Unsteady Operation and Rapid Start Up of Francis Turbines

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Abstract

Hydropower is a key component in a decentralised market with increasing penetration of often intermittent renewable energy resources. There is growing need for large-scale operators to develop a more comprehensive understanding of the full dynamic capabilities of a given hydropower plant, while developments in small-scale hydro will aid in waste energy recovery and rural electrification. In particular, greater confidence in the transient response characteristics of hydro generators will in turn enable and encourage increased contributions from other much needed large scale renewable energy sources, such as wind and solar power, without compromising grid stability.

The major drawback of current pump-as-turbine (PAT) micro-hydro solutions is the very narrow effective operating band and lack of universal performance prediction models. To address this, the design process and installed performance of a new 6.2 kW PAT micro-hydro turbine unit and test facility is presented. Steady state testing over the full operating range indicates a best efficiency of 79% and demonstrates near peak efficiency operation over an extended range of head and flow conditions. The developed turbine is a promising prototype of a lower cost alternative for installations where the capital costs associated with conventional Francis turbine units are often prohibitive.

The potential for hydraulic turbines to provide rapid reserve power generation is investigated experimentally by transitioning the micro-hydro unit from a modified synchronous condenser mode under varying levels of tail water depression and inlet guide vane (IGV) opening rates. Similar full-scale testing on a 116 MW Francis turbine generating set revealed an undesirable power draw and subsequent output power oscillations during the early stages of transition limiting the potential contribution of the tested unit. Direct shaft power measurements of the micro-hydro unit during transition demonstrate the presence of an inhibiting mechanical torque applied to the runner during low guide vane opening angles, of the order of 2% of rated, which is believed to initiate the electromechanical response seen at full-scale. The temporal location and duration of the applied negative torque was found to be highly dependent on IGV opening rate while the effect of tail water depression level was insignificant under constant head conditions. The magnitude of the power draw was independent of initial tail water depression level, while an increased opening rate marginally reduced the observed power draw at the laboratory scale.

An improved one-dimensional numerical model of a Francis turbine hydropower plant for dynamic response studies is presented. The model is based on the equations of motion
and continuity, while the conventional representation of the hydraulic turbine as an orifice is improved upon to account for machine behaviour away from design using known inlet flow velocity vectors. The new model remains valid at low IGV angles, where the traditional one-dimensional model breaks down, and is validated against full-scale transient test data. Furthermore, simulation results support the findings of experimental work indicating a high dependence on IGV opening rate in relation to output response during transition from tail water depression mode.

Ultimately, the research presented assesses the feasibility of transitioning a Francis turbine unit from tail water depression mode to generation for the purpose of providing rapid load support to the local grid. A three-stage transition mechanism responsible for the observed output response is proposed based on the findings from both experimental and numerical investigations, while key parameters and potential risks are identified. Finally an improved set of operational guidelines are presented for increasing the provision of rapid reserve power generation for maintaining system stability and ensuring a secure electricity supply.

**Keywords**: Francis turbine, micro-hydro design, transient response, rapid start up, hydraulic modelling, transient simulation.
Declaration

The research presented in this dissertation was conducted at the School of Engineering in the University of Tasmania between February 2010 and September 2014. This dissertation is the work of the author alone, and includes nothing which is the outcome of work done in collaboration, except where specifically indicated to the contrary. None of the work presented in this dissertation has been submitted to any other University or Institution for any other qualification.

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Supporting Publications

Several components of the work presented in this thesis have been published during the course of the project.

An initial review and preliminary experimental investigations into the feasibility of operating a hydraulic turbine in a tail water depression mode for rapid power generation was presented in Giosio et al. [31]. The design and steady-state performance of an improved micro-hydro turbine unit based on pump-as-turbine principles described in Chapters 4 and 5 was presented in Giosio et al. [32]. Additionally, comprehensive rapid start-up testing of the micro-hydro turbine unit, and a comparison to full-scale field test results, presented largely in Chapter 6 was published in Giosio et al. [33].
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Nomenclature

Latin Symbols

\( a \) \quad \text{Wave speed [m s}^{-1}] \\
\( f \) \quad \text{Darcy-Weisbach friction factor [-]} \\
\( g \) \quad \text{Acceleration due to gravity [m s}^{-2}] \\
\( \dot{m} \) \quad \text{Mass flow rate [kg s}^{-1}] \\
\( n \) \quad \text{Speed of revolution [s}^{-1}] \\
\( p \) \quad \text{Static pressure [Pa]} \\
\( t \) \quad \text{Time [s]} \\
\( v \) \quad \text{Mean flow velocity [m s}^{-1}] \\
\( z \) \quad \text{Elevation above given datum [m]} \\
\( A \) \quad \text{Cross sectional area of flow [m}^2] \\
\( A_0 \) \quad \text{Guide vane passage opening [m]} \\
\( C_m \) \quad \text{Meridian velocity component [m s}^{-1}] \\
\( C_u \) \quad \text{Circumferential velocity component [m s}^{-1}] \\
\( D \) \quad \text{Runner reference (outlet) diameter [m]} \\
\( D_p \) \quad \text{Penstock diameter [m]} \\
\( E \) \quad \text{Specific hydraulic energy [J kg}^{-1}] \\
\( G \) \quad \text{Guide vane function [-]} \\
\( H \) \quad \text{Hydraulic head [m], inertia coefficient (combined) [-]} \\
\( H_{Eu} \) \quad \text{Euler head [m]} \\
\( H_f \) \quad \text{Friction head [m]} \\
\( H_0 \) \quad \text{Static head [m]}
$J$  Inertia coefficient (combined) [kgm$^2$]
$L$  Conduit length [m]
$M$  Moment [Nm]
$N$  Speed of revolution [rpm]
$P$  Power [W]
$Q$  Discharge [m$^3$s$^{-1}$]
$R$  Radius [m]
$S$  Servomotor stroke (full-scale), actuator stroke (micro-hydro scale) [mm]
$T$  Torque [Nm]
$T_e$  Elastic water time constant [s]
$T_w$  Water acceleration time constant [s]
$V$  Fluid velocity [ms$^{-1}$]
$V$  Volume [m$^3$]
$U$  Circumferential velocity [ms$^{-1}$]
$W$  Relative flow velocity component [ms$^{-1}$]
$Z$  Hydraulic surge impedance [-]

**Greek Symbols**

$\alpha$  Absolute flow angle, guide vane opening angle [$^\circ$]
$\beta$  Relative flow angle [$^\circ$]
$\eta$  Efficiency [-]
$\rho$  Density of working fluid [kgm$^{-3}$]
$\sigma$  Thoma number [-]
$\omega$  Runner angular velocity [rads$^{-1}$]
$\zeta$  Loss coefficient [-]
Subscripts

0                   Initial
1                   High pressure side of machine, runner inlet
2                   Low pressure side of machine, runner outlet
a                   Atmospheric
b                   Blade
e                   Electric
h                   Hydraulic
m                   Mechanical
net                 Net value
nl                  No-load
o                   Overall
p                   Pump
r                   Rated
ref                 Reference
t                   Turbine
th                  Theoretical
v                   Volumetric
Ch                  Churn
G                   Generator
M                   Measurement section, micro-hydro
R                   Reece, runner

Superscripts

□ Non-dimensional quantity (normalised by rated value)
⇒ Vector quantity
Dimensionless Numbers

\( E_{nD} \) Energy coefficient [-]

\( Fr \) Froude number [-]

\( n_{QE} \) Specific speed [-]

\( P_{nD} \) Power coefficient [-]

\( Q_{nD} \) Discharge coefficient [-]

\( Re \) Reynolds number [-]

\( T_{nD} \) Torque coefficient [-]

\( \varphi = \frac{Q}{\pi w R^2} \) Flow coefficient [-]

\( \psi = \frac{2E}{\omega^2 R^2} \) Specific hydraulic energy coefficient [-]

Abbreviations

\( p.u. \) Per unit

BEP Best efficiency point

CV Control volume

FCAS Frequency Control Ancillary Service

IGV/GV Inlet Guide Vane/Guide Vane

MIV Main inlet valve

NL No-load

PAT Pump-as-turbine

SC Synchronous condenser

S.L. Referenced to sea level

TWD Tail water depression

UTAS-MH University of Tasmania micro-hydro unit

WAE Weighted-average-efficiency
Chapter 1

Introduction

Hydropower, the ability to harness the energy of moving water, has been utilised in various forms for centuries.

Many ancient cultures including the Greeks and Egyptians are known to have employed water wheels to lift and convey water for irrigation systems and for turning grindstones to help produce flour. By the fourteenth century water wheels were extensively used as a means to power machinery for grinding grain, sawmills or textile mills, either through a direct drive or pulley and gear system.

The current usage of the term hydropower has come to be associated with what is more correctly described as hydro-electric power. Hydro-electric power refers to the process of converting the kinetic and potential energy of water to a more useful and transportable form of energy, namely electricity.

It is fitting perhaps that now, as we are finally beginning to realise that a market based around renewable sources of energy production is the only sensible and prudent option we have going into the future, we will turn back to the earliest form of electricity production.

Electricity production is achieved by transforming the energy of the water, using some form of hydraulic turbine, into mechanical torque to drive an electric generator. The design of modern hydraulic turbines is generally credited to Benoit Fourneyron in 1833. An interesting and highly recommended discussion on the history and development of the hydraulic turbine and hydro-electric power in general can be found in Gulliver and Arndt [35].
For decades now around the world hydropower has been used as a base load provider for electricity generation. Additionally, due to the ability of hydropower plants to be brought on-line comparatively faster and more economically than conventional fossil fuel plants and other base load providers, hydropower has also been used to an increasing degree for peak load management. In a market with ever increasing penetration of often intermittent renewable energy resources, hydropower can provide flexible and reliable generation options for maintaining system stability and ensuring a secure electricity supply.

In the modern liberalised market the flexible and comparatively rapid response of hydropower also presents additional opportunities for operators in the form of ancillary service markets. As such there is a growing need for large-scale operators to develop a more comprehensive understanding of the full dynamic capabilities and safety limitations of a given hydropower plant, while developments in small-scale hydro will aid in waste energy recovery and rural electrification.

In particular, greater confidence in the transient response characteristics of hydro generators will in turn enable and encourage increased contributions from other, much needed, large scale renewable energy resources such as wind and solar power, without compromising grid stability.

This is ever more significant in the context of the Tasmanian electrical system. The relatively isolated and low inertia system is predominantly supplied by renewable energy sources with an increasing contribution of wind power. As such both base load and system stability services must, by and large, be provided by hydro turbine units. And while hydropower makes up 77% of the approximate 2968 MW\textsuperscript{1} generating capacity within the state all the hydropower stations within the state are conventional installations, as opposed to pumped-storage, somewhat limiting operational options and further reinforcing the need for understanding safe dynamic behaviour.

In the present study the potential for hydro turbines to provide fast frequency regulation to the grid, following rapid start-up from a modified synchronous condenser (SC) mode, is investigated. Experimental studies on a specifically designed micro-hydro scale Francis-type pump-turbine are given in both steady-state and transient operation. An improved numerical analytical model of a turbine unit, hydraulic circuit and generator is presented with simulation results compared to full-scale test data.

\textsuperscript{1}A 383 MW gas turbine power station and 310 MW of installed wind turbines make up the remainder of the Tasmanian generating potential.
1.1 The Francis Turbine Hydro-Electric Power Station

The general layout of a conventional hydro-electric power station is illustrated in Figure 1.1. The main components of a typical single machine station generally comprise of an upper reservoir, intake structure, penstock, main inlet valve, spiral casing, inlet guide vanes (also called wicket gates or the distributor), turbine-generator unit, draft tube and lower reservoir (tailrace).

![Figure 1.1: Schematic layout and main components of a single machine hydro-electric power station equipped with a Francis turbine unit (modified from [52]).](image)

Depending on the requirements of the station, a surge tower may also be incorporated for the control of pressure transients. A number of turbines may be installed in a given station depending on a variety of factors including capital and operating costs, flow availability and operational requirements. In this case a station may have duplicate hydraulic circuits or a shared penstock with manifold section for distribution of flow.

The net specific energy at various locations within the hydropower station is shown schematically in the form of an energy grade line in Figure 1.2. The mean specific hydraulic energy at a given point is given by equation Eq. 1.1.
\[ gH = \frac{p}{\rho} + \frac{v^2}{2} + gz \]  

(1.1)

Accounting for component losses and conduit friction losses, the net available specific energy of the turbine, \( E \) [J/kg\(^{-1}\)], defined as the difference in mean specific energy across the machine is given by Eq. 4.5.

\[ E = gH_1 - gH_2 = gH_{net} = g(z_A - z_B) - \sum gH_{losses} \]  

(1.2)

Working from either an energy-work approach or from the Bernoulli energy equation [105], it can be shown that the hydraulic power available to the turbine is a product of the specific energy and the mass flow rate as given by:

\[ P_h = \rho Q \cdot E \]  

(1.3)

This is the total theoretical power available to the turbine under the given operating conditions. The actual power that may be developed by the turbine however is somewhat reduced due to mechanical losses in bearings, leakage flow and hydraulic losses. The overall machine efficiency may be described by Eq. 1.4 which is shown as the product of mechanical efficiency, volumetric efficiency and hydraulic efficiency.

\[ \eta_o = \eta_m \cdot \eta_v \cdot \eta_h \]  

(1.4)
The power transformed by the hydraulic turbine is therefore given by Eq. 1.5:

\[ P_t = \eta_o \rho Q \cdot E \]  

(1.5)

This, by definition, is equal to the mechanical power transferred by the torque applied to the shaft, \( P_m = \omega \cdot T \), such that the overall efficiency of the turbine unit is given as:

\[ \eta_o = \frac{P_m}{P_t} = \frac{\omega \cdot T}{\rho Q E} \]  

(1.6)

### 1.1.1 Machine Specific Speed

Generally speaking there are three main, distinct types of hydraulic turbine in common large-scale usage: the Pelton, Francis and Kaplan turbines, each of which reaches their optimal efficiency at different conditions on the HQ-plane. As such depending on the site conditions, namely the available flow, \( Q \) [m\(^3\)/s], and the available net head, \( H_{\text{net}} \) [m], the most appropriate type of turbine may be chosen.

Turbines are classified based on the specific speed of the machine. The specific speed may be defined as given in Eq. 1.7, with nominal head and flow values taken at design operating point where efficiency is expected to be at a maximum, and machine speed, \( n \), given in s\(^{-1}\).

\[ n_{QE} = \frac{n Q^{0.5}}{E^{0.75}} \]  

(1.7)

The specific speed is a dimensionless parameter that defines the type of machine to be used for optimal performance under the stated operating conditions, although dimensional variations of this definition are in common usage. Not only does the specific speed define the type of machine to be used, but it also describes the required geometry of a given type of turbine as illustrated for the Francis turbine in Figure 1.3. It follows that geometrically similar machines having the same specific speed will exhibit equivalent internal flow pattern and achievable efficiency, regardless of the size or operating speed of the machine\(^2\).

---

\(^2\)This is generally true, however a reduction in physical size does result in an increase of the relative magnitude of losses. Efficiency majoration formulae are available to correct for this influence.
Figure 1.3: Francis turbine runner geometry for a range of specific speeds [76]. Runner geometry is seen to change from purely radial inlet flow at low specific speed to having a large axial component, or mixed inlet flow, at higher specific speeds.

1.1.2 Francis turbine operating principle

A section view showing all the main components of a typical Francis turbine is illustrated in Figure 1.4. Water is taken from the upper reservoir and directed via an intake structure to the pressurised penstock which delivers the flow to the turbine inlet.

Flow enters the Francis turbine via a spiral casing which surrounds the entire runner periphery and serves to maintain a constant radial inward flow into the runner by continuously decreasing the cross-sectional area of the flow passage. Flow passes through a ring of stationary (stay) vanes and is then directed into the runner at the desired angle for the current operating condition by adjustable guide vanes.

Upon entering the runner the flow direction is changed by the curved turbine runner blades to produce a corresponding change in tangential momentum which in turn imparts a torque on the runner as described by the Euler equation for energy transfer, Eq. 1.11 below.

Flow exits the turbine runner axially and is discharged into the draft tube to be transported to the tail race. The draft tube generally has the shape of a diffuser which serves to reduce
the velocity of the discharging fluid and increase the total head available to be converted into mechanical power.

Referring to Figure 1.5, working from the law of moment of momentum for one dimensional, steady flow it can be shown that the sum of the moments of all external forces acting on the fluid (equal to the torque applied to the shaft) is equal to the rate of change of angular momentum between inner and outer periphery of the turbine runner

$$T = \rho Q(c_{u1}R_1 - c_{u2}R_2)$$

(1.8)

where subscript 1 here represents the turbine runner inlet while subscript 2 represents the outlet. For a turbine rotating with an angular velocity of $\omega$ [rad/s], the rate at which the fluid does work on the rotor is given by Equation 1.9:

$$T\omega = \rho Q(U_1c_{u1} - U_2c_{u2})$$

(1.9)

since $U = \omega R$. 

Figure 1.4: Francis turbine three quarter section view showing all main components [53]
When equated with the equation for available hydraulic power, Eq. 5.7, gives:

\[
\rho g Q H_{net} = \rho Q (U_1 c_{u1} - U_2 c_{u2})
\]  

(1.10)

from which the Euler turbine equation is obtained where product \( g H_{Eu} \) represents the theoretical maximum energy transfer between fluid and turbine runner [94], as expressed in Eq. 1.11. The Euler turbine equation describes the ideal specific energy able to be extracted by the turbine under the given operating conditions:

\[
g H_{Eu} = U_1 c_{u1} - U_2 c_{u2}
\]  

(1.11)

The available specific energy becomes a maximum in the ideal case where the runner outlet tangential velocity component, \( c_{u2} \), is reduced to zero, Eq. 1.12. This is known as the no swirl condition and is shown vectorially in Figure 1.5 for the case where \( Q = Q_r \).

\[
g H_{Eu} = U_1 c_{u1}
\]  

(1.12)
1.2 The Role of Hydropower in Tasmania

1.2.1 Hydro Tasmania

Founded in 1914 as the State Government run Hydro-Electric Department, Hydro Tasmania is the sole electricity provider in the state and now generates approximately 10,000 GW-h of electricity per year with a generating capacity of 2615 MW. Hydro Tasmania’s assets include 27 hydro-electric power stations, numerous lakes, over 50 large dams, the Huxley Hill wind farm on King Island, two diesel power stations and a thermal power station situated at Bell Bay in the state’s north [46]. Until recently Tasmania, an island state of Australia, was isolated in terms of the electricity grid from the mainland of Australia. However with the installation of the high voltage direct current (HVDC) undersea Basslink cable in 2006 Tasmania is now trading in the national market.

1.2.2 The National Electricity Market (NEM)

The Australian National Electricity Market (NEM) is the wholesale electricity market that facilitates the exchange of energy between producers and consumers across five interconnected market jurisdictions. Currently only the eastern half of Australia is part of the NEM with the five regions including Queensland, New South Wales, Victoria, South Australia and Tasmania. The NEM currently supplies over $10 billion of electricity to more than 8 million end users annually [7].

The NEM operates as a logical spot market where the electricity output from all generators is effectively pooled together and redistributed to consumers based on current demand. The dispatch process is operated by the Australian Energy Market Operator (AEMO) (formally The National Electricity Market Management Company, NEMMCO). AEMO manages the NEM using 5 minute forecasts of energy demand and receiving supply side bids from the electricity generators. AEMO then determines which generators are required to meet the forecast demand in the most cost effective way and schedules the required generators into production.

Tasmania joined the NEM in May of 2005 however trading of energy did not become fully operational until the Basslink interconnector was commissioned in April 2006.

Basslink is a high-voltage direct current (HVDC) undersea cable rated at 500 MW that connects the sub station at George Town, Tasmania to the Loy Yang Power Station in Victoria.
Since activation of the Basslink interconnector in 2006 and inclusion into the NEM, Hydro Tasmania has been able to export electricity during peak times when the market price is high and import energy from the national market when the market price is lower. This not only provides economic benefits for Hydro Tasmania but also helps to manage and preserve water levels during times of low inflow.

### 1.2.3 Ancillary Services

In addition to meeting the current demands of the electricity market AEMO is required under the National Electricity Rules (NER), established by the Australian Energy Market Commission (AEMC), to maintain power system security [6] by ensuring adequate reserve capacity within the system should any deviation or contingency event occur.

Ancillary Services are those services used by NEMMCO (AEMO) to manage the power system safely, securely and reliably. These services maintain key technical characteristics of the system, including standards for frequency, voltage, network loading and system restart processes [8].

AEMO manages the power system through the procurement of three major categories of ancillary services from the same generating units that provide energy to the system. The three major categories of ancillary services are defined as:

- Frequency Control Ancillary Services (FCAS)
- Network Control Ancillary Services (NCAS)
- System Restart Ancillary Services (SRAS)

Since the inclusion of Tasmania into the national market the supply of fast FCAS, discussed in the following section (Section 1.2.4), has been problematic [40].

### 1.2.4 Frequency Control Ancillary Services (FCAS)

The power supply in Australia is generally maintained within a normal operating band (NOB) around the nominal frequency of 50 Hz. AEMO procures FCAS from energy providers to ensure that in the event of minor frequency fluctuations, due to normal changes in demand and/or generation, or the occurrence of a contingency event, such as the loss of the
single largest generating unit in a region, the system frequency is maintained within the standards set by the NER. As such, Frequency Control Ancillary Services can be divided into two categories:

- Regulation frequency control, and
- Contingency frequency control

Regulation frequency control is the correction of the minor frequency fluctuations of the system due to changes in load and/or generation and is continually used by AEMO to maintain system frequency. Contingency frequency control is the correction of a major frequency deviation resulting from the loss of a generating unit or major load. While being constantly enabled, contingency control it is only ever used in the case of a contingency event as shown in Figure 1.6.

If the system frequency deviates out of the NOB the NER specifies that AEMO must ensure that the frequency stays within a specified contingency band and correct the frequency deviation within 5 minutes of the occurrence. To achieve this, the standards have defined eight FCAS requirements, each with their own FCAS market.

As shown in Table 1.1 each type of contingency frequency control is defined as either ‘Raise’ or ‘Lower’ denoting the ability of the service to either increase the system frequency or decrease the system frequency. The frequency control service of particular interest in this case is the Fast Raise (R6) FCAS.
Table 1.1: Classification of the eight FCAS requirements as specified by AEMO [8].

<table>
<thead>
<tr>
<th>Contingency</th>
<th>Regulation Raise</th>
<th>Regulation Lower</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fast Raise (R6) and Fast Lower (L6)</td>
<td>(six second response to arrest the immediate frequency deviation)</td>
<td></td>
</tr>
<tr>
<td>Slow Raise (R60) and Slow Lower (L60)</td>
<td>(sixty second response to keep the frequency within the single contingency band)</td>
<td></td>
</tr>
<tr>
<td>Delayed Raise (R5) and Delayed Lower (L5)</td>
<td>(five minute response to return the frequency to the Normal Operating Band)</td>
<td></td>
</tr>
</tbody>
</table>

1.3 Motivation of Study

As mentioned previously, since Tasmania joined the NEM in 2006 Hydro Tasmania has had difficulty supplying adequate local fast raise frequency control ancillary services (R6 FCAS). Fast raise FCAS is the most difficult service to supply for hydro schemes [40] and this problem has only been worsened by the recent low inflows and low storage levels.

Currently in Tasmania the total amount of fast raise FCAS available is 90 MW at maximum efficiency and 233 MW at full FCAS capability. This however assumes full plant availability, which is significantly reduced with Basslink import and the increasing contribution of wind generation. Under low inflow and low storage conditions Hydro Tasmania strategy is to operate under high Basslink import, this however limits the access to cheap global FCAS and results in low system inertia reducing the amount of available local FCAS. Currently in low load conditions when Basslink import is high there is a shortage in fast raise FCAS of 10 - 25 MW.

In order to meet this fast raise FCAS deficit, while avoiding the need to reduce Basslink import, the John Butters Power Station is currently being operated at low output resulting in low efficiency water usage (down to 30%)\(^3\), increased operating hours, increased maintenance costs, and decreased asset life due to rough running [102].

The present study investigates the novel concept of running a turbine unit in a modified synchronous condenser mode such that the machine may be brought on-line rapidly to provide additional fast raise FCAS capacity. If successful the proposed operational scheme would also contribute a number of additional benefits and market products including inertia, fault level, voltage and reactive power control as well as the provision of fast raise FCAS [40].

\(^3\)Estimated cost of $0.5M to $1.0M p.a. in lost revenue in addition to increased wear and associated maintenance costs [41]
1.4 Thesis Structure

The main objective of the presented research was to develop a greater understanding of the flow physics and mechanisms associated with rapidly bringing on-line a Francis turbine for the purpose of providing frequency support to the local grid.

The research program was conducted in two streams: experimental testing was performed on a purpose built micro-hydro scale turbine unit, the design of which was part of the current work; while a new simulation model was developed, incorporating the findings from full-scale and laboratory testing, resulting in a useful predictive tool for determining unit power response under the proposed start-up procedure.

The overall objectives and motivation of study have been described in the present chapter along with some relevant background theory and a discussion on the role of hydropower in Tasmania, both at present, and looking forward into the future. A survey of the literature in regards to the off-design behaviour of hydraulic turbines, general experimental experience, and transient hydro power plant modelling is given in Chapter 2. Additionally, Chapter 2 describes the proposed mode of operation allowing rapid machine start-up. Chapter 3 presents the findings of investigative full-scale rapid starting tests performed by Hydro Tasmania on an identified Francis turbine power plant on the west coast of Tasmania, the results of which provided the motivation for the study presented.

The design and development of a novel micro-hydro turbine unit and facility is presented in Chapter 4. The entire facility and turbine unit was designed by the author as part of the present study. The turbine unit was designed during a 5 month engagement in which the author worked at Pentair Flow Technologies, Tasmania, Australia, under the guidance of one of the design engineers. The steady state performance of the new micro-hydro turbine is presented in Chapter 5, while results from laboratory transient start-up tests are given in Chapter 6.

The development of an improved Francis turbine power plant transient model; based on effective inlet flow velocity vectors and the conservation of angular momentum; is presented in Chapter 7.

Ultimately, the findings from both full-scale and laboratory testing, as well as results from numerical modelling, are brought together in Chapter 8 where a likely transition mechanism is proposed. The proposed mechanism is then incorporated into the improved numerical model of the Francis turbine power plant and simulation results are compared to full-scale observations. Finally, the improved numerical model is used to predict plant output based on a number of potential optimisation methods.
In summary, the work presented in this dissertation documents a number of original developments in the utilisation of hydropower for the rapidly evolving electricity market.

In the context of micro-hydropower, a novel turbine unit design is presented, built, and tested. Micro-hydropower is a key component in rural electrification, particularly in developing countries, and is also finding uses in industrial energy recovery applications. The unit was designed to bridge the gap between the high capital cost of conventional Francis turbines and the low-cost, but highly constrained, use of pump-as-turbines. The steady-state performance of the unit — the first of its kind reported — demonstrates that the effective operating range of a pump-as-turbine can be significantly extended by the addition of a suitably designed inlet flow control mechanism. The resulting high efficiency over a wide range of available head and flow conditions thus enables the turbine to vary output to meet demand more effectively, thereby increasing the economic viability of potential micro-hydropower installations.

The ability to rapidly inject power into the electricity network is of extreme importance in terms of maintaining system stability following contingency events. For such purposes a novel technique is proposed to considerably increase spinning reserve response. For the first time in the literature, the feasibility and factors influencing rapid transient start-up from a modified synchronous condenser mode are explored and visually documented. Key parameters relating to a units ability to generate and supply rapid load support are identified, along with potential operational concerns. The possibility of operating large scale turbine units in such a mode as to enable the provision of frequency control services to the grid may provide a valuable source of revenue and operational flexibility to utilities and minimise the need to operate units at damaging off-design conditions.

The analytical model developed provides an improvement to currently accepted dynamic response models by replacing the simplified, somewhat arbitrary, formulation for turbine power output with a consideration of the actual mechanism of energy transfer, namely the conservation of angular momentum, and machine geometry. The new analytical model is shown to consistently outperform current models, particularly — and importantly — at off-design conditions where all previous models reported in the literature perform poorly. The model is then extended to allow for simulation of the proposed method of rapid start-up which known current models simply cannot analyse. The simulation results are validated against full-scale test data and are shown to capture the major characteristics of the instantaneous power output observed at full-scale during the rapid transition. The model provides a more comprehensive, physics based, and adaptive formulation which is used to predict the effect of various guide vane opening profiles on response during the proposed start-up procedure.
Chapter 2

Literature Review

Due to the changing demands of an evolving modern electricity market, it is not uncommon to see hydropower units not only being operated for extended periods of time away from the plant design point, but also for the operating point to be frequently adjusted in order to meet the variable demands of the energy market. In terms of pumped storage power plants this can also involve switching between generation and pumping modes of operation.

This chapter will provide a review of various aspects of hydro turbine operation, particularly Francis turbines, concerning the transient response and behaviour of units away from design point. Operation of Francis turbines at off-design operating conditions is characterised by a high degree of swirling flow in the draft tube cone which has been the topic of much research in recent years, both experimental and numerical in nature. The study of the transient behaviour of hydro turbine units has largely been the domain of simulation models, starting with the 1973 IEEE Committee Report [49] on dynamic models for steam and hydro turbines in power system studies. A range of improvements that have been made to the original mathematical model over the years are discussed, however, critical limitations still exist. Finally, a general discussion on turbines operating in synchronous condenser mode, a plant requirement for the proposed rapid start-up method, is given as well as a presentation of the limited reported experience to date in transitioning a unit from synchronous condenser to generation mode.
2.1 Behaviour of Francis turbines at off-design operating conditions

In terms of singularly regulated (fixed pitch) reaction turbines, operation at off-design conditions results in a strong swirling flow component at runner outlet due to a misalignment between the guide vanes and runner blades. This swirling flow seen at the inlet to the draft tube is then decelerated in the draft tube cone and a highly complex and unstable flow condition arises. The rotating helical vortex, or draft tube vortex, which, depending on the Thoma cavitation number $\sigma$ may be seen to be cavitating in its core as shown in Figure 2.1, precesses in the draft tube and induces an unsteady rotating pressure field leading to potentially large pressure pulsations [20].

The additional compliance introduced by the cavitating vortex can result in a coupling between the natural frequency of the hydraulic circuit and the precession frequency which leads to draft tube surge, hindering the operation of the turbine and the entire plant [20]. As such the precessing helical vortex has been the subject of extensive research, both experimental and theoretical, in relation to the development and characterisation of the phenomena, the subsequent effect on the hydropower plant, as well as various methods aimed to mitigate or eliminate the development of the draft tube vortex.

Francis turbines operating at partial discharge are prone to the development of the precessing draft tube vortex. As the swirling flow is decelerated in the draft tube, vortex breakdown can occur above a certain value of swirl number [97]. The swirl number, $S$, as defined by Susan-Resiga et al. [96] relates the flux of swirl momentum in the axial direction divided by the flux of axial momentum in the axial direction as given in Equation 2.1

$$S = \frac{\int_0^{R_0} \rho u w(r) rdr}{\int_0^{R_0} (\rho u^2 + p - p_{R_0}) rdr}$$  \hspace{1cm} (2.1)

where $R_0$ is the survey section radius, $u$ is the axial velocity component, $w$ the circumferential component and $p_{R_0}$ the pressure at the wall.

This vortex breakdown leads to the formation of a precessing helical vortex which is recognised as the main cause of pressure fluctuations within the draft tube during part load operation of Francis turbines.
2.1.1 Experimental testing of scale model Francis turbines

Model testing continues to be a useful and much relied upon method of analysis for investigating flow phenomena in hydropower turbines and hydraulic structures. Even with recent developments and the level of sophistication of current computational fluid dynamics (CFD) packages, physical model testing remains a trusted and proven tool for use in design, research and the within the hydro industry at large [89]. Nowadays, with the cost effectiveness of CFD and the proven principles of physical model testing, particularly in certain situations where CFD is not yet thoroughly validated, both tools are being increasingly used in parallel to optimise component designs and investigate particular phenomena with great success. This is particularly the case in studies of transient and two-phase flow behaviour of hydraulic turbines where numerical models are simply not yet proven. The following gives a general overview of some noteworthy experimental investigations examining various aspects of Francis turbine behaviour including details on experimental facilities and techniques used.

Kirschner and Ruprecht [57] show that, for a constant speed equal to that of the best efficiency point (BEP), by varying the turbine discharge a spike in normalised wall pressure fluctuation, $\Delta \Psi = \frac{2 \rho}{R^2}$, is seen at approximately 72% discharge. It was observed that for operating points at constant speed below $Q_{BEP}$ the precession of the vortex was in the same
Figure 2.2: Simulated and measured part-load draft tube mean velocity profiles of tangential velocity component (top left), radial velocity component (top right), axial component (bottom left) and absolute velocity (bottom right) [20].

direction while for operation at 111% $Q_{BEP}$ the cavitating core displayed a straight central shape and at 135% $Q_{BEP}$ both the corkscrew shape and direction of rotation was reversed when compared to the vortex structure at operating points below $Q_{BEP}$.

Kubota [60] provides extensive visual observations of the draft tube flow for a low specific speed homologous scale model pump-turbine over the entire four-quadrant operating range. The $D = 0.285$ m model unit, developed for a 500 m head pumped storage unit, is tested at various operating points under steady conditions in each of the four quadrants as well as during pump start up from a depressed water level condition. Flow visualisation is achieved not only in the draft tube cone but also through the stay and guide vane flow passages with runner band, bottom case cover and draft tube cone all constructed from transparent acrylic resin. Using a stroboscope the flow phenomena was captured with a high speed camera capable of 3,000 frames per second with a small amount of air injection from upstream used to aid in the visualisation.

Numerous experimental investigations have been conducted under the FLINDT (Flow investigation in draft tube) project at the Laboratory for Hydraulic Machines (LMH) at the École Polytechnique Fédérale de Lausanne (EPFL). Arpe et al. [5], Ciocan et al. [20], Iliescu et al. [51], Susan-Resiga et al. [96] and Nicolet et al. [79] have conducted studies looking at
various aspects of draft tube flow on the same high specific speed\(^1\), \(n = 0.56\), scale model Francis turbine unit at the EPFL facility. For the first time, Ciocan et al. [20] presents a numerical simulation of the draft tube rotating vortex structure under cavitation free conditions validated against experimental measurements of the detailed flow field, presented in Figure 2.2. Previous studies by Ruprecht et al. [88], Scherer and Aschenbrenner [90], Miyagawa et al. [70] and Sick et al. [92] had only provided validation against the global flow characteristics such as precession frequency and pressure pulsations, however, with good agreement. The unsteady Reynolds-averaged Navier-Stokes (RANS) simulation results at partial discharge presented by Ciocan et al. [20] are compared to 2D laser Doppler velocimetry (LDV), 3D particle image velocimetry (PIV) and unsteady wall pressure measurements conducted on a scale model Francis turbine with the purpose built FLINDT elbow type draft tube at a chosen operating point of approximately 70% \(Q_{BEP}\). For a given operating point, defined by specific hydraulic energy and flow coefficients \((\psi, \varphi)\), the pressure pulsation amplitude, precession frequency and the vapour volume in the vortex is shown to be dependent on \(s\). The experimental investigation was carried out for a range of Thoma cavitation numbers varying from \(s = 1.18\) (cavitation free) to \(s = 0.38\) (maximum rope volume) however as the Ciocan model was limited to single phase flow, only the cavitation free condition \(s = 1.18\) was validated [20]. In terms of global quantities, the precession frequency of the draft tube vortex was found experimentally to be 0.3 times the runner rotation frequency, while numerically it was found to be 0.34 runner rotational frequency, 13% higher than experiment. The phase averaged wall pressure measurements are in good agreement throughout the draft tube measurement section with the difference in mean pressure level less than 2.5%. In regards to local flow structure the mean velocity field, phase averaged velocity field and vortex centre position show good agreement across each form of measurement.

