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Small-scale Wind-Powered Seawater Desalination Without Batteries

by

Marcos dos Santos Miranda

Doctoral Thesis
Submitted in partial fulfilment of the requirements for the award of

Doctor of Philosophy of Loughborough University
25th July 2003

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To Simone and Clara,
my renewable sources of life.
ABSTRACT

Potable water is a commodity taken for granted by many in modern society. In places where it is not naturally available, it is usually produced by adequate processing of the supply from other sources, as is the case with seawater desalination. Such processes require an energy supply, which just as well may not exist at many of these locations.

In view of the above, this work focuses on the study of two well-established technologies and their integration: water desalination by Reverse Osmosis (RO) and electricity generation using Wind Energy.

Based on the premise that no energy backup or storage devices would be employed, two alternative wind-powered RO system configurations are proposed. Their components are individually described and modelled. Control strategies are devised for both systems, aiming at making the best possible use of the energy available. The expected performances of both systems are assessed through simulation of computer models.

Based on the simulated performance results, one of the systems is chosen for further development. A prototype system is built and experimental tests carried out. The design of the prototype is detailed and the results obtained are presented. In the light of these results, the developed model is validated and the viability of the system is discussed.

Finally, practical implementation issues are discussed; a case study is introduced, including performance predictions and a simplified economic analysis presented.
ACKNOWLEDGEMENTS

I thank Prof. David Infield for his supervision, commitment, and support in both, the technical and personal fields, throughout this work. I would also like to thank Dr. Gordon Smith for his light-hearted and skilled co-supervision and invaluable help with part of the experimental work. I have gained much from both, and through their character, competence and dedication I have learned to respect and admire them.

I thank Murray Thomson for sharing with me the challenge of developing the test-rig used in this work. Murray is now finishing his thesis on PV-powered reverse osmosis, and although I have not seen his work, I am sure it will be of interest to those willing to know more about reverse osmosis systems powered by renewable energy sources. I also thank him and his family for all the help and support given to me and my family during this time.

I am grateful to the Brazilian National Council for Research and Development – CNPq, for their financial support, including my PhD studentship and university fees.

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Finally, but by no means less important, I thank my wife, Simone, and daughter, Clara, to whom I dedicate this work. Without them, this whole experience would not have been half as fun.
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CHAPTER 1
INTRODUCTION

The scope of the work is introduced, including the motivation and the main objectives. The methodology employed and the most significant achievements are briefly described and an outline of this thesis is presented.

1.1. Context

The availability of a suitable water resource is indispensable for the social and economic development of any settlement. This poses an important issue when arid and coastal regions (including islands) are taken into consideration. In some locations it becomes imperative to create a distribution infrastructure in order to meet the demand, such as pumping over long distances ([Childs, 1992]) or even transporting large volumes of potable water ([Avlonitis, 2002]), which leads to high water prices. In order to find technically and economically viable solutions for this problem, considerable attention has been given world wide to the research and development of water desalination technologies.

Another aspect to be considered is that these settlements may be located far from urban centres, having a precarious electrical supply, if any. In such locations electricity is often supplied by a weak grid or even generated locally, by means of diesel generators.
In view of the above, this work focuses on the study of two well-established technologies and their integration, viz. water desalination by Reverse Osmosis (RO) and electricity generation using Wind Energy.

1.2. Proposed Work

The use of renewable energy sources for the production of potable water is a matter of great interest worldwide, not least for its potential to provide a sustainable alternative for the further development of the communities concerned.

Within this wider context, the research work described in this thesis comprised the study of wind-powered seawater desalination systems. Amongst the desalination technologies available, reverse osmosis was chosen particularly for its suitability to both, seawater desalination and wind power.

The combination of wind power and reverse osmosis into an integrated system has been often addressed in the literature, however the resulting system configurations proposed have almost invariably included the use of either backup or energy storage devices typically lead acid batteries. In these circumstances, the use of reverse osmosis technology does not differ much from its already established market of grid connected desalination systems, which operate from a constant power supply.

Taking into account the additional problems that may arise from the introduction of either backup or storage devices, the concept of designing a system that could operate efficiently from a variable power supply, such as the wind, was incorporated. It should be pointed out that desalinated water has a higher energetic content than seawater and therefore itself represents an energy storage medium.

This work focused on the study of small-scale stand-alone wind-powered reverse osmosis seawater desalination.

It is acknowledged that such a system, particularly because of its size, may be of greater interest to a very specific niche market. Nevertheless, it may present itself as a sustainable and even economically viable alternative where no other local resources are present, the cost of water is considerably high, or at places with an exceptionally good wind resource ([Assimacopoulos, 2001]).
1.3. **Motivation**

The initial motivation for this work started with an earlier project carried out by Dulas Limited, UK ([DULAS, 1996]), in which the viability of stand-alone operation of a desalination system powered from PV panels was assessed. Despite the fact that the resulting system configuration had not been extensively tested, the results were promising and led to a subsequent project, this time in conjunction with CREST, Loughborough University, UK ([Thomson, 2001]). This latter project built on the experience acquired during the previous one and aimed at designing, building and testing a complete prototype system.

The aforementioned joint project served as the basis on which this work has been proposed and developed, albeit using the wind as the primary power source rather than photovoltaics. Much of the hardware (individual components) was inherited from the previous project, including the test rig, at the earlier stages (modelling of the components). As the work progressed, further components were acquired and a new prototype test rig was designed and built from scratch.

1.4. **Objectives**

The main objective of this work is to propose and assess the viability of a reverse osmosis desalination system having wind power as its sole energy supply. The actual product output of the system is ultimately dependent on the wind resource, but the design of a system that can provide over 3 and up to around 10 m$^3$/day of drinking water is envisaged.

No energy storage devices are employed, apart from the desalinated water storage tank, and therefore, the system must operate in a way to accommodate the fluctuations present in the wind. From the RO membranes perspective, this implies in variable flow and pressure operating conditions.

Besides, bearing in mind the uncertain nature of the wind, it becomes highly desirable that the system presents excellent overall performance in respect to the energy usage. Hence, improved energy efficiency was the main guideline adopted during the system design. This implies not only in careful choice of system components, but also in the selection of an adequate energy recovery device and the development of a
suitable system operation control strategy. It was anticipated that short term variations in wind speed (turbulence) would pose significant demands on the design of the control system.

1.5. Work Progress

The development strategy adopted consisted in the following stages: defining a preliminary configuration for the system; establishing satisfactory mathematical models for each of the system components from detailed laboratory testing; using the individual models to build the full system model, including the control strategy and finally, building a prototype system based on the analysis of the previously established system model.

The preliminary system configuration included a hydraulic motor as a means to recover the energy available in the concentrate stream. As the worked progressed, an alternative configuration using the Clark pump was suggested, tested and acknowledged as having better performance then the originally proposed system. In the interest of adequately assessing the new configuration, a new prototype rig was designed and built using the Clark pump. Nevertheless, the original configuration using the hydraulic motor was also analysed using computer simulations of the model and is also presented in this thesis.

1.6. Achievements

The idea of building a desalination system powered exclusively from renewable sources may not be original and the principles involved date back from as early as the 70's ([Lising, 1972]). The fundamental characteristic of the system presented in this thesis that sets it apart from those found in the literature is the absence of an energy back up/storage device, be it a diesel genset or a bank of batteries.

The most significant contribution of this work is undoubtedly the construction and operation of a fully functional stand-alone wind-powered reverse osmosis desalination system, with no storage devices. It must be pointed out that due to the time constraint, the final system was not exhaustively tested and further analysis would be needed before it could be used in a non-controlled environment, without close assistance.
Other achievements can be listed as the modelling and analysis of the system; the development and implementation of the system control strategy and the characterisation of the RO membranes over a wide range of operation.

1.7. Outline of Thesis Structure

In Chapter 1, the motivation and scope of the work are set out. The objectives as well as the main achievements are introduced and the text of the thesis is outlined.

In Chapter 2, a literature review that includes the main desalination technologies and the viability of their integration with renewable energy sources is presented. Special attention is given to wind-powered reverse osmosis systems, as they are the main object of this study.

In Chapter 3, a preliminary configuration is outlined and used as a guideline for the development of the whole system model. A description of the system components and their mathematical models is presented. Some details of the laboratory setup used in the modelling tests of some components are also described here.

The performance of different system configurations, including the control strategy, was assessed and compared by computer simulation, using the component models developed in Chapter 3. In Chapter 4, two of these configurations are analysed in greater detail. Their layouts are described and performance predictions are compared and discussed. One of the systems modelled is chosen for further analysis.

In order to improve the overall understanding of the most promising of the systems modelled in Chapter 4, a prototype was designed and built. In Chapter 5, the system model is analysed, adjusted and validated against data obtained from the initial experimental tests on the prototype.

Chapter 6 presents the construction characteristics and implementation issues regarding the prototype. Experimental results are shown and analysed in light of the expected system behaviour derived from the performance predictions previously discussed.

In Chapter 7, some of the aspects regarding the practical implementation of RO systems are discussed. From these practical considerations, some remarks regarding a possible implementation of the proposed system in the field are made. Also, a system
design exercise is conducted, using modified performance curves, similar to those established in Chapter 4. Finally, a simplified cost analysis is presented.

In Chapter 8, the conclusions of the work are presented. The contributions achieved are outlined and, in light of the results obtained, recommendations for further development of the work carried out are made.

Complementing the main text, a list of the parameters of components used to build the prototype system, including the controller gains, is shown in Appendix A. Details on the variables measured and the sensors used are given in Appendix B. The controlled load/voltage limiter (Buck converter) used both in the wind turbine modelling and the prototype testing is further detailed in Appendix C.
A literature review that includes the main desalination technologies and the viability of their integration with renewable energy sources is presented. Special attention is given to wind-powered reverse osmosis systems, as they are the main object of this work.

2.1. Water Desalination

Currently, many different desalination technologies are used for the purpose of supplying potable water. These technologies can be divided into two distinct groups according to the physical characteristics of the process. There are the thermal processes, in which the solvent (water) is submitted to a phase change in order to be separated from the solute (salts), and the membrane processes, where the solution is filtered, causing a reduction of its concentration ([Manwell, 1994]).

Thermal (phase-change) processes, in particular distillation (solar stills, multi-stage flash distillation, multiple effect distillation and vapour compression) and freeze separation methods, require energy inputs proportional to the amount of water produced.

Membrane processes, notably electrodialysis and reverse osmosis, depend on membrane properties and require energy inputs proportional to the salt content of the feed water.
It should be kept in mind that when the term “water desalination” is employed, two different types of feed solution are considered: seawater and brackish water. The former one presents higher solute concentrations, usually varying from 32,000 to 40,000 ppm (32-40 kg/m³) of total dissolved solids (TDS). The salt concentration range for brackish water is wider and generally taken from 2,000 to 10,000 ppm (2-10 kg/m³) TDS. This distinction is important with regard to system design, since the configuration, or even applicability, of some desalination technologies may depend on the nature of the source water.

2.1.1. Review of Desalination Technologies

It is desirable that this review on desalination techniques includes not only a description of the methods themselves (which are not necessarily recent), but also an analysis of latest developments, technical and economical comparisons and viability perspectives. Therefore, to reflect the most up to date information available, this section of the thesis was mostly based on reports written within the last few years ([THERMIE, 1998], [ETSU, 1996], [EURORED, 1996]).

a) Solar Distillation (SD)

In solar distillation plants, heat from the sun is directly used to evaporate brackish water or seawater in solar stills. These stills are very simple in construction and for well-designed plants daily production can reach up to 2.5 l/m² with a thermal efficiency of 50%.

Despite the existence of many different configurations, solar stills can generally be taken as being variations of the same basic construction (Figure 2.1), which resembles a greenhouse. The saline feed water is run continuously (or intermittently) along a shallow enclosed pool, which is covered with an inclined glass or plastic material roof. Part of the direct solar radiation will be absorbed by both the salt water and the still bottom, causing the water temperature and vapour pressure to rise. The air-vapour mixture is carried by convection to the internal roof surface, condenses and drains into side-mounted troughs, from where it is conveyed to storage. The remaining brine concentrate is removed from the pool to prevent salt deposition.
Although very simple in terms of operation and maintenance as well as inexpensive in terms of running costs (only electricity for pumping needs to be paid for), solar distillation's main restriction is related to capital costs ([Manwell, 1994]). The requirement for extensive land usage and building costs limit its application to very small-scale (about 0.75 m³/day) systems, and preferably to locations where land costs are low and solar radiation is high.

![Figure 2.1 - Typical solar still configuration](image)

b) Multi-stage Flash Distillation (MSF)

Multi-stage flash distillation is the most widely used desalination technique, accounting for more than 50% of installed capacity worldwide. The process consists of evaporating (flashing) the pressurised feed water along a series of stages at progressively lower pressures and temperatures. In each stage, part of the water "flashes", due to a sudden reduction of pressure and then condenses. The remaining feed, now at a lower temperature, will be conveyed to the next section, where its pressure will once again be reduced, causing more vapour to be released and condensed.

In order to increase system efficiency, an energy recovery scheme can be implemented. This is done by allowing the feed water pipeline to pass through the various plant stages, pre-heating the solution. After the last stage, it is boiled at its final temperature, usually ranging between 90 and 120 °C. Then, the heated solution is fed to the serial stages through a restriction, which causes the pressure to drop and part of the solution to flash. The condensed product collectors are connected in parallel to the
output pipeline. As the product flows to the next stage, it is also flashed so that heat is released, helping to increase the feed water temperature. This upper limit of 120 °C is generally applied to most distillation processes, due to chemical scaling problems at higher temperatures.

The number of stages to be implemented, generally ranging from 4 to 40, is one of the design aspects of a MSF unit. A compromise has to be achieved, since using more stages allows the heat exchange area to be smaller, but incurs in higher capital costs. It is a relatively simple process to operate and maintain and a modern plant life cycle is expected to reach up to 40 years.

c) Multiple Effect Distillation (ME)

This process was the first method to be used for seawater desalination and is mostly employed in the chemical industry, where it was initially developed. It is in some aspects similar to the MSF technique in which some of the vapour is generated by flashing, but most of it is obtained by boiling.

The multiple effect distillation process is based on the heat exchange between the evaporated product and the feed water. As the feed flows into the plant through the interconnected effects (stages), it is heated by the generated vapour. This heat transfer causes the feed to evaporate and the vapour to condense. The effective recycling of heat (condensation-evaporation) is accomplished by having consecutive reduced pressures in the effects, allowing the feed to boil at lower temperatures.

Modern ME systems can be either of vertical or horizontal tube type. In the former, the feed water is boiled inside the heat exchanging pipes while the vapour condenses outside. In the horizontal tube systems, the opposite occurs. By using the horizontal pipe configuration, a stacked unit can be designed, with consecutive effects built upon the other, allowing the brine to flow by gravity. Such a configuration is suitable for the implementation of high performance small systems.

d) Vapour Compression (VC)

The vapour compression technique is based on a very simple principle. In contrast with other distillation methods where heat is directly transferred to the feed solution, here higher temperatures are reached by means of an increase in system pressure. A
A typical system is composed of an evaporator, a compressor and a heat exchanger (Figure 2.2).

Part of the vapour produced by the evaporator is directed to the compressor and pressurised, increasing its condensation temperature. This pressurised vapour is then returned to the evaporator through tubes that work as heat exchangers between the vapour and the feed solution. As the vapour condenses and leaves the system as purified water, the tube is heated, increasing the feed solution temperature and causing more vapour to be generated. Besides promoting an increase in the pressure of the evaporated water - and consequently in its condensation temperature - the compressor also reduces the feed water pressure, thus reducing its boiling point.

Most of the energy input to the system is used to drive the compressor and can be either mechanical or thermal vapour compression. An additional heat source must be also provided for start up and to compensate for thermal losses.

To increase system efficiency a heat exchanger is used to pre-heat the feed solution using the remaining, and otherwise, wasted heat available in the brine and distillate outlets. Furthermore, increased surface area of evaporator tubes and lower compression ratios can lead to a higher productivity.

![Vapour compression system diagram](image)

**Figure 2.2 - Vapour compression system diagram**

e) **Freeze Separation (FS)**

Despite the very low energy consumption presented in prototype systems, freeze separation processes have not yet been widely used in production scale. The process is
very simple: a cold refrigerant removes the energy from the feed solution, cooling it down until ice crystals start to form on its surface. As an inherent characteristic of the phenomenon, impurities are left out during the formation of the crystals. Once a layer of crystal is built, it is removed from the solution and melted to form purified water.

Energy is mostly required for the freezer compressor and cooling coil, and the feed water temperature shows small effect on system performance. Although such a separation method is economically viable in the preparation of more expensive products, for water desalination it is not used in practice due to its low productivity and complex equipment requirements.

**Electrodialysis (ED)**

Electrodialysis is an electrochemical process where the ions dissolved in the feed solution are removed through selective membranes, and among the desalination technologies, it is the only one that demands electrical power as the main energy source.

Some input energy is required to pump water through the ED stacks and to operate them. The stacks are composed of alternate layers of positively and negatively charged ion-selective membranes and the power required to operate them is proportional to the amount of ions dissolved in the feed solution. Thus, this technology becomes more attractive for less concentrated solutions, i.e. below 10,000 ppm, and it is considered the most energy efficient technology for water concentrations below 1,500 ppm ([Manwell, 1994]).

As seen in Figure 2.3, as water flows along the stacks, the dissolved ions are deflected towards the charged membranes, which will allow them to pass through in only one direction. As a result, at the end of the stack the solution concentration is reduced and by using consecutive stacks, considerable reductions can be achieved.

Extra attention is required for the feed water pre-treatment, since membrane fouling may occur. A recent improvement to the ED process is called Electrodialysis Reversal. In this method, the polarity of the membranes is regularly reversed to prevent deposition, thus reducing the amount of chemicals to be added.
Figure 2.3 - Electrodialysis schematic diagram

b) Reverse Osmosis (RO)

Reverse Osmosis (RO) is a relatively new process, having experienced a great expansion due to materials development in membranes manufacturing in the last two decades. As a general separation process ([Slater, 1986]), reverse osmosis is widely used in industry for the purification of effluents (in the permeate stream) or reclamation of chemicals (in the concentrate stream). Nowadays, it is even available for home use, operating at tap pressure, to reduce water hardness. It is one of the most widely employed technologies due to its low specific energy consumption (kWh used per m³ of desalted water) and excellent performance (low concentration of product), particularly for high concentration feed solutions, such as seawater.

Macrofiltration processes (e.g. cartridge filters, bag filters, sand filters, multimedia filters) are usually limited to particles sized down to 1 micron. They are characterised by “throughflow” filtration, i.e. they present only one outlet (the filtrate). Contrary to macrofiltration, the removal of particles in RO (as in the other membrane separation technologies, e.g. Micro-, Nano-, and Ultrafiltration) is achieved by “crossflow” filtration (two outlets – one permeate and one concentrate).

Reverse osmosis is a separation process in which a feed solution (solute dissolved in a solvent) is pumped at high pressure along a semi-permeable membrane. If the pressure exerted on the solution is higher than the difference of the solution osmotic pressure across the membrane, the natural osmosis process is reversed. This means that
the solvent will flow through the membrane from the higher to the lower concentration side. This difference between the exerted pressure and the osmotic pressure difference is called trans-membrane pressure. There are two resultant effluents from the process: the permeate, or product, consisting of the purified solution, with low solute concentration; and the concentrate, or brine, with higher concentration than the feed solution. A diagram of the process can be seen in Figure 2.4.

![Figure 2.4 - RO schematic diagram](image)

where,

- $C_f$, $C_p$ and $C_c$ are the feed, product and concentrate streams concentration (ppm);
- $Q_f$, $Q_p$ and $Q_c$ are the feed, product and concentrate streams flow rate (l/s);
- $P_f$ and $P_c$ are the feed and concentrate streams hydraulic pressure (bar). As the product stream quite often flows into a collection tank for further pumping to another reservoir, its pressure is taken as the atmospheric pressure and is not represented above.

In water desalination, reverse osmosis systems can be divided into two main categories, according to the salt concentration of the feed water. The first one is the purification of brackish water (BWRO) with concentrations ranging from 2,000 to 10,000 ppm, implying in low osmotic pressures (up to 10 bar). The other application is seawater (SWRO) desalination, where salt concentrations are higher (between 32,000 and 40,000 ppm) and osmotic pressures are about 25 bar and higher.

### 2.1.1.1. Summary of Desalination Technologies

Table 2.1 provides a general figure on the usage of the different desalination technologies in the countries where such systems play an important role in the supply of fresh water.
Table 2.1 - Comparison of desalination technologies - Installed capacity (1,000 m$^3$/d)

<table>
<thead>
<tr>
<th>Country</th>
<th>MSF</th>
<th>ME</th>
<th>VC</th>
<th>RO</th>
<th>ED</th>
</tr>
</thead>
<tbody>
<tr>
<td>Saudi Arabia</td>
<td>2,700</td>
<td>50</td>
<td>50</td>
<td>1,000</td>
<td>94</td>
</tr>
<tr>
<td>USA</td>
<td>50</td>
<td>50</td>
<td>130</td>
<td>1,600</td>
<td>280</td>
</tr>
<tr>
<td>Kuwait</td>
<td>350</td>
<td></td>
<td>50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Libya</td>
<td>400</td>
<td></td>
<td>130</td>
<td>67</td>
<td></td>
</tr>
<tr>
<td>Spain</td>
<td>56</td>
<td>40</td>
<td>230</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>Italy</td>
<td>200</td>
<td>75</td>
<td>40</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Algeria</td>
<td>60</td>
<td>30</td>
<td>80</td>
<td>16</td>
<td></td>
</tr>
</tbody>
</table>

Source: [THERMIE, 1998]

Since the implementation of these technologies is based on distinct principles of operation, involving different process phenomena, some characteristic parameters need to be established to provide a common ground for comparison. Table 2.2 shows how some of these parameters compare to each other.

Bearing in mind that the main object of this work are wind power driven desalination systems, two factors should be carefully noticed. The first one is the specific energy consumption. Due to the uncertain nature of the power source, it becomes desirable to produce the most water possible out of the available energy at any time. Additionally, this parameter has a decisive influence on system costs.

The other aspect, which is also inherently related to the fluctuating nature of the wind, is the appropriateness of the desalination technology to the energy source, i.e. the tolerance to intermittent use operation. This is mostly critical to the thermal processes where start-up times are long and interruptions in the source would cause considerable energy waste.
Table 2.2 - Comparison of desalination technologies - Energy requirements and product quality

<table>
<thead>
<tr>
<th>Desalination Technology</th>
<th>Specific Energy (kWh/m³)</th>
<th>Product Water Quality (g/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Multi-stage flash (MSF)</td>
<td>10 – 14.5</td>
<td>~ 10</td>
</tr>
<tr>
<td>Multiple effect (ME)</td>
<td>6 – 9</td>
<td>~ 10</td>
</tr>
<tr>
<td>Vapour compression (VC)</td>
<td>7 – 15</td>
<td>~ 10</td>
</tr>
<tr>
<td>Electrodialysis (ED)</td>
<td>0.7 – 2.2</td>
<td>~ 350 - 500</td>
</tr>
<tr>
<td>Reverse Osmosis (RO)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brackish</td>
<td>0.5 – 2.5</td>
<td>~ 350 - 500</td>
</tr>
<tr>
<td>Seawater</td>
<td>4 – 13</td>
<td>~ 350 - 500</td>
</tr>
</tbody>
</table>

Source: [THERMIE, 1998]

Additionally, other aspects such as the ability to operate at variable conditions, ease of maintenance, portability of the system, should be also assessed. Considering all this, it is straightforward to see why reverse osmosis is so widely used for the implementation of wind powered desalination systems.

2.1.2. Reverse Osmosis

Although being a very widely used process nowadays, having applications in many areas such as biochemistry, microbiology, industrial separations and environment protection, RO is a relatively recent process. The roots of this technology lay back in the 18th century, when the osmotic process was first noticed in 1748 ([Glater, 1998]). Despite the apparently long time span, it was not until 1867 that the first inorganic membrane was developed.

By the 1930's, only few polymeric materials were known. Plastics and films were mostly derived from cellulose. Modern polymer science emerged in 1937, when nylon was developed, and experienced substantial growth after World War II. And in spite of the fact that pressure driven membrane desalination processes are about 100 years old, the first textbooks on desalination by synthetic membranes were only published in the mid 1960's.

The first effort to implement an RO system started in a project carried out by UCLA by Hassler, in 1954. At the time, reasonable yields were not accomplished and the project was abandoned in 1960.
In the mid 1950's, at University of Florida, Reid and Breton provided experimental details on membrane materials, cellulose acetate being the most promising, with high salt rejection levels, but poor product fluxes. Additionally, they proposed the first models for inter-membrane transport (permeation selectivity).

In 1958, also in UCLA, Loeb and Sourirajan focused on increasing the levels of product water, while maintaining high rejection levels. The major developments were achieved by improving the manufacturing process. This effort culminated in the development of first practical RO membrane.

During the last three decades, efforts in the research of RO systems led to the development of different membrane technologies as well as the proposition of alternative system configurations such as centrifugal RO ([Wild, 1997]). Even so, there is still considerable room for development mainly in respect to system energy consumption.

2.1.2.1. Working Principle

The spiral wound technology was chosen from the main types of RO module design available in the market when the initial research for this thesis was carried out. The four types of RO module available commercially are tubular; plate-and-frame; spiral wound and hollow fibre. Spiral wound units were favoured due to their widespread utilisation, low cost and easy availability from a number of manufacturers.

Effectively, a spiral wound module is a sandwich of membrane and spacer layers wrapped around a product water collection tube (Figure 2.5). The layers are glued together around the edges, with exception of the product tube end, which is connected to an opening on the tube. As the feed solution is pumped across the module, at high pressure, from one of its ends, the solvent permeates through the membrane, which blocks (rejects) most of the solute. The solvent, now in the spacer layer area, flows into the collection tube, by which it exits the module as the product stream. The feed solution, now with increased concentration of the solute, exits the opposite end of the module as the concentrate stream.
2.1.2.2. Energy Recovery

In all desalination techniques, energy consumption plays a very important role. It is a key selection criteria, and can be adopted as a means to place them all onto a common ground for comparison. This is due to the effect that energy costs have on the system total cost of production, i.e. price per volume of purified water. Therefore, this matter should be carefully considered in small stand-alone applications, where energy costs are significant.

Energy consumption is a prime concern when considering RO desalination mainly due to the relatively low utilisation of the supplied energy, which accounts for a substantial part of the costs related to system operation. This low energy utilisation takes place because, as illustrated in Figure 2.4, water pressurisation is commonly achieved by implementing some form of restriction (valve) to the concentrate pipeline and most of the energy used in the process (up to 90%) can be wasted as a pressure drop across this valve. Hence, an interesting alternative to increase system efficiency and reduce the product water costs is to make use of this wasted energy by means of an Energy Recovery mechanism.

