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Aerodynamic Analysis of a Wind Turbine with Telescopic Blades

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Abstract

Telescopic Blade Wind Turbine Systems offer great benefits to the wind industry and in particular to low speed wind site for potential development and enhancement. An inherent feature of a telescopic blade system is the discontinuity or step change in chord that exists between the fixed inner blade section, and the movable or telescopic outer blade section. This is its main distinctive feature when compared to a standard wind turbine blade, and it is therefore a major point of difference from an aerodynamic performance point of view as well.

This thesis attempts to investigate the effect of the sudden change in chord of a Telescopic Blade Wind Turbine System, using a combination of experimental, analytical, and computational studies. In particular, it is aimed at studying the dynamics of the flow around the step change region and its influence on blade aerodynamics; at investigating the influence of the step change and telescopic blade parameters on the turbine performance and energy output; and at developing a correlation for the losses that arise from the step change region.

Performance testing was carried out at model-scale in the wind tunnel, of a Telescopic Blade Wind Turbine System with two-stage telescopic blades (i.e. blades with two sections of different chords), having chord ratios of 0.6 and 0.4, and for blade extensions ranging 0 – 40% (in length of the first section). The power coefficient $C_p$ of the wind turbine was found to decrease with extension, with a 25% decrease in maximum $C_p$ obtained for a 20% blade extension. This is attributed to additional losses arising from the step change in chord. It was hypothesised that the step change region induces vortex rollup in a manner similar to that which happens at the tip. Correlations developed to quantify losses arising from the step change in the chord of a telescopic blade are in good agreement with experimental data, and this would pave the way for improved performance predictions.

Investigations into the aerodynamic performance of a single telescopic blade with a step change in chord were also carried out in the wind tunnel. From the surface static pressure measurements and flow visualisation studies, it was established that the sudden change in the blade chord causes an abrupt drop in pressure around the step region, which causes vortex roll, confirming the hypothesis drawn from performance testing.

Energy output analyses reveal that the percentage increase in the annual energy output for the telescopic blade wind turbine system over the corresponding non-telescopic blade system could be as high as ~48% for a blade with 40% extension at a chord ratio of 0.6 in wind class...
2. For the range of shape factors, chord ratios and blade extensions studied in this research, the reduction in annual energy output due to the step change in blade chord is found to be in the range of 15 to 1% for Wind Class 2 to 7 respectively.
Dedicated to my beloved Family
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# Nomenclature

- **a**: axial induction factor
- **\( \dot{a} \)**: angular induction factor
- **A**: wind turbine disc swept area (m²)
- **\( AR_{\text{eff}} \)**: effective aspect ratio
- **B**: number of blades
- **c**: scale parameter in performance chapter
- **\( c \)**: chord in blade aerodynamics chapter
- **C₁**: tip blade chord (mm)
- **C₂**: root blade chord (mm)
- **\( C_p \)**: power coefficient in performance chapter
- **\( c_p \)**: pressure coefficient in blade aerodynamics chapter
- **\( C_t \)**: thrust coefficient
- **\( C_T \)**: Tip blade chord
- **\( C_R \)**: Root blade chord
- **F**: total standard blade loss
- **Fx**: Force in X - direction
- **Fy**: Force in Y - direction
- **Fz**: Force in Z - direction
- **F_{ave}**: total standard blade loss
- **F_{hub}**: hub loss
- **F_{Step}**: step loss
- **F_{Telescopic}**: total telescopic blade loss
\( F_{\text{thrust}} \)  
thrust force (N)

\( F_{\text{tip}} \)  
tip loss

\( G \)  
constant

\( H \)  
constant

\( k \)  
shape factor

\( L_R \)  
Root blade length

\( L_T \)  
Tip blade length

\( M_x \)  
Moment in X - direction

\( M_y \)  
Moment in Y - direction

\( M_z \)  
Moment in Z - direction

\( P_e \)  
electrical power output

\( P_{\text{eR}} \)  
rated electrical power

\( p_i \)  
surface pressure at location \( i \) on the surface

\( p_{\text{stagnation}} \)  
stagnation pressure at the Pitot tube

\( P_t \)  
turbine shaft power output

\( P_{\text{t,ave}} \)  
average wind turbine power

\( p_{\infty} \)  
pressure in the free stream

\( r \)  
radius along the blade (m)

\( R \)  
wind turbine radius (m)

\( Re_x \)  
Wind tunnel Reynolds number

\( Re_{\text{root}} \)  
Reynolds number based on root blade chord

\( Re_{\text{tip}} \)  
Reynolds number based on tip blade chord

\( R_R \)  
root blade (m)
T  wind turbine torque (N m)

$U_\infty$  free-stream velocity (m s$^{-1}$)

$U_T$  wind velocity at the rotor (m s$^{-1}$)

V  inflow wind speed (m s$^{-1}$)

$V_c$  cut-in velocity (m s$^{-1}$)

$V_R$  rated velocity (m s$^{-1}$)

$V_F$  cut-out velocity (m s$^{-1}$)

X  distance from wind tunnel entry to test section

2D  two dimensional

3D  three dimensional

*Greek Letters*

$\omega$  rotational speed (rad s$^{-1}$)

$\rho$  air density (kg m$^{-3}$)

$\phi$  angle of relative wind(°)

$\lambda$  tip speed ratio

$\Gamma'(\chi)$  gamma function

$\sigma$  standard deviation

$\bar{P}_w$  average power in the wind

$\bar{v}$  mean speed

$\eta$  generator efficiency

$\delta$  boundary layer thickness

$\alpha$  3D angle of attack

$\alpha_{eff}$  effective angle of attack
\[ \alpha_i \]  induced angle,
\[ e \]  span efficiency (Oswald) factor

**Abbreviations**

AEO  Annual energy output
AR  aspect ratio
BEM  Blade element momentum theory
CF  Capacity Factor
CFD  Computational Fluid Dynamics
HAWT  Horizontal axis wind turbine
MPPT  maximum power point tracking
pdf  probability density function
PMG  permanent magnet generator
RBR  retractable blade rotor
rpm  revolutions per minute
TBHAWT  Telescopic blade horizontal axis wind turbine
VRG  Variable Ratio Gearbox
WECS  Wind energy conversion system
WTCS  Wind Turbine Conversion System
CR  chord ratio
ER  extension ratio
x  chord wise position on blade
Chapter 1. Introduction and background

The current energy crisis with limited reserves of fossil fuels compounded by inflated prices and their impact on the environment has led research institutions, organizations, and governments to develop new and renewable energy technologies. Of these, wind energy has been one of the largest sources of new electricity generation in many parts of the world in recent times. Wind energy is abundant, free, environmentally friendly and inexhaustible.

The world market for new wind turbines installed set a new record in the year 2011 and reached a total size of 42 GW, following 37.6 GW in 2010. According to the preliminary data gathered by the World Wind Energy Association (WWEA) and published on the occasion of the 3rd WE20 by 2020 conference in Coimbatore/India, the total capacity worldwide has come close to 239 GW (Figure 1.1), enough to cover 3% of the world's electricity demand [1].

According to the WWEA [1], amongst the individual countries, China kept its strong position reaching a level similar to that of the previous year 2010: China installed around 18 GW of new wind turbines within 2011, coming to a total capacity of 63 GW, more than one fourth of the global wind capacity. The second largest market for new wind turbines was again the USA with 6.8 GW, followed by India (2.7 GW), Germany (2 GW) and a surprisingly strong Canada with 1.3 GW of new installed capacity. Spain, France and Italy each added around 1 GW [1].

A strong increase in wind power utilization can be observed especially in the emerging markets, like China, India, Brazil, and Mexico [1]. This opens new windows for further growth, as these countries do have an increasing need for electricity which can be matched by wind power in a very economic, safe and timely way. On the other hand, several of the European markets showed stagnation or even a decrease.

According to the Global Wind Energy Council [2], New Zealand’s wind energy resource is sufficient to meet its annual demand several times. New Zealand installed 109 MW in 2011 for a total of 623 MW, a 20% increase in cumulative installed capacity [3]. Wind now supplies just over 4% of New Zealand’s electricity with no subsidy or special treatment whatsoever from the government.
There is no explicit target for wind energy set by the New Zealand government but it is planned that by 2025, 90% of power supply is to be sourced from renewable sources; to date, 70% of the energy supply is generated using hydroelectric and geothermal sources and only 26% is generated using non-renewable sources such as natural gas and coal [2, 4]. New Zealand’s wind energy industry is competing successfully with other energy sources; this globally unique achievement demonstrates the cost-effectiveness, competitiveness and the can-do attitude of wind energy sector without any form of government subsidy [2].

From the type of growth in 2011, despite the international economic meltdown, the wind energy sector looks very promising in terms of future renewable energy source.

![World total installed capacity (MW)](image)

Kaldellis and Zafrakis [5] in their paper “The wind energy (r)evolution: A short review of a long history” did a review of the wind energy history emphasizing on the main issues of global market facts, technology, economics, environmental performance, prospects and research and development (R&D) of wind power, providing some insight and presenting the highlights for each of the fields. From their review, the dynamics of wind power at the global energy scene during the last thirty years is illustrated. It was argued that the perspective of exceeding 1 TW of wind power installations by 2030 seems feasible, especially if the challenges introduced by the need of each country to safeguard security of supply and
promote clean power technologies are considered. Besides that, although the leading role of the European Union (EU) throughout the period of wind energy development has been designated, the return of the USA and the tremendous growth of the wind energy industry in China are also reflected. On top of that, the gradual adoption of wind energy by several developing countries of the world, and its ability to largely substitute fossil-fuelled power generation in the years to come are also illustrated.

In spite of the rapid growth, wind energy technology is nevertheless facing new technical challenges to achieve cost effectiveness especially in lower wind speed regimes. The decrease in cost of wind power generation has slowed due to the maturity of the technology. Notwithstanding this, designing wind turbine blades requires a lot of technical challenges to be overcome. “Successes in lower wind regimes and mature turbine design methodologies will require that more technically challenging innovative designs be explored” [6]. The Department of Energy (DOE) in the US initiated the Low Wind Speed Technology (LWST) program to move research in the direction of capturing more wind energy in low wind speed areas. The main issue surrounding the system is the cost, and if a Wind Turbine Conversion System (WTCS) can capture more energy, then the cost of energy goes down over its useful life. In designing a WTCS, there is a trade-off between energy capture and the mechanical loading of the system. A large diameter WTCS will definitely capture more energy, but when the wind speeds are higher, it will induce higher mechanical loading on the system. This will require stronger towers, foundations, bearings, and other elements which will increase the cost of the system. In any case, the rotor design of a WTCS will have a direct impact on both its aerodynamic performance as well as on its loading.

1.1. Overview
Several researchers have contributed insight to rotor design. Snel [7, 8] describes wind turbine aerodynamics in general and gives an overview of the available methods to compute the aerodynamic rotor performance. Tangler [9] gives a short historical overview of the rotor design investigations and aerodynamic optimization of rotors are described by Fuglsang and Madsen [10], Giguere and Selig [11] and Nygaard [12]. Details of rotor design to increase the power efficiency have been investigated by several researchers. Johansen et al [13] and Madsen et al. [14] especially investigated the root part of rotors and found that a new root design of the rotors did not increase the power performance significantly. Larwood and Zutech [15] described the dynamic analysis of a swept wind turbine blade concept. The swept-blade concept was used for increased energy capture without an increase in the turbine
loads. The blade works by twisting to feather under aerodynamic loads at the outboard region. The conceptual design of the blade resulted in a 28 m blade radius for eventual testing on a nominally 50 m diameter turbine. The blade was modelled with codes developed by the National Renewable Energy Laboratory, USA. Comparisons were made to an un-swept rotor of the same diameter and a baseline 50 m diameter rotor. The results demonstrated the twisting and load-reduction behaviour of the swept rotor. Their results demonstrated little detriment to the power curve with swept rotor and substantial power increase over the 50 m baseline in below rated power. One thrust of the LWST program was in advanced rotor control concepts. The main idea was to increase the rotor diameter for a given turbine rating without increasing the load envelope.

One concept investigated by Lobitz et al [16] was to incorporate flap/twist coupling into the rotor with off-axis fibre orientation into the blade construction. As the blade deflects in the flap-wise direction, the tip twists towards feather and reduces the aerodynamic loading.

A different physical mechanism, leading to similar aerodynamic load reduction, was analyzed by Zuteck [17] and proposed earlier by Liebst [18]. This concept was to sweep the outboard-region rotor plan-form in the plane of rotation aft of the pitch axis. The loads generated at the tip would introduce a moment about the pitch axis. With sufficient blade torsional flexibility, the tip would twist towards feather, thus reducing the loads. Zuteck [17] outlined key design parameters for the concept that included the sweep-curve geometry and tip sweep. It was estimated that 4° to 7° of tip twist would be needed to shed loads for a 30 m blade. This amount of tip twist would require that the torsional stiffness of the blade be decreased in comparison to typical straight-blade designs, but it was shown to be feasible through appropriate design modifications. Liebst [18] had also concluded that the torsional stiffness would have to be reduced for the concept to be successful.

Larsen et al [19] and Bossanyi [20] have shown that active load reduction for megawatt size wind turbines can alleviate load increments from yaw-errors, wind shear, gusts and turbulence considerably. The load reduction is realized either by pitching the blades independently (individual pitch) or equally for each blade but with a phase shift between each blade (cyclic pitch). According to Larsen et al [19], a comparison to collective pitch, where all blades are pitched equally, shows that cyclic pitch can reduce the blade flap fatigue loads at the hub by 15%, while it is reduced 28% when using individual pitch. They also show that the extreme load for the blade flap at the hub can be reduced 22% when using cyclic pitch.
and 14% when using individual pitch. This means that there is a significant potential in using advanced pitch control for load reduction. Recently Shen et al [21] studied the wind shear effects on the turbine blade with individual blade pitch control. They found that individual pitch control to control the fluctuating blade root flap-wise moment can reduce the flap-wise fatigue damage remarkably while the blade root edge-wise moments are less sensitive to the varying blade pitch than the blade root flap-wise moments.

Nakafuji et al [22] have investigated the aerodynamic loads on 2D airfoils in a wind tunnel by changing the airfoil trailing edge shape using a dynamic Gurney flap-like device, also called a micro-tab. They conclude that although the specifics of load control limitations will require further research, the wind tunnel results demonstrate a significant potential for using micro-tabs for active load control.

Several researchers [23-29] have shown that a further load reduction is possible by using Active Trailing Edge Flaps (ATEFs) by enabling the trailing edge to move quickly and independently at different radial positions of the blade. Local fluctuations in the aerodynamic forces can be compensated for by deformation of the airfoil geometry, like a hawk keeping its head and eyes fixed in standstill by fast adjustment of the shape of its wings.

For the control of wind turbine loads, Basualdo [26] carried out investigations with a 2D potential-flow camber-line panel model of an airfoil suspended with springs and damper and exposed to a turbulent wind field. The study showed that the standard deviation of the airfoil position normal to the chord, i.e. the direction corresponding to the flap direction on wind turbine blades, can be reduced significantly in a 2 s period. On the other hand, Troldborg [29] studied the aerodynamic performance of a 2D airfoil equipped with different ATEFs of different shapes and dynamic sequences using 2D Computational Fluid Dynamics (CFD) on a modified Risø-B1-18 airfoil and found that an oscillating airfoil superimposed with an oscillating ATEF could significantly reduce the amplitude of the lift generated on the airfoil for a wide range of reduced frequencies. With a 2D aero-elastic model, Buhl et al [27] showed that the standard deviation of the normal force on the 2D airfoil suspended with springs and dampers could be reduced by up to 95% if the airfoil experienced a sudden change in the wind speed and up to 81% if the airfoil experienced turbulent flow with 10% turbulence intensity. The predicted reductions depended strongly on the control algorithm and the delay in the signals from the sensors to the ATEF. Using a simplified aero-elastic model of a Vestas V66 wind turbine, Andersen et al [23] found that the equivalent flap-wise blade root
Introduction and Background

moment was reduced 60% for inflow with 10% turbulence using 7 m ATEF on the 33 m blade, where the actuators consume 100 W/m at maximum consumption rate.

The aerodynamic model used in the works of Buhl et al [27] and Andersen et al [23] was developed by Gaunaa [28]. The model is a computationally efficient unsteady 2D potential-flow model for the aerodynamic forces of a thin airfoil undergoing deflections of the camberline. A further development of this model including dynamic stall was presented by Andersen et al [24, 25].

Recently, Christian et al [30] showed in their wind tunnel tests that it is possible to control the loads on a wind turbine airfoil with an ATEF. They showed that even though optimal use of the ATEF shows load reductions of up to 80%, on the other hand, improper control of the ATEF should be avoided because this can cause load increases.

Sorensen [31] has recently provided a review of the Aerodynamic Aspects of Wind Energy Conversion. The review is focused on aerodynamic modelling, as it is used by industry in the design of new turbines, and on state-of-the art methods for analyzing wind turbine rotors and wakes. Specifically, the basics of momentum theory, which still forms the backbone in rotor design for wind turbines, are introduced along with recent results from computational fluid dynamics (CFD) simulations of wind turbines and wind turbine wakes.

Based on the Blade Element Momentum (BEM) theory, Vaz et al [32] proposed a mathematical model for the horizontal-axis wind turbine design, taking into account the influence of the wake on the rotor plane in the general form. This influence is considered when the tip-speed-ratio is small, justifying the development of formulations that predict the effects of the wake on the rotor plane. The proposed mathematical model in their work is an extension of the model developed by Andre et al [33] based on the BEM method, using the Glauert’s model modified for the wind turbine design. Similarly, Lanzafame and Messina [34] used the BEM Theory to optimise the turbine performance and also discussed the power curve of a micro wind turbine design [35].

Recently Okulov and Sørensen [36] developed an analytical method to determine the loading of an optimum wind turbine rotor. The method, which is basically a modification to the original model of Goldstein [37], is based on a new analytical solution to the wake vortex problem that enables the optimum circulation distribution at all operating conditions to be determined. In contrast to earlier models, the new model is consistent with the general
momentum theory and enables for the first time the theoretical maximum efficiency of rotors with an arbitrary number of blades to be determined.

The wake interference effect on the performance of a downstream wind turbine was investigated experimentally by Adaramola and Krogstad [38]. Two similar model turbines with the same rotor diameter were used in the study to find the effects on the performance of the downstream turbine of the distance of separation between the turbines and the amount of power extracted from the upstream turbine. The effects of these parameters on the total power output from the turbines were also estimated. It was found that the reduction in the maximum power coefficient of the downstream turbine is strongly dependent on the distance between the turbines and the operating condition of the upstream turbine. Depending on the distance of separation and blade pitch angle, the loss in power from the downstream turbine varies from about 20 to 46% compared to the power output from an unobstructed single turbine operating at its designed conditions. By operating the upstream turbine slightly outside this optimum setting or yawing the upstream turbine, the power output from the downstream turbine was significantly improved. In their study, they showed that the total power output could be increased by installing an upstream turbine which extracts less power than the following turbines. By operating the upstream turbine in yawed condition, the total power output from the two turbines could be increased by about 12%.

Philips [39] considered increasing the wind speed through the plane of a rotor using diffusers, but the idea was proven not to be feasible due mainly to the prohibitive costs in the wind resistant design of the diffuser. Ohya et al [40] similarly investigated a wind turbine system consisting of a diffuser shroud with a broad-ring flange at the exit periphery and a wind turbine inside it. They emphasized on placing of the flange at the exit of the diffuser shroud which generates a low-pressure region in the exit neighbourhood of the diffuser by vortex formation and draws more mass flow through the wind turbine inside the diffuser shroud. In order to obtain a higher power output of the shrouded wind turbine, they examined the optimal form of the flanged diffuser, such as the diffuser open angle, flange height, hub ratio, centre body length, inlet shroud shape and so forth. As a result, a shrouded wind turbine equipped with a flanged diffuser was developed, and power augmentation for a given turbine diameter and wind speed by a factor of about 4–5 compared to a standard (bare) wind turbine was demonstrated.
Several attempts at improving the performance of wind turbines have been made using different airfoil designs, different philosophies in the rotor design and in the control. The different rotor design philosophies have involved high design lift versus low design lift, large versus small root chord, use of winglets, and (active) stall regulation with constant rotor speed versus pitch regulation with variable rotor speed [41].

Afjeh and Keith [42] have discussed the idea of tip controlled rotor blade as shown in Figure 1.2. Tip control blades have been used to regulate the output power of large horizontal axis wind turbines from feather to full power. The rotor blades are divided into two sections: an inboard section with fixed pitch and an outboard section with variable pitch. Corrigan et al [43] found that the tip controlled rotors have better start-up and shut-down abilities, simpler pitch control system, reduced system weight and more flexibility in airfoil selection.

Lanzafame and Messina [44] also proposed a new design of wind turbine blade. The new wind turbine blade was subdivided into two, each with a different pitch angle to optimise aerodynamic flow, absence of twist, and carrying a variable chord along the blade itself as shown in Figure 1.3. The new blade reveals some energy loss due to the tip vortices of each blade part, which could be minimised by winglets. A numerical code based on blade element momentum theory was developed to design and evaluate the performance of the new wind turbine. The code was validated with the experimental data of Lindenburg [45].
In recent times, some researchers have examined extending or telescoping the blades as a means of increasing energy capture.

Figure 1.3 Double pitch – non-twisted WT blade [44].
1.2. Telescopic wing and wind turbine blades
The following sections give a brief overview of Telescopic Wing and Telescopic Wind Turbine Blade systems. Some work has been carried out in terms of telescoping wing concepts in the aeronautical sector with US patents [46-48] on it. A very brief description of what transpired from this invention as shown in the patents is discussed alongside some advantages of the Telescopic wing concept. Similarly, some work has been done on Telescopic Blade wind Turbines. This concept also has US patents (Energy unlimited [49]) on them and to date very little technical information regarding this concept is available.

1.2.1. Telescoping wing
Telescopic wing technologies [47, 48] such as Telescopic wing tips (Figure 1.4) or Convertible fixed wing (Figure 1.5) were proposed as early as the 1940s but there is little evidence that the early designs were ever built, or flown. The only information on those are found in US Patents, which give manufacturing guidelines and technical drawings but do not refer to any aerodynamic or structural studies or testing [46].

![Figure 1.4 Telescopic wing tips [48]](image)
More recently, in 1997, Gevers Aircraft, INC. developed a 6-seat 'triphibious' aircraft designed for unprecedented speed, utility, safety, and ruggedness. It uses a telescopic wing to adapt the aircraft geometry to the flight conditions. The wing is designed for high-speed cruise when retracted and enhanced low speed capabilities when extended. It is composed of a fixed centre section and two extendable outer sections, using an overlapping extension spar system. The centre section is a high-speed wing (low drag and strong) and the completely retractable high lift section moves in a span-wise direction. It appears that the aircraft actually flew, but no flight test data could be found [46].

Figure 1.5 Convertible fixed wing [47]

Figure 1.6 Telescopic wing Gevers Genesis [46]
Some of the advantages of the telescopic wing include [46]:

- High lift at low speeds without the drag penalty of flaps.
- Increased aspect ratio at low speeds which improves efficiency - unlike conventional flaps.
- The retracted configuration has much higher stiffness and resistance to bending and twist (i.e. flutter resistant) than a conventional wing.
- The retractable wing is lighter than a conventional wing with the same stall speed, maximum speed, and strength requirements

1.2.2. The Telescopic Blade Wind Turbine concept

The power output of a wind turbine $P_t$ depends upon its efficiency or power coefficient $C_p$, the swept area $A = \pi d^2 / 4$ given by the diameter of the turbine blades $d$, the density of air $\rho$ and the wind speed $v$,

$$P_t = C_p \frac{1}{2} \rho Av^3$$  \hspace{1cm} (1.1)

$$P_t = C_p \frac{\pi}{8} \rho d^2 v^3$$  \hspace{1cm} (1.2)

To improve the power output of a wind turbine, the turbine efficiency, $C_p$, could be enhanced. Betz law limits a wind turbine’s aerodynamic efficiency to no more than 59.3% [50]. After accounting for actual aerodynamic performance and efficiency losses in the gearbox and in the generator, the maximum achievable efficiency is approximately 52% [51]. State of the art wind turbines already have system efficiencies in the order of 50% [51] so there is little room for improvement in the performance from efficiency gains. Another method of improving the power output of a wind turbine is to increase the wind speed $v$ through the rotor. This may be achieved through the use of a diffuser or shroud as investigated by Philips et al [39] and others [40]. However, as discussed earlier, there are major drawbacks such as costs to the implementation of these concepts, rendering them unfeasible.

A wind turbine will also produce more energy at a given site with a larger diameter rotor. However, the size of the rotor, and hence the wind turbine’s energy capture, are limited in order to avoid damaging mechanical loads. Increases in rotor diameter also come at the expense of higher system capital cost. Wind turbine designs are often tailored to specific site characteristics with low wind speed sites having larger rotors and high wind speed sites.
having smaller rotors for the same power rating. The limiting mechanical loads are always developed in high wind conditions at any site and it is in those winds that it would be desirable to have a smaller rotor. Currently, most wind turbines use fixed length blades. Various control strategies such as variable speed and pitch, flexible blades, teetered rotors, feathering and furling have been used to increase energy capture and reduce system load. However, these control strategies are unable to limit loads during storm conditions when the rotor is parked. During a storm, the only technique for limiting loads is to reduce the rotor diameter and/or the tower height. There are no other technically feasible methods of reducing loads in stormy conditions.

A more recent concept [49, 51] to improve the power output and the Annual Energy Production (AEP) of a Wind Energy Conversion System (WECS) involves varying the rotor diameter, $d$, through the use of telescopic or variable length blades. Such a system is one of the future energy conversion technologies that have gained some interest around the globe. The telescopic blade concept offers opportunities to dramatically improve a wind turbine's energy capture and cost effectiveness by controlling the length of the blades and indirectly controlling the swept area. When wind speeds are low, the telescopic turbine blades extend thus increasing the swept area $A$ which captures more energy as compared with a fixed blade length turbine. Alternatively, when the wind speeds are high (above the rated speed) the telescopic blades could retract thus reducing the mechanical loading on the system.

1.2.3. Previous research on Telescopic Blade Wind Turbines

To test the possibility of building a telescopic or variable length blade system, Energy Unlimited Inc retrofitted one of their existing wind turbines with a proof of concept prototype [49, 51] as shown in Figure 1.7.