Using the same scale model turbine and operating point (as Ciocan et al. [20] above) Iliescu et al. [51] successfully employ two-phase particle image velocimetry (PIV) to "reconstruct" the helical vortex rope volume and position, and quantify the corresponding velocity field for a range of \(s\) values from \(s = 1.18\) (cavitation free) to \(s = 0.38\) (maximum volume), illustrated in Figure 2.3. The rope diameter and the relative position of the rope centreline are determined by image processing techniques, and the influence of turbine setting level on these parameters are investigated. This study represents the first time that the development of the cavitating rope in a Francis turbine scale model is quantified. The rope precession is detected by wall pressure measurement within the diffuser cone, which also has the benefit of working for non-cavitating flow, and it is this detection of the pressure pulsation

\[ v = \frac{\omega (Q_1 / \pi)^{0.5}}{(2E)^{0.5}} = 20.25 n^{0.5} n_{QE} \]
Figure 2.3: PIV overlay of velocity fields at different $\sigma$ values [51]

in the draft tube that triggers the PIV rather than runner rotation as the vortex precession frequency can change over one revolution [51, 57]. Ensemble averaging on phase is used to obtain mean velocity fields and cavity volumes for varying $\sigma$ values. The position and diameter of the rope in phase average are also calculated for various $\sigma$ values where the associated standard deviations of rope diameter are attributed to axial pressure waves acting on the cavity due to changes in local pressure distribution. These variations in diameter were relatively constant except for the case at $\sigma = 0.38$. At this value an eigenfrequency of the hydraulic system is excited, as shown by Nicolet et al. [77], resulting in the propagation of plane waves which consequently affect the cavity volume as they pass. A similar instability is seen in the rope centre position at this $\sigma$ value which is shown to be more variable in comparison to other settings.

Nicolet et al. [79] looks particularly at the upper part load vortex rope, defined as being between 0.7 and 0.85 times $Q_{BEP}$, where pressure fluctuations of 2 to 4 times the runner rotational frequency can be seen. Furthermore, at this setting the cavitating rope forms an elliptical cross-section which has its own self rotation. Kirschner and Ruprecht [57] also observe this elliptical vortex cross-section at 83% $Q_{BEP}$. 
While the vast majority of research has dealt with the cause, physics and consequences of the single helical draft tube vortex, Wahl [104] provides a study on the existence of a twin vortex flow structure using a 0.23 m scale model of the Grand Coulee Francis turbines with supporting measurements from the prototype unit. On the scale model, a sustained twin vortex was visually observed at 12 separate operating points, as shown in Figure 2.4, for guide vane openings between 26 - 44% of the maximum within the equivalent prototype operating head range, and were found to exhibit a higher frequency surge than the single vortex while also reducing the amplitude of pressure fluctuations.

2.1.2 Mitigation Techniques

Thicke [99] provides a review of various practical methods for the control of flow instability in the draft tube cone arising from operation at part load and advocates the use of runner cone extensions. Thicke [99] also evaluates various stabiliser fin arrangements attached to the walls of the draft tube as being beneficial to mitigate the swirling flow and associated pressure fluctuations, however a loss in efficiency (for low to medium head units) results. A study by Nishi et al. [80] showed that the natural frequency of the draft tube vibration system cannot always be changed with the installation of fins and the addition of fins can potentially help promote resonance due to the increased cavity volume generated downstream of the fins at low cavitation number. Various other similar solutions have been proposed however the addition of any physical structure into the flow passage for the pur-
pose of interrupting or suppressing the helical vortex incurs a loss in efficiency, particularly as the operating point is moved away from the partial discharge condition where the flow instability is no longer present.

A commonly used method for surge suppression is air admission either via the runner cone or directly into the draft tube using a jet pump, diverting water from the spiral casing, or via a compressed air supply. The injection of air into the central stagnated region produces a more stable axisymmetric air core which is still surrounded by the swirling main flow. This method of injecting air into the flow passage to relieve the effects of the helical vortex is long established and been shown to have a relatively small influence on overall machine efficiency while reducing the draft tube pressure fluctuations. By increasing the cavitation compliance the method of air injection moves the mass oscillating system within the draft tube away from the precession frequency of the helical vortex and thus reduces the risk of resonance being established. However, it is noted that this method does not actually eliminate the helical vortex, it only reduces the severity of its effects. Similarly Susan-Resiga et al. [97] cite work by Blommaert [13] whereby a small amount of water is injected into draft tube cone using a rotating valve to provide a forced excitation in order to offset the self-induced pressure fluctuations of the vortex rope. Again this method only reduces the effects of the pressure fluctuations rather than eliminating the source.

As \( \varphi \) is reduced below \( \varphi_{bep} \) the specific energy downstream the runner near the crown is reduced, creating a pumping effect near the crown. This is due to the residual swirl component and results in the establishment of the central quasi-stagnant region. Susan-Resiga et al. [97] aim to eliminate this quasi-stagnant region, which can lead to vortex breakdown and the formation of the helical vortex, by the injection of a high energy water jet directly into this region through the runner cone in order to accelerate flow within the region. This approach aims to avoid vortex breakdown altogether rather than controlling the pressure fluctuations resulting from the precessing spiral vortex following vortex breakdown. The method is analysed using 3D unsteady flow numerical simulations on a model that had been previously verified in respect to the vortex rope precession frequency and draft tube wall pressure fluctuation amplitude at cavitation free conditions \((r = 1.18)\) in the work by Arpe [4]. The simulation was carried out at the same operating point as the aforementioned study, and while not verified experimentally, the method seems to suggest the ability to reduce the magnitude of draft tube pressure fluctuations by at least one order of magnitude while not dramatically effecting the overall turbine efficiency. The method requires a small percentage of flow to be diverted from the spiral casing to supply the jet which is therefore unable to contribute to power generation and results in a corresponding efficiency loss. However at the part-load operating point considered, this efficiency loss, at an undisclosed jet flow rate, was in the order of 0.2%. 
Muntean et al. [72] experimentally evaluates the axial water jet mitigation technique proposed by Susan-Resiga et al. [97] by way of calculating the pressure recovery coefficient and relative root-mean-square pressure fluctuation coefficient on a specifically designed divergent test section with upstream swirl generator. It was found that a 10% jet discharge is required to completely mitigate the pressure fluctuation attributed to the precessing vortex and achieve maximum pressure recovery within the test section studied. The requirement for a flow-feedback of 10%Q does introduce some practicality issues however for larger installations.

2.2 Transient modelling of Francis turbine power plants

There has long been a focus on the ability to accurately model and predict the dynamic behaviour of hydro power plants and numerous models have been developed in the past to simulate various operational scenarios, to varying degrees of accuracy, in order to determine safe operating limits. In 1973 an IEEE Committee Report was published reviewing and compiling a number of models for speed-governing systems as well as turbines in power system stability studies currently being used by utilities in order to standardise the nomenclature and provide guidance on when models may and may not be used. The models given in the report cover speed governing systems and turbine models for both steam turbine and hydraulic turbine systems [49]. While these early models were considered adequate for first swing stability analysis it was acknowledged, even at the time, that more detailed models were required, particularly for the study of longer transient stability simulation up to and beyond 10 seconds, low frequency oscillations, islanding and isolated system operation, load rejection, load acceptance, water hammer dynamics and pumped storage generation with complex hydraulic structures [50].

The relatively recent restructuring and liberalisation of electricity markets around the world has seen an increased effort to produce more accurate dynamic models to improve control systems, estimate stability limits, predict restoration scenarios and increase overall performance of installed plants [37, 55, 106].

The 1992 IEEE Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies [50] presents some simple linear and nonlinear hydraulic turbine models taking into account both inelastic and elastic water column (the travelling wave, or water hammer effect). The report also examines models which include surge tank effects and models with multiple penstocks supplied from a common intake tunnel. The models given in the 1992 IEEE report have since been used extensively in the literature.
and in industry, often as a base model requiring refinements depending on the nature of the simulation [23, 36, 55, 74, 106]. While this conventional model addressed some of the shortcomings of earlier models it still has some serious drawbacks which are discussed extensively by Ng [74], and [75].

Hannett et al. [36] test the validity of the simple turbine model presented in the 1992 IEEE report [50] by comparing the output of the model simulation with field tests conducted on hydro units at three separate stations. The recommended transfer function for mechanical-hydraulic governors as given by Ramey and Skooglund [85] was found to be adequate in representing the governor controls, however the model structure of the turbine and hydraulic dynamics of the penstock as given by [50] was found to not fully represent the full characteristics of the units as indicated by the field tests.

Hannett et al. [36] offers two main model refinements; the first was to introduce additional damping for no load conditions as a result of discrepancies in observed and simulated speed excursions following a 25% load rejection, the second was to correct for the assumption of a linear turbine characteristic over the full range of guide vane position present in the IEEE model. Hannett et al. [36] provides evidence of this nonlinear behaviour in the form of simulated 5 MW load disturbances from two different initial loadings. Two alternative revisions are offered to incorporate this nonlinear characteristic depending on what data is available for the system being modelled.

Similarly De Jaeger et al. [23] also propose a nonlinear turbine model supported by field testing of load rejection for both the single unit case and the case with multiple turbines sharing a common tunnel. Hannett et al. [37] later re-examine both the IEEE Working Group [50] and De Jaeger et al. [23] models for multiple units for a system in which dissimilar turbines with different characteristics share a common tunnel and also consider the stability under islanding and black start operation where the traditional governor tuning criteria may result in unstable frequency control where dissimilar units are concerned.

The idea of modelling a multiple turbine system in regards to oscillatory behaviour was again visited by Jones [55] in examining the Dinorwig pumped storage hydroelectric power station which is used extensively for system frequency regulation. Jones [55] however uses a simpler linearised model assuming incompressible fluid flow and neglecting friction losses to highlight that the reduced stability and oscillatory behaviour is a direct consequence of loop interaction rather than higher order dynamic characteristics of the complex Dinorwig system. While linearised models are regularly used in the power industry and generally adequate for small perturbation studies they are not suitable for large frequency or output disturbances for example and are not recommended for time domain analysis [50].
Nicolet et al. [78] introduce a new simulation package for the purpose of modelling power networks under transient modes of operation developed in-house at the École Polytechnique Fédérale de Lausanne (EPFL). The code is based on the impedance method and uses electrical analogies to model system components representing pipe sections, valves, surge tanks and Francis turbine units as assemblies of RLC components with the piezometric head and discharge corresponding to the circuit voltage and current respectively. The resulting set of differential equations describing the electrical, hydraulic and regulation system are then solved simultaneously ensuring the interaction between each system is fully captured. Early simulations focus on the ability to reproduce mass oscillation and wave propagation in hydraulic systems and the ability to accurately model a francis turbine unit under transient conditions. A simulation of a 30% load perturbation is presented and shows good agreement with simulation results given by Wylie and Streeter [111].

The model is further developed in a study investigating pressure fluctuations of a scale model at part load operation to include a draft tube composed of two separate components and the introduction of a pressure source driven by the hydraulic characteristics of the turbine rather than a pure resistance, see Nicolet et al. [77]. The distinguishing feature of the SIMSEN draft tube model however is the distribution of the vortex rope compliance, $C_{rope}$, along the length of the draft tube based on experimental data. Additionally, an attempt is made to transpose draft tube parameters from scale model to full-scale plant. While the inductance terms are functions of geometry only the capacitance terms are dependent on the value of local compliance for which it is noted that Froude and Thoma cavitation number similitude must be obeyed if $C_{rope}$ is to be transposed.
2.3 Operation in Tail Water Depression (TWD) mode

Many hydroelectric units have the capacity to allow operation in a synchronous condenser (SC) mode, the feature allowing the supply of reactive power while motoring the generator/turbine with the tail water depressed [15, 64, 66, 68, 112]. In SC mode air is forced into the runner cavity such that the runner is able to spin at synchronous speed in air rather than water thus decreasing the friction on the runner and reducing the amount of power input required from the system. It is also noted that in SC mode the hydro unit typically operates with the main inlet valve (MIV) closed (unless there is no MIV in the case of lower head Francis turbines with guide vane seals) and the governor solenoid de-energised.

Historically this mode of operation has been employed in order to control reactive power and provide voltage support to the system [34, 68]. Due to the requirements of the market at the time, in the late 1970s to early 1980s there was a strong interest in the behaviour and operation of turbine units in synchronous condenser mode [15, 66, 112, 114]. The main focus of study during this time however was concerned with the need to efficiently dewater a pump-turbine unit for preparation of operation in SC mode.

Both Yamaguchi [112] and Zanetti and Rossi [114] reported from observations of model testing that delivered air became entrained in the flow and was transported out through the draft tube elbow. Regarding this, it has was found that there exists a certain critical flow rate, below which successful water level depression cannot be achieved regardless of the volume of air admitted or the time taken [66, 112, 114]. Furthermore, higher flow rates have been shown to depress the water level more efficiently using less air and in a shorter period of time [112]. The critical flow rate has been found to be dependent of a number of factors including machine speed (rpm), air admission hole location(s), inlet guide vane (IGV) opening, penstock pressure and possibly also the cavitation coefficient (depending on the position of air inlets) [112].

The position of the compressed air inlets has a significant effect on the ability to successfully depress the draft tube water level in pump-turbine machines. In tests conducted using the air admission set-up as shown in Figure 2.6, with inlet holes at the runner hub (A), between the IGVs and the runner (B) and the draft tube wall (C) the position requiring minimum critical flow rate was found to be at A, with position B requiring the highest flow rate [112]. Critical flow rate from a combination of both A and C was found to be between that of A and C alone. A similar study, using a model pump-turbine mounted on a frame for measurement of thrust, reported findings supporting the favourable admission from position A adding that a sudden axial thrust was observed with air admission from position B [114].
While operating in SC mode a wave-like phenomenon has been observed to appear on the free surface on the tail water below the runner of various pump-turbine models, the presence of which was also confirmed in a full-scale unit [15]. The effects on amplitude and frequency of the wave were investigated in relation to draft tube geometry, water level depression, operating speed and water leakage. The study was performed to gain understanding of the effect on the safety margin regarding possible interference of the wave with the runner and delivered air exiting through the draft tube elbow.

The main results of the study can be summarised as follows:

- The behaviour of the free surface is independent of draft tube shape and depth
- Water depression level is the major parameter influencing wave amplitude
- Wave amplitude decreases as water level approaches the runner, disappearing at a distance of 10 - 15% that of the outlet diameter
- Maximum wave amplitude occurs at a distance equivalent to one diameter from the runner
- Decreasing the water level to a distance below that of one diameter has no further influence on the wave amplitude
- A limiting speed exists, generally below rated speed, below which the wave is not primed
- For speeds over the limiting speed the wave priming is a function of water level
• For speeds over 1.25 times the limiting speed the wave is always present
• Water leakage into the draft tube causes a slight feathering of the wave surface
• Frequency of the wave is constant and does not depend on the machine speed

In recent years there have been many static var devices installed making SC mode rarely used. However, operation in a modified SC mode may also enable very rapid machine start-up [15, 64] and it has been demonstrated that hydro machines are successfully able to provide additional fast raise FCAS when operating in this pseudo synchronous condenser mode [41].

The following main modifications to SC mode are required in order to facilitate rapid machine start-up:

• The main inlet valve (if present) remains open, the water instead being held by the inlet guide vanes.
• The governor remains active, keeping the guide vanes in the closed position unless triggered to operate.

When required, due to a drop in the system frequency below a nominated level and/or the detection of a high rate of change of frequency, the inlet guide vanes are rapidly opened, flooding the runner and expelling the air volume through the draft tube, and with some delay (typically 1-2 sec), delivering rapid reserve power to the system. This new modified synchronous condenser mode, identified as tail water depression mode, will hereafter be referred to as TWD mode.

The proposed TWD mode of operation for the purpose of providing frequency control is not entirely new however; in fact one of the primary roles of the Dinorwig pumped-storage hydroelectric power plant is to regulate system frequency of the British national grid [65]. The Dinorwig plant consists of six 317 MW Francis type pump-turbines with the ability, when operating as spinning reserve, to be brought from zero to full power output in 10-15 seconds [65, 71].

Relatively little has been written regarding the transition of pump-turbines between operation in SC mode to generation mode, particularly concerning the rapid change over where the output in the first few seconds is of importance. Much of the literature concerning the provision of load relief and frequency response by spinning and non-spinning reserves focuses on the supply of load which must be deployed to at least 50% within 15 seconds.
Figure 2.7: Investigated locations of air and water inlets and outlets for operation in synchronous condenser mode as presented by Zanetti and Rossi [114]

from the contingency event and must be fully operational within 30 seconds as defined by the European Network for Transmission System Operators for Electricity (ENTSO-E) [28, 69].

The previously mentioned study by Zanetti and Rossi [114] also consider various methods of air evacuation and report that the location of air outlets was found to influence the reliability of air exhaust and runner start-up following operation in synchronous condenser mode. The most effective location of the air outlet was found to be through the crown of the runner, position A (refer to Figure 2.7), as this is where the remaining air collects and remains for the longest [114]. The study however noted that the filling (air exhaust) operation should occur so as to take up the load as gradually as possible and as such sudden air expulsion by way of air exiting the cone via the draft tube itself was not considered.

More recently, a study by Magsaysay et al. [64] on the rapid load response of one of the Dinorwig power plant 330 MVA units reports the observation of a reverse power flow into the machine upon transition from synchronous condenser mode which lasted for 2.7 seconds with a maximum magnitude of approximately 40 MW. A similar power dip has also been reported on a 325 MW unit at the Ohkawachi Power Plant, Japan where the negative power flow was found to last for up to 60 seconds with a maximum magnitude of 60 MW [64]. In this case the power reversal is attributed to a decelerating torque applied to the pump-turbine by the initial surge of water hitting the runner as the inlet guide vanes are opened. This deceleration results in a change in the electrical load angle causing the machine to momentarily be driven as a motor [64].

No mention is made of the precise time at which the tail water level impinges on the runner
or how much of a role this plays in the power reversal. It is stated, however, that water is only allowed to enter the runner chamber once the air is expelled, presumably via an air outlet valve, indicating that the rising tail water impingement on the runner is the primary cause.

The work by Magsaysay et al. [64] looks at the power dip observed by bringing a hydro unit online rapidly from TWD mode and considers the possible negative effects such a reverse power flow could have on a power system, particularly isolated or low inertia systems, in the form of large voltage and frequency drops [64]. If unmanaged, these frequency drops can have considerable consequences and could potentially lead to system blackout [16]. A solution to these negative effects of the power reversal is offered by the use of a static frequency converter (SFC) providing an asynchronous link to the network [64]. The solution however only mitigates the frequency problems associated with the power reversal and does not actually eliminate the reversal. Due to the energy required to power the SFC the use of the SFC actually extends the time required to compensate for the power reversal, thus reducing the ability of the unit to provide fast frequency response (in terms of the Australian system requirements), illustrated in Figure 2.8.

Mansoor et al. [65] and Muñoz Hernández and Jones [71] both report negative power flow following a step load increase and consider the decreased power generation in this case a result of the requirement for part of the available mechanical power to be used to accelerate the water column as the inlet guide vanes are opened, causing a drop in pressure at the turbine inlet and a subsequent decrease in generated power.
The existence of such negative power flow behaviour following a step load increase is well documented. Indeed, Jones et al. [56] have proposed a standard specification of responses for a hydroelectric power station operating in frequency-control mode in which following a step increase the negative power excursion (denoted $P_6$) and the time at which positive power generation begins (denoted $t_{p7}$) are assigned as critical test criterion, see Figure 2.9. This non-minimum phase (NMP) behaviour of hydroelectric power plants is well known and understood to be a key physical limitation in the rapid response of hydropower systems [71]. This phenomenon is well documented by Jones [55] among others.

It should be noted however that the studies by Mansoor et al. [65] and Muñoz Hernández and Jones [71] are both concerned with the step response of a hydro turbine unit operating at part load for the purpose of continuous frequency regulation, not the supply of instantaneous reserve following a contingency event by the transition from TWD mode to generation.

To the best of the author’s knowledge there has been no published work to date, either experimental or simulation based, to focus on the minimisation of, or the influential factors involved in, the size and duration of the observed negative power flow in regards to the rapid transition of a hydro turbine from TWD to generation mode.

For the low inertia Tasmanian system in the Australian energy market where the contribution to fast R6 FCAS is based on average energy delivered during the first 6 seconds from the time of the frequency trip, it is critical to develop a deeper understanding of the parameters involved with the observed power reversal in order to reduce its severity and maximise the potential for fast raise FCAS contribution.
Chapter 3

Field Study:
Reece Number 2 Machine FCAS Tests

The results of field studies designed, and carried out, by Hydro Tasmania on the Reece Power Station Number 2 machine in October 2008 during initial feasibility investigation and in February 2012 following governor re-tuning are presented. The results of the initial feasibility investigation were supplied by Hydro Tasmania and provided the starting point for the present study.

The purpose of the initial tests was to experimentally determine the transient response of the Reece Francis turbine unit, both from no-load and tail water depression mode, while also providing useful data to validate hydraulic models and identify key model parameters. Two additional TWD rapid start-up tests are presented from the 2012 return to service test series examining the effect of an increased inlet guide vane opening rate.

This chapter introduces the Reece Power Station, describing general station specifications and the identification and suitability of the plant for the proposed rapid start-up procedure. Details of test instrumentation and procedures associated with the aforementioned test series are given as well as a summary of the test results as found both before and after governor re-tuning, identifying improvements made and the potential for future upgrades. Findings from the field studies presented in this chapter are later used to validate simulation results generated from a new hydraulic model developed in Chapters 7 & 8.
3.1 The Reece Power Scheme

The Reece Power Station, located approximately 11.5 km inland on the West Coast of Tasmania, is the last and lowest of four power stations in the Pieman River scheme. The Pieman River scheme catchment comprises the Pieman River as well as two major lakes, Lake Mackintosh and Lake Murchison, and is the most recent of the state’s Hydropower developments with the final stage of development, the Tribute Power Station, completed in 1994.

The Reece Power Station was commissioned in 1987 and houses two identical vertical axis 116 MW Francis turbine units. The hydraulic circuit of each unit consists of independent intake structures and main gates for controlling inflow into each of the machines via dedicated and virtually identical 250 metre long power tunnels as seen in Figure 7.1. Each power tunnel has a 5.80 m diameter concrete lined section followed by a 5.80 m diameter horizontal steel lined section.

The hydraulic circuits were designed without a surge tower or chamber. Furthermore, no main inlet values are installed on either unit, instead guide vane seals are fitted. The hydraulic circuit of the station is discussed in further detail in Section 7.1. Details of the two hydro generating sets at the Reece Power Station are given in Table 3.1 below [42]:

![Aerial view of the Reece Power Station](image)
Table 3.1: Reece Power Station design specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Generator Manufacturer</td>
<td>Fuji Electric</td>
</tr>
<tr>
<td>Date of Manufacture</td>
<td>1984</td>
</tr>
<tr>
<td>Rated Alternator Output</td>
<td>136 MVA</td>
</tr>
<tr>
<td>Synchronous Speed</td>
<td>166.7 rpm</td>
</tr>
<tr>
<td>Turbine Type</td>
<td>Francis</td>
</tr>
<tr>
<td>Rated Turbine Output</td>
<td>115.6 MW</td>
</tr>
<tr>
<td>Flow at Rated Output</td>
<td>142 m³/s</td>
</tr>
<tr>
<td>Guide Vane Stroke</td>
<td>311 mm full stroke</td>
</tr>
<tr>
<td>Opening Time under maximum rate</td>
<td>13.4 seconds (fully closed to fully open)</td>
</tr>
<tr>
<td></td>
<td>16.5 seconds from Dry Opening Tests</td>
</tr>
</tbody>
</table>

### 3.1.1 Plant identification and suitability

In general most hydro machines are capable of providing some degree of fast raise FCAS by rapidly increasing machine output from either no-load or low load operation. However operating at no-load (with just enough flow to overcome machine losses) wastes valuable water resources; while operation at low load, particularly within the so-called rough running zone, is associated with excessive machine vibration and decreased asset life and therefore should be avoided whenever possible [30, 81, 102].

The proposed option of a machine operating in a modified synchronous condenser (SC) mode so that it is active and ready to be brought on-line if and when required would eliminate the inefficiencies and risk of machine damage associated with extended operation at no-load and low load output.

For a general description of hydro units operating in synchronous condenser mode and the required modifications to enable rapid generation start-up refer to Section 2.3. In short, a modified synchronous condenser mode has the turbine unit spinning at synchronous speed with an evacuated runner chamber such that there exists a free surface, the tail water level, within the draft tube cone region. The guide vanes are in the closed position but remain active and ready to be opened when triggered, the governor being temporarily by-passed. This modified synchronous condenser mode has been termed tail water depression (TWD) mode (Section 2.3).
For a machine to be able to operate in TWD mode there are a number of technical requirements and design features that must be met. The essential requirements are:

- An efficient method for dewatering the draft tube, and
- Sealed guide vanes enabling operation with main inlet valve open (if a MIV is present) to prevent excessive water leakage.

The potential for a given station to provide fast raise FCAS is increased if the following design criteria are met [41]:

i. High load factor (above 80%), so that there is a high probability the unit be be in service when required
ii. Short water acceleration time constant, $T_w$
iii. Short guide vane opening time
iv. Heavy, high inertia machine
v. Flexible (PLC) control, and
vi. Short tail race (minimising pressure surges in the draft tube)

The turbine water time constant, $T_w$, defined in Equation 4.7 will be discussed in more detail in Chapter 4 however is it essentially a measure of the time required to accelerate the water column within the penstock to rated flow, $Q_r$, under the influence of the rated head, $H_r$. Generally, a short water time increases turbine response and makes operation much more stable [40].

While larger hydro units were initially considered for their FCAS potential due to the relatively higher possible increase in output, these machines tend to have slower guide vane opening times, often upward of 15 - 20 seconds, and as such are unable to achieve significant FCAS contribution within the specified R6 window of 6 seconds. Following a survey of all of Hydro Tasmania’s assets the Reece Power Station was identified as a potential candidate for TWD operation.

The Reece Power Station has a simple hydraulic circuit with a short, single machine dedicated penstock and was originally designed for operation in synchronous condenser mode so it meets the first two essential requirements. Furthermore the station currently operates
at between 75 - 80% load factor has a water acceleration time constant of 1.65 seconds and a relatively short guide vane opening time. The Reece unit's actuation time, as designed, was 16.5 seconds (dry opening tests) although governor re-tuning has since significantly improved this and will be discussed in Section 3.4 below.

The Reece unit is fitted with two radial nozzles that protrude into the upper draft tube cone to provide air admission. Air is forced into the draft tube by two water jet pumps taking high pressure water from the spiral case. These are used during low load operation to mitigate rough running and pressure pulsations within the draft tube due to the formation of vortices below the runner, and also to depress the tail water in preparation for synchronous condenser operation. During synchronous condenser mode the draft tube water level is maintained within a 500 mm band by intermittent jet pump operation. Three measuring probes are located within a stand pipe, tapped at the top and bottom, into the draft tube. The low level probe is located 1.85 m below runner outlet while the high level probe is 1.35 m below the runner. A high level alarm probe is located 0.25 m above the high level indicator.

3.2 Experimental measurements

This section provides a brief summary of Hydro Tasmania measurements taken during the 2008 and 2012 test programs specifically relating to the rapid start-up potential of the Reece Francis turbine number 2 machine [42, 116].

Guide vane (GV) movement is measured in terms of the linear servomotor stroke, $S$ (mm), of the guide vane mechanism actuation arms. The arrangement of the Reece unit's guide vane assembly consists of two arms linked to a common ring. During movement one arm moves in a positive direction, the other in a negative direction. The reported values are taken from actuator arm C1 moving in the positive sense from closed to open. Maximum guide vane servomotor stroke is 311 mm however internal stops limit this in practice to 291 mm (or 93.2%). The maximum servomotor stroke corresponds to a guide vane opening, $A_0$, of 290 mm, the diameter of the largest cylinder able to pass through the fully opened guide vane passage.

During transient tests pressure was recorded in the penstock prior to spiral case inlet and also in the upper draft tube. The penstock pressure transducer was located on the upper diagonal, 1.2 m upstream of the spiral casing in the 4.72 m internal diameter section of steel lined penstock. The centreline of the penstock is at a level, referenced to sea level (S.L.) of 3.5 m. The pressure transducer protrudes 150 mm from the penstock giving a pressure
transducer location of S.L. 5.3 m. Prior to load acceptance tests in 2008, TimeStudio\textsuperscript{1} records indicate that the Lake Pieman level was S.L. 95.915 m. The pressure reading at the time gave a measured pressure of 890.2 kPa. From this the transducer offset was calculated as -0.105 m \cite{43}. The draft tube pressure transducer was located at approximately S.L. -0.61 m, 2.47 m below runner outlet. The diameter of the conical section draft tube at the pressure transducer location was calculated to be 4.602 m \cite{43}.

The active power output of the turbine generating unit was calculated from AC voltage and current transducers using a high accuracy three-phase wattmeter as specified by Ng \cite{74} connected to a station telemetry circuit. Three-phase voltage and current signals were stepped down prior to being input in to the wattmeter which gave a standard 4-20 mA current output, subsequently converted to a 0.4-2 V analogue signal of the active power measurement.

The rotational speed of the machine is measured by two digital revolution detection units using a toothed wheel attached to the turbine shaft \cite{39}. Proximity and vibration sensors were mounted in several locations described below to investigate the effect such a sudden loading may have on the machine:

- \textbf{S1a} Upper guide bearing proximity probe
- \textbf{S2a} Turbine bearing proximity probe
- \textbf{A1a} Upper guide bearing accelerometer
- \textbf{A2b} Turbine bearing accelerometer
- \textbf{A1v} Upper guide accelerometer
- \textbf{A4a} Penstock accelerometer
- \textbf{A5a} Draft tube accelerometere

There was no provision for flow rate or shaft torque (mechanical power) measurement during either the 2008 or 2012 test series.

Data acquisition was achieved using a National Instruments DAQCard-6024E, LabVIEW interface and two Yokogawa DL750 data recorder units. All data, for both presented test series, was recorded at a sample frequency of 20 Hz.

\textsuperscript{1}TimeStudio is an open source MATLAB toolbox for time series analysis
3.3 Tests performed during the Reece Power Station 2008 fast raise trials

In October of 2008 fast raise tests were performed on the Reece number 2 machine as part of an initial investigation to determine the feasibility of providing R6 FCAS by means of rapidly starting a unit from TWD mode. Load acceptance tests were performed from both generator No-Load (NL) position and from tail water depression (TWD) mode under various operational conditions as outlined in the test matrix given in Table 3.2. In addition to providing a base set of experimental data on TWD performance the results from these early tests were used to calibrate and validate a Hydro Tasmania hydraulic pressure transient study using the commercial software package HYTRAN. Simulation results were then able to provide guidance on upper limits in regards to the maximum possible guide vane opening rates that could be implemented following the planned governor retuning in 2012. Test results were also used during the development of the dynamic simulation model developed in Chapter 7.

To enable testing in this non-conventional manner the proportional-integral-derivative (PID) components of the governing system were essentially by-passed. Instead, analogue test signals generated using a LabVIEW signal generator program were directly injected into the GV opening controller stage. This allowed tests to be performed at the maximum possible guide vane opening rate given the current system configuration. Prior to start-up tests, guide vane dry stroking tests were undertaken to confirm current opening times from fully closed to open position. With the PID components disabled the maximum achievable guide vane opening rate results in a 16.55 second opening time while the delay between the injection signal and first recorded servomotor movement was less than 150 ms. However it was also established that the delay between the drop in system frequency below the minimum NOB level and first GV movement would be of the order of 0.5 seconds. This effectively reduces the R6 FCAS contribution window to 5.5 seconds.

Stroking tests also revealed the two stage nature of the guide vane servomotor movement. This is due to an inbuilt mechanical damping, a type of orifice port within the piston, the purpose of which is to cushion the guide vane movement. This cushioning is in place to minimise the generation of hydraulic transients, particularly during the closing stroke, however the opening movement is also affected. This behaviour can be seen in Figure 3.2 during actual start-up tests. The change in opening rates can be seen in both opening from no-load and synchronous condenser mode at approximately 75 mm servomotor stroke.

From the dry stroking tests the maximum obtainable opening rates were determined to be 19.2 mm/s for normal movement and 10 mm/s during cushioning.
Table 3.2: 2008 Reece 2 Raise Testing Summary [42]. Tests were performed from no-load (NL) and from tail water depression (TWD) mode. For NL tests the initial guide vane servomotor position ($S_0$) is 34.4 mm while for TWD raise tests guide vanes are initially fully closed.

<table>
<thead>
<tr>
<th>Test ID</th>
<th>Description</th>
<th>$S_0$ [mm]</th>
<th>Additional notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>NL1</td>
<td>Slow raise test from NL</td>
<td>34.4</td>
<td>0 MVAr</td>
</tr>
<tr>
<td>NL2</td>
<td>Fast raise test from NL</td>
<td>34.4</td>
<td>0 MVAr</td>
</tr>
<tr>
<td>NL3</td>
<td>Fast raise test from NL</td>
<td>34.4</td>
<td>+40 MVAr</td>
</tr>
<tr>
<td>$T_R1$</td>
<td>Slow raise test from TWD</td>
<td>0</td>
<td>0 MVAr</td>
</tr>
<tr>
<td>$T_R2$</td>
<td>Fast raise test from TWD</td>
<td>0</td>
<td>0 MVAr</td>
</tr>
<tr>
<td>$T_R3$</td>
<td>Fast raise test from TWD</td>
<td>0</td>
<td>0 MVAr, PSS off</td>
</tr>
<tr>
<td>$T_R4$</td>
<td>Fast raise test from TWD</td>
<td>0</td>
<td>0 MVAr, jet pumps active</td>
</tr>
<tr>
<td>$T_R5$</td>
<td>Fast raise test from TWD</td>
<td>0</td>
<td>+40 MVAr</td>
</tr>
</tbody>
</table>

The 2008 test series included three load acceptance tests from initially no-load machine operation and five load acceptance tests from the modified synchronous condenser mode.