Due to the importance given to this matter, several energy recovery methods have been proposed in the literature and, among these, the most relevant are considered below.

![Schematics of a spiral-wound membrane module](image)
a) Membrane array configuration

The use of lower operating pressure membranes is another possibility to recover the wasted energy in RO systems. These membranes, referred to as ultra-low pressure membranes [Nemeth, 1998], are used in a brine-staging configuration, connected to the brine outlet of a regular RO vessel and are suitable for the application in brackish water desalination. Ultra-low pressure membranes present a salt rejection characteristic similar to conventional brackish water membranes, but besides operating at reduced pressures, they also show a higher permeate flux for a given pressure. Therefore, they may offer significant advantages in small-scale systems, where the reduced energy costs brought by the addition of an energy recovery device may not payback the capital investment.

b) Centrifugal reverse osmosis

Centrifugal reverse osmosis is a new experimental configuration, firstly introduced by [Wild, 1997]. Since the system proposed presents a reduced energy consumption due to its inherent energy recovery, it was included in this section. In this configuration, system operational pressure is achieved by the centrifugal force developed inside a spinning rotor. A schematic representation of the system can be viewed in Figure 2.6.

![Figure 2.6 - Schematic representation of a Centrifugal Reverse Osmosis system](image)

In contrast to conventional RO systems, the concentrate stream energy content is negligible, since it leaves the system at a very small radius, near to the axis of rotation. An analysis of system losses shows that the system efficiency would be proportional to
system size, and other benefits such as reduced membrane costs and increased reliability could be achieved, basing on the results obtained form a prototype. However, there are still some problems to be overcome such as membrane dispositions; pressure gradients within the rotor; the influence of centripetal acceleration on concentration polarisation and fouling; and losses associated with the rotor windage. This technology is not commercially available.

c) Brine turbine - electrical generator

An alternative to recover the energy available in the brine stream of RO plants and, at the same time, add some control over it is the use of an electrical generator connected to a brine recovery turbine ([Raptis, 1996]). Since the pressure drop across the turbine is related to the rotational speed, the control of the generator load (energy recovered) could be used to control the pressure and system recovery ratio. This generated energy could then be fed back either directly or via a power electronics converter into the grid to which the driving motor is connected. The benefits brought by using such configuration would have to be carefully considered. This is because although a degree of controllability is added, making viable to control independently the pressure and flow rates, it would present the drawback of cascading several components (turbine, generator and converter), incurring in reduced overall recovery efficiency.

d) Brine turbine - high-pressure pump

If compared to the previous configuration, the option of connecting the recovery turbine to the high-pressure pump, directly or through pulleys and a belt, benefits from a reduction of components. Additionally to the increase in efficiency, this configuration is more robust, demanding less maintenance and being less prone to failure.

A few possibilities have been proposed regarding the use of recovery turbines, such as reverse running centrifugal pumps, pelton turbines and hydraulic motors. Although being simple and readily available commercially, the use of centrifugal pumps as a recovery turbine is not suitable for smaller installations, mainly because of their low efficiency in such applications. Furthermore, their efficiency is considerably reduced if operating at partial load. The pelton turbine is a very established technology, widely used at medium scale electricity generation, but its use in small systems would incur high capital costs.
Among these alternatives, the hydraulic motor is the one that presents the higher efficiencies. They operate similarly to positive displacement pumps and are suitable for application in a wide power range. Although being commercially available, their application in the energy recovery of RO systems has not yet been proven regarding component wear and lifetime. Even so, promising results were obtained in a preliminary experimental study carried out by [DULAS, 1996] and [Thomson, 2001]. The hydraulic motor is further discussed in Chapter 3.

2) Interstage boost pump – Hydraulic Turbocharger™

In order to achieve higher product rates in RO systems and reduce polarisation effects, it is common to cascade the pressure vessels using the pressurised output of one set of parallel elements as an input to the next one. In this configuration arrangement, referred to as "brine-staging", fewer elements are used in the consecutive stages. By allowing this lower-pressure higher-concentration feed to be used as input to a subsequent element, the element has its trans-membrane pressure reduced, altering its performance and incurring in higher product concentrations. To improve this situation, a boost pump can be used, but this will lead to further capital and operating costs.

An alternative solution to the addition of a motor-powered pump is to employ the energy available in the rejected pressurised brine of the last stage to drive an interstage boost pump ([Oklejas, 1992]).

This system is essentially a pump directly connected to a turbine and is known as the Hydraulic Turbocharger™. The high-pressure output concentrate impels the turbine and this in turn, drives the pump to boost the interstage feed pressure. An interesting feature of the Turbocharger is the presence of an auxiliary nozzle with a pressure control valve in the turbine inlet. This valve allows a fine adjustment of the recovered brine energy over a 30% range of variation for constant brine flow. An example of the resulting configuration is represented in Figure 2.7.

This device can also be used as a booster for the feed pressure in a single stage system. In [Silbernagel, 1992], this configuration was extensively tested (over 3000 hours) proving the advantage of reduced energy consumption of the high-pressure pump driving motor and consequently, its power rating. Additionally, longer lifetime of the pump components is expected, since its output pressure is reduced. For the tests carried
out, a payback period of less than one year was estimated based on the typical energy costs of remote locations. Another feature of this turbine/pump configuration is its robustness, having shown practically no wear during the period of testing.

![Figure 2.7 - System configuration with the Hydraulic Turbocharger](image)

Although the individual efficiencies of the turbine and pump stages are relatively high (about 60% each), the total energy recovery is small. This is because, apart from any friction losses, the device total efficiency can be taken as the product of both individual efficiencies, i.e. about 36%. This recovered energy provided an average of 24% reduction in the power consumption of the tested system.

**Pressure Exchanger**

The Pressure Exchanger is another type of commercially available energy recovery device and is conceptually similar to the Hydraulic Turbocharger™. It works by means of a pressure transfer from the pressurised brine discharge and the low pressure feed. In contrast to the Turbocharger, the main feature of this device is the absence of a shaft connection, avoiding the turbine/pump configuration, which yields a poor overall recovery efficiency. A schematic representation and a configuration diagram of a system using the Pressure Exchanger can be seen in Figure 2.8 and Figure 2.9, respectively.

Here, there is no physical barrier to completely prevent the contact between the two saline flows, but even so, little intermixing takes place in practice. Another feature of this system is that the output from the device is not directly fed to the RO unit. It is only used to boost the pressure of part (2/3) of the feed input to a high-pressure positive displacement pump. The pump will work part as a pressure booster (2 out of 3 plungers)
and part as a high pressure pump (for the remaining 1/3 of feed flow). The high-pressure and low-pressure streams are isolated within the pump.

Figure 2.8 - Schematics of the Pressure Exchanger (FIGURE SOURCE: [Hauge, 1995])

Figure 2.9 - Diagram of the system configuration using the Pressure Exchanger (HPX)

In the tests carried out by [Hauge, 1995], the Pressure Exchanger showed an overall efficiency of 85-90%, bringing a reduction in power consumption of 40-50%. From the results obtained, a decrease of about 30% in the cost of the final product was estimated for a 40,000 m³/day desalination unit. These costs were assigned not only to the energy recovery, but also to the reduction in the capital and operating costs.

g) Hydraulic Variable Flow Pump – VARI-RO™

The VARI-RO™ system is a very interesting alternative in which it is not only an energy recovery option, but also incorporates the pumping device as well. In fact, it is a variable flow positive displacement pumping and hydraulic energy recovery system in one unit, being mostly suitable for large capacity systems (up to 20,000 m³/day).

The recovery/pumping system consists of a modified positive displacement pump in which an extra chamber is used to convert the brine discharge pressure into motion of the displacement shaft. A schematic representation of the VARI-RO™ system is shown in Figure 2.10.

With this configuration, the high efficiency of positive displacement pumping and its inherent matching to system head and flow (in case fouling occurs, the pressure rises
to keep flow rates constant) is achieved. The displacement piston velocity is controlled to provide trapezoidal flow waveform (instead of the normal sinusoidal pattern of crank-driven pumps), thus reducing eventual flow/pressure fluctuations.

**Figure 2.10 - Schematic representation of the VARI-RO™ system**

Different configurations can be implemented depending on the energy sources used to power the system. The power sources could be either electricity or heat - supplied by natural gas or even waste heat from other processes. In one of these configurations an electrical motor/generator could be included so that excess heat could be employed to generate electricity either for local use or for sale, helping to reduce implementation capital costs.

Simulated performance projections presented in [Childs, 1995] show that specific energy consumption remains roughly constant for a wide range of recovery ratios. This type of pumping/recovery showed considerable savings in terms of energy consumption if compared to a system with centrifugal pumping and recovery turbine (reductions by 36%).

**b) Three-Chamber-Pipefeeder**

The three-chamber-pipefeeder is another energy recovery system that uses the same operational philosophy as the Pressure Exchanger and has been used in mining for long time ([Krumm, 1996] and [Geisler, 1998]). Here, the inlet/outlet ports of the chambers are valve-controlled allowing the water to flow in the desired directions. A
schematic diagram of one of the chambers can be seen in Figure 2.11.

![Schematic Diagram](image)

**Figure 2.11 - Schematic representation of one of the chambers of the three-chamber-pipefeeder**

The resulting system configuration is such that only part of the feed water is pressurised by a high-pressure pump. Most of the feed will be diverted and used as an input to the recovery device. Since the amount of feed diverted has to be equal to the amount of brine discharged from the RO unit and considering the mass balance within the RO unit, the amount of feed pressurised by the high-pressure pump is equal to the recovered product. The diverted feed is directed to the pipefeeder, and after having its pressure increased to nearly the brine pressure level, it is boosted by a smaller auxiliary pump so as to reach the same pressure output of the main pump. Both feed streams are joined and then fed to the RO unit.

During one operational cycle, the chamber is initially filled with water. Then, the chamber is pressurised and valves V1 and V3 are opened, allowing the incoming high-pressure brine to force the low pressure feed out, with increased pressure. When the chamber is filled with brine, valves V1 and V3 are closed, the chamber is depressurised and valves V2 and V4 are opened, allowing the low pressure feed to expel the low pressure brine from the chamber. Once it is full of feed water, the cycle restarts. Although it operates discontinuously with only one chamber, the output is smoothed out if three alternating chambers are used.

A minimum recovery efficiency of 96% was observed during the tests carried out in [Krumm, 1996], but this type of recovery device would mostly suit higher capacity plants (product flows greater than 2,000 m³/day), causing an average reduction of specific energy consumption by 35%. Another advantage of this recovery system is the
feasibility for implementation in multiple parallel-connected RO units.

i) Clark Pump

The Clark pump is a pressure exchanging device that uses the concentrate stream to drive a reciprocating piston, which in turn pressurizes a medium pressure (5 to 10 bar) feed. It was originally designed for use in very small systems (0.8 m³/day) but, because of its high efficiency, it has also been used in slightly bigger systems, in adapted configurations.

Similarly to some of the energy recovery devices previously described, the Clark pump has little losses associated to the recovery of the concentrate stream energy but, in contrast to them, it is primarily designed for use in small-scale systems. Because of these distinctive features, the Clark pump was chosen for further study in this work and more details on its operating and design characteristics are given in Chapter 3.

2.2. Renewable Energy Sources

There is no need to dwell on the importance of energy in the daily life of modern society and that its availability relies mostly on the existence of fossil fuels. Increasing awareness of the depletion of current sources has led to a global effort in the research and development of renewable energy technologies, such as Wind, Solar, Tidal and Geothermal energy.

This motivation for using renewable energy is even greater if stand-alone desalination applications are considered. This is because the energy required for the process is particularly expensive in the remote areas where fresh water is required. Renewable energy sources can provide a reliable energy supply alternative for water desalination. Initial cost and resource availability are the most significant limitations.

The utilisation of more established renewable energy sources, such as the sun (thermal and PV) and the wind, in stand-alone desalination systems has been widely discussed ([Petersen, 1979], [Hamester, 1981], [Alawaji, 1995], [Ehmann, 1996] and [Kiranoudis, 1997], to name a few) and even tested in the field ([Assimacopoulos, 2001] and [Kunczynski, 2003]). Additionally, tidal and geothermal sources have also been considered in [Hanafi, 1994]. Since desalination is an especially promising application that involves broad fields of study, many different solutions have been
proposed. Even if one focuses on one particular renewable source and a specific desalination method, there may still be many options available in terms of the final system configuration.

One of the most critical limiting factors to a wider implementation of renewable energy-driven desalination systems is the intermittency and unpredictability of the renewable source. Two distinct problems have been identified in the study presented in [THERMIE, 1998]. The first one is that most desalination technologies are not suitable for operation at variable power, and the second is the discontinuity of the energy supply, or even limited power availability over variable periods of time, which may cause the demand not to be met occasionally. A good example are the thermal distillation processes, in which frequent start-ups are not at all desirable. A common solution to these problems may be the use of energy storage, which can accumulate energy surpluses (long-term storage) and/or smooth out severe variations in the supply (short-term storage). On the negative side, the use of such energy storage devices results in an increase to both capital and operating costs.

Although the use of sources such as tidal and geothermal power is not yet proven, some feasibility studies have pointed out their viability based on technical aspects like power matching (supply-load energy balance) and system sizing. A summary of the technically possible combinations of renewable energy sources and desalination techniques is given in Table 2.3. Wind energy is a most attractive source in the short-term since the technology is well developed and relatively cheap, provided of course that an adequate wind resource exists. This alternative is further detailed in the next section.

Table 2.3 - Applicability of renewable energy sources to water desalination techniques

<table>
<thead>
<tr>
<th>WIND</th>
<th>SOLAR</th>
<th>TIDAL</th>
<th>GEOTHERMAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>RO</td>
<td>ME</td>
<td>RO</td>
<td>MSF</td>
</tr>
<tr>
<td>VC</td>
<td>MSF</td>
<td>VC</td>
<td>ME</td>
</tr>
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<td>FS</td>
<td>VC</td>
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<td>-</td>
<td>ED</td>
</tr>
<tr>
<td>-</td>
<td>SD</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Source: [Hanafi, 1994] and [Assimacopoulos, 2001]
2.2.1. Wind Energy

The use of wind energy for electricity generation can be divided into two main application areas. The first and foremost of these is the commercial generation of bulk electricity through grid-connected systems. This is currently a well-established application and mainly due to materials technology advances for the blades manufacturing, its capital costs have dropped to relatively low levels. Here the expression grid-connected applies to the use of large wind turbines (mainly 0.3 to 2 MW rated power), usually installed in large groups known as wind farms.

The second category, which involves small/medium wind turbines (up to 200 kW), is the electricity generation within stand-alone systems (including local grids, powered by diesel gensets). In contrast to grid connected systems, these are sometimes built to be used in sites where maintenance may be sporadic and technical assistance unavailable, thus greater robustness is required in their design incurring in higher capital costs than otherwise. Grid connected turbines are used as an additional supply, complementing the conventional base load power system. In stand-alone systems, the wind can often be the sole source of energy and this should be fully taken into consideration during the design stage.

In stand-alone systems, another sub-division can be made regarding the rating of the turbines used, i.e. medium and small sized turbines. The former (from about 20 up to 200 kW) is mostly used in hybrid wind-diesel systems. In general terms, their configuration do not differ much from the grid connected turbines, apart from having a more complex control system to avoid grid stability problems (depending on wind penetration). In the other field of application (small systems, up to 20 kW), the turbines generally present a different design from the previous ones. This is because they generally must be able to tolerate unattended operation for long periods of time and be self-protective under extreme weather conditions, being sometimes employed as the only energy source.
2.2.1.1. Stand-alone Wind Energy Conversion Systems

Due to the inherently random characteristic of the wind, certain key aspects must be attended to in the design process of stand-alone systems. Besides the wind resource potential, the nature of the electrical load should be given careful consideration, in particular whether disconnection is acceptable, and if so, for how long. Remote communication stations that must operate uninterruptedly can exemplify this. Since load matching is imperative, system oversizing may be unavoidable.

This consideration relates to the presence and sizing of any storage system that may be included in the system. Energy storage plays an essential role in determining system performance, as well as the aforementioned influence on capital and maintenance costs.

Once the basic design questions have been answered, i.e. the layout of the system has been defined, the technical aspects should be addressed. These aspects include turbine rating, typical wind speeds (cut-in, cut-out and rated speeds), electronic interfaces specification (power converters), control strategies, among others.

In small stand-alone systems, passive protection mechanisms (e.g. pitching, furling or conning) are generally the best alternative to limit power output from the wind turbine and the control system used is generally simple and rugged due to the low maintenance requirements.

In terms of electrical generators, the most common option is the use of permanent magnet machines. This configuration presents advantages in respect of simplicity and compactness, but becomes less economically viable as power levels increase. Self-excited induction machines are robust and cheap but generally require the use of gearboxes, which may incur higher maintenance. Furthermore, an external excitation source (either a capacitor bank or a static VAR compensator) must be provided, which increases system complexity.

Finally, for the design of a small stand-alone system, a key challenge is to find a good compromise between reliability and system complexity that meets the economical constraints. This is not a simple matter and it will mostly depend on the type of load and the local resources.
2.3. **Wind Energy & RO Integration**

As discussed in Section 2.1.2, Reverse Osmosis (RO) is now a well-established technology for the desalination of water and in particular seawater. Nevertheless, the use of RO in small stand-alone systems (in the range of a few cubic meters per day) is still an area of developing technology. Despite the many advances in RO membrane technology through the years, such as higher rejection rates and improved tolerance to pressure fluctuations, considerable attention should be given to their operational conditions. It is believed that higher efficiencies can be obtained when operating at certain flow and pressure conditions. Since the membrane is a very sensitive component of the system, exposure to extreme pressures or flow rates may cause reduction of lifetime or even in permanent damage.

The first experimental work addressing the unsteady operation of RO modules was carried out by [Lising, 1972]. Here, the membrane performance under variable operating conditions is studied as a feasibility analysis for connection to wind turbines (mechanical shaft power). An experimental set-up was implemented using a variable speed motor connected to a positive displacement pump, to simulate the variable input of the wind turbine. A sinusoidal pattern flow rate was imposed to a brackish water (3.5 kg/m³) solution and this operational condition (variable sinusoidal feed flow at constant pressure) was used in two different tests. In the tests, the flow variation range was nearly 2:1 between the minimum and maximum levels. The period for the fluctuation to occur was set to 90 seconds per cycle and the tests were carried out for a period of 7 days. Results showed that for this level of variation, the membranes behave as in steady state, provided the feed flow is not reduced to zero. An analysis of product water flow rate and concentration showed little difference between both tests. Membrane properties such as rejection and permeability were not affected, which indicated the technical viability of implementing a wind powered system, provided that the zero-flow problem could be circumvented.

Another important contribution dealing with variable flow and pressure operation of RO membranes, which looks at the feasibility of wind-powered systems, is that of [Feron, 1985]. In this work, an operational window is established (Figure 2.12) determining the operational parameter variation to which a membrane should be safely
submitted. The four limits that define this window are:

a) Maximum Feed Pressure - determined by the membrane mechanical resistance;

b) Maximum Brine Flow Rate - should not be exceeded to avoid membrane deterioration;

c) Minimum Brine Flow Rate - it should be observed to avoid precipitation and consequent membrane fouling;

d) Maximum Product Concentration - salt concentration in the permeate water is inversely proportional to the difference between the applied pressure and the osmotic pressure gradient across the membrane. If the applied pressure is less than a determined value, permeate concentration will exceed the limit of palatability of potable water (up to 1.0 kg/m³ may be considered acceptable, as suggested by the World Health Organization).

![Diagram showing the typical reverse osmosis membrane operational window](image)

**Figure 2.12 - Typical Reverse Osmosis membrane operational window**

2.3.1. Wind-RO System Configuration

Considering the number of possible configurations a classification of the different wind powered reverse osmosis systems found in the literature has been made. This was
based on some of the points previously discussed: the existence of an alternative electrical supply (weak grid or diesel generator); the matching of the available wind energy to the load; and the operational characteristic of RO membranes.

2.3.1.1. Systems with back up (diesel/grid)

In these systems, an additional energy source is provided (a diesel-powered generator or even the local grid) so that the power supplied to the RO is constant. The external source complements the power generated from the wind turbine or even supplies it all to match the RO unit power consumption.

The main benefit of these systems, as in any hybrid wind-diesel configuration is the achievement of fuel savings, which may increase the generator availability and reduce overall energy costs.

On the other hand, problems such as fuel shortages, diesel generator maintenance stops, interruptions or power cuts in the supply, may lead to unavailability of the RO system since it cannot be powered by the wind turbine alone.

2.3.1.2. Systems without back up

Systems without an external energy source can be divided in two categories, with emphasis on the RO unit operation: systems which run under approximately constant operating conditions, and those that experience variable operational conditions.

a) Near constant operating conditions

This first type of operation can be implemented by three different means: on/off switching of the RO units; usage of storage devices, such as batteries or pressure vessels; or de-rating the wind turbine. In all three cases, an attempt is made to supply the individual RO modules with approximately constant power.

• Storage devices

In this strategy, the storage devices are employed to accumulate energy surplus when the power generated by the wind turbine is greater than the load demand from the desalination unit. This surplus would then be used later when the generated power is insufficient to meet the load demand.
One common way of storing the surplus energy is by using batteries ([Infield, 1997] and [Hopkins, 1996]). In this case, the relation between operational pressure, storage sizing and average wind speed should be considered in the design stage. In addition, capital and maintenance costs should be carefully assessed. A disadvantage of this approach to the system design is the rating of the energy storage system, as this can make it economically unattractive at higher power levels due to the sizing of the battery bank.

Another possibility for implementing energy storage is the use of pressure vessels. In [Robinson, 1992] this alternative is analysed for the implementation of a low-pressure brackish water RO system. In this case, the pressurised container works as a surge suppressor, being a trade off between efficiency (existence of over-pressure relief valve) and the cost-reliability of the whole system. Another advantage is the viability of implementation using a multi-blade wind turbine for mechanical pumping (150 W) at low wind speeds, being of simple construction and easy maintenance. This is clearly a low power system, suitable for the production of small amounts of water (up to 0.4 m$^3$/d), at low pressures (5 to 15 bar), being insufficient for seawater desalination.

• RO unit switching

This strategy is based on the use of a higher power wind turbine connected to multiple smaller RO units. The power control is achieved by switching the units on and off so as to match the demand to the total power generated instantaneously by the turbine. There is no limitation concerning the system power rating, and this approach is feasible to power levels of hundreds of kilowatts.

Although frequent cycling of RO units is not usually recommended, this problem can be overcome by implementing different configurations. In [Rahal, 1997] and [Gonzalez, 1997], a higher power wind turbine operating at near constant speed (generation management) is connected to many equally smaller RO units switching on/off (load management). To smooth out the fluctuations, short-term storage (a flywheel in this instance) is used. Varying the pitch angle of the wind turbine blades controls the generated power.

Another possibility ([Neris, 1995]), suitable for smaller systems is the switching of few (two/three) desalination units with distinct power levels. Additionally, some
auxiliary loads (such as pumping/heating and dump loads) can be implemented to absorb any power surplus, keeping the system voltage and frequency constant.

- Wind turbine de-rating

This alternative consists of making use of the flat end of a pitch controlled wind turbine power curve to operate the RO unit at approximately constant power ([Feron, 1985]). An implication of this configuration is that, since the turbine rated power is only achieved at high wind speeds, it would have to be de-rated by changing the settings of the pitching mechanism. This will cause the generated power to be flattened at lower wind speeds and consequently to have lower values. Therefore, the original rating of the turbine rotor should be considerably higher than the RO unit rated power making the system more expensive.

**b) Variable operating conditions**

In contrast to systems that run at constant conditions, another strategy is based on the establishment of certain operational limits ([Feron, 1985], [Smulders, 1984]), as previously illustrated in Figure 2.12. This means that, based on the input power to the RO unit (flow times pressure), a control strategy is determined which adjusts the operating point of the unit so that it lies within the allowed region (the operational limits of the RO unit).

By doing this, an attempt is made to operate the system autonomously over a wider power range, without the need to use a back up unit or even storage devices. One aspect that should be emphasised is that very little is known about the consequences of variable operation of RO membranes. As previously stated, it is recognised that mechanical fatigue can occur and that lifetime of the elements may be shortened or their performance impaired.
Despite the fact that the final configuration of the system had not yet been decided at the beginning of the work, a provisional configuration was outlined and used as a guideline for the development of the whole system model. The models of individual components were gradually replaced by detailed models obtained from laboratory testing. This chapter presents a description of the system components and their mathematical models. Some details of the experimental setup related to the modelling tests of specific components are also described here. A more thorough description of the hardware/software used in the laboratory tests is given in Chapter 6.

3.1. Introduction

One of the main uncertainties in the system configuration design was related to the energy recovery mechanism, which would ultimately define the layout of the components within the system.

In light of the desired characteristics of the final system, i.e. high-efficiency, robustness and ability to operate from a variable power supply, specific components were chosen to meet these requirements. Another important aspect of the component selection was their compatibility with seawater, which invariably led to increased component costs.
A preliminary model of the complete system was built from standard models of the components found in the literature and from manufacturers data sheets. The system modelling work was carried out using the Matlab/Simulink environment, which allowed great flexibility in the programming due to its inherent modularity. Individual component models were developed in separate subsystems that could be improved, replaced or moved within the system model. An outline of a particular model of the complete system can be seen in Figure 3.1.

![Figure 3.1 - Outline of a complete system model in Simulink](image)

Following the preliminary modelling of the system, a phase of detailed laboratory testing of the components was carried out. Specific test procedures were designed for each component, mostly aiming at obtaining the steady-state input/output characteristics of the components over a wide range of operating conditions, as was the case of the wind turbine, pumps and RO modules. Additionally, simple dynamic performance tests were performed on the RO membranes to verify the product output flow and concentration behaviour as functions of variations in the input feed pressure and flow. The inertia of the wind turbine was taken from [Lloyd, 1998] and the motors inertia and equivalent circuit parameters were taken from the manufacturer's datasheets.

The data obtained from the component characterisation work was then used to create refined empirical models by fitting the data to polynomial functions, using the least-squares technique described in the next section. The old rig was used as a testbed for the individual component modelling stage. It was extensively modified, particularly
the plumbing, throughout this stage to accommodate the different test requirements.

The system configuration can be divided into five major components, which will be analysed in greater detail in the next sections. These components are:

- wind turbine;
- variable speed drives (inverter and induction motor);
- pumps;
- RO membrane modules;
- energy recovery device.

Although the control strategy constitutes a subsystem within the model, it will not be considered as a component at this stage and will be presented and discussed in Chapter 4. More of the specifications and further operational/construction parameters of the components are given in Appendix A.

