calculate the non-linear coefficient of head loss based on a theoretical formulation. The numerical simulations of the responses of the structure-TLCD with various fluids used in TLCDs were validated with the experimental results.

With the properties of the TLCD investigated, the structural response of an offshore wind turbine was next simulated with attached TLCD for controlling the vibrations induced within the structure. The Kaimal spectrum, which takes into account eddy currents of varying size acting between the structural nodes, was simulated in the time domain. The Kaimal spectrum for wind loading was combined with a simulation of
JONSWAP wave spectrum in the time domain to formulate correlated wind and wave loadings on the structure. The offshore turbine tower was modelled as a MDOF structure. The blades were discretized into 3DOF systems and the interaction between the wind excited rotating blades and the vibrating tower were coupled in the time domain. Cases for flat sea conditions under ‘moderate’ and ‘strong’ wind loading exciting the lumped mass structure, with which parallels to onshore wind turbines may be drawn, were simulated. Next, the lumped mass structure was excited under ‘strong’ wind and wave loading. Finally, the offshore wind turbine with rotating blades was excited by ‘strong’ wind and wave excitation. The reduction in bending moments and structural displacement response with TLCDs for each case were examined. An economic analysis on the benefits of installing a TLCD in an offshore wind turbine, with regards to extended fatigue life and reduced bending moments at the base of the structure, was presented.

To reduce transmission of vibration from wave action, an investigation into a novel design for an offshore wind turbine that incorporates a number of passive dampers has been undertaken. Base isolation (BI) has traditionally been used in an aseismic design approach in which the building is protected from the hazards of earthquake forces by a mechanism which reduces the transmission of horizontal acceleration into the structure. The main purpose of the BI system is to reduce the natural frequency of structural vibration to a level below the predominant energy-containing frequencies of the excitation. A secondary purpose of the BI system is to provide an additional means of energy dissipation, thereby reducing the transmitted acceleration into the system. The use of the BI system in reducing vibrations in an offshore wind turbine is investigated.

Following on from investigation of an offshore wind turbine with attached TLCD, a hybrid damping system was proposed and investigated. The hybrid damping system incorporates a BI system at the platform, a TMD connected to the BI system for increased lateral stiffness and a TLCD at the nacelle of the offshore wind turbine. The offshore wind turbine tower with hybrid damping scheme was subjected to a combined wind, wave and rotating blade excitation. A parametric investigation on the hybrid damping scheme was undertaken to both understand the interaction of the
various dampers and to optimisation the overall damping performance. In conjunction with the hybrid damping scheme, an additional method of energy generation derived from the vibrating TMD was proposed and explored.

Finally, following on from the investigations into the TLCD and the coupled offshore wind turbine – TLCD system, a long term probabilistic model of the wind turbine response was simulated. The wind turbine, with rotating blades, is modelled as a Multi-Degree-of-Freedom (MDOF) structure and is dynamically excited by a time-varying wind excitation. The acceleration response of the turbine system with and without TLCD was coupled with a probabilistic description of the dynamic properties of the wind turbine to produce an annual distribution of the wind hazard. Wind induced fragility curves were presented deriving from an annual distribution of the wind hazard to obtain a distribution of the acceleration response for various levels of wind speed. The overall effect of reducing unavailability in wind turbines by implementing TLCDs is obtained through use of the fragility curves.

7.2 CONCLUSIONS

This section presents the conclusions that may be drawn based on the numerical modelling and experimental studies presented throughout the thesis.