In each case a slow raise test was first conducted followed by tests at the maximum rate that was achievable at the time. The guide vane servomotor position following signal injection for the eight start-up tests conducted is shown in Figure 3.2. The no-load position occurs at a stroke of 34.4 mm while the closed position is equivalent to a 0 mm stroke. As the first six seconds is of primary interest, power output for R6 FCAS contribution is only counted for six seconds following a frequency deviation. The stroke was only taken up to approximately 37% before machine operation was stabilised and subsequently lowered in preparation for the following test.

The transition from generator no-load to synchronous condenser mode was achievable within 33 seconds, timed from the command to fully close the guide vanes to the time at which the low level tail water indicator was switched on. Once in the synchronous condenser mode the tail water level was held between S.L. 3.57 m and S.L. 3.07 m through intermittent jet pump operation, triggered by the high level indicator. However there was no way of knowing the exact location of the free surface at the time of guide vane opening during start-up tests $T_R1$ to $T_R5$.

The electrical power output for all the load acceptance tests are given in Figure 3.3. The no-load to generation tests (NL1 to NL3) show a generally constant increase in active power output until the final GV position is reached. The response shows very little over- or under-
Figure 3.2: Guide vane opening profiles tested during the 2008 TWD feasibility study at the Reece Power Station [42].

...shoot as may be expected following fast guide vane movement due to the need to accelerate and decelerate the water column in the penstock. This may be attributed to the short penstock length and water acceleration start time. Minor output fluctuations can be seen, particularly within the first 5 seconds.

Output response from loading the machine from TWD mode (T_R1 to T_R5) are also given in Figure 3.3. First of all, it is established that approximately 1.6 MW of power is absorbed during synchronous condenser mode with tail water depressed below the runner. This power draw, equivalent to 1.4% of rated output, is required to maintain synchronous speed while overcoming frictional and bearing losses of the unit. In general for the fast raise tests (T_R2 to T_R5) the rate of increase in power output is slightly slower than that of the no-load fast raise tests, particularly within the six second fast FCAS window. However the most notable feature of the rapid start-up from TWD mode is the large initial negative power incursion that begins at approximately 1.25 seconds after guide vane movement is initiated and reaches a maximum of 6 MW at just over 2 seconds. The 0 MW output position is not reached until 2.5 seconds after initial guide vane movement. This increased power draw is then followed by a succession of power oscillations that decrease in magnitude as output power is increased. The frequency of oscillations vary from approximately 1 Hz (T_R4) to 1.5 Hz (T_R5). Similar behaviour is also evident in the slow raise test (T_R1) although the
onset of the negative draw is later and the magnitude of oscillation is reduced slightly.

The penstock and draft tube pressure behaviour is given in Figures 3.4 to 3.8 for tests NL1, T_R1, T_R2, T_R4 and T_R5 with GV servomotor position and power output included for reference.

Figures 3.4 & 3.5 show the two slow raise tests, from no-load and TWD mode respectively, and show minimal hydraulic transients in both penstock and draft tube. The draft tube pressure before and after GV movement in the no-load case is seen to fluctuate considerably more than the TWD mode transition due to the high swirl runner exit flow caused by the low angle GV position. Upon start-up from TWD mode the draft tube pressure pulsations are suppressed due to the large cavity of air within the draft tube. In both cases the penstock and draft tube pressures are well within acceptable limits.

Figures 3.6, 3.7 & 3.8 show the pressure transients within the penstock and draft tube resulting from rapid GV movement from TWD mode. In each case the penstock pressure decreases from the pre-test static pressure of 890 kPa down to 790 kPa. A number of reflections are seen indicating a wave period of 0.95 seconds. Most importantly the negative
power draw is seen to occur well after the initial negative pressure spike and the subsequent power oscillations seem to be largely independent of the turbine inlet pressure. While the operation of the jet pumps during transition tends to increase penstock and draft tube fluctuations the general behaviour in each case is very similar and pressure are again well within the safe operating limits of the station.

The displacement and vibration levels at measurements points previously defined are given in Table 3.3. In general machine vibration levels during rapid transition were no more severe than during normal operation in synchronous condenser mode or during slow raise from synchronous condenser to generation.

Table 3.3: Maximum recorded displacement and vibration levels during operation in normal synchronous condenser mode and during slow and fast raise tests [42]. Designation of proximity and vibration sensors was given in Section 3.2.

<table>
<thead>
<tr>
<th></th>
<th>S1a (µm)</th>
<th>S2a (µm)</th>
<th>A1a (mm/s)</th>
<th>A2b (mm/s)</th>
<th>A1v (mm/s)</th>
<th>A4a (mm/s)</th>
<th>A5a (mm/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Synchronous condenser mode</td>
<td>50</td>
<td>43</td>
<td>0.19</td>
<td>0.19</td>
<td>0.11</td>
<td>0.90</td>
<td>0.37</td>
</tr>
<tr>
<td>SC-Gen transition</td>
<td>53</td>
<td>89</td>
<td>0.33</td>
<td>0.54</td>
<td>0.12</td>
<td>5.5</td>
<td>4.7</td>
</tr>
<tr>
<td>Fast TWD-Gen transition</td>
<td>58</td>
<td>89</td>
<td>0.36</td>
<td>0.57</td>
<td>0.11</td>
<td>6.3</td>
<td>4.8</td>
</tr>
</tbody>
</table>

While there is increased penstock and draft tube vibration levels (A4a and A5a) the mechanical duty is actually much lighter than operation at part-load within the rough running zone [41]. These measurements however give no indication of the stresses applied to the turbine runner by the sudden incoming jet of high pressure water.

Results of initial feasibility tests provide proof of concept that fast R6 FCAS can be successfully generated from a modified synchronous condenser mode (TWD mode) to alleviate and remedy large magnitude drops in system frequency following a contingency event. The contribution, however, is somewhat limited due to a large additional power draw upon transition.

The GV servomotor position, active power and calculated R6 FCAS contribution is summarised in Table 3.4 and shown graphically in Figure 3.9. The unfavourable influence of this negative power draw is clearly evident with FCAS not being provided until 3 seconds after initial GV movement. This leaves only 2.5 seconds for FCAS contribution at which stage only 6.3 MW has been generated.
Figure 3.4: Unit response during slow raise test from no-load (NL1) [42].

Figure 3.5: Unit response during slow raise test from TWD mode (T_R1) [42].
Figure 3.6: Unit response during fast raise test from TWD mode ($T_{R2}$) [42].

Figure 3.7: Unit response during fast raise test from TWD mode ($T_{R4}$) [42].
Figure 3.8: Unit response during fast raise test from TWD mode ($T_{R5}$) [42].

Figure 3.9: Calculated fast raise R6 FCAS contribution for raise tests from both no-load and TWD mode [42]. (Note: the 0.5 s response delay is not reflected in the presented FCAS contribution).
Values presented in Table 3.4 below are calculated from the time of the test input signal. It is estimated that there is approximately a 0.5 s delay between the detection of a contingency frequency deviation and the initiation of guide vane movement. As such the values given in Table 3.4 below are somewhat indicative values and would varying slightly under real operational conditions depending on the actual duration of delay.

Table 3.4: Output raise testing summary indicating guide vane servomotor opening stroke, active power output and calculated FCAS contribution at 5.0, 5.5 and 6.0 seconds post trigger time [42]. Estimated FCAS contribution calculated in accordance with the Market Ancillary Services Specification v 4.0 [9].

<table>
<thead>
<tr>
<th>Test ID</th>
<th>Servomotor stroke [mm]</th>
<th>Active power output [MW]</th>
<th>Calculated R6 FCAS [MW]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>5.0 s</td>
<td>5.5 s</td>
<td>6.0 s</td>
</tr>
<tr>
<td>NL1</td>
<td>62.9</td>
<td>66.7</td>
<td>70.5</td>
</tr>
<tr>
<td>NL2</td>
<td>94.3</td>
<td>101.6</td>
<td>111.5</td>
</tr>
<tr>
<td>NL3</td>
<td>96.8</td>
<td>104.7</td>
<td>111.6</td>
</tr>
<tr>
<td>TR1</td>
<td>28.1</td>
<td>31.8</td>
<td>35.5</td>
</tr>
<tr>
<td>TR2</td>
<td>48.6</td>
<td>53.8</td>
<td>59.0</td>
</tr>
<tr>
<td>TR3</td>
<td>48.3</td>
<td>53.5</td>
<td>60.0</td>
</tr>
<tr>
<td>TR4</td>
<td>49.4</td>
<td>51.3</td>
<td>56.4</td>
</tr>
<tr>
<td>TR5</td>
<td>46.4</td>
<td>51.3</td>
<td>56.4</td>
</tr>
</tbody>
</table>
3.4 Tests performed during the Reece Power Station 2012 governor upgrade

During return to service commissioning tests following a governor upgrade the opportunity was taken to test the new governor control and increased opening rates in relation to FCAS provision. An additional oil boost was incorporated into the servomotor actuation mechanism to overcome the two-stage nature of the guide vanes. The results of this boost and governor retuning is shown in the three load acceptance tests of Figure 3.10. The dotted line indicates the previous maximum achievable opening rate (initial 10 mm/s ramp) established during the 2008 test program while the solid lines show improved opening rates of 14.4 mm/s, 19.0 mm/s and 20.2 mm/s. The 20.2 mm/s ramp is the maximum opening rate presently available resulting in a single stage total opening time of 14.4 seconds from fully closed to fully open ($S = 290$ mm).

![Figure 3.10: Guide vane openings following governor upgrade](image)

Due to tight time constraints during re-commissioning only two fast raise TWD tests were able to be performed. Both tests ($T_{R6}$ and $T_{R7}$) were performed at the same opening rate of 20.2 mm/s however $T_{R7}$ was known to have had a higher tail water level prior to transition than $T_{R6}$, by how much it is not known. Power output during rapid transition from TWD mode to generation is given in Figure 3.12 and compared with a rapid load acceptance test from no-load at the same opening rate.
Figure 3.11: Guide vane openings and power response following governor upgrade showing comparison between transitions from TWD and NL condition [42]

Figure 3.12: Guide vane openings and power response following governor upgrade [42]
Figure 3.13: Penstock and draft tube pressure during transition from TWD mode to generation following governor upgrade [42]

Figure 3.11 compares the active power output following the transition from TWD mode and from no-load condition to generation mode under a 20.2 mm/s opening ramp rate. The output response may be compared to that of Figure 3.3 prior to governor re-tuning. It can be seen that at 5.5 seconds (equivalent to the 6 second FCAS contribution window) the active power output from an initial no-load condition is increased from 22.9 MW to 36.9 MW. The effect of the increased opening rate is even more significant for the transition from TWD mode to generation. Figure 3.12 compares the active power output from TWD mode with the output from the earlier feasibility study prior to governor re-tuning (Figure 3.3). The result of the increased opening rate is that the negative power draw is still present, however it occurs nearly 0.5 seconds earlier than previously while the magnitude is only slightly increased. Subsequent power oscillations are also still present, and slightly larger but the shortening of the negative power region has resulted in a much higher output power at the 5.5 second point up to 23.3 MW from 9.37 MW previously. This in turn increases the possible FCAS contribution to 17.1 MW, an improvement of 11.7 MW. It is also worth noting the highly repeatable response in terms of power output seen in transitioning the unit from TWD to generation mode in tests T₆ and T₇.

The increased guide vane opening had a significant impact on penstock pressure as shown in Figure 3.13, the initial low pressure spike decreasing from 790 kPa down to 645 kPa.
While draft tube pressure fluctuations increased, they are more comparable to the pre-upgrade results. The two sudden step increases in penstock pressure at approximately 5.5 seconds and 8.5 seconds that are seen in the current results but absent in the previous results are due to differences in guide vane movements and bear no significance to TWD performance.

3.5 Chapter Summary

This chapter has presented two sets of full-scale testing performed by Hydro Tasmania on the Reece Power Station number 2 machine. Tests performed in 2008 were part of an feasibility study looking into the ability of a Francis turbine to rapidly transition from a TWD mode to generation mode for the purpose of providing load and frequency support to the grid, the results of which provided the initial motivation for the present study. Following a governor upgrade in 2012, two rapid start-up tests were performed, in which the author was present, to gauge the FCAS generation increase under a higher rate of guide vane opening.

Initial tests revealed a significant delay in output power response following guide vane actuation followed by a increased power draw and subsequent power oscillations, ultimately reducing the amount of power generated during the 6 s FCAS window.

The amount of power required from the grid to operate in TWD mode and maintain synchronous speed spinning in air was established to be \( \sim 1.6 \) MW. Following guide vane actuation a large negative power incursion begins at approximately 1.25 s, and reaches a maximum magnitude of 6 MW at just over 2 s from the initial guide vane movement.

Following the 2012 governor retuning, higher guide vane opening rates were achievable. The two tests were performed with the same inlet guide vane opening rate but with differing, but unknown, initial tail water depression levels. The response, in terms of both output power and pressure, suggests the effect of TWD level to be negligible. The magnitude of the negative power incursion was only marginally increased, and while a comparable delay was still present, the temporal location of maximum power draw was reduced thereby increasing the net power contribution to the grid. The amount of FCAS contribution achieved was quantified for each test with the maximum contribution being 17.1 MW for both TR6 and TR7.

The combined set of transient TWD tests performed at the Reece Power Station and described in this chapter, TR1 to TR7, provide the basis for the work, both experimental and numerical, presented in this thesis.
Chapter 4

Design of a Novel Micro-hydro Turbine Unit and Test Facility

4.1 Introduction

Experimental investigations presented in Chapters 5 & 6 were performed on a micro-hydro test facility at the University of Tasmania Hydrodynamics Laboratory. The entire facility including the novel pump-turbine unit was designed and built as part of the current study. The design and steady-state performance testing of the micro-hydro unit are presented in Giosio et al. [32].

The unique design of the 6.2 kW turbine unit integrates an off-the-shelf pump impeller together with a customised housing for incorporation of inlet flow control. The unit, through utilisation of readily available parts and the addition of flow regulation, provides a lower capital cost generating system in comparison to conventional small scale Francis turbine units while providing a much greater operational range than pump-as-turbine (PAT) installations. The resulting design offers a reliable and cost efficient micro-hydro turbine for use in remote area power generation or waste energy recovery systems.

This chapter presents the conception, design methodology and construction of the laboratory facility and mid-range specific speed Francis-type pump-turbine unit. A general description of the facility and testing capabilities are given, along with details of the instrumentation and data acquisition system used for all presented experimental work.
4.2 Concept and methodology

The experimental facility at the University of Tasmania has been developed specifically to investigate the behaviour of hydraulic turbines and pump-turbines during transient operation. The laboratory scale vertical axis Francis-type pump-turbine unit was designed based on the general characteristics of the Reece Power Station, identified by Hydro Tasmania for its rapid start-up potential; although achieving exact geometric and dynamic similarity with the identified unit was beyond the scope and practical limits of the present study. Regardless, a number of key parameters pertaining to both the unit and facility layout were considered during design. The design was also guided by the desire to create a simple, cost effective micro-hydro turbine unit that would improve upon current pump-as-turbine performance in small scale installations.

In the context of micro-hydropower schemes the initial cost of conventional Francis turbine units is often prohibitive. As such there is growing interest in pump-as-turbine technology offering a more cost effective, yet still highly efficient, power generating alternative. However, implementation of a pump-as-turbine is highly problematic in terms of predicting the installed best operating point, which, when coupled with poor off-design performance due to the fixed geometry and absence of inlet flow control, results in a high probability of poorer than expected installed performance.

The ability for pumps to operate efficiently in reverse as turbines was first established by Thoma [101] in 1931 while mapping the full operating characteristic of a centrifugal pump. In recent decades there has been renewed interest in PAT technology which has found significant use in remote area power supply installations, both on- and off-grid, as well as industrial application in energy recovery systems where a high pressure water source exists that would otherwise require throttling.

In such instances, within the range designated as micro-hydropower systems (<100 kW), conventional turbine sets are generally not economically viable making PAT systems an attractive alternative with significantly shorter payback periods. However a major drawback of PATs is that the performance away from best efficiency point (BEP) is extremely poor due to the fixed internal geometry and absence of flow regulation. Various authors have provided, with positive results, a number of relatively simple modifications such as impeller tip and hub/shroud rounding in order to increase overall PAT performance [24, 94]. However the rapid efficiency drop-off at off-design conditions remains an inherent and major limitation of PATs.
This is further compounded by the current lack of accurate and reliable methods for predicting expected PAT performance (BEP) from available pump manufacturer data. In a review of PAT performance prediction methods Williams [108] defines a prediction criterion, $C$, as a means to assess the accuracy of eight prediction models suggested by various authors for predicting the turbine BEP based on either pump performance data at BEP or dimensional pump specific speed, $n_q = NQ^{0.5}/H^{0.75}$ where $N$ [rpm], $Q$ [m$^3$/s] and $H$ [m] at rated. Each method is compared to turbine test data of 35 pumps of various sizes and with $n_q$ ranging from 12.7 to 183.3. The method proposed by Sharma [91] was found to be the most accurate, however 20% of the tested pumps were still outside the acceptable range of the prediction criterion.

More recently Ventrone et al. [103] presents a detailed method of turbine performance prediction through the definition of a runner momentum coefficient, $\psi_R$, which describes the specific momentum work performed within the runner proper assuming zero incident losses. For the four pumps tested the proposed method gives values within ±4% of experimental values although a larger sample size is needed to verify the reliability of the proposed method. Moreover, the method requires specific knowledge of pump geometric parameters as well as performance curves and is therefore limited in its ability as a pump selection tool as these details are often not readily available, particularly at design stage.

Consequently, due to the nature and constraints of potential micro-hydro sites, if the installed BEP is found to differ to even a small degree from the predicted operating point the PAT will likely operate at sub-optimal efficiency for a high proportion of its operational life.

As such it is highly desirable to be able to produce a micro-hydro scale generating set that is cost effective for small installations but is not so constrained by predicted design operating conditions.

The aim of the facility design was therefore twofold:

- To design and construct an experimental facility with transient testing capabilities based on an existing Hydro Tasmania Power Station, and
- To improve upon current PAT design and performance to create a new class of low cost micro-hydro turbine unit
4.2.1 Similarity considerations

Machine type

According to the IEC International Standard 60193 [52] for hydraulic machine model acceptance tests, two machines, model and prototype, can be said to be operating under hydraulically similar operating conditions if the machines are geometrically similar and the ratios of corresponding flow velocity components, resulting in geometrically similar velocity triangles at runner, are identical.

As such, for a given operating point, both machines must have either:

- Identical discharge ($Q_{nD}$), energy ($E_{nD}$) and cavitation ($\sigma_{nD}$) coefficients, or
- Identical discharge ($Q_{ED}$), and speed ($n_{ED}$) factors and Thoma number ($\sigma$)

The specific speed of a hydraulic machine, expressed in Eq. 4.1 as function of discharge and energy coefficients, characterises the runner geometry and flow passage (the runner type) required for optimum operation under rated head and flow conditions. It is the specific speed that indicates which type of turbine; Pelton, Francis or Kaplan; would be most suitable for a given set of head and flow conditions.

$$n_{QE} = \frac{Q_{nD}^{0.5}}{E_{nD}^{0.75}}$$  \hspace{1cm} (4.1)

While specific speed similarity does not necessarily indicate geometric similarity, for the purposes of this project it was desirable to match specific speeds as closely as possible between the identified full-scale Francis turbine and the selected pump impeller (as calculated in turbine operation). This ensures to some degree a certain similarity of hydraulic performance, although results obtained can not be directly scaled up and any inferences made as to the full-scale performance from model results are qualitative only.

Other similarity conditions - Reynolds, Froude and Weber scaling terms

Generally, it is not possible to match Reynolds number, $Re$, between model and prototype for hydraulic machines, with the model $Re$ normally being smaller than that of the prototype. Consequently, frictional losses which are largely dependent on $Re$ are generally much
larger (relative to total losses) for the model, leading to lower efficiencies than seen on the prototype. Scaling factors are available to correct for this.

Froude similarity should be respected where large areas of two-phase flow are present such as large cavitation zones and draft tube vortices, or for any flows involving a free surface. Earlier researchers [15, 112, 114] performing scale model testing on pump-turbines running in synchronous condenser mode applied Froude similarity of operating conditions based on rotational speed equivalent to $Fr = N^2 D / g$ since the characteristic velocity is proportional to $ND$. The currently accepted definition of Froude number [52] for hydraulic machine is given by Eq. 4.2

$$Fr = \left( \frac{E}{g D} \right)^{1/2}$$  \hspace{1cm} (4.2)

Froude influence is particularly important for reaction machines of low specific hydraulic energy when performance becomes influenced by cavitating flow at the runner or in the draft tube. If Froude cannot be matched then Thoma number equality cannot be attained simultaneously for all homologous elevations on model and prototype. A correction exists for determining $\sigma$ values at different elevations.

The IEC standard [52] does not consider any influence of Weber number other than to say that it is usually not possible to match both Froude and Weber numbers simultaneously and that the effects due to Froude number generally dominate over Weber similitude effects. Similarly Lecoffre [63] states that "for the same reasons stated above (Reynolds similarity), equality of this number is also rarely possible" and mentions that it can be taken care of in a "roundabout manner" by injection of artificial nuclei.

**Cavitation number and turbine setting level**

The recommended definition of a cavitation coefficient in regards to turbines is a form of Thoma number expressed as the ratio of net positive suction specific energy (NPSE) to a specific hydraulic energy ($E$):

$$\sigma = \frac{NSPE}{E}$$ \hspace{1cm} (4.3)

where

$$NPSE = \frac{p_a}{\rho} - \frac{v^2}{2} - gh_s - \frac{p_v}{\rho}$$ \hspace{1cm} (4.4)
and
\[ E = \frac{p_{abs1} - p_{abs2}}{\rho} + \frac{v_1^2 - v_2^2}{2} + g(z_1 - z_2) = gH_1 - gH_2 \] (4.5)

Alternatively, a cavitation coefficient can be defined based on the runner diameter \((D)\) and rotational speed \((n)\):
\[ \sigma_{nD} = \frac{NSPE}{n^2D^2} \] (4.6)

**Water acceleration time constant**

A key indicator of a plant's ability to be rapidly brought online is the waterway inertia of the penstock. This is generally given in the form of a water start time, otherwise known as the water acceleration time constant. The water acceleration time constant, \(T_w\), is defined as:
\[ T_w = \sum \frac{L_i Q_r}{g A_i H_r} \] (4.7)

where \(L_i\) is the length, and \(A_i\) the cross-sectional area of the \(i^{th}\) conduit section.

The water acceleration time constant represents the time required for the water within the penstock to be accelerated up to rated flow, \(Q_r\), under rated head conditions, \(H_r\).

Typically for most commercial scale hydropower plants the water acceleration time constant can vary anywhere between 0.5 s and 4.0 s at full load [61]. For stations with multiple units and shared penstocks this value can vary significantly depending on the number of units on-line with time constants increasing as the number of machines operating increases.

Due to the inevitable difference in physical scale and operating head-flow of laboratory sized installations the water acceleration time constant is greatly reduced. In order to compare full-scale and laboratory-scale transient events the water acceleration time constant may be used to non-dimensionalise the time measurement during rapid start-up testing as will be discussed in Section 6.1.

**4.2.2 PAT design principles**

Prediction methods of PAT performance generally fall into one of four categories: formulae based on pump efficiency, formulae based on pump specific speed, empirical relations and theoretical procedures such as the method of Vetrone et al. [103] discussed earlier. To date there is no method that provides a genuinely accurate and reliable way to determine
the expected turbine performance from publicly available pump data, indeed due to the extremely wide variety of pumps on the market such prediction methods will only ever serve as a rough guide. Recent studies such as Derakhshan and Nourbakhsh [26], Fecarotta et al. [29] and Yang et al. [113] have begun to use computational fluid dynamics (CFD) as a means to predict pump-as-turbine performance. Such an approach requires detailed geometry of the pump in question, even more so than theoretical methods such as Ventrone et al. [103], and as such precludes the use of the method in the feasibility stage not to mention the cost and expertise required for such a study. The increased understanding gained from such CFD investigations on the other hand may provide useful insights into PAT performance leading to better prediction methods in the future.

As a first approximation the method of Sharma [91] was used to determine the required pump characteristics from the desired turbine characteristics. Sharma builds on the work previously put forward by Childs [19] that makes the initial assumption that turbine best efficiency will be approximately equal to that in pumping mode, $\eta_t \approx \eta_p$, and that the output power in turbine mode can be expected to be equal to that of the pump input power at pump best efficiency giving

$$\rho g Q_t H_t \eta_p = \rho g Q_p H_p \eta_p$$

(4.8)

Childs [19] then makes the assumption that the ratios of head and flow in pump and turbine mode are equal, i.e. $Q_t/Q_p = H_t/H_p$ which leads to the result

$$\frac{Q_t H_t}{Q_p H_p} = \frac{1}{\eta_p^2}$$

(4.9)

although it is now widely accepted that the head ratio $H_t/H_p$ is generally the larger of the two [103, 113]. Sharma uses an equation given by Engel [27] relating the expected turbine specific speed to the known pump specific speed by way of the pump efficiency which can be expressed in Eq. 4.10, assuming the pump and turbine operating speeds are equal.

$$\frac{\sqrt{Q_t}}{H_t^{0.75}} = \sqrt{\eta_p} \frac{\sqrt{Q_p}}{H_p^{0.75}}$$

(4.10)

Solving equations 4.9 and 4.10 results in the ratios as used by Sharma:

$$\frac{H_t}{H_p} = \frac{1}{\eta_p^{1.2}} \quad \text{and} \quad \frac{Q_t}{Q_p} = \frac{1}{\eta_p^{0.8}}$$

(4.11)
However, in order to allow for potential operation at a reduced speed, an additional correction was made based on turbine affinity laws as $Q \propto N$ and $H \propto N^2$ such that

$$H_t = \frac{N_t^2}{N_p} \frac{H_p}{\eta_p^{1.2}} \quad \text{and} \quad Q_t = \frac{N_t}{N_p} \frac{Q_p}{\eta_p^{0.8}}$$

(4.12)

In most site installations of PAT units an induction motor is used in place of a synchronous generator [95, 109]. A directly coupled induction generator removes the need for any belts or gearing, minimises any lateral force thereby prolonging bearing life, eliminates the requirement for a turbine shaft bearing, has low drive friction losses and they are low cost - particularly compared to synchronous generators of sizes up to 30 kW [109]. However this requires that one of the synchronous speeds, corresponding to the number of poles of the induction generator, must be chosen as the turbine operating speed [24].

### 4.2.3 System design curves

System design curves were plotted for specific speed similarity over the range of available head and flow able to be provided at the UTAS laboratory. Spatial constraints on the size and location of water supply tower restricted the design operating point to the $H \sim Q$ plane as shown in Figure 4.1. Operational speeds of 1000 rpm, 750 rpm and 600 rpm were considered as these correspond to 6, 8 and 10 pole synchronous speeds running at 50 Hz supply frequency.

Speed curves were plotted according to specific speed similarity according to Equation 4.13 while turbine diameter curves were plotted according to Equation 4.14 (i) and (ii) using values for the chosen rotational speed.

$$Q = \left[ \frac{n_{QE}(gH)^{0.75}}{n} \right]^2$$

(4.13)

$$H = \frac{E_n D (nD)^2}{g} \quad \text{and} \quad Q = Q_n D (nD^3)$$

(4.14)

As such, any turbine unit with a BEP located on one of the speed curves will have a specific speed equal to that of the full-scale unit, while the required diameter of that machine is indicated by the intersection of a turbine diameter line.

The ideal rated head and flow range and corresponding runner reference diameter for a scale model turbine of specific speed equal to that of the Reece unit, $n_{QE} = 0.201$, is shown
Figure 4.1: System design curves for a micro-hydro scale turbine with $n_{QE} = 0.201$ indicating required runner diameter and rotational speed required for specific speed similarity.

in Figure 4.1. In order to achieve a reasonable operational specific energy at a commonly available induction motor speed the design point was chosen as indicated by the highlighted area.

Knowing the available turbine net head and flow rate, $H_t$ and $Q_t$, and the associated turbine speed and diameter, $N_t$ and $D$, Equation 4.12 (i) and (ii) can be used to select a suitable pump impeller based on readily available manufacturers data. The selected pump impeller was a KSB Ajax I.S. series impeller with turbine outlet diameter, $D = 226$ mm. The rated values (BEP) of the pump unit from which the impeller is taken and the predicted and actual turbine performance are given in Table 5.1 in the following chapter.

4.2.4 Distributor design

With the pump impeller selected, the guide vane mechanism, stay vane ring and sub-assembly was designed. The starting point for the distributor design was the calculation of the anticipated optimal guide vane opening angle using velocity diagrams according to 2-dimensional analysis (Section 1.1).
Knowing the turbine theoretical flow rate, \( Q_t \), as calculated in the previous section, and the inlet geometry of the purchased impeller, the meridional component, \( C_{m1} \), of the absolute fluid velocity at design may be calculated (Equation 4.15). The peripheral component (Equation 4.16) of the absolute velocity can then be determined from the desired rated speed, \( U_1 = \omega R_1 \), and the inlet blade geometry, \( \beta_1 \) measured at the turbine blade mid-point (turbine centreline, \( \xi \)).

\[
C_{m1} = \frac{Q_t}{2\pi R_1 b} \\
C_{u1} = U_1 - \frac{C_{m1}}{\tan \beta_1}
\]

(4.15)  
(4.16)

The guide vane opening angle at design point can then be calculated as:

\[
\alpha = \arctan \frac{C_{m1}}{C_{u1}}
\]

(4.17)

Defining a maximum guide vane opening \( \alpha_{\text{max}} = 1.5 \times \alpha \) and including an appropriate runner~guide vane gap, the pitch circle diameter (PCD) of the guide vanes and the required number of vanes was determined. The number of guide vanes must not be equal to, or a multiple of, the number of runner blades so that periodical pressure fluctuations are prevented.

In a similar manner, taking the maximum guide vane opening, the radial position of the stay vane ring was determined. Being fixed in position the stay vanes were set at the predicted best efficiency guide vane opening angle, \( \alpha \). Again, the number of blades were set such that periodic fluctuations are avoided.
4.3 Experimental facilities

The UTAS micro-hydro experimental facility is located in the Hydrodynamics Laboratory at the University of Tasmania Sandy Bay campus, Hobart, Australia. The UTAS micro-hydro testing facility was designed specifically for the investigation of hydraulic transients in hydropower plants resulting from rapid changes in operational set point. Of particular interest is the behaviour and output response following rapid transition from modified synchronous condenser mode to generation mode for the purpose of frequency support to the grid. In order to allow full investigation of transient events a number of key design features were included into the design of the apparatus, infrastructure and auxiliary equipment not normally required in a typical hydraulic machine test facility.

The pump-turbine unit itself was designed in such a way as to be a prototype model for a new class of micro-hydro turbine for operation in remote area power supply and for industrial applications such as waste energy recovery systems. To this end the design and construction of the unit was required to be relatively simple and low cost while still providing competitive performance and payback period in comparison with existing micro-hydro options currently available.

The design of the micro-hydro experimental facility was carried out as part of the current study by the author, including all design calculations, 3D CAD model and the production of design drawings. Construction of infrastructure was largely carried out by the University.
of Tasmania Workshop with the exception of the elevated water storage support structure. The pump-turbine unit housing was manufactured and assembled by Pentair Flow Technologies, Tasmania.

### 4.3.1 Supply pump and hydraulic circuit

Flow is provided to the pump-turbine unit by a Southern Cross (Star-PRO model 250x200-315) centrifugal pump set fitted with a 22 kW 6 pole Monarch 3-phase motor and variable frequency drive (VFD). Water is pumped from a large under floor open air storage pit within the Hydrodynamics Laboratory to which it is returned downstream of the turbine unit via a tail water tank. In the current arrangement the pump set is capable of providing a volumetric flow rate up to 0.150 m$^3$/s at a total delivery head of 10 m.

![Schematic of dual configuration hydraulic circuit](image)

Figure 4.3: Schematic of dual configuration hydraulic circuit, micro-hydro turbine unit and downstream discharge tank installed at the University of Tasmania.

The dual configuration hydraulic circuit of the University of Tasmania test facility has been designed for operation in open loop, for isolation and study of hydraulic transients; or semi-closed loop arrangement for steady-state machine testing over the full operating range. In semi-closed loop operation inlet flow is provided directly from a supply pump, delivering up to 10 m head at 0.150 m$^3$/s, while in open loop configuration flow is gravity fed from an elevated supply tower with a static head supply range, depending on downstream gate height, of between 5.10 m and 7.15 m. The natural head supply allows penstock pressure transients, and subsequent reflections from the free surface, caused by rapid guide vane movements to be measured and analysed. Vortex breakers are fitted to both tank inlet and tank outlet to eliminate surface vortex formation and consequent air entrainment.

Notably the circuit is without the presence of a main inlet valve (MIV) upstream of the turbine inlet. This is an essential feature required of a given power station to be considered for
rapid start-up operation. For stations without a MIV, when not in operation the upstream penstock water is required to be held by the guide vanes in the closed position. Units at such stations are therefore fitted with guide vane insert seals. This has the advantage of the machine being able to be brought online much faster than a station in which a MIV is fitted which typically have a much greater opening time than guide vane actuation.

All upstream pipe sections are made from 10 bar rated polyethylene (PE) pipe, with inlet section 315 mm nominal diameter. To minimise inlet flow disturbances and non-uniformity all internal pipe welds were ground back and finished with a smooth surface prior to assembly. All pipe joins use slip ring flanges allowing the rotation of pipe segments depending on the required configuration.

In both open and semi-closed loop configurations water exits the draft tube and returns to the under floor storage pit via an open air tail water discharge tank. The tail water tank is fitted with an adjustable height weir for controlling the degree of back pressure on the turbine. During operation the tail water level may be adjusted from between -0.40 m to +0.28 m relative to the turbine centreline. This change in gate height corresponds to a static back pressure range of between -0.07 \( H_r \) to +0.05 \( H_r \). In comparison, the variation in tail water level at the Reece Power Station varies from between -0.03 \( H_r \) to +0.04 \( H_r \). Downstream of the weir a return channel conveys the turbine outflow back to the under floor storage pit.

In open loop configuration, flow is provided via an elevated storage tank to isolate any hydraulic coupling between pump and turbine. Operation under natural head supply also lends itself to simulation using 1-dimensional analytical models, as discussed later in Chapter 7. Depending on tail water gate height the net static head available can be set between 0.85 \( H_r \) to 1.20 \( H_r \). In comparison, the variation in net static head at the Reece Power Station depending on tail water conditions varies from between 0.92 \( H_r \) to 1.13 \( H_r \).