3.2. Component Modelling and Data Fitting

The initial strategy for processing the experimental data from a number of the components, as part of the determination of the submodels, was to use the Matlab's `polyfit` function. This routine is very useful for finding complex polynomial relationships between two variables, but that is also its main limitation: it only works for two variables (single-input-single-output). Since many of the components in the system have variables that depend on more than one input variable, an alternative procedure was sought.

3.2.1. Modelling Procedure

It was noticed that even for the single-input-single-output case, the adoption of a more flexible modelling procedure - one that was not based solely on polynomial relationships - could be beneficial in reducing the curve fitting errors.

A more refined modelling procedure was applied instead, which considered the influence of more than one variable on any other variable. This procedure, referred to as "Linear-in-the-parameters Multiple Regression", is described in [Mathworks, 1998]. It was also implemented in Matlab and consists of a data regression routine, which fitted
the data points to a predefined non-linear relationship, through least-squares error minimization. It uses Matlab's `backslash` operator to calculate the coefficients of a predefined non-linear model, as explained below.

For instance, to calculate the relationship between an output variable, $Y_1$, and two input variables, $X_1$ and $X_2$, the following non-linear model (linear-in-the-parameters) can be defined:

$$Y_1(X_1, X_2) = k_1 + k_2 \cdot X_1 + k_3 \cdot X_2 + k_4 \cdot X_1 \cdot X_2$$

(3.1)

where,

$Y_1$ is the variable wanted as the model output;

$X_1$ and $X_2$ are the variables used as the model inputs.

A new variable ($X_3 = X_1 \times X_2$) was generated from the original experimental data set, equal to the desired non-linear term, in the equation above, producing the target model matrix $X = [I \ X_1 \ X_2 \ X_3]$. Matlab would than calculate the coefficients vector $k = [k_1 \ k_2 \ k_3 \ k_4]'$, that gave the minimum error between both sides of Equation 3.1 ($k = X\,y$). The RMS error between the original data and the model fit would be calculated, and if needed, the target non-linear relationship adjusted, giving a different target equation.

An rms error of less than 5% was used as a guideline for the suitability of the data fitting. This tolerance was obtained for most of the derived models with exception of the membranes (product flow, 5.4% and salt passage, 5.3%) and the wind turbine (torque coefficient, 5.8%). In the case of the membranes, this was not considered a problem since the manufacturer's datasheets suggest a possible variation of $\pm15\%$ in each individual element performance from the expected quoted values. For the wind turbine, it was also an acceptable error margin, considering the possible errors introduced during its modelling procedure (see Section 3.3.4).

Another guideline used was the "overdetermined system" warning that Matlab would produce if the data matrix were "rank deficient", i.e. if it did not have linearly
independent columns. In this case, the least squares solution to $Xk=Y$ (following the nomenclature used above) was not unique and the backslash operator, $X\backslash Y$, issued a warning. This indicated that the established non-linear model was not suitable and a different target matrix $X$ would be chosen.

This curve-fitting procedure of defining a non-linear relationship based on expected physical relationships between variables and refinement of the target function based on previous results was used for many of the components, and through example is explained in greater detail in the section describing the plunger pump (Section 3.5.1).

3.2.2. Components Models

The purpose of modelling the components individually was to characterise their behaviour within the expected range of operation of each component during the complete system operation. This means that all components were tested and the models established for operation within the range tested (interpolation). The models are not guaranteed to work outside the tested range (extrapolation) and although this approach may limit the applicability of the developed models in other applications, it was found to be satisfactory in the scope of this work.

This also means that the complete system model may not be representative in all conditions the system is expected to operate, e.g. when there is not enough pressure in the feed stream to overcome the osmotic pressure, say in low wind speed conditions, there will be no permeate flow and the product concentration (amount of salt per amount of product) would mathematically equal infinity. To overcome this problem a discontinuity was implemented which allowed the concentration to be calculated only when the permeate flow was greater than a threshold value (0.001 l/s).

An overview of the guidelines and assumptions used in the development of the components models is presented below. Regarding the wind turbine, the objective was to establish a simple equivalent static $Cq = f(\lambda, \omega)$ surface (due to the conning of the rotor, $Cq$ vs. $\lambda$ curves were obtained at different rotor speeds), which could be representative of the average rotor shaft torque driving the turbine generator at each simulation time-step (for a given wind and rotor speed). Dynamic interactions and detailed aeroelastic modelling of the turbine rotor were thus not considered appropriate. Again for
simplicity, a simple wind speed time series was derived, using a 1st order autocorrelation function, which did not take phenomena such as rotational sampling of turbulence or tower shadow into consideration. The $C_q$ vs. $\lambda$ curves were obtained by using a high sampling rate (above 1 kHz) and consequent averaging the data. The procedure is described in Section 3.3.4.

With respect to the variable speed drives (inverters and induction motors), the initial plan was to build all the power electronics to be used in the system. Therefore, more complex models for the motors and inverters were developed and used from the very beginning. At a later stage, it was decided that standard off-the-shelf inverters were to be used instead as these were readily available and cheap, but the models already developed and known to work were kept, as presented in Section 3.4. It is understood that the complexity of these models is not consistent with the levels of complexity used in other system components and in hindsight simpler models could have been used without compromising the system model performance. Such simplifications would encompass the use of a 3rd order model for the induction motors and a simple V/f model for the inverters.

The pumps used in the system were positive displacement pumps and hence, it was expected that the flow rate and developed pressure would be directly related to the rotational speed and applied shaft torque, respectively. The water was considered as incompressible and therefore, the pumps were modelled as direct input-output relationships. As they were connected to the induction motors, the flow rate output followed the dynamics of the motors' speeds. Similarly, the energy recovery devices used were also positive displacement devices and were also modelled as functions of their steady-state input-output characteristics.

The RO membrane models and parameters used initially were taken from the literature ([Slater, 1985], [Slater, 1992] and [DOW, 1996]). At a later stage, empirical models were derived using experimental data as described in Section 3.6. The dependence between the variables observed in the models described in the literature was used as a guideline in the development of the empirical model functions.
3.3. Wind Turbine

The wind turbine used in the system has a rated power output of 2.5kW and was manufactured by Proven Engineering, from Scotland (Figure 3.2). It is a three-bladed horizontal-axis downwind turbine with power limiting at high wind speeds achieved passively, through blade coning.

The turbine is installed at the Loughborough University campus, next to the CREST building. Although this is a practical location regarding electrical connection to and easy access from facilities available nearby within the CREST workshop, it is considered a poor site in terms of the surrounding features (tall trees and buildings), which greatly influence the local wind characteristic, and in particular result in high levels of turbulence. This will be returned to later as such high turbulence intensity increases the demands on the system design and control.

![Proven wind turbine installed at CREST](image)

*Figure 3.2 - Proven wind turbine installed at CREST*

The turbine was originally a 2.2kW pre-production prototype, and had been previously modelled in [Lloyd, 1998]. Since then, one of the blades had suffered a mechanical failure, which caused the complete set of blades to be replaced by the ones used in the current model (2.5kW). Also, the maximum output voltage (rectified DC) in the old model was around 240V (in no-load conditions). As this value was not suitable for direct connection to standard off the shelf inverters (an important aspect of component selection was that they should be commercially available wherever
possible), the windings were replaced by high voltage ones (600V rectified DC). Significant changes in the stator frame had to be made in order to accommodate the new stator windings, as they had new dimensions and were attached to the stator frame in a different way. Due to all these changes, the machine had to be extensively modelled once again, and despite not being supplied by the manufacturer as a completed product, the performance data obtained were in excellent agreement with the data provided by the manufacturer for their current model. Also, it must be added that both the turbine inertia (rotor plus generator) and the shaft friction torque characteristic were not obtained experimentally. The values used were taken from [Lloyd, 1998].

In the following sections, all the variables related to the wind turbine (either the rotor or the generator) are followed by the subscript “WT”. The only exception are the torques, which are represented by $T_{WTR}$ and $T_{WTG}$, for the rotor mechanical torque and generator electromagnetic torque, respectively.

3.3.1. Wind Speed Time-series

As seen in Figure 3.1, the wind speed time series is an important component of the modelling, on which all the results will depend. It is the input to the system and a good representation of the wind behaviour is essential to properly characterise the resulting system dynamics.

There are different techniques available to simulate wind turbulence and, being it a stochastic phenomenon, a useful approach is to refer to its spectral characteristics. This can be very demanding in terms of computational costs, thus a simple technique to implement was sought.

A technique initially considered was that developed by [Baran, 1995] based on the relationship between the spectral distribution and the autocorrelation function. A schematic representation of this approach is shown in Figure 3.3.
Initially, a spectral function is chosen and Fourier transformed to obtain the target autocorrelation function. The Kaimal power spectrum was chosen ([Kaimal, 1973]) by Baran, as it is known to provide a good description of the sort of wind sites under consideration here. The procedure used in [Baran, 1995] is outlined below.

The Kaimal power spectrum can be represented by Equation 3.2:

\[
\frac{\omega \cdot S(\omega)}{\sigma^2} = \frac{0.164 \cdot \left( \frac{f}{f_0} \right)}{1 + 0.164 \cdot \left( \frac{f}{f_0} \right)^{3/2}}
\]  

(3.2)

where
\( \omega \) is the cyclic frequency;
\( S(\omega) \) is the power spectral density;
\( \sigma \) is the standard deviation of the wind speed;
\( f = \omega \cdot z / U_{ave} \), and
\( f_0 = 0.041 \cdot z / L \).

In the latter two equations, \( z \) is the height above the ground, \( U_{ave} \) is the mean wind speed, and \( L \) is the turbulence length scale.
From a $p$th-order autoregressive model of the wind speed (Equation 3.3), the autocorrelation function (Yule-Walker equation) is derived (Equation 3.4).

\[ U_t = \Phi_1 U_{t-1} + \Phi_2 U_{t-2} + \cdots + \Phi_p U_{t-p} + Z_t \]  

(3.3)

\[ \rho_k = \Phi_1 \rho_{k-1} + \Phi_2 \rho_{k-2} + \cdots + \Phi_p \rho_{k-p} \]  

(3.4)

In the equations above, $U_t$ is the wind speed at time $t$, $\Phi_i$ are the autoregressive parameters, $Z_t$ is a random term (white noise) and $\rho_k$ is the $p$th-order autocorrelation function.

Through an optimisation routine based on the least-squares method, the estimated autocorrelation function is then fitted to the target function and the parameters $\Phi_i$ are found. These parameters are used in the autoregressive model to generate the wind speed time series. The computational cost of this iteration is minimal, since it is the autocorrelation function that is fitted, instead of the more computationally demanding power spectrum as used in most of the alternative techniques.

Finally, it was decided that the complexity of the 4th-order autocorrelation function was unnecessary. Instead, a simple 1st-order autocorrelation function, with the autocorrelation at unit lag, or $\Phi_1$ in the previous notation, was used to model the wind time series. Therefore, Equation 3.3 can be rewritten as:

\[ U_t = \Phi_1 U_{t-1} + Z_t \]  

(3.5)

where $\Phi_1 = 0.9873$. This means that the autocorrelation function is exponential and the corresponding spectrum is known as the Dryden spectrum (which can be derived by analytically carrying out the Fourier transform). The time-lag (sampling interval) was chosen to be 0.1s and the value of $\rho = 0.9873$ chosen to represent typical wind turbulence.
In order to implement this routine, a script program was developed in Matlab. The program is run off-line and the output file is used as the input to the system model. Three examples of the time series obtained from the program, using average speeds of 8, 10 and 12 m/s are shown in Figure 3.4. For this set of results a standard deviation equal to the unity was chosen, therefore each of the series presents different turbulence intensity (ratio between the standard deviation and average).

### 3.3.2. Wind Turbine Rotor

In the Proven wind turbine, the rotor blades are connected to the hub by flexible hinges and a set of springs. These allow the blades to cone away from the tower as the thrust on them increases (at high wind speeds) and move towards the plane of the tower as the centrifugal force increases (at high rotational speeds). The resulting conning action will effectively reduce the power coefficient ($C_p$) of the rotor in high winds at high rotor speeds, thus limiting the power transferred to the generator.

Due to the conning of the rotor blades, the rotor shaft power output cannot be represented solely as a function of the wind speed. Indeed, the aerodynamic torque model employed is based on a polynomial fitting of a 2-dimensional data set. The wind
speed, $U_w$, and the turbine rotational speed, $\omega_{WT}$, are given as the inputs and the torque coefficient, $C_q$, is taken as the output. The data set was obtained from experimental tests and the procedure used is described in Section 3.3.4. The torque coefficient is then used to calculate the torque developed by the wind turbine rotor using Equation 3.6.

$$T_{WTR} = \frac{1}{2} \cdot C_q \cdot \rho \cdot A_{WT} \cdot R_{WT} \cdot U_w^2$$ (3.6)

where

$\rho$ is the air density (1.225 kg/m$^3$);
$R_{WT}$ is the turbine rotor radius, $R_{WT} = 1.75$ m;
$A_{WT}$ is the turbine rotor swept area, $\pi \times R_{WT}^2$ (m$^2$).

3.3.3. Permanent Magnet Synchronous Generator

The rotor shaft is directly connected (no gears) to a 3-phase axial-flux permanent magnet synchronous generator (PMSG), which will generate an output voltage proportional to its rotational speed ([Miller, 1989]). At maximum speed (just over 40 rad/s), the 8-pole generator provides an open circuit line-neutral voltage ($V_{LN}$) slightly higher than $220V_{rms}$ (Figure 3.5) at a frequency of approximately 26 Hz. As the frequency is lower than the supply frequency on which electrical motors are designed to operate, some form of power conditioning is required. Therefore, the generator output is rectified by an uncontrolled bridge rectifier, providing a DC bus with voltage proportional to the generator speed (Figure 3.5).
3.3.4. Modelling Procedure

In order to obtain a satisfactory steady-state model of the wind turbine, it was desirable that the inertial component (rotor speed variation) of its power output was minimised, i.e. that the output power directly represented the instantaneous power available in the wind. Thus, a continuously variable controlled load, was built for constant speed operation of the turbine. This was implemented by means of a Buck converter and a resistive load connected to the DC bus (Figure 3.6).

This controlled load is also used in the system final configuration as a DC voltage limiter. Connected to the DC bus, in parallel with the inverters, the converter switches in the resistive load to prevent the DC voltage at the inverters input to rise above 400V.

A detailed description of the controlled load / voltage limiter, with diagrams of the gating circuitry and the converter itself, is presented in Appendix C.

The closed-loop control of the wind turbine speed is digitally implemented in a PC. The speed is calculated from the stator voltage waveform frequency, compared to a
constant speed reference value and the resulting error is processed by a PI controller. The controller outputs a reference value that is then produced as a voltage signal at one of the analog outputs of the data acquisition board. This analog signal is compared to a saw-tooth waveform in the gating circuit PCB, generating a PWM gate signal to the IGBT switches of the Buck converter (switching frequency of 1.35 kHz). The gating circuit PCB, which was kindly supplied by Marlec Engineering Co Ltd, was originally designed as a battery charging controller. Some minor modifications were made in order to adapt it to drive the Buck converter for both applications: as a controlled load and as an overvoltage protection device.

![Figure 3.6 - Controlled load diagram (Buck converter)](image)

3.3.4.1. Experimental Setup and Data Processing

A set of operating speeds from 16 to 38 rad/s was chosen and the wind turbine was left to run at each of the speeds for a period of time long enough to experience a reasonable range of wind speeds. The data was sampled at 4 kHz and the control loop was executed - and its output updated - with the data averaged over every 200 ms. Although the hardware allowed faster actuation, this was not implemented because of a limitation in the rotational speed detection algorithm. As it needed at least two complete cycles of the stator voltage waveform for precise frequency estimation, a faster execution rate (more than 5 times a second) would imply in the reduction of the speed measurement range due to the increase of the minimum speed detected. At 200 ms, rotational speeds from around 15 rad/s (10 Hz electrical, considering the stator winding...
has 4 pairs of poles) could be measured, which was enough for proper characterisation of the turbine.

All the data collected was then processed for use in the modelling stage. The processing was implemented in Matlab and consisted of three stages:

a) filtering out the data which was not representative of the conditions desired. The main objective of these tests was to operate the wind turbine at a constant given speed, and verify its power output for different wind speeds. Therefore, some conditions were established to assure data representativeness, such as: points must have an absolute speed error no greater than 1 rad/s; the variation of the speed from the previous point must be less than 0.1 rad/s; and the duty cycle of the Buck converter must be greater than 0.01 (just to indicate power flow).

b) calculating other system variables derived from the variables acquired. Variables such as powers, torques, losses, etc. were derived from the tests readings. Also, the wind speed reading is corrected at this stage (see Section 3.3.4.2).

Despite the data filtering, some variation of rotational speed is still present in the remaining data. To account for the contribution of the rotor inertia in the power output, the inertial component was deducted from the power output of the wind turbine at each point. This was achieved by calculating the difference in the rotational kinetic energy at two consecutive points and dividing by the sampling period (Equation 3.7):

\[ P_{JWT}(T) = \frac{1}{2} \cdot J_{wr} \left[ \omega_{WT}^2(T) - \omega_{WT}^2(T - \Delta T) \right] \]

(3.7)

where,

- \( P_{JWT}(T) \) is the inertial component of the power output at sampling time \( T \) (W);
- \( J_{wr} \) is the rotor inertia (\( J_{wr} = 9 \text{ kg.m}^2 \));
- \( \omega_{WT}(T) \) is the wind turbine rotational speed at sampling time \( T \) (rad/s);
- \( \Delta T \) is the interval between consecutive updates of the controller (\( \Delta T = 200 \text{ ms} \)).

c) binning/averaging variables. Variables pertaining to one set of speeds were grouped (and averaged) in wind speed bins, so that the resulting set of data is an
averaged performance as a function of rotational speeds and wind speed. This procedure is similar to that adopted by [Lloyd, 1998].

One aspect related to the constant speed control strategy that should be stressed here is the inherent non-linear characteristic of the process. Constant speed operation consists in transferring any excess rotational kinetic energy the rotor may have to a fixed resistance, via the generator and through the Buck converter. Obviously, if the speed falls below the setpoint speed, nothing can be done but to reduce the load as fast as possible to allow the wind to accelerate the turbine again, which renders the system uncontrollable in this situation.

The instantaneous power dissipated on the Buck converter resistive load can be written as:

\[ P_{RL} = \frac{V_{RL}^2}{R_L} = \frac{(D_{Buck} \cdot V_{DC})^2}{R_L} \]  \hspace{1cm} (3.8)

where

- \( P_{RL} \) is the resistive load power (W);
- \( V_{RL} \) is the resistive load voltage (V);
- \( R_L \) is the resistive load (\( R_L = 36 \Omega \));
- \( D_{Buck} \) is the Buck converter duty cycle, and
- \( V_{DC} \) is the DC bus voltage, or the input voltage to the converter (V).

The power dissipated on the resistance is proportional to the square of the product of the DC bus voltage and the converter duty cycle. The DC bus voltage is proportional to the generator speed (Figure 2.5), and the converter duty cycle is proportional to the PI controller output voltage, \( V_{cont} \) (Figure 3.7).
Therefore, a given speed error $\Delta \omega_{WT}$ (regardless the speed setpoint) corresponds to a $V_{\text{cont}}$ value, which in turn represents a given duty cycle to the inverter. And, as seen in Equation 3.8, the power transferred to the load will be proportional to the square of the speed.

This means that, for the same speed error, the wind turbine would slow down much faster (more power transferred) at higher rotational speeds than in lower speeds. The solution used to circumvent this problem was to make the proportional gain of the PI controller inversely proportional to the reference speed. By doing this, an attempt is made to keep the numerator of Equation 3.8 less dependent on the speed setpoint. The values used in the implementation of the speed regulator PI controller were found empirically and are given in Appendix C.

The wind turbine characterisation tests were carried out along several days, totalling almost ten hours worth of data. For presentation purposes the data was concatenated in one long sequence, which is displayed below.

Figure 3.8-a shows the actual rotational speed of the turbine and the reference setpoint applied to the controller, which was varied from 16 to 38 rad/s. The wind speed
at the moment of the tests is shown in Figure 3.8-b. A clear relationship between the wind speed (or the wind turbulence) and the deviation of the rotational speed from the setpoint value can be noticed, although this gets less pronounced at higher speeds. It is possible that the use of a different resistance, of lower value, in the low speed tests would have made the system more controllable under wind gusts, but this was not found to be necessary at the time.

![Graph showing speed and wind speed](image)

*Figure 3.8 - a) Measured rotational speed and reference, b) Wind speed.*

The speed error (setpoint minus actual value) is displayed in Figure 3.9, as a percentage of the reference speed. As previously explained, due to the very nature of the system, nothing can be done as the rotational speed fell below the reference (positive errors) other than remove the load as quick as possible. For this reason, another adjustment was made to the controller. Whenever a positive error was present (actual speed below reference), the proportional gain would have its value doubled, so that the turbine could be unloaded faster and allowed to speed up again.

Although not formally examined (only verified empirically), the changes implemented in the speed controller were found to be useful in improving its performance and appropriate for a highly non-linear system in nature.
3.3.4.2. Wind Speed Correction

One of the problems often encountered in wind turbine modelling and performance assessment is the correct measurement of the incident wind speed. Proper positioning of the anemometer(s) and subsequent treatment of the data, if need be, is essential for reliable characterisation.

As seen in Figure 3.2, the anemometer is placed right in front of the rotor, at a distance of about 1.2 times the rotor radius. According to the IEA recommendations, a distance of at least 4 times the radius is required between the turbine rotor and the anemometer ([IEA, 1990]). This spacing of 1.2 times the radius was not enough to completely avoid the upwind slow down effect that the rotor has on the wind stream, thus some form of correction of the data had to be implemented.

Further to the wind speed correction, the prevailing wind direction is also adjusted using the reading provided by the wind vane, placed next to the anemometer, in front of the turbine rotor.

The wind speed correction procedure adopted was the same used in [Lloyd, 1998]. It is originally based on a piece of software developed by Mr. David Sharpe (CREST, UK) and makes use of blade element momentum theory in order to find the relationship between the axial induction factor (AIF) and the tip speed ratio (λ). As this
code was not available at the time of this work, a printed graph displaying this relationship was digitalised and a polynomial function was fitted to it and used in the wind speed correction code.

![Graph](image)

Figure 3.10 - Axial induction factor as a function of the tip speed ratio

As seen in Figure 3.10, the solid line represents the original digitalised curve and the markers are the values found for the axial induction factor by the polynomial function described in Equation 3.9:

\[
AIF = 0.159 - 2.407 \cdot \lambda^{0.5} + 15.213 \cdot \lambda - 51.258 \cdot \lambda^{1.5} \\
+ 95.256 \cdot \lambda^2 - 81.635 \cdot \lambda^{2.5} - 17.273 \cdot \lambda^3 \\
+ 101.399 \cdot \lambda^{3.5} - 81.874 \cdot \lambda^4 + 22.410 \cdot \lambda^{4.5} + 0.050 \cdot \lambda^5
\]  

(3.9)

where \( \lambda \) is the tip speed ratio of the turbine, \( \lambda = \frac{\omega_{WT} \cdot R_{WT}}{U_{wm}} \), calculated using the original uncorrected measured wind speed data \( (U_{wm}) \).

Having found the axial induction factor, the velocity deficit upstream from the rotor can be found applying \( A_{up} = AIF \cdot (1 - \frac{\gamma}{\sqrt{1 + \gamma^2}}) \), where \( \gamma = \frac{R_{up}}{R_{WT}} \), \( R_{up} \) being the
distance of the anemometer upstream from the rotor and $R_{WT}$, the turbine rotor radius. The corrected wind speed is then calculated using:

$$U_w = \frac{U_{wm}}{1 - A_{up}}$$

(3.10)

3.3.4.3. Experimental Results

Having gathered and filtered the data for the constant speed tests, the variables were then processed to generate the wind turbine model. The mechanical power of the rotor is depicted in Figure 3.11 as a function of the tip speed ratio for each of the rotational speeds tested. The turbine rotor shaft mechanical power was calculated by adding the measured electrical power generated to the estimated losses of the turbine, both electrical (stator resistance) and mechanical (shaft friction).

![Graph showing rotor mechanical power as a function of the tip speed ratio at different speeds]

*Figure 3.11 - Rotor mechanical power as a function of the tip speed ratio at different speeds*

The main objective of the turbine modelling was to produce a reliable $C_q$ function in respect to the wind speed and rotational speed of the turbine, $C_q = f(U_w, \omega_{WT})$. The best way of analysing the results was to plot the “$C_q$ vs $\lambda$” curves for the different speeds, due to their familiar shape. This was also because the “$C_q$ vs $\lambda$” curve peaks
slightly to the left (lower $\lambda$), when compared to the respective "$C_p$ vs $\lambda$", and therefore a reasonable $C_q$ curve up to its peak would provide a clear $C_p$ curve, including points to the left of the peak.

The interest in the peaks of the $C_p$ curves is clearly due to the fact that at these points the energy conversion is most efficient, i.e. the wind turbine rotor is able to extract the most power for a given wind speed. Knowledge of these points is paramount for the control strategy as it will be shown in the next section.

During the data processing stage, it was noticed that although the $C_q$ curves were well defined at the high speed end, they were not so clear at the lower lower speeds, specifically at the lower tip speed ratios (low speeds, high winds). This was possibly due to the fact that the constant speed controller was not so effective in these conditions, as previously explained. In order to provide a better approximation, the power curves were modelled as polynomials as a function of the wind speed, for each of the rotational speeds tested. These new estimated power curves were then used to generate the $C_q$ points used in the modelling.