The tuning of the natural frequency of the TLCD with the fundamental mode of a wind turbine reduces to a considerable extent the vibration response of the turbine. The effectiveness of the TLCD in suppressing harmonic excitations at resonance for SDOF systems has been experimentally demonstrated. It is observed that changes in the orifice size in the pipe have a direct impact on the damping characteristics of the damper. Orifice sizes to optimise the damping of the TLCD based on the experimental results were proposed for the SDOF system. The experimental results and the equivalent damping ratio values obtained suggest that TLCD performance is dependant on excitation amplitude. In specific cases, due to an optimal coefficient of damping not being provided, the structure-TLCD response may increase for narrow banded (harmonic in the case investigated) excitations. It was found that orifices sizes between 0.5d and 1d produce optimal vibration reduction performance.
throughout the frequency spectrum. Thus, even though a TLCD may be specifically designed to operate at a certain frequency that induces resonance in the structure, by controlling the opening ratio one can still obtain a satisfactory performance from the TLCD at non-resonant frequencies. This proves useful as soil properties, structural properties and excitation levels may alter over the lifetime of an offshore wind turbine. By changing the opening ratio, one may continuously optimise the performance of the TLCD with regards to varying system parameters. Through expressions formulated for the structure-TLCD system response, one may accurately simulate the real life performance of the system. This observation was validated by comparing the theoretical response with the experimental response of the SDOF-TLCD system.

By implementing both water and glycol in the TLCD, one achieves significant structural response reduction to that of the undamped structure. The same can be said regarding the MR fluid, which also achieves substantial reductions in the structural response. The non-linear quadratic damping theory accurately models the behaviour of a TLCD in the resonant range of excitation frequencies where there is significant liquid motion leading to increased damping through head loss in TLCDs in addition to the effect of tuning. The difference in viscosity between water and glycol does not transmit to a significant difference in the passive damping characteristics of the system. Hence, glycol or glycol/water mixtures are viable for structures were freezing temperatures might abound. In terms of volumetric efficiency, MR fluid is almost as effective as water within the TLCD in passively damping structural responses on the top of a SDOF structure. This is promising for the potential use of MR fluids in a semi-active damping context within the TLCD where the damping due to head loss is controlled. In addition, under certain circumstances, such as low excitation forces or in a disruption of the electrical supply to the semi-active control system, the passive damping properties of the MR-TLCD liquid will be exclusively relied upon. The displacement response of the MR fluid within the TLCD is significantly lower than the other liquids, indicating its potential use in situations where space for the vertical columns in the TLCD is limited. It is observed both experimentally and theoretically that an appropriately low viscous MR fluid may be implemented in MR-TLCDs, thus making MR-TLCDs practical for applications in wind turbines. An MR fluid with relatively low viscosity may be relied upon in an MR-TLCD, although the designer
will have to make adequate arrangements to cater for the increased mass ratio and ensure that the benefits of the semi-active MR-TLCD outweigh those of the passive TLCD. Appropriate MR fluids that perform efficiently in the MR-TLCD must be formulated.

Through modelling an offshore wind turbine as a MDOF system with rotating blades under wind and wave excitations, one may present an accurate dynamic portrait of a very complex slender structure. The WTT assembly was discretized into ultimately an eighteen DOF system, the tower with dampers incorporating nine DOFs and the three blades each represented by three DOFs. The wind excitation may be accurately represented by the Kaimal spectrum. Spatial correlation is accounted for by coherence function and simulating the excitation in the time domain. As the input PSDF has cross drag terms, spectral decomposition theory and random processes with orthogonal increments are used to generate spatially correlated time histories and apply these at respective nodes in the offshore wind turbine. Cases of the blades lumped at the nacelle and of rotating blades were examined. Wave excitations were generated using the JONSWOP spectrum and then simulated in the time domain. A correlation between the wind and wave excitations using joint distribution is possible through integrating the mean wind speed with the formation of the JONSWAP spectrum. When an offshore wind turbine is equipped with a TLCD and subjected to wind and wave forces, reductions of up to 55% in the tip response of the same system without TLCD may be achieved. The TLCD performs consistently well, in terms of reducing base bending moments and tip responses in the structure, across all excitations considered. The option open to design the wind turbine more efficiently with less steelwork and less foundation expenses is thus afforded when one implements TLCDs in offshore wind turbines. It was also observed through use of the rain-flow calculation method for fatigue, that implementation of a TLCD in an offshore wind turbine greatly increases the fatigue life of the wind tower assembly.