The support structure was designed to support the weight of a 4500 L storage tank and filled vertical penstock sections. Independent, adjustable support is provided for the horizontal penstock and pump outlet sections for ease of changing over between open and semi-closed loop configurations. Ladder access and a guarded elevated workspace around the tank is provided to comply with occupational health and safety regulations. All pipe sections are rigidly attached to pump, tank and turbine unit so as not to dampen hydraulic transients.

Cooling lines take high pressure water from the spiral case to labyrinth seals located at runner band, crown and shaft seal housing.
4.3.2 Micro-hydro scale turbine unit

![Figure 4.4: CAD design model of the UTAS micro-hydro turbine unit.](image)

The micro-hydro scale turbine unit described herein was based around the selected pump impeller which was chosen according to pump-as-turbine prediction theory and design methodology previously discussed in Section 4.2. The novel aspect of the design is the inclusion of inlet flow regulation in the form of a guide vane assembly used in conjunction with a readily available pump impeller. In order to accommodate the assembly a simplified spiral case was designed and manufactured as a typical pump housing lacks the required space between impeller periphery and volute section to house the required guide vanes. The result is a unit with the operational flexibility of a high cost conventional Francis turbine but with a capital cost only slightly higher than that of a typical PAT set-up, depending on the level of complexity of the designed casing. It is thought that the better utilisation of generating potential, in terms of water usage, and the increased percentage of operational time at or near design point may even provide a payback period less than that of a comparable sized PAT installation, although a comprehensive economic study has not been performed to date.

Essentially the turbine unit consists of spiral case, stay vane ring, top and bottom plate assemblies, guide vane assembly, rotating assembly and draft tube (Figure 4.4). Key component details are given in the following sections.
Spiral case

The spiral case, as the name suggests, is a spiral shaped casing, symmetrical around the turbine centreline, that totally surrounds the opening on the periphery of the stay vane sub assembly. The purpose of the spiral case is to deliver water to the turbine with uniform velocity along the circumference. In order to achieve this, the cross-sectional area of the spiral case must decrease from Sect 1. to Sect. 31 (Figure 4.5) as water is directed and transported in to the turbine inner. Typically for large scale units the spiral case is manufactured using a circular section lobster-back design, while for medium to small units spiral cases may be cast either as a single piece or in two halves joined at turbine centreline. For simplicity and ease of manufacture a square section spiral case was designed from four pieces of sheet metal, the internal sheet being cut in two to accommodate the stay vane sub assembly. The stay vane sub assembly consists of the stay vane ring and top and bottom plate assemblies. The primary function of the stay vanes is structural, holding the top and bottom plates together under the large internal pressures. The stay vanes were manufactured from flat plate steel GR 250 with a circular leading edge and tapered trailing edge as seen in Figure 4.7.

![Figure 4.5: Spiral case template sections for the micro-hydro turbine unit. The top image indicates the radial positioning, while the middle and lower image are templates for the outer and inner side sections, respectively.](image)
Guide vane mechanism

The purpose of an adjustable guide vane mechanism is to control both the velocity and direction of the flow entering the runner proper such that flow is uniform around the entire circumference. According to the available head~flow condition, adjustment can be made such that the incoming flow velocity matches the inlet blade angle for the given operating condition. Without the presence of guide vanes, this optimal flow alignment is only possible at a single value of head and flow. Additionally, guide vanes are used to control the amount of flow entering the turbine unit depending on current system power demand. In this case it is more a matter of regulating output rather than matching inlet flow angle. As a result, in fixed pitch singularly regulated machines such as Francis turbines and PATs, such regulation will alter the inlet and outlet velocity diagrams resulting in increased losses and an associated decrease in efficiency in order to control the power output, see Figure 1.5, Section 1.1.

The guide vane assembly consists of 13 hydrofoil shaped vanes with individual linkages to a common ring. Position is controlled by a programmable 5.3 kN SEW electro-mechanical linear actuator with position feedback. The guide vane mechanism is shown undergoing testing in Figure 4.6, installed in the case sub assembly consisting of spiral case, stay vane ring, top and bottom plates and electro-mechanical actuator mounted on the case support. Control of the actuator is incorporated into the LabVIEW data acquisition system (see Section 4.4.10) with the ability for real-time control or pre-programmed opening and closing

Figure 4.6: Guide vane mechanism operational testing prior to final assembly
profiles. Due to the lack of a MIV, each guide vane was fitted with a recessed rubber sealing pad flush mounted to the surface for minimising leakage flow (Figure 4.7). Additionally the top and bottom plate of the turbine case was lined with a thin plate of Ertacytal® C enabling extremely tight top and bottom guide vane tolerances with low coefficient of friction.

For the purposes of this project existing guide vanes were sourced. The hydrofoil vane and shaft are cast as a single piece of Duplex, a high strength and corrosion resistant form of stainless steel. The guide vane profile is asymmetrical on the front half chord after which the camber line becomes completely linear. The only modifications required was to reduce the vane height to that of the runner inlet passage height and to recess a sealing pad flush with the vane surface at the calculated contact point of each vane (see Figure 4.7). It is noted that for the purpose of flow control only the outlet portion of the vane is of any real importance and that the shape of the first half is only to minimise losses [73]. As such it is reasoned that for low cost turbine units a simple guide vane profile, possibly even flat plate sections, would provide adequate regulation to significantly improve the turbine operating range, although the ability to predict the best efficiency point of a given PAT may be affected due to the increased hydraulic losses. As will be shown in the following chapter however the ability to control the inlet flow angle mitigates any deviation from predicted to actual turbine best efficiency point.

Figure 4.7: UTAS micro hydro unit guide vane (left, middle), cast as a single piece of Duplex with sealing pad shown at the contact point, and stay vane design detail (right).
Draft tube

Modern draft tube design generally consists of a straight circular diffuser section, the draft tube cone, followed by a transition bend from circular to square section. This transition to square section allows for continual increase in draft tube internal flow area without significant additional civil works costs [105]. A pier is often included to prevent flow detachment. In the laboratory however where spatial constraints are not present, a simplified circular section draft tube design is both cost effective and easier to manufacture.

The entire section from runner exit to tail water tank is made from 8 mm clear acrylic enabling full visual access of the draft tube flow, see Figure 4.2. Comprised of draft tube cone (diffuser) section at a 4° half angle expansion, length of 2.4D, followed by a constant diameter 7-section lobster back bend, the resulting inclination angle of the draft tube is 15° from the horizontal. A second diffuser section follows the bend increasing the draft tube diameter to 1.8D designed to give an optimal draft tube exit velocity of 1 ms⁻¹ at rated conditions. The draft tube also features a compressed air inlet 0.3D below runner exit for air admission. Two pressure tappings are also provided on a plane 0.44D below runner exit 180° apart (Section 4.4.4).

Figure 4.8: UTAS micro hydro turbine draft tube design drawing. The circular section draft tube is constructed from 8 mm clear acrylic from cone to draft tube exit for full visual access.
Rotating assembly

The rotating assembly consists of runner, runner cone and turbine shaft with rigid coupling to the motor/generator shaft. The direct coupling and a short shaft length eliminates the need for a separate turbine bearing with axial and radial load taken by the induction motor bearing offering significant cost benefit. The turbine shaft is partially hollow from the top seal housing to runner cone allowing air admission via the runner cone for alleviating draft tube pressure pulsations during part-load operation (Figure 4.9). During experiments undertaken in this study this was blocked off. While the direct coupling design reduces complexity and cost of the unit due to the elimination of the turbine bearing it does mean that common in-line torque sensor units were unable to be used. As such a strain gauge was used for shaft torque measurement as detailed in section 4.4.6.

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Top</th>
<th>Middle</th>
<th>Bottom</th>
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</thead>
<tbody>
<tr>
<td>Inlet angle</td>
<td>36°</td>
<td>32°</td>
<td>43°</td>
</tr>
<tr>
<td>Outlet angle</td>
<td>33°</td>
<td>24°</td>
<td>15°</td>
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Figure 4.9: Rotating assembly consisting of turbine runner (pump impeller) nose cone, turbine shaft and rigid tapered coupling for direct connection to motor/generator shaft.
**Induction motor/generator**

A 15 kW, 8-pole (750 rpm) WEG High Efficiency E3 industrial three phase induction motor is used as turbine generator. The motor is directly coupled to the turbine shaft, and flange mounted on the generator support plate, designed to allow adjustment in all three axes to ensure correct shaft alignment. A control circuit provides automated grid synchronisation at 720 rpm. Once synchronised the induction motor may either draw power from the grid, acting as a motor, or, supply power to the grid, depending on the present operating condition of the turbine. This grid connectivity allows for operation in TWD mode (motoring) and the fast transition to generation mode.

**4.4 Instrumentation**

**4.4.1 Overview**

The following sections outline the specific measurement equipment used throughout this project. The operating conditions of the turbine were determined by head and flow measurements at rated speed. The net head across the turbine was measured using digital gauge pressure transducers located at spiral case inlet and draft tube outlet, while flow rate measurement was achieved using an ultrasonic flow meter upstream of the spiral case inlet. Output power of the turbine was measured at the shaft as determined by shaft torque and speed measurement. As such, the efficiency of the generator is not required to be taken into account giving a direct indication of the exchange between working fluid and turbine runner with mechanical losses accounted for. During steady state operation control of the operating condition was possible via either supply pump variable frequency drive (VFD) or by guide vane position adjustment using an electromechanical actuator. Dynamic pressure sensors were flush mounted in the penstock for determination penstock characteristics and for the study of hydraulic transient behaviour following rapid changes in turbine set point.

**4.4.2 Hydraulic flow rate**

The hydraulic flow rate through the turbine was measured $5.0D_{pipe}$ upstream of the spiral case inlet using a Yokogawa US300PM ultrasonic flow meter with externally mounted transducers. The Yokogawa unit has a manufacturer specified accuracy of $\pm 0.01 \text{ m/s}$ over the sonic path and is independent of liquid pressure or conductivity. The transducer set
was located at a distance of $10.7D_{pipe}$ downstream of the 90° pipe bend in natural head configuration.

### 4.4.3 Gauge pressure

Gauge pressure for the determination of net available turbine head is measured at both spiral case inlet and draft tube outlet according to IEC standards. The specific hydraulic energy across the turbine is calculated from separate gauge pressure measurements and calculated inlet and outlet velocities from measured flow rate according to Equation 4.18:

$$ E = gH = \left( p_{M1} - p_{M2} \right) / \rho + \left( v_1^2 - v_2^2 \right) / 2 + g(z_{M1} - z_{M2}) $$  \hspace{1cm} (4.18)

For determination of test specific energy Druck UNIK 5000 gauge pressure transducers, 0 - 100 kPa range, were fitted to ring manifolds with individually valved tappings at the four diagonals and an air bleed valve at the top according to IEC standard. Each unit has a frequency response of 3.5 kHz and was calibrated over the full range with a maximum deviation of 0.06% full scale.
4.4.4 Dynamic pressure

Three high frequency PCB piezoelectric dynamic pressure sensors are flush mounted in the penstock at 0.93 m, 2.43 m, and 6.80 m from turbine inlet for detection and characterisation of pressure transients following rapid guide vane operation. Two high frequency sensors are also located in the draft tube cone on the same horizontal plane, 180° apart. A sixth dynamic sensor is located at mid-passage of a guide vane channel at rated opening.

4.4.5 Temperature

Water temperature measurements were taken using a platinum resistance temperature sensor (Temtrol Technologies model MIR-HPC-TX RTD), 0 - 50°C operating temperature range with head mount 4-20 mA transmitter located at the inlet line to the supply pump. Readings were taken concurrently at the same sample rate as all other measurements for both steady state and transient testing.

The test temperature, $\theta \, [^\circ C]$, is used in the calculation of water density, $\rho$, and vapour pressure, $p_{va}$, which are subsequently used in the determination of available hydraulic...
power, $P_h$, machine specific energy, $E$, and net positive suction energy, NPSE.

Based on the empirical equation of Weber [107] according to IEC standards the specific volume of water, $v$, in m$^3$kg$^{-1}$ is given by:

$$v = 1/\rho = v_0[(1 - A \cdot p_{abs}) + 8 \cdot 10^{-6} \cdot (\theta - B + C \cdot p_{abs})^2 - 6 \cdot 10^{-8} \cdot (\theta - B + C \cdot p_{abs})^3] \quad (4.19)$$

where $v_0 = 1 \cdot 10^{-5}$ m$^3$kg$^{-1}$, $A = 4.6699 \cdot 10^{-10}$, $B = 4$, $C = 2.1318913 \cdot 10^{-7}$ and absolute pressure, $p_{abs}$, in Pa [52]. Vapour pressure, $p_{va}$ [Pa] of water for temperatures between 0 and 40°C may be calculated according to the following empirical relation [52]:

$$p_{va} = 10^{2.7862 + 0.0312\theta - 0.000104\theta^2} \quad (4.20)$$

Prior to each individual test the temperature reading was recorded and entered in to the Yokogawa control unit as an input.

### 4.4.6 Torque

Shaft torque measurement is achieved with a full bridge 350 Ohm strain gage installed on an exposed section of turbine shaft with data acquisition via a KMT digital telemetry system. The KMT transmitter (model TEL1-PCM-BATT) is mounted to the rotating shaft and provides a pulse code modulated signal (PCM) to an inductive winding around the shaft [59]. An inductive pick-up is mounted at a distance of 15 mm from the transmitter coil which then outputs to the decoder. The encoder scanning rate is 6.944 kHz and system accuracy is stated at ±0.2%. The strain gage was shunt calibrated with a 1000 ppm change in electrical resistance across each element giving a $F_c$ coefficient of 15.019 ppm/volt.

Torque, $T$, is calculated from the measured shaft strain as:

$$T = \frac{\pi G D^4 F_c}{8D 10^8} \quad [Nm/v] \quad (4.21)$$

where $G$ is the shear modulus of the solid stainless steel shaft (of diameter $D_s$) in GPa, calculated as $G = E/2(1+\nu)$ with Young’s Modulus, $E = 200$ GPa and Poisson’s ratio, $\nu = 0.275$. 
4.4.7 Rotational speed measurement

A 500 hole optical encoder is rigidly mounted to the non-drive end of the generator shaft for high resolution rotational speed measurement for capturing speed deviation during rapid machine loading (Figure 4.13). At rated speed this equates to an effective sample rate of 2250 pulses/sec. A separate Hall effect sensor is used as an indexing pulse while an induction pick-up, completely independent of all data acquisition equipment, is mounted on the drive end for grid synchronisation and over speed protection.

4.4.8 Tank levels

Water level in the elevated head tank was monitored using an ultrasonic ToughSonic®/PC Distance Sensor mounted in the roof of the storage tank. The sensor has an optimal working range of 102 mm - 3.25 m, covering the full height of the tank, with a resolution of 0.09 mm. The 4-20 mA output, adjusted to the desired range, continually recorded tank water level and was used as a trigger to shut down the supply pump at the designated full supply level. Additionally, a mechanical float switch was installed to serve as a secondary full supply level trigger.
4.4.9 Actuator position and GV angle calibration

Although the SEW electro-mechanical linear actuator had built-in position feedback there was found to be an unsatisfactory delay in communication between the actuator and the DAQ system during rapid transient testing. As such a long stroke linear potentiometer with 4-20 mA analog output was rigidly attached to the actuator body and drive arm to provide a secondary measurement of the actuator stroke length.

An indicator arm was mounted to a guide vane shaft to provide a true physical measurement of the actuator stroke ~ opening angle relationship, against a stationary angular scale. As with any mechanical system, a small degree of hysteresis was found in the actuator movement, Figure 4.14. The magnitude of backlash was highly repeatable, irrespective of the opening/closing profile tested. A calibration curve was incorporated into the LabVIEW DAQ and control system so that the required actuator stroke could be corrected for based on desired guide vane angle and direction of movement. The calibrated GV angle for a full opening and closing cycle is given in Figure 4.15.
Figure 4.14: Measured inlet guide vane opening angle and actuator stroke length prior to calibration. Actuator stroke was opened in 5 mm increments from closed, up to 90 mm, and returned to the fully closed position.

Figure 4.15: Calibrated inlet guide vane opening angle. Actuator stroke was opened in 10 mm increments from closed, up to 90 mm, and returned to the fully closed position.
### 4.4.10 Data acquisition

Data acquisition for shaft torque, rotational speed, dynamic pressure and actuator position required during rapid transient tests was achieved using a National Instruments 8 channel, 16-bit NI PCI-6123 simultaneous sampling card, while global measurements such as turbine flow rate and available head, as well as tank levels and water temperature were acquired using a NI PCI-6251 standard High Speed 16-bit card.

The control and measurement system was implemented using the National Instruments LabVIEW system design platform. The designed control system provided real-time monitoring of turbine performance with the ability to manually adjust the turbine operating point on-the-fly as required, or perform pre-programmed guide vane actuation profiles including start-up procedures from TWD mode. Data acquisition and recording was initiated manually, which may also be used to provide the trigger for pre-programmed actuation profile test scenarios.

An independent control system monitored turbine over-speed and supply tank level. If either of these were triggered the supply pump circuit breaker was tripped, shutting off the pump, and the grid connection cut.
Chapter 5

Performance Evaluation of a Novel Micro-Hydro Turbine Unit

5.1 Overview

This chapter presents the results of performance tests conducted on the novel micro-hydro turbine unit developed at the University of Tasmania (UTAS-MH unit) introduced in the previous chapter, and presented in Giosio et al. [33]. Steady-state tests are performed to determine the operating characteristics of the turbine facility and assess the suitability of such a unit to be employed in either an industrial setting or a remote area power system. Measurement of global variables; namely upstream and downstream head, flow, rotational speed and torque are recorded during steady-state tests to establish the installed best efficiency point as well as to evaluate the off-design performance. Additionally the performance at very low range flow is examined.

The main driver for the design of the turbine unit was to improve off-design efficiency, recognising that in many situations the conditions at a given site will vary and that pump-as-turbine prediction methods are presently unreliable. Accordingly, a key criterion for assessment of the new turbine unit is the weighted average efficiency (WAE), based on the definition of Singh [94]. As defined, the WAE takes into account the relative performance of the turbine at three pre-determined load conditions, weighted based on the anticipated share of operational time expected at each setting.
5.2 Turbine performance characterisation

The global performance characteristics of primary concern for assessment of a micro-hydro unit are measurements of net head, $H_{net}$, and flow rate, $Q$, in addition to output power. From these measurements the overall efficiency of the turbine unit may be determined at any given operating point, defined in terms of available head and flow and operating speed.

The point at which power is measured will effect the stated efficiency and turbine specific speed \[^{3}\]. In the case of micro-hydro installations, the overall efficiency and electrical power output of the system is of interest and should therefore take into account the generator performance. The definition for overall turbine efficiency, $\eta_o$, as given in Eq. 5.1, however, is a measure of the power transmitted to the turbine shaft.

\[
\eta_o = \frac{P_t}{P_{hyd}} = \frac{2\pi n.T}{gH_{net}.(\rho Q)}
\] (5.1)

This definition was chosen because of the nature of the proposed transient testing in relation to rapid start-up ability and a desire to determine the mechanical torque applied on the shaft as distinguished from the active power output of the generator. Shaft power also allows comparison with efficiency data reported in the literature \[^{3, 21}\]. The generator efficiency curve is of course known and can be used to determine the full system’s operational efficiency if required, for example in determining the installations weighted average efficiency (WAE) (see Section 5.4).

Measured at the shaft, the overall efficiency incorporates mechanical efficiency, taking into account mechanical losses in shaft seals and bearings; volumetric efficiency, taking into account leakage and cooling water flow; and hydraulic efficiency, defining the amount of power extracted by the runner from the power available from the water. Owing to the tight internal tolerances achieved during manufacture the volumetric efficiency can be approximated to $\eta_v \approx 1$, with only small losses due to cooling water flow, such that

\[
\eta_o \approx \eta_m \eta_o \eta_h \approx \eta_m \eta_h
\] (5.2)

Mechanical efficiency due to losses in bearings, shaft seals and wear rings was estimated by running the turbine up to rated speed and connecting to the grid. Guide vanes were then closed and air injected into the draft tube in order to vacate the runner chamber. With cooling lines open the torque required to maintain constant speed, $T_{loss}$, was recorded.

The mechanical efficiency as determined by Eq. 5.3 was assumed constant over the entire operating range, however in practice this may vary to a small degree.
\[ \eta_m = \frac{P_i}{2\pi n(T + T_{loss})} \quad (5.3) \]

Therefore the hydraulic efficiency can be found by dividing Eq. 5.1 through by Eq. 5.3:

\[ \eta_h = \frac{\eta_o}{\eta_m} = \frac{2\pi n \cdot (T + T_{loss})}{gH_{net} \cdot (\rho Q)} \quad (5.4) \]

The available specific energy of the hydraulic circuit under given operating conditions is given by:

\[ E = gH_1 - gH_2 \quad [J/kg] \]
\[ = gH_{net} \quad (5.5) \]

where subscripts 1 and 2 refer to the high and low pressure measuring points, respectively. In accordance with IEC Standards [52] the net specific energy across the machine is calculated as

\[ E = \frac{p_{M_1} - p_{M_2}}{\rho} + \frac{v_1^2 - v_2^2}{2} + g(z_{M_1} - z_{M_2}) \quad [J/kg] \quad (5.6) \]

where \( M_1 \) and \( M_2 \) refer to the measurement instruments at high and low pressure side. The hydraulic power available to the turbine is therefore a product of the test specific energy and the mass flow rate calculated as:

\[ P_{hyd.} = E(\rho Q) \quad [W] \quad (5.7) \]

The power transferred to the turbine shaft, \( P_t \), is given by:

\[ P_t = 2\pi n.T \quad [W] \quad (5.8) \]

where \( n \ [s^{-1}] \) is the rotational speed of the turbine and \( T \ [Nm] \) is the measured shaft torque per volt output, calculated from the measured shaft strain as:

\[ T/V = \frac{\pi G D_s^4 F_c}{8D_s^{10^5}} \quad [Nm/V] \quad (5.9) \]

where \( G \) is the shear modulus of the solid stainless steel shaft (of diameter \( D_s \)) and \( F_c \) is the strain gauge calibration coefficient (see Section 4.4.6).

\[ G = \frac{E}{2(1 + v)} \quad [GPa] \quad (5.10) \]
For the stainless steel shaft the modulus of elasticity, \( E = 200 \text{ GPa} \), and Poisson’s ratio, \( \nu = 0.275 \), is assumed.

In addition to the physical variables, turbine performance is presented in terms of dimensionless coefficients as defined by IEC 60193 standards [52]. The energy coefficient, \( E_{nD} \), discharge coefficient, \( Q_{nD} \), power coefficient, \( P_{nD} \), and torque coefficient, \( T_{nD} \), are defined in equations Eq. 5.11 to 5.14.

\[
E_{nD} = \frac{E}{n^2D^2} \quad (5.11)
\]

\[
Q_{nD} = \frac{Q}{nD^3} \quad (5.12)
\]

\[
P_{nD} = \frac{P}{\rho n^3D^5} \quad (5.13)
\]

\[
T_{nD} = \frac{T}{\rho n^2D^5} \quad (5.14)
\]

5.3 Steady-state performance

5.3.1 Test procedure

Performance evaluation of the UTAS micro-hydro unit was carried out in the semi-closed loop test configuration with direct connection to the supply pump to achieve the full range of test specific energies available in the laboratory. Prior to operation the penstock, turbine unit and draft tube are primed; an air bleed valve is located at the highest point of the inlet section to assist the filling. The start-up of the unit is achieved using the VFD to slowly increase pump speed at a guide vane opening of 10° until synchronous speed is reached.

Grid connection of the induction motor acting as generator is automated to occur at a turbine rotational speed of 720 rpm (8-pole motor, 750 rpm synchronous speed). Speed measurement for grid connection is via an induction pick-up mounted beneath the generator support plate picking up the shaft keyway and is independent of all other instrumentation, data acquisition system and LABVIEW control.

The same sensor also monitors overspeed, in which case a circuit breaker trips and cuts power to the pump at a detected speed of 780 rpm. Once brought up to speed the unit is locked to the electrical grid and is therefore subject to any minor fluctuations around the 50 Hz system frequency. Furthermore grid synchronisation limits the test facility in the current set-up to one nominal operating speed.
Steady state test data were recorded at each operational point with a sample frequency of 100 Hz acquired by a NI M Series Multifunction DAQ SCB-68 card. Convergence and repeatability tests were conducted to determine an adequate settling time between test measurements and the duration of each test required to ensure accurate results.

Due to synchronisation with the grid, turbine speed was found to exhibit very low frequency variations, an example of which is shown in Figure 5.1. The nature of the variations was irregular, arising from the normal operation and continual adjustment of the electrical grid based on the present load and demand on the system. The long term average grid frequency is of course maintained at 50 Hz. For this reason steady-state test runs at each individual operating point are time-averaged over a period of five minutes to mitigate any effects of turbine speed variation.

For the construction of the turbine efficiency hill chart tests were conducted for guide vane opening angles ranging between 20° and 35°, equivalent to a range of between 53% and 92% full stroke. Tests were conducted at net specific energy values of 34.3 J/kg up to 83.4 J/kg (3.5 to 8.5 m net head) in 5 J/kg increments to ensure a thorough exploration of the entire operating range. In total over 300 individual 5 minute tests were conducted in constructing the Hill chart diagram and the supporting steady state performance tests.

5.3.2 Performance evaluation

The efficiency of the designed micro-hydro pump-turbine unit over the entire operating range is illustrated in the hill chart shown in Figure 5.2. Overall efficiency contours are given while guide vane opening angles are represented by the dotted lines running diagonally left
Figure 5.2: Performance Hill Chart diagram of the new UTAS micro-hydro pump-turbine, designed as part of the current study.

to right. The best efficiency point was found to be at $E_{nD} = 7.27$, $Q_{nD} = 0.925$, corresponding to values of rated head, $H_r$, and flow rate, $Q_r$, of 5.98 m and 0.133 m$^3$/s respectively. The maximum, time-averaged, overall turbine efficiency obtained was 79.0 %, measured at turbine shaft. This result is in good agreement with the predicted best efficiency of a pump impeller acting as turbine which is generally reported to be within the range of $\pm 2\%$ of the best efficiency in pump operation [25, 108].

Of practical interest is the general flatness of the efficiency map, particularly around the BEP. This is also shown in Figure 5.3 presented for three cases of constant specific energy. Test cases chosen correspond to tank minimum supply level, BEP condition and tank maximum supply level. The variation in head represents a range of static head between approximately $0.85 \cdot H_r$ and $1.2 \cdot H_r$, a significant range in the context of micro-hydro installations and within expected bounds of PAT performance predictions. The efficiency in each case remains considerably high even with significant decreases in flow; indeed in all cases a 30\% flow reduction is required to see a 10\% reduction in overall efficiency, and a further 20\% decrease in flow for an additional 10\% efficiency reduction at rated $E_{nD}$. 
This is a significant improvement upon previously presented data on PAT performance by Yang et al. [113], Williams et al. [109], Derakhshan and Nourbakhsh [25] and Singh [94] that all report significant reduction in turbine operation efficiency due to the inherent lack of inlet flow control and associated large incidence losses. In the sphere of micro-hydro remote area power installations, the ability to operate efficiently at off-design conditions is extremely important as water storage reservoirs are often limited in capacity and catchments are subject to highly fluctuating, often seasonal inflows.

<table>
<thead>
<tr>
<th>Values at BEP</th>
<th>$n_{QE}$ [-]</th>
<th>$H_r$ [m]</th>
<th>$Q_r$ [m$^3$/s]</th>
<th>$N_r$ [rpm]</th>
<th>$P_r$ [kW]</th>
<th>$\eta$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump as rated</td>
<td>0.314</td>
<td>12.25</td>
<td>0.222</td>
<td>1450</td>
<td>34.0</td>
<td>78.5</td>
</tr>
<tr>
<td>Turbine predicted</td>
<td>0.278</td>
<td>4.38</td>
<td>0.139</td>
<td>750</td>
<td>4.68</td>
<td>78.5</td>
</tr>
<tr>
<td>Turbine actual</td>
<td>0.215</td>
<td>5.98</td>
<td>0.133</td>
<td>754</td>
<td>6.20</td>
<td>79.0</td>
</tr>
</tbody>
</table>

Performance of the micro pump-turbine unit in comparison to the parent pump and predicted turbine values is given in Table 5.1. It can be seen that while the maximum efficiency shows a slight increase of 0.5% from that in pumping mode, the net head and output power
at rated is considerably higher than predicted by the method of Sharma [91], while the flow rate in turbine operation is slightly less than that expected.

It is widely accepted that PATs require a higher head and flow at BEP point than in pump mode, with the head ratio generally the larger of the two. Ventrone et al. [103] suggests that this is due to a combination of a significant increase in head coefficient with increasing flow and the simultaneous decrease in mechanical losses relative to power output as well as a residual negative velocity at operating condition just greater than design. This is in contrast with the belief that slip, due to a relative rotational velocity within the blade passages, is only present in pump mode such that flow should be increased in order to minimise incidence losses. However a degree of slip is also present in turbine operation and an increase in incidence loss due to higher flow is balanced in some degree by the mechanical efficiency. This decreasing influence of mechanical losses can be seen in Figure 5.4 whereby the overall efficiency approaches the hydraulic efficiency at normalised flow $Q/Q_{BEP} > 1$. This is also evident in Figure 5.3 at low flow coefficients where the efficiency is maintained for higher specific energy while the lower specific energy curve shows sharper decline below approximately 70% $Q_{ND}$. The increase in net head required at BEP could also be due in part to the additional head loss across the distributor which is not accounted for in the method of Sharma [91].
Mechanical losses of the UTAS-MH unit were calculated based on experiments performed under no flow conditions following normal operation while still connected to the grid. With the inlet guide vanes fully closed air was injected into the draft tube, evacuating the runner chamber of water. In such operation the torque required to maintain synchronous speed is provided by the induction generator acting as a motor. As such the torque measured by the shaft strain gauge is negative, the magnitude of which is proportional to the power required to overcome losses. A similar test is performed without air admission to determine the increase in power required due to the added resistance of water within the runner chamber. The results of the above described tests are given in Table 5.2 in terms of rated power and compared to that of the Reece Power Station. It can be seen that the percentage of rated power required to maintain speed for the full-scale unit is significantly less than that of the micro-hydro unit as mechanical losses are proportionately much larger for the smaller output machine.

Table 5.2: Input power required to maintain synchronous speed with inlet guide vanes in the fully closed position (Q = 0 m³/s), normalised by rated power output.

<table>
<thead>
<tr>
<th>Operating condition</th>
<th>Reece PS</th>
<th>UTAS-MH</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spinning in air</td>
<td>-0.01</td>
<td>-0.05</td>
</tr>
<tr>
<td>Spinning in water</td>
<td>-0.13</td>
<td>-0.28</td>
</tr>
</tbody>
</table>

A summary of the UTAS-MH unit in terms of dimensionless parameters is given in Table 5.3. Values are also given for the Reece Francis turbine unit. With the exception of the cavitation coefficient and Reynolds number (see Section 4.2.1) the achieved performance parameters of the designed micro-hydro turbine show good agreement with the target values of the Reece unit. While the two machines are not geometrically, or hydraulically similar in the strict sense as defined in Section 4.2.1, the full scale and micro-hydro turbine are both mid-range mixed flow reaction machines with comparable degree of reaction. It may be assumed that, generally speaking, the internal flow patterns and energy transfer mechanism of the two machines are similar and that findings from the laboratory scale turbine may provide useful insight into the transient behaviour of the full scale unit, particularly concerning the start-up from TWD mode. Results, however, may not be meaningfully scaled up.
Table 5.3: Dimensionless terms based on rated values for the Reece Power Station and realised micro-hydro turbine rated values

<table>
<thead>
<tr>
<th>Dimensionless Parameter</th>
<th>Symbol</th>
<th>Definition (IEC 60193)</th>
<th>Reece</th>
<th>UTAS-MH</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy coefficient</td>
<td>$E_{nD}$</td>
<td>$E/(nD)^2$</td>
<td>7.08</td>
<td>7.27</td>
</tr>
<tr>
<td>Discharge coefficient</td>
<td>$Q_{nD}$</td>
<td>$Q/(nD^3)$</td>
<td>0.761</td>
<td>0.917</td>
</tr>
<tr>
<td>Power coefficient</td>
<td>$P_{nD}$</td>
<td>$P/(n^3r^2D^3)$</td>
<td>5.00</td>
<td>5.23</td>
</tr>
<tr>
<td>Specific speed</td>
<td>$n_{QE}$</td>
<td>$nQ^{0.5}/E^{0.75}$</td>
<td>0.201</td>
<td>0.215</td>
</tr>
<tr>
<td>Cavitation coefficient</td>
<td>$\sigma_{nD}$</td>
<td>$NPSE/(n^2D^2)$</td>
<td>0.088</td>
<td>1.25</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>Re</td>
<td>$\pi D^2n/\nu$</td>
<td>1.23x10^8</td>
<td>1.71x10^6</td>
</tr>
<tr>
<td>Froude number</td>
<td>Fr</td>
<td>$(E/gD)^{0.5}$</td>
<td>4.76</td>
<td>5.14</td>
</tr>
</tbody>
</table>

5.4 Weighted average efficiency (WAE)

In assessing the performance of micro-hydro installations the ability for a given unit to utilise the available head and discharge on site while also measuring the efficiency with which it is able to convert the available resources to deliverable output power must be considered. The WAE provides developers of micro-hydro installations a means to quantitatively compare generation alternatives. Singh [94] has noted that the Uttarakhand Renewable Energy Development Agency (UREDA), India, has used the definition of WAE in issuing tenders for micro-hydro projects stating a minimum compliance WAE of 60%.

As mentioned previously, it is well known that the major operational problem with pump-as-turbine units is the significant drop in efficiency away from installed best efficiency point. The installed best efficiency point of a PAT is highly likely to be different to the design BEP calculated using current PAT prediction theory. It is therefore unlikely to operate at peak efficiency upon installation and this fact should be taken into account when assessing micro-hydro options.

Furthermore when considering a site with a constant supply head, a PAT operating at part-load will have a reduced head utilisation, the excess pressure being throttled by a control valve. This decreased head utilisation wastes the potential power able to be extracted from the source.

As such in the calculation of the WAE the efficiency should be based on rated head, $H_r$ [m] at 100% load, rather than at the local operating head which is invariably less [94]. For PATs with no inherent inlet flow control, in order to operate at reduced power output both the head and flow must be decreased; or as would more likely be the case in a practical sense, if the available head increases or decreases the output power and flow will increase or
decrease accordingly accompanied by a drop in machine efficiency (assuming the machine was initially operating at BEP). In the case of the UTAS micro-hydro unit with added flow control it is possible to decrease power output without lowering the machine head by way of reducing the guide vane opening angle, and if the available head increases or decreases the guide vane position can be altered to minimise any observed efficiency reduction. For this reason in the following calculations of WAE the efficiency of the UTAS micro-hydro does not need to be adjusted at part-load as head utilisation remains constant.