![Figure 3.12 - Target (black) and modelled (gray) $C_q$ curves as a function of the tip speed ratio at different speeds](image)

The modelling procedure used was the one described at the beginning of this chapter, in which two or more variables are used (tip speed ratio and rotational speed) to
define a relationship function to an output variable \((C_q)\). Equation 3.11 shows the relationship used and Figure 3.12 displays the matching between the modelled and the target \(C_q\) curves.

\[
C_q = -43.82 - 15.05 \cdot \lambda^{-1} + 41.16 \cdot \lambda^{-0.5} + 22.16 \cdot \lambda^{0.5} - 5.45 \cdot \lambda + 0.51 \cdot \lambda^{1.5} \\
+ 6.99 \cdot \omega_{WT}^{-1} + 0.018 \cdot \omega_{WT} - 0.0003 \cdot \omega_{WT}^2 \\
- 10.76 \cdot (\lambda \cdot \omega_{WT})^{-1} + 0.0006 \cdot \lambda \cdot \omega_{WT} + 4.31 \cdot (\lambda \cdot \omega_{WT}^2)^{-1}
\]  

(3.11)

Having defined a function that provided a good match at the speeds tested, the final \(C_q\) function used in the modelling could be visualised within the operating range with a smaller step between the variables. This \(C_q\) surface is plotted in Figure 3.13.

\[\text{Figure 3.13 - Resulting } C_q \text{ surface, as a function of speed and tip speed ratio}\]

Another characteristic verified from the data acquired was the relationship between the generator electromagnetic torque, \(T_{WTG}\), and the rectified DC current. This characteristic, seen in Figure 3.14, was approximated to:
\[ T_{WTG} = 11.875 \cdot I_{DC} \] (3.12)

![Generator electromagnetic torque vs. rectified DC current](figure)

**Figure 3.14 - Generator electromagnetic torque vs. rectified DC current**

3.3.4.4. Optimal \( C_p \) Operation

From the wind turbine model developed above, the optimal \( C_p \) operation curve was found. Despite not being related to establishing the model of the turbine (indeed it is derived from the model), this curve is presented here as part of the characterisation of the turbine. It is used as the setpoint for the system controller for maximum power extraction from the available wind. Further details of the control strategy can be found in Chapter 4.

As seen in the model derived in the previous section, the Proven wind turbine has a different "\( C_p \) vs. \( \lambda \)" curve for every rotational speed, \( C_p = f(\lambda, \omega_{WT}) \). From each of these curves, the optimal (or maximum) \( C_p \) point was found and the DC current associated with this point, determined.
Hence, a set of DC currents was obtained, one for each of the rotational speeds. By plotting these points against the speed, a curve representing the DC current that would optimally load the wind turbine, i.e. make it operate at optimal $C_p$, was defined (Figure 3.15).

![Original Data vs. Polynomial Fitting](image)

**Figure 3.15 - Optimal DC current as a function of rotational speed**

In order to verify the performance of the wind turbine under these conditions, a polynomial representation of this curve was derived (Equation 3.13) and used to generate the setpoint value of the DC current controller. In contrast to the speed controller previously described, the aim of the DC current controller is to optimally load the wind turbine through the Buck converter.

$$I_{DC_{opt}} = 0.0000457 \cdot \omega_{WT}^5 - 0.00572 \cdot \omega_{WT}^4 + 0.284 \cdot \omega_{WT}^3 - 6.985 \cdot \omega_{WT}^2 + 85.116 \cdot \omega_{WT} - 408.716$$

(3.13)

Controller objectives aside, the implementation of the DC current controller is very similar to that of the speed controller. The acquired rotational speed data is used in Equation 3.13 to generate a DC current reference, which is then compared with the actual DC current value. A PI controller processes the error signal and provides the
value for the gating signal to the Buck converter as its output. This value is then produced at an analog output, which is connected to the IGBT block gating circuitry. The control loop runs (variables averaged and outputs updated) every 200ms.

A 2½ hour test was carried out and the results are presented below. The DC current and its reference value can be seen in Figure 3.16. Only about 3 minutes of data are displayed in the graph, for better visualisation. Due to the dominantly resistive characteristic of the load (small electrical time constant), the reference is followed very closely (RMS error = 0.36 A).

The acquired data was binned into wind speed bins of 0.25 m/s. Unfortunately, there were not many high wind speeds at the time of the test. In fact, there were very few points over 10 m/s (about 0.1% of the data).
Figure 3.17 - Wind turbine power curve

Figure 3.17 shows the power output of the wind turbine obtained from the data binning. It also shows some data points extracted from a power curve graph provided by the manufacturer on their website. The scantness of data at the higher wind speeds may explain the inaccuracy of the curve above 10 m/s. And although this comparison does not purport to contest or confirm the data provided by the manufacturer in any way, the matching between the curves was very satisfactory. It also provides further confirmation of the adequacy of the control approach adopted to collect the data.

3.3.5. Matlab Model

The Matlab model of the wind turbine is illustrated in Figure 3.18. The wind time-series and the rotational speed are fed into the turbine rotor model to deliver the output of the rotor aerodynamic torque (Equation 3.6), after calculating the torque coefficient $C_q$ using Equation 3.11. The generator model uses the rotational speed and the load current to calculate the output terminal voltage and the electromagnetic and frictional torques.
All the torques outputs, from the rotor and the generator are than used in the mechanical dynamics equation that models the connection between the turbine rotor and the generator (Equation 3.14).

\[
J_{WT}\cdot \omega_{WT} = T_{WTR} - T_{WTfric} - T_{WTG}
\]  

(3.14)

From the electrical load connected to the generator, the stator current is used to calculate the electromagnetic torque ( \( T_{WTG} = 15.75 \cdot I_{sw} \) ). The net or acceleration torque is then calculated by subtracting the load torque (\( T_{WTG} \)) and the frictional torque (\( T_{WTfric} = 1.72 \cdot 10^{-3} \omega_{wr}^2 - 0.247 \omega_{wr} - 4.145 \)) from the aerodynamic driving torque (\( T_{WTR} \)). Given the turbine rotor and generator inertia (\( J_{WT} \)), rotational speed calculation is straightforward.

The generator terminal voltage is calculated by subtracting the stator resistance and inductive reactance voltage drops from the line-neutral air-gap voltage, which is in turn proportional to the rotational speed ( \( E_{WT} = 5.25 \cdot \omega_{wr} \) ). Since the generator is directly connected to a 3-phase diode bridge rectifier, the effect of inductive loads was considered to be negligible and, therefore, not included in the model.
3.4. Variable Speed Drives

As mentioned in the previous section, the output frequency range of the wind turbine generator was not suitable for direct connection to the conventional electrical motors used to drive the pumps of the RO system. Therefore, power converters (inverters) were used to link the turbine to the pumping motors.

The introduction of inverters in the system meant that the system could be smoothly controlled, but this required the development and implementation of a suitable dispatch procedure. A control strategy was devised whose main objective was to maximise the system overall efficiency, from source (wind turbine) to sink (RO unit) over a wide range of operation. The strategy implemented by the variable speed drives is described in Chapter 4.

As previously explained, the models used for the variable speed drives (inverters and induction motors) were developed in the earlier stages of the work and include a vector controlled model for the inverters and a 5th-order model for the motors.

3.4.1. Motors

Induction motors were used to drive the pumps due to their ready availability, robustness and low cost. High efficiency models were chosen in accordance with the design guideline of low energy consumption. The machines used were provided by Brook-Hansen (3 kW) and Siemens (1.5 kW).

A standard 5th-order dynamic model, based on 'dq' axis variables, was used to represent the induction motors ([Vas, 1992]). The motors stator currents and rotor magnetizing currents were used as the state variables and the model was developed using a stator-fixed reference frame in the calculation of the 'dq' variables (Equation 3.15).

By choosing the currents as state variables, the electromagnetic torque can be calculated using Equation 3.16. Thus the 5th state variable, the rotational speed, can be readily obtained using the equation of motion (Equation 3.17).
\[
\begin{align*}
\dot{i}_{sD} &= \frac{v_{sD}}{L_s} \left[ 1 + \frac{1-\sigma}{T_s} \right] i_{sD} + \frac{1-\sigma}{T_r} i_{rD} + \frac{1-\sigma}{\sigma} \omega_{rM} i_{rQ} \\
\dot{i}_{sQ} &= \frac{v_{sQ}}{L_s} \left[ 1 + \frac{1-\sigma}{T_s} \right] i_{sQ} + \frac{1-\sigma}{T_r} i_{rQ} + \frac{1-\sigma}{\sigma} \omega_{rM} i_{rD} \\
i_{rD} &= \frac{i_{sD}-i_{rD}}{T_r} - \omega_{rM} i_{rQ} \\
i_{rQ} &= \frac{i_{sQ}-i_{rQ}}{T_r} + \omega_{rM} i_{rD}
\end{align*}
\]
(3.15)

\[
T_{eM} = \frac{3}{2} P M \frac{L_m^2}{L_r} \left( i_{rD} \dot{i}_{sQ} - i_{rQ} \dot{i}_{sD} \right)
\]
(3.16)

\[
J M \omega = T_{eM} - T_{lM} - T_{fM}
\]
(3.17)

In the equations above, \( \dot{X} \) represents the time-derivative (\( \frac{d}{dt} \)) of the variable "\( X \);

\( i_{sD}, i_{sQ}, i_{rD} \) and \( i_{rQ} \) are the 'dq' components of the stator and rotor currents, respectively;

\( v_{sD} \) and \( v_{sQ} \) are the stator voltage 'dq' components;

\( L_s, R_s, L_r, R_r \) and \( L_m \) are the stator inductance and resistance, rotor inductance and resistance, and the magnetizing inductance;

\( T_s \) and \( T_r \) are the stator and rotor time constants, \( \frac{L_s}{R_s} \) and \( \frac{L_r}{R_r} \), respectively;

\( \sigma \) is the leakage factor (\( 1 - \frac{L_m}{L_s L_r} \)).
$T_s^{'}$, and $T_r^{'}$ are stator and rotor transient time constants, $\sigma \cdot T_s$ and $\sigma \cdot T_r$, respectively;

$\omega_{T_M}$ is the speed of the rotor;

$T_{eM}$, $T_{LM}$ and $T_{FM}$ are the motor electromagnetic torque, the load torque and the friction torque;

$J_M$ and $P_M$ are the combined inertia of the motor and the load and the number of pole pairs.

### 3.4.2. Inverters

For simplicity and robustness, two off the shelf inverters were used in the prototype: a 2.2 kW and a 1.5 kW standard V/f inverters (FID1000), supplied by FKI. Since the inverters used were commercial models, the details of their internal control software were not known.

The inverters are assumed as being almost ideal converters, i.e., delays in response and waveform distortions are neglected; this is supported by the manufacturers technical specification ([FID, 2000]) which states that the maximum response delay expected is 8 ms, which was very small compared to other time constants in the system. Also, it was considered that the inverters were able to ideally reproduce the reference set-points given by its controllers and therefore the references were directly applied to the motor model. In part, this simplification has been adopted because the representation of a high-frequency switching pattern would greatly reduce the model simulation integration step and, consequently, increase the simulation running time. The efficiencies were modelled from a test carried out on the 2.2 kW inverter. It was represented as a function of the inverter power output (motors input power), as seen in Figure 3.19. A model of a 'dq' indirect vector controlled (rotor flux oriented) voltage source inverter was developed and used to represent both inverters (Figure 3.20).
Figure 3.19 - Inverters efficiency

The inverter model uses the system controller reference speed output and the rotor flux as the input reference values. On the $d$-axis branch, the rotor flux reference is used to calculate the rotor magnetizing current reference, which is then compared to the motor actual current. The error signal is then fed to a PI controller, which outputs the $d$-axis component of the stator current reference ($i_{sd}^*$). Had a current controlled inverter been implemented (e.g. with a hysteresis comparator), these reference values could then be used directly as the inputs to the controller. But for the voltage controlled inverter, a reference voltage signal is needed. Therefore, this current reference is then compared with the actual value of $i_{sd}$ and the error signal, after passing thorough another PI controller, provides the $d$-axis stator voltage reference ($v_{sd}^*$). A voltage decoupling term is then applied to this reference, providing the final direct axis component of the stator voltages ($v_{sd}^*$).
In a complete inverter model, this value, together with the q-axis component, would be transformed into 'abc' values, which would then be compared to a saw-tooth waveform in order to generate a PWM switching pattern. The inclusion of the "dq" dynamic model thus allows this work to be extended at a later date to include a more complete drive representation.

A similar calculation routine is performed in the q-axis branch of the inverter, with the difference of an additional torque control loop between the speed reference input and the q-axis current control loop.

As previously mentioned, the input voltage range to the inverters is limited by a voltage limiter, which consists of a Buck converter connected to a resistive load. This auxiliary dump load is used to prevent the DC voltage to rise above 400V, the upper limit of the inverters input voltage range.

A slight modification was made to the gating circuit of the IGBT block when the Buck converter was used as a voltage regulator. Instead of using the analog output from the data acquisition board in the PC to provide the reference voltage to be compared to the saw-tooth waveform, the DC voltage signal is taken directly from the DC bus by means of a voltage divider. This signal is then compared to a voltage reference (approximately 5 V) in an analog comparator and the output is then used in the PWM
generator (saw-tooth waveform). The gating circuit is further detailed in Appendix C.

Figure 3.21 shows the DC voltage as a function of the wind turbine rotational speed when the Buck converter is connected to the DC bus as a voltage limiter. The overvoltage protection action can be clearly noticed (compare with Figure 3.5). In this plot, the converter starts to load the wind turbine at around 390V and the resistive load is fully applied at voltages just over 405V. In the final system configuration it was decided that it would be safer to have the voltage limiter set to a slightly lower level, so that the DC voltage did not get anywhere near the inverters recommended limit of 400V.

![Figure 3.21 - DC bus voltage with Buck converter (overvoltage protection action)](image)

3.4.3. Model Performance

The performance of the variable speed drives is illustrated in the following test simulation. The motors are started with zero reference speed (Figure 3.22 and Figure 3.23) for 0.5 s while the machines are magnetised (Figure 3.24 and Figure 3.25). A reference speed equal to the motors rated speed (1415 rpm for VSD1 and 940 rpm for VSD2) is then applied. The load is made to vary linearly with the speed so that the rated load is applied at rated speed. At instant \( t = 1 \) s the reference is then changed to 1000
and 500 rpm, respectively. And finally, at $t = 1.5$ s the reference is made equal to the rated speed once again.

3.5. Pumps

The use of positive displacement pumps in seawater RO desalination systems to provide the high pressure (up to 70 bar) feed is a common practice in industry. Although such pumps have considerably high discharge pressures, they cannot operate at high negative NPSH (Net Positive Suction Head), if at all ([Henshaw, 1987]). Therefore, a boost pump, usually centrifugal, is employed to pump the water from the feed tank/well to the high pressure pumps.
Two positive displacement pumps were used in the system. A progressive cavity pump is employed to pump the feed water from the feed tank (or, in a real system, a beach well) and a plunger pump, to provide the high pressure feed to the RO modules.

The model of the pumps was based on the fact that, being positive displacement devices, the flow and pressure of the water should be directly related to the pump rotational speed and shaft torque, respectively.

Therefore, the modelling tests were carried out over a wide range, at several points of the speed-pressure plane. This was done by keeping the driving motor at a constant speed while adjusting the discharge pressure of the pumps by means of a needle valve. After covering the pressure range at one fixed speed, the speed would be changed to a new setpoint and the pressure varied once again. For each speed-pressure pair, 30 seconds of steady-state operation data were recorded and averaged to produce the data point used in the modelling.

For the plunger pump, as exemplified below, the data points were processed in Matlab using both the `polyfit` function (to find a simple linear relationship between the flow and the speed and also the torque and pressure) and the non-linear data regression procedure described in Section 3.2.

3.5.1. Plunger Pump

The plunger pump used (CAT 317) was manufactured by CAT Pumps, USA, and has a triplex plunger action (three strokes per revolution). This multiple stroke per revolution configuration, has a ripple reduction effect in the output pressure, analogous to a rectified 3-phase AC waveform, if compared to a 1-phase waveform. Despite that, the ripple is still considerable and pressure dampener (a capacitor, following the analogy) was used to smooth out the pressure output.

The pump modelling consists essentially in finding the relationships between four variables: speed, flow, torque and pressure. According to Figure 3.1, the flow rate and the load torque to the driving motor were taken as the outputs, whereas the rotational speed and the pump input/output pressures as the inputs to the model.

Initially, the flow and torque data were fitted to the speed and the pressure data using Matlab `polyfit` function (Equation 3.18 and Equation 3.19):
\[ Q_{pp}(\omega_{pp}) = k_I Q_{pp} + k_2 Q_{pp} \cdot \omega_{pp} \]

\[ T_{pp}(\Delta P_{pp}) = k_I T_{pp} + k_2 T_{pp} \cdot \Delta P_{pp} \]

where,

- \( Q_{pp} \) is the plunger pump output flow (l/s);
- \( \omega_{pp} \) is the plunger pump rotational speed (rpm);
- \( T_{pp} \) is the plunger pump shaft torque (N.m);
- \( \Delta P_{pp} \) is the plunger pump differential pressure, i.e. the discharge pressure minus the suction pressure (bar);
- \( k_I Q_{pp} \) are the coefficients of the outlet flow function;
- \( k_I T_{pp} \) are the coefficients of the shaft torque function;

Then, a refined model was formulated, which added the effect of both the pressure in the flow calculation (Equation 3.20), and the speed in the torque calculation (Equation 3.21).

\[ Q_{pp}(\omega_{pp}, \Delta P_{pp}) = k_I Q_{pp} + k_2 Q_{pp} \cdot \omega_{pp} + k_3 Q_{pp} \cdot \Delta P_{pp} + k_4 Q_{pp} \cdot \omega_{pp} \cdot \Delta P_{pp} \]

\[ T_{pp}(\omega_{pp}, \Delta P_{pp}) = k_I T_{pp} + k_2 T_{pp} \cdot \Delta P_{pp} + k_3 T_{pp} \cdot \Delta P_{pp}^2 + k_4 T_{pp} \cdot \Delta P_{pp}^3 + k_5 T_{pp} \cdot \omega_{pp} + k_6 T_{pp} \cdot \omega_{pp}^2 \]

A comparison between the error at each data point for both curve fitting methods (Matlab polyfit and the non-linear data regression – NLDR) is presented below.

Figure 3.26 shows the variation of the error found for the two flow modelling functions (Equation 3.18 and Equation 3.20) in respect to the pressure. According to Equation 3.18 the flow depends solely on the speed and bears no relationship with the pressure. However, it is clear from the plot that when using the linear approach (polyfit),
there is an almost linear trend in the error with respect to the pressure. On the other hand, for the NLDR method, the error is evenly spread around zero, throughout the pressure range.

Figure 3.27 displays a similar comparison, where the error of the torque modelling functions, Equation 3.19 and Equation 3.21, are depicted as a function of the pump rotational speed. Once again, a clear trend in the error can be seen when Equation 3.19 is used.

Finally, the plunger pump model (using the NLDR method) output curves are shown in Figure 3.28 and Figure 3.29, where they are compared with the experimental data. The influence of the variation in pressure (from 10 to 70 bar, at approximately every 10 bar) can hardly be noticed in the flow characteristic (Figure 3.28), as the points for each speed are almost superimposed. On the other hand, the effect of a variation in speed is more noticeable in the torque function (Figure 3.29). Nevertheless, the model output is in very good agreement with the experimental data.
3.5.2. Moineau Pump

Centrifugal pumps are often used as the primary pumping device in desalination systems, conveying the feed stream from the feed tank to the high pressure pumps. One of the problems found with their use in the system was their efficiency, typically rated at less than 40% in small units, and considerably worse at partial load.

The selection was not straightforward as, for one of the possible system configurations, the output pressure of this pump should start at about 4 bar and reach values in excess of 10 bar. This operating range is considered to be too high for centrifugal pumps and too low for plunger pumps (this would imply in a considerable drop in their efficiency). After some consideration, a compromise was struck by choosing a progressive cavity pump (or Moineau pump).

![Moineau pump diagram](image)

**Figure 3.30 - Moineau pump stator/rotor diagram**
The Moineau pump is a "screw-type" pump in which pumping action is achieved by turning the rotor eccentrically within the stator (Figure 3.30). The particular model used was supplied by Netzsch Pumps. Built with seawater compatible materials, the rotor is made of 316 stainless steel and the stator, of a rubbery composite (nitrile).

The modelling procedure of the Moineau pump followed the same approach used for the plunger pump. The pump was operated at a range of rotational speeds and discharge pressures and the data collected was fitted to the non-linear functions shown in Equation 3.22 and Equation 3.23.

\[
Q_{MP}(\omega_{MP}, \Delta P_{MP}) = k_1 Q_{MP} + k_2 Q_{MP} \cdot \omega_{MP} + k_3 Q_{MP} \cdot \omega_{MP}^2 + k_4 Q_{MP} \cdot \Delta P_{MP}
\]  

(3.22)

\[
T_{MP}(\omega_{MP}, \Delta P_{MP}) = k_1 T_{MP} + k_2 T_{MP} \cdot \Delta P_{MP} + k_3 T_{MP} \cdot \Delta P_{MP}^2 + k_4 T_{MP} \cdot \omega_{MP} + k_5 T_{MP} \cdot \omega_{MP}^2 + k_6 T_{MP} \cdot \omega_{MP}^3 + k_7 T_{MP} \cdot \omega_{MP}^4
\]  

(3.23)

where,

- \( Q_{MP} \) is the Moineau pump output flow (l/s);
- \( \omega_{MP} \) is the Moineau pump rotational speed (rpm);
- \( T_{MP} \) is the Moineau pump shaft torque (N.m);
- \( \Delta P_{MP} \) is the Moineau pump differential pressure, i.e. the discharge pressure minus the suction pressure (bar);
- \( k_1 \) are the coefficients of the outlet flow function;
- \( k_1 \) are the coefficients of the shaft torque function;

The fitting of the model equations can be seen below, in comparison with the experimental data. As expected for a positive displacement pump, the flow is directly related to the pump rotational speed (Figure 3.31). But in contrast to the plunger pump, the shaft torque also bears a close relationship with the rotational speeds (Figure 3.32). This is due to a significant friction component in the shaft torque, in addition to the pressure component, derived from the contact between the rotor and the rubbery stator.
3.6. RO Membrane Modules

Spiral wound membranes were chosen amongst the main types of RO module designs available. The membranes used in the final system were manufactured by Koch Membrane Systems. The particular model used is a polyamide membrane, with 4 inches in diameter by 40 inches long (TFC® 1820HF).

During the initial modelling stage, 2.5-inch membranes manufactured by DOW Chemical were used in the laboratory tests. Later on a 2.5-inch module from Koch Membrane Systems was also tested with the purpose of comparing the performance of modules supplied by different manufacturers. From the system simulation results it was noticed that by increasing the number of cascaded modules – effectively, the membrane area – better overall efficiency was obtained. This led to the adoption of 4-inch modules in the final design. The 4-inch membranes are designed to operate at considerably higher flow rates, and therefore present almost negligible pressure drop at the low flows used in the proposed system.

The initial model equations and parameters values for the RO membrane modules as well as the procedure used to solve the equations were taken from [Slater, 1985] and [Slater, 1992]. The main problem found in the ready implementation of these equations in the system model was that some of the parameters were given as simple constants and no mention of their behaviour with respect to other variables, such as the temperature, was made.
A more detailed formulation of the membranes model was obtained from DOW Chemical, with their system design software (Reverse Osmosis System Analysis - ROSA) and its accompanying reference guide [DOW, 1996]. Both the software and the guide were very helpful in providing a deeper understanding of the working principle of the membranes and the general issues involved in designing RO systems. The guide also describes in detail the equations used by the software, including the specific parameters used to model the membranes, as follows.

The product stream flow rate is defined according to Equation 3.24:

\[ Q_p = A_w (OsmP_f) \cdot S_m \cdot TCF \cdot FF \cdot P_d \]  
(3.24)

where:
- \( A_w (OsmP_f) \) is the membrane solvent permeability, defined as a function of the feed stream osmotic pressure;
- \( S_m \) is the membrane area (m²);
- TCF is the temperature correction factor;
- FF is the fouling factor, an empirical measure of the ageing/plugging of the membrane;
- \( P_d \) is the trans-membrane, or resulting driving pressure (bar), expressed by:

\[ P_d = P_f - \Delta \frac{P_{fc}}{2} - P_p - OsmP_f - OsmP_p \]  
(3.25)

where:
- \( P_f \) and \( P_p \) are the feed and the product stream pressures (bar);
- \( OsmP_f \) and \( OsmP_p \) are the osmotic pressures of the feed and the product streams (bar);
- \( \Delta P_{fc} \) is the pressure difference (drop) between the membrane feed inlet and the concentrate outlet (bar);

Equation 3.26 defines the concentration of the solute in the product stream:

\[ C_p = B_s \cdot C_{fc} \cdot TCF \cdot PF \cdot \frac{S_m}{Q_p} \]  
(3.26)

where:
$B_s$ is the membrane solute permeability constant (s/m);

$C_{fc}$ is the average solution concentration on the feed side of the membrane, between the feed and the concentrate streams (mg/l);

$PF$ is the polarization factor.

The polarization factor is a measure of the concentration polarization, i.e. the increase in the concentration of the solute at the wall of the membrane on the feed side (see Figure 3.33).

![Figure 3.33 - Concentration polarization effect](image)

This is because as the solvent permeates through the membrane, the solute is mostly blocked (NaCl rejection rates of 99% are typical). This causes a temporary increase in the concentration within the boundary layer, as shown in Figure 3.33. This increased concentration is counteracted by the diffusion of the solute back into the bulk solution flow and, in steady-state operation, no solute should precipitate onto the membrane surface.

The polarization factor (Equation 3.27) is empirically determined as function of the solvent recovery ratio ($Q_p/Q_f$) and is responsible for an increase in the product concentration of about 5%.
The remaining variables - from the concentrate stream - are found by applying the relations for flow balance and mass transfer, which can respectively be expressed by Equation 3.28 and Equation 3.29:

\[
P_F = e^{0.7 \frac{Q_p}{Q_f}}
\]  
(3.27)

\[
Q_f = Q_c + Q_p
\]  
(3.28)

\[
Q_f \cdot C_f = Q_c \cdot C_c + Q_p \cdot C_p
\]  
(3.29)

The osmotic pressure of an aqueous solution is a function of the concentration and the nature of the solute, as well as the temperature of the solution, and its precise characterisation is by no means simple ([Stoughton, 1965] and [Gauwbergen, 1997]).

In the case of seawater, there is no single solute, but a mix of salts, with NaCl being a major component (almost 90%). For general purpose calculations, it is common practice in the desalination industry to use a standard composition for seawater as the same all over the world, as shown in Table 3.1. And although the concentration of the total salt content also vary from place to place, the average concentration of 35,000 ppm is widely adopted as being the standard seawater concentration.