When an offshore wind turbine is fitted with the BI-TMD-TLCD damping system, substantial reductions in displacement and acceleration response are attained. In one case, when the WTT is subjected to fluctuating wind and wave loadings deriving from a mean wind speed of 30 m/s, the tip acceleration and displacement response are reduced from the undamped WTT by 72% and 68%, respectively. Thus, the BI system
is successful in preventing the predominant energy-containing frequencies of the excitation from entering the superstructure and hence drastically reduces in the acceleration and displacement response of the WTT nacelle. The mass ratio of the BI system has a large impact on the performance of the damping system. The mass of the BI system, and hence stiffness, must be such as to prevent substantial lateral displacement responses at the platform level. The mass ratio of the BI system to the SS2 should be greater than 3.5%, with 10% fulfilling the operational performance of the BI system. Additional energy may be generated from the vibration of the TMD, which ultimately derives from the wind and wave excitations. It was found that under operational conditions, an additional 13.1 kWhs of energy are produced from the WTT.

Wind-induced acceleration fragility curves introduced in this study indicate that undamped wind turbines are prone to failure from acceleration-sensitive components. High acceleration levels are predicted for the nacelle at the onset of current shutdown wind speed criteria ($v_w \sim 25$ m/s). For acceleration thresholds $a_{LS}$ of 5, 10, 15, and 20 m/s$^2$, the probabilities of being exceeded for the undamped WTT are 86%, 66%, 51%, and 40%, respectively. These levels of acceleration can clearly induce malfunctioning in critical wind turbine components such as generators, inverters, mechanical controls, electrical controls, and any movable part. When a TLCD is implemented in the WTT, for acceleration thresholds $a_{LS}$ of 5, 10, 15, and 20 m/s$^2$, the probabilities of being exceeded for the undamped WTT are 67%, 44%, 30%, and 23%, respectively. Thus, TLCDs in WTTs significantly reduce the probability of wind turbine subcomponents malfunctioning.

7.3 RECOMMENDATIONS FOR FURTHER STUDY

The application of a smart sliding isolation system with independently variable semi-active MR dampers to an offshore wind turbine could be investigated. With offshore wind turbines being isolated structures at sea and subjected to large wind and wave excitations, the reliability of the semi-active system would have to be studied in depth.
Floating wind turbines with TLCD could be studied. Floating wind turbines negate the need for expensive foundation systems plus allow the possibility of implementing large scale wind turbine farms in very deep water. TLCDs would help reduce the vibration response of the wind turbine systems and also reduce the stress levels in the anchored cables holding the wind turbines in place.
REFERENCES


Glotman Simpson Consultant Engineers. (2001). "One Wall Centre Project."


Stewart, R. H. "Introduction to physical oceanography." Texas A&M University.


A.1 Linear Variable Differential Transformer

The essence of achieving accurate results in any testing procedure lies with the accurate measurement of displacement changes of the model structure. An accelerometer may have been used to calculate any displacement changes but it was felt that a linear variable differential transformer (LVDT), which directly gives displacement changes, would prove to be the more efficient and accurate device for the testing undertaken in Chapter 1. The lack of friction between the hollow shaft and the core prolong the life of the LVDT and enable very good resolution. In addition, the small mass of the core allows for good sensitivity in dynamic tests.

The LVDT relies on an inductance principle to derive displacement changes in the structure at which it is connected. The LVDT is constructed with two secondary coils placed symmetrically on either side of a primary coil contained within the hollow cylindrical shaft. Any induced mechanical movement will change the characteristics of the flux path and these changes are monitored circuit electrical output.

Figure A.1 LVDT circuit
When the primary coil is excited by an a.c. signal, voltages are induced within the two secondary coils. The circuit is shown above in Figure A.1. With the variation of the core positioning, the induced voltages \( v_1 \) and \( v_2 \) when summed up give the output voltage \( v_0 \), which reflects the displacement change of the mechanical device to which the LVDT is attached, as long as the displacement is within the working range of the LVDT. For the LVDT used in this project, the range was approximately 100mm. The LVDTs started at an amplitude value of 0 on the system 6000 programming graphs when the arm was 76mm extended. Thus, it proved to be time consuming setting up the LVDTs in order that they were placed horizontally in elevation and plan views, and also with the arm extended 76mm.