As defined by Singh [94] the weighted average efficiency is determined at \(1.0 \cdot P_r\), \(0.8 \cdot P_r\) and \(0.6 \cdot P_r\), where \(P_r\) is the power output at rated, according to Eq. 5.15:

\[
WAE = (a_1 \cdot \eta_{t,1.0} \cdot \eta_{G,1.0}) + (a_2 \cdot \eta_{t,0.8} \cdot \eta_{G,0.8}) + (a_3 \cdot \eta_{t,0.6} \cdot \eta_{G,0.6})
\]

(5.15)

where \(a_1\), \(a_2\), \(a_3\) are weight factors for the three load conditions of \(1.0 \cdot P_r\), \(0.8 \cdot P_r\) and \(0.6 \cdot P_r\), respectively. The efficiencies, \(\eta\), of the turbine and generator are indicated with a subscript ‘t’, for turbine and subscript ‘G’ for generator while the subscript number represents the load condition. The weight factors represent the percentage of time the unit is expected to operate at the corresponding load point. To enable direct comparison with data from Singh [94] the weight factors will be taken as \(a_1 = 0.6\), \(a_2 = 0.2\) and \(a_3 = 0.2\) although these values can be adjusted based on known flow availability for a given site. Potential unit options would then all be assessed using the same nominated factors.

As an illustrative example a comparison of weighted average efficiency for the University of Tasmania micro-hydro (UTAS-MH) unit (\(n_{Q_E} = 0.215\)) and a PAT (\(n_{Q_E} = 0.119\)) as tested by Singh [94] is provided. Table 5.4 shows the performance characteristics of both turbine units.

<table>
<thead>
<tr>
<th>Load point</th>
<th>UTAS-MH</th>
<th>Optimised PAT [94]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load (kW)</td>
<td>6.2</td>
<td>22.8</td>
</tr>
<tr>
<td>Head, (H_{net}) (m)</td>
<td>5.98</td>
<td>5.98</td>
</tr>
<tr>
<td>Flow, (Q) (m(^3)/s)</td>
<td>0.133</td>
<td>0.110</td>
</tr>
<tr>
<td>Turbine efficiency, (\eta_t) (%) (corrected for head utilisation)</td>
<td>79.0</td>
<td>75.9</td>
</tr>
<tr>
<td>Generator efficiency, (\eta_G) (%)</td>
<td>88.5</td>
<td>85.0</td>
</tr>
<tr>
<td>(\eta_t \cdot \eta_G) (%)</td>
<td>69.9</td>
<td>64.5</td>
</tr>
</tbody>
</table>
The weighted average efficiency of the UTAS-MH based on Eq. 5.15 is:

\[ WAE_{UTAS-MH} = (0.6 \cdot 0.790 \cdot 0.885) + (0.2 \cdot 0.759 \cdot 0.850) + (0.2 \cdot 0.710 \cdot 0.830) = 66.6\% \] (5.16)

While the weighted average efficiency of the optimised PAT by Singh [94] is:

\[ WAE_{PAT} = (0.6 \cdot 0.834 \cdot 0.860) + (0.2 \cdot 0.731 \cdot 0.840) + (0.2 \cdot 0.614 \cdot 0.835) = 65.5\% \] (5.17)

As calculated above the WAE takes into consideration the efficiency of the chosen generator for the micro-hydro installation. As such this method of assessment is ideal for determining the total package that is most suitable for a given site. As can be seen the generator efficiency is an important factor in the overall unit output and WAE value. The generator chosen for the UTAS micro-hydro facility is considerably oversized for the current installed turbine unit and so does not reach greater than 41% of rated output power at the turbine BEP. A correctly sized unit operating in the upper range of generator efficiency would yield a more favourable WAE. Even so, in comparison with the PAT unit tested by Singh [94], the UTAS-MH generator efficiency is superior. As a result the UTAS-MH system weighted average efficiency is slightly higher than that of the optimised PAT system. To negate the effect of generator efficiency and assess the turbine unit itself a turbine weighted average efficiency, \( \text{WAE}_t \), may be calculated as:

\[ \text{WAE}_t = (a_1 \cdot \eta_{t,1.0}) + (a_2 \cdot \eta_{t,0.8}) + (a_3 \cdot \eta_{t,0.6}) \] (5.18)

Using the same weight factors as above the two unit’s \( \text{WAE}_t \)'s are now:

\[ \text{WAE}_{t,UTAS-MH} = (0.6 \cdot 0.790) + (0.2 \cdot 0.759) + (0.2 \cdot 0.710) = 76.8\% \] (5.19)

\[ \text{WAE}_{t,PAT} = (0.6 \cdot 0.834) + (0.2 \cdot 0.731) + (0.2 \cdot 0.614) = 76.9\% \] (5.20)

In this case the \( \text{WAE}_t \)’s of the two units are practically equal at ~77%. It must be noted that the PAT as tested by Singh [94] had undergone numerous optimisation stages including impeller blade tip rounding and tapered ring inserts, resulting in the rather high peak efficiency of 83.4%. The UTAS-MH runner on the other hand had not undergone any modifications and the peak efficiency of 79% was within the expected ±2% of parent pump peak efficiency [25, 108]. Even so the improved performance at off-design conditions has essentially made up for a 4.4% disparity in peak efficiency. If turbine efficiency is normalised by peak efficiency the benefit of guide vane control is clear, Figure 5.5a and 5.5b.
(a) PAT efficiency at rated head  

(b) Normalised PAT efficiency

Figure 5.5: Weighted-average-efficiency comparison of the new UTAS-MH and previously published conventional PATs at three load points: $1.0 \cdot P_r$, $0.8 \cdot P_r$ and $0.6 \cdot P_r$. The optimised PATs (A) and (B) presented by Singh [94] are of specific speed $(n_{QE}) = 0.119$ and $0.284$, respectively, while Derakhshan et al. [24] presents a conventional PAT of specific speed $(n_{QE}) = 0.049$ before (C), and after (D), blade profile optimisation.

When considering that a micro-hydro installation will never exclusively run at BEP conditions, the WAE criterion provides a realistic assessment method for comparing turbine units. In fact, owing to the uncertainty of prediction models, the WAE as calculated is likely to report slightly higher values than should be expected because the likelihood of the installed BEP matching the design BEP is relatively low. Being somewhat conservative it may be more prudent to calculate the WAE based on load points of $0.9 \cdot P_r$, $0.8 \cdot P_r$ and $0.6 \cdot P_r$ instead, in which case the UTAS-MH WAE would compare even more favourably.

The new design of the UTAS-MH has shown that the inclusion of inlet flow control has a significant effect on the overall performance of small scale hydro turbines by increasing the effective range of operation. Such turbines are ideally suited to any site where head or flow variations are expected. In such circumstances the traditional solution was to install multiple smaller units to cover the expected conditions adding significant cost to the project.

5.4.1 Performance at low flow conditions

In addition to measurements taken across the normal micro-hydro operating range as presented in the preceding sections, studies were undertaken at extremely low flow conditions largely below the point of positive power production. Tests were performed in semi-closed loop configuration as operating heads required were below the obtainable range as provided by the gravity fed configuration.
The turbine was brought on-line and grid connected using the same method and test procedure as described in Section 5.3.1. The pump speed control dial was then used to manually bring down the supply head and flow to the required operating point while synchronous speed was maintained through connection to the grid. In the case where low test specific energies were required the downstream gate was raised to prevent the formation of the cavitating draft tube vortex. The range of operating points explored is shown in Figure 5.6 and represents flow rates between $0.15 \cdot Q_r$ to $0.45 \cdot Q_r$ requiring test specific energies from $9.09 \text{ Jkg}^{-1}$ to $91.7 \text{ Jkg}^{-1}$. Performance is plotted in terms of dimensionless torque coefficient, $T_nD$, whereby a negative value indicates that power is being supplied from the grid to the unit, the generator effectively being driven as a motor in order to overcome bearing frictional forces and the resistance to rotation due to the low momentum water volume present within the runner chamber.

For the range of flows tested performance is generally below the positive torque region indicating that the incoming low energy flow is not sufficient to overcome the frictional and resistive forces present.

The results indicate that for a given guide vane opening angle, a drop in turbine flow rate results in a corresponding decrease in applied shaft torque. If, for a given guide vane opening angle, the flow rate entering the turbine runner is sufficiently low, the torque coefficient will be negative. Furthermore, the magnitude of the negative torque will increase
with opening angle unless the flow entering the runner is also adequately increased. The implications of this behaviour will be further discussed in Chapter 8.

The extent to which the resistive forces of the water has on the input power required is illustrated in Figure 5.7 for a constant machine flow rate of $Q = 20 \, \text{L/s}$. To provide some quantification of the influence of the water churn, tests were performed in which $\sim 15 \, \text{L/s}$ of air, measured using a standard 0 - 20 L/s Thorpe tube flowmeter, was injected into the draft tube during operation at low flow rate, decreasing the amount of water within the runner proper. At $Q = 0.15 \cdot Q_r$ the admission of air decreased the power consumption at $22^\circ$ opening by over half, equivalent to 17.3% of rated $T_n D$, while at $3^\circ$ opening the shaft torque became positive increasing by 11.7% of rated. This result will be looked at in more detail in a following chapter. Similar trends were seen for all low range flow rates tested.

While the range of test conditions presented in Figures 5.6 and 5.7 are not practically obtainable for a traditional site with a relatively constant head supply, the behaviour of the UTAS-MH unit under such conditions provides some valuable insight that will be used in the following chapters in developing a new simulation model. In relation to FCAS provision it is already clear that the presence of low energy water within the runner chamber during low inflow will be detrimental to the production of net positive power.
5.5 Chapter Summary

A 6.2 kW micro-hydro scale pump-turbine unit has been designed, built and tested at the University of Tasmania. The design concept of the micro-hydro turbine, presented in Chapter 4, offers a considerable improvement over traditional pump as turbine (PAT) systems due to the use of existing pump components and the incorporation of guide vane flow control within a custom designed case assembly, a functionality that traditional PATs lack.

Performance over the full operating range is presented in the form of a turbine performance hill chart demonstrating near peak efficiency over a wide range of head and flow conditions not seen with traditional PAT systems. Maximum efficiency was determined to be 79%, marginally higher than the parent pump maximum efficiency. The design operating point, as calculated by the method of Sharma [91], was determined to be at a head of 4.38 m and a discharge of 0.139 m$^3$/s. The installed best efficiency operating point, however, was found to be at a head of 5.98 m and a discharge of 0.133 m$^3$/s. Accordingly, the predicted output power was 4.7 kW, while the achieved output power at best efficiency was 6.2 kW.

In the majority of micro-hydro systems, the turbine is often required to operate well away from the design operating condition due to highly fluctuating head and flows, whether in a remote area power system or an industrial setting. This, coupled with the well acknowledged difficulty in predicting the installed best efficiency operating point, results in PATs generally producing less power, and at a lower efficiency, than expected.

A more indicative measure of the performance of a micro-hydro turbine installation is given in the form of a weighted-average-efficiency (WAE) as defined by Singh [94]. The designed and tested UTAS-MH unit was found to achieve a WAE of 67%, while a similar PAT which had undergone numerous modifications, resulting in a substantially higher peak efficiency of 84%, was reported to have a WAE of 66% [94].

The resulting unit, while slightly more complex than a traditional PAT system, demonstrates near peak efficiency operation over a wide range of flow conditions, thus addressing the main drawback of pumps operating in turbine mode.
Chapter 6

Rapid Start-up of a Laboratory
Micro-Hydro Turbine

This chapter presents results of rapid start-up tests carried out on the University of Tasmania micro-hydro unit (UTAS-MH) from a depressed tail water mode of operation. Work contained in this chapter was previously presented in Giosio et al. [33]. Tests were performed with the unit on-line and connected to the grid such that the machine was spinning at near synchronous speed, effectively being driven as a motor, while the guide vanes were closed and the tail water depressed prior to the transition to generation mode. Operation in this manner replicates the proposed mode of operation for full scale units identified for their potential to provide fast FCAS. Unlike full scale operation, the trigger for switching from TWD mode to generation is performed manually via direct guide vane actuation rather than the detection of a frequency deviation and/or detection of a high negative rate of change of system frequency.

The current chapter focuses on the effect of two key parameters: guide vane actuation rate (GV rate) and the initial tail water depression level (TWD level), as identified in Giosio et al. [31]. The establishment of the magnitude and type of influence of these two parameters was deemed particularly important in relation to the provision of FCAS and the origin of the adverse power draw identified in the early stages of transition in initial full scale studies. Furthermore, these two parameters are, to a limited degree, able to be adjusted at the full scale level without the need for additional infrastructure or machine modifications. Other parameters which may increase FCAS production that do require additional equipment or modifications are discussed in Chapter 8.
Transient measurements of rotational speed, upstream and downstream pressure, shaft torque and guide vane stroke are recorded simultaneously at a sample rate of 2 kHz. Key findings are presented in the time domain in terms of shaft mechanical power and penstock pressure in relation to guide vane position.

### 6.1 Scope of investigation

The objective of the transient laboratory test series was to establish the influence of two identified parameters, GV opening rate and TWD level, on the ability of a hydro turbine to rapidly transition from a modified synchronous condenser mode to generation mode. Therefore during testing all other variables, such as supply tank level and downstream gate height, were held constant. All tests were performed at rated conditions with a net turbine head of 5.9 metres.

Due to differences in scale, in order to compare laboratory tests with full scale results, time measurements during the transition tests are non-dimensionalised based on the water acceleration time constant, $T_w$ [s], for each unit, as suggested by Abreu et al. [1], according to Eq. 6.1.

$$\tilde{t} = \frac{t}{T_w}$$

where

$$T_w = \frac{Q_r L}{gAH_r}$$

as defined previously in Section 4.2.1.

In the following results the GV opening rate is described as a value of non-dimensional time, corresponding to the time required to reach rated GV position under continuous, single stage opening for the given opening rate, that is

$$\tilde{t}_{GV} = \frac{\text{physical time required to reach rated GV opening}}{T_w}$$

As such, a $\tilde{t}_{GV}$ of 2 still represents an opening rate three times as fast as a $\tilde{t}_{GV}$ of 6, however guide vane opening rates between full-scale and the more rapid opening of the micro-hydro unit may now be compared. Furthermore, penstock pressure events and unit power response can be related between full- and micro-hydro scale tests in relation to the non-dimensional time, $\tilde{t}$, with events being propagated at the same (relative) frequency.
The level of tail water depression is given as a proportion of turbine runner diameter, $D$, and indicates the vertical distance of the tail water free surface below the runner exit. As such a TWD level of $0.5D$ is depressed more than a TWD level of $0.25D$. A higher value of TWD level indicates a greater volume of air contained within the runner chamber and upper draft tube section, however due to the internal geometry of the machine it is not a linear relationship and a TWD level of $0.5D$ does not have an air volume twice that of a TWD level of $0.25D$.

The selection of the presented start-up procedures was based on the current capabilities of the full-scale Francis turbine machine number 2 at the Reece Power Station. Guide vane opening rates tested are chosen at $0.5\times$, $1\times$ and $2\times$ the equivalent existing maximum initial opening rate. Selection of water depression level is based on the positions of the two water level indicators in the full-scale unit draft tube, discussed in Section 3.1.1. A summary of presented tests is given in Table 6.1.

Table 6.1: Testing summary of selected laboratory transient start-up tests performed on the University of Tasmania micro-hydro unit.

<table>
<thead>
<tr>
<th>Test ID</th>
<th>TWD level</th>
<th>$\overline{t}_{GV}$, s</th>
<th>Test description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_M1$</td>
<td>$0.23D$</td>
<td>17.6</td>
<td>High TWD indicator level, 0.5x existing max. opening rate</td>
</tr>
<tr>
<td>$T_M2$</td>
<td>$0.23D$</td>
<td>8.97</td>
<td>High TWD indicator level, existing max. opening rate</td>
</tr>
<tr>
<td>$T_M3$</td>
<td>$0.56D$</td>
<td>17.6</td>
<td>Low TWD indicator level, 0.5x existing max. opening rate</td>
</tr>
<tr>
<td>$T_M4$</td>
<td>$0.56D$</td>
<td>8.97</td>
<td>Low TWD indicator level, existing max. opening rate</td>
</tr>
<tr>
<td>$T_M5$</td>
<td>$0.56D$</td>
<td>4.39</td>
<td>Low TWD indicator level, 2x existing max. opening rate</td>
</tr>
<tr>
<td>$T_M6$</td>
<td>$1.0D$</td>
<td>8.97</td>
<td>Below low TWD indicator level, existing max. opening rate</td>
</tr>
</tbody>
</table>

The R6 FCAS contribution is calculated as a function of power produced within a six second window from the time of detection of a frequency deviation, which when non-dimensionalised by the full-scale unit water start time corresponds to $\overline{t} = 3.64$. As such tests were performed from the fully closed GV position to $10.3^\circ$ opening, corresponding to 0.35 of rated, at which point the GV position was held constant before closing and re-calibrating for the following test. For ease of comparison, all values of power, pressure and GV position given are presented as normalized by rated values, $P_r$, $H_r$ and $GV_r$, respectively, where rated GV position is defined as the guide vane servomotor stroke required under rated head and flow to produce rated power.
6.2 Flow visualisation during rapid start-up from TWD mode

The rapid transition from TWD mode to generation mode for test case $T_M5$ is shown sequentially in Figures 6.1(i) - (xii).

Prior to guide vane actuation, depending on the level of tail water depression, a rotating wave-like phenomenon can be observed on the tail water free surface below the runner, see Figure 6.1(i), as previously reported by Ceravola et al. [15]. Indeed, the rotating wave reaches a maximum amplitude at a TWD level of approximately $1.0D$, however, more importantly, the amplitude diminishes as the level is decreased towards the runner and at no level is the surface seen to impinge on the turbine runner as suggested by Magsaysay et al. [64], whether in TWD mode or during the transition to generation mode.

Following GV opening, the initial surge of water enters the runner chamber, proceeding into the draft tube cone after passing through the runner, with a highly tangential flow as seen in Figures 6.1(ii) - (iv). During these initial stages an inner cylinder of air remains within the draft tube as the incoming water on the periphery works to push the surface down and begins entraining the air in the rotational flow, Figures 6.1(iv) - (vii).

This results in a highly aerated flow region that grows in size to fill almost the entire draft tube as air begins to be expelled beyond the draft tube bend, Figures 6.1(viii) - (x). For a considerable period of the transition an air/bubbly-region interface remains clearly visible below the runner exit.

As the two-phase flow becomes more established a number of rotating regions, equal to the blade number, can be seen to form (Figures 6.1(ix) & (x)) eventually combining into one large poorly defined bubble vortex, Figure 6.1(xii).

Depending on the final operating condition, this vortex coalesces and reduces in size as the remaining air is entrained and transported downstream through the draft tube until normal operation is established. This, however, was not realised during test case $T_M5$, shown below, in which GV opening was stopped at 0.35 of rated.

The process described above, and illustrated in Figures 6.1(i) - (xii) below, occurs within a time frame of approximately 1.8 s ($\tau = 5.0$).
Figure 6.1: $T_{M5}$ Transition of the UTAS-MH unit from TWD mode to generation (1 of 3).
Figure 6.1: $T_{M5}$ Transition of the UTAS-MH unit from TWD mode to generation (2 of 3).
Figure 6.1: $T_{M5}$ Transition of the UTAS-MH unit from TWD mode to generation (3 of 3).
6.3 Effect of single stage guide vane opening rate on transition

The influence of guide vane opening rate with a constant TWD level of 0.56\(D\) is shown in Figures 6.2 and 6.3 presenting results of start-up procedures \(T_{M3}\), \(T_{M4}\) & \(T_{M5}\). Figure 6.2 presents normalised guide vane position and power during transition while Figure 6.3 presents the resulting pressure transient in the penstock due to rapid GV movement.

Prior to start-up the turbine unit draws 0.05\(\cdot P_r\) of power in order to overcome losses. At \(\bar{t} = 0\) the GV command is sent and GV movement commences soon thereafter. In each case an increased reverse power flow is observed. The slower opening \(T_{M3}\) shows the largest magnitude power draw of the order of 0.07\(\cdot P_r\) while the fastest opening rate, test case \(T_{M5}\), is of the order of 0.06\(\cdot P_r\).

As such, it is evident that the recovery and return from reverse power flow to power generation is increased as GV opening rate is increased. This is further aided by the location of the power dip in relation to initial movement with the more rapid opening case, \(T_{M5}\), experiencing peak negative power flow substantially earlier than the slower case, \(T_{M3}\).

The point at which the power starts to increase, however, does not seem to be a function of GV position, even though a more rapid opening does lead to an earlier recovery. The normalised GV opening at the point of peak power draw for test cases \(T_{M3}\), \(T_{M4}\) & \(T_{M5}\) is 0.12, 0.08 and 0.06, respectively. This suggests that another factor is preventing the ability of the turbine to produce power, rather than solely the GV opening angle. Furthermore, there is no correlation to the penstock pressure response in regards to the point at which the power draw is at its maximum. The penstock pressure at the time of peak power draw for test case \(T_{M5}\) is 0.8\(\cdot H_r\), while the pressure for the slowest opening rate, \(T_{M3}\) is 0.9\(\cdot H_r\) at the same instant.

The corresponding penstock transients show that a higher opening rate significantly increases the magnitude of the low pressure transient which can be seen to increase from 11% to 28% of rated turbine head. Faster opening rates also display more pronounced and sustained hydraulic transients for the remainder of the transient test.

In each case the power response immediately following the negative incursion shows a high rate of power increase followed by a levelling off. This decrease in the rate at which output power is produced occurs in each case and, again, there is no direct correlation to penstock pressure. The sudden increase in rate of power generation seen at \(\bar{t} = 2.6\) in \(T_{M5}\) is due to the deceleration of guide vane movement upon reaching the 10.3° set point. This movement consequently increases the turbine inlet pressure, aiding power production, and initiating a subsequent transient event within the penstock.
Figure 6.2: GV position and mechanical power output during rapid transition of the micro-hydro unit for varying GV opening rates at TWD = 0.56D.

Figure 6.3: Upstream inlet pressure during rapid transition of the micro-hydro unit for varying GV opening rates at TWD = 0.56D.
6.4 Effect of tail water depression level on transition

The effect of tail water depression is much less significant than the rate of guide vane opening as seen in Figures 6.4 and 6.5 presenting results of test cases $T_M2$, $T_M4$ & $T_M6$. Figure 6.4 presents three start-up procedures with varying levels of TWD and a constant setting of guide vane opening rate $\bar{t}_{GV} = 8.79$, the current maximum opening rate at full scale. In all cases the power reversal is clearly present with a consistent magnitude of $0.065\cdot P_r$. The magnitude of the penstock pressure reduction is the same in each case at $18\% H_r$ and is unaltered by the volume of air initially contained in the draft tube prior to start-up.

The level of depression does seem to have an effect on the frequency of pressure transients within the penstock following GV opening with a higher value of TWD level resulting in higher frequency transients.

By considering the results obtained for varying TWD levels and through the visual observations presented, it has been established that the location of the free surface in the draft tube prior to transition to generation mode has negligible influence on either the peak magnitude or duration of the negative power flow. Moreover, the nature of the power response following rapid start-up for varying initial TWD levels is shown to be highly repeatable.

6.5 Comparison to full scale performance

Transient data acquired during rapid start-up feasibility studies of the 116 MW Reece Francis turbine unit are compared to scale model test results in Figure 6.6 & Figure 6.7. The turbine is of a similar specific speed to the experimental model. Data for penstock pressure, active power and guide vane servomotor position are normalised based on rated values. All measurements at full-scale were taken at a sampling frequency of 20 Hz. The discrepancy in power draw during TWD mode between model and full-scale unit can largely be attributed to scale effects. The full-scale test presented in both Figure 6.6 and Figure 6.7 is for rapid start-up from TWD mode at maximum guide vane opening rate, $\bar{t} = 8.79$, and a low TWD level of TWD = $0.56D$.

The transient response is compared in Figure 6.6 with the analogous experimental start-up procedure $T_M4$. While the general trend of the response agrees well with full-scale data there are some key discrepancies. The most notable in terms of FCAS provision is the difference in magnitude of the initial reverse power draw. At full scale this equates to 5% of
Figure 6.4: Guide vane position and mechanical power output during rapid transition of the micro-hydro unit for varying TWD levels at $\tau = 8.79$.

Figure 6.5: Upstream inlet pressure during rapid transition of the micro-hydro unit for varying TWD levels at $\tau = 8.79$. 
Figure 6.6: Comparison of experimental ($T_{M4}$) and full-scale ($T_{R6}$) response during rapid start-up from TWD mode

Figure 6.7: Comparison of experimental ($T_{M5}$) and full-scale ($T_{R6}$) response during rapid start-up from TWD mode
rated output whereas experimental results indicate only a 1% of rated additional power draw. The severity of power oscillations in early stages of positive generation are also increased at full scale. Furthermore the magnitude of the initial drop in inlet pressure, and subsequent reflection, is approximately twice that seen on the reduced scale model. This may be attributed to the GV opening profile at full-scale which indicates a slightly more rapid initial rate before a constant rate is established. The diminished reflections may also results from the slight flexibility inherent in the laboratory PE pipe work.

In order to match the initial inlet pressure reduction and determine the consequent effect on power output the same full-scale data is compared to experimental start-up procedure $T_{M5}$ with an opening rate equivalent to twice the existing full-scale maximum. While the inlet pressure condition was matched, the scale model does not experience a power flow equivalent to the full-scale unit, indeed it is diminished. Again the magnitude of the returning hydraulic transient in the scale model is significantly reduced.

In all test cases ($T_{M1}$ to $T_{M6}$) a sharp rise in shaft power is seen following peak power draw. In each case this appears to occur following the return of the high pressure wave. In each case, at this point, the available hydraulic power is not yet sufficient to overcome the mechanical losses and the unit is still drawing power from the grid. This correlation is not seen on the full-scale unit, however, which is still experiencing an increasing negative power flow when the high pressure transient is seen at the inlet. Indeed peak negative power flow is seen to occur directly after the returning high pressure transient at inlet indicating that another mechanism is contributing to the prolonged power draw.

### 6.6 Chapter Summary

This chapter has presented key laboratory rapid start-up tests performed on the UTAS-MH turbine unit. Tests were conducted starting from a TWD mode whereby the micro-hydro unit was brought online and grid connected before fully closing the turbine guide vanes. Compressed air is then introduced into the draft tube cone in order to depress the tail water below the runner exit plane. During operation in TWD mode the required power draw from the grid to maintain near synchronous speed was 5% of rated power. This is compared to the much smaller 1% of rated power at full-scale due to scaling effects on friction losses.

The increased negative power incursion observed at full-scale was also observed in the laboratory, however the subsequent power oscillations were not observed due to the differing methods of power measurement in each case. This, however, proved to offer a valuable
insight into the nature and cause of the full-scale response. The magnitude of the additional power draw was much less on the micro-hydro unit, where start-up procedure $T_{M4}$ produced a $0.01 \cdot P_r$ power incursion compared to the analogous full-scale test $T_{R6}$ of $0.05 \cdot P_r$.

The influence of two key parameters, guide vane opening angle and initial TWD level, were investigated in relation to the effect of each, individually, on the measured power response. The rate of single stage guide vane opening was shown to significantly reduce the transition time from the moment of initial guide vane movement to the production of positive power. Conversely, the power response was found to be independent of the initial level of tail water depression.

Flow visualisation within the draft tube cone during rapid transition from TWD mode is presented and discussed. Most notably, in the proposed method of transition in which the air cavity is not evacuated prior to guide vane movement, the tail water free surface does not impinge on the runner at any stage during the transition. The results presented represent the first time that the transition of a hydro turbine from TWD mode to generation has been visualised, and documented in a laboratory setting.

The results presented in this chapter will be further discussed in Chapter 8 when describing the proposed transition mechanism.
Chapter 7

Hydraulic Modelling of a Single Machine Francis Turbine Power Plant

This chapter presents a comprehensive one-dimensional hydraulic model of a single machine Francis turbine power station for transient simulation and analysis. The model is based on the equations of motion and continuity, while the conventional representation of the hydraulic turbine as an orifice is refined to account for machine behaviour away from normal operating range using effective machine geometry. The new model remains valid at low inlet guide vane opening angles, as well as for large magnitude transient events where traditional one-dimensional models tend to break down.

The chapter begins with a description of the Reece Power Station hydraulic layout. The development of the non-linear elastic waterway model is discussed including mathematical derivations, assumptions and limitations. The structure and formulation of the model, implemented using the commercial MATLAB/Simulink software package, is presented along with the identification of key hydraulic model parameters.

Finally, the numerical model is validated against commissioning tests performed at Reece during the 2012 governor retuning return to service, with simulation results also compared to two conventional models.
7.1 Reece Power Station Hydraulic Layout

Figure 7.1: Schematic layout of the Hydro Tasmania Reece Power Station showing conduit details [43].

The Reece Power Station has been identified by Hydro Tasmania for its potential and suitability to operate in a tail water depression (TWD) mode (described previously in Section 2.3) according to the design criteria given in Section 3.1.1. The simple hydraulic circuit configuration also makes the power station ideal for the performance validation of a new transient simulation model.

The Reece Power Station is situated at the base of the Reece Dam on Lake Pieman in the state’s west coast. The station houses two identical 116 MW vertical axis Francis turbine generating sets, identified simply as Reece machine number 1 (R1) and Reece machine number 2 (R2). Each unit has a completely independent and dedicated penstock and intake structure, and while these are more or less identical in design all specifications given in the following two chapters relate to the hydraulic circuit of R2 as this was the machine used in both the 2008 and 2012 fast raise tests presented in Chapter 3.

The rectangular intake structure for each machine is identical in design to those employed at the Bastyan and Mackintosh Power Stations and was designed to eliminate the formation of surface vortices and provide a smooth, streamlined inlet to the penstock [45, 74]. The penstock, which conveys water from Lake Pieman directly to the turbine, is comprised of a 164.5 m long concrete-lined conduit inclined at 24° to the horizontal followed by a steel lined 83.4 m long horizontal power tunnel. Both penstock sections have an internal diameter of 5.8 m and provide a net head of approximately 92 m to each unit depending
on reservoir and tail water levels. The penstock is reduced in diameter to 4.72 m prior to turbine spiral case inlet bringing the total penstock length to 250.3 m. Water is discharged from the turbine to the Pieman River via a 29.2 m long elbow type draft tube. The hydraulic circuit does not contain a surge tower or chamber. Furthermore, there is no main inlet valve installed, however each intake structure is equipped with a hydraulically operated gate designed to cut off full flow.

7.2 Transient Operation and Behaviour of Francis Turbine Power Plants

Changing demands of the modern electricity market over recent decades has seen a change in the way in which hydro-electric power plants are being utilised in electricity systems across the globe.

Increasingly, due to both the deregulation of markets and the growing contribution of non base load renewables, hydro-electric power plants are being relied upon to stabilise electricity networks. As such, the power plants are routinely subject to frequent changes in operating set-point, as well as load acceptance and load rejection procedures, leading to potentially dangerous transient phenomena.

Additionally, transient plant behaviour may arise through unplanned events. Electrical faults such as short-circuit, earth faults, loss of synchronisation, or numerous other scenarios which may lead to an emergency plant shutdown will also initiate transient behaviour, both mechanical and electrical in nature [76].

In the initial stages of hydro power plant design, transient studies determine critical features of the system such as penstock and power tunnel dimensions, and requirements for water-hammer control devices such as a surge tower or chamber, or pressure relief valves, based on the dynamic response to small magnitude disturbances [93]. Anticipated response and safety margins of operation may also be considered following large magnitude disturbances, while any proposed modification in regards to plant operation, system control or component upgrades will certainly require adequate transient analysis before implementation.

Transient analysis studies are therefore of paramount importance. The transient behaviour and response of a Francis turbine power plant involves hydraulic considerations, such as water inertia and fluid compressibility; mechanical considerations such as conduit wall flexibility and machine inertia; as well as electrical considerations such as regulation and
excitation systems, speed deviations and system inertia. Therefore a multi-physics model of the plant is required.

In terms of convenient computation time and required accuracy, detailed one-dimensional analytical models provide the most practical method of exploring the dynamic response of Francis turbine power plants subject to transient operation [76].

### 7.3 Fundamental Equations

Hydraulic transients are initiated in pipe systems following an adjustment, such as a valve movement, that results in a change in flow conditions. Rapid variations in flow may initiate local pressure changes that will then propagate throughout the pipe system. The pressure transients will generally undergo damping due to energy dissipation in the form of conduit friction and through component losses, however certain situations may result in resonance, and lead to potentially catastrophic damage.

The study of unsteady flow, and the propagation of hydraulic transients brought about by flow disturbances in piping systems, is governed by the equation of motion (conservation of momentum) and the continuity equation. The subject has a long history and a more comprehensive analysis can be found in numerous dedicated texts [17, 110]. However, a brief outline and derivation/description of the equations will be given below.

In developing a one-dimensional model of the dynamic behaviour of a water filled pipe the following assumptions are made [83]:

1. The pipe is uniform, and the pipe length \( L \) is much greater than the internal diameter \( D_p \).
2. Pressure and velocity do not vary across the conduit cross-section and flow is in the direction of the centreline axis, and
3. The pipe remains full of water and the pressure inside the pipe remains above the vapour pressure of water (i.e. no cavitation, or column separation).

Applying the equation of momentum to a control volume of length \( dx \) along the \( x \)-axis yields:

\[
\frac{\delta Q}{\delta t} + gA \frac{\delta H}{\delta x} + f \frac{Q|Q|}{2D_pA} = 0
\]  

(7.1)
Similarly, performing a mass balance on the control volume of length $dx$, the continuity equation may be expressed as:

$$\frac{gA}{a^2} \frac{\delta H}{\delta t} + \frac{\delta Q}{\delta x} = 0 \quad (7.2)$$

Where $g$ is the acceleration due to gravity, $A$ the conduit cross-sectional area, $D_p$ the internal pipe diameter and $a$ is the wave speed for the given pipe. The shear stress, $\tau_0$, applied to the control volume is expressed in terms of the Darcy-Weisbach friction factor, $f$, where

$$\tau_0 = \frac{1}{8A^2} f \rho Q |Q| \quad (7.3)$$

Equations 7.1 and 7.2 are coupled partial differential equations of momentum and continuity with dependent variables of hydraulic head, $H(x,t)$, and discharge, $Q(x,t)$, while the distance along the pipe, $x$, and time, $t$, are the independent variables.

### 7.4 Non-linear Francis Turbine Model

Standard non-linear hydro turbine hydraulic models are comprised of a component for calculating conduit dynamics as well as a component representing the actual turbine unit characteristics. The water conduit model may be modelled as either inelastic or elastic depending on the type of studies and scenarios to be tested by the model. The current section will give an overview of the derivation of both inelastic and elastic waterway models as well as two representations of the turbine characteristic leading to the two well known conventional one-dimensional hydro turbine hydraulic models as formulated by the IEEE Working Group [50] and Kundur [61].