Unfortunately for experimental purposes, seawater is not available everywhere, and despite the existence of commercial “instant sea” salt mixes, the use of NaCl solutions is a common workaround used in laboratory tests, allowing standardization and repeatability in test conditions and results. This alternative was adopted in this work.
Table 3.1 - Standard Seawater Composition

<table>
<thead>
<tr>
<th>Ion (+)</th>
<th>Concentration (mg/l)</th>
<th>Ion (-)</th>
<th>Concentration (mg/l)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calcium</td>
<td>410</td>
<td>Silica</td>
<td>0.04-8</td>
</tr>
<tr>
<td>Magnesium</td>
<td>1,310</td>
<td>Chloride</td>
<td>19,700</td>
</tr>
<tr>
<td>Sodium</td>
<td>10,900</td>
<td>Sulfate</td>
<td>2,740</td>
</tr>
<tr>
<td>Potassium</td>
<td>390</td>
<td>Fluoride</td>
<td>1.4</td>
</tr>
<tr>
<td>Barium</td>
<td>0.05</td>
<td>Bromide</td>
<td>65</td>
</tr>
<tr>
<td>Strontium</td>
<td>13</td>
<td>Nitrate</td>
<td>&lt;0.7</td>
</tr>
<tr>
<td>Iron</td>
<td>&lt;0.02</td>
<td>Bicarbonate</td>
<td>152</td>
</tr>
<tr>
<td>Manganese</td>
<td>&lt;0.01</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

SOURCE: [DOW, 1996]

The properties of a pure NaCl solution are slightly different than those of standard seawater. Amongst these properties, the osmotic pressure is of particular interest and in order to obtain a solution isosmotic to 35,000 ppm seawater, a solution of 32,000 ppm NaCl concentration was used. The osmotic pressure of a NaCl solution can be represented by Equation 3.30, which is derived from the definition given in [van't Hoff, 1887]:

\[
OsmP_{NaCl} = K_{NaCl} \left( T_{NaCl} + 273 \right) \frac{C_{NaCl}}{1000 - \frac{C_{NaCl}}{1000}}
\]

(3.30)

where,

\(C_{NaCl}\) is concentration of the NaCl solution (mg/l);

\(T_{NaCl}\) is the temperature of the NaCl solution (°C);

\(K_{NaCl}\) is the osmotic constant of NaCl (\(K_{NaCl} = 2.6545\) bar.l/°K.g).

In addition to ROSA, from DOW Chemical, another piece of proprietary software called ROPRO, supplied by Koch Membrane Systems, was used in order to increase understanding of the operation of the membranes.

In accordance to the modelling procedure adopted for other components, the membranes were tested in a range of operating conditions, consistent to what they would experience in a system with a variable power supply.
After some initial experimental tests with a 2.5-inch membrane operating over a wide range of pressure, flow, temperature and feed concentration values, it was noticed that the models used by the membrane manufacturers in their software packages were not suitable for characterisation of the modules over the same range. The operating conditions to which the membrane was submitted are shown in Table 3.2.

Table 3.2 - Operating conditions of RO module for performance comparison

|   | 1   | 2   | 3   | 4   | 5   | 6   | 7   | 8   | 9   | 10  | 11  | 12  | 13  | 14  | 15  | 16  | 17  | 18  | 19  | 20  | 21  | 22  | 23  | 24  |
|---|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| C_x (x10^3 ppm) | 32  | 32  | 32  | 32  | 32  | 32  | 32  | 32  | 32  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  | 44  |
| P_e (bar) | 45  | 45  | 65  | 65  | 45  | 45  | 65  | 65  | 45  | 45  | 65  | 65  | 45  | 45  | 65  | 65  | 45  | 45  | 65  | 65  | 45  | 45  | 65  |
| Q_f (l/min) | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  | 7.5 | 19  |

The operating conditions for each of the 24 test points were set and plenty of time given for the settling of transients. Experimental points were then collected as an average of one minute of steady-state operation. Each of the conditions listed above was also simulated in the manufacturers software packages.

![Figure 3.34 - Experimental and simulated (from manufacturers software) product flow](image_url)

A comparison of the results gathered is discussed next. Figure 3.34 shows the desalinated product flow rate, as estimated by ROSA and ROPRO and the values obtained experimentally. It is clear that, although good agreement was found between both manufacturers software, the membrane tested underperformed throughout the...
range. The reason for this discrepancy is not known, specially because the membrane tested was brand new and should therefore be at its very best (the performance of new RO modules decays quickly and stabilises after a few days of use, due to initial compaction and settling of the membrane inside).

Assuming that the salt passage through (or salt rejection by) the membrane was consistent amongst the simulated and experimental results, the reduced product flow found experimentally would result in higher salt concentration in the product. This can be verified in Figure 3.35. Although the product concentrations are very similar at the lower feed concentration of 32,000 ppm, at 44,000 ppm - and specially at the low concentrate pressure of 45 bar (points 13, 14, 17, 18, 21 and 22), the results differ considerably.

One might argue that seawater with such high concentrations would be very unlikely to be found in practice, but in fact, in a train of cascaded modules the feed water concentration seen by the last of the modules could easily be within this range.

![Diagram](image)

**Figure 3.35 - Experimental and simulated (from manufacturers software) product concentration**

This disparity between the results suggested that, although very reliable for the design of standard desalination systems, the models implemented by the manufacturers in their software would not be suitable to model a system operating at variable conditions (particularly one which would be expected to often operate at lower pressures).
This conclusion led to the adoption of an alternative approach to the membranes modelling. One that could take into account the influence of several input variables with a typical non-linear influence on the output. The non-linear data regression technique, previously described and used to model the wind turbine and the pumps, seemed an obvious choice.

3.6.1. Modelling Procedure

Similarly to the tests carried out with the 2.5-inch membrane already described, the 4-inch membrane modelling tests consisted in gathering the steady-state performance characteristics of one module within a range of expected operating conditions.

The feed stream characteristics changed in the tests were the same: pressure, flow, temperature and concentration. Each of them were given three values (setpoints) within their expected operating range. The test procedure is explained below:

The permutation of the conditions was implemented in the sequence shown in Table 3.3. For each concentration setpoint, the test would start at 25 °C/60 bar (setpoint 1) and the flow would be changed (setpoints 1, 2 and 3) and the data collected. The pressure would be reduced to the next lower setpoint and the flow varied once again. The pressure setpoints 2 and 3 are dependent on the feed concentration, and therefore show three different values. The higher the feed concentration, the higher its osmotic pressure. The increase in the lower limit of the feed pressure aimed at achieving similar trans-membrane pressures throughout the test for the three concentrations.

Table 3.3 - RO modules modelling test operating conditions

<table>
<thead>
<tr>
<th>Variable\Setpoint</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_f (x10^3$ ppm)</td>
<td>33</td>
<td>39</td>
<td>45</td>
<td>----</td>
</tr>
<tr>
<td>$T$ (degC)</td>
<td>25</td>
<td>15</td>
<td>25</td>
<td>40</td>
</tr>
<tr>
<td>$P_f$ (bar)</td>
<td>60</td>
<td>40/45/50</td>
<td>35/40/45</td>
<td>----</td>
</tr>
<tr>
<td>$Q_f$ (l/min)</td>
<td>18</td>
<td>12</td>
<td>7</td>
<td>----</td>
</tr>
</tbody>
</table>

Following the sequence, the temperature would be reduced to 15 °C (setpoint 2), and the pressure and flow adjusted once again. In order to verify the consistency of the
data, another sequence of samples would be collected at 25 °C. Finally, the temperature would be raised to 40 °C (setpoint 4). This procedure was adopted for the three concentration setpoints used. Each data point consisted of an average of 60 seconds (1 kHz sampling) of steady-state operation.

3.6.2. Matlab Model

This experimental procedure yielded 108 points that were then used to model the product stream flow and concentration. Taking the relationships between variables suggested by Equation 3.24 (product flow) and Equation 3.26 (product concentration) as a starting point and using the non-linear data regression procedure already described, the relationships were derived as described below.

The first variable to be modelled was the product flow ($Q_p$). The input variables were the feed flow ($Q_f$), concentration ($C_f$), temperature ($T$) and the trans-membrane pressure ($P_d$). During the modelling, the feed pressure ($P_f$) was also tried as one of the model inputs, but it was noticed that the dependence of the product flow on the trans-membrane pressure was significantly more noticeable (this dependence is clearly indicated in Equation 3.24). The resulting model equation and the model estimated output can be seen below.

$$Q_p = 0.012 - 0.011 \cdot Q_f + 7.1 \times 10^{-4} \cdot P_d - 2.83 \times 10^{-7} \cdot C_f$$

$$+ 1.03 \times 10^{-4} \cdot T - 3.24 \times 10^{-5} \cdot (P_d \cdot T) - 5.36 \times 10^{-7} \cdot \left(\frac{C_f}{T}\right)$$

$$- 0.97 \times 10^{-9} \cdot (P_d \cdot T \cdot C_f) - 1.30 \times 10^{-4} \cdot (P_d \cdot Q_f \cdot T)$$

$$\text{Equation 3.31}$$

Noticeably, from Equation 3.31, modelling the water permeation process through a semi-permeable membrane is not a simple task and relationships between variables are far from obvious, particularly, if account is taken of the extent of the operating range this expression purports to represent. Surely, the equation above is one of many representations that could be used, but amongst the ones tested, it was the one which provided the best matching between the measured data and the model outputs, as plotted in Figure 3.36.
The next variable of interest was the product concentration ($C_p$). As indicated by Equation 3.26, the salt concentration in the permeate stream (milligrams/litre or ppm) can be represented by the ratio between the salt passage (or the solute flux, in milligrams/sec) and the water flow rate (solvent permeation, in litres/sec) through the membrane.

Considering that the product flow was already one of the variables represented in the model, the salt passage was chosen as the variable of interest, instead of the concentration, which could be found from the ratio between both variables.

$$SP = 4.34 + 3.82 \times 10^{-5} \cdot C_f - 0.19 \cdot P_d - 0.3 \cdot T$$
$$+ 8.28 \times 10^{-3} \cdot T^2 + 3.10 \times 10^{-6} \cdot (C_f \cdot T)$$
$$+ 1.95 \times 10^{-2} \cdot (P_d \cdot T) - 4.17 \cdot \left( \frac{Q_{P_{EST}}}{Q_f} \right)$$

Equation 3.32

The relationship found to best represent the salt passage is shown in Equation 3.32. Following the theoretical concept that the product concentration is related to the recovery ratio (concentration polarisation), the previously estimated value for the product flow ($Q_{P_{EST}}$) is used in the calculation of the salt passage.
The matching between the measured and estimated salt passage is depicted in Figure 3.37. In this case, the term measured is used not to indicate that the salt passage was effectively measured, but derived from measured variables (product between the permeate flow and concentration). Finally, the estimated product concentration is calculated and compared to the measured values, as shown in Figure 3.38.

![Figure 3.37 - Salt Passage (measured and estimated)](image1)

![Figure 3.38 - Product concentration (measured and estimated)](image2)

The overall agreement between the modelled and the measured variables was very satisfactory, with RMS errors in the range of 5%. To complete the steady-state characterisation of the membranes, the estimated concentrate stream variables are calculated using Equation 3.28 and Equation 3.29. Good representation of these variables is of great importance, as they are used as the input variables for subsequent RO modules in a cascaded arrangement. A comparison between the modelled and experimental points is shown in Figure 3.39 and Figure 3.40. Here, once again, the measured points are obtained by manipulation of other variables, also employing Equation 3.28 and Equation 3.29.
The use of steady-state relationships to represent such a complex process seems straightforward and justifiable if applied to a system that will be run under constant operating conditions. On the other hand, if one considers the variable nature of the wind – and the fact that the system proposed in this work does not include an energy storage device - this approach becomes questionable.

Preliminary laboratory results obtained in [Lising, 1972], and theoretical evidence discussed in [Rahal, 2001], refute this possibility. Both studies confirm the idea that steady-state characterisation can be effectively used in the analysis of a system operating under variable conditions.

Nevertheless, the introduction of some dynamic behaviour, even if not greatly refined, was seen as a possibility to improve the process representation. [Alatiqi, 1989] demonstrates the strong influence of the feed pressure input on the product flow and concentration outputs in the dynamic modelling of a hollow fibre RO module (although it uses the feed pH to effectively control the product concentration, due to its greater influence). As the pH is assumed to be constant in this work, a test to verify the response of the product characteristics to a variation in the feed pressure was carried out.

The test was performed at three different feed flow rates, with the special intent to investigate its influence on the product concentration dynamics. This is because, at a reduced feed flow, less mixing should be expected to occur between the boundary layer (at the membrane surface) and the bulk feed flow (less turbulence), hence the effect of
concentration polarisation could be more pronounced.

The procedure consisted in analysing the response of the product stream characteristics to step inputs in the feed pressure, performed in three distinct pressure ranges. Starting from 35 bar, the pressure would be taken to 45, 55, 45, and then returned to 35 bar once again. By doing this, a range of trans-membrane pressures would be covered. The feed was kept at 25 °C and 33,000 ppm during the test.

The dynamic response was modelled by a simple first order transfer function of the type \( G(s) = k_d \frac{1}{T \cdot s + 1} \), where the time-constant \( T \) was deducted from direct inspection of the response curves ([Dutton, 1997]). The objective of the test was to characterise only the dynamic behaviour, not the absolute values. Therefore, the direct gain \( k_d \) was taken as the unity and the resulting transfer function applied to the trans-membrane pressure contribution to the steady-state model previously developed, as shown in Figure 3.41, for the product flow.

![Simulink representation of the product flow dynamic model](image)

Figure 3.41 - Simulink representation of the product flow dynamic model

Figure 3.42 shows the trans-membrane pressure and the product flow for the three feed flow rates used. The response of the product flow to pressure variations was almost instantaneous and resulted in the transfer function \( G_{Q_p}(s) = \frac{1}{0.1 \cdot s + 1} \).
The determination of the product concentration response was not as straightforward. The results can be seen in Figure 3.43. It was noticed that although the response - not the absolute value - was not related to the feed flow rate, it showed an inverse relationship with the trans-membrane pressure being applied. The higher the pressure applied, the faster it would respond. An average value was chosen and the resulting function used in the model was \( G_{cp}(s) = \frac{1}{25s+1} \).
3.7. Energy Recovery Device

As discussed in Chapter 2 the presence of an energy recovery device can make a substantial difference in the energy consumption of a reverse osmosis system. And since one of the main objectives of the design exercise was to reduce the overall specific energy, two possibilities were considered suitable for a small system and worthy of further investigation: the Clark Pump and a hydraulic motor.

3.7.1. The Clark Pump

The Clark pump is a very efficient energy recovery device developed originally for the small reverse osmosis systems used in yachts (using a single membrane module). It works by recovering the hydraulic energy from the concentrate stream and returning it directly to the feed stream. The Clark pump has been developed by Spectra Watermakers (California, USA).

3.7.1.1. Working Principle

The Clark pump is composed of two piston bases connected by a rigid rod and a set of passive control valves (Figure 3.44). Its working principle is based on the pressure transfer (recovery) along the pistons, from the concentrate (brine) to the low-
During the system initial start-up, a by-pass valve is open, which allows the feed to "flood" all the components, flowing across the membrane modules and back to the concentrate chamber. At this stage, the high-pressure feed pressure is very low, and there is no product flow through the membrane. Due to the presence of a rod connecting the two piston bases, the volume of the feed and concentrate chambers are different.

When the by-pass valve is closed, this difference in volume causes the concentrate stream to push the piston and, consequently, the feed stream pressure to rise. Since the net driving pressure (high-pressure feed pressure minus osmotic pressure) is negative during the start-up, there is no product flow. This pressure build-up loop continues until stability is reached. This means that the pressure will rise until the feed and concentrate chambers are full and the difference in volume between both (volume of the connecting rod) is leaving the membrane as product water. At the end of a stroke, the direction of displacement of the piston is smoothly reversed, keeping the feed pressure almost constant.

Therefore, were the losses ignored, the Clark pump should present - at any operational speed - a constant recovery ratio, which is the ratio between the product and the feed stream flows (or the ratio between the connecting rod cross-section and the
piston base area). This design feature implies inherent "control" of the feed pressure, because if any characteristic of the feed water such as flow, concentration or temperature varies, the feed pressure will vary accordingly in order to keep the recovery ratio constant.

Regarding the energy flow, the Clark pump transfers the concentrate stream hydraulic power \( P_{\text{ow}} = P_c \cdot Q_c \) straight to the low-pressure feed, presenting no intermediate energy conversion stage. Therefore, the only losses in the energy recovery process are due to the internal piston friction and flow leakage between adjacent chambers. And despite being one of the advantages of the device, the inherent fixed recovery ratio is also a drawback as it does not allow much flexibility in a standard system design. Fortunately, this problem may be circumvented by the alternative system design proposed in Chapter 4.

3.7.1.2. Modelling

The Clark pump working principle theory developed in [Thomson, 2003] was used, together with comprehensive lab testing, to build a model in Matlab-Simulink. The inputs to the model are on the left of the Simulink diagram shown in Figure 3.45 and the outputs, on the right.

![Matlab model of the Clark Pump](image)

Figure 3.45 - Matlab model of the Clark Pump
In the diagram, $Q_f$, $P_h$, $P_c$ and $P_e$ are the feed flow and pressures of the high-pressure, concentrate and exhaust streams respectively. The "R (RECTheory)" block is the ratio between the connecting rod and the piston bases cross-sectional areas or, in fact, the theoretical recovery ratio.

The model uses the input/output relationships derived from its working principle theory to calculate both the volumetric (leakages) and mechanical (friction) losses. These are respectively represented in the diagram by the "FlowLosses" (Equation 3.33) and "PressureLosses" (Equation 3.34) subsystems.

\[
Q_{Loss} = 0.000178 \cdot Q_f \cdot P_h + 1.5 \times 10^{-5} \tag{3.33}
\]

\[
P_{Loss} = 49.22 \cdot Q_f^2 + 0.071 \cdot P_{drop} + 0.53 \tag{3.34}
\]

In Equation 3.34, $P_{drop}$ is the difference between the high-pressure feed and the concentrate stream pressures (or the pressure drop across the membranes the Clark pump is connected to).

3.7.2. The Hydraulic Motor

As discussed in Chapter 2, one simple alternative energy recovery option that can be used in RO systems to reduce energy consumption is to connect a reverse running pump to the concentrate stream. A more efficient alternative to this approach is the use of a hydraulic motor. The hydraulic motor tested was manufactured by Danfoss Water Hydraulics (MAH5). It has also been modelled so that comparisons between the Clark pump and other alternative options could be made.

3.7.2.1. Working Principle

In principle, the hydraulic motor is like a reverse running positive displacement pump. It uses the axial piston principle to make the high pressure inlet flow spin an angled plate (swash plate), which in turn drives the shaft. Being a positive displacement device, its rotational speed is directly related to the inlet water flow rate. The specific model used was very compact, and could stand up to 140 bar of input pressure at speeds
above 3500 rpm (rated power of 4.1kW).

This specification far exceeded the rating needed for the proposed system, but features such as high starting torque, few wear parts and low maintenance costs, and also the fact that it was lubricated by the driving concentrate stream (no oil lubrication) made it a strong candidate for a small system. On the other hand, it introduces at least two further conversion stages (hydraulic - mechanical - hydraulic) in the energy recovery process, which may ultimately incur in lower system efficiency.

3.7.2.2. Modelling

Similarly to the positive displacement pumps, a model that described the relationships between flow, torque, speed and pressure was devised. The problem with this choice of variables was the direct measurement of the hydraulic motor shaft torque (it had no sensors connected to it) and inlet flow (none of the sensors could withstand pressures above 20 bar).

Thus, in order to test the hydraulic motor, it had to be assembled on to the rig. As its operating speeds were much higher than those of the pumps, the shaft was connected to the plunger pump shaft through a pulley and belt arrangement. The membrane concentrate stream was used as its main flow inlet. This arrangement can be seen in Figure 3.46.

![Figure 3.46 - Hydraulic motor test setup](image-url)
With this setup, the concentrate flow ($Q_c$) could be calculated directly from the difference between the feed ($Q_f$) and the product ($Q_p$) stream flows. The measurement of the torque was not so straightforward and relied upon the model previously derived for the plunger pump. The torque sensor was fitted to the induction motor driving the plunger pump ($T_{IM}$). The torque delivered by the hydraulic motor at the plunger pump shaft ($T_{HML}$) was then calculated by subtracting the actual torque reading of the induction motor from the torque the plunger pump would produce ($T_{PP,EST}$) in those operating conditions had the hydraulic motor not been connected to it (estimated from its model), or $T_{HML} = T_{PP,EST} - T_{IM}$.

Then, to find the actual torque output of the hydraulic motor, at the high-speed end of the pulley/belt, the above estimated value was divided by the gear ratio used. This means that the torque expression below represents the high-speed torque at the hydraulic motor shaft, but already accounts for the losses in the pulley/belt transmission. The functions found to represent the flow and the torque as functions of the speed and pressure are:

\[
Q_{HM} = 4.51 \times 10^{-3} + 77.1 \times 10^{-6} \cdot \omega_{HM} + 1.72 \times 10^{-9} \cdot \omega_{HM}^2 \\
+ 6.42 \times 10^{-4} \cdot P_{HM} - 4.2 \times 10^{-6} \cdot P_{HM}^2
\]

\[
T_{HM} = -1.99 + 1.08 \times 10^{-3} \cdot \omega_{HM} - 2.3 \times 10^{-7} \cdot \omega_{HM}^2 \\
+ 70.78 \times 10^{-3} \cdot P_{HM} + 33.36 \times 10^{-6} \cdot P_{HM}^2
\]

where $Q_{HM}$ and $T_{HM}$ are the hydraulic motor flow (l/s) and torque (N.m), respectively; $\omega_{HM}$ and $P_{HM}$ are the hydraulic motor speed (rpm) and inlet pressure (bar), respectively.

3.8. Summary

In this chapter, the main components of a prospective wind powered RO system were introduced, and their mathematical models and modelling procedures presented.
Despite the fact that the final system design had not yet been determined at the earlier stages, the notion of which requirements such configuration should meet was clear: high energy efficiency and ability to operate at controllable, variable conditions.

These requirements were used in the selection of the components, including the energy recovery devices. As the use of such devices was mandatory, if reduced energy consumption was to be achieved, two alternative solutions were proposed and modelled: a Clark pump and a hydraulic motor.

The use of the individual component models supported the development and assessment of different system configurations, through the analysis of computer simulation results. Two of these configurations are described in greater detail in the next chapter.
CHAPTER 4
SYSTEM MODELLING AND PERFORMANCE PREDICTIONS

Using the component models developed in Chapter 3, the performance of different system configurations, including the control strategy, could be assessed and compared by computer simulation. This chapter presents two of these configurations. Their layouts are described and performance predictions are compared and discussed.

4.1. Introduction

From preliminary simulation results obtained for some tentative system layouts, two configurations were chosen for further consideration and detailed modelling: one using a Clark Pump and the other using a hydraulic motor as the concentrate stream energy recovery mechanism.

The power supply stages of both systems are the same, from the wind turbine, passing through the 3-phase bridge rectifier, to the DC bus and the overvoltage protection circuit (Buck converter - resistive load).

4.2. Clark Pump System

The first configuration layout uses the Clark pump for high-pressure pumping and energy recovery. As described in Chapter 3, the Clark pump presents a fixed recovery ratio of about 10%. This is an undesired limitation to the system as RO seawater
desalination plants can usually achieve recovery ratios between 30 and 40% and sometimes, depending on the feed water quality and membrane modules layout, in excess of 40%.

The fixed recovery ratio limitation was overcome by using a high-pressure injection pump, also operated at variable speed, connected "in parallel" to the Clark pump. A diagram of the system is shown in Figure 4.1.

![Figure 4.1 - Clark pump system diagram](image)

The Proven wind turbine provides a 3-phase variable-voltage variable-frequency supply to the whole system. This three-phase supply is rectified by an uncontrolled diode bridge rectifier, creating a variable-voltage DC bus.

Two variable speed drives (inverter/induction motor) are directly connected to the DC bus. They are the primary means by which the system control strategy is implemented. The voltage regulator is also connected to the DC bus to prevent the DC voltage to rise above the inverters input voltage upper limit (400 V).

One of the induction motors (1.5 kW) drives the medium-pressure Moineau pump, which is the primary pumping unit of the system. It provides the suction from the well, delivering the seawater feed stream to the other pumps. The other induction motor
(3.0 kW) drives the high-pressure plunger pump.

As mentioned above, the Clark pump recovery ratio is fixed at about 10%. This means that it will develop as much pressure in the high-pressure side of the feed stream as needed to allow that 90% of its feed flow returns as the concentrate flow stream. Thus, if more feed water is injected into the system by another pump, the Clark pump will raise the pressure to make this “excess” injected water leave the system as product water, therefore, increasing the system overall recovery ratio. This is the role of the plunger pump, also referred to as the injection pump. It operates at high pressure and very small flow rates. Its outlet is connected to the Clark pump high-pressure outlet, just before the first RO module.

As both the Moineau and the plunger pumps are positive displacement pumps, the overall system recovery ratio can be adjusted simply by varying the speed relationship between the Moineau pump and the plunger pump. This relationship defines how the feed flow is shared between the pumps.

In the diagram shown in Figure 4.1, the block “Reverse Osmosis Modules” represents an array of four cascaded 4-inch modules. In this arrangement, the concentrate outlet of the first module is connected to the feed inlet of the second and so on, until the fourth and last module. The last module concentrate outlet is connected to the Clark pump concentrate inlet.

4.2.1. Control Strategy

Albeit indirectly, the use of the injection pump makes it possible to control the feed stream pressure and flow independently. This extra degree of freedom is not generally present in RO systems with energy recovery. For instance, in a system using only a Clark pump, the pressure varies with the flow, as the recovery ratio must be kept constant. A control strategy was developed so as to take advantage of this design characteristic.

A block diagram of the control strategy is shown in Figure 4.2. It uses two model-based functions to generate the control setpoints. The control variables are the rotational speeds of the induction motors, which in turn determine the feed stream flow rate through the Moineau and the plunger pump.
The control functions are based on the models developed in Chapter 3, and although they are both used to generate the reference speed to the motors, each one performs a different control objective.

![Control strategy diagram of the Clark pump system](image)

*Figure 4.2 - Control strategy diagram of the Clark pump system*

The first one, represented in the block diagram by the function ‘Idc\text{ref} vs. \omega’", is responsible for extracting the maximum power available from the wind. As discussed in Chapter 3, for a given wind speed, the turbine output power is a function of its rotational speed, which in turn is related to the load supplied by the generator (output current). Therefore, the first controller will aim at establishing and maintaining optimal current output so that the energy input to the system is maximised. The function ‘Idc\text{ref} vs. \omega” is the polynomial function defined in Section 3.3.4.4 (Optimal C_p Operation). It is worth mentioning that the power input to the system may not be the maximum power available at all times, for at high wind speeds the turbine output exceeds the system rated power (around 1.7 kW) and there are no means to store the energy surplus.

This optimal reference current is compared to the actual DC current and the error is used as the input to a standard PI controller, represented by the block named “Control” in the diagram. The output of the PI controller is the reference speed to the variable speed drive connected to the Moineau pump (medium pressure pump). Since the current cannot be controlled at the very high wind speed conditions (the load current
is limited by the system maximum power), integral wind-up protection is implemented by limiting the integral component of the PI controller output.