Originally, to provide a strong bond between the LVDT head and the structure, the head was glued onto the base plate and the mass platform of the structure. However, the glue proved inadequate at remaining in place over successive tests, so a connection involving two brackets, shown below in Figure A.2, was applied. An added bonus with this connection was that, whilst providing a virtually unbreakable connection in terms of the forces that would be applied to it over the course, the LVDT could be easily dissembled from the structure, which proved effective when the test conditions had to be altered. Figure A.3 shows a rendered three dimensional representation of the LVDT – Structure interaction.
Figure A.2 LVDTs

Figure A.3 Representation of the LVDT-Structure connection
A.2 Accelerometer

The accelerometer used in the testing was a piezoelectric accelerometer, accurate to changes of 0.01g. Inside a piezoelectric accelerometer, the sensing element is a crystal which has the property of emitting a charge when subjected to a compressive force. In the accelerometer, this crystal is bonded to a mass such that when the accelerometer is subjected to a 'g' force, the mass compresses the crystal which emits a signal. This signal value can be related to the imposed 'g' force. The accelerometer can be easily attached to the test structure, without imposing any frictional resistance on the structural response.

A.3 Signal Generator and Magnetron

The frequency generator, shown in Figure A.4, that was utilized in the experiment was used because a wide variety of amplitudes and frequencies could be inputted instantly into the system in order to test the model structures under a variety of loading conditions. The signal generator allowed the various frequencies and amplitudes to be implemented into the system via two control dials. The diversity of frequencies operated under a range of between 1.5Hz and 20Hz. It was found that when a frequency was inputted into the generator, the generated frequency did not match the frequency value given on the dial. Thus, a calibration was performed which can be seen below in Table A.1. The amplitudes operated under a range of between 0.5mm and 15mm, although the generator tended to turn off when the amplitudes reached 6.5mm. The amplitudes were also calibrated from the values on the input dial. This can also be seen below in Figure A.4.
In Table A.1 the Frequency Table is in terms of Hertz and the Amplitude table on the right hand side is in terms of mm.
The signal generator is connected to the magnetron. The magnetron in Figure A.5, which contains an electromagnet, can receive a signal and subsequently change the value of the frequency or amplitude that it is inducing in the shaking table. The main problem with the magnetron materialized when lengthy tests were performed. When the magnetron overheated, it tended to stop, and remain motionless until it cooled off appropriately. This problem was overcome by connecting the magnetron to a motorized fan, which had the ability to consistently blow cool air into the magnetron, hence preventing overheating. Also as the frequencies increased, it was seen that the sine waves became more inaccurate. This problem arose because the electromagnet was not able to cope with high frequencies, and instead of outputting the correct frequencies, the magnet "buzzed", thus reducing the amplitudes of the base of the structure. However this problem did not affect the results significantly as the testing was concerned with low frequencies.
A.4 The Model 6000 Scanner

The System 6000 was chosen as the scanner as it provides fast simultaneous acquisition and digitization of multiple channels of various analogue inputs. The system incorporates full-featured software that provides graphical presentations along with data reduction and scanning interval control. The System 6000 was connected to a new PCI interface card, which made data acquisition quick and efficient. Thus, a graph of an experiment could be viewed within seconds of the experiment taking place.

As noise was an all too common aspect of working in the laboratory, it was important to note that the System 6000 contained Input Shielding, which had the ability to reduce noise and protect the integrity of the bridge signal from Electro-Static Discharge (ESD) and Radio Frequency (RF) noise. The shield connection was connected to Pin 9 of the input connector, which formed the soldering side of the Input Plug Connection. For additional protection, the external transducer elements were completely shielded.
The System 5000 was also available in the laboratory, however in contrast to the System 6000’s ability to take an LVDT reading at up to 10,000 times per second, the System 5000 was limited to 10 readings per second. To this end, the System 6000 was chosen as at least 1,000 LVDT readings per second had to be taken in order to provide an accurate plot of the displacements of the model structure. Figure A.7 below shows the System 6000 scanner.