Field tests conducted by Energy Unlimited Inc [49, 51] have shown that the energy yield of a nominally 120 kW machine fitted with variable length blades increases by 33% at an IEC Class II site with a 20% blade length extension. They estimated that the increase in capital cost of the wind turbine as a result of variable length blades was expected to be below 10% of the total system cost.
In a report prepared by Energy Unlimited [49, 51], the telescopic blade concept is promoted to be a simple yet powerful concept with the following advantages:

1. Improves energy production in low winds.
2. Allows the turbine to continue generation even in higher wind speeds with blade retraction as compared with standard bladed turbines.
3. Improves capacity factor without running turbines beyond rated capacity.
4. Allows areas with low wind speeds that could not support wind energy technology, to become viable for development.
5. Reduces the need for different size blades for different wind regimes.
6. Allows owner to control power output of the wind farm easily and in an efficient way
7. Reduces array losses. Turbines in the rear of an array for any wind direction could easily have larger rotor with blade extension to compensate for the lower wind speeds.
8. Lowers the cost of wind energy by improving annual power output.
10. Allows blades to be self cleaning for dirt and ice.
11. Limits damage in extreme winds as they can be shortened to less than standard length during heavy winds.
12. Compatible with existing technologies of fixed pitch, variable pitch, and variable speed turbines.
13. Allows existing projects to increase their annual production without increasing their power ratings.
14. Can be retrofitted to existing blades, as well as being designed into new blades.
15. Makes the wind industry more competitive.
Following the pioneering work of Energy Unlimited Inc [20, 21], a number of researchers have since investigated the telescopic blade wind turbine system through computer modelling.

Sharma et al [52-54] also investigated the telescopic blade concept using a mathematical model based on the blade element – momentum theory. They showed that the concept would be feasible if the cost of the rotor could be kept less than 4.3 times the cost of a standard rotor with fixed length blades.

The concept of retractable blades has been analysed by McCoy and Griffin [49, 55]. In their investigation, they studied a WindPACT 90 m diameter rotor with a rated power output of 2.5MW with variable pitch and speed control. Retractable blade lengths of 81 - 110 m, being approximately -10% to +20% of the base length were studied. The result of their study showed that the Annual Energy Output (AEO) increased by up to 22.8% for Class 4 winds and 19.4% for Class 6 winds (refer to Table 1.1 in section 1.2.4).

More recently, the writer has analysed a 10kW wind turbine with telescopic blade lengths of 6.3 – 9.4m, being -10% to +33% of base length, for a Wind Class 7 site in New Zealand and predicted an 18% increase in annual energy production over a fixed length blade system [56]. Figure 1.8 shows the extracted power curves for the standard and the telescopic blade wind turbine systems from the study. The power output for the wind generator with telescopic blade is enhanced below rated speed due to the increase in the diameter of the rotor. This leads to increases in the energy capture in Region 2 of Figure 1.8 which is where the turbines can extract most energy.
1.2.4. Wind Classes and the telescopic blade concept

Since our discussions so far have centred around Wind Classes, it deserves further comment. Wind classes range from 1 to 7 with 7 having the fastest winds with an average of 8m/s or more at 10 metres tower height as shown in Table 1.1. A Class 4 site is a low wind speed site with an average wind speed of between 5.6m/s and 6.0m/s at 10m hub height.

Generally, there are more Class 4 wind sites (about 20 times more [6]) than Class 5 and 6 sites. Many Class 5 and 6 sites have already been developed for wind energy production, while some are too inaccessible for development such as in rugged mountain passes, or are too remote to have access to transmission lines. On the other hand, Class 4 sites while being plentiful are also generally nearer load centres, such as near big cities, averaging 100 miles versus 500 miles for Class 6 sites [6].

Figure 1.8 Power curve for wind, standard and telescopic blade systems
Table 1.1 Wind Classes

<table>
<thead>
<tr>
<th>Wind Power Class</th>
<th>Wind Power Density (W/m²) at 10 metres height</th>
<th>Wind Speed (m/s) at 10 metres height</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0-100</td>
<td>0.0 – 4.4</td>
</tr>
<tr>
<td>2</td>
<td>100-150</td>
<td>4.4 – 5.1</td>
</tr>
<tr>
<td>3</td>
<td>150-200</td>
<td>5.1 – 5.6</td>
</tr>
<tr>
<td>4</td>
<td>200-250</td>
<td>5.6 – 6.0</td>
</tr>
<tr>
<td>5</td>
<td>250-300</td>
<td>6.0 – 6.4</td>
</tr>
<tr>
<td>6</td>
<td>300-400</td>
<td>6.4 – 7.0</td>
</tr>
<tr>
<td>7</td>
<td>400-1000</td>
<td>7.0 – 9.0</td>
</tr>
</tbody>
</table>

With this information, the value of the telescopic blade concept can be revealed further when we plot the increase in energy output obtained by different researchers over different wind classes in Figure 1.9. This shows that telescopic blade wind turbine could enhance the annual energy output by more than 15% for any class of wind. However the studies conducted by Pasupulati et al [51] and McCoy et al [55, 57] in Wind Classes 2 and 4 respectively show that the increase in annual energy output (AEO) is more than 20%. Therefore, the telescopic blade wind turbine concept could possibly be most effectively applied to Class 2 - 4 sites.

Figure 1.9 Percent increase in energy output Vs Wind Class
1.2.5. Further discussions on Telescopic Blade Wind Turbines
While the telescopic blade concept has been analyzed to demonstrate its potential, the studies found in the literature appear to be rather limited. Further investigations into possible variations of the concept, as well as analysis for a whole range of wind sites could reveal the full extent of its potential. Three possible configurations of the telescopic blade concept are shown in Figure 1.10. For a site with high probability of low wind speeds (i.e. more low wind speed hours), the concept shown in Figure 1.10 (a) is anticipated to be more appropriate. Additional swept area is required to compensate for the low wind speeds. In the same way, for a site with high prospects of moderate wind speeds (i.e. more medium wind speed hours), the concept shown in Figure 1.10 (b) would likely be appropriate as there would be a lot more times when the wind speeds were greater than the rated wind speed as compared with the low speed regime. Also the system would have additional mechanical loading at higher wind speeds which could be reduced by reducing the swept area i.e. the rotor diameter. Similarly, for a site with a high probability of high wind speeds (i.e. more high wind speed hours), a fully extended blade would probably be used at rated speed as shown in Figure 1.10 (c), and when the wind speeds went higher, the blades would retract, reducing the system mechanical loading while the power production was maintained near its rated level. For any of these systems, when the wind speeds are high, the blades retract thus reducing the thrust and the overall bending moment on the tower. This reduction in load reduces the foundation and tower strength requirements thus reducing the cost of the system. However, a full analysis is necessary to enable a better understanding of the correlation between wind speed characteristics and telescopic blade configurations for best performance.
Figure 1.10 Telescopic wind turbine concepts (a) Low wind speed site, (b) Moderate wind speed site, (c) High wind speed site
Irrespective of the configuration that is ultimately found to be most appropriate for a given wind condition, the telescopic blade concept inevitably introduces a discontinuity between the fixed (root) and movable (tip) sections of each blade.

Figure 1.11 Two dimensional view of telescopic blade concepts

A plan-form view of the telescopic blade previously shown in Figure 1.7 is shown in Figure 1.11 (a) showing such a discontinuity or step change. Figure 1.11 (b) shows a slightly modified situation with a more gradual change. The aerodynamics of such a discontinuity or a step change along the blade appear not to have been investigated, as published literature cannot be found in either the wind energy domain or in the broad fluid and aerodynamics area. There are likely to be additional losses to the blade efficiency similar to the tip and hub losses (Figure 1.12 (a)) which are incorporated in turbine design models. The step change in chord presents an opportunity for the ‘underside’ positive pressure fluid particles to ‘bleed’ around the step to the ‘topside’ suction pressure regions and generate tip-type vortex roll up as shown in Figure 1.12 (b).

In the development of the telescopic blade concept, gaining an understanding of the aerodynamic characteristics of the discontinuity region is important for a number of reasons. Firstly, it will enable quantification of the level of loss of aerodynamic efficiency, if any, and secondly it will aid in the minimization of the loss through modifications to the design of the region. Generation of such information would also be helpful from the modelling point of view. Realistic losses could be incorporated with the modelling process that should then enable improved predictions based around the use of the telescopic blade concept. Only when such effects are accounted for in the modelling process will the predicted turbine characteristics be close to reality. Optimization can then be carried out with much confidence.
Figure 1.12 Typical vortex roll up from a wind turbine blade
1.3. Power curves

1.3.1. Review of power curves for Telescopic Blade Wind Turbines

Figure 1.13 shows power curves from different telescopic blade wind turbine studies found in the published literature. Included also are the power curves from some analyses that were conducted during this research. These demonstrate that the power output of the turbine increases in Region 2 (refer Figure 1.8) which is where most of the harvestable wind speeds lie in any probability density function of wind speeds.

![Power curves for Telescopic Blade Wind Turbine](image)

(a) McCoy et al [55]

Figure 1.13 Power curves for Telescopic Blade Wind Turbine
The literature cited thus far has shown that the telescopic blade wind turbine offers considerable promise, but further research is required before its full benefits are realised.
1.4. Research motivation and a summary of gaps in literature

Very little technical research on the aerodynamics and performance aspects of a Telescopic Blade Wind Turbine system has been found in the literature to date. In fact there are no data available for this type of wind energy system in any controlled conditions. As shown in the literature by the limited research available, the potential benefits of a Telescopic Blade Wind Turbine system are tremendous. Full scale tests carried out by Energy Unlimited [51] showed great benefits of this system. Mc Coy and Griffin [55] carried out a study on a virtual extendable/telescopic blade system for a 2.5MW turbine and showed that the percentage increase in the annual energy for Class 4 and 6 winds (configuration 6) are 21.4 and 17.3% respectively. A 4% reduction in energy output due to losses associated with the step change in blade chord and blade actuation was assumed [55]. It was also shown that the reduction per kWh of energy produced is 3.9 and 0.6% respectively for Class 4 and 6 wind sites. Sharma and Madawala [54] showed that for the telescopic blade concept to be feasible, the cost of the rotor should be kept less than 4.3 times the cost of a standard rotor with fixed length blades, which indicates the economic feasibility of the concept.

The author derived his motivation not only from the demonstrated potential of the concept as discussed above, but also from the “knowledge gaps” identified in the early stages of this research.

The gaps in the knowledge can be summarised as follows:

1. There is no systematic study conducted on the telescopic blade wind turbine system under controlled conditions, to correlate telescopic blade parameters to turbine performance. A single field study [51] has demonstrated the potential of such a system, however a knowledge base remains to be established on how the telescopic blade parameters (extension ratio and chord ratio) affect performance.

2. There is no study found in the literature that provides an understanding and quantification of the aerodynamic performance of a blade with a step or sudden change in its chord. In the numerical study of McCoy [55], a 4% loss in energy output was assumed to arise partly from such a change in chord of the blade. On the other hand, Sharma and co-workers [52-54] ignored the possibility of such losses in their numerical analysis. The writer believes that a study to quantify the aerodynamic performance and losses associated with the step change in blade chord is essential to obtain realistic predictions of turbine performance and energy yield.
3. There is very little information on how the energy yield enhancement of a telescopic blade wind turbine system correlates with wind class, and whether the concept is suited better to certain Wind Classes.

This served as an additional motivation for this research on aerodynamics and performance aspects of a Telescopic Blade Wind Turbine system which would to some extent fill the knowledge gap. With better understanding of this system, more energy could be harvested which in return may make wind energy industry more competitive thus reducing carbon emissions and help save the Planet Earth.

1.5. Objectives
The fundamental aim of this research is to study the Aerodynamics of a Telescopic Blade Wind Turbine system. The worst case scenario of a step or sudden change in blade chord, with constant chord and zero twist, is chosen for the study. This is because of (a) practicality and (b) because if this shows improvements, then good aerodynamic cases will likely be even better. The research goal can be further explained with the help of the following objectives met in this research:

1.5.1. Investigate the aerodynamic characteristics of telescopic blades, in particular the influence of the discontinuity along the blade.
A second set of wind tunnel performance tests to be carried out on a model scale telescopic blade wind turbine. A series of wind tunnel tests to be carried out to investigate the aerodynamic effect of a step change in the chord of a telescopic blade. Surface pressure measurements in the chord and span-wise directions to be made in the wind tunnel to understand and quantify aerodynamics around the step change region. The tests to be carried out for different Reynolds numbers, various angles of attack, different tip blade lengths and different configuration (2D and 3D). Flow visualisation to be carried out to see if there are any vortex roll ups around the step change region.

1.5.2. Develop a model Telescopic Blade Wind Turbine System (TBWTS), and to conduct aerodynamic performance tests in the wind tunnel, for a range of blade configurations
A set of root blades to be designed and manufactured with provisions for the tip blade to slot in the root blade. Two different sets of tip blades with different chords to be made and assembled for performance testing. Torque, thrust and rotor rpm measurements to be carried out using an existing in-house built measuring system. Wind speed and/or rotor speed to be varied in order to identify the optimum tip speed ratio for different blade configurations.
Similar to the tip and hub loss model, a correlation for step loss to be formulated in order to quantify this loss. From the performance testing data, the step loss correlation to be validated and implemented in the wind turbine rotor analysis software and in this case WT Perf [58]. WT Perf is a software which uses BEM theory to calculate the performance of a wind turbine. Once the validated correlation is implemented in WT Perf, power curves to be obtained for energy output analysis.

1.5.3. Analyse different variations of the Telescopic Blade Wind Turbine concept, for a whole range of wind sites, towards developing a guide between each variation and wind site characteristics.

Once the power output curves are available from WT Perf, these are then to be used for energy output analysis. For different Weibull parameters, the wind speeds are distributed differently therefore the energy output is expected to vary. Energy output analysis to be carried out for different blade extensions and correlated with different wind classes. A simple but realistic rotor and generator control strategy to be assumed.

1.6. Scope

From the preceding discussions, it is clear that the issue of losses due to a step change in the chord of a telescopic blade wind turbine system (also referred to as step loss) needs further investigation, with regard to individual blade aerodynamic performance, as well as the turbine performance. This research involves experimental, computational and analytical studies of the aerodynamics of the telescopic blade wind turbine rotor. In light of this situation, the following issues are within the scope of this PhD research:

- Telescopic blade design (constant chord and no twist for both the blade sections) to be designed and constructed for performance testing in the wind tunnel. A set of root blade and two sets of tip blades with different chords to be constructed to analyse the chord ratio effects. For tip extensions, the same set of blades to be utilised by varying the tip blade length.

- Performance tests for the telescopic blade wind turbine with different chord ratios and tip extension lengths to be carried out in the wind tunnel. A correlation for step loss to be formulated and validated using the experimental data and implemented in the wind turbine analysis software and in this case, in WT Perf.

- Aerodynamic analysis of the step change in telescopic blade in the wind tunnel for various Reynolds numbers, angles of attack, different blade configuration (2D and
3D). The effect of the step change in chord from root to tip blade, on the blade pressure coefficient to be explored. These results could give a clearer picture of the flow around the step change region and the effect of different chord Reynolds numbers. Are there any signs of vortex roll ups? Basic flow visualisation to be carried out in the wind tunnel to see if there are any vortex roll ups.

- Once the step loss correlation is developed and validated, modelling of the turbine performance to be carried out using WT Perf software. This could let the wind turbine designer quantify the reduction in power output due to step change thus the reduction in energy output.

- Establish a method to choose appropriate blade configurations using the Weibull parameters. This would enable the wind turbine design engineer to choose appropriate blade configurations for different sites. Different controls strategies are beyond the scope of this research.

Wind turbine design engineers, wind farmers, energy policy makers, potential investors and researchers are expected to benefit from the outcome of this work that aims to place a quantification of step loss at the design stage rather than assuming it.

It is important to point out that while the following issues might be of much value to the overall topic of telescopic blade wind turbine systems; these are however outside the scope of the present study:

- Optimisation of the blade profile
- Blade telescoping mechanism
- Actuation and control strategies for the telescoping mechanism
- Structural analysis
- Cost benefit analysis of telescopic blades, etc.
1.7. Thesis outline
The work carried out in this research as mentioned in the objectives in the preceding section, is sequentially documented in this thesis in several chapters. The research process and its different outcomes are described by organizing them into six chapters including the current one. Each chapter has its own introduction and literature review that introduces the current state of the art in that aspect of the problem before going on to tackle it in detail.

Chapter 2 describes briefly the theory behind the aerodynamics of wind turbines. The fundamental blade momentum, blade element and blade element and momentum theories are described very briefly followed by an overview of blade losses. Standard blade turbine power output calculations together with telescopic blade output are discussed. Literature on wind turbine tip and hub losses are also discussed in detail.

Chapter 3 describes the performance testing of a model-scale telescopic blade wind turbine that was carried out in the wind tunnel. A brief description of the twisted flow wind tunnel is presented with a detailed description of the experimental setup, measuring system and instrumentation, telescopic blade configurations and assembly, and experimental uncertainties. The results of the performance tests carried out are presented in conjunction with a validated step loss correlation. Different blade extension and chord ratio test result are presented in several sub-sections and discussed.

Chapter 4 describes the aerodynamic study carried out on a telescopic blade to establish the effect of step change on blade performance. Telescopic blades with 300 pressure taps installed on the surface of the blade were tested in a wind tunnel. Reynolds number, blade extension and angles of attack were varied, and 2D and 3D configurations were also studied. Benchmarking of the experimental measurements was done with the analytical solutions from XFOil. The effects of step change, 2D versus 3D configuration, Reynolds number and blade extension are discussed in detail.

Chapter 5 describes the energy output analysis of a telescopic blade wind turbine system. A brief introduction on the statistical analysis of wind, together with the well known theory of Weibull distribution is discussed. A reference wind turbine is chosen for energy analysis as a case study in this research. A telescopic blade wind turbine with maximum 40% tip blade extension and different chord ratios are analysed for energy output analysis. A control strategy based on limiting the blade bending moment and Maximum Power Point Tracking (MPPT) strategy is chosen for rotor and generator respectively. The power curves obtained
for various chord ratios and blade extensions are presented. The effect of step change, blade extension, chord ratio and Weibull parameters on the energy output are discussed in detail.

Finally, conclusions and recommendations for future work are presented in Chapter 6.
Chapter 2. Aerodynamics of wind turbine

A wind turbine is a device for extracting kinetic energy from the wind. By removing some of its kinetic energy the wind must slow down but only that mass of air which passes through the rotor disc is affected. Assuming that the affected mass of air remains separate from the air which does not pass through the rotor disc and does not slow down, a boundary surface can be drawn containing the affected air mass and this boundary can be extended upstream as well as downstream forming a long stream-tube of circular cross section. No air flows across the boundary and so the mass flow rate of the air flowing along the stream-tube will be the surfeit for all stream-wise positions along the stream-tube. Because the air within the stream-tube slows down, but does not become compressed, the cross-sectional area of the stream-tube must expand to accommodate the slower moving air (Figure 2.1).

Figure 2.1 The energy extracting stream tube of a wind turbine [59]

Wind turbine power production depends on the interaction between the rotor and the wind. The wind may be considered to be a combination of the mean wind and turbulent fluctuations about that mean flow. The major aspects of wind turbine performance are determined by the aerodynamic forces generated by the mean wind. Periodic aerodynamic forces caused by wind shear, off-axis wind and rotor rotation and randomly fluctuating forces induced by turbulence and dynamic effects are the source of fatigue loads and are a factor in the peak loads experienced by a wind turbine.

Practical horizontal axis wind turbine designs use airfoils to transform the kinetic energy in the wind into useful energy. A number of authors have derived methods for predicting the
steady state performance of wind turbine rotors. The classical analysis of the wind turbine was originally developed by Betz and Glauert [60]. The theory was expanded and adopted for solution by digital computers (Wilson and Lissaman [61], Wilson et al [62]). In all of these methods, momentum theory and blade element theory are combined into a strip theory that enables calculation of the performance characteristics of an annular section of the rotor. The characteristics for the entire rotor are then obtained by integrating, or summing the values obtained for each of the annular sections.

2.1. Momentum theory and blade element theory

A wind turbine rotor consists of airfoils that generate lift by virtue of the pressure difference across the airfoil, producing the same step change in pressure seen in the actuator disc analysis. The analysis here uses momentum theory and blade element theory. Momentum theory refers to a control volume analysis of the forces at the blade based on the conservation of linear and angular momentum. Blade element theory refers to an analysis of forces at a section of the blade, as a function of blade geometry. The results of these approaches can be combined into what is known as strip theory or blade element momentum (BEM) theory. This theory can be used to relate blade shape to the rotors ability to extract power from the wind.

2.1.1. Momentum theory

The forces on a wind turbine blade and flow conditions at the blade can be derived by considering conservation of momentum since force is the rate of change of momentum. Hence, considering an annular section of the rotor of thickness, $dr$ at radial locations $r$, gives the differential thrust $dT$ on the rotor,

$$dT = 4a(1 - a)\rho V^2 \pi r dr$$

(2.1)

Similarly, from the conservation of angular momentum equation, the differential torque, $dQ$ imparted to the blades (and equally, but oppositely, to the air) is:

$$dQ = 4a'(1 - a)\rho V\Omega \pi r^3 dr$$

(2.2)

Thus from momentum theory one gets two equations that define the thrust and torque on an annular section of the rotor as a function of the axial ($a = \left(\frac{U_{\infty} - U_T}{U_{\infty}}\right)$) and angular induction ($a'$) factors (i.e. of the flow conditions - refer Manwell et al [63] for detailed explanation).
2.1.2. **Blade element theory**

The forces on the blades of a wind turbine can be expressed as a function of lift and drag coefficients $C_L$ and $C_D$, and the angle of attack. Wind turbine blades use airfoils to develop mechanical power. The cross sections of wind turbine blades have the shape of airfoils. The width and length of the blade are functions of the desired aerodynamic performance, the maximum desired rotor power, the assumed airfoil properties and strength considerations. Thus, from blade element theory, one also obtains two equations that define the normal force (thrust) and tangential torque producing force as shown in Figure 2.2 on the annular rotor section as a function of the flow angles at the blades and airfoil characteristics.

\[
dF_N = \frac{1}{2} \rho U_{rel}^2 (C_l \cos \varphi + C_d \sin \varphi) c \, dr \tag{2.3}
\]

\[
dQ = \frac{1}{2} \rho U_{rel}^2 (C_l \sin \varphi - C_d \cos \varphi) c \, dr \tag{2.4}
\]

![Figure 2.2 Blade geometry [63]](image-url)
2.2. Blade Element Momentum (BEM) theory

Blade element momentum (BEM) theory is one of the oldest and most commonly used methods for calculating the induced velocities on wind turbine blades. This theory is an extension of actuator disk theory, first proposed in the pioneering propeller work of Rankine and Froude in the late 19\textsuperscript{th} century. The BEM theory, generally attributed to Betz and Glauert [60], actually originates from the two different theories discussed above: blade element theory and momentum theory (see Leishman [64]). Blade element theory assumes that blades can be divided into small elements that act independently of surrounding elements and operate aerodynamically as two-dimensional airfoils whose aerodynamic forces can be calculated based on the local flow conditions. These elemental forces are summed along the span of the blade to calculate the total forces and moments exerted on the turbine. The other half of BEM, the momentum theory, assumes that the loss of pressure or momentum in the rotor plane is caused by the work done by the airflow passing through the rotor plane on the blade elements. Using the momentum theory, one can calculate the induced velocities from the momentum lost in the flow in the axial and tangential directions. These induced velocities affect the inflow in the rotor plane and therefore also affect the forces calculated by blade element theory. This coupling of two theories ties together into what is known as the blade element momentum (BEM) theory and sets up an iterative process to determine the aerodynamic forces and also the induced velocities near the rotor.