Alternative control techniques were considered such as variable gain PI control (gain scheduling) to improve dynamic performance at different operating regions; sliding mode control to account for the system non-linearities; and also hill-climbing control to avoid the use of a model-based function altogether, as the actual plant may deviate from the established model with time. These alternatives were not formally assessed and remain as possibilities for further development.

The other control function, represented by the block "$K_T$ vs. $P_{dc}$", manages the energy delivered to the RO unit. Its output value, $K_T$, represents a fictitious gear ratio between both induction motors, and hence, defines the speed relationship between the positive displacement pumps. This function implements the share of power between the motors, for a given power input ($P_{dc}$), that will incur in minimal specific energy consumption by the system. This function was derived from a simulation of the system model (without the wind turbine) in equilibrium condition (steady-state).

In this simulation, the speeds of the motors are varied in steps over a certain range. This was done by fixing the speed of one motor and varying the speed of the other; then changing the first one to another setpoint and varying the second one once again, until both motors had covered the operating range of the system. The limits to these operating ranges were the maximum pressures and flows that the components could operate at. For each combination of speeds, the specific energy and the DC power consumption were taken. At the end, for each of the DC power inputs found, the minimum specific energy and its respective speed combination were determined. This collection of points was then approximated by a polynomial function (Equation 4.1), which described the relationship between the speed ratios ($K_T$) and the power input ($P_{dc}$).

$$K_{Tref} = -0.177 \times 10^{-6} \cdot P_{dc}^2 + 0.834 \times 10^{-3} \cdot P_{dc} + 0.37$$

This essentially empirical function, plotted in Figure 4.3, incorporates the efficiencies of all the components over the system operating range and yields the
optimal steady-state operating point for a given input power condition.

Figure 4.3 - Speed ratio for minimum specific energy consumption as a function of input DC power

By implementing both control functions, one responsible for the energy capture and the other, for the delivery of this energy, it was expected that optimal system operation could be achieved (from the energy usage standpoint). This was considered critical for a system without energy backup or storage, operating from an intermittent source such as the wind.

4.2.2. Performance Predictions

The following performance predictions are taken from a detailed Matlab-Simulink model of the entire system, built using the models of the components characterised and described in Chapter 3. The salinity of the feed is taken to reflect that of standard seawater (35,000 ppm), which is isosmotic to a 32,000 ppm solution of pure NaCl. Throughout the simulation, the feed temperature is assumed constant and equal to 25 °C.

The performance prediction analysis was divided in two distinct types of simulation: one that finds the equilibrium points, or the steady-state characteristics, and another, that verifies the dynamic behaviour of the system.
4.2.2.1. Long-Term (Steady-State) Performance

The first analysis carried out aimed at establishing the “input-output” characteristic of the system, i.e. the relationship between mean (say, ten-minute or hourly) wind-speed and fresh water production, as depicted in Figure 4.4.

![Graph showing product flow vs. wind speed](image)

*Figure 4.4 - Product flow vs. wind speed*

The importance of this characteristic relies on the fact that it can be used in a system siting study. Similarly to a wind turbine power curve, it would give the expected output from the system, given the resource in a certain location. A statistical analysis can be conducted by applying the wind speed probability (Weibull distribution) for a candidate location against the characteristic of Figure 4.4. The result is the water production probability expected for this location.
Figure 4.5 and Figure 4.6 show the system recovery ratio and specific energy as a function of the wind speed, respectively. Although the recovery ratio varies considerably, from 30% to almost 45% over the range of wind speeds, the specific energy remains very low. It presents an overall value below 4 kWh/m³, which is very satisfactory considering the system size and compared to the typical values between 4 and 10 kWh/m³ found for bigger systems with energy recovery. And although the values are slightly higher at the lower wind speeds, it is very even throughout the operating range. This translates into the fact that water production water is directly related to the available power, as verified by Figure 4.4. The abrupt reduction in the specific energy in the lower wind speeds (between 6 and 8 m/s) can be explained by the fact that although the generated power roughly doubles in this range, the product flow is almost three times as much, indicating an improvement in efficiency of the system for the slightly higher wind speeds.

4.2.2.2. Short-Term (Dynamic) Performance

A dynamic model of the system was also developed to verify its performance during normal transient operation, under turbulent wind. This is particularly important in the determination of the control strategy and fine-tuning of the controller parameters.

No formal approach, such as root locus analysis, was employed to adjust the PI controller gains since this would have been problematic given the complexity of the overall system and the non-linear nature of a number of key components, not least the passive pitching wind turbine. Instead, the gains were adjusted from the results of
simulated tests using step changes in the wind speed input.

The following figures show the behaviour of the system for the wind speed time series shown in Figure 4.7-a. This series, used as the input to the system, is 5 minutes long and has an average wind speed just above 8 m/s. The desalinated product water flow is plotted in Figure 4.7-b. Its shows an average value of 6.75 m$^3$/day. The dependence between both curves is obvious, as expected from the steady-state analysis. But because the control system uses the turbine rotational speed as its main input, the system variables have smoother variations, as the turbine rotor inertia acts as a low-pass filter for the fast changing wind speed values.

![Graphs](image)

*Figure 4.7 - a) Wind speed time series, b) Product flow rate*

Figure 4.8-a shows the feed hydraulic and osmotic pressures at the input of the RO modules. The net driving pressure (trans-membrane pressure) is always positive, so despite the dip at $t \approx 200$ s, there is no interruption in the product flow. Contrary to the water (solvent) flow, the salt passage through the membranes is mostly a function of the difference in concentration between the feed and the product streams (which is approximately constant for each of the modules). Therefore, the product water concentration (Figure 4.8-b) shows a behaviour inversely proportional to the flow. The concentration shown is resultant of the mixing of the different product concentrations of the four modules, and presents an average value of almost 550 ppm which can be regarded as high quality potable water without any significant salinity taste.
Figure 4.8 - a) Feed and osmotic pressures, b) Product concentration

Figure 4.9-a shows the product recovery ratio, which varies between 25 and 42%. No restriction was imposed upon this range and it is possible that in a practical situation the upper limit may need limitation to prevent membrane fouling/scaling.

Figure 4.9-b displays the instantaneous specific energy, obtained by dividing the DC power input to the inverters (in kW) by the product water flow (m$^3$/h). The DC power was used, instead of the turbine output power, because the voltage limiter (Buck converter) becomes active at t = 250 s due to the higher wind speeds.
The specific energy graph is therefore a function of the desalination system energy consumption and not of the total energy generated by the turbine. The peak in energy consumption verified at lower recovery ratios is due to an overall reduction in the efficiency of the motors, pumps and inverters at partial loads. The only component that presented improved performance at these conditions was the Clark pump. This behaviour was attributed to the low flow rates, and hence, reduced friction losses caused by the piston displacement.

Moving from the overall system performance and focusing on the system control, Figure 4.10 shows some of the variables related to the first controller, responsible for managing the power input to the system.
Figure 4.10 - a) Wind turbine speed, b) Simulated DC current (reference and actual), c) DC current error

The wind turbine rotational speed, depicted in Figure 4.10-a, is used to calculate the reference DC current, using the polynomial function already described. This reference and the actual current value are shown in Figure 4.10-b. Even with integral wind-up limitation implemented, the reference value was limited to 5 A, to prevent excessive errors due to saturation of the system output. The controller performance is very good throughout, as displayed by the current error ($I_{dc\text{ERROR}} = I_{dc\text{REF}} - I_{dc}$) in Figure 4.10-c. Clearly, the only moment in which the DC current is not able to follow the reference is at $t \approx 250$ s, when the Moineau pump motor is at its maximum speed (800 rpm) and the controller output is saturated. Even with a considerable error during this time, the overall RMS error was only 0.28 A.

The efficacy of the control action can be verified by its influence on the wind turbine performance. The power coefficient of the turbine rotor is plotted against the tip speed ratio, in Figure 4.11. Both values are calculated using the turbulent wind input. Despite its rough characteristic, it is apparent that by appropriately loading the turbine, it is possible to operate it within the vicinity of the optimal power conversion points, or
the peak of the $C_p$ vs. $\lambda$ curve.

![Graph of Wind turbine power coefficient ($C_p$) vs. tip speed ratio ($\lambda$)](image)

**Figure 4.11 - Wind turbine power coefficient ($C_p$) vs. tip speed ratio ($\lambda$)**

Finally, Figure 4.12 shows the input and output variables of the second controller, linked by the relationship shown in Equation 4.1. During the simulation, the DC power (Figure 4.12-a) ranged from just under 420 W to almost 1,650 W (average of 960 W), a 4:1 variation which although highly desirable, is not possible with standard commercially available RO desalination units. The fictitious gear ratio between the injection pump motor and the Moineau pump motor, $K_T$, is plotted in Figure 4.12-b. Both motors speed curves present the same pattern of the product flow curve. They follow their respective references with no noticeable error, due to their fast dynamics, when compared to the slower variation of the system variables (or indeed, the wind turbine rotational speed) and therefore, they are not shown here. This again supports the relatively simple inverter models used in this study.
4.2.3. Remarks on the Clark Pump Configuration

One important conclusion regarding the overall system design is that despite its rating, the 2.5 kW Proven wind turbine may be too small for the RO system proposed, if the normal operation at lower wind speeds is considered.

The plunger pump modelled in Chapter 3 and used in this system performance study as the high-pressure injection pump was already available in CREST, from a previous project ([Thomson, 2001]). As its rated flow rate was much higher than the required for it to run as the injection pump, its driving induction motor (a 4-pole 3kW machine also reclaimed from the same project) would have to operate at extremely low speeds at the expanse of a much reduced efficiency. Therefore, a toothed pulley and belt arrangement was used in the simulation (with an assumed efficiency of 90%) to increase the motor operating speeds. Despite that, the pump still operated at very low speeds and it is believed that the use of an even smaller model (CAT 217) with a suitable driving motor could bring some improvements, albeit marginal, to the overall system efficiency.

Even so, the overall specific energy values found seemed remarkably low when compared with the typically quoted 5 kWh/m³ or more, achieved by other systems with energy recovery.
4.3. Hydraulic Motor System

As described in Chapter 3, a hydraulic motor was also considered for the recovery of the concentrate stream hydraulic energy. An alternative configuration to the one using the Clark pump was proposed, modelled and simulated. This alternative configuration aimed at achieving the same design objectives of the Clark pump system, namely:

- Similar rated output;
- Ability to operate from a variable power input, e.g. a wind turbine;
- High efficiency (low specific energy) throughout the operating range.

The power supply end of this alternative configuration was very similar to the Clark pump system, as seen in Figure 4.13. The main difference resides in the energy recovery mechanism: the hydraulic motor.

![Figure 4.13 - Hydraulic motor system diagram](image-url)
The operational speed of the hydraulic motor is well above the motor-pump speed range, therefore, a pulley-belt arrangement was used to connect its shaft to the motor-pump.

The main high pressure pump is the plunger pump, which is directly connected to the motor shaft. The Moineau pump ("Boost Pump" in the diagram) is used to provide the feed stream to the plunger pump. The RO array configuration is the same as used in the previous system - four in-line modules.

The hydraulic motor model uses the rotational speed (imposed by the motor) and the concentrate stream flow as inputs. The shaft torque and back-pressure to the last RO module are the outputs. Since both the high-pressure plunger pump (in the feed stream) and the hydraulic motor (in the concentrate stream) are positive displacement devices, the system recovery ratio should be constant and directly adjusted by proper choice of pulley ratios (assuming constant hydraulic efficiency for both). In practice, the recovery ratio varies significantly due to the distinct behaviour of their hydraulic efficiencies over the operating range. A gear ratio of 1.6 between the plunger pump and the hydraulic motor speeds was chosen and used in the following analysis.

4.3.1. Control Strategy

In this configuration, the theoretical recovery ratio is fixed by the ratio of the pulley/belt connection. Therefore, the system only controllable variable is the feed flow, or the high-pressure pump rotational speed (Figure 4.14).

The objective of the control strategy is simple: to try and optimise the energy extraction from the wind, using the optimal $C_p$ curve "$\text{Id}\text{c}_{\text{ref}} \text{ vs. } \omega$", by adjusting the plunger pump rotational speed.
Although in this configuration there are no means to directly control the feed pressure, a careful design procedure could greatly improve the system performance. This procedure would take into account the variation of the characteristics of both the feed water (specially the temperature) and the wind, for proper choice of pulley ratios. It is possible even to have scheduled pulley changes with a view to adapt the system to the seasonal variation of the natural resources.

4.3.2. Performance Predictions

As with the Clark pump system, the performance predictions for the hydraulic motor system are taken from a Simulink model. The input wind time series and the feed seawater characteristics are the same used in the Clark pump system simulation.

4.3.2.1. Long-Term (Steady-State) Performance

Similarly to the Clark pump system, a system model was built using steady-state equations in the components modelling. The water production as a function of the wind speed is shown in Figure 4.15, and the expected specific energy can be seen in Figure 4.16. From the specific energy curve, it becomes apparent that the hydraulic motor system is less efficient than the Clark pump configuration, presenting slightly higher energy consumption over most of the wind speed range and considerably higher values at lower power inputs.
4.3.2.2. Short-Term (Dynamic) Performance

The input wind speed time series is plotted in Figure 4.17-a. It is the same series used in the analysis of the Clark pump system. Figure 4.17-b shows the product water flow. Once again, the relation between both curves is clear, but less water is produced for the same wind input in the system with the hydraulic motor, as expected from the graph in Figure 4.15. An average flow rate of 3.5 m$^3$/day was found, just over half as much as the estimated value for the Clark pump system.

Figure 4.17 - a) Wind speed time series, b) Product flow rate
In fact, the product flow is interrupted during the dip in the wind speed at \( t \approx 200 \) s as the trans-membrane pressure becomes negative (Figure 4.18-a). The considerably lower pressure developed is responsible for the decrease in product water flow shown above, as well as the increase of its concentration (average of 730 ppm) as shown in Figure 4.18-b.

![Figure 4.18 - a) Feed and osmotic pressures, b) Product concentration](image)

The slightly inferior energy recovery performance of the hydraulic motor system, when compared to the Clark pump results, can be attributed to the cascaded losses in the hydraulic motor, pulley-belt connection and the high-pressure pump. This becomes particularly important at reduced power input conditions, because the hydraulic motor is actually designed to operate at substantially higher pressures (up to 140 bar).

With the hydraulic motor, the recovered hydraulic energy is firstly converted to mechanical energy (shaft torque), which is then used to drive the main high-pressure pump. Thus, the efficiency of the main high-pressure pump (and any other components in the line) must be included when considering the efficiency of the overall energy recovery process. Although the high-pressure pump is very efficient (80% plus), the combined efficiency is considerably lower than for the Clark Pump system.

As previously stated, the system should present approximately constant product recovery ratio. Nonetheless, the recovery ratio found was substantially variable, as depicted in Figure 4.19-a.
The wide variation of the system overall efficiency has a direct impact on the energy consumption, as indicated by the system specific energy, plotted in Figure 4.19-b. Because the specific energy is calculated from the ratio between the instantaneous power and the product flow, as the flow approaches zero, the specific energy tends to infinity. In the curve shown, this ratio is limited to 20 kWh/m^3. Values greater than this are plotted as zero. The curve shows considerable variation of the specific energy values within the system operating range, but even so, these values are quite respectable for a system of this size.

![Figure 4.19 - a) Recovery ratio, b) Specific energy](image)

At this stage, it should be mentioned that the design of the RO system (with four in-line modules and with the employed pulley-belt ratio) was not “optimised” for the hydraulic motor. Therefore, it is possible that the odd peak in the specific energy be smoothed out by a more careful design, although it would be very difficult to further reduce its minimum value and get much below the 5 kW/m^3 mark. It is also worth mentioning that the use of the Moineau pump as the boost pump may not be the most efficient way of conveying the feed water from the feed tank or well to the high-pressure pump. As the plunger pump does not require that the feed be pressurised at its suction inlet, as does the Clark pump, any other pump able to deliver the feed more efficiently could be employed.
On the control side, Figure 4.20-a shows the turbine speed values used to generate the reference DC current, which is compared to its actual (simulated) value in Figure 4.20b. Because of its comparatively increased energy consumption, the hydraulic motor system can still load the wind turbine under the higher wind speeds present at $t \approx 250$ s without saturating. As a result the DC current error is very low - RMS error of 0.13A – as seen in Figure 4.20-c.

![Graph of turbine speed, simulated DC current, and DC current error](image)

**Figure 4.20 - a) Wind turbine speed, b) Simulated DC current (reference and actual), c) DC current error**

As with the Clark pump system, the efficacy of the control can be observed in Figure 4.21, the wind turbine $C_p$ vs. $\lambda$ curve, where once again operation within the region surrounding the peaks of the curve is verified.
4.3.3. Remarks on the Hydraulic Motor Configuration

Although not nearly as promising as the results found for the system using the Clark pump, the simulated performance for the hydraulic motor system was nevertheless impressive. It was shown to be viable over a wide range of input powers (from 400 up to 1,700 W) with energy recovery values as low as 5.1 kW/m³.

Similarly, system performance would greatly benefit from the use of a wind turbine of higher rating. Also, a more detailed design could lead to a smoother system specific energy curve over the operating range, although it is believed that its minimum value may not be easily reduced.

The simplicity of the configuration is its forte, as well as its main disadvantage. The use of a single variable speed drive makes the configuration more robust but the pulley-belt coupling between the hydraulic motor and the high-pressure pump takes away the possibility of controlling the system recovery ratio independently during operation. Notwithstanding, scheduled pulley changes can be programmed to fine-tune the system operation and allow for seasonal changes in the available wind and feed water characteristics.
4.4. **Final Remarks on the Proposed Configurations**

Although preliminary, the simulated performance predictions of the two candidate configurations suggested that both systems could meet the established design objectives of a small desalination system, able to operate from a variable power input and with high efficiency throughout the operating range. Unfortunately, due to the great number of non-linearities and constraints imposed in the model, the simulation results obtained are not realistic at very low wind speeds (low feed pressure). Hence, the slightly higher wind speeds used in the systems simulations presented.

The low figures found for the specific energy in both systems emphasise the importance of assessing components efficiency individually, in an attempt to minimise system overall losses. The concern in selecting these additional components showed to be worthwhile.

The next step consisted of selecting one of the configurations for laboratory prototype testing. This would provide not only a better understanding of all the individual processes involved and their integration (the wind turbine control, operation of the variable speed drives, the desalination process itself), but also a deeper insight on the practical issues of implementing such a system in a “real world” situation.

The Clark pump system was chosen for implementation and further analysis.

It not only presented better efficiency (lower specific energy) throughout the operating range but also showed greater flexibility in the control, despite being the more complex of both systems. The details of its implementation are described in Chapter 6.

4.5. **Summary**

Two candidate systems were presented. One using a Clark pump and another using a hydraulic motor as the concentrate stream hydraulic energy recovery devices. Both configurations aimed at meeting the design requirements of operation at variable power input and low energy consumption.

The integration of the several components - which were individually tested and modelled - and their performance over a wide operational range could be established through simulated performance predictions. A control strategy was proposed for each of the systems. These strategies aimed at improving system efficiency for the integrated...
system of energy source (wind turbine) and sink (RO unit).

Although preliminary, the simulation results obtained suggested that both configurations could be viable alternatives for the desalination of seawater using renewable energy sources.

The performance predictions indicated clearly in favour of the Clark pump as a more efficient system. Even so, it is believed that the gap between both systems can be shortened by a more careful design of the hydraulic motor system.

A more detailed assessment of the Clark pump system is only possible after prototype testing, which is presented in the next two chapters. It is believed that prototype testing can provide a better understanding of the system performance, particularly considering the transients present in a system supplied by an extremely variable and unpredictable source, such as the wind power in a turbulent site.

Finally, it should be mentioned that energy recovery – using turbines, reverse running pumps or other devices - is a relatively common practice in bigger systems. Improved performance in such systems is also expected due to the improved efficiencies associated to the higher rating of individual components (especially motors and pumps). These higher efficiencies are not easily achieved in smaller components of the sort being used in this research.
CHAPTER 5
MODEL REFINEMENT AND VALIDATION

The commissioning of the prototype provided important data characterising the test rig, linking model development with the experimental part of the work. Validation of the developed model and refinements to the controller were undertaken alongside these initial experiments.

During the commissioning stage of the Clark pump system, a few test runs were carried out to verify the integration between the rig components and also to check the suitability of the developed system model. These initial runs provided important data characterising the implemented system, particularly related to the controllers. This data was also used to help refine the model. Also, a test run of the system without the wind turbine and control system was performed to validate the performance of the model against the prototype system. Both studies are presented in the following sections.

5.1. Controller Performance

The same controller gains used in the system modelling stage were also used as a starting point for the prototype commissioning, but it was soon realised, empirically, that they were not the best choice for the implemented system. This was mainly due to the fact that the control implementation was significantly limited in speed by the hardware. As a result the control loop could not be executed fast enough in practice and
this created a considerable difference between the continuous control simulated by the model and the discrete control action which was implemented. Rather than undertake a formal, mathematically based stability analysis a more practical control tuning process was followed based on observation of the prototype performance and subsequent adjustment, as described below.

Despite the efforts to implement the data acquisition and processing routines as fast as possible in LabView, the smallest time achieved in the execution of one loop of the control system was 0.5 seconds. The sampling frequency was maintained at 1 kHz so that rapid system dynamics could be observed in the data, but for control purposes the data was averaged every 500 samples. As the wind speed could not be controlled so as to provide step inputs to the prototype, the computer model had to be used as the reference. It was modified and a zero-order hold was introduced to represent the actual control action. The gains were adjusted and a simple sensitivity analysis of the control with this 0.5-second zero-order hold was then carried out. These gains were then verified against the performance data from the prototype.

![Figure 5.1 - DC current controller performance with discrete actuation and different integral gain parameters](image-url)
The simulation runs used in the analysis consisted of a constant input wind speed of 7 m/s, which at time $t = 10$ s was stepped up to 9 m/s and then at $t = 20$ s stepped down again to 7 m/s. The system response showed greatest sensitivity to variations in the integral component of the controller, so the proportional gain was kept constant ($P = 20$) during the simulations.

Figure 5.1 shows the performance of the modelled system DC current controller for different values of the integral gain $I$. A value of $I = 300$ was used in the simulated continuous model of the system (Chapter 4), but this was found to be too slow to achieve effective control in the implemented prototype system. It was then increased to $I = 380$ and the improvement in performance was evident. The prototype system was clearly able to react better to the rapid variation of the incident wind speed. It was also noticed during the experimental tests that if the gain was further increased, the controller would become highly unstable and unable to properly follow the reference signal. An attempt to reproduce this behaviour in the simulated system can be also seen in Figure 5.1, where a integral gain of $I = 460$ is used.

The value of $I = 380$ was than adopted for use in the prototype experimental tests presented in the next chapter.

5.2. Model Validation

In order to validate the developed model, a comparison between the RO rig performance and a simulation run with similar inputs was performed. Due to the lack of wind speed signal during the testing of the prototype, the wind turbine was not included in the following analysis. This was not considered a major drawback as the wind turbine model performance had already shown good agreement against the manufacturer's data (Section 3.3).

During the test, the prototype was powered from the grid and step variations in the speed of the induction motors driving the positive displacement pumps were applied to obtain different operating conditions. The feed water was kept at approximately 25°C and with a concentration of 32,000 ppm NaCl. After performing the experimental test, a simulation run was devised so that the motors would follow the same speed pattern, with the feed water at the same temperature and concentration of the experimental tests. The test used two speed settings (high and low) for each motor/pump. Both pumps
started at the lower setting and after permuting the speeds between both settings, the motors should return to their initial operating conditions (lower speed).

The experimental (EXP) and the simulated (SIM) rotational speeds for the Moineau pump (Figure 5.2) and the plunger pump (Figure 5.3) can be seen below.

![Figure 5.2 - Moineau pump speed (experimental and simulated)](image1)

![Figure 5.3 - Plunger pump speed (experimental and simulated)](image2)

The feed and product water flow rates can be seen in Figure 5.4. The difference between the experimental and simulated values is almost negligible, with a more noticeable difference in the feed flow. This difference is directly reflected in the system recovery ratio, shown in Figure 5.5.

![Figure 5.4 - Feed and product flow rates (experimental and simulated)](image3)

![Figure 5.5 - Recovery ratio (experimental and simulated)](image4)
Figure 5.6 shows the simulated and experimental outputs for the product water concentration. Here the difference is more noticeable, in the range of 10%. This is mostly due to the difference in the input pressure to the membrane array (the osmotic pressure is calculated for both experimental and simulated tests), as seen in Figure 5.7.

![Figure 5.6 - Product concentration (experimental and simulated)](image1)

![Figure 5.7 - Feed and osmotic pressure (experimental and simulated)](image2)

The discrepancies shown in the previous figures between the experimental and simulated results cannot be attributed to one single reason, as indeed shouldn't any other mismatches between both results be. The flow rates of the pumps are a reflection of both their speed and output pressure - and possibly other variables not included in the model (such as the feed temperature), though in a smaller scale. Similarly, the feed pressure is a reflection of the behaviour of the membranes, which in turn is dependent on the feed water characteristics. Because the system has four cascaded membrane modules, any errors small as they may be, present in the membrane model are propagated throughout the array and the rest of the system.

As previously stated, the model of the RO membranes was obtained from tests carried out on one single module, when the membranes were new. Also, according to the manufacturer, a variation in performance of ±15% can be expected between modules and their performance vary considerably with use. This variation in membrane characteristics is thus the most likely explanation for the discrepancies shown in Figure 5.6 and Figure 5.7.
Finally, the specific energy is depicted in Figure 5.8. It is one of the most important performance indicators of the system, relating the electrical power consumption with the product water output. The most significant conclusion that can be drawn for these results is that the prototype maintains its energy efficiency throughout a wide range of operating conditions, as designed. And despite being slightly less efficient, the model corroborates this performance characteristic.

Figure 5.8 - Specific energy (experimental and simulated)

5.3. Summary

As the rig was being commissioned, it was realised that some adjustments to the model and the test rig would be needed, specifically in the implementation of the controller, which had to be adjusted to allow for the slower rate of execution of the control loop used in the actual system.

A zero-order hold was introduced in the model and step response tests were simulated, using the wind speed as the input parameter. The performance of the rig was interactively assessed against the simulation using the same gains. It was empirically concluded that by using an integral gain of \( I = 380 \) in the rig controller a satisfactory compromise between fast response and stability could be achieved.
A model validation test was also devised and carried out. The induction motors speeds were used as the inputs for the test. These were varied between two fixed settings (low and high) so that the performance of the rig could be analysed for a range of conditions. The match between experimental curves and the simulated results was very good. This was somehow expected because of the results obtained from the individual components models. Nevertheless, some significant differences in the results were noticeable and it is believed that these could be attributed mostly to errors in the membrane models resulting from use dependent changes to the membrane characteristics.
CHAPTER 6
EXPERIMENTAL ANALYSIS

In order to improve the overall understanding of the Clark pump system modelled in Chapter 4, a prototype system including the pump was designed and built. Constructive characteristics and implementation issues are presented. Experimental results are shown and analysed in light of the expected system behaviour derived from the performance predictions previously discussed.