![System 6000 Scanner](image)

Figure A.7

**A.5 The Tuned Liquid Column Damper**

In order to optimize the performance of the TLCD, and to make the solution of the TLCD for a structure viable both structurally and economically, it is a prerequisite that the mass ratio (mass of the liquid/ mass of the structure) of the structural system is kept below 5%. As the mass of the structure is 8kg, a weight limit of 400g was imposed in the design of the TLCD.
Furthermore, it quickly became axiomatic that the length of the pipe section was obliged to be in the range of 60-90mm, so that the liquid would have sufficient room to travel through the pipe and interact with the orifice. Conversely, so that the two tanks could function with an adequate supply of liquid, a suitable portion of the liquid length \( L_L = 100.9 \text{ mm} \) was required to provide sufficient space to build the tanks. If the length of the pipe had of been in the order of 90mm, then only 5mm would have been rationed to each tank for the length of the liquid, \( L_L \), which would have made the construction and operation of the tanks impractical. Consequently, a pipe length of 64mm was selected, as it was predetermined that enough length was left over to construct satisfactory tanks, whilst allowing sufficient interaction between the flowing liquid and orifice. Also, a pipe length of 64mm provided enough space to unscrew the bolts that held the orifice plate in position.

In order to enable a variety of different orifice openings to be tested, an adequate pipe diameter was required. After numerous iterations it was decided that a pipe diameter of 35mm would provide a commodious selection of orifice openings to test, whilst keeping to weight and liquid length limits (as the diameter of the pipe has direct implications on the design of the height of the water in the tanks). The formula for the weight of liquid in the pipe is given by

\[
m_p = \frac{\rho \pi d^2}{4} \cdot L_{pl},
\]

where \( m_p \) is the mass of the liquid in the pipe, \( \rho \) is the density of water which is taken as \( 1000 \text{ kg/m}^3 \), \( L_{pl} \) is the length of the liquid in the pipe and \( d \) is the diameter of the pipe. With \( L_{pl} = 64 \text{ mm} \), solving Equation (A1) yields \( m_p = 61 \text{ g} \). And assuming that the height of the water in the tanks is \( 50 \text{ mm} \), the total weight of the TLCD becomes \( 381 \text{ g} \), which gives a mass ratio of 4.95%.

For the length of the Liquid, \( L \), it can be seen that it is a summation of the length of the pipe, the length of the fillet connecting the pipe and the tanks, and the length from the top of the fillet to the top of the water.
Summing up these lengths yields \[ 64 + 2 \cdot \left( \frac{2\pi r}{4} \right) + 2 \cdot 10 = > L \approx 115.4\text{mm}. \]

Thus, one yields a natural frequency for the TLCD of 13.04 radians per second, which matches up quite well to the natural frequency of the model structure.

A schematic showing the elevation of the TLCD can be seen in Figure A.8.

\[ \text{Figure A.8} \]

Figure A.9 shows a rendered three dimensional view of the TLCD system.
The predominant aspect of the TLCD’s construction and operation concerned the ability to change the orifice size in the TLCD. Thus, a system was devised in which the centre of the TLCD could be dismantled by unscrewing four bolts, allowing the orifice to be changed. Figure A.10 shows a cross section of the unscrewed components of the TLCD.
The pen in the photo is an indication of the scale of the TLCD. It can be clearly seen how the screws connect the two halves of the TLCD together. The blue substance around the LHS halve of the TLCD is silicon, which guaranteed a water tight connection. The orifice plate is 6mm thick, so 3mm were left on each side of the connection so that the orifice could fit perfectly into the connection, and hence guarantee a water tight connection. This can be seen above where the black pipe resides 3mm back from the orifice connection.