In practice, BEM theory is implemented by breaking the blades of a wind turbine into many elements along the span. As these elements rotate in the rotor plane, they trace out annular regions, shown in Figure 2.3, across which the momentum balance takes place. These annular regions are also where the induced velocities from the wake change the local flow velocity at the rotor plane. BEM can also be used to analyze stream tubes through the rotor disk, which can be smaller than the annular regions and provide more computational fidelity.
Because of its simplicity, BEM theory does have its limitations. One primary assumption is that the calculations are static; it is assumed that the airflow field around the airfoil is always in equilibrium and that the passing flow accelerates instantaneously to adjust to the changes in the vorticity in the wake. In practice, it has been shown that the airfoil response takes time to adjust to a changing wake resulting from new inflow or turbine operating conditions (Snel and Schepers [65]). One other limitation is that BEM theory breaks down when the blades experience large deflections out of the rotor plane. Because the theory assumes that momentum is balanced in a plane parallel to the rotor, any deflections of the rotor will lead to errors in the aerodynamic modelling. Another limitation of BEM theory comes from blade element theory. This theory is based on the assumption that the forces acting on the blade element are essentially two-dimensional, meaning that span-wise flow is neglected. This assumption also implies that there is very little span-wise pressure variation (which would create span-wise flow), and the theory is therefore less accurate for heavily loaded rotors with large pressure gradients across the span. Some other limitations of the original theory include the absence of modelling of tip or hub vortex influence on the induced velocities and an inability to account for skewed inflow. However, corrections to the original theory have provided some methods to model these aerodynamic effects and are explained in more detail below. In spite of the limitations listed above, BEM theory has been used widely as a reliable model for calculating the induced velocity and elemental forces on wind turbine blades.
The advantage of the BEM theory is that each blade element is modelled as a two-dimensional airfoil. Figure 2.4 (a) is an example of an airfoil with the velocities and angles that determine the forces on the element and also the induced velocities from the wake influence. Figure 2.4 (b) shows the resultant aerodynamic forces on the element and their components perpendicular and parallel to the rotor plane. These are the forces that dictate the thrust (perpendicular) and torque (parallel) of the rotor, which are the dominant forces for turbine design. In Figure 2.4 (b), the angle relating the lift and drag of the airfoil element to the thrust and torque forces is the local inflow angle. As shown in Figure 2.4 (a), this inflow angle is the sum of the local pitch angle of the blade, $\beta$, and the angle of attack, $\alpha$. The local pitch angle is dependent on the static blade geometry, elastic deflections, and the active or passive blade pitch control system. The angle of attack is a function of the local velocity vector, which is in turn constrained by the incoming local wind speed, rotor speed, blade element velocities and induced velocities. Note in Figure 2.4 that the velocities of the element from blade deflections ($v_{e-op}$ and $v_{e-ip}$) affect the inflow angle and angle of attack, but are not directly affected by the induced velocities from the wake. This assumption is consistent with momentum theory, but it might not be the appropriate physical model for the element-wake coupling. Because the angles of attack are required to be obtained to determine the aerodynamic forces on an element, the inflow angle must first be determined based on the two components of the local velocity vector. Assuming that the blade motion is very small, the resulting equation is dependent on the induced velocities in both the axial and tangential directions as well as the local tip speed ratio ($\lambda_r$):
However, if the blade motion is significant, the local velocities must be included in the calculation of the inflow angle, as follows:

$$\tan \varphi = \frac{U_\infty (1 - a)}{\Omega r(1 - a')} = \frac{1 - a}{\lambda_r(1 - a')}$$

(2.5)

This equation holds for all elements of the blade along the span, although typically the inflow angle changes with element location. The induced velocity components are a function of the forces on the blades and we use BEM theory to calculate them. A thorough derivation of these equations can be found in most wind turbine design handbooks (Manwell et al [63]; Burton et al [59]), and so it is only summarized here. From blade element theory and Figure 2.4 (b), the thrust distributed around an annulus of width $dr$ (see Figure 2.2) is equivalent to

$$dT = B \frac{1}{2} \rho U_{rel}^2 (C_l \cos \varphi + C_d \sin \varphi) cdr$$

(2.7)

and the torque produced by the blade elements in the annulus is equivalent to

$$dQ = B \frac{1}{2} \rho U_{rel}^2 (C_l \sin \varphi - C_d \cos \varphi) crdr$$

(2.8)

Now, to relate the induced velocities in the rotor plane to the elemental forces we must incorporate the momentum part of the theory, which states that the thrust extracted by each rotor annulus is equivalent to

$$dT = 4 \pi a \rho U_\infty^2 (1 - a)rdr$$

(2.9)

and the torque extracted from each annular section is equivalent to

$$dQ = 4U_\infty \pi \rho' (1 - a)a'r^3 dr$$

(2.10)

Thus, when the two-dimensional airfoil tables of lift and drag coefficient are included as a function of the angle of attack, $\alpha$, a set of equations that have to be solved iteratively for the induced velocities and the forces on each blade element. However, before solving the system of equations, several corrections to the BEM theory needs to be accounted. These corrections
include tip- and hub-loss models to account for vortices shed at these locations, the Glauert correction to account for large induced velocities \( a > 0.4 \), and the skewed wake correction to model the effects of incoming flow that is not perpendicular to the rotor plane.

2.2.1. Tip and hub loss model
One of the major limitations of the original blade element momentum theory is that it does not account for the influence of vortices shed from the blade tips into the wake on the induced velocity field. These tip and hub vortices create multiple helical structures in the wake, as seen in Figure 2.5, and they play a major role in the induced velocity distribution at the rotor. The effect on induced velocity in the rotor plane is most pronounced near the tips of the blades, an area that also has the greatest influence on the power produced by the turbine. To compensate for this deficiency in the original BEM theory, a theory originally developed by Prandtl (see Glauert [67] and Vries [62]) may be applied. According to this method, a correction factor, \( F \), is introduced into the thrust and torque equations. This correction factor is a function of the number of blades, the angle of the relative wind and the radial position on the blade.

![Figure 2.5 Idealized vortex system of a two bladed rotor [61]](image)

2.2.1.1. Tip loss
Prandtl simplified the wake of the turbine by modelling the helical vortex wake pattern as vortex sheets that are convected by the mean flow and have no direct effect on the wake itself. This theory is summarised by a correction factor to the induced velocity field, \( F_{\text{tip}} \), which is expressed as:
The tip loss correction factor characterises the reduction in the force along the blade due to the losses at the blade tip.

2.2.1.2. Hub loss

Much like the tip loss model, the hub loss model serves to correct the induced velocity resulting from a vortex being shed near the hub of the rotor. The hub loss model uses a nearly identical implementation of the Prandtl tip loss model to describe the effect of this vortex:

\[
F_{hub} = \frac{2}{\pi} \cos^{-1}e^{-\left(\frac{B/\rho R}{r \sin \phi}\right)}
\]

(2.12)

where \( R_{hub} \) is the distance to the root of the blade as shown in Figure 2.6.

The total correction factor due to hub and tip losses is then given by the product of the terms [66, 68]

\[
F = F_{tip} \times F_{hub}
\]

(2.13)

Figure 2.6 Wind turbine blade

The tip loss correction factor affects the forces derived from momentum theory. Thus:

\[
dT = 4F\pi \rho U^2 (1 - a) r dr
\]

(2.14)

and

\[
dQ = 4FU_\infty \Omega \pi \rho'(1 - a)a' r^3 dr
\]

(2.15)

Because of its reasonable accuracy for most operating conditions and easy formulaic implementation, the Prandtl model is often used in engineering codes. However, like most
engineering models it has limitations that affect its accuracy. One limitation of this model is that it assumes the wake does not expand, limiting its validity to lightly loaded rotors. Also, Glauert [60] showed that the accuracy of this model, relative to the more accurate and computationally expensive Goldstein solution [37], decreases with lower numbers of blades (less than three) and higher tip speed ratios. Previous researchers (e.g. Wilson and Patton [69]) have suggested various other corrections to the BEM theory. These corrections include accounting for the blade thickness effect on local angle of attack, cascade width for high solidity turbines, and span-wise gaps for partial span pitch control. Blade thickness and cascade effects can be aerodynamically significant near the rotor hub and may affect the in-plane yaw forces on the rotor.

2.2.2. Calculation of power coefficient

Once the axial induction factor \( a \) has been obtained for each section, the overall rotor power coefficient may be calculated by integration (Wilson and Lissaman [61]) of the expression for the differential torque and the definition of the local tip speed ratio:

\[
C_p = \left( \frac{8}{\lambda^2} \right) \int_{\lambda_{hub}}^{\lambda} \lambda^3 a' (1 - a) \left[ 1 - \left( \frac{C_d}{C_l} \right) \cot \varphi \right] d\lambda_r
\]

(2.16)

where \( \lambda_r = \frac{\lambda}{R} \) and \( \lambda = \frac{\Omega R}{U} \).

The power coefficient is derived from the power contribution from each annulus. The power contribution of each annulus is:

\[
dP = \Omega dQ
\]

(2.17)

The total power from the rotor is then

\[
P = \int_{r_{hub}}^{R} dP = \int_{r_{hub}}^{R} \Omega dQ
\]

(2.18)

and the power coefficient, \( C_p \), is

\[
C_p = \frac{P_{\text{turbine}}}{P_{\text{wind}}} = \frac{\int_{r_{hub}}^{R} \Omega dQ}{\frac{1}{2} \rho \pi R^2 U^3}
\]

(2.19)
2.2.3. Calculation of power coefficient for a Telescopic Blade Wind Turbine

A telescopic blade wind turbine concept is schematically shown in Figure 2.7

![Figure 2.7 Schematic diagram of a telescopic blade wind turbine](image)

The power coefficient for this is similarly derived from the power contribution from each annulus. The total power from the rotor with telescopic blade is thus

\[
P_{\text{turbine}} = \int_{r_{\text{hub}}}^{R} dP = \int_{r_{\text{hub}}}^{R_R} \Omega dQ + \int_{R_R}^{R} \Omega dQ \tag{2.20}
\]

Therefore the power coefficient of telescopic blade rotor, \( C_P \), is

\[
C_P = \frac{P_{\text{turbine}}}{P_{\text{wind}}} = \frac{\int_{r_{\text{hub}}}^{R_R} \Omega dQ + \int_{R_R}^{R} \Omega dQ}{\frac{1}{2} \rho \pi R^2 U^3} \tag{2.21}
\]

The telescopic blade configuration is expected to have an additional loss due to the step change in blade chord. This sudden change in chord is likely to create a vortex roll up and these vortex roll ups would create further three dimensional effects to the overall blade.
2.3. A review of blade loss corrections for wind turbine blades
To take into account the three dimensional effects at the tip and hub of the blades, Prandtl introduced the concept of tip and hub losses. Prandtl showed that the circulation of a real rotor tends to zero exponentially when approaching the blade tip as shown in an appendix to the dissertation of Betz [70]. The blade element momentum (BEM) theory was later developed by Glauert [71] as a computational tool to predict aerodynamic loading and power of a wind turbine or propeller. In order to make more realistic predictions, Glauert introduced an approximation to Prandtl’s tip loss correction to be included in BEM computations. In his analysis, Glauert assumed that the tip loss only affected the induced velocities but not the mass flux. The BEM theory is based on one dimensional momentum theory in which forces are distributed continuously in the azimuth direction, corresponding to an infinite number of blades with no tip loss. In order to make BEM computations more realistic, Glauert [71] showed how the tip loss effect is integrated in a simple manner into the BEM model. He corrected the induced velocity in the momentum equations by exploiting the fact that, the ratio between the average induced velocity and the induced velocity at the blade position tends to zero by the expression developed by Prandtl.

A refined tip loss model was later introduced by Wilson and Lissaman [61] who suggested that the mass flow through the rotor disc should be corrected in the same manner as the induced velocity in the wake. This, however, leads to a formulation in which the orthogonality of the induced velocity to the relative velocity at the blade element is not satisfied. In order to satisfy the orthogonality condition, de Vries [62] refined the tip correction of Wilson and Lissaman by correcting the mass flux in the tangential momentum equation in the same way as in the axial momentum equation. Apart from BEM computations, advanced axisymmetric actuator disc (AD) models based on solutions to the axisymmetric Navier–Stokes/Euler equations have been introduced [72-75]. In these models the kinematics is described by first principle methods. Corrections for tip loss are still needed but have to be introduced in different ways from those used for BEM models. Recently, Mikkelsen et al. [72] proposed a technique to model the tip loss in which the induced velocities are first computed by the Navier–Stokes solver, after which they are corrected with the Prandtl tip loss function in the rotor plane before applying aerofoil data. The technique, however, is in principle a variant of the Glauert tip loss correction.

Looking more closely into the various models, Shen et al [76] found that they all lack rigorous consistency when the tip of the blade is approached. By analysing the basic
equations of the tip loss correction, they found that the computed axial interference factor always tends to unity when approaching the tip. This implies that the axial velocity, independently of the pitch setting, tip shape, aerofoil type and operating conditions, always tends to zero at the tip. Furthermore, comparisons of BEM computations with experiments show that the Prandtl/Glauert tip loss correction method overestimates the loading close to the tip. As a first attempt to derive a consistent tip loss correction model and overcome the difficulties of the classical modelling of the tip region, a new tip loss correction model was proposed by Shen et al. [76].

2.3.1. Tip loss correction models

The BEM method is the most widely used technique for computing loads and power output of wind turbines. The method is a one-dimensional approach based on the actuator disc principle (i.e. the forces are distributed evenly in annular elements without azimuth dependence). In order to simulate a real wind turbine with a finite number of blades, tip loss effects are introduced into the BEM method by using the Prandtl tip loss function [70],

\[
F = \frac{2}{\pi} \cos^{-1} \left( \frac{B(R-r)\sqrt{1+\lambda^2}}{2R} \right)
\]

(2.22)

where \( \lambda = \frac{\omega R}{V} \) is the tip speed ratio. The Prandtl tip loss function, which takes values in the range from zero at the tip to one at the root section, is derived by assuming that the wake consists of a system of straight vortex sheets. A more realistic and much more complicated model was introduced later by Goldstein [37] who used the inviscid screw surface structure of the wake to compute the circulation along an optimal rotor blade and compared it with the one obtained with the Prandtl tip loss function. For a four-bladed rotor, good agreement between the model of Goldstein and the Prandtl tip loss function is achieved. For a two-bladed rotor the results are in good agreement for tip speed ratios greater than \( \lambda = 7 \), whereas large differences are found at smaller tip speed ratios (e.g. at \( \lambda = 5 \) the difference is about 6%)

An approximate formula for the Prandtl tip loss function was introduced by Glauert [71], namely
Theory

\[ F = \frac{2}{\pi} \cos^{-1} \left( e^{\frac{B(R-r)}{2R \sin \phi_R}} \right) \]  
(2.23)

where \( \phi_R \) is the flow angle at the tip. In order to make the formula easier to use in practical BEM computations, the tip loss function was further changed to

\[ F = \frac{2}{\pi} \cos^{-1} \left( e^{\frac{B(R-r)}{2r \sin \phi}} \right) \]  
(2.24)

where \( \phi = \phi (r) \) is the angle between the local relative velocity and the rotor plane.

There are several ways of making tip loss corrections. Shen et al. [76] summarized three well-known tip loss correction models and analysed the resulting expressions. Since all models are derivations of the model of Glauert, its implications were discussed in detail. It should be noted that the following derivation is quite general and in principle covers all existing tip correction models.

2.3.1.1. The Tip loss correction of Glauert

Glauert [71] corrected only the induced velocity in his model. The final expressions for the interference factors read as follows:

\[ \frac{a}{1-a} = \frac{\sigma C_x}{4 \sin^2 \phi} \]  
(2.25)

\[ \frac{a'}{1+a'} = \frac{\sigma C_y}{4 \cos \phi \sin \phi} \]  
(2.26)

where the solidity \( (\sigma) \) is defined as

\[ \sigma = \frac{B \rho}{2 \pi r} \]

The coefficients \( C_x \) and \( C_y \) are related to the lift and drag coefficients \( (C_l, C_d) \) by,

\[ C_x = C_l \cos \phi + C_d \sin \phi \]  
(2.27)

\[ C_y = C_l \sin \phi - C_d \cos \phi \]  
(2.28)

where \( C_l \) and \( C_d \) depend on the local aerofoil shape and are obtained using tabulated 2D aerofoil data corrected with 3D rotating effects. For detailed derivation of the above equation, please refer to Shen et al [76].
2.3.1.2. The Tip loss correction of Wilson and Lissaman
In the work by Wilson and Lissaman [61] the concept of circulation was employed in order to reformulate the tip loss correction relationship. Since lift is basically generated by circulation, they only considered the lift force in their analysis. The tangential interference factor $a'$ that is obtained is of the same form as that of the Glauert tip loss correction. For the axial induction factor $a$, however, the mass flux is corrected in the same way as for the induced velocity,

$$\frac{a'F}{1 + a'} = \frac{\sigma C_l}{4 \cos \phi}$$  \hspace{1cm} (2.29)

$$\frac{(1 - aF)aF}{(1 - a)^2} = \frac{\sigma C_l \cos \phi}{4 \sin^2 \phi}$$  \hspace{1cm} (2.30)

A similar analysis to the one done for the Glauert tip correction shows that, if the chord at the tip is not zero, the axial interference factor $a$ tends to 1.0 and the flow angle $\phi$ tends to 0 when approaching the tip, whereas the lift coefficient $C_l$ does not tend to 0. Thus a similar inherent inconsistency exists for this model [76).

2.3.1.3. The Tip loss correction of de Vries
In his research De Vries [62] pointed out that the tip loss correction of Wilson and Lissaman [61] does not satisfy the orthogonality between the induced velocity and the relative velocity at the blade element. The induced velocity and mass flux were corrected. Thus,

$$\frac{(1 - aF)aF}{(1 - a)^2} = \frac{\sigma C_x}{4 \sin^2 \phi}$$  \hspace{1cm} (2.31)

$$\frac{a'F(1 - aF)}{(1 + a')(1 - a)} = \frac{\sigma C_l}{4 \cos \phi \sin \phi}$$  \hspace{1cm} (2.32)

In practice, the tip loss correction of de Vries gives almost the same results as those obtained by the original correction of Wilson and Lissaman. Further, it contains the same inherent inconsistency as the other two correction models.
2.3.1.4. Tip loss correction of Shen et al

Similar to de Vries, Shen et al. [76] corrected both the induced velocities and the mass flux for tip loss effects. However, Shen corrected the lift coefficients by introducing the correction factor, $F_1$, which has a similar form to $F$,

$$C_x^r = F_1 C_x$$

(2.33)

$$C_y^r = F_1 C_y$$

(2.34)

where

$$F_1 = \frac{2}{\pi} \cos^{-1} \left( e^{-\frac{g B(r-r)}{2 R \sin \phi}} \right)$$

(2.35)

The function, $g$, generally depends on $\lambda$ and $B$ as well as the chord distribution and pitch settings. However, Shen et al [76] give a simplified function dependent only on the number of blades and the tip speed ratio,

$$g = \exp[-C_1 (B \lambda - C_2)]$$

(2.36)

Experimental data were used by Shen et al [76] to quantify the constants which were 0.125 and 21 for $C_1$ and $C_2$ respectively. In order not to degenerate the formula in the case where $\lambda$ tends to infinity, Shen et al [76] shifted the coefficient $g$ with a small constant of 0·1. Thus the final form of the $g$ function reads as:

$$g = \exp[-0.125(B \lambda - 21)] + 0.1$$

(2.37)

Then the resulting expressions for $a$ and $a'$ become

$$\frac{(1 - aF) aF}{(1 - a)^2} = \frac{\sigma C_x}{4 \sin^2 \phi} F_1$$

(2.38)

$$\frac{a' F (1 - aF)}{(1 + a')(1 - a)} = \frac{\sigma C_1}{4 \cos \phi \sin \phi} F_1$$

(2.39)
2.3.1.5. The Tip loss correction of Xu and Shankar

Similar to Prandtl model, Xu and Sankar [77] extended Prandtl tip loss model and developed two empirical relationships for the attached and stalled flow conditions for the tip loss based on the Navier-Stokes solutions. The empirical relationships are described in the following equations:

**Attached Flow:**

\[
F_{\text{new}} = \frac{\left( F_{\text{prandtl}}^{0.85} + 0.5 \right)}{2}, \quad \text{for } 0.7 \leq \frac{r}{R} \leq 1
\]

(2.40)

\[
F_{\text{new}} = 1 - \left( \frac{r}{R} \right) \left( \frac{1 - F_{r=0.7}}{0.7} \right), \quad \text{for } \frac{r}{R} < 0.7
\]

(2.41)

**Stalled Flow:**

\[
F_{\text{new}} = 0.8, \quad \text{for } 0.8 \leq r/R \leq 1
\]

(2.42)

\[
F_{\text{new}} = 1, \quad \text{for } r/R < 0.8
\]

(2.43)

The discrepancy between Prandtl’s tip loss model and the CFD simulations executed by Xu and Sankar [77] is due to tip vortex stretching effects. These relationships are a correction for the Prandtl model and must be used in conjunction with Prandtl loss model. However, Xu and Sankar [77] also pointed out that this correction was based on a specific turbine design (UAE Phase 6, Hand et al. [78]) at one wind speed and may not be applicable to all turbine configurations. It also results in a tip-loss factor greater than zero at the tip, which is physically unrealistic at the tip blade station.

Several authors have carried out investigations to improve the tip loss corrections factors based on initial work carried out by Glauert [60]. For the telescopic blade system, the loss generated due to sudden change in chord (step loss) could be developed based on similar principles of work carried out by the authors above. This will however have to rely on experimental data from performance testing of a telescopic blade wind turbine system and perhaps also from the aerodynamic analysis of a step change blade. Performance testing is considered next.
Theory
Chapter 3. Performance testing of a Telescopic Blade Wind Turbine

The performance characteristics of a model-scale two-stage Telescopic Blade Horizontal Axis Wind Turbine (TBHAWT) was investigated during the course of this research. Experimental tests were carried out in the ‘Twisted Flow Wind Tunnel’ facility at The University of Auckland (Figure 3.1) with the vanes removed. A uniform free stream is produced in the semi-open-jet flow, which has a floor and roof but no side-walls. The flow streams were fairly uniform across the wind tunnel test section with turbulent intensities of less than 2%.

3.1. Blade configuration and test procedure
The two-stage blades used in the experiments are made from two blade sections: a section connected to the hub (hereafter referred to as the root blade) and a movable section that can move relative to the root blade (hereafter referred to as the tip blade), as shown in Figure 3.2. The tip blade is slotted into the root blade and is held using the grub screws as shown in Figure 3.2. In this study, the torque, thrust and rotor speed were measured for 4 different tip blade lengths ($L_T$ – different extensions) being equivalent to 0, ½, 1 and 2 root blade chords ($C_R$) i.e. ~0, 10, 20 and 40% of the root blade lengths. Chord ratios ($C_T/C_R$) of 0.6 and 0.4 (tip blade chord/root blade chord) were chosen for experimental tests. These chord ratios were chosen because of structural strength reasons for the experimental prototype. Furthermore the full scale typical chord ratios are in this range as found in the literature [57]. The symmetrical NACA0018 airfoil, with a constant chord of 70 mm was chosen for the root blade; while constant chords of 42mm and 28mm were chosen, giving chord ratios of 0.6 and 0.4 respectively, for tip blades as shown in Figure 3.3. The root blade length ($L_R$) was fixed at
340 mm and the tip blade length ($L_T$) was varied to give different extensions, with the maximum being a 40% extension based on the root blade length giving a tip blade length of approximately 140 mm. Therefore the maximum overall blade length tested was 480 mm.

Figure 3.2 Telescopic blade

Figure 3.3 Telescopic blade with different chord ratios
Both blades (root and tip) had no twist, so that they were simple and practical to fabricate. The aluminium rotor blades were CNC machined to the required shape, assembled and finally attached to the hub as shown in Figure 3.5. Different blade extensions, i.e. different tip blade lengths were achieved by extending or retracting the tip blade.

![Figure 3.4 Telescopic blades assembled on hub](image)

The blade attachment to the turbine hub had provision to change the pitch angle. The hub was attached to the test rig (Figure 3.5) enabling the measurement of torque, thrust and rotor rpm to be obtained. The test rig had a high specification motor attached to convert the turbine shaft power into electrical energy which was dumped through resistors that were air cooled.

![Figure 3.5 Photographs of the experimental setup with telescopic blades viewed from the front](image)

Torque and thrust measurements were carried out by in-house built torque and thrust transducers. Strain gauges attached to metal strips were used to measure by deflection, the normal and tangential forces exerted by the rotor. The strain gauge readings were amplified
through a strain gauge amplifier and then fed to the A/D card to the Lab view program specifically built for this. The torque and thrust measuring devices were constantly subjected to calibration for accurate data output. Experimental uncertainties associated with strain gauges were quantified and are discussed in the experimental uncertainty section.

**Figure 3.6 Photograph of torque, thrust and speed measuring device setup**

Variations in power coefficient ($C_p$) and thrust coefficient ($C_t$) versus tip speed ratio ($\lambda$) were studied. These parameters are generally defined by

$$\lambda = \frac{\omega R}{V} \quad C_p = \frac{T \omega}{0.5 \rho AV^3} \quad C_t = \frac{F_{\text{thrust}}}{0.5 \rho AV^2} \quad (3.1)$$

The values for experimental parameters were chosen to obtain a range of tip speed ratios corresponding to those obtained with full scale wind turbines (i.e. $\lambda = 3-7$). To vary the rotor speed, the motor was subjected to different loads. These loads were changed by connecting across different resistors in the load bank, as shown in Figure 3.7.

**Figure 3.7 Electrical load bank**
The wind turbine shaft torque \( T \) was measured for different blade lengths. The rotor revolutions were measured using a photo-sensor-type tachometer. Rotor speeds of 0 – 1100 rpm were chosen to get the desired tip speed ratios for different wind speeds. Wind tunnel speeds were varied from 0 to 8 m/s.

### 3.2. Experimental uncertainties

Two types of uncertainty can occur in experimental measurements [79]. One is stochastic uncertainty due to dispersion of the data, which can be evaluated by repeating the experimental measurements, and the other is systematic uncertainty i.e. instrument inaccuracies. The systematic uncertainties cannot be evaluated by statistical analysis. To minimize uncertainty, the measurements were repeated from three to six times for each configuration and an example of the level of scatter in the data is shown in Figure 3.8. Experimental errors arise from errors in the torque, thrust, revolution counter and the velocity measurement. The dominant experimental errors were identified to be in the measurement of \( C_P \), \( C_t \), and the tip speed ratio as defined in equation 3.1. The methodology used to calculate the systematic errors is based on that presented by Coleman and Steels [80] and ITTC [81, 82]. The uncertainties of a variable \( X \) are categorised as being systematic \( (b_X) \), in that they are inherent in all tests, or random \( (s_X) \), and varying between each test. The total uncertainty of variable \( X \) is given as

\[
    u_X^2 = s_X^2 + \sum_{k=1}^{M} b_k^2
\]

where \( k \) is the number of systematic sources of uncertainty.

The sources of systematic errors inherent in the measurements of the power and thrust coefficient of the rotor blade are considered to be composed of three dominant uncorrelated elemental errors (torque, rotational speed and wind speed measurements) and (thrust force and wind speed measurements) respectively. The torque measurement calibrations (strain gauge) were carried out using M1 standard weights and lever arm. Similarly, for the thrust coefficient, thrust force measurement calibrations were again carried out using M1 weights and the wind speed measurement calibration were carried out using the Cobra probe[83]. The rotor rotational speed photo-sensor tachometer was calibrated based on the methodology described by ITTC [84] where the data reduction was carried out based on standard error estimate (SEE).
In usual conditions, the torque and thrust, revolution meter and the velocity anemometer measurement introduced a 2%, 2% and 3% error respectively. In particular, the free stream measurement error of 3% associated with the anemometer is expected to contribute a 9% error in energy and a 6% error in torque and thrust. The overall systematic error was estimated using the Taylor series method for propagation and by application of the quadrature rule for product and quotients, the uncertainties in these values were calculated. Therefore the total maximum experimental errors are approximated to be 13% in $C_p$ and 10% each in $C_q$ and $C_t$.

The uncertainty ranges from 3% to 13% and for each experimental configuration, the graphs show the average measurements. The measuring instruments were subjected to regular calibrations in order to minimize systematic uncertainties.