6.1. The Test Prototype

The prototype used in this work's experimental analysis is a resource which was also used in another concurrent project at CREST ([Thomson, 2003]). As such, its design and construction were shared tasks.

The test prototype can be divided into two main groups of components: the instrumentation, data acquisition and control system; and the reverse osmosis rig. These components are described in more detail in the next sections. A schematic diagram of the complete prototype is depicted in Figure 6.1.

A wind turbine simulator was also designed and built. Unfortunately, it had not yet been fully commissioned by the end of this work and therefore no experimental results were available. In spite of that, it is briefly described in Section 6.1.2.
The power input to the prototype can be selected from three different sources: the Proven wind turbine, the wind turbine simulator or the grid. The latter is of particular importance for the commissioning and characterisation tests that need be carried out on the reverse osmosis rig or any of its components.

Figure 6.1 - Schematic diagram of the test prototype
The electrical end of the system, from the power supply input to the inverters, was located separately from the RO rig itself due to safety considerations and all the relevant variables were measured using carefully chosen instrumentation. Further details on the variables measured and the sensors used can be found on Appendix B.

Proper feed water temperature control was needed for the membranes characterisation tests, and therefore a domestic immersion heater was fitted to the feed tank and a heat exchanger introduced in the concentrate stream. Both product and concentrate streams were recombined in the feed water tank so as to provide a continuous constant salinity feed.

6.1.1. Instrumentation, Data Acquisition and Control System

The instrumentation, data acquisition and control system (hardware and software) is responsible for the supervision and control of the prototype. Dedicated software manages the data acquisition, processes the data and also executes the control loop and outputs the control variables. All the sensors were chosen to meet the specific requirements (e.g. sensors in contact with the salt water had to be made of stainless steel) and operating range of the variables to be measured.

The sensors outputs are connected to a signal conditioning board, which in turn is connected to the data acquisition (DAQ) board in an Intel-586 PC. Due to the number of variables sampled, two 16-channel DAQ boards were used. They were supplied by National Instruments (PCI-6024E) and have an A/D conversion rate of 200 kSamples/s, with 12-bit resolution. In order to keep the data acquisition synchronised in both boards, they were hard-wired to each other through a comms cable, which allowed the triggering of one of board by the other, through the DAQ software. In addition to the analog inputs, the DAQ boards have two analog outputs each (also 12-bit resolution). These are used to control the wind turbine simulator (torque reference), the Buck converter (when operated as a controlled load, for the wind turbine modelling) and the two pumping units (speed reference to the inverters) of the RO rig.

The supervisory software code was written using LabView, also from National Instruments. It manages the DAQ boards, acquires and processes the data, runs the control loop and stores the data files in the PC hard disk. It also provides a front panel from which one could manage the data acquisition process, actuate on the prototype
(anologue outputs) and visualise any of the system variables. A picture of the supervisory software front panel is shown in Figure 6.2.

Each variable was sampled at 1 kHz. At every 0.5 seconds (500 samples) all the variables were averaged and some processed for use in the control loop, which was also executed at this same rate (every 0.5 s). The variables were then stored in a buffer of a chosen size. When the buffer was full, e.g. after 20 loops (every 10 seconds), the variables were again averaged, processed and written on the PC hard-disk. This was implemented so that the control loop could run at a faster rate than the data processing and logging, if required.

![Supervisory software front panel](image)

**Figure 6.2 - Supervisory software front panel**

### 6.1.2. Wind Turbine Simulator

A wind turbine simulator was designed and built so that the RO rig could be tested independently of the available wind. Unfortunately, for a number of reasons
outside the authors control it was not ready for use when this work was written. It would have been particularly important for the performance analysis and consequent improvement of the control scheme, due to the fact that different wind speed time series with specific characteristics could have been repeated over and again.

An important design choice was the representation the wind turbine rotor inertia in the simulator. A standard 30kW variable speed drive was used to drive the wind turbine generator. Since the induction machine rotor inertia was much lower than the turbine rotor inertia, the difference between both had to be allowed for if the simulator was to have the same dynamic behaviour of the wind turbine. The options were either to add a fly-wheel with the desired inertia to the driving shaft or to introduce the lower inertia effect in the control software so that the reference torque signal to the drive could compensate it (in effect, be reduced).

The former option was chosen. A new 2.5 kW wind turbine generator was kindly supplied by Proven Engineering. The fly-wheel was designed based on the wind turbine rotor and the driving induction motor inertias ($J_{WTR} \approx 6.33$ kgm$^2$ and $J_{IMS} \approx 0.4$ kgm$^2$, respectively). Given the desired inertia ($J_{FW} \approx 5.93$ kgm$^2$), the density of steel ($\rho_s = 7.800$ kg/m$^3$), the inertia of a rotating cylinder ($J_{Cyl} = 0.5 \rho H \pi R^4$) and the driving shaft height constraint (the radius of the generator), a cylindrical fly-wheel was built and connected to the simulator shaft, between the motor and the generator. The fly-wheel design dimensions were: thickness equal to 10.0 cm and diameter 52.6 cm.

6.1.3. RO Rig

The disposition of the components and sensors onto the rig frame was carefully planned, based on the experience acquired during a previous project ([Thomson, 2001]). It was decided that flexible plumbing should be used to facilitate changes in the connections between components. High-pressure stainless-steel braided flexible hoses were used throughout. Being it a prototype, a didactic approach was employed to attempt an uncluttered and functional distribution of components. The result can be seen Figure 6.3.
The main components and sensors were placed at one side of the frame while the feed tank, heat exchanger and signal conditioning board, on the other. Both pumping units were placed on the frame base, while the four membrane modules were displaced on top of the backplane. The instrumentation and plumbing connectors went on the lower part of the same backplane.

### 6.2. Experimental Procedure

Unfortunately, due to time restrictions, the experimental performance curves here displayed are only preliminary results, obtained in late May 2003, at the end of the windy season in Loughborough. At the time these tests were carried out, the system had just been commissioned and further tests were required for better understanding of its operation and, specially, fine tuning of the control strategy.
Nevertheless, many lessons were learnt from these tests. Some of these are further related below, before the presentation and discussion of the experimental results.

6.2.1. Remarks on the Experimental Setup

The most important realisation during the tests, which had already been suggested by the simulation results, was that the Proven turbine rating was not ideal for a system this size (not at the Loughborough site, at least). Clearly the problem was not related to its rated output (2.5 kW), which was more than enough, but with its output at average wind speeds. This simple fact made the commissioning and testing of the system all the more challenging, as considerably higher wind speeds were needed. A larger rotor, giving a lower rated wind speed, would clearly have been preferable.

Another important realisation related to the speed of execution of the control loop. The execution period of half a second per loop was apparently too long, particularly taking into account the high levels of turbulence at the wind turbine site nearby CREST. The final version of the data acquisition and control software had many unnecessary features inherited from earlier versions and the PC was not able in practice to cope with faster loop rates. This problem could be solved by redesigning and optimising the software with views to increasing its execution speed.

The presence of noise (EMI) in some of the sensors output signals was also very noticeable, initially. Due to the number of sensors installed and the use of two industrial inverters plus the purpose built Buck converter it was very difficult to keep power and signal cables completely apart from each other. Low-pass filters were used in the software, for some of the variables (although of course this does degrade control response) and the earthing reinforced at many points on the rig to try and reduce this effect. Such measures considerably improved the quality of the acquired data, albeit not absolutely.

The connection of the Buck converter to the DC bus, as a voltage limiter, in parallel to the inverters also brought up an unexpected problem. As soon as the voltage at the DC bus rose above the preset limit (conservatively set just under 370 V), and the Buck converter started switching to clamp the DC voltage, one of the drives would trip with an “output current overload” message. Apparently, the current transients caused by the switching at the inverter input were upsetting the inverter current measurement and
protection system at its output. The solution adopted was to introduce a choke between the Buck converter and the inverters, smoothing out the DC current waveform and preventing the drives from tripping.

In addition to the maximum input voltage limit, the drives control software also had a minimum voltage detection feature, which turned them off every time the DC voltage went below 230V.

Although the energy content at such low voltages (or wind turbine speeds) was not itself significant, the drives required a couple of seconds to become functional after being restarted. Depending on the wind conditions, a gust could suddenly speed up the turbine and before the drives could be started, the Buck converter would switch in. Despite the fact that the choke introduced in the DC link prevented the transient currents to trip the drives, some spurious currents could still be detected by the sensors in the DC link.

As described in Chapter 4, the DC current was used in the feedback loop of the control algorithm. These undesired spurious readings would mislead the control, resulting in reduced speed demand to the drives and wasted wind power. Eventually, the drives would speed up, the turbine would be loaded and the DC voltage would drop, causing the Buck converter to stop switching. In situations where this was followed by another wind gust, this time with the motors already running, the reaction of the control was fast enough to accelerate them and properly load the turbine as expected. It is possible that the use of inverters with higher input voltage limits would circumvent the problems created by the interaction with the voltage limiting converter.

Finally, the use of a pulsation dampener in the medium pressure feed line (between the Moineau pump and the plunger/Clark pumps) had been anticipated, as shown in Figure 6.1. Unfortunately, it was not sized sufficiently to properly absorb the pressure transients caused by the shifting of the Clark pump piston at the end of each stroke. The pressure peaks were reflected all the way back to the DC link, as small current spikes, which had a further unwanted impact on the operation of the control loop. This effect was greatly reduced by the introduction of an improvised dampener (devised by Murray Thompson) connected at the Clark pump inlet. The dampener was built from an empty 2.5-inch RO pressure vessel with a bicycle tire inner tube, pre-
charged to about 4 bar.

6.2.2. Experimental Results

From the limited experimental data collected, a 30-minute window was selected and displayed below for a selection of key system variables, for illustrative purposes. Figure 6.4 shows the wind turbine rotational speed and the product flow rate. The feed pressure and the calculated osmotic pressure can be seen in Figure 6.5-a, while the product concentration is shown in Figure 6.5-b. The feed water concentration remained around 31,500 ppm (representative of seawater) and the temperature, just under 25 °C during the test. As previously stated, the data was sampled at 1 kHz and, in the results here presented, variables were recorded every time the control loop was executed, i.e. every half a second.

Within this half-hour window, a 5-minute zoom (750 sec < t < 1050 sec) was selected in the presentation of the results for better visualisation of the variables.

Unfortunately, the anemometer had been damaged a few days before these tests were carried out and could not be fixed in time for this experiment. Hence, no wind speed data was available, which is regrettable as it precludes a direct appraisal of the turbine maximum $C_p$ tracking algorithm. The wind turbine rotational speed was used instead for comparison against the product flow rate, as shown below in Figure 6.6.
As suggested by the simulation results, the correspondence between both curves is clear. Here, the RO membranes performance should be given special attention. Compared with the values found when they were modelled, the desalinated water production showed considerable reduction. At the time the characterisation tests were performed, the membranes were new and almost six months - of not particularly careful use - had passed since.

Some reduction in the product flow rate is normally expected during the first days of operation, as a consequence of membrane compaction and settling. A simple verification of what would be expected of the membranes, under the same input conditions, using the developed models indicated a reduction of about 30% in the product flow rate. This realisation stressed the importance of the practical issues of the RO technology, which would need further attention in a possible field implementation of the system.

The feed stream pressure and the osmotic pressure calculated from the feed water concentration and temperature can be seen in Figure 6.7-a. The resulting product water concentration is shown in Figure 6.7-b. An interesting phenomenon was observed, related to the dynamic behaviour of the membranes under the unsteady operating conditions they were submitted to. If the product flow was uninterrupted, or even if it was only discontinued very briefly as it happened during these tests, the product
concentration would not suffer great variations. However, if the system had stopped for a longer period of time (the exact value was not formally assessed), the standing pressure would drop considerably and upon system restart, the product concentration would overshoot to higher values (between 2,000 and 3,000 ppm), taking a few of minutes until it returned to normal operation. This may be explained by the fact that once the pressure on the membrane decreases, due to leakage that has not been modelled, to a level below the osmotic pressure, natural osmosis will occur.

![Graphs showing feed and osmotic pressures and product stream concentration](image)

*Figure 6.7 - a) Feed and Osmotic Pressures, b) Product stream concentration*

This somewhat emphasised the importance of keeping the feed flowing, even if at lower pressures. This remark, added to the fact that the inverters would shut down at lower voltages, led to the adoption of an overall system management procedure to try and keep the system running for as long as possible, albeit at low outputs, even if at the expense of reduced wind energy capture. A series of “flags” introduced in the control software was used to determine the current operation mode at any moment, allowing the implementation of the procedure described below.

When the system was switched on, the control would wait until the DC voltage rose above 300V to start the Moineau pump alone, at a constant speed of 200 rpm. If the DC voltage did not drop below 300 V for a couple of seconds, then it would switch the reference speed to the Moineau pump over to the output of the PI controller and start the plunger pump using the speed ratio defined by the control function already described.
The system would operate in this mode until the DC voltage dropped below 290 V, when the plunger pump was stopped and the Moineau pump was operated alone, still following the reference provided by the PI controller. If the voltage rose again above 300 V, the plunger pump would be restarted and if dropped below 270 V, the reference to the Moineau pump would be once again switched to the constant value of 200 rpm, to try and keep the system running with very low energy consumption, not stalling the wind turbine and preventing the DC voltage to drop even further, to the point of having the drives switched off. If, however, the voltage kept dropping, and reached 240 V, the Moineau pump would stop to unload the wind turbine and allow it to speed up again. The flags would be reset and the system would wait until the voltage reached 300 V once again.

Despite the unsteady conditions introduced by often starting and stopping the motors, this pragmatic procedure was found to be very useful in keeping the system running continuously. It did however result in frequent operation of the system out of the optimal targets previously defined. Therefore, system overall efficiency could greatly benefit from further analysis and development of this management procedure, in order to establish a balance between performance within the control targets and continuous operation.

Irrespective of that, system performance was very satisfactory. The product water recovery ratio and the system specific energy consumption can be seen in Figure 6.8. Both curves values are calculated from the instantaneous readings of flow and power data. This calculation approach accounts for some of the spurious values depicted.

In Figure 6.8-a the recovery ratio curve was calculated by the ratio between the product and the total feed flow. Therefore, when the feed flow is greatly reduced or even interrupted, the product may still be flowing due to the standing pressure in the membrane modules. This would cause the recovery ratio value to display unrealistic values in excess of 50 % product recovery.
A similar explanation applies to Figure 6.8-b, the specific energy consumption, which is the ratio between the instantaneous DC power and the product flow rate. Although very efficient, the inverters consumed some power in stand by mode. So at “zero” or very low product flow rates, the specific energy would show extremely high values. These were limited to 20 kW/m³ in the figure above. In spite of that, stretches of operation at values below 4 kW/m³ can be seen in the graph, which corroborates the predictions derived from the simulation results. Likewise, operation at values much above 5 kW/m³ can be observed, which are related to the operation of the Moineau pump alone, when the plunger pump was not running.

The reference and the measured DC current values, used by the control to generate the reference speed to the Moineau pump, are plotted in Figure 6.9. When zoomed in further, some delay can be noticed in between both curves.

Some of it can be attributed to the slow execution time of the control loop, i.e. at any instant, the recorded values will be the reference demand at that instant and the measured values that resulted from the actuation in the previous execution. Fortunately, the wind turbine rotor inertia helps by not allowing the turbine speed, and hence the reference current, to suffer great variations abruptly.
It is also important to mention that in order to protect the RO rig, especially the membranes, from surges in pressure and flow, the rate of change of the motors reference speed signal was also monitored and clamped to a maximum of 200 rpm/sec. This meant that, for example, if the output of the controller to the Moineau pump were saturated to its maximum (700 rpm in these tests) and the pump were stationary, it would take seven loops (i.e. 3.5 seconds) for the motor to reach the setpoint speed. This value was not optimised, but empirically chosen since the system seemed less stable when higher variations were used. Clearly, a higher speed demand to the drives strongly affected the DC current, which in turn reflected in the control as another high demand. The reference and measured speed curves for the Moineau pump can be seen in Figure 6.10.
Finally, Figure 6.11-a shows the instantaneous DC power during the test. As expected, the system was able to operate over a wide range of power inputs. As explained in Chapter 4, the DC power was used to calculate the speed ratio between the motors driving the pumps (plunger over Moineau), $K_T$, which is seen in Figure 6.11-b. Here the unsteady operation of the system can be clearly appreciated by the number of start/stop cycles of the plunger pump (stopped at $K_T = 0$).
A prototype test rig was designed and built for the assessment of the Clark pump system configuration. Details of its construction, hardware and software, were presented, as well as experimental results obtained from tests carried out using the Proven wind turbine.

During the commissioning stage of the rig, a number of practical issues came to light. It was noticed that these issues were mainly related to the interaction between the specific components used, and by no means they refuted the validity of the proposed configuration. Although some of these problems had been anticipated to an extent, the complete picture only became clear after initial tests were carried out.

As previously suggested, and further verified by the results presented, system operation would greatly benefit from a turbine with a bigger rotor, especially at the Loughborough University site. Unfortunately, due to time constraints, further improvements and subsequent tests could not be carried out before the writing of this thesis. Such improvements would clearly include redesigning the data acquisition and control software, to allow faster actuation, and the inclusion of the wind speed data in the results. In spite of the improvements achieved by the management routine used in maintaining continuous operation of the system, it is believed that further testing would certainly bring additional benefits.

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Figure 6.11 - a) Measured DC Power, b) Induction motors speed ratio, $K_T$
Despite the adversities encountered, the system behaviour confirmed the expected overall performance, being able to operate under a considerably wide range of input power, with remarkably low energy consumption throughout.
CHAPTER 7
PRACTICAL ISSUES AND APPLICATION OF THE MODEL

Some general aspects regarding the practical implementation of RO systems are discussed. From these practical considerations, some remarks regarding the proposed system implementation in the field are drawn. Also, a simple case study exercise is conducted, using adapted long-term performance curves similar to those established in Chapter 4. Finally, a simplified cost analysis is presented.

7.1. Practical Considerations

Although most of the practical considerations here described are applicable to any RO desalination system, these are presented in this chapter, instead of in the general literature review, because they were thought to be also very pertinent to a possible field implementation of the proposed system. The considerations discussed below are taken from various sources, but are mostly based on the general guidelines suggested in [DOW, 1996] and [Lior, 1986].

Feed water quality (and adequate pre-treatment) is a key design aspect for any reverse osmosis system. Its characteristics are to reverse osmosis what the wind resource is to wind energy. Disregarding it may mean the difference between membranes that last years or weeks. Therefore, one of the first steps when considering siting an RO system is to proceed a complete analysis of the candidate water resource.
The two major problems regarding the feed water quality and membrane performance are fouling and scaling. Fouling is associated with the plugging of the membrane by colloids (particulate fouling) and bacteria or other living organisms (biofouling), whilst scaling relates to the precipitation of elements with little solubility onto the membrane surface.

Scaling represents a bigger problem for brackish water applications, where higher recovery ratios (up to 90%) are common and unusual salts may be present in the water. In seawater plants, the feed osmotic pressure acts as the limiting factor for the recovery ratio, so scaling is not usually a major problem. Furthermore, standard commercial anti-scaling additives can also be dosed into the feed should the need arise.

On the other hand, colloids, organic matter and bacteria abound in seawater and attention should be given to these if the membranes are not to be prematurely damaged. One simple way of identifying fouling would be by monitoring the pressure drop along the elements (feed pressure minus the concentrate pressure). A significant increase (above 10%) in this pressure difference could indicate membrane fouling.

Membrane particulate (clay, colloidal silica, organic substances, etc.) fouling should not present a problem, provided appropriate filtration is employed. Membrane bio-fouling prevention, however, may not be so simple. It consists of the growth of living organisms onto the membrane surface and, as they can reproduce given appropriate conditions, it can cause considerable reduction in the permeate flux in a short period of time.

Although very efficient, domestic ultra-violet (UV) pre-treatment units are not perfect. Therefore, it is possible that even a small amount of living matter not killed by the UV light could, given the time, cause membrane fouling, as they would feed off the dead matter in the water. The use of underground water, e.g. from a beach well, instead of surface water can substantially reduce the biofouling potential, and is strongly recommended. The use of UV radiation is recommended for small systems, but the feed water should be clear in order to improve radiation penetration.

Another approach would be to prevent these organisms to reach the RO membranes by isolating (filtering) and “growing” them in a confined area of the system. Additionally, a UV unit could be fitted further downstream in order to kill any remnant
living matter. In this context, the idea of using slow sand filters (filtration by gravity) was considered but despite the interesting philosophy, little practical experience was available to confirm it as a viable solution.

In view of the above, it is believed that the use of a properly designed beach well, in conjunction with cartridge filtration (for undissolved matter greater than 1 μm) and a UV unit, is preferable for the proposed system. It would provide the filtration of a great part of the organic matter through the beach sand and the remaining living matter could be killed by the UV unit. Recently, [Kunczynski, 2003] presented a very positive feedback on the use of this relatively simple configuration. Finally, chlorination of the product tank should be used to prevent reinfection of the stored product water.

Another operational matter of great importance for proper conservation of the membranes is the implementation of a flushing procedure whenever the system is stopped. System flushing consists of washing the feed-concentrate line at very low pressures to remove any settled material (living or not) onto the membrane surface. Clean feed or product (permeate backflush) water can be used for this. A small permeate collection tank placed above the rig and a set of controlled valves can be used to passively backflush the permeate thorough the modules.

In any case, care must be taken to prevent the membranes from getting in direct contact with the air. If air is sucked back into the system, the permeate side of membrane may be exposed to airborne organisms; the air may affect a subsequent start-up; the membranes may dry, affecting their performance in the longer term; the air (oxygen) in the membrane may cause reduction of ions present in the feed water and subsequent fouling.

The lack of a flushing procedure in the prototype rig together with operation at very low flow rates, may well account for the sudden drop in performance of the RO elements during the first six months of operation.

7.2. Performance Analysis

As already suggested by both the simulated predictions and the experimental results, the 2.5 kW Proven wind turbine does not have the adequate rating to supply the RO system. Therefore, an approximated wind turbine model, using a bigger diameter
rotor, was derived.

7.2.1. Modified Long-Term Performance Prediction

In the steady-state model of the system, a higher value (4.5 m) was used for the turbine diameter in the aerodynamic torque equation, instead of the actual diameter of the 2.5 kW turbine (3.5 m). The generator parameters were kept the same. It was not desired that this change in diameter affected the torque coefficient polynomial function, which is a highly non-linear function based on the tip speed ratio and the turbine rotational speed, as described in Chapter 3. It is understood that a bigger radius would imply in lower rotational speeds, if the tip speed ratio was to be kept the same. Therefore, in the calculation of the tip speed ratio, the original radius was kept. The increase in torque was balanced by a proportionally higher DC current reference.

By doing this, the model of a wind turbine with higher torque, but roughly the same rotational speed range of the original turbine was obtained. As the main interest of such an approximation was to produce a model that could generate more power at the same incident wind speeds, it was found to be very satisfactory. A cut-in wind speed of 3 m/s was established and a polynomial function was fitted to the power curve points. The power curve of the resulting wind turbine model can be viewed in Figure 7.1. As the new turbine model was simulated within the system, not separately, the curve shown is limited to the operating range of the system. Nevertheless, the values obtained were compared to the manufacturer's 6 kW model (5.5 m diameter) and were found to be realistic.

The approximated turbine model, used within the system model, produced a new long-term performance characteristic that represented the product flow against the average wind speed. The result was encouraging and is displayed in Figure 7.2. The system "cut-in speed" was set to 4 m/s and, as shown in the curve, the maximum output of the system happens at 8 m/s. A polynomial function was then fitted to the points for use in the following performance predictions. For wind speeds higher than 8 m/s, the system output saturates, and the values obtained for 8 m/s are also used at higher wind speeds. The particular turbine technology is assumed, like the Proven turbine, to continue generating at arbitrarily high wind speeds, in other words, there is no furling wind speed beyond which the turbine ceases to generate electricity for water
production.

\[ \text{Figure 7.1 - Estimated output power curve of 4.5 m (diameter) wind turbine and polynomial fitting} \]

\[ \text{Figure 7.2 - Estimated product flow with 4.5 m (diameter) wind turbine and polynomial fitting} \]

7.2.2. Output Estimates

The polynomial function derived from Figure 7.2 was then used in conjunction with a year-long hourly wind speed data series to produce a water probability curve for a given site. The wind speed data used is a set of 10-minute averages, recorded once every hour. The particular data set analysed was taken from an off-shore location to the East of the UK, during the year of 1995.

The first step was to represent the 8,760 data points of the hourly wind speed time series (Figure 7.3) in probabilistic terms. The data was binned in 1 m/s bands, and represented by the Weibull probability distribution function, as shown in Figure 7.4. An average wind speed \( U_w = 9.2 \) m/s and the Weibull parameters \( c = 10.26 \) and \( k = 2.025 \) were found for the given time series, using Matlab's \texttt{weibfit} function.

The product flow rate vs. wind speed polynomial fitting can be combined with this probability associated with each wind speed bin, producing the desalinated water probability curve. This curve can be presented either as the probability distribution, as in the wind speed curve above, or as the cumulative probability distribution. Based on the this curve, the probability of producing a certain amount of water can be established and used either in a viability study or in a design analysis of a new system. This is further discussed in the next section.

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The next step in the analysis was to effectively use the hourly (10-minute averages) wind speed time series data (Figure 7.3), applied to the water output function (Figure 7.2), to produce a water production flow rate time series. This analysis was carried out according to the procedure described below.

Initially, the total production of one year was calculated, by applying the wind data to the water output function. The average daily output was then calculated (total volume over 365 days) and the value of 10.4 m$^3$ was found. Then, a storage tank size was defined and a daily consumption of 10 m$^3$ assumed. The first 24 hours of wind data were used to calculate one day of water production. At the end of the day, 10 m$^3$ of water (daily consumption) were deducted from the total amount produced and the remaining, left in the tank.