#### 7.4.1 Inelastic water column assumption

Examining the continuity equation above, as the wave speed increases, i.e. as $a \to \infty$, or as $\delta H/\delta t \to 0$, Equation 7.2 reduces to [1]:

$$\frac{\delta Q}{\delta x} = 0 \quad (7.4)$$

Such that the discharge is purely a function of time, $Q = f(t)$. The two conditions above correspond to assumptions of an incompressible flow inside the conduit and a perfectly
rigid, non-deformable pipe, since the increase in mass within the control volume is due to the combination of the water compressibility and pipe elasticity [61].

The inelastic waterway model thus assumes a rigid conduit and water column. Therefore, a change in flow condition caused by a guide vane movement will result in an instantaneous change in flow throughout the whole system - the water column undergoing mass oscillation as a single body of fluid.

The momentum equation can be integrated to give:

$$\frac{\delta Q}{\delta t} = (H_0 - H - H_f) \frac{gA}{L}$$

Where $H_0$ is the static head of the water column between reservoir and tailrace, $H$ is the head at turbine admission, and the head loss due to friction, $H_f$, is given by:

$$H_f = \frac{fL}{2gD_pA^2}Q|Q| = k_pQ|Q|$$

The integrated unsteady momentum equation (Eq. 7.5) is the fundamental equation for modelling of turbine conduit dynamics as given in the foundational paper of the IEEE Working Group on Prime Mover and Energy Supply Models for System Dynamic Performance Studies [50].

The inelastic waterway model takes into account the inertia effect of the water column which is represented by the water acceleration time constant, $T_w$ [s]. Expressed in per unit form by dividing through by rated values Eq 7.5 becomes:

$$\frac{\delta Q}{\delta t} = \frac{1}{T_w} (H_0 - H - H_f)$$

where

$$T_w = \sum \frac{Q_{rated}L_i}{gA_iH_{rated}}$$

as previously defined (4.2.1). The water acceleration time constant represents a delay between guide vane movement and the time required for the turbine flow to respond. The typical functional block implementation of Equation 7.7 is shown in Figure 7.2.
Figure 7.2: Functional block implementation of the inelastic waterway model for determining conduit dynamics. The model calculates turbine flow by integrating the right hand side of Equation 7.7 for each simulation time step.

7.4.2 Elastic waterway model

The elastic waterway model offers a more detailed representation of the hydraulic system by considering the effect of fluid compressibility and pipe deformations, resulting in a finite wave speed within the conduit.

The effect on pressure and flow can be considerable and must be accounted for in the case where the plant penstock length is significant or the guide vane opening is considered "rapid", generating a flow change within a time interval less than then characteristic time of the plant \[ T_{plant} = \frac{2L}{a} \], where \( T_{plant} = \frac{2L}{a} \).

The same assumptions hold as given in Section 7.3, however unlike the assumption made for the inelastic model, the wave speed \( a \) is now considered to be finite. Accordingly, a rapid guide vane movement will give rise to a travelling pressure wave within the conduit due to the compression of water and the elasticity of the pipe wall, commonly referred to as the water hammer effect.

The propagation velocity of the pressure wave is a function both of the conduit pipe properties; including pipe diameter \( (D_p) \), wall thickness \( (e) \) and Young’s modulus of elasticity \( (E) \); as well as the bulk modulus of compression \( (K) \) of water.

The model assumes that the boundary condition at penstock inlet does not change (i.e. constant head) and that during transient operation flow will continue to enter the the conduit with the added water mass being accounted for by the conduit and water elasticity.
The governing equations of momentum and continuity (Eq. 7.1 and Eq. 7.2) may be solved using a number of approximate methods such as graphical, linear impedance, method of characteristics, or a finite difference method [111].

The method of characteristics (MoC) is a commonly used tool for analysis of pressure transients and has been incorporated into various models studying hydropower plant response with great success [17, 86, 111]. However, the method requires detailed knowledge of the system to be studied and requires significant set-up by an experienced user. Furthermore, boundary conditions are a function of spatial location and solutions are prone to numerical dampening. In the application of system optimisation, a MoC model requires to be set-up anew for each case, while application to another power station will require a completely new model set-up.

The linear impedance method is based on electrical transmission line theory where the head and flow are analogous to the transmission line voltage and current, respectively [61, 111].

The general solution of the partial differential equations in time and space, normalised by rated head and flow, is given by [82]:

\[
\begin{align*}
\bar{H}_2 &= \bar{H}_1 \text{sech}(T_e s) - Z \bar{Q}_2 \tanh(T_e s) - \bar{H}_f \\
\bar{Q}_1 &= \bar{Q}_2 \cosh(T_e s) + \frac{1}{Z} \sinh(T_e s)
\end{align*}
\]

(7.9)

(7.10)

where
- \(\bar{H}_2\) = per-unit hydraulic head at turbine inlet
- \(\bar{H}_1\) = per-unit hydraulic head at reservoir
- \(\bar{H}_f\) = per-unit flow conduit head losses
- \(\bar{Q}_2\) = per-unit turbine flow
- \(\bar{Q}_1\) = per-unit upper penstock flow
- \(Z\) = normalised hydraulic surge impedance = \(T_w / T_e\)
- \(T_e\) = elastic water time constant
- \(T_w\) = inelastic water time constant
- \(s\) = the Laplace operator

The elastic wave travel time is simply the total length of the water conduit, divided by the wave speed, \(T_e = L / a\), where the wave speed is a function of the conduit properties and the bulk modulus of water:

\[
a = \sqrt{\frac{1}{\rho \left( \frac{1}{K} + \frac{\nu E}{\rho} \right)}}
\]

(7.11)
where \( \rho \) = water density
\( K \) = the bulk modulus of water
\( D_p \) = internal penstock diameter
\( c \) = pipe constraint coefficient
\( e \) = penstock wall thickness
\( E \) = Young’s modulus of the pipe wall material

Depending on the construction and material of the water conduit the wave speed may vary from anywhere between 1000 m/s up to 1400 m/s.

For a hydro power plant with no surge tower and a constant upstream reservoir head, Eq. 7.9 may be simplified to:

\[
\bar{H}_2 = -\frac{ZQ_2 \tanh(T_e s)}{1 + e^{-2Te s}} - \bar{H}_f
\]

or,

\[
\bar{H}_2 = \frac{ZQ_2(1 - e^{-2Te s})}{1 + e^{-2Te s}} - \bar{H}_f
\]

The functional block implementation of the elastic waterway model as given by Eq. 7.13 is illustrated in Figure 7.3.

Figure 7.3: Functional block implementation of the elastic waterway model for determining conduit dynamics. The model calculates the hydraulic head across the turbine according to Equation 7.13.
7.4.3 Francis turbine characteristics

The representation of the Francis turbine itself is a key component of the non-linear one-dimensional model. While the waterway model is well tested and has been thoroughly validated by experimental data [17, 23, 111] the turbine model is a complex and much less understood element. Accurate modelling of the turbine characteristics is essential for predicting the plant behaviour over the full range of operating conditions. The performance of the turbine unit in terms of its output power, $P$, is a function of head ($H$), flow ($Q$), guide vane position ($g$), rotational speed ($N$), geometric parameters, water density ($\rho$), and viscosity ($\mu$) [74].

For an ideal, lossless turbine the power output of a Francis turbine is equal to the available power:

$$ P = \rho Q g H_{net} $$  \hfill (7.14)

In order to account for turbine inefficiencies the IEEE Working Group [50] recommends the following equation for calculating turbine output, $P_m$, in per unit form based on the generator MVA rating:

$$ P_m = A_t H (Q - Q_{nl}) - D G (N - N_r) $$  \hfill (7.15)

where

- $P_m$ = per-unit mechanical output power of the machine
- $A_t$ = turbine gain factor
- $H$ = per-unit head at turbine admission
- $Q$ = per-unit turbine flow
- $Q_{nl}$ = per-unit no-load flow
- $D$ = turbine damping factor
- $G$ = per-unit guide vane function
- $N$ = per-unit turbine rotational speed
- $N_r$ = per-unit turbine rated rotational speed

The head at turbine admission is calculated by considering the turbine as a valve, modelled by the standard orifice equation [61]:

$$ Q = G \sqrt{H} $$  \hfill (7.16)
For the conventional IEEE turbine model \( \bar{G} \) represents the per unit guide vane position such that \( \bar{G} = 1 \) at rated head and flow.

The turbine gain factor, \( A_t \), converts the calculated power output to turbine power in terms of the generator MVA base and takes into account the turbine gain [50]. The factor is assumed constant over the entire operating range and is defined as:

\[
A_t = \frac{\text{Turbine MW rating}}{(\text{Generator MVA rating})} \frac{H_r}{(Q_r - Q_{nl})}
\]

The no-load flow, \( Q_{nl} \), is used to account for a collection of losses including bearing friction, internal flow losses and windage losses in both the turbine and generator [74]. Friction losses are taken into account by the head term defined in Eq. 7.6 although this is often neglected in simple models.

A damping effect is also included based on the speed deviation. The damping factor, \( D \), is also assumed constant and, for Francis turbines, is given the arbitrary value of \( D = 0.5 \) by the IEEE Working Group [50].

The block diagram implementation of the non-linear model with inelastic waterway is given in Figure 7.4. This represents both the turbine and conduit dynamics, taking guide vane position and rotational speed as inputs to determine turbine mechanical power, or torque, as the output.

Figure 7.4: Functional block diagram of the inelastic IEEE non-linear turbine model [50] with non-dimensional gate position (\( \bar{G} \)) and speed deviation (\( d\bar{N} \)) as model inputs and mechanical power (\( \bar{P}_m \)) as output.
Kundur [61] presents a slightly different formulation of the turbine model which is also widely employed in hydro power transient analysis.

Kundur also assumes the turbine head-flow behaviour is described by the standard orifice equation as given in Eq. 7.16. However, the non-dimensional guide vane function $\bar{G}$ is the effective gate (guide vane) opening which is assumed to vary linearly from zero at no-load flow to a value of unity at full (rated) power output. The effective guide vane opening is therefore related to the actual gate position, $\bar{g}$, by the turbine gain factor $A_t$ as

$$\bar{G} = A_t \bar{g}$$  \hspace{1cm} (7.18)

where $\bar{g}$ is the actual guide vane position expressed in per unit using the maximum opening position as base [61]. The turbine gain factor, based on guide vane positions at no-load (nl) and full load (fl), is therefore

$$A_t = \frac{1}{\bar{g}_{fl} - \bar{g}_{nl}}$$  \hspace{1cm} (7.19)

This simple relationship is illustrated graphically by Kundur [61] in Figure 7.5.

Figure 7.5: Graphical representation between turbine gain factor and normalised guide vane position (from [61])

\[ ^1\text{not to be confused with the IEEE definition} \]
The mechanical power output is then simply:

\[ P_m = (\overline{Q} - \overline{Q}_{nt})\overline{H} \]  \hspace{1cm} (7.20)

which is then converted to the generator base by the factor, \( P_r \),

\[ P_r = \frac{\text{Turbine MW rating}}{\text{Generator MVA rating}} \]  \hspace{1cm} (7.21)

The turbine model presented by Kundur [61] does not include a damping factor in relation to speed deviations. The block diagram of the Kundur inelastic turbine model is given by Figure 7.6.

Figure 7.6: Functional block diagram by Kundur [61] with inelastic waterway assumption. Non-dimensionalised gate position (\( \overline{g} \)) is used as the model input, while non-dimensional turbine power (\( \overline{P}_m \)) is output.

Both standard turbine models assume single phase flow throughout thereby neglecting the possibility of cavitation, particularly at low load conditions, and the associated effects on pressure transients within the turbine, draft tube and penstock. Recently, considerably more complex models have been presented by Chen et al. [18], Koutnik et al. [58], and Nicolet [76] that attempt to take the cavitation compliance into account.
7.5 Model development

A new non-linear hydraulic model of a Francis turbine power plant has been developed based on the previously described conventional IEEE turbine and waterway model. The model takes into account the conduit dynamics as well as a simplified representation of the rotor dynamics and grid while providing a new formulation of the turbine model component. The conventional representation of the turbine as a simple orifice is replaced by a calculation of inlet and outlet velocity vectors and two-dimensional Euler turbomachinery theory to more accurately represent turbine behaviour over the a wider operating range.

The functional block diagram of a power system is given in Figure 7.7 indicating the scope of the current model. Servomotor position, \( S(t) \) [mm], is therefore the required system input, either as a programmed opening profile or taken from actual field test measurements, while electrical (active) power, \( P_{eo} \), is the output variable of primary interest.

The model was implemented in the Simulink environment, a graphical simulation package within the MATLAB commercial software platform [67]. As with the conventional model, the turbine flow and penstock pressure are calculated and used in the computation of turbine mechanical power output. Additionally, the generator rotor dynamics are considered using the Simulink Synchronous Machine block from the SimPower library from which the rotor speed deviation is calculated and fed back into the turbine model.

For comparison, simulation results from the conventional IEEE turbine and waterway model, as well as the Kundur variation, are given. The determination of the standard hydraulic model parameters and the development of the new turbine model are given in the following sections.
7.5.1 Determination of standard hydraulic model parameters

In systems modelling it is common practice to represent variables as per unit (normalised) values. The base value chosen to normalise by is somewhat arbitrary, however, must be consistent throughout all calculations. In terms of the hydraulic modelling of hydro power plants it is most common to specify base values as those corresponding to either the rated operating point or at full gate opening.

For the following simulations, base values are chosen at the best efficiency operating point according to the original manufacturers model test data [98]. The physical values at the best efficiency operating point of the Reece Power Station are summarised in Table 7.1.

Table 7.1: Base values used for expressing variables in the per unit system taken at the best efficiency operating point of the Reece Power Station

<table>
<thead>
<tr>
<th>Parameter value at BEP</th>
<th>Base value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Net head [m]</td>
<td>92</td>
</tr>
<tr>
<td>Flow rate [m$^3$/s]</td>
<td>128.1</td>
</tr>
<tr>
<td>Machine speed [rpm]</td>
<td>166.7</td>
</tr>
<tr>
<td>Actuator stroke (B.E.P.) [mm]</td>
<td>251</td>
</tr>
<tr>
<td>Actuator stroke (full) [mm]</td>
<td>290</td>
</tr>
<tr>
<td>Turbine rated power output [MW]</td>
<td>116</td>
</tr>
<tr>
<td>Generator rated power [MVA]</td>
<td>136</td>
</tr>
</tbody>
</table>

The per unit total static head, $H_0$, is defined as the difference in elevation between the upper reservoir surface level and the downstream tail water surface level, normalised by net head at BEP. Although 45 km upstream of the Pieman River mouth, the tail water level of the Reece Power Station is subject to tidal variations with expected no flow levels of between 0.8 m to 5.0 m above sea level (S.L.). During plant operation the tail water level can be expected to vary between S.L. 3.0 m (1 machine minimum level) up to S.L. 8.0 m (2 machines maximum). Although tail water levels were not measured during the test series presented in Chapter 3, a survey of previously recorded data over the span of a year indicates an approximate average tail water level of S.L. 3.5 m, which was assumed for the simulation model. The reservoir level was also not measured during the test series. However, during normal periods of operation the head water surface level can be expected to be between S.L. 91.7 m and S.L. 93.6 m. Full supply level of the dam is S.L. 97.5 m. For the purpose of transient simulation, with a constant downstream surface level of S.L. 3.5 m assumed, the upstream head was adjusted such that the simulated penstock pressure was equal to the test data prior to a given transient test. For all test cases presented in Section 7.6 the required upstream head was within the expected range.
Friction loss within the conduit is calculated assuming a quasi-static model [1, 11]. If the acceleration term of Eq 7.5 is set to zero (i.e. \( dQ/dt = 0 \)) the equation reduces to the classical Darcy-Weisbach equation for steady pipe flow as given in Eq. 7.6 where:

\[
k_p = \frac{fL}{2gD_pA^2}
\] (7.22)

The internal walls of the conduit are assumed to be in reasonable order, although penstocks are prone to biological growth and increased surface roughness due to wall deposits or general degradation [10]. As such a Darcy friction factor of \( f = 0.03 \) was assumed for the entire conduit based on experimental test data by Hydro Tasmania [43]. For implementation in the waterway model the loss coefficient, \( k_p \), is expressed in per unit as:

\[
f_p = \left( \frac{k_p}{A^2} \right) \left( \frac{Q^2}{2gH_r} \right)
\] (7.23)

The no-load flow, \( Q_{nl} \), is a term used in the conventional models to account for a collection of assumed constant losses. The no-load condition is defined as the operation point at which just enough power is being generated to account for losses at low flow resulting in a zero megawatt output. From R2 machine test data the no-load condition corresponds to a servomotor position of 34.5 mm, (although this varies slightly depending on reservoir head level). Using the calculated nonlinear guide vane function discussed in a following section (see Figure 7.10) and the standard orifice equation (Eq. 7.16), the no-load flow was calculated to be 12.75 m³/s. This is expressed as a per unit value based on BEP flow rate.

Turbine gain factors are calculated according to equations Eq. 7.17 and Eq. 7.19 for the conventional IEEE and Kundur turbine model respectively, using rated values as per Table 7.1. For the conventional IEEE model a damping factor of \( D = 0.5 \) is assumed as given by the IEEE Working Group [50].

The water acceleration time constant, \( T_w \), is calculated according to Eq. 7.8 with rated head and flow as given in Table 7.1. The conduit length is taken as the distance from intake structure to spiral case inlet, which for Reece machine R2 is 275.4 m while the conduit area is calculated from the diameter of 5.8 m, giving a water acceleration time constant of \( T_w = 1.65 \).

For the elastic waterway model, the pressure wave propagation speed is required. Based on Eq. 7.11, the wave speed in the concrete lining section was found to be 1386 m/s while for the steel lined section a slightly higher speed of 1473 m/s was calculated [83]. However, if the contact between the steel liner and the surrounding tunnel wall is such that an air gap exists, the wave speed may be reduced to as low as 722 m/s [43]. As the penstock
and tunnel condition was not known, an average value of 1000 m/s was used for the steel lined conduit. Implementing the elastic waterway equations (Eq. 7.9 and Eq. 7.10) requires a single value of wave speed so a weighted average was adopted based on the relative lengths of each section. As such the effective wave speed used in the model was 1260.6 m/s.

7.5.2 Direct calculation of torque from conservation of angular momentum

As will be shown in the following section, and as reported in the literature, the two conventional models (IEEE [50] and Kundur [61]) are inadequate for accurately calculating the dynamic response of Francis turbine power plants over the full operating range. Although both models provide a valuable base model which may be adapted for modelling a particular hydro power unit, there are still a number of inherent short-comings that make their use problematic, particularly at low machine output.

The inaccuracies generally arise from the rather simple formulation of the turbine model. The turbine power equation for the conventional IEEE model as given by Eq. 7.15 includes a number of correction factors, assumed constant over the operating range, the choice of which are either arbitrary or simply not sufficient to describe low load turbine operation. This is also true for the Kundur formulation.

A more realistic turbine model based on the actual principle of energy transfer for turbo-machines, which includes loss factors of known origin that can be readily determined from field test data, is presented in the following.

The proposed new turbine model is based on the Euler turbine equation (discussed in detail in Section 1.1.2), which is re-stated in Eq. 7.24:

\[
E_{th} = \omega (C_{u1} R_1 - C_{u2} R_2)
\]

(7.24)

In the development of this model the following assumption are made:

1. Euler turbine equation holds over the entire operating range
2. Original manufacturer scaled model test data accurately represents the installed turbine performance
3. The turbine is considered as axi-symmetric, with geometric parameters as defined in Figure 7.8, and
4. The working fluid is water in its liquid phase (i.e. the effects of cavitation on performance and pressure transient wave speed are neglected)
Figure 7.8: Turbine section view and geometrical definitions used in the formulation of the simulation model (adapted from [38]).

The new model retains the conventional IEEE elastic waterway model as shown in Figure 7.9 however the turbine model is replaced, essentially by a formulation of the Euler equation with appropriate loss components as determined in the following sections.

The model requires a set of steady-state data for calibration of model parameters of the unit to be studied that gives turbine performance, at rated head, over an adequate operational range in terms of servomotor position, flow rate, and output power. Additionally, rated speed must be known as well as some required unit geometry, namely the runner diameter at inlet mid-section, shroud outlet diameter, and diameters at the pressure measuring sections (generally spiral inlet and draft tube outlet). The development of the model from the above mentioned parameters will be discussed in the sections below.

Figure 7.9: Functional block diagram of the new non-linear turbine model with elastic waterway. The new model calculates the turbine power output based on the conservation of angular momentum incorporated within the Euler block.
Nonlinear guide vane function

In both of the standard models presented above, the conventional IEEE turbine model [50] and the Kundur formulation [61], the guide vane function is assumed to vary linearly with servomotor position. This is a significant simplification that has been addressed by numerous authors in the literature [23, 36, 74]. The guide vane function actually consists of two nonlinear relationships: the opening area in relation to servomotor position, as well as the nonlinear nature of the discharge coefficient itself [74]. This nonlinearity is usually addressed based on observations made during field tests either by way of a relationship between servomotor position and flow or servomotor position and electrical power [36].

The servomotor position \( S \) is determined from the measured guide vane opening area, \( a_g \) [mm], assuming a three segment piece-wise linear function based on actual full-scale measurements. Assuming a volumetric efficiency, \( \eta_v \approx 0.99 \), the guide vane function is determined based on scaled up flow rate determined from model test data at rated head as

\[
G = \frac{Q}{\sqrt{2gH}}
\]  

(7.25)

Where \( Q = f(S) \) is known and the hydraulic head across the turbine is calculated based on the known test net head and the inlet and outlet velocity heads calculated at the pressure measuring sections. The resulting nonlinear relationship between servomotor position (actuator stroke) and guide vane function is given in Figure 7.10.

![Figure 7.10: Calculated nonlinear guide vane function \( G \) based on Reece model test data over the full operation range as a function of servomotor position.](image)
Inlet and outlet turbine blade angles at design

In order to calculate the energy transferred to the runner by the incoming water, the blade angle, at both inlet and outlet, must be known. In reality the shape of the turbine runner blade is highly complex and three-dimensional and will not adequately be described by a single value. However Euler theory, based on the principle of the conservation of angular momentum, simplifies the analysis by considering an axi-symmetric geometry and the change in angular momentum between the blade inlet and outlet.

Figure 7.11: Velocity triangles at turbine inlet and outlet at design where the zero outlet swirl and zero inlet incidence is assumed.

The runner outlet blade angle is calculated based on the assumption of zero swirl component at best efficiency operating point. As such, the tangential component of absolute outlet flow, $C_{u_2}$, is equal to the angular velocity of the runner blade trailing edge, $U_2$. The assumed runner outlet angle, $\beta_{2b}$, can be determined from Eq. 7.26 and 7.27:

\[ C_{m_2} = \frac{Q}{A_2} \quad (7.26) \]
\[ U_2 = \left( \frac{C_{m_2}}{\tan \beta_{2b}} \right) \quad (7.27) \]

Similarly, the runner blade inlet angle may be determined assuming that the flow approaches the runner blade with zero incidence at design operating point. The meridional flow component is calculated taking into account a blockage effect of the runner blade inlet thickness, $t$, for the thirteen blades,

\[ C_{m_1} = \frac{Q}{\pi D_1 B_R - (13 \times B_R \times t)} \quad (7.28) \]

where $B_R$ is the runner passage height in metres. From the Euler equation (Eq. 7.24) multiplied by the mass flow rate ($\rho Q$), the peripheral component of the inlet absolute velocity, $C_{u_1}$, is determined by Eq. 7.29 based on the output power and flow from test data and the zero outlet swirl assumption. The assumed runner inlet angle, $\beta_{1b}$, is then determined from the inlet velocity triangle as given in Eq. 7.30:
\[ P_{th} = \rho Q \omega (C_{u1} R_1 - 0) \quad (7.29) \]
\[ C_{u1} = U_1 - \left( \frac{C_{m1}}{\tan \beta_{1b}} \right) \quad (7.30) \]

Equations 7.27 and 7.30 provide a calculation of the effective blade angles at turbine inlet and outlet based on velocity triangles and Euler turbine theory, calculated at the best efficiency point with the assumptions made in Figure 7.8. Actual blade angles may vary from those calculated and may also vary from hub to band.

**Determination of loss components and effective inlet flow angle**

The Euler turbine equation calculates the theoretical maximum specific energy transfer possible, \( E_{th} \), based on the incoming flow velocity, flow angle and turbine blade geometry. In reality there exist a number of system losses limiting the energy transfer achievable at a given operating point.

\[ E_{th} = g (H - \sum h_L) = E - \sum E_L \quad (7.31) \]

In a study examining the validity of the accepted scale effect (from model to prototype hydraulic machine) away from the optimum operating condition, Ida [47] describes a method for the determination of the loss distribution coefficient, \( V \), used in the scale up formula.

Ida [47] considers specific energy loss components incurred within each flow field of the hydraulic turbine, from upstream pressure measurement section to draft tube exit, along a representative streamline. Specific energy loss formulae are proposed for each loss type within the five identified flow domains such that the total specific energy loss may be expressed as:

\[ \sum E_L = E_c + E_s + E_G + E_R + E_D \quad (7.32) \]

where \( c \) represents the turbine spiral case, \( s \) the stay vane ring, \( G \) the guide vane distributor, \( R \) the runner and \( D \) the turbine draft tube.

Individual loss components are described in Table 7.2. Losses can be classified into being either frictional losses, or kinetic losses which include incidence loss, wake loss, residual swirl loss and draft tube diffusion and bend losses.
The residual swirl flow loss is calculated using the tangential component of the absolute
adopted.

For the current model a value of
where the loss coefficient, \( \zeta_{R_s} = 0.75 \sim 1.0 \), Albuquerque et al. [2] cites a similar incidence loss model in relation to the design of axial flow hydraulic turbines, however recommends a coefficient within the range of 0.5 to 0.7. For the current model a value of \( \zeta_{R_s} = 0.75 \) was adopted.

The residual swirl flow loss is calculated using the tangential component of the absolute

Of the kinetic energy losses, two were identified as being particularly influential at operation away from design. These were the shock or incidence loss at runner inlet, and the loss of residual swirl flow at runner outlet. The remaining kinetic energy losses, such as wake loss and other minor shock losses within the distributor, were considered negligible.

The incidence loss at runner inlet is based on the difference of peripheral component (\( \Delta W_R \)) of the relative flow velocities entering the runner and just beyond runner inlet \([47]\) as expressed in Eq. 7.33 and 7.34:

\[
E_{R_s} = \zeta_{R_s} (\Delta W_R^2 / 2) \quad \text{for positive incidence} \quad (7.33)
\]

\[
= \zeta_{R_s} (\{\Delta W_R \times \cos \alpha_{R1}\}^2 / 2) \quad \text{for negative incidence} \quad (7.34)
\]

\[
E_{R_s} = \zeta_{R_s} (\Delta W_R^2 / 2) \quad \text{for positive incidence} \quad (7.33)
\]

\[
= \zeta_{R_s} (\{\Delta W_R \times \cos \alpha_{R1}\}^2 / 2) \quad \text{for negative incidence} \quad (7.34)
\]

\[
E_{R_s} = \zeta_{R_s} (\Delta W_R^2 / 2) \quad \text{for positive incidence} \quad (7.33)
\]

\[
= \zeta_{R_s} (\{\Delta W_R \times \cos \alpha_{R1}\}^2 / 2) \quad \text{for negative incidence} \quad (7.34)
\]
flow velocity at runner exit, $C_{u2}$, as shown in Figure 7.12 according to:

$$E_{D_u} = \zeta_{D_u} \left( C_{u2}^2 / 2 \right) \quad (7.35)$$

where Ida [47] sets a value of $\zeta_{D_u} = 1.0$, while Albuquerque et al. [2] recommends a loss coefficient of $\zeta_{D_u} = 0.2 \sim 0.4$. Based on Reece model test data a value of $\zeta_{D_u} = 1.0$ is used in the simulation model.

![Velocity triangles at turbine inlet and outlet for $Q < Q_r$.](image)

**Figure 7.12**: Velocity triangles at turbine inlet and outlet for $Q < Q_r$.

Friction losses are present in each of the flow zones, the draft tube friction losses being included within the draft tube diffusion loss, $E_{Du}$. Recommended values, or equations, for the various skin friction coefficients, $C_f$, are given. In general, the friction loss formulae take the form:

$$E_{i_f} \propto v_i^2 / 2 \quad (7.36)$$

where $E_{i_f}$ is the specific energy frictional loss of the particular flow domain, $i$, and $v_i$ is the characteristic velocity of the domain. Considering the system as a whole, the total frictional losses, $E_{tot,f}$, may be combined such that:

$$E_{tot,f} = k_f Q^2 \quad (7.37)$$

where $k_f$ is the combined friction loss coefficient.

Since at design operating point both the runner inlet incidence and residual outlet swirl velocity are close to zero, losses at best efficiency point can be assumed to be almost entirely due to hydraulic friction losses across the various fluid domains. As such, Eq. 7.31 becomes:

$$E_{th} = gH - E_{tot,f} = gH - k_f Q^2, \quad \text{where } k_f = 0.022 \quad (7.38)$$

For operation away from design, the assumed model for turbine specific energy transfer is:

$$E_{th} = gH - (E_{Re} + E_{Du} + k_f Q^2) \quad (7.39)$$
Where both $E_{th}$ and $E_{Rs}$ are a function of the inlet flow angle $\alpha_1$. Equation 7.39 can therefore be solved over the entire operating range to determine the effective runner inlet flow angle as a function of servomotor stroke. The resulting stroke $\sim$ angle relationship is given in Figure 7.13, while the associated specific energy losses are given in Figure 7.14.

Figure 7.13: Calculated effective runner inlet angle over the full operating range based on velocity triangle theory and major loss components

External mechanical losses in bearings, shaft seals and wear rings are estimated by the power required to operate the Reece Francis turbine unit in synchronous condenser mode (see Section 2.3). As shown in test results presented in Chapter 3 the power draw by the unit spinning in air at rated speed was 1.6 MW. This is the assumed power loss due to mechanical friction, $P_f$, and is assumed constant over the operating range.

Finally, in order to correctly represent behaviour at very low flow operation, churning losses are included into the model. Churning losses are a result of the resistance to rotation applied to the turbine runner by low momentum fluid within the runner proper. As identified by Daugherty [22] in an examination of centrifugal pumps, this loss is neither a mechanical loss, as it is clearly a function of flow through the turbine, nor is it classed as a typical hydraulic loss as this would tend toward zero as the flow rate is decreased. On the contrary, the churning loss is greatest at zero through-flow and reduces in magnitude as a more favourable flow is established within the runner as flow rate is increased. For this reason it must be considered separately.
Churning losses may be determined by inspection of test data. Figure 7.15 shows the results of tests conducted on a single stage centrifugal pump with calculated loss components as reported by Daugherty [22]. The contribution of churning loss is determined based on the total of calculated losses and the observed pump output at low flow.

Similarly for the new turbine model, churning losses ($P_{Ch}$) are determined according to the difference between calculated losses and reported turbine performance. This is implemented into the model as a lookup table based on flow rate.
The equation for output power of the Reece Francis turbine unit is therefore:

\[
P = P_{th} - P_f - P_{Ch}
\]

\[
= \rho \eta_v Q E_{th} - P_f - P_{Ch}
\]

\[
= \rho \eta_v Q [gH - (E_{R_v} + E_{D_u} + k_f Q^2)] - P_f - P_{Ch}
\]

(7.40)

The calculated output power implemented into the new one-dimensional hydraulic model is shown in Figure 7.16. The contribution of each of the identified loss components is shown. At low load churning losses dominate up to approximately 40 m$^3$/s, where the runner incidence loss becomes the more significant factor. The residual swirl loss is greatest at part load although remains relatively constant up until near design operation where both the residual swirl and incidence losses go to zero. Above design point friction losses dominate as expected due to the direct proportionality with $Q^2$. The predicted steady-state output power is shown to be in close agreement with model test data.

Figure 7.16: Predicted power output based on the new model formulation with calculated loss components. Power output is shown with loss components sequentially added, whereby the final predicted power output also includes the determined churning loss.
7.5.3 Synchronous machine model and grid representation

A synchronous machine model is included in the new model to more accurately simulate the electrical power response from a rapid transient event by taking into account the rotor dynamics. The Synchronous Machine pu Standard block from the MATLAB SimPower [67] library, which operates in both motor and generator modes, is used in conjunction with a standard excitation system model. The network is represented by a 13.8 kV, 50 Hz infinite bus, approximating an infinitely stiff system [61].

The Synchronous Machine block takes the per unit mechanical power from the turbine model as input. The block output is a vector containing 24 signals which may be demuxed to obtain the required output. For the purpose of the current model the stator voltages, $V_d$ and $V_q$, rotor speed deviation, $d\omega$, and output active power, $P_{eo}$ are chosen as block outputs.

The model represents the synchronous machine by an equivalent circuit in the $d-q$ reference frame which takes into account dynamics associated with the stator field and damper windings [61, 67]. The required block parameters and values are given in Table 7.3

<table>
<thead>
<tr>
<th>Reece synchronous machine parameter</th>
<th>Value</th>
<th>Reece synchronous machine parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal power [MVA]</td>
<td>136</td>
<td>Stator resistance [p.u.]</td>
<td>0.00363</td>
</tr>
<tr>
<td>Line-to-line voltage [kV]</td>
<td>13.8</td>
<td>Reactances</td>
<td></td>
</tr>
<tr>
<td>Frequency [Hz]</td>
<td>50</td>
<td>Direct axis</td>
<td></td>
</tr>
<tr>
<td>Pole pairs [-]</td>
<td>18</td>
<td>synchronous, unsaturated</td>
<td>1.03</td>
</tr>
<tr>
<td>Inertia coefficient [sW/VA]</td>
<td>7.073</td>
<td>transient, saturated</td>
<td>0.33</td>
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<tr>
<td>Friction factor [-]</td>
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<td>sub-transient, saturated</td>
<td>0.25</td>
</tr>
<tr>
<td>Time constants</td>
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<td>Quadrature axis</td>
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</tr>
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<td>2.6</td>
<td>synchronous, saturated</td>
<td>0.64</td>
</tr>
<tr>
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<tr>
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<td>0.06</td>
<td>Stator leakage reactance</td>
<td>0.18</td>
</tr>
</tbody>
</table>

The synchronous machine model is included in the new simulation model, as well as the two conventional models, so that simulated output active power from each model may be directly compared to the active power recorded during field tests. In each case, the speed deviation output, $d\omega$, is fed back into the model in place of a system input according to the block diagrams shown in Figure 7.4 and Figure 7.6, while the new model uses the speed deviation as an input to the Euler equation computation.

The stator voltages, $V_d$ and $V_q$, along with the reference stator terminal voltage are inputs to the excitation system model. The excitation system block implements a DC exciter as described in the IEEE standard [48] and outputs the per unit stator field voltage for input into the synchronous machine block.
7.6 Model verification

The ability of the new turbine model to simulate the transient response of a Francis turbine hydro power station is assessed against full-scale testing performed as part of the return to service test schedule described in Chapter 3.