![Figure 3.8 Variation in $C_p$ as a function of tip speed ratio for several test runs](image)

**Figure 3.8 Variation in $C_p$ as a function of tip speed ratio for several test runs**

**3.3. Rotor analysis using the Blade Element Momentum (BEM) method**

Because of its simplicity, BEM theory has been the mainstay of the wind industry for predicting rotor performance. For this research, rotor performance predictions were obtained by using a recent version of BEM theory in the software suite, WT Perf [85]. One of the major limitations of the original BEM theory is that it does not account for the influence of vortices shed from the blade tips into the wake on the induced velocity field. These tip vortices create multiple helical structures in the wake. The effect of the wake on the induced velocity in the rotor plane is most pronounced near the tips of the blades, an area that also has the greatest influence on the power produced by the turbine.
3.3.1. Standard blade
To compensate for this deficiency in the original BEM theory, a theory originally developed by Prandtl (see Glauert [67] and Vries [62]) may be applied, and is incorporated in WT Perf. According to this method, a correction factor, $F$, is introduced into the thrust and torque equations. As discussed already in Chapter 2, this correction factor is a function of the number of blades, the angle of the relative wind and the radial position on the blade.

The total correction factor due to hub and tip losses is given by the product of the equations (2.11) and (2.12) as shown in equation (2.13).

3.3.2. Telescopic blade
With the step-change in blade section occurring between the root and the tip blade sections of a telescopic blade, it is expected that losses similar to those at the hub and tip of a conventional blade would arise. In order to be able to predict the performance of a TBHAWT, it is thus prudent to develop appropriate correlations to account for such losses arising from the step change in the chord for the telescoping blade. In this thesis, the quantification of the overall losses are taken to be similar to the standard blade loss terms, but with an additional loss hereafter defined as the step loss ($F_{step}$), which is created by the introduction of a tip-blade that has a different chord length. With reference to Figure 3.9, the total loss at some radius $r$ in the telescopic blade then becomes

$$F_{Telescopic} = F_{tip} \times F_{hub} \times F_{Step}$$

(3.3)

where $F_{tip}$ and $F_{hub}$ are given by Eq’s (2.11) and (2.12) respectively.

![Figure 3.9 Losses in a telescopic blade](image-url)
In the present study, $F_{\text{step}}$ is developed by assuming two losses around the step, a tip-type loss applied to the root blade, and a hub-type loss applied to the tip blade as shown in Figure 3.10.

The concept of a constant $G$ is introduced into Eq’s (3.5) and (3.6) based on a proposed new tip loss correction model of Shen et al [76]. Thus Eq’s (3.5) and (3.6) are modified by introducing a coefficient $G$, which needs to be determined for the case of step loss.

$$F_{\text{Step}} = F_{\text{Step-tip}} \ast F_{\text{Step-hub}} \quad (3.4)$$

$$F_{\text{Step-tip}} = \frac{2}{\pi} \cos^{-1}\left[ G \ast e^{-\left(\frac{B}{\pi} \frac{(R_R - r)}{r \sin \phi}\right)} \right] \quad \text{for } R_{\text{hub}} \leq r \leq R_R \quad (3.5)$$

$$F_{\text{Step-hub}} = \frac{2}{\pi} \cos^{-1}\left[ G \ast e^{-\left(\frac{B}{\pi} \frac{r - R_R}{R_R \sin \phi}\right)} \right] \quad \text{for } R_R \leq r \leq R \quad (3.6)$$

It is expected that the step loss, and therefore the coefficient $G$ would depend on the number of blades ($N$), the tip speed ratio ($\lambda$), chord distribution ($c(r)$), chord ratio ($CR$), extension ratio ($ER$) and pitch settings. However, for simplicity, the function was set to be dependent on the chord ratio, extension ratio, and tip speed ratio, and the following form was developed in this research:

$$G = X_1 \exp^{X_2 \ast CR} \left( \frac{\lambda}{Z_1} \right)^{CR \ast (H \ast ER)} \quad (3.7)$$

$$H = Y_1 \ast ER^{Y_2} \quad (3.8)$$
Experimental Investigation

where

\[ ER = \frac{R - R_R}{R_R} \]  
(3.9)

\[ CR = \frac{c_1}{c_2} \]  
(3.10)

and \( X_1, X_2, Y_1, Y_2 \) and \( Z_i \), are coefficients that has to be determined from experimental data. It was found that a good fit to the experimental data was obtained (as shown later) when \( G \) and \( H \) were defined with the values of the constants shown in equations (3.11) and (3.12). Regression analyses to the experimental data were carried out to come up with these values for the range of tests conducted. The experimental data is then compared with the data predicted with the step loss incorporated in the WT Perf code.

\[ G = 0.084e^{3.7185 \cdot CR} \left( \frac{\lambda}{1.25} \right)^{[CR + (H \cdot ER)]} \]  
(3.11)

\[ for \ 0.4 \leq CR \leq 1 \ and \ 0 < ER \leq 0.4 \]

and

\[ H = 13.102 \cdot ER^{1.5814} \]  
(3.12)

\[ for \ 0 < ER \leq 0.4 \]

The step loss, as shown later, is strongly dependent on the size of the step change represented by the chord ratio \( (CR) \). It is therefore useful to consider the two limiting cases for the chord ratio, namely \( CR = 0 \) and \( CR = 1 \) and then when \( CR \) is between these limits.

Case 1: if \( \frac{c_1}{c_2} = 1 \), a blade with no step change, so no step losses, thus \( F_{\text{Step}} = 1 \), and therefore the overall blade correction factor is given by equation (2.13);

Case 2: if \( \frac{c_1}{c_2} = 0 \), there is no tip blade, thus no step change, hence no step losses, thus \( F_{\text{Step}} = 1 \), and therefore the overall blade correction factor is given by equation (2.13)

Case 3: if \( 0 < \frac{c_1}{c_2} < 1 \), a step change between the root and tip blades, and thus the overall blade loss is given by equation (3.3)
These empirical relationships for the loss due to the step change were implemented in WT Perf in the course of this research.
3.4. Results and discussion

3.4.1. Experimental setup validation and benchmarking
The validation of the experimental setup was carried out by testing the blades without any extension. As illustrated in Figure 3.11, the predicted and experimental data for the blade with zero extension (refer Figure 3.5(a)) are in good agreement within the uncertainty range, which serves to validate WT Perf simulations which were carried out with the Prandtl tip and hub losses. These tests ensured that the setup and the data measurement and acquisition system are valid.

![Figure 3.11 Power and thrust coefficients from experiment and predictions as a function of tip speed ratio for the wind turbine without any blade extension](image)

3.4.2. Effect of blade extension
The variations in the power and thrust coefficients with respect to the tip speed ratio for a $CR = 0.6$ and extension ratios 0, 10, 20 and 40% studied in the present work are shown in Figure 3.12. Performance characteristics predicted using the computational procedure based on WT Perf with $F_{step}$ implemented as described above, are also plotted for comparison purposes. The results in Figure 3.12(a) for a 10% blade extension show that the experimental data are in good agreement with the WT Perf simulations without $F_{step}$ applied. For a 20% extension however, the experimental data shown in Figure 3.12(b) are different and higher than the standard Prandtl tip and hub loss model prediction of WT Perf. Not surprisingly, WT Perf under-predicts the losses, thereby over-predicting the power and thrust coefficients. It is important to note that WT Perf has not been developed to model the additional losses which
are deemed present in a telescopic blade system. The incorporation of step loss corrections in WT Perf is one of the subjects of the present study. Similar to that of 10% extension, Figure 3.12(c) for a 40% extension shows that both prediction models (with and without step loss corrections) give similar results, and are in good agreement with experimental data, which indicates that the effect of the step loss on the overall blade power coefficient is small in this case.

The zero (Figure 3.11) and the 10% (Figure 3.12a) extension blades have the highest maximum power coefficients of approximately 0.32 and 0.30 respectively, at tip speed ratios of approximately 4. There is therefore a 6% reduction in the blade performance (power coefficient) with a 10% extension when compared to a zero extension blade, which suggests that a 10% extension does produce some additional losses at the step-change region. It might be speculated that in this case, since the step is quite close to the tip, that the tip region of flow more or less interacts or overlaps with the flow in the step change region and thus produces an additional loss. The maximum thrust coefficients for zero and 10% extensions are 0.6 and 0.57 respectively. The reduction in the thrust coefficient of the 10% extended blade may be viewed as the effect of a higher aspect ratio when compared to the zero extension blades. Alternatively, this could be viewed as an influence of the reduced solidity of the extended blade system.

On the other hand, for blades with a 20% extension, the maximum power coefficient reduces significantly by approximately 25% to 0.24, and the maximum thrust coefficient by 22% to 0.47, at a tip speed ratio of ~ 4. Similarly, for a 40% extended blade, the power coefficient reduces by approximately 16% to 0.27 and the maximum thrust coefficient by 23% to 0.46. It should be noted that these comparisons are made to the zero extension blades and the power coefficient is calculated using the overall swept area or in other words with the actual extended blade length if the blades are extended.
Figure 3.12 Power and thrust coefficients from experiment and predictions as a function of tip speed ratio for several blade tip extensions for a chord ratio of 0.6.
Table 3.1 Maximum power and thrust coefficient with solidity for different blade configuration

(i) Maximum power coefficient

<table>
<thead>
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<th></th>
<th>WT Perf: CR 1.0</th>
<th>Experiment: CR 0.6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>70 mm chord</td>
<td>42 mm chord</td>
</tr>
<tr>
<td>No extension</td>
<td>0.32</td>
<td>0.26</td>
</tr>
<tr>
<td>10% extension</td>
<td>0.30</td>
<td>0.25</td>
</tr>
<tr>
<td>20% extension</td>
<td>0.29</td>
<td>0.22</td>
</tr>
<tr>
<td>40% extension</td>
<td>0.28</td>
<td>0.21</td>
</tr>
</tbody>
</table>

(ii) Maximum thrust coefficient

<table>
<thead>
<tr>
<th></th>
<th>WT Perf: CR 1.0</th>
<th>Experiment: CR 0.6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>70 mm chord</td>
<td>42 mm chord</td>
</tr>
<tr>
<td>No extension</td>
<td>0.60</td>
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<td>0.37</td>
</tr>
<tr>
<td>40% extension</td>
<td>0.50</td>
<td>0.36</td>
</tr>
</tbody>
</table>

(iii) Solidity

<table>
<thead>
<tr>
<th></th>
<th>CR 1.0 (70mm)</th>
<th>CR 1.0 (42mm)</th>
<th>CR 0.6</th>
<th>CR 0.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>No extension</td>
<td>0.16</td>
<td>0.10</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>10% extension</td>
<td>0.15</td>
<td>0.09</td>
<td>0.14</td>
<td>0.14</td>
</tr>
<tr>
<td>20% extension</td>
<td>0.14</td>
<td>0.08</td>
<td>0.13</td>
<td>0.12</td>
</tr>
<tr>
<td>40% extension</td>
<td>0.12</td>
<td>0.07</td>
<td>0.11</td>
<td>0.10</td>
</tr>
</tbody>
</table>
Figure 3.12 (a) to (c) also compare power coefficients for hypothetically extended blades without any step change, that is with chord ratio $CR = 1$, for two different chords equivalent to the chords of the root and tip blades of 70 and 42 mm respectively. The blade configurations for these comparisons are shown in Figure 3.13. For all blade extensions, as shown in Figure 3.12(a) to (c) and Table 3.1(i), the blade with $CR = 1.0$, and 70 mm chord, having the highest solidity ratio as shown in Table 3.1 (iii), has the highest maximum power coefficient, and the blade with $CR=1.0$ and 42 mm chord, having the lowest solidity ratio has the lowest maximum power coefficient, as anticipated. For the blade with $CR = 0.6$ having an intermediate solidity ratio, the maximum power coefficient is in-between that of $CR = 1.0$ with a 70 mm chord and $CR = 1.0$ with a 42 mm chord. Similar trends are evident in the thrust coefficients of Figure 3.12 (a) to (c) and Table 3.1(ii). At 40% extension, a turbine with a $CR = 1.0$ and 42 mm chord blades has approximately 25% reduction in maximum power coefficient when compared to a turbine with a $CR = 1.0$ and 70 mm chord blades. At the same 40% extension, the turbine with step change blades with $CR = 0.6$ has a 4% reduction in maximum power coefficient. Thus the blade with $CR = 0.6$ performs better than that of a $CR = 1.0$ with 42 mm chord blade. The corresponding thrust coefficients are reduced by 28 and 8% respectively.

Figure 3.14(a) shows the correction factor based on equation (3.4), as a function of the radial position for various tip extensions. It can be seen that for the blades tested, the lowest step change correction factors are 0.46 and 0.84 for 20 and 40% extensions respectively occurring at the step location. The 10% extended blade correction factor is superimposed on top of the
zero extension correction factor. As speculated earlier, it is likely that the wake generated due to the step change on the 10% extended blade interacts and overlaps with the extent of tip vortices, thus the tip loss dominates the losses. To get an insight into the vortices generated by the tip and step region of a telescopic blade, a study titled ‘Flow around a step change wind turbine’ [86] was carried out using computational fluid dynamics (CFD) modelling alongside the present study which provides some insight. The same aerofoil (NACA0018) profile was used for the CFD simulation in order to maintain similarity with the blades in the turbine used for performance testing purposes. The vorticity maps obtained from CFD in Figure 3.15 show that there are vortices rolling up from the step change area, much like those from the tip. The step vortices and the tip vortices overlap each other for the tip blade length equivalent to more or less half of the root blade chord. This is equivalent to the 10% extension blade performance testing case. For this reason, the step loss prediction in Figure 3.12 (a) demonstrates that the $C_p$ values using tip and hub losses agree well with the experimental data. A full CFD analysis was beyond the scope of the present study.

On the other hand, for the blade with a tip blade length equivalent to the root blade chord, corresponding to 20% extension in the performance tests, the vortices from the step change and the tip are relatively independent and separate from each other, and so have individual impacts on the rotor performance. In this case, the additional step-loss to some extent nullifies the lift contribution from the additional tip blade length. Afjhe et al [42] gave a similar explanation for tip-controlled horizontal axis wind turbine performance.

The maximum power coefficient for a 40% extended blade is 11% better than that of a 20% extended blade. It is implicit from Figure 3.14(a) that the step loss is significant in the 20% configuration (correction factors are lower) when compared to that of a 40% extended blade as shown in Figure 3.14 (b). The lowest correction factors only due to step change are 0.56, and 0.85 for 20 and 40% extensions as shown in Figure 3.14(b) and Figure 3.14(c) respectively. On the other hand, the overall blade area for the 40% extension is higher than that for 20% extension; therefore the lift force contribution of the tip blade is higher. Another reason for the better performance of the 40% extended blade would be the aspect ratio of the tip blade ($AR = 3.33$) which is higher than that of 20% extended tip blade ($AR = 1.67$), and therefore would have a better lift to drag ratio; in other words, the region of influence of the step is a smaller proportion of the total blade length in the case of 40% extension.
Experimental Investigation

Figure 3.14 Blade correction factor as function of radius for various extensions (Fixed pitch variable speed)

(a) Zero, 10, 20 and 40% extension

(b) 20% extension
(c) 40% extension

Figure 3.14 Blade correction factor as function of radius for various extensions (Fixed pitch variable speed) – cont’d
Figure 3.15 CFD visualization of step vortices at different blade extensions [86] Q-criterion isosurface

Figure 3.16 shows the power and torque coefficient distributions along the blade computed using WT Perf and tip, hub and step loss corrections, for different blade configurations. These are plotted against non-dimensionalised radial position \((r/R)\) where \(R\) is the rotor radius. Figure 3.16 (a) demonstrates the manner in which the step loss correction, that is developed in the research, works. There is a drop in power and torque coefficient along the blade for all blade extensions considered here, when compared to the blade without extension. The blade with 20\% extension demonstrated the lowest power and torque coefficient at the step change region. Figure 3.16(b) demonstrates the power and torque coefficient of a 20\% extended blade with different chords and chord ratios. The step change region has a detrimental effect on the power and torque coefficient as shown in experimental investigations. For the blade with 40\% extension, (see Figure 3.16 (c)) there is a sudden drop
in power and torque coefficients at and around the step change region. This is because the step change in a 40% extended blade is in close vicinity of the peak power generation region of the blade. Any small change in the flow around this region would have a large impact on the blade power and torque coefficients. On the other hand, for blade with 20% extension, as demonstrated in Figure 3.16 (b), a remarkable drop in power and torque coefficients using the proposed step loss model is obtained whereas the the tip and hub loss model show a very small decrease in the power and torque coefficients.

(a) Zero, 10, 20 and 40% Extension

Figure 3.16 Blade correction factor as function of radius for various extensions (Fixed pitch variable speed)
Figure 3.16 Blade correction factor as function of radius for various extensions (Fixed pitch variable speed) – cont’d (for CR =1.0 and 0.6)

Since the rotor radius is normalized, the area under each curve in Figure 3.14 represents an effective or average correction factor ($F_{ave}$) for the entire blade. For the blade with zero
extension, the correction factor is a maximum (or losses are minimum) when compared to a blade that has an extension.

![Blade loss factor (tip, hub and step loss) as a function of blade extension](image)

**Figure 3.17 Blade loss factor (tip, hub and step loss) as a function of blade extension**

Figure 3.17 shows a plot of \(1 - F_{ave}\) against tip blade extension at maximum power coefficient for various blade extensions at a fixed pitch. \(F_{ave}\) may be viewed as a turbine performance factor, as it is computed from the areas under the curves plotted in Figure 3.14. \(F_{ave} = 1\) or \((1 - F_{ave}) = 0\) means no losses at all. Hence plotting \((1 - F_{ave})\) against tip extension indicates the level of loss for each turbine blade configuration. Figure 3.17 shows that the blade with zero extension has the minimum loss \((1 - F_{ave})\). The blade with the 20% extension, includes a step loss, and has higher \((1 - F_{ave})\) than those for the 10% and 40% extended blades. It is also evident from Figure 3.17 that there exists a maxima in terms of the loss at 20% extension, which is the extension equivalent to one root blade chord.

It is reasonable to assume that any sudden change in the chord of the blade of a wind turbine causes the circulation around it to drop. As a result of the change in circulation, vorticity is shed into the wake. The step change in chord of a telescopic blade therefore must have an effect on the blade performance.

Figure 3.18 shows plots of power against wind speed. It is important to note that the wind turbine was tested at a range of wind speeds while maintaining an optimum tip speed ratio. The predicted and the experimental data are in good agreement. The power output with a zero blade extension is the lowest, as expected, and it is highest for the fully extended blade (40%
extension). At approximately 7m/s, the power output of the 40% extended blade is approximately 74% higher than that of the no-extension blade, despite having a lower power coefficient. For blades with 10% and 20% extension, the increase in power output at optimum power coefficients when compared with the no-extension blades are 18% and 13% respectively.

Despite the step change having a detrimental effect on the local power coefficient for a telescopic blade, the power output of a telescopic blade between the cut-in and cut-out velocities increases significantly because of the increase in swept area. Any flow improvement around the step change region would result in a larger power coefficient, thus better turbine power output and higher capacity factor.

![Figure 3.18 Power output of a telescopic blade wind turbine at optimum TSR and comparing the experiments with modified WT Perf predictions](image)

**Summary**

The performance of the telescopic blade concept has been analysed experimentally and computationally. It has been established that the effect of a step change in blade section is significant for the range of blade extensions studied (0 to 40%). It is recognised that the current simulation model in WT Perf incorporating Prandtl tip and hub loss models, was not developed to include step losses, and therefore over-predicts the rotor performance when there is a step change along a blade. Correlations have been developed to quantify losses arising from the step change in the chord of a telescopic blade based on Prandtl tip and hub loss concepts. Results obtained (predictions) utilizing the new correlations were found to be in good agreement with the experimental data. It was also found that the maximum step loss
occurs at a blade extension of 20%, which is equivalent to one root blade chord. The power output of a telescopic blade wind turbine increased for all blade extensions considered in this study, in spite of the detrimental effects of a step change in blade chord. As shown in this research, the effect of step change in the blade chord of a telescopic blade system has significant impact for lower tip blade extensions. For tip blade lengths of 2 times the root blade tip chord and beyond, it is assumed that the effect of step loss would reduce as shown in this research for the case of 40% extension.

3.4.3. Effect of chord ratio

The experimental data (power and thrust coefficients) for the two different chord ratios, $CR = 0.4$ and $CR = 0.6$, are presented in Figure 3.19 and Figure 3.21 respectively. Figure 3.19 (a) shows the data for a 10% blade extension. As shown and discussed already, the WT Perf simulations agree with the experimental data. For the blade with a 20% extension, as shown in Figure 3.19(b) the predictions using the proposed new step loss model described above agree quite well with the experimental data. However, including only the Prandtl tip and hub loss model in WT Perf under-predicts the loss thereby over-predicts the power and thrust coefficients. Likewise, the prediction for the 40% extended blade as shown in Figure 3.19(c) is only slightly different for the two prediction models.

As noted previously, the blade with zero extension (Figure 3.11) has the highest maximum power coefficients of approximately 0.32 at tip speed ratios of ~4 whereas the maximum power coefficient with 10% extension (Figure 3.19 (a)) are 0.30 and 0.29 respectively for $CR = 0.6$ and $CR = 0.4$. There is ~6% reduction in the blade performance (power coefficient) with 10% extension for the chord ratio of 0.6 when compared with a zero extension blade and ~9% for $CR = 0.4$. As also shown in Table 3.2, there is thus a further reduction in power coefficient of 3.3% for a 10% extended blade with chord ratio 0.4 when compared with a blade of chord ratio 0.6.

For blades with a 20% extension as shown in Figure 3.19 (b), the power coefficient reduces significantly by approximately 25% to 0.24 for $CR = 0.6$ and by 31% to 0.22 for $CR = 0.4$ at a tip speed ratio of ~4 and 4.2 respectively when compared with a zero extension blade. Similarly, for a 40% extended blade, when compared with a zero extension blade, the power coefficient reduces by approximately 16% to 0.27 and 25% to 0.24 for $CR = 0.6$ and 0.4 respectively as shown in Figure 3.19 (c). It is clear from Figure 3.20 that the blade extension has a much more significant effect on the power coefficient, than the chord ratio, as seen
already for $CR = 0.6$, the reduction in power coefficient for $CR = 0.4$ increases with the increase in blade extension up to 20%, and then starts to reduce beyond a 20% extension. For blade extensions of 10%, 20% and 40%, the difference in the reduction of the power coefficient between the two chord ratios studied with reference to zero extension blades are 3%, 6% and 9% respectively.

The difference in reduction in power coefficient here was calculated as follows:

$$Reduction \ in \ Power \ Coefficient = \frac{Power \ Coefficient_{CR \ 0.6} - Power \ Coefficient_{CR \ 0.4}}{Power \ Coefficient_{No \ extension}}.$$  

But the reduction in power coefficient of a blade with $CR = 0.4$, relative to the blade with $CR = 0.6$ are 3, 8 and 11% (Table 3.2) respectively, for 10, 20 and 40% extensions; these were calculated as follows:

$$Reduction \ in \ Power \ Coefficient = \frac{Power \ Coefficient_{CR \ 0.6} - Power \ Coefficient_{CR \ 0.4}}{Power \ Coefficient_{CR \ 0.6}}.$$  

On the other hand, the 40% extended blade performance is 12.5% and 9% better than that of the 20% extended blade, for CR of 0.6 and 0.4 respectively, which were calculated from.

$$\frac{Power \ Coefficient_{CR \ 0.6,40\% \ extension} - Power \ Coefficient_{CR \ 0.6,20\% \ extension}}{Power \ Coefficient_{CR \ 0.6,40\% \ extension}}.$$  

and

$$\frac{Power \ Coefficient_{CR \ 0.4,40\% \ extension} - Power \ Coefficient_{CR \ 0.4,20\% \ extension}}{Power \ Coefficient_{CR \ 0.4,40\% \ extension}}.$$  

This shows that there is a reduction of 3.5% in power coefficient of a 40% extended blade with $CR = 0.4$ when compared with similar extension and $CR = 0.6$. 
Experimental Investigation

Figure 3.19 Experimental and predicted performance
For the blade with no extension, the thrust coefficient was 0.6 but for 10% extension blade, with $CR = 0.6$ and $CR = 0.4$, these are 0.57 and 0.5 respectively as shown in Figure 3.21 (a). This corresponds to approximately, 5% and 17% reduction in the thrust coefficient for $CR = 0.6$ and $CR = 0.4$ respectively, when compared with the no extension blade. Similarly, comparisons were made for the blades with a 20% extension, and the thrust coefficients are 0.47 and 0.45 for $CR = 0.6$ and $CR = 0.4$ respectively as shown in Figure 3.21 (b), corresponding to 22% and 25% reduction. For a 40% extended blade, the thrust coefficient reduces by approximately 23% to 0.46 for $CR = 0.6$ and ~ 30% to 0.43 for $CR = 0.4$ as shown in Figure 3.21 (c). It is evident from these results that the thrust force for $CR = 0.4$ is lower than that of $CR = 0.6$ because of the higher aspect ratio and lower blade area in other words lower solidity; therefore the thrust loading on the turbine will be lower for the blade with lower chord ratio, but this has an adverse effect on the power coefficient.