Following this sequence, a whole year was simulated for that chosen size of tank and the system reliability calculated. The reliability was defined as the ratio between the number of days that the tank was not empty after having the demand deducted and the total number of days, 365. After employing this procedure for different storage tank sizes (at 10 m$^3$ steps), a curve representing the system reliability against the tank size was produced.
The result is seen in Figure 7.5-a. Clearly, it follows a diminishing returns pattern and in order to support the decision of which tank size to use, Figure 7.5-b was generated. It shows the variation, or the gain, in reliability for one tank size in respect to the immediately smaller one. So, for instance, having a 10 m$^3$ tank makes the system almost 20% more reliable than not using a storage tank, and so on. Taking this into account, a tank size of 30 m$^3$ was chosen (only 1% gain in reliability would be obtained if a 40 m$^3$ tank was used). The simulation was run again using this size and the resulting volume in the tank (water level), calculated daily, is shown in Figure 7.6. Over one year, the demand was not met on 37 days, representing a reliability of almost 90%. Given this tank size, another important factor is the water usage, i.e. the amount of water actually used (not spilled out of the tank, when it was full) over the total water produced. A water usage of 92.6% was found for a tank this size.
It is important to remember that the desalinated water vs. wind speed curve (Figure 7.2) was originally produced considering a feed temperature of 25 °C and a feed water concentration of 32,000 ppm. Although the feed concentration might be taken as a constant for one specific location over a year, this will usually not hold true for the temperature. In this case, a rough approximation may be obtained by taking the known average feed water temperature (if the variance is not too high), or by generating a more complex function that correlates seasonal variations in the wind speed and the feed water temperature, or even using different product water vs. wind speed curves. Note that although monthly sea water temperatures are often available for coastal regions, the temperature of water from a beach well is expected to vary significantly from this and the supplied water will also be affected by the details of the pipe runs and whether they are exposed to solar gain.

Clearly, in such a system with a considerably bigger wind turbine, there will often be a surplus in the generated power. To make best use of this surplus, other deferrable loads such as water pumps could be added (to convey water to and from the system), or even a pre-heating element for the feed water, as membrane permeation increases with higher feed temperatures (at the expense of higher concentrations in the product). Certainly, the wind turbine rotor size (not necessarily its rating) may count as one of the
7.2.3. Economic Analysis

Taking into account the estimated performance prediction described above, a simplified economic analysis was carried out. A *Net Present Value* calculation was performed in order to provide a cost figure for the water produced by the system over its lifetime.

A lifetime of 20 years was established and the total amount of water calculated. The daily average production of 9.63 m$^3$ was used and an estimated system availability of 90% was chosen. The daily average was found by multiplying the water usage (92.6%) by the daily average considering the total amount of water produced (10.4 m$^3$/day). A yield of nearly 63,270 m$^3$ of desalinated water was found over the system lifetime.

In the system costs calculations, the actual price paid for the components modelled and used in the prototype were taken as a reference, with exception of the wind turbine. The list price of a 6 kW turbine (including the tower) was used instead. A conservative estimate was made for the costs of other parts, consumables and installation (£15,000.00) and added to the components price, producing a system total capital cost just under £33,000.00.

The most uncertain cost estimates were related to the system maintenance. It was assumed that the complete set of membranes (all four) would be replaced every year at a total cost of just over a thousand pounds. Also, an extra expenditure for other possible costs such as parts replacements, chemicals, etc. was allowed for, totalling £1,500.00 in maintenance costs per year.

Considering the capital costs, the additional maintenance costs for a period of 20 years and an interest rate of 5% p.a. a total net present value of just under £51,000.00 was found. Dividing this cost by the aforementioned predicted yield, the estimated cost of 0.80 £/m$^3$ was obtained for the water produced by the system.

Moreover, considering the same water usage figure (92.6%) obtained in the previous section as a reasonable value, a more generalised analysis can be carried out. As suggested in the previous section, the Weibull probability distribution can be used in...
conjunction with the system performance curve (product water vs. wind speed) to support in the system design assessment or viability study. In this context, Equation 7.1 can be used to calculate the mean water yield for a given probability function.

\[
\overline{Q}_p = \int Q_p(U) \cdot p(U) \, dU
\]  

(7.1)

where

- \( \overline{Q}_p \) is the mean water yield;
- \( Q_p(U) \) is the product water vs. wind speed function;
- \( p(U) \) is the probability of the wind speed "\( U \)" to occur, from the Weibull distribution.

Different Weibull probability distribution functions were generated by assuming the shape parameter, \( k = 2 \), and varying the scale parameter \( c \), according to the annual mean wind speed, \( \overline{U} \) \((c = \frac{2 \cdot U}{\sqrt{\pi}}, \text{for } k = 2)\).

The average water yield for the different annual mean wind speeds was then calculated using Equation 7.1. The result can be seen in Figure 7.7. Using these values and by considering the same water usage found for the system analysed in the previous section (92.6%), the total yield of the system over its lifetime was estimated. Taking into account the previously stated net present value of the system, a curve displaying the cost of water as a function of the mean wind speed was produced, as seen in Figure 7.8.
The results obtained are very encouraging even at much lower mean wind speeds than the one simulated in the previous section, specially if compared to the costs suggested by [Avlonitis, 2002] for shipped water in some of the Greek islands of up to $2.50 \text{ £/m}^3$ (3.75 US$/m^3)$.

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**Figure 7.7 - Daily water production as a function of the site annual mean wind speed**

---

**Figure 7.8 - Product water cost as a function of the site annual mean wind speed**
7.3. Summary

Some of the practical aspects involved in the implementation of reverse osmosis RO systems were discussed. Special attention was given to those issues that may be more pertinent to the proposed system, in particular biofouling. From these practical considerations, some suggestions applicable to the implementation of the proposed system in the field were made.

A modified system model was built using a bigger diameter wind turbine. From this model, new long-term performance prediction curves were presented. These were used with a year-long wind speed time series in a case study, and based on the estimated output, a procedure for sizing the storage tank was sketched.

Finally, a water cost analysis was presented for the particular wind speed time-series previously used, based on the expected output over a system lifetime of 20 years and the system net present value. This analysis was then generalised, using the Weibull probability distribution function and the system performance curve (product water vs. wind speed), to produce a water cost vs. annual mean wind speed curve. The results were very encouraging and can serve as a simple site assessment tool.
The most significant steps of the work are highlighted. In light of the procedures adopted and the results obtained, the research contributions are discussed and recommendations for further development are outlined. Finally, the main conclusions of the work are drawn.

The lack of a reliable source of drinking water poses an evident obstacle for human development. In light of that, much has been done in the research and development of alternatives to solve or even alleviate such problems. In this context, water desalination has been widely used and is now considered a mature technology.

The reference to, and even the effective use of, renewable energy sources have been a constant in proposed solutions targeting isolated locations with undependable power supply.

Considering this background, this work focused on the study of small-scale wind-powered reverse osmosis seawater desalination. The use of batteries as an energy storage device was discarded from the outset, due to the performance issues commonly found in hot climates applications and unattended operation. In addition, experience of battery application in remote regions, where they have been used to compensate for wind power fluctuations, suggests they have very limited lifetimes under such operational conditions. In part this is because they are poorly suited to the particular
stochastic variations exhibited by wind turbines which have a local peak at frequencies characterising turbulence.

The proposed system working principle was based on operation at variable conditions, according to the available input wind energy. Energy storage would be implemented by means of desalinated water, which has a higher energy content than seawater.

Two possible configurations - one using a hydraulic motor and the other, a Clark pump - were suggested and assessed through computer simulation. The Clark pump system was chosen for further development with the construction of a test prototype. The system was designed with the end of improving energy utilisation in view.

Throughout the development of the work, a system approach was adopted. This meant that, although the final configuration design was aided by individual component modelling, the models were not unnecessarily detailed. Therefore, enough details of the components were included to support this approach as, for instance, is the case of the inverters.

An overview of the work carried out, from the perspective of the contributions made and the suggestions for further work, is presented below.

8.1. Overview and Recommendations

Regarding the progress of the work, from start to conclusion, the adopted procedure of component-model aided design proved very beneficial. The representation of the simplified component models, later replaced by the experimentally based models obtained for individual components, as Simulink blocks was very helpful. This allowed the visualisation of the system performance, albeit in general qualitative terms, from the early stages of development, reinforcing the system approach adopted throughout.

The curve fitting technique used (non-linear data regression) was satisfactory for most components, specially for those whose parameters do not change considerably with time, which is unfortunately not the case with the RO modules themselves. The modelling of the RO membranes over a wide operating range is one of the contributions of this work, although it is believed that further improvement could be achieved, particularly with regard to the deterioration of their performance with time and any
specific influence of operation at variable conditions. A consistent modelling campaign over a long period of time would be required for that.

Besides, a formal system identification routine could be implemented for the membranes, based on a series of dynamic tests (step changes in operating conditions). The dynamic influence of variations in the feed water temperature and concentration could be disregarded, as these variables tend to vary slowly. In spite of that, knowledge of the expected temperature variations over a longer period, e.g. one year, may be important in a system viability study for a specific site. The dynamic influence due to variations in the feed water pressure and flow rate are more pronounced and these variables should be used in the system identification modelling procedure. In order to include some dynamic characteristics in the membranes model, the simpler idea of characterising the pressure dynamic influence through step response tests was suggested. Despite its simplicity, this approach showed to be satisfactory in the overall system analysis.

In respect to the characterisation of the Proven wind turbine, the developed model seemed appropriate. It is acknowledged that the models used for the turbine and the generation of the wind speed time series are simple and based on single point u-component wind speeds only. But in the overall system approach adopted, and considering the limited size of the wind turbine, they proved adequate. The establishment of a control function based on the model proved satisfactory and was corroborated by the experimental results that showed good matching with the manufacturer's data.

With regard to the positive displacement pumps, the models are believed to be sufficient, whereas for the drives and motors, simpler models could have been used. The 5th-order dynamic 'dq' models used for the motors were more complex than necessary and simpler 3rd-order models could have been used. Despite, their speed dynamics coupled with the steady-state characteristic employed for the pumps proved satisfactory. The representation of the inverters by their control functions and their steady-state efficiencies seemed enough in the system context. It is believed that the inclusion of high-frequency switching patterns would have made the model simulation much slower and would not contribute much to the overall picture. The same approach
was used in the modelling of the Buck converter. In spite of that, the control functions used could be those of standard V/f inverters, as those used in the prototype.

On the system modelling as a whole, both configurations proposed presented encouraging results. The hydraulic motor system is clearly simpler and improved performance may be obtained by further optimisation of this approach. The disadvantage of a reduced degree of control over the system may be mitigated by programmed pulley changes. On the other hand, the Clark pump system inherently allowed independent variation of the feed flow and pressure. In order to make best use of this advantage, a control strategy based on independently controlling both, the energy capture and usage, was proposed.

The efficacy of the control scheme proposed has been shown to be very promising from the simulation results obtained, although they were based on continuous measurement and actuation of the controller. To allow similar implementation in practice, direct measurement of the turbine speed (e.g. a photo-sensor used on its shaft), a more complex frequency detection algorithm or even the use of a dedicated analog circuit would be need. Unfortunately, experimental demonstration of the control over the power extracted by the wind turbine could not be presented for the system operation (only for the turbine modelling tests), due to the lack of wind speed data during the experimental tests. Notwithstanding, as an outcome of the long period (500 ms) between consecutive executions of the control loop routine, it is believed that the results would not have been too impressive. It is noteworthy that, ultimately, a trade-off must be made between the stable and continuous operation of the system, and the extraction of the maximum power available. This means that using a conservative approach in the tuning of the controllers - instead of allowing severe excursions in the system operating parameters - may prove more productive in the long run.

With regard to the validity of both model-based functions used in the system control, it is clear that changes in the system characteristics, more specifically the wind turbine and the RO membranes, can seriously compromise their effectiveness. Such threat may not be too significant for the wind turbine, but it certainly seems sensible with respect to the RO modules. Although there is no doubt that the performance of RO membranes does decline with use, whatever the conditions they may be subjected to, it is not believed that these changes will cause a significant shift in the point of better
performance, i.e. the membrane will permeate less water overall, but will still produce the most water at or very near the points defined by the control function.

To account for the drifting of the parameters in time, an improved version of the model-based control strategy used of both the wind turbine and the membranes could be developed. A routine that could systematically test neighbouring values to those established by the control functions could be devised. The results obtained would be confronted with those obtained using the initially defined function and if the new point provided better performance, it would replace the previously used point in the curve, which could be stored in a memory chip. Even though it may not be difficult to implement such procedure in terms of hardware requirements, it would certainly take time to be thoroughly validated. The added complexity may prove to be outweighed by the simplicity of the model-based control strategy.

On the tuning of the controllers, it is acknowledged that the gains were set empirically, through observation of simulation results and not formally, by application of classical control methodologies. Appropriate stability analysis using established criteria (such as pole placing) could be used, in order to define performance targets for the controllers actuation, although the highly non-linear nature of the system may well limit the effectiveness of such approaches. Besides, considering the non-linearities present in the system, other techniques, such as gain-scheduling or adaptive control, could be studied to enable an “operating point/region” specific control action.

Finally, as with any renewable energy powered system, its economic viability is ultimately dependent on the available local resources. According to the simplified analysis performed, the choice of a bigger wind turbine than the one used throughout this work is strongly recommended. Encouraging water costs, under 1.00 £/m³, were found for a system lifetime expectancy of 20 years. It is acknowledged that many assumptions had to be made in this analysis and the implementation of a pilot system would certainly provide better estimates, particularly related to component wear, membrane lifetime, and other considerations related to the system operation and maintenance (O&M).

The main contributions of this work can be stated as the modelling, implementation and analysis of an alternative configuration for stand-alone
wind-powered reverse osmosis desalination, without the use of energy back-up or storage devices; and the development of a suitable control strategy for the system, focused on the optimal use of the resource and based on the independent control of the energy input and usage. The development of a model of the RO membranes that is able to characterise their behaviour over a wide range of operating conditions is also a contribution. Albeit, it is acknowledged that the membranes model could be improved, specially regarding the characterisation of their long term performance deterioration and their dynamic behaviour. An interesting study would be to assess their ability to stand variable operating conditions (in a real system environment) over a long time span, with views to validate the use of Reverse Osmosis, as a desalination technique, in conjunction with renewable energy sources. This could be verifiable by the implementation and close monitoring of a pilot system.

8.2. Final Comments

The aim of this work was to propose and assess an alternative system configuration, which would enable the desalination of seawater using the reverse osmosis technology and supplied by a wind powered generator, without the use of energy backup or storage.

The performance predictions obtained from computer simulated results and the experimental data acquired from a prototype test rig are encouraging and corroborate its technical viability. A simplified economic analysis indicates that it is also economically viable.

Despite its suitability to a limited market niche, it is believed that the proposed system would be very useful to reliably supply potable water in a sustainable way. The practical issues that may arise with the implementation of the proposed system are not fully known, and in the same way that the prototype brought to light some of these aspects, a more thorough assessment could only be possible after a pilot system study.

8.3. Published Work

As a result of parts of this work, the following publication were produced:


Besides, the following publications are planned for the near future:

a) On the Clark Pump (description, modelling and experimental results - focus on energy recovery efficiency);

b) On the Membranes (testing, modelling and experimental results – focus on variable operating conditions);

c) On the RO Rig (experimental results for the configuration proposed – focus on low specific energy consumption);

d) On the Complete System, with Wind Turbine (renewable-energy powered RO system performance – focus on modelling and experimental results);

e) On the System Control (adopted strategy, simulation and experimental results).
REFERENCES


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[van't Hoff, 1887]: van't Hoff, JH. *The Role of Osmotic Pressure in the Analogy between Solutions and Gases*. Zeitschrift fur physikalische Chemie (1887), 1, 481-508.


APPENDIX A
SYSTEM COMPONENTS
PARAMETERS

A.1. Wind Turbine Generator
Manufacturer/Model: Proven Energy - WT2500
Rated power: 2.5 kW
Rotor diameter: 3.5 m
Total inertia of blades and generator ($J_{WT}$): 9 kgm$^2$
Generator:
   Number of poles ($P$): 8
   Stator resistance ($R_s$): 3.65 $\Omega$
   Stator inductance ($L_s$): 15.8 mH
   Flux constant ($k_w = E_{WT}/\Omega_{WT}$): 5.25

A.2. Variable Speed Drives

A.2.1. VSD1 (Plunger pump)
Inverter:
Manufacturer/Model: FID 1000 Series (FKI-22220)
Rated power: 2.2 kW
Induction Motor:
Manufacturer/Model: Brook-Hansen Motors
Rated power: 3 kW
Number of poles (P): 4
Inertia (J): 0.009 kg.m\(^2\)
Equivalent circuit parameters:
\[
\begin{align*}
\text{Stator resistance (Rs): } & 2.36 \, \Omega \\
\text{Stator inductance (Ls): } & 290.6 \, \text{mH} \\
\text{Mutual inductance (M): } & 285.2 \, \text{mH}
\end{align*}
\]

\[\text{Rotor resistance (Rr): } 2.29 \, \Omega \]
\[\text{Rotor inductance (Lr): } 295.4 \, \text{mH}\]

A.2.2. VDS2 (Moineau pump)

Inverter:
Manufacturer/Model: FID 1000 Series (FKI-12150)
Rated power: 1.5 kW

Induction Motor:
Manufacturer: Siemens Energy & Automation, Inc.
Rated power: 1.5 kW
Number of poles (P): 6
Inertia (J): 0.0065 kg.m\(^2\)
Equivalent circuit parameters:
\[
\begin{align*}
\text{Stator resistance (Rs): } & 4.15 \, \Omega \\
\text{Stator inductance (Ls): } & 402.5 \, \text{mH} \\
\text{Mutual inductance (M): } & 369.2 \, \text{mH}
\end{align*}
\]

\[\text{Rotor resistance (Rr): } 4.71 \, \Omega \]
\[\text{Rotor inductance (Lr): } 378.9 \, \text{mH}\]

A.3. Pumps

A.3.1. Plunger pump
Manufacturer/Model: CAT Pumps (CAT 317)
Flow: 15 l/min
Speed: 950 rpm
Stroke: 18 mm
Bore: 20 mm
Output pressure range: 7 to 155 bar
Inlet pressure range: -0.35 to 4 bar

\( k_{QPP} \), coefficients of the outlet flow function – Equation 3.18:

<table>
<thead>
<tr>
<th>( k_{1_{QPP}} )</th>
<th>( k_{2_{QPP}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>-1.3x10^{-3}</td>
<td>2.9x10^{-4}</td>
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</tbody>
</table>

\( k_{TPP} \), coefficients of the shaft torque function – Equation 3.19:

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<thead>
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</thead>
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<tr>
<td>1.66</td>
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</table>

\( k_{QPP} \), coefficients of the outlet flow function – Equation 3.20:

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<th>( k_{2_{QPP}} )</th>
<th>( k_{3_{QPP}} )</th>
<th>( k_{4_{QPP}} )</th>
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</thead>
<tbody>
<tr>
<td>2.6x10^{-3}</td>
<td>2.77x10^{-4}</td>
<td>7.23x10^{-5}</td>
<td>-1.18x10^{-7}</td>
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</tbody>
</table>

\( k_{TPP} \), coefficients of the shaft torque function – Equation 3.21:

<table>
<thead>
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<th>( k_{2_{TPP}} )</th>
<th>( k_{3_{TPP}} )</th>
<th>( k_{4_{TPP}} )</th>
<th>( k_{5_{TPP}} )</th>
<th>( k_{6_{TPP}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.08</td>
<td>2.72x10^{-1}</td>
<td>3.22x10^{-4}</td>
<td>-5.03x10^{-6}</td>
<td>-3.16</td>
<td>2.80</td>
</tr>
</tbody>
</table>

A.3.2. Moineau pump

Manufacturer/Model: Netzsch Pumps (NEMO Series - NM021SY02S12B)
Flow: 20 l/min
Speed: 900 rpm
Pressure range: up to 12 bar

\( k_{QMP} \), coefficients of the outlet flow function – Equation 3.22:

<table>
<thead>
<tr>
<th>( k_{1_{QMP}} )</th>
<th>( k_{2_{QMP}} )</th>
<th>( k_{3_{QMP}} )</th>
<th>( k_{4_{QMP}} )</th>
</tr>
</thead>
<tbody>
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<td>-1.68x10^{-3}</td>
<td>3.83x10^{-4}</td>
<td>-5.31x10^{-9}</td>
<td>-3.17x10^{-4}</td>
</tr>
</tbody>
</table>

\( k_{TMP} \), coefficients of the shaft torque function – Equation 3.23:

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<thead>
<tr>
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<th>( k_{2_{TMP}} )</th>
<th>( k_{3_{TMP}} )</th>
<th>( k_{4_{TMP}} )</th>
<th>( k_{5_{TMP}} )</th>
<th>( k_{6_{TMP}} )</th>
<th>( k_{7_{TMP}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.99</td>
<td>0.31</td>
<td>1.56x10^{-3}</td>
<td>-1.48x10^{-2}</td>
<td>3.44x10^{-5}</td>
<td>-4.11x10^{-8}</td>
<td>1.87x10^{-11}</td>
</tr>
</tbody>
</table>
A.4. **RO Modules**
Manufacturer/Model: KOCH Membrane Systems (TFC 1820HF)
Membrane area: 6.8 m²
Maximum operating pressure: 82 bar
Maximum operating temperature: 45°C
Performance specs @ 32,800 ppm NaCl, 55 bar, 25°C, pH 7.5, 7% recovery:
   - Permeate Flow: 5.7 m³/day
   - Chloride Rejection: 99.4%

A.5. **Hydraulic Motor**
Manufacturer/Model: Danfoss A/S (MAH5)
Rated power: 4.1 kW
Speed: 300 - 4000 rpm
Volumetric displacement: 5.07 ml/rev
Inlet pressure range: up to 140 bar

A.6. **Clark Pump**
Manufacturer: Spectra Watermakers Inc.
Flow: up to 15 l/min
Design recovery ratio (A_{rod}/A_{piston}): 0.1012
Output Pressure: up to 65 bar
Inlet pressure: up to 13 bar

A.7. **Control Loop (PI Controller)**
Input: DC Current error (A)
Output: Moineau pump speed reference (rpm)
Proportional Gain, P: 20
Integral Gain (model), I: 300.
Integral Gain (experimental), I: 380.
APPENDIX B
PROTOTYPE INSTRUMENTATION

In total, 26 variables were sampled by the DAQ programme. The signal conditioning board used had dedicated high-gain inputs for strain-gauge/thermocouple readings. These were used in the measurement of the feed water temperature, all the pressures and motor torques. A brief summary of the instrumentation used is presented below:

- **Voltage** (4): Four isolated Hall-effect voltage sensors were used to measure the wind turbine AC output (1 phase), the DC bus, and both inverters outputs.

- **Current** (5): Five isolated Hall-effect current sensors were used to measure the wind turbine AC output (1 phase), the DC bus connection to both inverters, and both inverters outputs.

- **Wind Characteristics** (2): An anemometer and a wind vane were used in the characterisation of the incident wind speed. The measured yaw angle was used to correct the incident wind speed measured by the anemometer in the DAQ software.

- **Temperature** (1): A platinum resistance temperature probe (PT100) was inserted in the low pressure feed water pipeline.

- **Pressure** (3): Three silicon diaphragm pressure sensors were used to measure the medium pressure output from the Moineau pump and both the membrane modules input and output pressures (high pressure feed and concentrate).
• **Flow** (3): Three oval gear flow sensors were used to measure the product flow and the inlet flow to the Clark pump and the plunger pump. The total feed flow was calculated from the latter two measurements. The sensors output pulses were converted into flow measurements by the DAQ software.

• **Conductivity** (2): The feed and product water concentrations were measured by two conductivity sensors. The temperature compensation facility was disabled in both sensors and the correction was implemented in the DAQ software, using the data provided by the temperature sensor.

• **Torque** (2): The torque of both induction motors were measured by load cells (strain gauges). The motors were suspended by purpose-built cradles, to which the load cells were connected.

• **Speed** (2): Photocells were used in the measurement of the motors rotational speeds. Markers were placed on the motors shafts and the photocells used to read the pulses, which would then be converted into speed measurements by the DAQ software.

• **WTG Simulator** (2): The analogue outputs of the variable speed drive used in the wind turbine simulator were used to verify the driving induction motor speed and shaft torque.
APPENDIX C
BUCK CONVERTER

A Buck converter was purpose-built for use in both the wind turbine modelling stage, as a speed or power regulator, and the system testing stage, as a voltage limiter to the variable speed drives.

Figure C.1 - Buck converter (controlled load and voltage limiter)
It was built using an IGBT block containing six switches (Figure C.1). The upper bridge gating ports were connected to a 1 kΩ load, to prevent undesirable triggering as a consequence of EMI. A 36Ω resistive load was connected in parallel with the upper bridge and the anti-parallel diodes of the switches used as the free-wheel diode of the converter. The bottom bridge gating ports were connected to the gating circuit board, supplied by Marlec Engineering Co Ltd, for simultaneous triggering. The original circuit was modified to allow the gating of the switches either directly from the PC (digitally controlled load) or from its own control circuit (analog controlled voltage limiter), as shown in Figure C.2.

![Diagram of the gating circuit to the Buck converter](image)

**Figure C.2 - Diagram of the gating circuit to the Buck converter**

### C.1. Controlled Load

As described in Chapter 3, a feedback control loop was implemented in a PC in order to operate the wind turbine either at constant speed (for the development of the model) or at optimal \( C_p \) (for the control function validation). The gating signal to the
converter was supplied by a PWM generator IC already present in the gating circuit board. The converter duty ratio was proportional to the input analogue signal, which varied between 0.8 and 3.8 V (the output range of the PI controller implemented in the control PC).

When operated as a speed regulator, the value used for the proportional gain of the PI controller was set to be inversely proportional to the set-point speed. The values used for the gains were:

- Integral Gain, \( I = 0.05 \).
- Proportional Gain, \( P = -(0.1)x + 3.9 \)

Where "\( x \)" is the set-point rotational speed in rad/s. This meant that the gain could vary from \( P = 2.3 \) (at \( x=16 \) rad/s) down to \( P = 0.1 \) (at \( x=38 \) rad/s).

When operated as a current regulator, the values for the PI controller gains were fixed and equal to:

- Integral Gain, \( I = 1 \)
- Proportional Gain, \( P = 0.2 \)

C.2. Voltage Limiter

When used as a voltage limiter, the converter duty ratio is proportional to the output signal of a voltage comparator, built in the board. Using a voltage divider, built with a 512K\( \Omega \) fixed resistor and a 10K\( \Omega \) variable resistor, the DC voltage limit could be directly adjusted. Due to the comparator gain, the duty ratio of the converter was fully covered (from 0 to 1) by a 20 V variation of the input DC voltage. This meant that having set an initial triggering voltage of, for instance, 390 V by adjusting the variable resistor, it was expected that the converter resistive load would be fully connected to the DC bus by the time the voltage had risen to around 410 V. Given the value of the load (36\( \Omega \)), this would imply in the dissipation of just over 4.5 kW of power, more than enough to prevent the wind turbine to speed any further. Because of the that, this situation did not happen in practice, and the wind turbine would usually stop accelerating at lower speeds.