Figures 7.17 through to 7.22 present a range of fast raise and lower tests from various initial power outputs, while Figure 7.23 presents a slow load to rated output test for validation over the full operating range. The recorded servomotor position during the tests was used as model input while the simulated active electrical output power and penstock pressure response are compared to the recorded test data. Simulation results are also presented for the two conventional models previously discussed.

The simulation results show that the new model performs well over the entire operating range, given in Figure 7.23. In comparison, the two conventional models can be seen to significantly over-predict performance at low load operation, while from approximately 35% output and above the IEEE model significantly under-predicts turbine output power. Although a number of minor improvements are generally made to these conventional models for use in commercial transient analysis studies (discussed previously in Section 2.2), the new model is shown to be a much improved method for predicting full-scale transient response, particularly at low load operation.

The new model does over-predict the active power above rated output as shown clearly in Figure 7.23. Low load operation is extremely well simulated as illustrated in the fast raise from no-load test (Figure 7.17) and the fast lower from 15 MW output (Figure 7.18).

It may be noted that the new model does tend to over predict the initial response following a rapid guide vane movement and can be seen to be related to an associated over-prediction of penstock pressure response, in fact this is a characteristic of each of the simulation models. This is likely due to the servomotor recording being used as a system input variable which is fed directly into the waterway model. In reality, this linear movement of the servomotor arm does not instantaneously translate into a guide vane movement as a degree of backlash is inevitably present in the guide vane linkage mechanism. A simple time lag may be incorporated into the model to correct for this, however this was not included in this case. Additionally, the presence of cavitation and the associated increase in water column compliance, particularly at low load, may slightly alter the machine response.

The new model behaviour has been sufficiently verified by transient field tests and outperforms the standard models. The new turbine model will now form the basis of a model for predicting response from the proposed rapid start-up from tail water depression mode.
Figure 7.17: Fast raise to ~15 MW from no-load test (negative frequency ramp with FCAS boost ON). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.18: Fast lower from ~15 MW test (frequency ramp with FCAS boost ON). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.19: Fast FCAS raise test from ~86 MW (negative frequency ramp with FCAS boost OFF). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.20: Fast FCAS raise test from \( \sim 86 \text{ MW} \) (negative frequency ramp with FCAS boost ON). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.21: Fast FCAS lower test from ~86 MW (frequency ramp with FCAS boost OFF). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.22: Fast FCAS lower test from \(~115\) MW (frequency ramp with FCAS boost ON). Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
Figure 7.23: Step-wise load to 115.6 MW. Simulation results are given and compared to full-scale data for active output power and penstock pressure based on the measured full-scale servomotor stroke.
7.7 Chapter Summary

This chapter has presented the formulation of a new, one-dimensional numerical model of a Francis turbine power plant for use in system dynamic studies. The model builds upon the conventional 1992 IEEE Working Group model [50], however, the representation of the turbine unit, a key component of the model that has remained drastically simplified, is replaced by considering inlet and outlet velocity vectors and calculating machine output based on steady Euler turbine theory. Furthermore, individual loss components such as runner inlet incidence loss, residual swirl flow loss [47] and churning loss [22] are included to ensure model performance across the entire turbine operating range.

The development of the model requires sufficient steady-state test data that gives turbine performance at rated head over an adequate operational range in terms of guide vane position, turbine discharge and output power. From this, and some basic turbine geometry measurements; namely the mid-blade inlet diameter, outlet diameter and inlet blade height; effective inlet flow angles may be calculated over the entire range.

Simulations of fast raise and fast lower tests at the Reece Power Station are performed using the actual recorded guide vane servomotor position as model input. Model output is validated against measured field test data of Reece penstock pressure and active power. Additionally, simulations are performed using two well known conventional models, the 1992 IEEE inelastic waterway model [50] and the model by Kundur [61].

The new model is shown to outperform the two conventional models in each test case, particularly at very low machine output. The two conventional models are still widely used throughout the hydropower industry, however, they are often used as a base model and various modifications, such as nonlinear guide vane functions or friction losses [23], are made as required. Regardless, the new formulation provides a relatively simple and much more physically realistic model that is thoroughly verified by full-scale test data.

Furthermore, the model requires relatively little information to set up, especially in comparison with a method of characteristics analysis, and may in the future be adapted for use in other stations, and for other types of machines such as Kaplan and Pelton turbines.
Chapter 8

Rapid start-up from Tail Water Depresssion Mode

This chapter builds on the new simulation model presented in the previous chapter to examine the process of rapidly loading a Francis turbine from tail water depression (TWD) mode. The new model, which calculates output power according to the conservation of angular momentum and identified sources of loss, is used in parallel with full-scale and model-scale test observations to elucidate the events and mechanisms occurring during this proposed transition.

The ability to accurately and confidently simulate the operation of a hydro-electric plant is an invaluable tool, particularly when considering the plant behaviour under a new mode of operation. Accurate simulation models provide a quick, low cost and low risk method of studying the effects of various system parameters which may be used to inform project feasibility, possible future outcomes or required modifications. Numerous scenarios may be run in a short period of time without risks concerning plant damage or personnel safety. Such scenarios may not be easily tested physically due to either a lack of infrastructure, required works, or operational constraints.

Based on the analysis of initial full-scale TWD test results and key findings from the transient testing of the UTAS micro-hydro unit, a detailed description of the TWD transition process is given. The formulation of an improved turbine model for use in the new one-dimensional hydraulic model of a hydro-electric plant is then presented with simulation results compared to full-scale data. Finally, potential guide vane opening profiles are presented for maximising FCAS contribution.
8.1 Fast Raise FCAS Contribution

The main driver for the proposed operation of a Francis turbine unit in TWD mode, and the need to develop a greater understanding of the flow physics occurring during the rapid transition to generation, is for the provision, and ultimately the optimisation, of FCAS contribution.

As previously noted, FCAS is the ancillary service the Australian Energy Market Operator uses to maintain system frequency within the required normal operating band. Fast raise FCAS, also called R6 FCAS, is a particular classification of service to arrest the immediate frequency deviation following a contingency event and may be realised by either rapid generation or rapid load shedding (see Section 1.2.4 for more).

In terms of providing fast raise FCAS by way of increasing generation output, according to the Market Ancillary Services Specification the amount of fast raise FCAS contribution is defined as:


twice the time average of the raise response between zero and six seconds from the frequency disturbance time, excluding any inertial response'

where the time average is determined as the integral of the increase in power generation from the initial value over the time interval divided by the time duration. The FCAS contribution is therefore described by Equation 8.1 as:

\[
FCAS \ [\text{MW}] = 2 \times \int_{t_0}^{t} \frac{(P_t - P_{e0})}{(t - t_0)} dt
\]  

(8.1)

where the contributing response is in respect to the initial value of electrical power output, \(P_{e0}\), and the time duration, \((t - t_0)\), in regards to R6 FCAS is six seconds from the moment of frequency deviation.

Assuming that a 0.5 s delay exists between the detection of a contingency frequency deviation and the initial movement of the guide vane mechanism, the calculated R6 FCAS contribution prior to governor retuning was 6.93 MW. Following retuning, during which the initial opening rate of 10 mm/s was boosted to 16.5 mm/s, the maximum FCAS contribution was determined to be 17.1 MW. The shortage of local fast raise FCAS within the state is currently 25 MW under high Basslink import condition, however this value is only expected to rise with increases in renewables such as wind turbine generation and the tightening of the Tasmanian Frequency Operating Standards [42].
8.2 Key Rapid Starting Observations

A brief summary of some pivotal observations pertaining to the proposed rapid turbine starting procedure is presented below\(^1\).

Initial full-scale testing of the Reece Francis turbine unit revealed a significant delay in active power response followed by an undesirable negative power flow, and subsequent power oscillations during transition from TWD mode to generation. As the fast raise FCAS contribution is calculated based on the time averaged MW output over the 6 seconds following the frequency trigger, both the delay and the negative power dip significantly inhibit FCAS production.

During governor retuning in 2012, the mechanical dampener limiting the early guide vane opening rate was bypassed. This removed the inherent two-stage opening and resulted in faster possible guide vane opening rates. Even with the increased opening rate, rapid TWD transition tests still displayed a significant delay in positive power generation while the magnitude of the negative power flow was marginally increased. Representative tests from before (\(T_R2\)) and after (\(T_R7\)) governor retuning are presented in Figure 8.1.

It can be seen that although guide vane movement commences at 0.5 s following the detection of a contingency event trigger, there is a delay in power response of approximately 1 s, one-sixth of the FCAS window, for both \(T_R2\) and \(T_R7\) tests. In both cases the power drawn from the grid then increases, reaching a maximum power draw of approximately 5.5 MW in each case, before then transitioning to generation mode. While the magnitude of the power draw is largely unchanged, the duration of negative power flow is reduced with the increased opening rate so that positive power generation is established at approximately 2.1 s for \(T_R7\) compared to 2.7 s for \(T_R2\).

As expected, the decrease in penstock pressure following guide vane opening is increased following the more rapid opening of test case \(T_R7\). In both cases however the first low pressure wave, initiated by the guide vane movement, occurs well before any power response is seen. Furthermore there is no apparent correlation between penstock pressure and the turbine power output during the transition which occurs between 1 s and 6 s in both cases.

In terms of the peak power draw during transition, this occurs at 2.4 s for \(T_R2\) and 1.9 s for \(T_R7\) corresponding to a servomotor stroke of 19.9 mm and 22.9 mm respectively. At this point it may be assumed that the mechanism causing the increased power draw is alleviated.

\(^1\)A thorough description and the full range of performed full-scale TWD tests is given in Chapter 3, while TWD transient tests performed on the UTAS micro-hydro unit are presented in Chapter 6.
Following the establishment of positive output power, active power oscillations with a frequency of approximately 0.9 Hz are present in both test cases.

Additionally, as previously presented (see Figure 3.12, Section 3.4), the level of tail water depression, effectively the volume of air contained within the runner chamber and draft tube cone prior to transition, does not influence the response of the turbine unit.

Figure 8.2 presents two representative cases for tests conducted on the new micro-hydro turbine unit designed and built at the University of Tasmania as part of this study. The two test cases given, $T_M3$ and $T_M4$, are both transition tests from TWD mode to generation corresponding to full-scale guide vane opening rates of 10 mm/s and 16.5 mm/s as previously defined in Section 6.1. The turbine unit performance is given in terms of per unit mechanical power recorded at turbine shaft and the penstock pressure.
Figure 8.2: Shaft power and penstock pressure response following rapid starting of the UTAS micro-hydro unit from TWD mode. Results are given for test cases T\textsubscript{M}3 and T\textsubscript{M}4 with guide vane opening rates (equivalent to full-scale) of 10 mm/s and 16.5 mm/s respectively.

As with the full-scale tests, the micro-hydro tests also revealed a considerable delay from the first movement of the guide vane actuator to a discernable power response. An increased power draw is seen followed by a short, rapid positive increase in power output before a more steady loading is established. Again, as observed on the Reece unit, the penstock pressure immediately responds to the guide vane movement, decreasing more rapidly and to a greater extent for the faster opening T\textsubscript{M}4 test case.

The duration of the increased negative power flow is reduced considerably for the faster opening case with positive power generation occurring at approximately 0.65 s for T\textsubscript{M}4 compared to 1.05 s for T\textsubscript{M}3. Unlike the full-scale response, the peak negative power flow appears to be somewhat less for the more rapid guide vane opening which occurs at an actuator stroke of 0.095 p.u., while again the peak negative power flow of the slower opening
test occurs at a lower actuator stroke of 0.065 p.u.

Rapid starting tests were performed on the micro-hydro turbine unit with tail water depression levels ranging from $0.23D$ to $1.0D$ below the runner exit plane (see Figure 6.4, Section 6.4). The shaft power during transition was found to be completely independent of the initial depression level.

Noticeably, the shaft power recording during micro-hydro unit tests does not display the periodic fluctuations following the initial power dip as observed at full-scale in which active electrical power measurements were recorded.

### 8.2.1 Summary of Findings

The findings of full-scale testing and the tests performed on the micro-hydro unit may be summarised as follows:

1. An increased power draw during the initial stage of transition is observed for both full-scale and micro-hydro scale units;

2. Visual observations on the micro-hydro unit indicate that the tail water does not impinge on the runner during the transition;

3. The tail water depression level prior to transition was found to be irrelevant in relation to the power output for both micro-hydro and full scale units;

4. There exists a significant delay between the initial guide vane movement and any noticeable power response;

5. The penstock pressure response does not exhibit the same delay, indeed the first low pressure wave passes prior to any significant power draw;

6. The magnitude of the negative power flow is not significantly influenced by guide vane opening rate;

7. The duration of negative power flow is reduced under an increased opening rate;

8. The temporal location of the maximum power draw decreases with increasing opening rate, however the servomotor position at the peak draw is actually increased;

9. Power oscillations are observed at full-scale where electrical output power is measured, while these are not observed on the micro-hydro unit where shaft power is recorded.
8.3 Proposed Transition Mechanism

The findings given above are interpreted in the following section. Based on these findings a three-stage transition mechanism is proposed to explain the nature of the power response observed in rapidly switching a Francis turbine unit from TWD to generation mode. The proposed mechanism is discussed and described by considering the law of conservation of angular momentum as applied to two connected control volumes which is then used as the basis for the improved turbine simulation model developed in Section 8.4.

8.3.1 Interpretation of Findings

Findings 1, 2 and 3

Finding 1 indicates that the behaviour observed at the Reece Power Station during trial rapid starting tests is a universal characteristic for mid-range reaction turbines transitioning from the proposed TWD mode to generation. There had been some suggestion of this based on the behaviour reported by Magsaysay et al. [64], although the start-up procedure used at the Dinorwig and Ohkawachi Power Stations were not exactly the same as the procedure under consideration in the present study. The start-up procedure cited [64] involved firstly evacuating the draft tube cone and runner chamber of air prior to the guide vane movement being actuated, although the exact timing of the starting sequence is not entirely clear. It was suggested that the combination of the two actions results in a ‘surge’ of water striking the runner as being the cause of the negative power flow. For such a procedure, the rising tail water would certainly impinge on the runner resulting in an increased power draw from the grid, as the unit is effectively operating as a motor to overcome the inertia of the tail water entering the runner chamber and maintain synchronous speed.

This procedure of ‘assisted’ air evacuation also significantly increases transition time to generation. In the European system this is not as critical as in the Australian system as fast raise services are calculated over a much larger window [69]. As such, the proposed start-up procedure relies on the air volume being transported out through the draft tube by the incoming water flow. Visual observations of the micro-hydro unit during the proposed start-up procedure did not indicate any tail water impingement on the runner as stated in Finding 2 which was further supported by experimental measurement (Finding 3). The fact that an increased negative torque is observed on a small scale micro-hydro unit starting from TWD without assisted air evacuation indicates that another mechanism is present impeding the transition.
Findings 4 and 5

Findings 4 and 5 provide evidence of an accumulation stage in the transition from TWD mode. Without assisted air evacuation, when the guide vanes are initially opened the runner chamber, including the annular space between the maximum runner inlet diameter and the guide vane PCD (pitch circle diameter) is vacated of water.

This annular area (designated as ‘Region 1’ in Figure 8.3) for the Reece Francis turbine unit is 3.45 m$^3$. During the early stages of guide vane opening the water enters the runner chamber as an almost purely tangential flow and continues to have a high tangential component (relative to the absolute velocity) for a significant proportion of the opening. Under this highly tangential inflow a rotating annulus of water persists in the vaneless space until a sufficient volume of water has been introduced, or the guide vane angle is such, as to direct water into the runner proper (‘Region 2’).

Findings 6, 7 and 8

Findings 6, 7 and 8 all indicate a relationship between the inlet flow condition at the moment of, and directly following, the incursion of the rotating annulus, and the negative torque applied to the runner causing the increased power draw. Due to the dependence of flow rate on the guide vane opening angle and the constant annular volume, this is not directly related to servomotor stroke as evidenced by Finding 8, although for the range of opening rates of interest the maximum negative torque can be said to occur at a stroke of between approximately 19 mm and 25 mm, after which a positive applied torque is quickly established.

Finding 9

The experimental findings on the micro-hydro turbine unit confirmed the existence of a negative applied torque on the turbine runner by direct measurement of shaft torque that was not available during full-scale tests. Furthermore, the lack of subsequent power oscillations following the initial power dip indicates that this aspect of the Reece response is electromechanical in nature rather than a purely hydraulic phenomena. As such, the expected machine response from a rapid start-up will be influenced to a degree by the machine inertia and electrical characteristics, as well as the strength of the connected grid at the time of start-up.
8.3.2 Description of the transition from TWD mode

The transition sequence may be considered in three key stages:

i. Initially, as the guide vanes are opened, flow is accumulated in the annular volume that extends from the maximum runner inlet diameter to the guide vane PCD (Region 1, Figure 8.3). The flow, due to the initially high tangential velocities, remains largely on the periphery of the enclosed volume resisting entry into the runner proper.

ii. At a given point following opening, dependent on opening rate, water will begin to enter the runner proper (Region 2), but, due to the low volumetric flow rate, will do so with a reduced inlet tangential velocity in relation to that required for positive power production as described by Euler turbine theory. This results in a negative torque as seen by the runner which is required to essentially bring the incoming flow up to speed. As the guide vanes continue to open, the volumetric flow rate continues to increase and the inhibiting torque applied to the runner by the low momentum fluid reaches a maximum.

iii. Finally, the incoming tangential velocity component of the flow attains a value such that, for the current machine flow rate, positive power output is realised. This rectification of the inlet velocity triangle occurs relatively suddenly and results in a sudden change in electrical phase angle of the generator causing the observed power oscillations. As load is increased at a steady rate the power fluctuations subside and normal operation is established.

Figure 8.3: Control volumes (Region 1 and Region 2) used in the analysis of the transition from TWD mode to generation based on the conservation of angular momentum.
The mechanical aspect of this three-stage process can be adequately described by considering the conservation of angular momentum across two control volumes, Region 1 and Region 2, as defined in Figure 8.3. The conservation of angular momentum applied to a single control volume is given by:

\[
\sum \vec{M} = \frac{d}{dt} \int_{CV} (\vec{R} \times \vec{V})\rho dV + \sum_{out} \vec{R} \times \vec{\dot{m}} \vec{V} - \sum_{in} \vec{R} \times \vec{\dot{m}} \vec{V}
\] (8.2)

which implies that the sum of all external moments acting on a control volume is equal to the time rate of change of angular momentum contained within the control volume plus the net rate of flow of angular momentum out of the control surface boundaries by mass flow [14]. As the Euler turbine equation is just a particular application of the conservation of angular momentum, the mechanical power output is encapsulated in the model as the sum of the external moments acting on Region 2.

The process of rapid transition from TWD mode is illustrated graphically in Figure 8.4 by considering each term of Equation 8.2 separately for Region 1 and Region 2.

Figure 8.4 is comprised of five plots: the first shows a representative linear guide vane opening profile from zero (fully closed) to maximum opening. The four remaining plots show the flow rate of angular momentum into the control volume, the flow rate of angular momentum out of the control volume, the rate of change of angular momentum within the control volume (the accumulation term) and the sum of the moments acting on the control volume. Each angular momentum plot shows two curves: the dotted line represents the specific angular momentum term within Region 1, while the solid line represents the specific angular momentum term within Region 2.

The process is shown as three distinct phases: Accumulation, Transition, and Normal behaviour according to Euler theory. The transition from TWD mode as described above is concerned with the first two phases, the transition phase being broken up into two of the key stages identified.

During the accumulation phase the guide vanes are opened from the closed position and flow enters the turbine but remains within the annular volume between the guide vanes and the runner vaned area (Region 1). Therefore the angular momentum within the annular volume increases from zero, the accumulation term of Region 1 increasing at the same rate as the inlet term of Region 1. The outlet term of Region 1 remains zero as flow is not yet crossing the boundary outlet surface. The resultant moment acting on Region 1 is zero, and in fact remains zero for the entire process. During the accumulation phase all terms for Region 2 remain at zero.
Accumulation phase | Transition | Normal behaviour according to steady Euler theory

Figure 8.4: Flow rate of angular momentum balance during rapid transition from TWD mode to generation. The dotted line represents the region between guide vanes and runner inlet (Region 1), while the solid line represents the region between runner inlet and outlet (Region 2). (note: not to scale).
The start of the transition phase is marked by entry of fluid into the runner proper. This occurs gradually at first owing to the shape of the free surface of the annular body of water in Region 1 and the blade shape of the mid-range specific speed turbine runner. The exact nature of the flow within Region 1 prior to incursion into Region 2 is not at this stage known. It is likely that the flow remains on the periphery of the annular space due to the centripetal force associated with the highly tangential velocity vector, in which case the free surface will take the shape of a parabola, analogous to a partially filled cylindrical container revolving around its major axis [87]. Additionally, the flow may be highly aerated, forming a non-distinct air-water boundary due to the throttling of the flow through the guide vanes. In any case, the flow rate of angular momentum entering Region 2 is not established instantaneously (and is the major contributing factor in the observed power draw and response delay).

While the angular momentum entering the control volume of Region 1 continues to increase with increasing guide vane opening, the Region 1 outlet term gradually increases from zero. As there is no moment transfer within Region 1 this increase in momentum outflow is reflected by a decrease in accumulation.

Neglecting any leakage flow, the momentum entering Region 2 is equal to that leaving Region 1. However, the flow rate of angular momentum entering Region 2 at this stage is extremely low. Some of the incoming flow is accumulated within the runner, while the remaining flow leaves the runner. The flow rate of angular momentum leaving Region 2 does so however with a higher angular momentum than what it entered Region 2 with. Referring to Equation 8.2, the difference between the incoming angular momentum and the sum of the outgoing and stored angular momentum must be provided by an external torque. With the rotor locked to system frequency, this external torque is provided by an increased draw in power from the grid. Referring to Equation 8.2, this external torque is applied to the control volume and so the sum of moments temporarily becomes positive. As the guide vanes are opened further the flow condition, in terms of flow rate and inlet angle, becomes such that the flow rate of angular momentum entering Region 2 becomes greater than the sum of outgoing and stored angular momentum flows thereby imparting a positive torque on the runner which is seen as a decrease in the sum of moments applied to the control volume.

As the positive torque increases the power flow is reversed: the synchronous machine switching from acting as a motor and drawing power from the grid, to acting as a generator and providing power to the system. At the end of the transition phase both storage terms go to zero and normal turbine operation according to traditional Euler theory is established.
8.4 Model Implementation

In the previous chapter the new one-dimensional hydropower plant hydraulic model was shown to accurately predict electrical power output and penstock pressure over the full operating range. Test cases presented included rapid raise tests from no-load operation, rapid lower tests from low load, as well as a number of transient tests performed at part-load and full load operation.

Figures 8.5 and 8.6 present the simulation of rapid start-up events $T_R^2$ and $T_R^7$ which were presented and discussed in Section 8.2. Simulation results are given for the new turbine model as well as the predicted response according to the conventional IEEE model and the model given by Kundur [50, 61]. In each case the the electrical power draw at zero and low guide vane opening is severely over predicted. As the load is increased each of the models performs significantly better. The new turbine model however actually performs the worst over the initial loading phase. This is due to the fact that the new model takes into account and accurately represents the churning losses present at low turbine flow based on experimental data, while for the two conventional models the predicted power at low opening is a function of the no-load flow estimate and the linear guide vane function which, in this case, does give a reasonable estimate of the power output given the crude assumption.

As the runner chamber is vacated of water prior to the rapid start-up from TWD mode the predicted power output for the new turbine model is also given in Figures 8.5 and 8.6 with churning losses set to zero. In both cases the predicted power output is much closer to the response recorded during field tests at the Reece Power Station however as the load is increased the model with excluded churning losses does tend to over estimate the active power generation. Indeed it can be seen that during the transition from TWD mode to generation mode the recorded power output at Reece transitions from the output response predicted excluding churn losses, to that with churn losses, as the flow through the turbine is increased.

While the new turbine model with excluded churning losses provides a reasonable prediction of the response during a rapid start-up sequence, it does not fully capture the behaviour of the unit, particularly the negative power flow which is of particular concern in relation to the provision of FCAS.
Figure 8.5: Simulation of a rapid starting sequence (test case TR2) using the conventional IEEE and Kundur hydraulic models, as well as the new hydraulic turbine model as developed in the previous chapter with, and without, churning losses.

Figure 8.6: Simulation of a rapid starting sequence (test case TR7) using the conventional IEEE and Kundur hydraulic models, as well as the new hydraulic turbine model as developed in the previous chapter with and without churning losses.
The formulation of the new hydropower plant hydraulic model presented in Chapter 7 was based on the direct calculation of torque by the Euler turbine equation, a specific application of the more general law of conservation of angular momentum. The proposed transition mechanism discussed in the previous section was described largely in terms of the conservation of angular momentum applied to two control volumes sharing a common boundary. In order to more accurately represent the physical processes occurring during the rapid start-up a number of additional features are required to be incorporated in to the hydraulic model. The modifications resulting in an improved turbine model, suitable for simulation of rapid start-up events, are given below.

8.4.1 Flow accumulation

During the first phase of transition flow enters Region 1 as soon as the guide vanes are actuated. There is a delay, however, before the flow enters Region 2 as water is accumulated in the annular vaneless space between guide vanes and runner. The flow rate, as calculated by the orifice equation, is integrated over the simulation time to determine the volume of water that has passed into Region 1. Referring to Figure 7.9, the calculated flow rate is still passed through to the waterway model, as this still affects the penstock dynamics, however the flow rate input into the Euler equation calculation block is held to a value of zero until the accumulated volume has reached a predetermined proportion of the annular volume.

Figure 8.7: Simulated flow rate at inlet to Region 1 and Region 2 for rapid starting sequence $T_{R7}$ with accumulation coefficient $k_a = 0.6$. A volumetric efficiency, $\eta_v = 0.99$, accounts for flow leakage prior to Region 2.
To account for the nature of the accumulated water within Region 1 an accumulation coefficient, $k_a$, is introduced. The accumulation coefficient was varied according to opening rate as the flow rate entering Region 2, as implemented in the improved turbine model, was assumed to instantaneously increase to that entering Region 1 when the required accumulated volume was reached. Field and laboratory tests however indicate that the process is not instantaneous, where interaction between the water and runner actually occurs before the full volume has entered Region 1. For test cases $T_{R1}$, $T_{R2}$ and $T_{R7}$ accumulation coefficients of 0.6, 0.8 and 1.0 were used, respectively, to reflect the different opening rates tested. Simulated flow rate during rapid start-up test $T_{R7}$ is given in Figure 8.7. As the runner chamber is initially filled with air at near atmospheric pressure, and, as evidenced by laboratory tests, a large cavity remains for a considerable period during transition, a cavitation compliance term is not included within the model. The presence of a large cavity would undoubtedly affect the pressure transients to some degree which has not been included in the model. It is shown later in Section 8.5 that initial pressure transient magnitudes have marginal affect on power output at six seconds post trigger.

### 8.4.2 Reduced tangential velocity

The reduced tangential velocity assumption accounts for the behaviour of the flow entering Region 2 following the accumulation phase. As the flow begins to enter Region 2 it will do so with a considerably lower flow rate of angular momentum as previously described due to the low mass flow rate, confused inlet flow angle and the potentially highly aerated air-water boundary. Therefore, working in conjunction with the flow accumulation, the calculated inlet flow angle, which is a function of servomotor stroke as given in Figure 7.13, is modified to reflect the initially low rate of flow of angular momentum. A linear assumption is made whereby the apparent inlet angle varies from a purely radial flow, $a'_R = 90^\circ$, at the end of the accumulation phase to the calculated angle at a servomotor opening determined by inspection of test data. This has the effect of gradually increasing the rate of flow of angular momentum into Region 2, bridging the accumulation and transition phases. The simulated flow rate of angular momentum entering Region 1 and Region 2 during the rapid start-up $T_{R7}$ test case is given in Figure 8.8.

While the linear approximation succeeds in gradually increasing the rate of flow of angular momentum (since $C_{u_1} = C_{m_1} / \tan(a'_R)$) it does result in a sudden switch to the calculated angle which is reflected in the rate of angular momentum flow entering Region 2 as seen in Figure 8.8. This sudden rectification of the effective inlet flow angle would, in reality, be more of a transition as illustrated in Figure 8.4. This results in over predictions of the power response at this point as will be shown in the Results Section following.
It can also be seen that the rate of angular momentum flow leaving Region 2 is instantaneously increased at the end of the accumulation phase. This is due to the absence of an accumulation term associated with Region 2, the runner. As such both terms are effectively lumped together into the \( \bar{r} \times \dot{m} \bar{V} \) outlet term. The sudden increase in outlet flow rate of angular momentum also results in a slight over prediction of the negative power flow.

### 8.4.3 Machine Inertia and Grid Strength

The transient testing results of the micro-hydro turbine unit in which shaft torque measurements were taken indicated that the oscillations in power response were electromechanical in nature rather than purely mechanical as suggested by Finding 9. Due to the inertia of the rotor, the generator rotor angle is not able to instantaneously adjust to a new equilibrium position when subject to a sudden increase in applied torque. This is illustrated in an example given by Kundur [61] in examining a sudden step increase in mechanical power of a generic generating unit connected to a large network represented by an infinite bus, Figure 8.9.

The case shown neglects all resistances and sources of damping and so the power and rotor angle is seen to continually oscillate between the initial position and the position of maximum overshoot. In reality however there are a number of factors that influence the
machine response to a transient event. These included, but are not limited to: how heavy
the generator is loaded, generator output at the time of the event, transmission system
properties, generator reactance and internal voltage magnitude, generator inertia and the
size of the network the generator is connected to [61].

Figure 8.9: Power angle and rotor angle time response following a step change in applied
mechanical power from \( P_{m0} \) to \( P_{m1} \) [61].

The influence of machine inertia is examined in Figure 8.10. The inertia constant, \( H \), is
defined as the kinetic energy at rated speed divided by the unit VA rating as given by
Equation 8.3:

\[
H = \frac{1}{2} J \omega_r^2 \quad \text{(8.3)}
\]

where \( J \) [kgm^2] is the combined moment of inertia of the generator-turbine unit, \( \omega_r \) [rad/s] the rated angular velocity of the rotor, and \( VA_r \) is the rating of the generator. The inertia constant is used to normalise the swing equation implemented within the Synchronous Ma-
chine Simulink block to express the resulting accelerative torque applied, \( T_a \) [Nm], to the
unit:

\[
J \frac{d\omega_0}{dt} = T_a = T_m - T_e \quad \text{(8.4)}
\]

where \( T_m \) [Nm] is the mechanical torque applied to the rotor and \( T_e \) [Nm] is the electro-
magnetic torque of the generator.
Figure 8.10 presents results of test case TR7 as simulated by the improved turbine model incorporating the flow accumulation and reduced tangential inlet velocity mechanisms as described above, with various values of inertia constant, H.

![Figure 8.10: Simulated output power response during rapid transition from TWD mode for various turbine-generator inertia constants with the connected network represented as an infinite bus.](image)

The calculated combined inertia constant for the Reece Francis turbine unit was $H = 3.86$, with an inertia ratio of generator inertia constant to the turbine inertia constant of approximately 10. This was however based on inertia values as given by the original manufacturer which may not be completely accurate, particularly if work has been done on either, or both, of the turbine runner and generator rotor. Typically the inertia ratio is between 10 and 40 however developments in design and materials is resulting in lighter generator rotors [12]. As the Reece turbine unit is at the lower limit of the range of expected inertia ratio, inertia constants of $H = 3.0, 5.0, 7.0$ and $9.0$ were chosen for examination.

A higher inertia constant is seen to reduce the frequency of power oscillations due a reduction in the rate of change of rotor angle experienced following a disturbance. In regards to R6 FCAS production, it is uncertain however as to whether this is beneficial as the response is slower to achieve positive power output and establish a steady acceptance of load in comparison to a similar low inertia machine. The full-scale power output is also given in Figure 8.10. Simulated power response oscillations appear to dampen out much sooner in comparison to actual Reece output due in large part to the representation of the connected network as an infinite bus.
A more representative network model is given in Figure 8.11 in which the synchronous machine is connected to a 220 kV, 1,600 MVA network through a Delta-Wye 210 MVA transformer. A local resistive load of 5% of the rated generator capacity is also provided.

Figure 8.11: Alternative network representation based on a model by L. A. Dessaint and R. Champagne (Ecole de Technologie Superieure, Montreal) [67].

The size of the Tasmanian network typically varies between 1.0 - 2.0 GW depending on the time of day and year, with peak generation reaching 2.2 GW for the year 2013 [44]. The influence of grid strength on the simulated output response is presented in Figure 8.12 for representative network sizes of 0.8, 1.2, 1.6 and 2.5 GW.

The simulated response is shown to capture the general nature of the power response observed during Reece FCAS testing with more sustained oscillations of comparable frequency. The influence of the grid strength is such that response is increasingly dampened as the size of the network is increased. In the limiting case as the grid strength is increased the response would approach that of the infinite bus assumption.

The combined effects of machine inertia and grid strength are presented in Figure 8.13. A representative grid strength of 1.6 GW was chosen while inertia constants ranging from $H = 3.0$ to 9.0 are given as for Figure 8.10.

All simulation results presented in the following section were performed using the network model provided by Figure 8.11 representing a Tasmanian grid of 1.6 GW. The inertia constant used was that calculated based on the available manufactures data.
Figure 8.12: Influence of network strength on simulated output power response during rapid transition from TWD mode. Simulations performed using the network representation of Dessaint and Champagne [67].

Figure 8.13: Simulated output power response during rapid start-up from TWD mode for various turbine-generator inertia constants connected to a representative 1.6 GW network based on that of Dessaint and Champagne [67].
8.5 Model Verification

Simulation results for the improved turbine model, implemented within the hydropower plant hydraulic model described in Chapter 7, are presented for three separate start-up scenarios from TWD mode. The three test cases were chosen to cover the full range of previously tested start-up procedures at the Reece Power Station during the 2008 feasibility study and the 2012 governor upgrade return to service tests.

Start-up scenarios $T_R1$ and $T_R2$ were performed during the initial rapid start-up feasibility study. Figure 8.14 presents simulation results of test case $T_R1$ which represents a slow transition from TWD depression mode with an average opening rate of 7.3 mm/s while Figure 8.15 presents results of test $T_R2$ in which a moderate two-stage opening is performed. The initial rate of 10 mm/s increases to approximately 16.5 mm/s at 75 mm servomotor stroke due to the inherent mechanical cushioning. Simulation results of test case $T_R7$, performed during the return to service test series following governor retuning in 2012, are presented in Figure 8.16. At the time the in-built mechanical cushioning was bypassed allowing higher initial opening rates. Test $T_R7$ shows a single stage opening of 16.5 mm/s with a high initial tail water free surface level.