These results are summarised in Table 3.2 and Figure 3.22 which shows that the blade extension has a significant effect on the thrust coefficient when compared with the chord ratio. The reduction in thrust coefficient increases with the increase in the blade extension for all blade extensions. For blade extension of 10%, 20% and 40%, the difference in the reduction of the thrust coefficient when compared with the different chord ratio ($CR = 0.6$) with reference to no extension blades are 11.7%, 3.3% and 5% respectively. In other words, there
is a further reduction in the thrust coefficient of the blade with $CR = 0.4$ when compared with zero extension blade, the percentage reduction being calculated as:

$$\frac{Thrust \ Coefficient_{CR=0.6} - Thrust \ Coefficient_{CR=0.4}}{Thrust \ Coefficient_{zero \ extension}}$$

But the reduction in thrust coefficient of a blade with $CR = 0.4$, when compared with that with $CR = 0.6$ is ~12, 4 and 7% (Table 3.2) respectively for 10, 20 and 40% extended blades; the reduction being calculated from

$$\frac{Thrust \ Coefficient_{CR=0.6} - Thrust \ Coefficient_{CR=0.4}}{Thrust \ Coefficient_{CR=0.6}}$$

Figure 3.21 Experimental and predicted performance

(i) CR = 0.6

(ii) CR = 0.4

(a). 10% extension

(i) CR = 0.6

(ii) CR = 0.4

(b). 20% extension
(i) CR = 0.6

(ii) CR = 0.4

(c) 40% extension

Figure 3.21 Experimental and predicted performance – cont’d

Figure 3.22 Percent reduction in maximum thrust coefficient
### Table 3.2 Maximum power and thrust coefficient for different chord ratios

<table>
<thead>
<tr>
<th>CR</th>
<th>Maximum Power Coefficient</th>
<th>CR</th>
<th>Maximum Thrust Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>% reduction</td>
<td>0.4</td>
<td>% reduction</td>
</tr>
<tr>
<td>No extension</td>
<td>0.32</td>
<td>0.32</td>
<td>0.0</td>
</tr>
<tr>
<td>10 % extension</td>
<td>0.30</td>
<td>0.29</td>
<td>3.3</td>
</tr>
<tr>
<td>20 % extension</td>
<td>0.24</td>
<td>0.22</td>
<td>8.3</td>
</tr>
<tr>
<td>40 % extension</td>
<td>0.27</td>
<td>0.24</td>
<td>11.1</td>
</tr>
</tbody>
</table>

#### 3.4.4. Step change correction factors

For the blades tested, the lowest step change correction factors that had to be applied in WT Perf to match experimental data, as shown in Figure 3.23 (a) for CR = 0.6 are 0.14, 0.45 and 0.82; and for CR = 0.4 are 0.04, 0.22 and 0.68 for 10, 20 and 40% extensions respectively. For the blade with CR = 0.4, it is anticipated that the influence of the wake generated at the step change region is more severe when compared with that of CR = 0.6 as demonstrated in Figure 3.23. Hence the blade with CR = 0.4 had to be modelled with the lowest correction factor at the step change region when compared with that for CR = 0.6 at every extension. This represents that smaller chord ratios increases the losses, and therefore further reduction in the correction factor had to be applied. For 10% extended blades, the differences are small as shown in Figure 3.23 (b). On the other hand, for the blade with a 20% extension, the lowest correction factor at the step change region for CR = 0.6 and CR = 0.4 are 0.48 and 0.22 respectively as shown in Figure 3.23 (c). As discussed earlier, the vortices from the step change and the tip are likely to be independent and separate to each other, and have individual impact on the rotor performance. Figure 3.23(c) therefore shows that the step loss is significant in this configuration when compared with that of a 40% extended blade shown in Figure 3.23 (d).
As discussed previously, since the rotor radius is normalised, the areas under the curves in Figure 3.23 represent the correction factor ($F$). For the blade with zero extension, the correction factor is maximum (or losses are minimum) when compared with a blade that has an extension.
Figure 3.24 Blade loss factor as a function of blade extension

Figure 3.24 shows a plot of $(1-F_{ave})$ against tip blade extension and different chord ratios. As discussed in Section 3.4.2, $F_{ave}$ may be viewed as a turbine performance factor as it is computed as the area under the curves plotted in Figure 3.23. Figure 3.24 shows that the blade with $CR = 0.4$ has higher losses for all blade extensions. The highest losses are indicated for a blade with 20% extension and $CR = 0.4$.

It might be reasonable to assume that higher sudden change in the chord of the blade of a wind turbine causes the excessive circulation of flow around it to drop. As a result of the drop in circulation, vorticity is shed into the wake. This is discussed further in Chapter 4. The step change in chord of a telescopic blade with lower chord ratio therefore has even more significant effect on the blade performance. Figure 3.25 shows the power coefficient distributions along the blades computed using WT Perf, for different blade configurations. It shows that there is a drop in power coefficient along the blade for all blade extensions at different chord ratios, when compared to the blade without an extension. The blade with a chord ratio of 0.4 has a severe effect on the blade power coefficient as is anticipated during the modelling. For the blade with $CR = 0.4$ and 10% extension, the change in thrust, power and torque is obvious when compared with $CR = 0.6$ as shown in Figure 3.26 (a). The blade with 20% extension and $CR = 0.4$ has the lowest power coefficient at the step change region. Figure 3.26(b) shows the thrust, power and torque coefficient of a 20% extended blade with different chords thus different chord ratios. It is also evident from this figure and as anticipated that the step change region has a detrimental effect on the torque coefficient thus
has a lower power coefficient. Similarly, for the blade with 40% extension, Figure 3.26 (c) shows that there is a sudden drop in thrust, power and torque coefficients at and around the step change region as discussed earlier. This is because the step change in a 40% extended blade is in close vicinity to the peak power generation region of the blade. Any small change in the flow around this region would have a large impact on the blade power and torque coefficients.

![Figure 3.25 Local power coefficient as function of radius for different chord ratios and various blade extensions (Fixed pitch, variable speed) - separated for clarity](image-url)
Figure 3.26 Thrust, Power and Torque coefficient as function of radius for various extensions (Fixed pitch, variable speed) – separated for clarity
Figure 3.26 Thrust, power and torque coefficient as function of radius for various extensions
(Fixed pitch, variable speed) – separated for clarity - cont’d

3.4.5. Power and thrust output
Figure 3.27 and Figure 3.28 shows the plot of power and thrust against wind speeds for different blade extensions and chord ratios respectively. The wind turbine was tested for various wind speeds at optimum tip speed ratio. The result obtained after the implementation of the correlations in WT Perf and the experimental data are in good agreement. The power output of a zero extension blade is the lowest, as expected, and is highest for the fully extended blade (40% extension) for both the chord ratios. At approximately 7m/s, the power output of the 40% extended blade with \( CR = 0.6 \) is approximately 74% (Figure 3.27 (a)) and \( CR = 0.4 \) of approximately 55% (Figure 3.27 (b)) higher than that of a no extension blade despite having a lower optimum power coefficient. This is due to a larger swept area for the 40% extended blade. For blades with 10% and 20% extensions, the increase in power output at optimum power coefficient when compared with no extension blades are 18% and 13% respectively for \( CR = 0.6 \); while for \( CR = 0.4 \), these are 17% and 3% respectively for 10% and 20% extensions as shown in Table 3.3. It is clear from Figure 3.27 (b) that the blade with \( CR = 0.4 \) and blade extension of 20% does not add much to the power output of the wind energy conversion system, being also less than that of a 10% extended blade. Despite having a detrimental effect of the step change in a telescopic blade for any chord ratio, the power
output of a telescopic blade between the cut-in and cut-out velocities increases; the increase is calculated from the equation:

\[
\% \text{ increase in power output} = \frac{\text{Extended Blade Power} - \text{Zero Extension Blade Power}}{\text{Zero Extension Blade Power}}
\]

Similarly the thrust force of a zero extension blade is the lowest, as expected, and is highest for the fully extended blade (40% extension) for both the chord ratios. At approximately 7m/s, the thrust force of the 40% extended blade with \( CR = 0.6 \) is approximately 54% (Figure 3.27).
3.28(a)) and $CR = 0.4$ of approximately 43% (Figure 3.28(b)) higher than that of a no extension blade. This is due to a larger swept area for the 40% extended blade.

**Figure 3.28 Thrust output of a telescopic blade wind turbine**
### Table 3.3 Percentage increase in power output at 7m/s wind speed

<table>
<thead>
<tr>
<th></th>
<th>CR 0.6 (W)</th>
<th>% increase in power output</th>
<th>CR 0.4 (W)</th>
<th>% increase in power output</th>
</tr>
</thead>
<tbody>
<tr>
<td>No extension</td>
<td>23.7</td>
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<td>23.7</td>
<td>0.0</td>
</tr>
<tr>
<td>10 % extension</td>
<td>27.8</td>
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<td>27.8</td>
<td>17</td>
</tr>
<tr>
<td>20 % extension</td>
<td>26.6</td>
<td>13</td>
<td>24.4</td>
<td>3</td>
</tr>
<tr>
<td>40 % extension</td>
<td>41.1</td>
<td>74</td>
<td>36.5</td>
<td>55</td>
</tr>
</tbody>
</table>

### 3.4.6. Summary
The concept of telescopic blades with different chord ratios has been experimentally and computationally analysed for rotor blade performance. It has been established that the step change in the blade chord has significant effect on blade performance for the range of blade extensions and chord ratios studied. The blade with chord ratio of 0.4 when compared with chord ratio of 0.6 had an adverse effect on the power coefficient but favourable effect on the blade thrust as anticipated due to lower solidity. Despite having a detrimental impact on the power coefficient of a telescopic blade, the power output for all blade extensions at different chord ratios were seen to be higher than that of zero extension. It can be deduced that with a lower chord ratio, the power coefficient would be lower, thus lower energy output compared with that of higher chord ratios. Therefore it can be concluded that to minimise the effect of step loss, the chord ratio needs to be closer to unity and the minimum tip blade extension to be more than 2 times the root blade tip chord.
Chapter 4. Wind tunnel investigation of the aerodynamic characteristics of a NACA 0018 telescopic blade at low Reynolds numbers

4.1. Introduction
As discussed in chapter 3, the performance testing of a telescopic blade wind turbine at low Reynolds number revealed additional losses that were attributed to the step change in chord of the telescopic blades. This warranted an investigation into the aerodynamics of such a blade at low Reynolds number and thus an experimental investigation was carried out to understand the effects of the step change region on the aerodynamic performance. Several engineering devices employ airfoils operating at relatively low chord Reynolds numbers. Specifically, airfoil performance in the Reynolds number range from $10^4$ to $10^6$ is of interest for such applications, as it is for small-to-medium scale non-telescopic blade wind turbines and unmanned aerial vehicles [87-89]. In the performance testing discussed earlier, the blade Reynolds number ranged from $4.0 \times 10^4$ to $2.0 \times 10^5$. In this Reynolds number range, the laminar boundary layer on the upper surface of an airfoil is susceptible to separation, even at low angles of attack. When laminar separation occurs, the evolution of the separated shear layer has a strong influence on the entire flow field. For Reynolds numbers below about $3.0 \times 10^4$, the flow does not reattach [90], leaving a wide wake behind the airfoil. For higher Reynolds numbers, the separated shear layer undergoes laminar-to-turbulent transition over the airfoil surface. This may result in flow reattachment, closing the re-circulating flow into a separation bubble. Below a Reynolds number of about $3.0 \times 10^5$, the separation bubble may occupy upwards of 15% of the chord [91]. Independently of the flow regime, flow separation usually has a detrimental effect on airfoil performance and may also contribute to undesirable noise generation.

Weibust et al [92] did an experimental investigation of laminar separation bubbles and found that the static pressure fluctuations grow very rapidly at the rear of the laminar-free shear layer. Mueller and Batill [93] did an experimental study of separation on a two-dimensional airfoil at low Reynolds numbers. They found that a leading edge separation bubble forms on the smooth NACA 66 018 airfoil at 8° angle of attack which induces transition which accounts for a sudden increase in lift. Bastedo and Mueller [94] carried out an investigation on a Wortmann FX63 137 airfoil section and three rectangular wings at Reynolds numbers.
ranging from $8.0 \times 10^4$ to $2.0 \times 10^5$ to determine the effects of the laminar separation bubble on the performance and the span-wise variation in the bubble due to the influence of the tip vortex. They showed that the tip vortex reduced the local angle of attack along the span. Their comparison of pressure distributions showed that the flow-field at span-wise stations on the wings were identical to two-dimensional distributions at the equivalent effective angle of attack.

Taira and Colonius [95] investigated the three dimensional flows around low aspect ratio flat plate wings at low Reynolds numbers. They found that due to the influence of the tip vortices, the three dimensional dynamics of the wake vortices were found to be quite different from the two dimensional von Karman vortex street in terms of stability and shedding frequency.

The effect of Reynolds number on low Reynolds number airfoil behaviour was examined thoroughly by Marchman and Abtahi [96] for a Wortmann FX63 137 wing with an aspect ratio of 8. Bastedo and Mueller [97] and Marchmann et al [98] have investigated the three dimensional effects for aspect ratios of 1-10 for a low Reynolds number airfoil. They show that there is a range of Reynolds number between about $7.5 \times 10^4$ and $4.0 \times 10^5$ where the separation bubble behaviour dominates the flow and determines the stall behaviour. The limits of this Reynolds number range are dependent on the aspect ratio, and as the aspect ratio decreases, the vortical flow around the wing tip becomes more of a factor in the overall upper-surface flow behaviour [99]. Below some aspect-ratio-dependent Reynolds numbers near $7.5 \times 10^4$ and $1.0 \times 10^5$ [99], the airfoil behaves like a thin plate with separated upper-surface flow at almost all angles of attack.

Gerakopulos [87], Timmer [100] and Nakano et al [101] have investigated a NACA 0018 airfoil in low Reynolds number flows. Gerakopulos [87] carried out an experimental study on the lift and separation bubble characteristics of a NACA 0018 airfoil. They presented the surface pressure measurements for Reynolds numbers from $8.0 \times 10^4$ to $2.0 \times 10^5$ and angles of attack from $0^\circ$ to $18^\circ$. Two distinct regions were identified in the lift curves, being a region of rapid and linear growth of the lift coefficients at low angles of attack, and a region of more gradual and linear growth at higher pre-stall angles as shown in Figure 4.1. The slope of the lift curve in each region was found to be linked to the rates of change in separation, transition, and reattachment locations with the angle of attack.
Figure 4.1 Two distinct regions of linear growth in lift coefficient curves [87]

Timmer [100] reports lift coefficient data for several angles of attack from 0° to 30° and Reynolds numbers ranging from $1.5 \times 10^5$ to $10^6$. On the lower surface, laminar separation bubbles for span-wise wake rake traverse measurements showed an irregular three-dimensional pattern. Nakano et al [101] measured surface pressure distributions for $\alpha = 0°$, $6°$ and $15°$ and measured the velocity field at $\alpha = 6°$ for $Re_c = 1.6 \times 10^5$.

Yarusevych and Boutilier [102, 103] studied the vortex shedding characteristics of a NACA0018 airfoil at low Reynolds numbers. Their focus of investigation was on the effects of the Reynolds number and airfoil thickness on the shedding frequency of the coherent structures. They found that with the increase in Reynolds number, the shear layer reattaches on the upper surface of the airfoil, thus the wake shedding frequency decreases significantly and the vortical structures become less coherent.

A lot of research has been carried out on the NACA0018 airfoil at low Reynolds numbers, but there is no study to show the effects of span-wise pressure variation of a telescopic blade. In a telescopic blade configuration, the tip blade slots inside the root blade for different lengths to obtain different blade extensions. This research is aimed at looking at the effect of a step change in blade chord of a telescopic blade at low Reynolds numbers. Wind tunnel tests were carried out in a subsonic boundary layer wind tunnel at The University of Auckland.
4.2. Experimental setup
Experimental investigations were carried out with a specifically designed telescopic blade to get an insight into the flows around the step-change region of the blade. The tests were carried out in the low speed section of a closed loop subsonic wind tunnel at the University of Auckland. The 8.0m long rectangular test section of the tunnel has a height of 1.2m and a width of 1.83m. Flow enters the test section through a honeycomb and screens, with background turbulence intensity level of less than 1%. The uncertainty of the free-stream velocity measurements was estimated to be less than 2.5%. A NACA 0018 airfoil made of PVC, with a constant chord length of 0.25m and a span of 0.5m (root blade) and a constant chord length of 0.15m and a variable span of 0 to 0.25m (tip blade) which gave a chord ratio ($Chord\ ratio = \frac{tip\ blade\ chord}{root\ blade\ chord}$) of 0.6, was mounted vertically in the test section as shown in Figure 4.2. The root blade span was fixed and the tip blade lengths were varied in order to get different blade extensions. The tip blade could go inside the root blade to get different extensions. The airfoil model spanned according to the blade extension studied in this research. As shown in Figure 4.2, the angle of attack adjustments were facilitated by a turntable mechanism, with an angular resolution of $1\degree$. Surface pressure measurements were obtained from 256 pressure taps symmetrically distributed on the upper and lower surfaces along the model. A Pitot-static probe installed in the test section upstream of the model served as a reference free-stream static pressure. A multichannel digital pressure scanner was employed to acquire pressure measurements. For data presentation, the origin of the coordinate system is located at the leading edge of the airfoil. For all of the cases investigated, the uncertainty of the surface pressure measurements is estimated to be less than about 5% of the dynamic pressure.

![Figure 4.2 Photograph of model in wind tunnel looking downstream showing blade configuration](image-url)
4.2.1. Blade configurations

Four different tip blade lengths of 0, 65, 125 and 250 mm (0%, 13%, 25% and 50% tip blade extensions) were chosen, which are equivalent to 0, ¼, ½ and 1 root blade chords respectively. The tests were divided into several parts and for each blade configuration, two dimensional (2D) and three dimensional (3D) tests were conducted. The 2D and 3D tests were carried out for each blade extension by having a false wall at the end to get a 2D setup, and the same removed to get a 3D configuration as shown in Figure 4.3. The 2D and 3D setups in the wind tunnel are shown for tip blade lengths of 0, 65, 125 and 250mm in Figure 4.3 (i) to (iv) respectively.

(a) 2D        (b) 3D

(i) Zero extension (no tip blade)

(a) 2D        (b) 3D

(ii) Quarter extension (Tip blade length 65mm)

Figure 4.3 Photograph of wind tunnel blade setup
4.2.2. Pressure tap locations
In-line pressure taps were used rather than the staggered configuration which is believed to minimise the effect of the upstream taps on the downstream taps, because in this research it was necessary to determine the character of the flow field at various span-wise stations due to the sudden change in the blade chord at the tip blade. Based on the number of available pressure channels, limited numbers of pressure taps were installed on the blades. In total 135 and 120 taps were installed on the tip blade and root blade respectively. Figure 4.4 shows the chord-wise distribution of the taps, while Figure 4.5 shows the span-wise distribution. Both the blades had pressure taps at the same position relative to their leading edge i.e. the non-dimensional distance from the leading edge \((x/c)\) is the same for both the root and tip blades.
as shown in Table 4.1. Table 4.2 and Figure 4.5 show the span-wise distribution of the pressure taps from the top of the tip blade. No pressure taps were installed close to the wind tunnel floor region i.e. there were no taps from the floor up to 200 mm of the root blade span.
Table 4.1 Chord-wise pressure tap distribution

<table>
<thead>
<tr>
<th>Tap</th>
<th>1</th>
<th>2/3</th>
<th>4/5</th>
<th>6/7</th>
<th>8/9</th>
<th>10/11</th>
<th>12/13</th>
<th>14/15</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative distance from leading edge (Distance/chord)</td>
<td>0</td>
<td>0.024</td>
<td>0.064</td>
<td>0.12</td>
<td>0.12</td>
<td>0.4</td>
<td>0.64</td>
<td>0.92</td>
</tr>
</tbody>
</table>

Figure 4.4 Chord-wise pressure tap locations [104]

Table 4.2 Span-wise pressure tap distribution

<table>
<thead>
<tr>
<th>Row</th>
<th>Distance from top (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>2</td>
<td>25</td>
</tr>
<tr>
<td>3</td>
<td>55</td>
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<tr>
<td>4</td>
<td>105</td>
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<td>5</td>
<td>115</td>
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<td>6</td>
<td>150</td>
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<td>230</td>
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<tr>
<td>15</td>
<td>345</td>
</tr>
<tr>
<td>16</td>
<td>405</td>
</tr>
<tr>
<td>17</td>
<td>555</td>
</tr>
</tbody>
</table>

Figure 4.5 Overview of span-wise pressure tap locations
4.2.3. Pressure measurement setup
Surface pressures were measured on the turbine blades with a multi-channel pressure measurement system developed by the Aero-lab technician Dr. Nick Velychko at The University of Auckland. The system has 256 channels, and each of them can acquire up to 3900 samples per channel per second. The transducers have a pressure range of 450 Pa and a resolution of 9.25mV/Pa with an accuracy better than ±0.5 Pa [105, 106]. All the transducers were pneumatically connected to a reference static pressure from a pitot-static probe. The system is made of 4 boxes, which contain 64 transducers each as shown in Figure 4.6.

![Figure 4.6 Pressure module](image)

(a) Tubings attached to the pressure module  (b) Pressure modules with calibrator

The boxes were connected to a computer, which ran the data acquisition software. The pressure measurements were acquired at 600Hz for 1 min and the mean values were used for the analysis. Each transducer was connected to a pressure tap through a tube as shown in Figure 4.6 (a); the tubes were run inside the airfoil. The pressure transducers were calibrated using the pressure transducer calibrator as shown in Figure 4.6 (b). Figure 4.7 shows a schematic drawing of the data acquisition system from the blade pressure transducers to the data reduction and storage used for this research.
The pressure distribution on the airfoil is expressed in dimensionless form by the pressure coefficient \( c_p \)

\[
    c_p = \frac{p_i - p_\infty}{\frac{1}{2} \rho U_\infty^2}
\]  

(4.1)

where \( p_i \) is the surface pressure measured at location \( i \) on the surface, \( p_\infty \) is the static pressure in the free stream, \( \rho \) is air density, and \( U_\infty \) is the free-stream velocity given by

\[
    U_\infty = \sqrt{\frac{2(p_{\text{stagnation}} - p_\infty)}{\rho}}
\]

(4.2)

where \( p_{\text{stagnation}} \) is the stagnation pressure measured at the tip of the Pitot static tube.

### 4.2.4. Wind tunnel boundary layer

The test section (1.83 by 1.2 m) of the wind tunnel has a turntable that is ~ 8m away from the wind tunnel flow entry. This configuration means a significantly thick boundary layer develops on the floor and walls. The wind tunnel boundary layers were measured before the tests were carried out. The well known theory developed by Schlichting [107] for turbulent flow was used to find the theoretical boundary layer thickness at the model location,

\[
    \frac{\delta}{x} = 0.38Re_x^{-1/5}
\]

(4.3)
where $\delta$ is the boundary layer thickness, at a distance $X$ from wind tunnel entry to the test section; $Re_x$ is the Reynolds number.

Figure 4.8 shows the variation of the theoretical boundary layer thickness based on the Schlichting equation. In order to establish the boundary layer in the wind tunnel test section, a traversing rig with a Cobra probe [83] was used as shown in Figure 4.9. Different wind speeds were used to establish the boundary layer profile. The measured profiles shown in Figure 4.10 (a) and (b), are for wind speeds of 3m/s without the false wall, and 6m/s with a false wall respectively. A false wall was erected in the wind tunnel to get the 2D setup and these were removed to get the 3D setup. Figure 4.10 (a) and (b), shows that the floor boundary layer thicknesses are approximately 170 and 150 mm for wind speeds of 3 and 6 m/s respectively. Figure 4.10 (b) shows the velocity profile for a 2D case with the false wall. It is evident from these that the boundary layer at the floor is approximately 150mm and at the top near the false wall is ~ 50mm as theoretically predicted. For the root blade, the first set of pressure taps were installed 200mm away from the floor in order to avoid these being in the boundary layer region.

![Figure 4.8 Theoretical boundary layer thickness](Image)

![Figure 4.9 Traversing rig with cobra probe](Image)
The pressure distributions on the airfoil surfaces are shown and discussed in the following sections for various Reynolds numbers and different angles of attack.

4.2.5. Benchmarking the experimental setup
In this research, tests were carried out for 2D and 3D blade configurations. In a 3D situation, the effects of the floor boundary layer and the downwash due to the tip vortex and low aspect ratio wing are present, whereas in a 2D situation, the floor and false wall boundary layers are present.

Due to resource limitations (wind tunnel height and turntable orientation), the blades used for the aerodynamic study of the telescopic blade had a low aspect ratio. The aspect ratio for the root blade was 2 (fixed) while for the tip blade, it varied over the range 0 - 1.7 as the tip blade lengths were varied to get different blade extensions. As the aspect ratio of the blade is low, this would have an adverse effect on the blade aerodynamics when compared with a larger aspect ratio blade. Another adverse effect on the blade aerodynamics would be from the wind tunnel boundary layer. As discussed earlier, the turntable was ~ 8m away from the flow entrance therefore it developed a significant boundary layer thickness of ~ 160mm at 6m/s wind speed.

Before the pressure taps were installed on the blades, a set of experiments were carried out with the JR3 [108] (model # 30E12A-I40) force balance in order to get some insight into the
stall angle due to 3D effects. The JR3 is a six-component force balance which measures the three force and three moment components. The capability of the load cells used in this research are 44N, 44N and 89N for Fx, Fy and Fz respectively and the moments 3.39Nm each for Mx, My and Mz. But the electrical load settings were amplified to increase the sensitivity of the system thus reducing the maximum load ratings as this JR3 is specifically designed for building aerodynamics studies in the wind tunnel. Therefore, the maximum the system could electrically handle before it saturates is 8N, 8N and 16N for Fx, Fy and Fz respectively and the moments 1.5Nm for Mx, My and 0.18Nm for Mz. This limited the direct measurement of the lift and drag over the range of Reynolds number in this study. Nevertheless, the tests that were able to be carried out using the JR3 force balance, were compared with the theoretical values calculated using Xfoil [109]. The JR3 force balance was attached to the balance holder as shown in Figure 4.11 (a). The telescopic blade with the hub attached at ¼ chord were attached to the JR3 and calibrated using the procedure described by Bhatt [110] as shown in Figure 4.11 (b).

(a) JR3 Force balance with turntable cover (b) Blade attached to JR3 force balance for calibration

Figure 4.11 JR3 force balance assembly

With the JR3 force balance, 3D tests were carried out for one Reynolds number (8.5*10⁴). Above this Reynolds number, the JR3 load cell reached its electrical limits.

Experiments carried out in the wind tunnel showed delayed stall angles as shown in Figure 4.12. The wind tunnel 3D experimental data were corrected for the 2D case using the correlation from the literature [111]. Due to the three dimensional downwash effects, the effective angle of attack for a blade in 3D changes. The following equations were utilised to
make corrections to the angle of attack and the experimental data for comparison with theoretical values,

\[ \alpha_i = \frac{C_L}{e\pi AR_{eff}} \]  

(4.4)

where \( \alpha_i \) is the induced angle, \( e \) is the span efficiency (Oswald) factor, \( AR_{eff} = 2 \times AR \) [111] is the effective aspect ratio and \( C_L \) is the 3D lift coefficient. The span-wise efficiency of 0.65 [112] was assumed for this analysis. Span efficiency is a correction factor that represents the change in drag with lift of a three dimensional wing as compared with an ideal wing having the same aspect ratio and an elliptical lift distribution. Therefore, the effective angle of attack is then:

\[ \alpha_{eff} = \alpha - \alpha_i \]  

(4.5)

where \( \alpha \) is the 3D angle of attack. The wind tunnel 3D experimental data were corrected to the 2D case with infinite aspect ratio and compared with the 2D data generated using XFOIL as shown in Figure 4.12. The trends are very similar with slightly lower magnitudes noted in the measured data up to the stall angle. From these results, a correlation for 3D (finite aspect ratio) to 2D (infinite aspect ratio) angle of attack i.e. the effective angle of attack was deduced which was then used for other benchmarking analysis.

Figure 4.12 Characteristic of a 3D NACA 0018 blade
The two-dimensional tests, as shown in Figure 4.13, were carried out for the blade with zero extension to show the effect of the boundary layer on the span-wise pressure distributions and were compared with data from Xfoil. As shown in Figure 4.14, the pressure coefficients along the chord were compared with Xfoil data for pressure taps along the span-wise direction with rows 10 and 17 representing the taps close to the false wall and floor respectively (see Figure 4.13). For rows 10 to 13, the experimental pressure coefficients obtained do not match with those from Xfoil (see Figure 4.14). This is due to the false wall boundary layer, which in this case was approximately 50 mm thick; thus taps in rows 10 to 13 were in the boundary layer region giving lower pressure coefficients. For rows 14 to 17, the experimental pressure coefficients are symmetrical and agree well with those obtained through Xfoil. These results and comparisons validated the experimental setup and measuring system, and thus the subsequent tests were carried out with confidence. Regular calibrations were carried out after several tests in order to maintain the integrity of the whole experimental and measurement setup.

Figure 4.13 2D experimental setup and schematic

Figure 4.14 2D Span-wise pressure coefficient of root blade ($\alpha = 0^\circ$, $Re_c \ 1.2 \times 10^5$)
Once the experimental setup was tested and calibrated as discussed earlier for $\alpha = 0^\circ$, a series of 2D tests were carried out for various angles of attack in the wind tunnel and compared with data from XFOil for benchmarking purposes. The effective angles of attack were calculated for XFOil and the experimental data were compared. Results from row 14 (a row with no boundary layer influence) are presented in Figure 4.15, which shows that the experimental data is in good agreement with that of XFOil for the range of angles of attack studied. These results gave further confidence in the setup, thus the tests were carried out for other blade configurations.
Figure 4.15 Pressure coefficient for various angles of attack (2D, Re $1.2 \times 10^5$) for row 14
4.3. Results
Large amount of data, both quantitative and qualitative were obtained for the telescopic blade with a NACA0018 airfoil section. Several tests were carried out for different blade configurations (zero, quarter, half and full extensions) which consisted of both 2D and 3D experimental setups. The experimental chord Reynolds number ranged from $4.0 \times 10^4$ to $2.0 \times 10^5$ based on tip and root blade chords respectively. The following sections describe the in-depth analysis of the aerodynamics of a telescopic blade.

4.3.1. Effect of the step-change in chord
Figure 4.16 shows the chord and span-wise pressure coefficient distribution for a 3D fully extended blade at different angles of attack, with the Reynolds number based on the root and tip blade chords equal to $1.8 \times 10^5$ and $1.1 \times 10^5$ respectively. It is evident from these graphs as shown in Figure 4.16, that the magnitude of the pressure coefficients is higher for the tip blade when compared with those for the root blade over pre-stall angles of attack from 0 to 12°. The higher pressure coefficients on the tip blade could be due to Reynolds number effects.

It is known that at Reynolds numbers of less than $5.0 \times 10^5$, the formation of laminar separation bubbles usually have major impact on aerodynamic performance of aerofoils [93, 94, 113]. Xfoil data in Figure 4.17 (a) shows, for Reynolds numbers of up to $1.8 \times 10^5$, the lift curve slopes are steeper for alpha up to 6°, and beyond this the gradient of the slope reduces before it stalls. For Reynolds number of $5.0 \times 10^5$ and $1.0 \times 10^6$, the gradient of the lift curve is smaller than that at lower Reynolds numbers. These Reynolds number effects are due to the laminar separation bubbles which are classified as ‘long’ and ‘short’ separation bubbles. Long bubbles may be as much as 20-30% of the chord in length and adversely affect the airfoil performance [94] whereas the short bubbles are usually only a few percent of the chord in length causing very little modification of the pressure distribution. These bubbles serve primarily as a tripping mechanism to allow reattachment of an otherwise separated shear layer and increases the performance [94, 114]. Associated with this, a long bubble tends to increase in length as incidence is increased, decreasing the slope of the lift curve as discussed earlier. A short bubble, on the other hand, decreases in length with increasing angle of attack, yielding improved airfoil performance as observed in Figure 4.17 (a) for alpha up to 6°. The important physical parameters that affect formation of laminar separation bubbles have been determined to be the airfoil geometry, angle of attack, chord Reynolds number and the free stream disturbance environment [94].
It is also apparent from the sudden drop in measured $c_p$ in Figure 4.16 that the tip blade stalls somewhere between $12 \sim 15^\circ$ and the root blade stalls at $23.5^\circ$. There is clear evidence of a drop in pressure as shown in Figure 4.16 around the step change region for the root blade for all angles of attack, and as the angle of attack increased, the pressure drop also increased. On the other hand, the pressure coefficient of the tip blade around the step change region increased with increase in angle of attack up to the tip blade stall angle. As shown in Figure 4.16, at $\alpha = 15^\circ$, the tip blade is not fully stalled at the tip region and the drop in pressure coefficient is significant around the step change region, whereas the tip blades for half and quarter extension are fully stalled at $15^\circ$ (refer to Appendix A).

It is therefore assumed that the step change which caused some 3D effects caused the tip blade to stall earlier than would otherwise happen. Nicholas et al [115] and Bastedo and Mueller [94] have shown that rectangular wings of relatively low aspect ratio have a very three dimensional flow structure due to the effect of tip vortices and significant variations in their separation bubbles. A similar situation is experienced in this research where the aspect ratios tested were low. Generally any sudden change in the chord of the blade causes the circulation to drop and as a result of the drop in circulation, vortex roll up takes place and sheds into the wake as shown in Figure 4.18, thus disturbing or tripping the flow.

From Figure 4.17 (a), it is worth noting that the blade with Reynolds number of $1.1*10^5$ stalls at $14^\circ$, whereas the blade at $1.8*10^5$ Reynolds number stalls at around $15^\circ$, therefore the difference is $1^\circ$ but the difference in the telescopic blade stall angles at higher Reynolds number and the one with lower Reynolds number is large ($\sim 10^\circ$). This difference in the stall angle could be attributed to phenomena in the wind tunnel including buoyancy, solid blockage, wake blockage and the streamline curvature [116]. The significance of these were not studied in this research as we were more interested in understanding the effects of step change in a telescopic blade.

It is also apparent from Figure 4.16, at $\alpha = 15^\circ$ that the pressure coefficient at the tip of the root blade has further reduced when compared with angles of attack less than $15^\circ$. Similar trends were evident for blades with half and quarter extensions except that the tip blades are fully stalled across the span-wise direction at $15^\circ$ (refer to Appendix A).

As shown in Figure 4.17 (a), the lift coefficients for a blade with a Reynolds number of $1.1*10^5$ is higher than that of $1.8*10^5$ for alpha of up to $6^\circ$, and thereafter the lift coefficients are similar up to the stall angle. Then the foil with Reynolds number $R_e = 1.1*10^5$ stalls
before the foil with \( R_e = 1.8 \times 10^5 \). Similar trends of Reynolds number effects were shown in the telescopic blade testing. Figure 4.17 (b) shows the pressure coefficient at 3° and 9° for a 2D airfoil for Reynolds numbers of \( 1.8 \times 10^5 \) and \( 1.1 \times 10^5 \) based on root and tip blade chords respectively. It is evident from this figure that at 3°, the pressure coefficient of the tip blade is higher than that of root blade, and at 9° the pressure coefficient of the root blade is higher than that of tip blade. Similar trends in terms of pressure coefficients for effective angles of attack were observed with the telescopic blade testing that were carried out in this research.

From these observations it can be deduced that longer tip blades would probably not stall as early as the fully extended ones tested in this research, in other words if the aspect ratio of the tip blade were higher than those tested (see Figure 4.16), the tip blades would stall closer to the root blades.

**Figure 4.16 Experimental pressure coefficients for full extension, 3D, \( R_e \text{ root} 1.8 \times 10^5 \) (suction side)**
Figure 4.16 Experimental pressure coefficients for full extension, 3D, $R_{c, root} \times 10^5$ (suction side) – cont’d
Figure 4.16 Experimental pressure coefficients for full extension, 3D, $Re_{root} \ 1.8 \times 10^5$ (suction side) – cont’d
Figure 4.17 Xfoil data for a NACA0018 airfoil, AR=∞, (a) Lift coefficient (various Reynolds numbers) (b) Pressure coefficient (Re_root $1.8 \times 10^5$, Re_tip $1.1 \times 10^5$)

Figure 4.18 Flow visualisation at step region, alpha = 9°
4.3.2. 2D and 3D effects on the telescopic blade
For the range of tests carried out in this research, the 2D versus 3D configuration effects were also considered. When an end plate is attached to the end or tip of a 3D model, the setup becomes 2D and in this research, as discussed already, a false wall was used. It is very clear from Figure 4.19 that the pressure coefficients for all angles of attack for the 2D experimental configuration are higher than those from the 3D setup.

4.3.2.1. 2D flow over telescopic blade
For the 2D configuration, one would expect the pressure coefficient to be the same as those obtained from Xfoil and shown in Figure 4.17 (b), but this is not evident in the 2D case shown in Figure 4.19. For the 2D root blade, this could be due to low aspect ratio of the airfoil thus the three dimensional effects and also the influence of the step vortex (see Figure 4.18) which has likely altered the upper surface flow-field. Bastedo and Mueller [94] found similar effects caused by the tip vortex on a Wortmann FX63 137 airfoil with an aspect ratio of 2, which is the same as the aspect ratio used in this research and at similar Reynolds numbers.

Moving inboard away from step change region along the root blade and thus away from the step vortex, the induced flow component strength decreased and the pressure coefficient rose to a maximum and then started to decrease. In the maximum $c_p$ region, the airflow and the boundary layers can be assumed to be closer to two dimensional and can be approximated by airfoil data at the angle of attack equal to the effective angle of attack at that span station. After this maximum plateau, the pressure coefficients start to drop towards the root of the root blade i.e. nearer the wind tunnel floor and this is believed to be caused by the floor boundary layer. In the test section, the floor boundary layers measured were approximately 150 to 170 mm in thickness. This boundary layer region has a reduced velocity resulting in a span-wise pressure gradient which could induce the flow to skew thereby changing the effective angle of attack, thus reduces the pressure coefficient. It is worth noting that for $\alpha = 12^\circ$, the span-wise pressure coefficient distribution of the root blade is fairly uniform but that the tip blade has started to stall. The maximum pressure coefficient of the root blade kept on rising with increasing angles of attack, whereas the pressure coefficient around the step change region reduced till it stalled.

On the other hand for the 2D tip blade, around the tip region, the pressure coefficient is low when compared with those at span-wise locations towards the step change region. As shown in Figure 4.19, for 2D at $\alpha = 0^\circ$, around or close to the step change region, the pressure
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coefficient is similar to that generated using the Xfoil (e.g. \( C_{p_{\text{max}}} \sim -0.6 \)). Ideally, in a 2D situation one would expect a uniform pressure coefficient distribution in the span-wise direction, but the pressure coefficient around the tip region is not uniform. This is due to the boundary layer that is present on the end plate, and the corner flow. The end plate forms a boundary layer and the formation of corner flow (refer Mueller and Pelletier [117]) induces flow in other directions, thereby reducing the pressure coefficient. The strength of this induced flow reduces when moving away from the end plate thus the effect reduces. Selig et al [118] also showed that for low Reynolds numbers, two-dimensional airfoil aerodynamics might be affected by the strong three-dimensional effects originating from the endplates. Somewhere between 9°-12°, the tip blade starts to stall, and as shown in Figure 4.19, for \( \alpha = 12^\circ \), the tip blade is not fully stalled. It is worth noting that the tip region of the tip blade, up to \( \alpha = 21^\circ \) is not completely stalled. It is anticipated that this could be due to the presence of a separation bubble present around the tip of tip blade. The presence of end plate boundary layer is also obvious for all angles of attack as the pressure coefficients are reduced in this region.

4.3.2.2. 3D flow over telescopic blade

For the 3D tests that were carried out, the end plate was removed. It is clear from Figure 4.16, that for the 3D configuration, the general trends show lower pressure coefficients for all angles of attack when compared to the 2D configurations. For the 3D root blade, the pressure coefficient profiles are similar to those of the 2D root blade, but the magnitudes are different. This could be due to the presence of stronger three dimensional effects created by the tip vortex, evident in the span-wise direction in Figure 4.20 which would have a strong influence on the span-wise pressure distribution. In Figure 4.20, a trail of tip vortex is evident in the span-wise direction. It is known that this vortex would change the effective angle of attack, which is also evident from the test as the 3D root blade stalled at 23.5°, whereas the 2D root blades stalled at 23°.

For the 3D tip blade, for alpha up to 6°, the pressure coefficients of the 3D tip blade are similar (at \( \alpha = 6^\circ \)) or higher (over \( \alpha = 0^\circ \) and 3°) than that of the 2D tip blade around the tip region. For alpha 0° and 3°, it could be argued that the presence of weak tip vortices in 3D tip blade results in possibly a flat or low gradient effect on the pressure coefficient in the span-wise direction away from the tip. Whereas for the 2D tip blade, at alpha 0° and 3°, the pressure coefficient around the tip is low, which is likely due to a stronger influence of the boundary layer and the end plate corner flow as discussed earlier. Generally however, the
overall pressure coefficients of the 3D tip blade along the span-wise direction are low in magnitude when compared with those for the 2D tip blade. The 3D tip blade started stalling somewhere between 12° and 15°, and stalled completely at around 18°. Figure 4.21 (a) shows a three dimensional representation from which it is evident that for $\alpha = 15^\circ$, the 3D tip blade is not fully stalled. This could also be due to a separation bubble which might be present and sitting in between the tip and step vortices.

Figure 4.19 Full extension, 2D and 3D effects, various angle of attack, Re_root - 1.8x10^5
Figure 4.19 Full extension, 2D and 3D effects, various angle of attack, Re_root - 1.8x10^5 – cont’d
Figure 4.19 Full extension, 2D and 3D effects, various angle of attack, Re_root - 1.8x10^5 – cont’d

Figure 4.20 Flow visualisation at the tip region, alpha = 9°

Figure 4.21 (a) shows that, at $\alpha = 12^\circ$, the 2D fully extended tip blade has started to stall whereas the half and quarter extension blades have not started (see Figure 4.21 (b) and (c)). The half and quarter extension 2D tip blades stalled somewhere between 12° and 15°.

The 2D root blade, at $\alpha = 12^\circ$, with the tip blade fully extended had an overall low pressure coefficient in the span-wise direction when compared with that of half and quarter extensions (see Figure 4.21 (a) to (c)). This could be due to stronger step vortices creating larger three dimensional effects, thus reducing the pressure coefficients at shorter extensions.
On the other hand, the 3D root blade had similar pressure coefficients for full and half extension blade at $\alpha = 12^\circ$ whereas for the quarter extension blade, the span-wise pressure coefficients were low when compared with other extensions; see Figure 4.21 (a) to (c). The 3D tip blade started to stall between $12^\circ$ to $15^\circ$ for full and half extensions and before $12^\circ$ for quarter extension. This further reduction of the stall angle could be due to the tip and step vortices interacting with each other due to smaller blade extension. This interaction of tip and step vortices could possibly intensify their strength perhaps inducing a stronger three dimensional flow compared with other three dimensional effects for other tip blade extensions.

(a) Full extension

Figure 4.21 3D view, 2D and 3D effects, various angles of attack, $R_{e, root} = 1.8x10^5$
Figure 4.21 3D view, 2D and 3D effects, various angles of attack, $Re_{\text{root}} - 1.8 \times 10^5$ – cont’d
4.3.3. Effects of Reynolds number
Several tests were carried out for various Reynolds numbers, and the results are shown in Figure 4.22 (a) and (b) for 3D and 2D configurations respectively. These plots show that as the Reynolds number increases, the magnitude of the pressure coefficients increases for the range of Reynolds number tested. Figure 4.22 (a) shows the 3D pressure coefficient for Reynolds number of $7.0 \times 10^4$ to $1.8 \times 10^5$ and alpha $9^\circ$ to $15^\circ$. As previously shown in Figure 4.17(a), for 2D blades with infinite aspect ratio, with Reynolds number effects, similar trends are also present in these 3D tests. The stall angles for the blade with a Reynolds number of $7.0 \times 10^4$ is around $6^\circ$ to $9^\circ$, and for $1.0 \times 10^5$ and $1.8 \times 10^5$ it is between $9^\circ$ to $12^\circ$ and $12^\circ$ to $15^\circ$ respectively. On the other hand, for the 2D tests, the magnitude of the pressure coefficients is higher as expected, but the effect of step vortices are stronger when compared with 3D tests. The stall angles for the 2D tests at Reynolds number of $7.0 \times 10^4$, $1.0 \times 10^5$ and $1.8 \times 10^5$ are around $9^\circ$ to $12^\circ$ for all blade extensions. The presence of a separation bubble as discussed already is likely to be present in all Reynolds number cases studied for the 2D tests.

Figure 4.22 Pressure coefficient, variation with Reynolds numbers - 3D view, full extension,

$R_{c, \text{root}} = 1.8 \times 10^5$
Figure 4.22 Pressure coefficient, variation with Reynolds numbers - 3D view, full extension, 

Re_{root} = 1.8*10^{5} – cont’d
4.3.4. Effect of blade extension

Figure 4.23 compares the pressure coefficient distribution on the basis of blade extension at a number of angles of attack. The blade pressure coefficients are seen to increase in magnitude with increase in tip blade length for the 3D configuration. Increasing the tip blade length, to have different tip blade extensions, effectively increases the aspect ratio of the blade. Increase in aspect ratio means only a smaller proportion of the blade suffers from the three dimensional tip effects. This increases the overall pressure coefficient of the wing, and thus the lift curve slope. The tip blade aspect ratios for the range of tests carried out were 1.67, 0.83 and 0.4 for full, half and quarter extensions respectively, and for the root blade the aspect ratio was fixed at 2. Bastedo and Mueller [97] showed that as the aspect ratio was reduced, the maximum value of the lift coefficient as well as the lift to drag ratio decreased, while the magnitude of the minimum drag coefficient increased due to the velocity field induced by the tip vortex. Similar trends were observed in this research whereby as the blade extension increased (in other words with the increase in aspect ratio) there was a clear trend of increasing maximum pressure coefficient with respect to increase in tip blade lengths.

For the root blade, the span-wise pressure coefficients for zero and quarter extension blades are similar and lower than that of half and full extensions. This again could be caused by the 3D effects due the presence of stronger tip and tip-step vortices present for no and quarter extensions respectively. Overlapping of tip and step vortices at small extensions could possibly intensify the strength of the vortices, thus producing stronger 3D flows with a greater influence on the span-wise pressure distribution. On the other hand, the span-wise pressure coefficients for blades with half and full extensions increased in magnitude as the blade length increased. It is assumed that with this increase in tip blade length, the tip and step vortices are localised around that region, whereby it could not intensify their strength, thus the effects are somewhat less than those of no and quarter extensions. It is also evident from Figure 4.23, that a fully extended blade has the highest maximum pressure coefficient on the root blade around the step change region and that the drop in span-wise pressure coefficient is localised close to the step change region. As shown in Figure 4.23, the tip blades stall between 12° and 15° for all blade extensions and it is worth noting that once the blades with higher aspect ratio (half and full extension blades), stall, they have a larger influence on the root blade span-wise pressure distribution when compared to zero and quarter extension blades.

From all these results, it is very clear that the change in the blade chord causes a sudden drop in pressure around the step region. This suggests vortex roll ups into the wake at the step which in
turn creates a 3D flow thereby affecting the span-wise pressure distribution of a low aspect ratio and low Reynolds number telescopic blade system. From this it can be expected that tip blades with higher aspect ratios would perform better with reduced influence of the step, thereby affecting the span-wise pressure distribution.

Figure 4.23 3D, Various blade extensions, 2D view - $R_{e_{\text{root}}} \times 10^5$
On the other hand, for 2D flows, as shown in Figure 4.24, the maximum pressure coefficient for the tip blade is similar across all blade extensions and the magnitude is higher than those for the 3D situation, as expected. For the root blade, the span-wise pressure coefficient...
distributions are very similar for all blade extensions studied. In the 2D test configuration, Figure 4.24 suggests that, the effect of tip and step vortices are localised, thereby do not significantly affect the span-wise pressure distributions. As expected, the three dimensional effects are less significant because the pressure coefficients are higher in 2D test configuration as evident in Figure 4.24.

Figure 4.24 2D, Various blade extensions, 2D view - $R_e, \text{root} \ 1.8 \times 10^5$
4.4. Summary
The present wind tunnel investigation aimed to describe the effects of a step change in the chord of a telescopic blade. Surface static pressures were measured for different tip blade extensions at various angles of attack and Reynolds numbers. Two-dimensional and three-dimensional effects were also investigated through different configurations.

From the tests carried out, it is very clear that the sudden change in the blade chord causes an abrupt drop in pressure around the step region. This suggests vortex roll up is occurring into the wake at the step which in turn creates a 3D flow thereby affecting the span-wise pressure distribution of a low aspect ratio and low Reynolds number telescopic blade system. It has been found that lower aspect ratio or smaller length of the tip blade has an adverse effect on the pressure distribution of the root blade of a telescopic blade system. It can be asserted that longer tip blades would have less effect around the step region, thus the span-wise pressure distribution would not be affected as much. From this it can also be deduced that longer tip blades would not stall as early as the fully extended ones tested in this research. In other words if the aspect ratio of the tip blade were higher than those that were tested, the tip blades would stall at angles closer to those of the root blades.
It has been established that with longer tip blade sections, the maximum pressure coefficient of both tip and root sections of a telescopic blade system also increased.

As the Reynolds number increases, the blade performance also increases. For the telescopic blade system studied, the tip and root blades had different chord Reynolds numbers for the same free stream velocity due to different chord lengths. The effects of different chord Reynolds numbers were quite pronounced in this research as the tip blade stalled first when compared with the root blade. In other words, if the tip blade chord were closer to that of the root blade (higher chord ratio), this Reynolds number effect would be minimised or eliminated.
Chapter 5. Energy output analysis for a Telescopic Blade Wind Turbine

5.1. Introduction and background
With the current energy crisis and growing environmental realisation, the global perspective on energy conversion and consumption is shifting towards sustainable resources and technologies. This has resulted in huge increases in the installation of renewable energy all over the world, with wind energy conversion being no exception. One of the major reasons for strong growth in wind energy conversion systems is the improvement in wind turbine technologies that have resulted in lower costs.

In dealing with any wind energy project, accurate technical knowledge of wind regime characteristics and wind turbine technology is a pre-requisite for the efficient planning and implementation of any wind energy project. Due to wind speed variability, a wind turbine seldom operates at its rated output. Therefore, the Capacity Factor ($CF$) of a turbine is usually used to estimate its average energy production, which in turn can be used for the economic evaluation of wind power projects at potential sites. Several researchers have carried out energy analyses by varying different parameters affecting the energy output. These parameters could be those associated with the wind speed distribution or with the turbine technology. To get some insight into the energy research carried out, a few of the many available wind assessment studies are discussed here.

Abed et al [119], Salameh et al [120] and Jangamshetti et al [121] carried out some research on turbine site matching purely based on the capacity factor. On the other hand, Albadi and Saadany [122] in their research presented a new formulation for the turbine-site matching problem, based on wind speed characteristics at any site, the power performance curve parameters of any pitch-regulated wind turbine, as well as turbine size and tower heights.

Several researchers have attempted to evaluate the wind energy potential of different regions by using various probability distribution functions. Most of these researchers have indicated that the Weibull wind speed distribution is accurate enough for wind energy estimation. Based on the two-parameter Weibull wind speed distribution, Jamil et al [123] developed an evaluation method to estimate wind energy density and other wind characteristics in Iran. Rosen et al. [124] used two kinds of the Weibull distribution to analyze wind energy potential.
of two windy sites located in the coastal region of the Red Sea in Eritrea. Similarly, Islam et al [125] carried out an assessment of wind energy potentiality at Kudat and Labuan, in Malaysia. Li [126] and Lu et al [127] conducted mathematical investigations using the two-parameter Weibull wind speed distribution to examine wind power potential and wind turbine characteristics in Hong Kong. Mathew et al [128] presented an analytical approach in their study, in order to come up with a method for characterizing wind regimes, bringing out their energy potential, i.e. they developed an expression to compute the energy density, energy available in the wind spectra in a time period and the energy received by turbine using the Rayleigh wind speed distribution. A method to identify the most frequent wind speed and velocity that carries maximum amount of energy with it, and the effect of cut-in and cut-out wind speed on the turbine performance were also discussed. Bustamante et al [129] evaluated the influence of the Weibull fitting in monthly wind energy estimation for five wind farms in Spain. Similarly, Chang et al [130] carried out an assessment of wind and wind turbine characteristics based on Weibull parameters for Taiwan. Ullah et al [131] carried out wind energy potential at Kati Bandar, Pakistan. Rehman et al [132, 133] investigated the wind energy potential for coastal locations of the Kingdom of Saudi Arabia. Ouammi et al [134] carried out wind energy potential in Liguria region, Turkey. Al-Yahyai et al [135] carried out an assessment of wind energy potential locations in Oman. Similarly an assessment of wind energy potential in Gaza Strip was carried out by Alyadi et al [136] and Akpinar et al [137] carried out an assessment on seasonal analysis of wind energy characteristics and wind turbine characteristics. These are some of the wind resource assessment carried out by several researchers. From all this research, it is evident that the wind energy output of wind turbines fluctuates due to the stochastic nature of the wind speed. This fluctuation in the wind speed could to some extent be compensated for by the telescopic blade concept where the blades would extend when the wind speed drops in order to maintain the power output at its rated capacity thus enhancing the capacity factor. Any increase in energy output would reduce the per kWh cost of the energy produced.