The predicted power output in each case is shown to capture the general response characteristic of the test, and in the case of tests $T_R2$ and $T_R7$ the output of the improved turbine model is much improved from the initial simulations presented in Figures 8.5 and 8.6. It should also be noted that the Reece field test data for $T_R1$ and $T_R2$ display momentary 0 MW output power ‘dead spots’ that are not indicative of the actual response at that time.

Generally, the simulated power response tends to slightly over predict the magnitude of both the negative power flow and the subsequent positive power flow due to the model assumptions made concerning the flow accumulation and reduced tangential velocity at Region 2 inlet. Furthermore, the absence of an accumulation term within Region 2, and the associated requirement for the inclusion of some form of transitional churning loss, causes a significant over prediction in power output following the initial transient phase where load is being steadily increased. As such, the predicted output power at six seconds post trigger signal is consistently greater than the actual power output based on measured test data. As the churning loss is implemented as a function of flow rate, the output power prediction is actually affected the most for intermediate opening rates. For slower opening rates the excluded churning loss assumption holds due to the low proportion of water within the runner chamber at six seconds, while for higher opening rates the flow rate is such that normal operation is established within the FCAS window limiting the influence of the churning losses.
Figure 8.14: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given for the opening profile based on test case TR1 - a single stage opening profile with a rate of 7.3 mm/s.
Figure 8.15: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given for the opening profile based on test case TR2 - a two-stage opening profile with an initial rate of 10 mm/s followed by a second stage opening rate of 16.5 mm/s initiated at 75 mm servomotor stroke.
Figure 8.16: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given for the opening profile based on test case TR7 - a single stage opening profile with a rate of 16.5 mm/s.
Figure 8.17: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given for the opening profile based on test case TR7 - a single stage opening profile with a rate of 16.5 mm/s with penstock pressure from Reece field test data used as model input.
The ability of the proposed model to accurately predict penstock pressure following rapid guide vane movement, verified in the previous chapter, provided an interesting result in regards to TWD start-up. For test cases TR1 and TR2 the magnitudes and frequency of the penstock pressure transients were well predicted, although it is apparent that the actual turbine guide vane movement does exhibit a degree of backlash resulting in slightly delayed and gentler pressure transients not captured in the model.

The interesting result is the under-prediction of the initial low pressure wave seen for test case TR7. In this case where the guide vanes are opened at a single stage rate of 16.5 mm/s the penstock pressure reduces by 23 m in approximately 0.5 s to 67 m S.L. The simulated penstock pressure on the other hand predicts a low of only 77 m. This discrepancy is due to the calculation of the non-linear guide vane function, $G$, which is based on steady state data and the head difference between upstream and downstream measurement sections. In the case of start-up procedures from TWD mode where flow is essentially throttled through the guide vane assembly to the runner chamber filled with air at near atmospheric pressure, the discharge coefficient in this case is expected to be significantly higher than the case in which the discharge coefficient is calculated between spiral inlet and draft tube outlet with the additional associated head losses. As previously mentioned, there is also an additional effect, similar to a ‘cavitation compliance’, associated with the large volume of air within the runner chamber influencing the pressure transients within the turbine and penstock.

In order to verify that the improved turbine model was correctly representing the proposed transition mechanism, and that the inability to predict this low penstock pressure was not affecting the predicted power response, the hydraulic model was modified to include the measured penstock pressure as a model input, essentially replacing the elastic waterway equation and feeding directly into the orifice equation. The flow rate calculated by the measured penstock pressure was then used as input into the improved turbine model as normal to determine the accumulated volume and runner torque according to the Euler equation block. The results of this simulation are shown in Figure 8.17 which also includes the predicted power response from the full improved turbine model previously presented for reference.

It is shown that the low pressure measured during field tests has very little influence on the simulated power output during the rapid start-up as it occurs during the accumulation phase of transition. It does however indicate that the rate of flow entering Region 1 during this time would be under predicted to a degree.

Furthermore it does highlight the inability of the improved turbine model to predict the penstock pressure during the first moments when the turbine is started in this way, due to the changed discharge characteristic. Following the initial low pressure transient within
the penstock the model does then adequately simulate penstock pressures as shown in Figures 8.14 to 8.17 but also in Figures 7.17 to 7.23 of the previous chapter.

The fact that the initial pressure transient was not under predicted in test cases T_R1 and T_R2 is likely due to the relatively small stroke of the servomotor during the accumulation phase in both cases where the opening rate is low. In these cases the sudden decrease in pressure is more a function of the measured servomotor stroke being used as model input which neglects any backlash present in the system.

### 8.5.1 Predicted FCAS contribution

The new simulation model also enables the predicted FCAS contribution to be calculated according to Equation 8.1. The predicted contribution from the time of frequency disturbance detection is presented in Figure 8.18 and compared to the calculated FCAS contribution based on full-scale data. The amount of R6 FCAS contribution able to be provided is equal to the value at six seconds following the disturbance and is summarised in Table 8.1.

![Figure 8.18: Expected FCAS contribution based on the improved turbine model simulation data as compared to full-scale FCAS contribution, calculated according to Equation 8.1.](image)

Taking into consideration the moderate over prediction of power output mentioned previously, the simulated FCAS contribution is well predicted in each case, slightly greater than the contribution calculated based on full-scale data. As with the power output, this discrepancy is due to the lack of an accumulation term in the simulation model associated
Table 8.1: Simulated and actual output power and calculated FCAS contribution during rapid start-up tests from TWD mode. FCAS contribution is calculated based on the predicted MW output during the first six seconds following a frequency trigger as described by Equation 8.1.

<table>
<thead>
<tr>
<th>Test case</th>
<th>( T_{R1} )</th>
<th>( T_{R2} )</th>
<th>( T_{R7} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Active power output at 6 seconds</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Actual [MW]</td>
<td>4.91</td>
<td>9.32</td>
<td>24.0</td>
</tr>
<tr>
<td>Simulation [MW]</td>
<td>6.27</td>
<td>12.8</td>
<td>29.9</td>
</tr>
<tr>
<td>FCAS contribution at 6 seconds</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Actual [MW]</td>
<td>2.82</td>
<td>6.93</td>
<td>17.1</td>
</tr>
<tr>
<td>Simulation [MW]</td>
<td>3.91</td>
<td>9.15</td>
<td>21.3</td>
</tr>
</tbody>
</table>

with Region 2 and will be the most severe for intermediate guide vane opening rates. For test case \( T_{R7} \) the predicted FCAS contribution was 21.3 MW as compared with the actual calculated value of 17.1 MW. It is expected, however, that the simulation prediction would become more accurate at higher guide vane opening rates.

### 8.6 Possible Improvement Methods

Results presented in the previous section indicate that the proposed transition mechanism, implemented in the improved turbine simulation model, appropriately describes the unit behaviour during rapid starting from TWD.

Power output is somewhat over-predicted due to the absence of an accumulation term for Region 2 however it is suggested that this would become less significant as the opening rate is increased, thereby decreasing the period of time expected to be influenced by these churning losses.

Penstock pressure was found to be adequately predicted except in the very early stages of transition during the accumulation phase. It is during this time that the peak low penstock pressure is experienced owing to the changed machine discharge characteristic due to the evacuated runner chamber. Under a 16.5 mm/s opening rate the penstock pressure reaches a minimum of 67 m S.L. Although the steel lined section of penstock is rated to withstand a full vacuum internal pressure of -10 m [43], it is recommended that the initial opening rate is not increased beyond the tested 16.5 mm/s as the likely magnitude of the associated pressure response is not known.
In order to increase the potential for FCAS contribution a number of methods were considered. Possible improvement methods include:

- Increasing or decreasing machine inertia
- Assisted air expulsion
- Pressure transient control measures (surge tower or chamber)
- Auxiliary, possibly directed, flow source(s)
- Increased guide vane opening rate
- Continual air admission, and
- Optimised guide vane opening profiles

**Increased machine inertia**

Through modification of either turbine runner, generator rotor, or both, or the addition of a flywheel, it may be possible to significantly alter the machine inertia. Alternatively, the combined inertia constant may be used as a selection tool in identifying other potential hydropower plants suitable for FCAS production via the proposed start-up procedure.

The effect of machine inertia was shown previously in Section 8.4.3. The predicted FCAS contribution however, between similar units with inertia constants of $H = 3.0$ and $H = 9.0$, was calculated to be 21.2 MW and 21.1 MW respectively, practically insignificant.

**Assisted air expulsion**

One of the significant factors in the ability to provide rapid generation is the requirement to fill the evacuated runner chamber and draft tube cone before normal operation can be established. This leads to a delay in machine response of approximately 1 second in the case of $T_{RZ}$, longer for slower opening rates, which is a significant proportion of the 6 second fast FCAS window. The ability to eliminate this delay would dramatically increase potential FCAS provision. Reports in the literature on the transition of the Dinorwig and Ohkawachi power plants [64] from synchronous condenser mode indicate that the air inside the turbine is expelled before the guide vanes are initiated and flow is allowed to enter. This procedure however would further delay power generation, dependent on the speed at which the air can be exhausted. Furthermore the air expulsion would lead to the tail water rising and impinging on the runner, providing a considerably large braking torque prior the guide vane movement as evidenced by the magnitude of power reversals reported.
Pressure transient control

As the Reece Power Station hydraulic circuit does not include a surge tower, and the ability to retrofit a surge chamber is limited due to spatial constraints, the proposition of controlling pressure transients is not practical in this case. A surge tower (or chamber) could easily be included within the new simulation model as described by the theory outlined in references [54], [62] and [115]. A simulation was run however for the absolute limiting case in which the penstock pressure was set to be constant for the duration of the TR7 start-up event in order to gauge the potential for improvement. In this case the predicted FCAS contribution was calculated to be 22.9 MW, an improvement of 7.5%. While this is a comparatively minor improvement, the presence of a surge tower/chamber at a FCAS providing power plant could allow for significantly increased initial guide vane opening rates, potentially increasing FCAS appreciably.

Auxiliary flow source(s)

To maximise FCAS provision the unit must be able to be safely loaded as rapidly as possible while minimising the response delay and subsequent negative power flow during the initial stage of transition. As the response delay is due to the requirement to fill the annular volume between guide vanes and runner inlet, an auxiliary water source could potentially be used to assist in the filling process. The volume itself is less than 3.5 m$^3$ so the size of the storage vessel would be minimal. The vessel would be required to have the same pressure rating as that of the penstock, however the operating pressure need not be as high, as the air pressure inside the evacuated runner chamber is approximately equal to atmospheric (depending on downstream reservoir height). Potentially, multiple nozzles could be positioned in such a way as to increase the flow rate of angular momentum entering Region 2. A measure of this nature however has a number of practical issues in terms of implementation and would require considerable works to be undertaken. The potential for increased FCAS is unknown and would require further studies to determine whether feasible or not.

It may also be worth investigating whether it is possible to admit a small amount of water while running in TWD mode to somewhat pre-fill Region 1, and whether this would remain upstream of the runner due to the effective pumping action of the turbine runner being run as a motor. If the leakage flow was not too great this may be a simple measure to decrease the response delay, although as a result the earlier impingement of low angular momentum water at runner inlet may exacerbate the magnitude and duration of the reverse power flow.
Continual air admission

The results of the improved numerical model that neglects churning losses for TWD simulations indicates that by eliminating the churning losses that are present at the low part-load range of turbine operation, power generation, at least within the early stages, may be increased. While this has been indicated by simulation results, the practically of administering a large enough flow rate of air may be problematic.

8.6.1 Guide vane opening profiles

In regards to the Reece Power Station, the method most likely to produce substantial R6 FCAS gains is by increasing the guide vane opening rate or by modifying the opening profile, or both.

As mentioned above, the initial low pressure transient within the penstock is greater in magnitude than predicted by the model due to the altered discharge characteristic. However, penstock pressure is well predicted following the Region 1 accumulation phase. As such, it is possible to confidently test a range of two-stage opening profiles, of various second stage opening rates, following an initial 16.5 mm/s opening for the first 2 seconds.

Figure 8.19 presents a range of simulation tests examining the effect of increasing the second stage of opening from 16.5 mm/s (effectively single stage opening) to 66 mm/s (4× the maximum opening rate physically tested). The guide vane opening profile in each case is such that the profile remains linearly increasing at least until the 6 second point following the frequency trigger signal. At 6 seconds the two-stage profile of 16.5/33 mm/s is leveled off at a servomotor stroke of 156 mm, profiles with second stage rates less than 33 mm/s are also brought to 156 mm before opening is ceased. For the 66 mm/s second stage opening the servomotor stroke reaches 289 mm by 6 s post frequency trigger before movement is ceased.

The power response is found to predictably increase with increasing second stage opening rate with no adverse power oscillations limiting FCAS production. The response in each case is identical prior to the 2 s point meaning that the negative power flow magnitude and duration is unaffected, however power output is shown to increase from 26.5 MW to 70.1 MW for profiles of 16.5 mm/s (single stage) to 16.5/66 mm/s (two-stage). This results in a simulated FCAS contribution increase from 19.8 MW up to 47.9 MW, more than double the actual FCAS contribution of test case TR7 of 21.3 MW.
Figure 8.19: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given examining various opening rates following an initial 2 second, 16.5 mm/s opening as per test case TR7.
It must be restated however that the tendency of the simulation model to consistently over-predict power output will in turn over-predict FCAS contribution. For test case TR7 the simulated power output was 25% greater than measured, although this is expected to be less significant at higher opening rates. If 25% is taken as a conservative error prediction the FCAS contribution expected from the maximum two-stage opening is reduced to 35.9 MW, still a considerable increase from TR7 FCAS contribution. The results presented in Figure 8.19 are summarised in Table 8.2.

All penstock pressures were well above the minimum allowable pressure, the minimum pressure observed being 53.8 m for the two-stage opening rate of 16.5/66 mm/s.

![Figure 8.20: Calculated FCAS contribution as a function of the second stage guide vane opening rate based on simulation data.](image)

The effect of increasing the second stage guide vane opening rate is illustrated by Figure 8.20. The simulation results indicate that significant gains can be made by increasing the second stage opening rate of a two-stage opening profile with the initial opening rate equal to that of test case TR7. However, the relative increase in FCAS contribution does decrease with increasing rate.

Additional FCAS contribution may be realised by modifying the guide vane opening profile. Sudden stopping of the guide vane movement just before the end of the 6 s FCAS window results in a rise in penstock pressure (Figure 8.19). With normal turbine behaviour more or less established by this point, the water column inertia and the sudden increase in pressure results in a spike in output power. This behaviour may be exploited to further increase FCAS contribution as shown in Figure 8.21 and summarised below in Table 8.3.
Table 8.2: Simulated active power output and calculated FCAS contribution for various two-stage guide vane opening profiles. The percentage FCAS improvement is given in relation to the simulated FCAS contribution of test case Tr7.

<table>
<thead>
<tr>
<th>Opening Profile</th>
<th>Simulated active power [MW]</th>
<th>Simulated FCAS contribution [MW]</th>
<th>Simulated FCAS Improvement [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulated test case Tr7</td>
<td>29.9</td>
<td>21.3</td>
<td>-</td>
</tr>
<tr>
<td>Single stage ramp, 16.5 mm/s</td>
<td>26.5</td>
<td>19.8</td>
<td>-7.0</td>
</tr>
<tr>
<td>Two-stage ramp, 16.5 to 20 mm/s</td>
<td>31.1</td>
<td>22.7</td>
<td>+6.6</td>
</tr>
<tr>
<td>Two-stage ramp, 16.5 to 25 mm/s</td>
<td>37.9</td>
<td>26.7</td>
<td>+25.4</td>
</tr>
<tr>
<td>Two-stage ramp, 16.5 to 33 mm/s</td>
<td>47.8</td>
<td>33.0</td>
<td>+55.0</td>
</tr>
<tr>
<td>Two-stage ramp, 16.5 to 66 mm/s</td>
<td>70.1</td>
<td>47.9</td>
<td>+124.9</td>
</tr>
</tbody>
</table>

By stopping guide vane opening however the maximum power output level reached at 6 s is decreased. If opening is stopped too soon any gains in FCAS made by the sudden spike in output power are offset. For the 16.5/66 mm/s two-stage opening, while the maximum power output is less, Table 8.3 shows that maximum FCAS gains, of the order of 4.4 MW, are obtained for a reduced second stage ramp duration of 2.6 s. For ramp durations shorter than this however FCAS gains begin to decrease. Furthermore, relative gains are shown to be more substantial for higher second stage opening rates with only marginal increases achieved for the 33 mm/s second stage opening.

Table 8.3: Simulated active power output and calculated FCAS contribution for two-stage guide vane opening profiles with various second stage ramp durations. Percentage FCAS improvement is given in relation to the simulated FCAS contribution of test case Tr7.

<table>
<thead>
<tr>
<th>Opening Profile</th>
<th>Simulated active power [MW]</th>
<th>Simulated FCAS contribution [MW]</th>
<th>Simulated FCAS Improvement [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulated test case Tr7</td>
<td>29.9</td>
<td>21.3</td>
<td>-</td>
</tr>
<tr>
<td>Two-stage, 4.0 s ramp, 16.5 to 33 mm/s</td>
<td>47.8</td>
<td>33.0</td>
<td>+55.0</td>
</tr>
<tr>
<td>Two-stage, 3.2 s ramp, 16.5 to 33 mm/s</td>
<td>50.7</td>
<td>33.3</td>
<td>+56.3</td>
</tr>
<tr>
<td>Two-stage, 4.0 s ramp, 16.5 to 66 mm/s</td>
<td>70.1</td>
<td>47.9</td>
<td>+124.9</td>
</tr>
<tr>
<td>Two-stage, 3.2 s ramp, 16.5 to 66 mm/s</td>
<td>88.2</td>
<td>50.3</td>
<td>+136.2</td>
</tr>
<tr>
<td>Two-stage, 2.6 s ramp, 16.5 to 66 mm/s</td>
<td>81.3</td>
<td>52.3</td>
<td>+145.5</td>
</tr>
<tr>
<td>Two-stage, 2.0 s ramp, 16.5 to 66 mm/s</td>
<td>63.2</td>
<td>50.6</td>
<td>+137.6</td>
</tr>
</tbody>
</table>

Similar tests were performed whereby the guide vane movement began closing after the nominated ramp duration rather than just maintaining position. Small increases in FCAS production were seen depending on the rate of guide vane closure, however penstock pressures became increasingly high with greater closure rates before any substantial FCAS benefit was realised.
Figure 8.21: Simulation of the rapid start-up of the Reece Power Station Francis turbine unit from TWD mode. Active power output and penstock pressure response are given examining various ramp durations for two-stage opening profiles.
8.7 Chapter Summary

This chapter has brought together results from full-scale field test data and experimental laboratory testing of a micro-hydro turbine unit to propose a mechanism of transition that occurs when a mid-range specific speed Francis turbine is rapidly brought online from a tail water depression mode.

The proposed mechanism is supported by evidence which suggests a three-stage transition process consisting of an accumulation phase, a transition phase, and the establishment of normal turbine behaviour as described by Euler turbine theory.

The transition process is shown to be adequately described by considering the conservation of angular momentum applied across two control volumes sharing a common boundary. The control volumes represent the annular space between guide vane ring diameter and the runner blade inlet edge (Region 1), and the space between runner inlet and outlet (Region 2). The rate of flow of angular momentum between Region 1 inlet and Region 2 outlet is described, and presented graphically, illustrating the accumulation, transition and normal behaviour phases in each region.

The three phase transition mechanism is then incorporated into the new turbine numerical model presented in Chapter 7, utilising the law of conservation of angular momentum and the compatibility of the structure of the new model.

The following modifications are made for simulation of TWD rapid start-up procedures: churning losses are turned off, an accumulation term within Region 1 is included, a reduced Region 2 inlet tangential velocity is assumed based on a linear relationship with guide vane position, and a more representative network model is used.

Simulation tests are run for test cases T_R1, T_R2 and T_R7. The improved turbine model is shown to fully capture the entire characteristic of the observed active power response of the Reece Francis turbine unit adding further validation to the proposed mechanism. The power output during the later stages of the transition tests were consistently over predicted by the simulation model due to the lack of an accumulation term within Region 2 and the assumption of no churning losses. Additionally, for the more rapid initial guide vane opening rate of test case T_R7 the resulting low penstock pressure was under predicted by 15%. It is thought that this is due to the fact that the nonlinear guide vane function was calculated based on single phase steady-state operating conditions. With flow discharging into an evacuated runner chamber the discharge characteristics will be altered.
From the simulated active power output within the first 6 s from a frequency trigger the simulated FCAS contribution was calculated based on the definition provided by the Australian Energy Market Operator. Taking into consideration the slight over prediction of power, simulated FCAS contribution is well predicted in each case.

Finally, a number of potential improvement methods are proposed in order to increase the rapid start-up response. Of those put forward, it is suggested that the guide vane opening profile remains the most likely to substantially increase FCAS contribution while also being the most practical to implement.

Simulation tests were run for various two-stage opening profiles, the initial stage kept to the current maximum initial opening rate so that the penstock pressure response was known. FCAS contributions are shown to steadily increase with increasing second stage opening. The highest second stage opening was 66 mm/s, \(4 \times\) the initial stage, and the simulated FCAS was 47.9 MW, just over twice that determined by simulation for test case T_{R7}.

By modifying the opening profile such that servomotor movement stops before the 6 s FCAS window the resulting pressure increase brought about by the guide vane movement may be exploited. Various ramp times were investigated ranging from 2.0 s to 4.0 s at a 33 mm/s and 66 mm/s second stage rate. Optimal FCAS contribution was found for a 2.6 s ramp at 66 mm/s second stage opening rate. The amount of FCAS contribution predicted by simulation in this case was 50.6 MW.
Chapter 9

Conclusions and Recommendations

In the coming years, with increases in capacity of non base load renewables, the role of hydropower will be increasingly tied to managing and maintaining system stability. Furthermore, small scale hydropower, in rural electrification and waste energy recovery schemes will be of great importance.

The ability to rapidly transition a Francis turbine unit from tail water depression (TWD) mode to generation has been investigated experimentally, at both full-scale and on a micro-hydro turbine unit, and numerically. Full-scale tests were performed by Hydro Tasmania at the Reece Power Station on the west coast of Tasmania, while laboratory tests were performed at the University of Tasmania on a test facility designed and developed as part of the research program presented in this dissertation.

The initial feasibility study demonstrated that a modified synchronous condenser mode, whereby air is forced into the draft tube cone with the governor remaining active, could be used to facilitate a rapid transition in order to provide fast generating capability. However, test results indicated the presence of a significant response delay, followed by an undesirable increased power draw during the early stages of transition and subsequent active power oscillations. The result was a considerably reduced contribution within the 6 s window specified by the Australian market for the provision of the fast raise frequency control ancillary service (R6 FCAS).
As such, the specific objective of the current work was to investigate the cause and mechanisms behind the observed unit response in an effort to develop optimised operational procedures for increasing rapid generating potential, and thus the provision of frequency support services, for the future. Additionally, in developing a laboratory test rig for studying the proposed operation, an improved micro-hydro turbine (UTAS-MH) was designed and tested.

A new micro-hydro turbine design was presented, addressing the main drawbacks of current small scale hydropower installations that had heretofore limited their suitability and application in many instances. In the context of micro-hydro power schemes, the capital cost of conventional Francis turbines are often prohibitive, while implementation of a pump-as-turbine (PAT) is problematic due to unpredictable best efficiency operating points and low off-design efficiencies.

Therefore, a new class of low cost micro-hydro turbine was developed to provide an alternative generating solution for application in remote area power supply and industrial energy recovery systems. Working initially from established PAT design principles, a suitable pump impeller was selected based on available head and flow capacities. A customised housing was built to allow for the incorporation of inlet flow control by way of a simple guide vane assembly. The resulting turbine was found to have a maximum efficiency of 79%, marginally higher than that of the parent pump. This result is in agreement with the literature which generally reports PAT efficiencies to be within ±2% of best efficiency pump operation [25, 109].

The advantage of the new micro-hydro unit, however, is in the much wider operating range achievable due to the included flow control. Using the weighted average efficiency (WAE), defined by Singh [94] as a means to more accurately assess the performance of a micro-hydro installation, the new unit was shown to achieve a WAE of 67%. In comparison, Singh [94] presents the results of performance testing on an optimised PAT that had undergone numerous modifications such as impeller blade tip rounding and tapered ring inserts resulting in a relatively high peak efficiency of 84%. While the best efficiency of the PAT presented by Singh [94] was considerably higher than the unmodified UTAS-MH unit, due to the poor off-design efficiency, the optimised PAT had a WAE of only 66%. This result is extremely important in the sphere of micro-hydro installations where ideal operating conditions are rarely realised and the prediction of the expected PAT best efficiency point itself is highly uncertain.
The resulting 6.2 kW micro-hydro unit, while slightly more complex than a traditional PAT system, demonstrates near peak efficiency over a wide range of head and flow conditions thereby addressing the main drawback of pumps operating in turbine mode.

Rapid start-up tests were performed on the UTAS-MH turbine unit from a tail water depression mode. The increased power draw present at full-scale was also observed in laboratory tests, however to a lesser degree. The full-scale peak power draw, relative to the small amount of power required while operating in TWD mode was 5% of rated, whereas the micro-hydro unit exhibited only a 1% increase in power draw during the transition. Additionally, the measured response of the micro-hydro unit did not display the power oscillations that were observed at full-scale. Since shaft power, as opposed to active power, was measured in the laboratory it was concluded that the full-scale response was, in part, electro-mechanical in nature, initiated by a sudden inhibiting torque applied to the runner.

The influence of two key parameters, guide vane opening rate and initial TWD level, were investigated on the UTAS-MH turbine unit. A range of tests were performed at guide vane opening rates equivalent to $0.5 \times$, $1 \times$ and $2 \times$ the current full-scale maximum rate, when non-dimensionalised by the water acceleration time constant of the system. Additionally, tests were performed with the tail water free surface depressed to $0.23D$, $0.56D$ and $1.0D$ runner diameters below the runner exit plane. The range of $0.23D$ to $0.56D$ is equivalent to the location of water level sensors installed in the full-scale draft tube cone, while $1.0D$ is beyond the practical range of tail water depression level.

The rate of guide vane opening was shown to have a considerable effect on the time required to switch from TWD to generation mode of operation. Interestingly, while the more rapid opening rates resulted in a faster production of positive power, the guide vane position at peak power draw actually increased with increasing opening rate. This result suggests that the cause of the inhibiting torque is not merely due to the incoming flow angle and that a storage effect may be present. This hypothesis is also supported by the existence of significant delays seen between the penstock pressure response and the measured power response at both full-scale and micro-hydro scale.

The magnitude of the associated low pressure transient caused by the sudden guide vane opening ranged from 11% of rated turbine head for the slowest opening rate up to 28% for the most rapid opening tested.

Conversely, tests performed under varying TWD levels were shown to have a negligible effect on the power generated during transition. This was also supported by full-scale test results.
To further examine the transient behaviour and dynamic response of a Francis turbine power plant an improved one-dimensional numerical model was proposed. The model is based on the formulation of the hydraulic model for system dynamic studies provided by the 1992 IEEE Working Group [50], however, the conventional representation of the hydraulic turbine as a simple orifice was improved upon by considering inlet and outlet flow velocity vectors, effective machine geometry and identified loss components in the calculation of turbine output. Additionally, a nonlinear guide vane function was incorporated to account for the nonlinear relationship between servomotor stroke and machine flow as previously suggested by numerous authors [23, 36, 74].

The numerical model was thoroughly validated against field test data. Simulation tests were run for a range of (non-TWD) transient tests performed at the Reece Power Station. The recorded guide vane servomotor stroke was used as model input, while simulated penstock pressure and active power were recorded as model outputs. The proposed numerical model was shown to be a significant improvement upon two of the standard model formulations, given by the IEEE Working Group [50] and Kundur [61], in predicting machine behaviour in terms of both power output and penstock pressure following a rapid guide vane movement. Importantly, the new model remained valid even at low part-load operation. While the model representation of the generator and system dynamics was relatively basic, the formulation of the new turbine model may be easily incorporated into an existing system model as commonly used within the hydropower industry.

Based on findings from laboratory and full-scale TWD transition tests, a mechanism is proposed to describe the transition process from a depressed tail water mode following rapid guide vane opening. Key pieces of evidence from various tests were discussed and interpreted in terms of the law of conservation of angular momentum as applied to two connected control volumes. The control volumes represent the annular volume between guide vane ring and runner inlet (denoted Region 1), and the region encompassing the runner from blade inlet to outlet (denoted Region 2).

The transition sequence from TWD to generation may be considered as occurring in three key stages: i) accumulation, ii) transition and iii) normal behaviour according to steady flow Euler theory. The proposed transition mechanism may be summarised as follows:

i. Initially, as the guide vanes are opened, flow is accumulated within the annular volume of Region 1. The flow, due to the initially high tangential velocities and the pumping action of the turbine runner, remains largely on the periphery of the enclosed volume resisting entry into the runner proper.
ii. At a given point following opening, water will begin to enter the runner (Region 2). The flow, with a reduced tangential flow velocity in relation to that required for power production, will initially impart a negative torque on the runner. As the guide vanes continue to open, the volumetric flow rate continues to increase and the inhibiting torque applied to the runner by the low momentum fluid reaches a maximum.

iii. Finally, the incoming tangential velocity component of the flow attains a value such that, for the current machine flow rate, positive power output is realised. This rectification of the inlet velocity triangle occurs relatively suddenly and results in a sudden change in rotor phase angle of the generator, causing the observed power oscillations. As load is increased at a steady rate the power fluctuations subside and normal operation is established.

The transition mechanism described above was incorporated into the new turbine numerical model. Simplified mechanisms describing flow accumulation, reduced inlet tangential velocity, churn-free operation and a more representative network model were included within the improved turbine model formulation. Simulated penstock pressure and power response was validated against field test data for rapid TWD start-up tests performed at the Reece Power Station before, and after, governor retuning.

The improved turbine model was shown to capture the general nature of the Reece Francis turbine unit during rapid start-up, providing validation of the proposed transition mechanism. Moreover, the model provides a predictive tool that may be used to calculate the expected FCAS contribution for a given opening profile. The model, however, was shown to consistently over-predict the power output in the later stages of transition. This is thought to be due to the absence of an accumulation term associated with Region 2 and a related requirement to include some form of transitional churning loss into the numerical model.

Following a discussion on several potential improvement methods, the improved turbine numerical model was used to explore a number of selected guide vane opening profiles to maximise potential FCAS contribution. The simulated FCAS generated by the selected opening profiles were compared to the validated simulation response of the T_{R7} test case giving a base FCAS contribution of 21.3 MW.

A two-stage opening of 16.5 mm/s followed by a 2 s ramp at 66 mm/s was found to provide the largest gains in FCAS contribution of the profiles tested, producing a minimum penstock pressure of 53.8 m, well above the safe operating level. The simulated FCAS contribution was 50.6 MW, a percentage increase of 138%.
The improved turbine model represents a new method of more accurately modelling the
general dynamic behaviour of Francis turbine power plants while also, for the first time,
providing a simulation tool for prediction of unit behaviour during the proposed rapid
machine start-up from TWD mode. The structure of the model enables easy testing of
other stations by changing a small number of key parameters and may also in the future be
adapted for use in Kaplan or Pelton power station studies.

9.1 Recommendations for Future Research

9.1.1 Micro-hydro Turbine Unit Design Improvements

The improved pump-as-turbine design presented in this thesis presents exciting new re-
search opportunities with potentially wide reaching benefits. Impeller modifications such
as those investigated by Singh [94] including blade tip rounding, or more complex proce-
dures such as blade profile optimisations as performed by Derakhshan and Nourbakhsh
[25], could be undertaken on the UTAS-MH unit to significantly improve peak efficiency.
It would also be interesting to examine the effect of such modifications on the off-design
performance as quantified by the weighted-average-efficiency (WAE).

Additionally, tests on other converted pumps would provide a more comprehensive view of
the potential applications in the context of micr-hydro power installations. Furthermore, the
development of a simpler design of micro-hydro unit that still included inlet flow control
would make the unit more commercially attractive.

9.1.2 Increased Flow Visualisation and Instrumentation on the UTAS-MH Unit

Increased flow visualisation and instrumentation on the existing UTAS-MH unit would
allow for a more complete exploration of the transition between tail water depression (TWD)
mode and generation mode. The addition of flow visualisation in the space between guide
vanes and runner (the region denoted Region 1 in this document) would provide further
verification of the proposed transition mechanism which includes an accumulation phase
and a (relatively speaking) gradual increase in negatively applied torque on the runner.

Additionally, flow field measurements within this space would provide valuable data in
terms of the inlet velocity vectors and flow condition responsible for the inhibiting torque
which could then be more accurately represented in the improved numerical model.
Perhaps more importantly, now that the transition process is more understood, is the need to determine the resulting runner stresses caused by the rapid start-up procedure. The work presented in this thesis has proven the ability of Francis turbines to provide fast frequency control ancillary services via this start-up procedure, however, the motivation for doing so was to limit the exposure of Hydro Tasmania turbine units to operation within the rough running part-load zone which is known to decrease asset life. The next step before this mode of operation can be confidently implemented is to quantify the stress on the runner and weigh up this effect with that of running constantly at part-load operation.

9.1.3 Laboratory Tests Examining Potential Improvement Methods for Increasing Rapid Start-Up Response

Further laboratory testing should be performed to examine the potential for improvement in rapid response. As identified in Section 8.6, such improvement methods may include, but are not limited to, increased machine inertia, assisted air expulsion, pressure transient control, auxiliary flow source(s), as well as further investigations into guide vane opening profile influence.

9.1.4 Improved Turbine Numerical Model Refinements

A number of further refinements may be made to the presented turbine model to improve the predictive capabilities under a wider range of rapid start-up conditions. The most important area of improvement is in understanding the very first moments following guide vane actuation, and being able to accurately capture the penstock dynamics resulting from opening the guide vanes into an evacuated space with altered discharge characteristics. Furthermore, the simplified mechanisms incorporated into the current model may be improved upon with additional full-scale and laboratory test data.

9.1.5 Full-scale Field testing of Other Francis Turbine Power Plants

It is recommended that further full-scale testing be performed both at the Reece Power Station and on other suitable turbine units. Field test measurements of flow and shaft torque, in addition to penstock and draft tube pressure and active power output, would provide valuable information. Rapid start-up testing on other Francis turbine units would also aid in developing an understanding of the key parameters required for optimal rapid generation. Data from units of varying specific speed and turbine geometry would further enhance the understanding gained from this research project.
9.1.6 Detailed investigation into the bubbly flow within the draft tube cone during transition

The free surface deformation within the draft tube prior to, and during, transition from TWD mode is highly complex due to the aerated bubbly layer that forms and the sloshing that results from the interaction with the rotating runner above. This bubbly layer, along with the presence of a large air cavity above may significantly modify resonant frequencies and affect the propagation, and magnitudes, of pressure transients associated with rapid guide vane movement and therefore warrants a more detailed investigation.
Bibliography