Several researchers have studied different control strategies to enhance the energy output of a wind energy conversion system. To mention a few, Tan and Islam [138] came up with a control strategy based on mechanical sensor-less control to optimise the energy conversion of a permanent-magnet synchronous generator. Chedid et al [139] and Mohamed et al [140] developed a maximum power point tracking (MPPT) control strategy which enabled the selection of the optimal MPPT for each WECS project. Recently, Narayana et al [141]
proposed a generic maximum power point tracking controller for small-scale wind turbines in order to enhance their energy output. Muljadi et al [142] came up with a control strategy for a variable speed stall regulated wind turbine. In their investigation, the turbine was controlled to operate near maximum efficiency (energy capture) in low and moderate wind speeds, while at high wind speeds, the turbine was controlled to limit its rotational speed and output power which was accomplished by forcing the rotor into an aerodynamically stalled condition.

Hall and Chen [143] carried out their research on the performance of a 100 kW wind turbine with a Variable Ratio Gearbox (VRG) which is a device which can be used in the drive train of a small to medium-size wind turbine to improve its aerodynamic efficiency. A VRG allows wind turbines, with constant-speed generators, to discretely vary rotor speed and to achieve greater aerodynamic efficiency. In their research the authors proposed a method to identify turbine sites that provide the VRG with the greatest opportunities to increase production. Their overall findings suggest that the VRG can benefit all wind turbines, irrespective of wind class.

The studies carried out so far that are available in the literature in terms of energy analysis of a wind energy conversion system are mostly for the standard blade wind turbine. The energy analysis carried out in the present research is aimed to show the impact of various parameters such as blade extension, chord ratio and the Weibull shape and scale factors that could affect the energy output of a telescopic blade wind energy conversion system.

5.2. Wind speed and the probability density function
To analyse the energy output for any Wind Turbine Conversion System (WTCS), it is very important to first analyse the wind data. Wind is stochastic in nature and for this study the probability density function (pdf) used was the Weibull distribution. Figure 5.1 shows the average annual wind speed variations from 1960 to 2007, as well as the mean wind speed for the entire period of 48 years for a site in Wellington, New Zealand [144]. From this figure it is evident that the annual energy output would fluctuate a lot, and if there were a telescopic blade system in place, it could compensate favourably for these fluctuations.
Once the mean and the standard deviation of the wind speed are known, the Weibull parameters can be obtained.

5.2.1. Probability density function
In practice, three basic methods are used in wind energy assessments: (i) statistical analysis of the existing wind energy potential, with other meteorological data and topographical information; (ii) qualitative indicators of long term wind speed levels; and (iii) application of boundary layer similarity theory and the use of surface pressure observations [145]. There are several density functions that can be used to describe the wind speed frequency curve. Here, the Weibull distribution is used as it is widely used to describe the wind speed distribution. The probability density function (pdf) $f(v)$ of wind speed $v$ can be described as [145, 146]:

$$f(v) = \left(\frac{k}{c}\right)\left(\frac{v}{c}\right)^{k-1} \exp\left[-\left(\frac{v}{c}\right)^k\right] \quad (k > 0, v > 0, c > 1)$$  \hspace{1cm} (5.1)

where $k$ is a shape factor and $c$ is a scale parameter.

The two Weibull parameters ($k$ and $c$) and the average wind speed $\bar{v}$ are related by

$$\bar{v} = c \Gamma \left(1 + \frac{1}{k}\right)$$ \hspace{1cm} (5.2)

where $\Gamma (x)$ is the gamma function, equation (5.3)

$$\Gamma (x) = \int_0^\infty e^{-t} t^{(x)-1} \, dt$$ \hspace{1cm} (5.3)
The gamma function can be approximated by the so-called Stirling approximation \([147]\) in equation (5.4).

\[
\Gamma (x) = \sqrt{2\pi x} \left( x^{x-1} (e^{-x}) \left( 1 + \frac{1}{12x} + \frac{1}{288x^2} - \frac{139}{51840x^3} + \ldots \right) \right)
\]  

(5.4)

The standard deviation of the wind speed variations is given by equation (5.5).

\[
\sigma = c \left( \Gamma \left( 1 + \frac{2}{k} \right) - \Gamma^2 \left( 1 + \frac{1}{k} \right) \right)^{1/2}
\]

(5.5)

Once the mean and standard deviation of the wind speed are known, the following approximations can be used to calculate the Weibull parameters \(c\) and \(k\) \([146]\).

\[
k = \left( \frac{\sigma}{\bar{v}} \right)^{-1.086} \quad (k \leq 1 \leq 10)
\]  

(5.6)

\[
c = \frac{\bar{v}}{\Gamma(1+1/k)}
\]

(5.7)

where \((Mean\ Speed)\)

\[\bar{v} = \frac{1}{n} \sum_{i=1}^{n} v_i\]

(5.8)

and \((variance)\)

\[\sigma^2 = \frac{1}{n-1} \sum_{i=1}^{n} (v_i - \bar{v})^2\]

(5.9)

and the mean wind energy density as a function of Weibull parameters is given by equation (5.10).

\[
\bar{p} = \frac{1}{2} \rho c^3 \Gamma \left( 1 + \frac{3}{k} \right)
\]

(5.10)

The average power in the wind, \(\bar{P}_w\), over the swept area \(A = \pi d^2/4\) of the turbine can be expressed as \([127, 146]\):

\[
\bar{P}_w = \frac{1}{2} \rho A \int_{0}^{\infty} v^3 f(v)dv
\]

(5.11)

If \(f(v)\) is the Weibull density function, the average power in wind becomes:

\[
\bar{P}_w = \frac{\rho A \bar{v}^3 \Gamma(1+3/k)}{2[\Gamma(1+1/k)]^3}
\]

(5.12)

For comparison, typical Weibull distributions with different shape and scale parameters are shown in Figure 5.2. The probability distribution function shown in Figure 5.2 (a) is for various Wind Classes with the scale parameter \(k = 2\) which is a special case of Weibull
distribution, known as the Rayleigh distribution. For a Wind Class 2 site with \( k = 2 \), there are more low speed wind hours indicated by the \( f(v) \) plot when compared with a Wind Class 7 site for example. Similarly, Figure 5.2 (b) shows a Class 4 wind site with different scale parameters from \( k = 1.5 \) – 2.5. From this we can see that a site with low \( k \) value has more low speed wind hours than a site with high \( k \) value for the same wind class.

\[ f(v) \]

\[ k=2, \text{ varying Wind Class} \quad \text{(b) Wind Class 4, varying } k \]

\[ P_t(v) = C_p \frac{1}{2} \rho A v^3 \] (5.13)

where \( C_p \) is the power coefficient. The average power produced by the wind turbine \( P_{t,ave}(v) \) is given by:

\[ P_{t,ave} = \int_0^\infty P_t(v) f(v) dv \] (5.14)

The generated electrical \( P_e \) power is expressed in terms of generator efficiency \( \eta \) that accounts for losses in power conversion (mechanical, electrical, etc),

\[ P_e = \eta P_t \quad \text{or} \quad P_e = \eta C_p \frac{1}{2} \rho A v^3 \] (5.15)
Different wind generators have different power output performance curves, so the model used to describe the performance is also different, and in most published literature [127, 137, 148], the following equation is used to simulate the electrical power output, $P_e$, of a model wind turbine:

$$
P_e(v) = \begin{cases} 
0 & (v < v_c) \\
\frac{P_{eR} v_k}{v_k - v_c} & (v_c \leq v \leq v_R) \\
P_{eR} & (v_R \leq v \leq v_F) \\
0 & (v > v_F) 
\end{cases}$$

where $P_{eR}$ is the rated electrical power, $v_c$ is the cut-in wind speed, $v_R$ is the rated wind speed and $v_F$ is the cut-out wind speed. The model is simplified when the wind speed is higher than rated speed.

The annual energy output can be estimated using the Weibull function and the above model. The capacity factor can also be computed. It is defined as the ratio of average energy output over a time period to the rated energy output,

$$
\text{Capacity Factor} = \frac{\text{Actual Amount of Energy Produced over a time period}}{\text{Energy that would have been produced if the turbine operated at maximum output 100\% of the time}}
$$

(5.17)
5.2.2.1. The reference wind turbine
The reference wind turbine with fixed length blades considered in the study was the Bergey XL1 rotor with a 2.5m rotor diameter; see Figure 5.4 [149]. It has a power rating of 1000 W at a wind speed of 11 m/s. Its blades are made of pultruded fibreglass and have fixed pitch and a constant chord of 100 mm. The NACA 0018 airfoil section (Sandia Laboratories [150]) was assumed since Bergey XL1 uses a patented SH3045 airfoil, data for which is not available. The optimum fixed blade pitch for this rotor was first determined by simulating the results for different blade pitch angles as shown in Figure 5.5. The pitch angle which gave maximum power coefficient for all blade extensions was chosen.

$$\beta_{optimum} = 0.7° \text{ and } C_{p(max)} = 0.375$$

For the energy analysis carried out in this research, the inertial effects were neglected. A generator efficiency of 0.65 was used. This was calculated as follows: Since $$P_{rated} = 1000 \text{ W}$$ at $$U_{rated} = 11 \text{ m/s}$$, then using the already computed $$C_{p(max)}$$ gives,

$$\eta_{rated} = 0.65$$

The analysis also assumed optimal generator operation, meaning a constant tip-speed ratio at all wind speeds, which is typical of permanent magnet generators connected to wind turbines. Thus the rotational speed of the rotor varies according to: $$\Omega = \lambda_{rated} \frac{U}{R}$$. Further assumptions included: (a) zero power output below the cut in wind speed of 2.5 m/s; and (b) constant power output above a wind speed of 13 m/s.

<table>
<thead>
<tr>
<th>Type</th>
<th>3 Blade Upwind</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor Diameter</td>
<td>2.5 m – zero extension</td>
</tr>
<tr>
<td>Start up Wind Speed</td>
<td>3 m/s</td>
</tr>
<tr>
<td>Cut in Wind Speed</td>
<td>2.5 m/s</td>
</tr>
<tr>
<td>Rated Wind Speed</td>
<td>11 m/s</td>
</tr>
<tr>
<td>Rated Power</td>
<td>1000 W</td>
</tr>
<tr>
<td>Maximum Power</td>
<td>1300 W</td>
</tr>
<tr>
<td>Blade Pitch Control</td>
<td>Fixed Pitch</td>
</tr>
<tr>
<td>Generator</td>
<td>Permanent Magnet</td>
</tr>
<tr>
<td>Cut out wind speed</td>
<td>None</td>
</tr>
<tr>
<td>Root blade chord</td>
<td>100 mm</td>
</tr>
<tr>
<td>Tip blade chord</td>
<td>60 and 40 mm for CR = 0.6 and CR = 0.4</td>
</tr>
</tbody>
</table>

**Figure 5.4 Bergey XL1 [149] wind turbine details**
Figure 5.5 Power coefficient for different pitch angles and blade extensions at a rotor speed of 500rpm.

The telescopic blade wind turbine with different blade extensions was analysed based on the reference wind turbine. In the energy analysis, 4 different tip blade lengths, equivalent to 0, 1.3, 2.5 and 5 root blade tip chords ($C_2 = 100$ mm) as shown in Table 5.1 (i.e. ~0, 10, 20, 30 and 40% of the root blade lengths) were considered.

**Table 5.1 Blade length and extension**

<table>
<thead>
<tr>
<th>Blade Length (m)</th>
<th>% Extension (Extension Ratio)</th>
<th>Extension based on number of Root blade tip chord ($C_2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.25</td>
<td>0 (Zero extension)</td>
<td>0</td>
</tr>
<tr>
<td>1.38</td>
<td>~10%</td>
<td>1.3* $C_2$</td>
</tr>
<tr>
<td>1.5</td>
<td>20%</td>
<td>2.5* $C_2$</td>
</tr>
<tr>
<td>1.625</td>
<td>30%</td>
<td>3.75* $C_2$</td>
</tr>
<tr>
<td>1.75</td>
<td>40%</td>
<td>5* $C_2$</td>
</tr>
</tbody>
</table>
5.3. Telescoping blade system and control strategy
There are several different telescopic blade systems and control strategies that could have been used for energy output analysis. In this research, the control strategy adopted by Sharma et al [53, 54] has been used for controlling the rotor actuation at various wind speeds, as explained below.

5.3.1. Control strategy
The telescopic blade concept involves multiple blade sections that extend out when the wind speed drops below the rated level. The simplest form would be a two-stage telescopic blade which consists of two sections capable of extending to almost twice the original rotor radius when fully extended; see Figure 5.6. With telescopic blades, the rotor diameter or radius is controlled according to the wind speed level, especially when the speeds fall below the rated level. In this study, a simple but nevertheless realistic criterion developed by Sharma et al. [53, 54] based on limiting the blade root bending moment was used to control the blade length. As a first approximation, the thrust force on the blade was assumed to act at the midpoint of each blade section between its ends. Consequently, the blade root bending moment at the rated wind speed is given by

\[
M_{\text{rated}} = C_T \frac{1}{2} \rho U_{\text{rated}}^2 \left( C_2 (R_{\text{rated}} - R_R) \right) \times \frac{1}{2} (R_{\text{rated}} + R_R)
\]

(5.18)

When the second stage is extended at a wind speed \( U < U_{\text{rated}} \), then the blade root bending moment becomes

\[
M = C_T \frac{1}{2} \rho U^2 \left[ C_2 (R_{\text{rated}} - R_R) \times \frac{1}{2} (R_{\text{rated}} + R_R) + C_1 (R - R_{\text{rated}}) \times \frac{1}{2} (R + R_{\text{rated}}) \right]
\]

(5.19)

The maximum second stage blade extension \( R \) can be found by equating the two equations i.e. when the blade root bending moment at wind speeds below rated is maintained at the rated level. Hence a relationship between rotor radius \( R \) and the wind speed \( U \) is obtained,

\[
\frac{R}{R_{\text{rated}}} = \sqrt{\frac{1 - (R_R/R_{\text{rated}})^2}{(C_1/C_2)(U/U_{\text{rated}})^2} - \frac{1 - (R_R/R_{\text{rated}})^2 - (C_1/C_2)}{(C_1/C_2)}}
\]

(5.20)

If the chord length of the second stage is close to that of the first i.e. \( \frac{c_1}{c_2} \approx 1 \) and if \( \frac{R_R}{R_{\text{rated}}} \ll 1 \), then equation (5.20) simplifies to:
Various blade extensions and chord ratios for different wind speeds are possible. Figure 5.7 shows different possible blade extensions at a range of wind speeds for $\frac{R_R}{R_{rated}} = 0.15$. This suggests that if a two-stage blade system that is capable of approximately doubling the blade radius is to be used, then the maximum advantage can only be extracted to about half the rated wind speed for $\frac{c_1}{c_2} \approx 1$. For example, if $R_{rated} = 1.25$ m and $U_{rated} = 11$ m/s, then a variable length blade system would involve second stage actuation between radii of 1.25 to 2.5 m maximum from wind speeds of 11 m/s down to 5.5 m/s. Below 5.5 m/s, the theoretical total blade radius would remain at 2.5 m. For the energy analysis carried out in this research, the second stage blade was assumed to have a maximum radius of 1.75 m, then a variable length blade system would involve second stage actuation between radii of 1.25 to 1.75 m for wind speeds of 11 m/s to 7.9 m/s for $\frac{c_1}{c_2} \approx 1$. For chord ratio of 0.6 and radii of 1.25 to 1.75 m correspond to wind speeds of 11 m/s to ~8.8 m/s. Below 8.8 m/s, the total blade radius would remain at 1.75 m.
5.3.2. Telescopic blade and generator controller

For this research, a generic maximum power point tracking (MPPT) controller proposed by Narayana et al [141] is assumed. In small-scale applications, variable-speed fixed-pitch (VSFP) wind energy conversion systems (WECSs) are generally more efficient compared to fixed speed counterparts, and hence are becoming increasingly popular. Wind turbines with variable-pitch control are generally costly and complex. Typically, variable-speed wind turbines are electrically controlled, usually by using power electronics, to regulate the torque and speed of the turbine in order to maximize the output power. Therefore, VSFP approach is becoming more popular for low cost construction, and is the most common scheme for small wind turbines. In this scheme, a Maximum Power Point Tracker (MPPT) is used to control the restoring torque of the electrical generator for optimum system operation [142, 151]. Accordingly, the performance of a VSFP wind turbine could be optimized without the need for a complex aerodynamic control. The maximum output power from the turbine is usually obtained by controlling the system such that the relevant points of wind rotor curve and electrical generator operating characteristic coincide. In order to achieve this, it is necessary to drive the turbine at optimal rotor speeds for a particular wind speed profile. Techniques that employ wind sensors generally perform well with wind speed variations, as the control system responds to variation in wind conditions [152]. However, in practice it is difficult to accurately measure the wind speed by an anemometer installed close to the wind turbine, as it is exposed to different speeds due to wake rotation. Generally the wind rotor blades experience coning and flapping effects due to wind forces. The wind speed varies across the
swept area of the wind rotor as a result of the yaw behaviour, wind shear, and so forth. Accordingly, wind rotor dynamics vary and become very difficult to predict in real systems. Therefore, it would be useful to implement a sensor-less control strategy, which is proposed by Narayana et al [141], for small-scale wind turbine systems that operate without predetermined turbine characteristics.

The wind rotor aerodynamic characteristics are usually represented by the $C_p$ vs $\lambda$ relationship, as shown in Figure 5.8. Note that $C_p$ is maximum at a certain value of tip speed ratio $\lambda_{optimal}$ (optimal ratio) which results in a maximum power extraction. For variable-speed WECSs, when the wind speed varies, the rotor speed should be adjusted to follow the optimum operating point for maximum power generation.

The aerodynamic torque of a wind rotor is also a function of wind speed ($v$) and rotational speed ($\omega$) of the rotor. The aerodynamic torque of a wind turbine rotor can be obtained as:

$$T_e = \frac{1}{2} \rho \pi R^3 Av^2 \frac{C_p}{\lambda}$$

or

$$T_e = \frac{1}{2} \rho \pi R^3 Av^2 C_t$$

where $C_t = \frac{C_p}{\lambda}$ is the torque coefficient and is also plotted in Figure 5.8. The maximum $C_p$ and $C_t$ occur at different tip speed ratios ($\lambda$).

Figure 5.8 Wind turbine blade characteristics (zero extension)
Generally the electromagnetic torque of a generator can be varied by controlling the permanent magnet generator current, and the rotational speed of a wind turbine rotor can be varied by controlling the generator load. As discussed by Narayana et al [141], the inherent wind rotor characteristics shown in Figure 5.9 indicate that maximum torque is delivered by the wind rotor at a lower rotational speed compared to that for maximum aerodynamic power. Also, the PMG loss varies with the level of torque transmitted through the system. Therefore, optimum operation does not occur at the maximum points of the wind rotor aerodynamic power curves, but at the maximum points of the electrical output power curve, as shown in Figure 5.9. Consequently in order to obtain maximum power output, the restoring power curve of the generator as shown in Figure 5.9 needs to be adjusted to tally with the actual maximum operating point of the PMG output power curve. MPPT controllers usually employ wind speed sensors (anemometers) to provide a feedback reference signal to the MPPT controller to set the turbine speed. MPPT control may also be implemented using a sensor-less control, where the generator output frequency and power (or torque) mapping techniques are used to track the MPP [138]. Detailed information regarding a generic MPPT controller can be found in Narayana et al [141].

![Figure 5.9 Operating point of wind power system [141]](image)

Thus by using a generic MPPT controller, the wind turbine system always operates at optimum, and thus the AEO of the wind turbine system becomes:
The analysis assumes optimal wind turbine rotor and generator operation for every blade extension. For the generator, as discussed earlier, a constant efficiency of 0.65 was assumed.

For the wind turbine rotor analysis, the simulations were carried out for different blade extensions. Figure 5.10 (a) to (e) show the rotor performance obtained for different rotor rpm and wind speeds for the blade with CR = 0.6. In the energy analysis carried out in this research, it is assumed that the MPPT controller tracks the rotor performance at its optimum power curve as shown in Figure 5.10 (a) in order to capture maximum power out of the turbine rotor. The simulations for different chord ratios are supplied in Appendix B.

\[
AEO = \sum_{t=1}^{8760} \eta_{opt} C_{p, opt} \frac{1}{2} \rho A v^3 \delta t
\]  

(5.24)

Figure 5.10 TBWT power curves for different blade extensions at different rotor speeds
Another important parameter to consider is the tip speed ratio ($\lambda$). Every blade configuration has its own performance characteristics at different rotor speeds. Figure 5.11 (a) to (e) shows the $C_p$ vs $\lambda$ curves for different blade extensions with $CR = 0.6$. For the blade with no extension as shown in Figure 5.11(a), the tip speed ratio at optimum $C_p$ is $\sim 6$. Similarly, for 10, 20, 30 and 40% extended blades, the tip speed ratios at optimum $C_p$ are $\sim 6.6$, 6.9, 7.3 and 7.7 respectively. Figure 5.11 shows that every blade extension has a different tip speed ratio for optimum $C_p$. Therefore, the controller has to adjust the rotor rotational speed (Equation 5.25) at different wind speed for various blade extensions in order to maintain the optimum $C_p$, which is why MPPT controllers were chosen for this research. Similar trends are exhibited for $CR = 0.4$ and $CR = 1.0$, and are given in Appendix C.
(a) Zero extension

(b) 10% extension

(c) 20% extension

(d) 30% extension

(e) 40% extension

Figure 5.11 $C_p$ curves for different blade extensions ($CR = 0.6$)
5.3.3. Blade actuation and power curves

The telescopic blade actuations were modelled on the control strategy described above. Figure 5.12(a) to (e) shows the power outputs obtained for different maximum extensions with blades having a chord ratio (CR) of 0.6. The power curves were generated using the software WT Perf. For example, if a design is for a blade system with a maximum of 40% extension, then the adopted control strategy leads to the power curve shown in Figure 5.12 (e). The rated wind speed is 11m/s for the wind turbine used in this study, so for any wind speeds beyond the rated wind speed, the blades are fully retracted (i.e. zero extension). When the wind speeds drop below rated, the blades start to extend, up to their maximum possible extension.

Figure 5.12 (b) to (e) show two different actuation methodologies available i.e. continuous or step actuation. In a continuous actuation, the tip blade is actuated instantaneously when the wind speed fluctuates. However, it is assumed that this instantaneous actuation could induce unnecessary dynamic loading and excessive movements could create excessive wear of blade and more energy required for actuation than practical realisation of the system. On the other hand, step actuation actuates the blade in steps. For example, when the wind speed drops to a certain level based on the strategy described above, the controller then actuates. Therefore, in this research, the step type blade actuation is assumed as it simplifies the energy output analysis and is more appropriate as it reduces the tip blade frequency of actuation.

With the step-type actuation and the strategy used for energy analysis in this research, the telescopic blade will be fully retracted (i.e. no extension: blade length of 1.25m) up to the wind speeds of 10.4m/s. When the wind speed drops below 10.4m/s, the blade would actuate to 10% extension (blade length of 1.38m). Similarly, the blades would actuate to 20% (blade length of 1.5m), 30% (blade length of 1.625m) and 40% (blade length of 1.75m) extensions when the wind speed drops to 9.8, 9.3 and 8.8m/s respectively for CR = 0.6. For other chord ratios, refer to Appendix D. It is worth noting that if a different strategy is used based purely on power output or wind speed or perhaps continuous actuation instead of step actuation, the overall power output improves further. For example, as soon as the power output of the turbine drops below rated level, the system could start extending the blade up to its maximum length if necessary, in order to maintain the rated power. Whereas in the present study, it is only when the wind speed drops to certain level, that the next extension is done and vice-versa. The continuous blade actuation analysis is beyond the scope of this research.
Energy Analysis

Figure 5.12 Power curves with blade actuation ($CR = 0.6$)
5.4. Power curves and power coefficient

Several telescopic blade wind turbine simulations were carried out using WT Perf with different blade lengths and chord ratios. Simulations were carried out for every configuration with tip and hub losses and with and without the step loss. These were done to quantify the loss contributed by the introduction of the step change in the blade chord of a telescopic blade wind turbine.

5.4.1. Effect of tip length on the power curves with step corrections

The simulations were carried out based on the reference turbine described earlier. Generally as anticipated, the power output of the wind turbine with telescopic blade increases with increase in blade length as shown in Figure 5.13. For every blade extension, the power output in region 2 (between cut-in and rated wind speed) increased with the blade extension. This increase in region 2 will not increase the maximum system power output but the overall energy output of the system, in other words it will enhance the system capacity factor.
5.4.2. Effect of chord ratio on the power curves with step corrections

Similar to the blade extensions, the power output of telescopic blade wind turbine with different chord ratios and blade extensions were simulated to gain an insight into the effect of chord ratio. It is shown in Figure 5.14 that for all blade extensions, the smaller chord ratios have a detrimental effect on power output. For a 10% extended blade, as shown in Figure 5.14 (a), the power output of the blade with chord ratio of 0.4 is somewhat similar to that of blade with no extension. The 10% extension blade had a tip length of 1.3 times the chord of root blade, which had the highest losses therefore lowest power output when compared with other blade extensions associated with the step change region. These show that the extended blade turbine performance improves with increase in chord ratio, and the benefits of the
extended blade idea are reduced if the chord ratio is low. As shown in Figure 5.14 (a) to (d), the power output increased with the increase in blade length and for individual blade extension, it also increased with the increase in chord ratio. As shown in Figure 5.14 (d), for a turbine blade with 40% extension, the power output for chord ratios (CR) 1.0, 0.6, 0.4 and 0 are ~ 1000, 880, 710 and 430 Watts respectively at wind speeds of ~ 8.6 m/s. It is evident from this that the blades with CR = 1.0, 0.6 and 0.4 produce 130%, 104% and 65% more power when compared with no extension blade respectively. This shows that the telescopic blade turbine concept would harvest more energy below its rated wind speed when compared with a no extension blade.

![Graphs showing effect of chord ratio on power output at different wind speeds](image)

(a) 10% blade extension  
(b) 20% blade extension  
(c) 30% blade extension  
(d) 40% blade extension

Figure 5.14 Effect of chord ratio on power output at different wind speeds
5.4.3. Effect of step losses on the power curves
Figure 5.15 shows that, the power output of a turbine with extended blade reduces with the introduction of losses associated with the blades. It is evident from Figure 5.15 (a) to (d), that the step loss has a lot of influence on blades with lower tip blade lengths, as the power output for the blade with 10% extension is more or less the same as that with no extension. But as the tip blade length increases, Figure 5.15 (b) to (d) show that the power output also increases with increasing blade extension.

![Power curves as a function of wind speed with and without step losses (CR = 0.4)](image)

5.4.4. Effect of blade extension on the product of power coefficient and swept area
Figure 5.14, shows that it is not very clear as to what really is the effect of blade extension on the power output of a telescopic blade system. In order to get more insight into the effect of blade extension, the power coefficient is multiplied by the swept area ($C_p \times A$) for individual blade extension, and are shown in Figure 5.16 (a) to (c). It is clear that the $C_p \times A$ for every
blade configuration increases with increase in tip blade lengths. It is also evident that for \( CR = 0.4 \), there is reduction in \( C_p^* A \), and for \( CR = 1.0 \) and 0.6, the \( C_p^* A \) is very similar. These curves confirm that any blade extension would definitely increase the power output of a wind turbine; this is primarily due to increase in the swept area, and this is an important and significant findings of this research.

Figure 5.16 Effect of blade extension on the product of power coefficient and area at different chord ratios and at 500rpm rotor speed
5.4.5. Effect of chord ratio on the product of power coefficient and swept area

Figure 5.17 (a) to (d) shows the product of power coefficient and swept area $C_p*A$ for different chord ratios for different blade extensions as a function of tip speed ratio. For the blade with 10% extension, the $C_p*A$ is not increased significantly relative to the zero extension blade. But for extensions of 20 to 40%, the increase in $C_p*A$ is pronounced for all chord ratios. There is a clear trend of $C_p*A$ increasing with increasing blade lengths. For all blade extensions, the blade with $CR = 0.4$ has the lowest $C_p*A$.

![Graphs showing $C_p*A$ vs Tip speed ratio for different chord ratios and blade extensions.](image)
5.5. Annual Energy Output
Annual Energy Output (AEO) analysis for different blade extensions, chord ratios, wind class and Weibull shape factors were carried out. The discussion is mostly based around the increment in the AEO when telescopic blades are used compared with blades with zero extension.

5.5.1. The effect of blade extensions on annual energy output
Annual energy output calculations for blades with $CR = 0.6$ and different blade extensions were carried out in order to elucidate the effect of blade extension on the AEO. In this case, the shape factor ($k$) of 2 was chosen for Wind Class 2. It is apparent from Figure 5.18 that the AEO and thus the capacity factor, increase with blade extension. Figure 5.19 demonstrates the percentage increase in the AEO. It has a fairly linear trend upwards for blade extensions 0 to 40%. For the blade with 40% extension, as shown in Figure 5.19 and Table 5.2, the increase in the AEO is the highest at 48% when compared with other blade extensions. For blade extensions of 10, 20 and 30%, the percentage increase in AEO is 10, 23 and 33% respectively. This demonstrates that 2 wind turbine systems with telescopic blades of 40% blade extension, in a wind regime with $k = 2$ and Wind Class 2, could produce the same amount of energy as that from 3 standard blade wind turbines. This definitely would reduce the initial capital investment thus making wind energy conversion system more attractive for investors. On the other hand, the same generator would produce more energy (increased capacity factor), thus the price per unit kilowatt hour reduces.

![Figure 5.18 Annual energy output and capacity factor for different blade lengths ($CR 0.6$), $k=2$, Wind Class 2](image-url)
5.5.2. The effect of chord ratio on annual energy output

The annual energy output for different chord ratios \((k=2\), Wind Class 2\) are shown in Figure 5.20 (a). It shows that the blade with chord ratio \((CR)\) 1.0 has the highest AEO followed by that with \(CR = 0.6\) and \(CR = 0.4\), as expected. For the blade with \(CR = 0.6\), the AEO is very close to that of \(CR = 1.0\). Figure 5.20 (b) shows the percentage increase in AEO for different chord ratios. The percentage increases in the AEO for a blade with 40% extension are 50, 48 and 27% for \(CR = 1.0, 0.6\) and 0.4 respectively. The blade with \(CR = 0.4\) has a huge impact on the AEO as shown in Figure 5.20 (b). For a blade with \(CR = 0.4\) and 10% blade extension, the percentage increase in the AEO is negligible. It is clear from this that as the blade extension and chord ratio increases, the capacity factor also increases.
5.5.3. The effect of shape and scale parameter on annual energy output

The Weibull parameters were varied to determine annual energy output over a range of wind characteristics; with shape factors from 1.5 to 2.5 over Wind Classes 2 to 7. The AEO increases, as anticipated, with the increase in wind class and chord ratio, as shown in Figure 5.21 (a) to (c). Figure 5.22 shows the percentage increase in the AEO and the capacity factor when $k=2.0$ and for three different chord ratios ($CR = 1.0$, 0.6 and 0.4) for various wind classes. These curves show that as the wind class increases, the relative increase in AEO
reduces for all chord ratios and blade extensions. On the other hand, with increase in Wind Class, the capacity factor increases for all chord ratios and blade extensions. For a blade with a 40% extension and \( CR = 1.0 \), the increase in AEO from Wind Class 2 to 7 is in the range of 50 to 16% respectively as shown in Figure 5.22 (a). Similarly, for a blade with a 40% extension and \( CR = 0.6 \), the increase in AEO from Wind Class 2 to 7 is in the range of 48 to 16% respectively as shown in Figure 5.22 (b). However, for a blade with 40% extension and \( CR = 0.4 \), the increase in AEO from Wind Class 2 to 7 is in the range of 28 to 11% respectively, as shown in Figure 5.22 (c). Similar trends are obtained for \( k = 1.5 \) and 2.5, for which full results can be found in Appendix E. From these figures, any design engineer could easily determine the level of increment in AEO or capacity factor, with different blade extensions for various chord ratios as a rule of thumb. As an example, consider a blade with \( CR = 0.6 \). If an engineer wants to design a wind energy conversion system for a Class 4 site with a scale parameter \( k = 2 \) as shown in Figure 5.22 (b), a rough estimate of the percentage increase in the annual energy out of the system can be obtained from such a design ‘chart’. In a feasibility study for a wind farm, this information could be used as a starting point for the design of a telescopic blade wind turbine system. Generally the trends show a larger increase in the energy output of the extended blade system in lower wind speed regimes, and this is of huge practical interest as there are many low wind speed sites.

![Figure 5.21 Annual energy output as a function of Wind Class with \( k = 2 \)](image-url)
Figure 5.21 Annual energy output as a function of Wind Class with $k = 2$ – cont’d

(a) $CR = 1.0$ (k=2)

Figure 5.22 Effect of Weibull scale parameter on annual energy output and capacity factor for different blade extensions
Similarly the effect of chord ratio on the percentage increase in AEO for different blade extensions for $k = 2.0$ and wind class 2 to 7 are shown in Figure 5.23. It is apparent from these graphs that the percentage increase in AEO is more pronounced in lower wind classes. As shown in Figure 5.23 (a), the blade with 10% extension and $CR = 0.4$ shows no major effect on AEO. The effect of chord ratio is more prominent on lower blade extensions and wind classes as shown in Figure 5.23 (a) and (b). As the blade extension increases and the wind class increases, as shown in Figure 5.23 (c) and (d), the percentage increase in the AEO
tails off with increase in wind class. For the blade with 40% extension, the effect of chord ratio beyond $CR = 0.6$ has very little effect for all wind classes. Similar trends are obtained for $k = 1.5$ and 2.5; see Appendix E.

Figure 5.23 Effect of Weibull scale parameter on annual energy output and capacity factor for different chord ratio $(k=2.0)$
Figure 5.23 Effect of Weibull scale parameter on annual energy output and capacity factor for different chord ratio ($k=2.0$) – cont’d

Figure 5.24 (a) to (d) shows the effect of shape factor on the percentage increase in AEO for $CR = 0.6$ and different blade extensions. These curves show that as the shape factor ($k$) increases, the percentage increase in AEO increases for all blade extensions. This is due to the fact that as the shape factor ($k$) increases, the peak on the probability density functions
shifts to the right as already shown in Figure 5.2. With this shift to the right, there are higher velocity winds distributed over the period with higher shape factor. Also due to higher shape factor, there is less wind speed distribution below cut-in speeds therefore higher energy is produced with higher shape factors. With the increases in blade extension as shown in Figure 5.24, the percentage increase in AEO is significant at lower wind class. For a blade with 10% extension, and Wind Class 2, the percentage increase in AEO is 8, 11 and 13% for $k = 1.5$, 2.0 and 2.5 respectively. On the other hand, for a blade with 40% extension, and Wind Class 2, the percentage increase in AEO is 35, 48 and 62% for $k = 1.5$, 2.0 and 2.5 respectively. As the wind class increases, the percentage increase in AEO decreases for all shape factors.

Figure 5.24 Effect of Weibull shape factor on annual energy output and capacity factor for different blade extensions (CR = 0.6) as a function of Wind Class
5.6. The effect of step loss on annual energy output
Any reduction in the wind turbine rotor performance has an adverse effect on the amount of energy a wind energy conversion system can produce. In the case of a telescopic blade wind turbine, due to sudden change in the chord of the tip blade, the performance of the wind turbine rotor decreases, thus the amount of energy that can be produced is reduced. The effects of blade extension, chord ratio and the Weibull parameters on step loss are discussed in detail in the following sections. The energy output analysis of the telescopic blade were carried out with tip, hub and step losses. In order to quantify the influence of the step loss on the AEO, the simulations were carried out with step, tip and hub losses and then subtracted from the energy output with tip and hub losses. The difference is then the energy lost due to step loss.

5.6.1. The effect of blade length, chord ratio and Weibull parameters on step loss
Figure 5.25 (a) shows that as the blade extension increases, the reduction in annual energy output due to step loss reduces which indicates that the step loss becomes less significant as the blade extension increases. For Wind Class 2, the reduction in annual energy output due to step loss is ~12, 8, 5 and 4% for 10, 20, 30 and 40% blade extension respectively. On the other hand, for wind class 6, the reduction in annual energy output due to step loss is ~6, 4, 3 and 2.5% for 10, 20, 30 and 40% blade extension respectively. Therefore, as the Wind Class increases, the reduction in annual energy output due to step loss decreases for all blade extensions. On the other hand, Figure 5.25 (b) shows the effects of chord ratio ($CR = 0.4$ and 0.6) in terms of the reduction in annual energy output due to step loss. The blade with lower chord ratio contributes higher losses as anticipated, which are reflected in the reduction in annual energy output. For a wind class 2, 10% blade extension, the reduction in annual energy output due to step loss for $CR = 0.4$ and 0.6 are ~12 and 8% respectively. Similar trends are obtained for different blade extensions. In their research, McCoy et al. [55] made an assumption of 4% reduction in the energy output due to step loss and energy used for blade actuation of the extended blade turbine system for both class 4 and 6 wind sites. As expected, from Figure 5.25 (b) shows that the reduction in annual energy output is less for blade with $CR = 0.6$ when compared with $CR = 0.4$. 
To get an insight into the effect of the shape factor on the reduction in annual energy output due to step loss, energy analysis were carried out for different blade extensions, Wind Classes and Weibull shape factors. It is evident from Figure 5.26 (a) to (c) that with increase in Wind Class, the reduction in annual energy output due to the step loss reduces for all shape factors. The effect of shape factor is higher for lower blade extensions and wind class. The reduction in annual energy output due to step change for 10% extension blade, Wind Class 2, are ~15, 12 and 9% for \( k = 2.5 \), 2.0 and 1.5 respectively and for the same blade length, wind class 6, the reduction in annual energy output due to step change for \( k = 2.5 \), 2.0 and 1.5 are around 6%.
Figure 5.26 Effect of Weibull shape factor on step loss for different Wind Classes (CR = 0.4) as a function of blade extension

For the range of shape factors, chord ratio and blade extensions studied in this research, the reduction in annual energy output due to the step change in blade chord is in the range of 15 to 1% as shown in Figure 5.27 for wind class 2 to 7, respectively. In reality, the chord ratio is not expected to be below 0.4 due to structural strength reasons. Therefore as shown in Figure 5.27, the maximum loss in annual energy production due to the step loss would be ~15% for the worst case scenario.
5.7. Summary

The energy output analyses for telescopic blade wind turbines were carried out for different blade extensions, chord ratios and Weibull parameters. It is shown from the analysis that as the blade extension increases, the power output increases and so does the annual energy output. The annual energy output could be up to ~48% higher for a blade with 40% extension with a chord ratio of 0.6 in a Wind Class 2 site.

It is also apparent that the chord ratio has a significant impact on the annual energy output of a system. For wind classes 2 to 7, and \( k = 2 \), the increase in energy output of the extended blade wind energy system with 40% extension, for chord ratios of 0.4 – 1.0 are in the range of 11 – 51% respectively. On the other hand, for a blade extension of 10%, \( CR 0.4, k = 2 \), the percentage increase in the annual energy output is basically negligible.

For the range of shape factors, chord ratio and blade extensions studied in this research, the reduction in annual energy output due to step change in blade chord is in the range of 15 to 1% for wind class 2 to 7 respectively.

Wind speed distribution as described by the Weibull scale factor \( k \) also has an impact on the energy output of telescopic blade wind turbines. As \( k \) increases, the energy output also increases for all Wind Classes, but the increases are more significant in low Wind Class sites. This is important because there are more low wind speed sites available for wind farms. The
increases in energy output are in the range of 16-62% for Wind Classes 7 -2 respectively for a blade extension of 40%, \( k = 2.5 \) and \( CR \, 0.6 \). For wind class 2, blade extension of 40% and \( CR \, 0.6 \), the percentage increase in annual energy output for \( k = 1.5 \), 2.0 and 2.5 are 35, 48 and 62% respectively.
Chapter 6. Conclusions and recommendations

This research was divided into several different sections. Experimental, computational and analytical studies were carried out for a Telescopic Blade Wind Turbine System. The following sections summarise some of the findings and main conclusions from this research regarding the Telescopic Blade Wind Turbine System.

6.1. Performance testing of Telescopic Blade Wind Turbine: the effect of blade extension

The performance of the telescopic blade concept has been analysed experimentally and computationally. It has been established that the effect of a step change in blade chord is significant for the range of blade extensions studied (0 to 40%). It is recognised that the current simulation models based on the blade-element-momentum theory incorporating Prandtl tip and hub loss models, do not incorporate step losses, and therefore over-predict rotor performance when there is a step change in chord along a blade. Correlations have been developed to quantify losses arising from the step change in the chord of a telescopic blade based on Prandtl tip and hub loss concepts. Predictions utilizing the new correlations are in good agreement with the experimental data. It was also found that the maximum step loss occurs at a tip blade length which is equivalent to one root-blade-tip chord representing a 20% blade extension in this research. The power output of a telescopic blade wind turbine increased for all blade extensions considered in this study, due largely to the accompanying increase in swept area, in spite of the detrimental effects of the step change in blade chord. As shown in this research, the effect of a step change in the blade chord of a telescopic blade system has a more significant impact for lower tip blade extensions. For tip blade lengths of 2 times the root-blade-tip chord and beyond, it can be deduced that the effect of step loss reduces as shown in this research for the case of 40% extension.

6.2. Performance testing of Telescopic Blade Wind Turbine: the effect of chord ratio

It has also been established from this research that the aerodynamic effect of a step change in chord is significant over the range of chord ratios studied; chord ratio being defined as the ratio of the chords of the tip to root blade sections. The blade with a chord ratio of 0.4 when compared with a chord ratio of 0.6 had a lower power coefficient as well as a lower blade thrust. This was due to the size of the step change in blade chord and a lower solidity of the blade with the lower chord ratio. Despite having a detrimental impact on the power
coefficient of a telescopic blade, the energy outputs of the telescopic blade wind turbine were seen to be higher for all blade extensions at different chord ratios when compared to that of zero extension. It was found that with a lower chord ratio, the power coefficient is lower, thus there is a lower energy output. Therefore it can be concluded that to minimise the effect of the step loss, the chord ratio needs to be as close to unity as possible and the minimum tip blade extension to be at least 2 times the root-blade-tip-chord.

6.3. Aerodynamics of telescopic blades: the influence of step change
The present research aimed to describe the effects of a step change in chord of a telescopic blade. Surface static pressures were measured with different tip blade extensions at various angles of attack and Reynolds numbers. Two dimensional and three dimensional flow effects were also investigated.

From the tests carried out on a low aspect ratio telescopic blade at low Reynolds numbers, it was found that the sudden change in the blade chord causes an abrupt drop in pressure around the step region. It is well known (for example in the case of a blade tip) that this is accompanied by a vortex roll up in the wake, creating a 3D flow thereby affecting the span-wise pressure distribution of the low aspect ratio and low Reynolds number telescopic blade system that was studied. It can be deduced that longer tip blades would have this influence over only a smaller proportion of their length. Hence the span-wise pressure distribution and thus the blade performance would not be affected as much. The results also suggest that longer tip blades would not stall as early as the fully extended ones tested in this research. There is also a suggestion that if the aspect ratio of the tip blade is high, the tip blades stall closer to that of the root blade. Also, it is found that a lower aspect ratio or shorter tip blade has an adverse effect on the root blade of a telescopic blade system and reduces performance.

By increasing the tip blade length to have different tip blade extensions, the maximum pressure coefficient over each of the tip and root sections of the telescopic blade also increased giving increased aerodynamic performance.

As the Reynolds number increases, it was found that the blade performance also increases. For the telescopic blade system studied, the tip and root blades had different chord Reynolds numbers for the same free-stream velocity due to their different chord lengths. The effects of different chord Reynolds numbers were pronounced in this research, as the tip blade stalled before the root blade, due to Reynolds number effects. This implies that if the tip blade chord
were closer to that of the root blade (i.e. with a higher chord ratio), this Reynolds number effect could be minimised or eliminated.

6.4. Energy analysis of a Telescopic Blade Wind Turbine System
The energy output analyses for the telescopic blade wind turbine system were carried out for different blade extensions, chord ratios and wind characteristics via the Weibull parameters. It is shown from the analysis that as the blade extension increases, the power output increases and so does the annual energy output. The percentage increase in the annual energy output could be up to ~48% for a blade with 40% (five times the root-blade-tip chord) blade extension at a chord ratio of 0.6 in a Wind Class 2 site.

It is also apparent that the chord ratio has a significant impact on the annual energy output of a system. For wind classes 2 to 7, and \( k = 2 \), the increase in energy output of the telescopic blade wind energy system with 40% extension, for chord ratios of 0.4 – 1.0 are in the range of 11 – 51% respectively. However, for blade extension of 10% (approximate one times the root-blade-tip chord), a chord ratio of 0.4, and \( k = 2 \), there is no increase in the annual energy output.

The wind speed distribution which is described by the Weibull shape factor \( k \) also has an impact on the energy output of the extended blade wind turbine. As \( k \) increases, the energy output also increases for all wind classes, but the increases are more significant in low Wind Class sites. The increases in energy output are in the range of 16.5-62% for wind classes 7 - 2 respectively for blade extension of 40% (five times the root-blade-tip chord), \( k=2.5 \) and a chord ratio of 0.6. For wind class 2, a blade extension of 40% and a chord ratio of 0.6, the percentage increase in annual energy output for \( k = 1.5, 2.0 \) and 2.5 are 35, 48 and 62% respectively.

For the range of shape factors, chord ratios and blade extensions studied in this research, the reduction in annual energy output due to the step change in the blade chord is found to be in the range of 15 to 1% for wind classes 2 to 7 respectively compared to what it would be if the step loss were ignored in the calculation. Any reduction in the step loss could be capitalised as enhanced energy output which would increase the capacity factor and make the telescopic blade wind turbine system even more attractive.
Conclusion
6.5. Recommendations

6.5.1. Performance testing of Telescopic Blade Wind Turbine

- Optimisation of the Telescopic Blade Wind Turbine System geometry: the blade geometry could be optimised. Performance testing could then establish the configuration that produces a high performance low cost Telescopic Blade Wind Turbine System. The configurations could include variable chords and introduction of twist.

- Performance testing of Telescopic Blade Wind Turbine System could be carried out at higher Reynolds numbers that are typical of full scale systems and with winglets to see their effects.

- Different blade actuation mechanisms need to be identified and investigations carried out to establish the energy required for actuation and control of the telescopic blades. Could the centrifugal force, incorporated with governor technology be used for blade actuation?

- A wake analysis for the Telescopic Blade Wind Turbine System, if carried out could reveal further insight into its aerodynamics.

- An optimum control strategy to enhance the energy output of a Telescopic Blade Wind Turbine System in a cost effective manner could be investigated.

6.5.2. Aerodynamics of telescopic blade

- The effect of step change needs to be established for higher aspect ratio blades at different chord ratios and Reynolds numbers.

- Wake analysis to be carried out and correlated with the blade pressure measurements to get better understanding of the step vortex. Since the blades have different chords, they will have different shedding frequencies. These different shedding frequencies could induce the wake to be skewed which could therefore affect the flow-field. Another adverse effect of the different shedding frequency could be that it will induce a pulsating effect on the blade which can increase the fatigue loading.

- Investigate the use of a fillet at the step change region to effect a gradual decrease in chord.
Recommendations

- Extensive Computational Fluid Dynamics (CFD) work could be carried out to get more insight into the fluid dynamics around and away from the telescopic blades.

6.5.3. Energy analysis of a Telescopic Blade Wind Turbine System

- Effectiveness of different control strategies to be carried out for different setups. As discussed earlier, the strategy used in this research is a step-type blade actuation. When the wind speed changes to certain value based on the root bending moment, then the blade actuates accordingly. In contrast to this, a continuous blade actuation analysis to be carried out to see the improvements it does to the energy enhancements. On the other hand, the effect of continuous actuation on the blade wear and tear also needs to be established.

- In this research, the telescopic blade concept of fully retracted blades at rated wind speeds is used. When the wind speed drops below rated, the blades extend and remain extended for wind speeds below rated. Future work could investigate intermediate and fully extended blades that might be suitable for medium and high wind sites respectively.

- The overall load on the turbine blade, tower and foundation needs to be quantified for a Telescopic Blade Wind Turbine System.
Appendices

A. Appendix Part A

A.1 Effect of step change

A.1.1 2D and 3D view zero extension, 2D Configuration, $Re_{\text{root}}$ 180 000
A.1.2  2D view Zero extension, 3D Configuration, $Re_{root}$ 180 000
A.1.3 2D and 3D view full extension, 2D Configuration, \( R_e,\text{root} 180,000 \)
A.1.4  3D view full extension, 3D configuration, $R_{e,root}$ 180 000
A.1.5  3D view half extension, 2D configuration, $R_{e,root} = 180000$
Blade length (mm)
Appendices

A.1.6 3D view half extension, 3D configuration, $R_e_{\text{root}}$ 180 000
A.1.7 3D view quarter extension, 2D configuration, $R_{e,\text{root}}$ 180 000
Appendices
A.1.8 3D view quarter extension, 3D configuration, \( R_{e, \text{root}} 180 \, 000 \)
A.2.2D and 3D effects

A.2.1 3D view full extension, \( \text{Re}_{\text{root}} 180000 \)
A.2.2 2D view half extension, $R_{e, root} = 180000$
A.2.3  3D view half extension, \( R_e \text{ root} \ 180\ 000

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Appendices

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Appendices

2D, Alpha = 12°

3D, Alpha = 12°

2D, Alpha = 15°

3D, Alpha = 15°

2D, Alpha = 18°

3D, Alpha = 18°

2D, Alpha = 21°

3D, Alpha = 21°
Appendices

A.2.4 2D view quarter extension, $R_{e,\text{root}}$ 180 000

[Images of 2D and 3D plots for various angles and blade lengths]
A.2.5 3D view quarter extension, $Re_{root} 180\ 000$
A.2.6 2D view zero extension, $R_{e,\text{root}}$ 180 000
A.3 Effect of blade extension

A.3.1 3D view, 3D configuration, $R_{e\_root} 180 000$
A.3.2  2D view, 3D configuration, $R_{e,root}$ 180 000
A.3.3  3D view, 2D configuration, $R_{e,root}$ 180 000
A.3.4  2D view, 2D configuration, $R_{e, root} 180 000$
A.4 Reynolds number effects

A.4.1 2D view, 3D configuration
A.4.2  2D view, 2D configuration
B. Appendix Part B

B.1 Power curves for different blade extensions at different rotor rpm

(a) No extension

(b) 10% extension

(c) 20% extension
Appendices

(d) 30% extension

CR 0.4

(e) 40% extension

(a) 10% extension

(b) 20% extension
(c) 30% extension

(d) 40% extension
C. Appendix Part C

C.1  \(C_p\) curves for different blade extensions and chord ratios

(a) 10\% extension  
(b) 20\% extension  
(c) 30\% extension  
(d) 40\% extension

CR 1.0
Appendices

(a) 10% extension

(b) 20% extension

(c) 30% extension

(d) 40% extension

CR 0.4
D. Appendix Part D

D.1 Power output using the blade bending moment strategy for different chord ratios

(a) No extension

(b) 10% extension

(c) 20% extension
Appendices

(d) 30% extension

CR 1.0

(e) 40% extension

(a) 10% extension

(b) 20% extension
(c) 10% extension  
(d) 20% extension

CR 0.4
E. Appendix Part E

E.1 The effect of shape and scale parameter for different chord ratios on annual energy output

(i) $k=1.5$

(ii) $k=2.0$
(iii) $k=2.5$

(a) CR 1.0

(b) CR 0.6 $k=1.5$
Appendices

(b) CR 0.6
CR 0.4 \(k=2.5\)

(c) CR 0.4
E.2 The effect of shape and scale parameter on annual energy output

10%, $k=1.5$

- CR - 1.0
- CR - 0.6
- CR - 0.4

Capacity factor CR 1.0
Capacity factor CR 0.6

20%, $k=1.5$

- CR - 1.0
- CR - 0.6
- CR - 0.4

Capacity factor CR 1.0
Capacity factor CR 0.6
Capacity factor CR 0.4
References


83. Turbulent flow instrumentation. *Cobra probe - 100 series.*


