Analysis and Modelling of Pressure Exchangers

by

Mahaganaapathy Magasvran

Dissertation submitted in partial fulfilment of
the requirements for the Bachelor of Engineering (Hons)
(Chemical Engineering)

AUGUST 2013

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

Analysis and Modelling of Pressure Exchangers

by

Mahaganaapathy A/L Magasvran

A project dissertation submitted to the
Chemical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(CHEMICAL ENGINEERING)

Approved by,

____

(DR. RAJASHEKHAR PENDYALA)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

August 2013

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

____

MAHAGANAAPATHY A/L MAGASVRAN

ABSTRACT

In this dissertation, the use of pressure exchangers as an energy recovery device was highlighted as a solution to the global energy shortage issue that we are currently facing. As such, different designs of pressure exchangers with respect to their intended applications were discussed. Apart from that, one of the main obstacles preventing this device from resolving the world's energy crisis was also addressed through a detailed mathematical modelling. In the first phase of the project, the basic components and working principles of a pressure exchanger were introduced through conceptual explanation. Then, the major features of a pressure exchanger were studied to determine their advantages and barriers in the pressure transmission process. Based on these features, several criteria were identified to classify the various existing pressure exchanger designs. The classifications were made based on the type or phases of the working fluids, the operating principles used in a pressure exchanger, the use of a separation mechanism and other additional features that can be incorporated into a pressure exchanger. The research methods that were used include studying the basic structure and working principle of a pressure exchanger, conducting a theoretical analysis of different designs while determining the advantages and disadvantages of each design, classification of each design based on certain criteria, identifying the various industrial applications of the device and studying its economical feasibility as compared to a conventional method. The second phase of the project involves performing a detailed mathematical modelling to represent the dynamic flow of fluids in a basic rotary pressure exchanger design. The model can also be used to determine the overall pressure transfer efficiency of the device. Finally, opportunities for further study on this subject were identified in which this basic pressure exchanger model can be further developed to facilitate a specific industrial application.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

1. INTRODUCTION

1.1 Background of Study

1.1.1 General Description

In today's era of globalisation, the world's energy consumption has been on a tremendous rise since the transformation of human civilization from an agricultural society to the age of industrialisation. This change has ever since invoked rapid inventions and discoveries in terms of machineries and industrial processes that aim to assist human beings in fulfilling their daily needs as they aspire for a higher standard of living.

As such, for the purpose of this research paper, the study on energy consumption will be based on the trend in energy consumption and production in the last three decades as significant technological advancements were made during this period. The world's energy consumption by fuel in 1973 was 4672 million tonnes of oil equivalent (MTOE) (1.96×10^{11}GJ) as compared to 8677 MTOE (3.63×10^{11}GJ) in 2012 which is an increase of 85.7% (2012 Key World Energy Statistics, 2012). If we were to consider the energy consumption specifically for the industrial sector, there is an increase of 56.8% from 1544.86 MTOE (6.47×10^{10}GJ) in 1973 to 2422.94 MTOE (1.01×10^{11}GJ) in 2010.

Hence, to meet this ever increasing energy demand, the total primary energy supply by fuel has been increasing steadily from 6107 MTOE (2.56×10^{11}GJ) in 1973 to 12717 MTOE (5.32×10^{11}GJ) in 2010 that accounts for a rise of 108.2% within 37 years. The statistics (2012 Key World Energy Statistics, 2012) also states that fossil fuels, being the main source of energy have contributed 86.7% out of the total energy supply from all sources in 1973 and 81.1% in 2010. This decrease is primarily due to technological enhancement in the use of renewable energy such as hydro, bio fuel, solar, geothermal, wind and heat energies that make up the alternative energy resources. However, the increase in alternative resources usage to produce energy is not proportional to the increase in global energy demand.

One of the reasons behind this limitation is the difficulty to generate equal amount of electricity as those produced by traditional fossil fuel generators. This means that we may need to reduce the amount of energy we use or simply build more energy facilities which would be economically unfeasible (World Energy Consumption, 2013). Apart from that, renewable energy sources are also highly reliable on their supply which often relies on the weather. For example, hydro generators need rain to fill dams to supply flowing water, wind turbines need wind to turn the blades, and solar collectors need clear skies and sunshine to collect heat and produce electricity. Hence, it was made clear that mankind still has to depend on non-renewable fossil fuels as their primary energy source until a significant breakthrough takes place in search for a replacement to the depleting fossil fuel reserves on earth (World Energy Outlook 2012, 2012).

Another reason that causes a high energy demand for industrial applications particularly is due to improper energy management that leads to energy losses and wastage. Instead of maximizing the energy produced, in 2008 only 31.9% or 98022 terra-watt hour (TWh) of energy is used from 143851 TWh of energy supplied (World Energy Outlook 2012, 2012). One of the primary reasons for this deficit is because, when designing a process plant, the role of pressure is not taken into account in predicting the minimum utility requirement for a chemical process. Utility requirement is fully based upon temperature effect that can be demonstrated through composite curve diagram and temperature against enthalpy plot. According to Homsak's (1996) article, Pressure Exchangers in Pinch Technology, this is due to the assumption that the pressure change is very small and insignificant. However, he stressed that both temperature and pressure have an important influence on thermodynamic analysis of a process because enthalpy changes do not tell enough about the quality, quantity and flow direction of energy between two streams and the true cost of energy for a process depends on both quantity and quality of energy used.

Thus, an approach to counter this unaccounted loss of energy regardless of heat or pressure is through the use of energy recovery devices (ERD) that recycles or reuses high output energy from one process to be fed for the same process or for another process. This application is particularly significant in achieving the required parameters which are the operating temperature and pressure of a process. Sackrison (2001), states that the application of an ERD with respect to temperature is the use heat exchangers called sensible devices that transfer sensible energy (or heat energy) that allow us to heat and cool a feed stream using a hot or cold output stream from another process. Hence energy savings result from the reduced need for mechanical heating or cooling.

Similarly, with respect to pressure transmission which is the scope of our study, the use of pressure exchangers can reduce the energy consumption of pumps and compressors for pressurizing feed streams. Furthermore, high pressure technology has been developed to the extent that many modern chemical processes operate at high pressure (Cheng & Cheng, 1970). These include refrigeration or air conditioning, gas turbine engines, Rankin cycle engines, water desalination and fuel cell pressurization (Garris Jr., 2009). Pressure exchangers are defined as machines for exchanging pressure energy from a high pressure fluid stream to a low pressure fluid stream (Andrews, 2009; Seipple, 1946; Shumway, 1998). In other words, Cheng & Cheng (1970) mentioned that a pressure exchanger can be considered analogous to a heat exchanger where one fluid is heated while another is cooled as heat is exchanged between them. Similarly, in a pressure exchanger, one fluid is pressurized while the other is depressurized through pressure exchange between them. The fluids involved may include gas, liquid and pumpable mixture of liquid and solid slurries (Shumway, 2007). These devices are also referred to as flow-work exchangers, isobaric devices and pressure-wave machines. In addition to that, Barnes and Vickery (1965) stated that pressure exchangers can also act as a pressure equalizer or a pressure divider. It is said to be a pressure equalizer when two fluids of relatively different pressures are introduced into the cells of a pressure exchanger and combined to produce an outlet stream of intermediate pressure. In contrary, a pressure divider separates a single fluid stream of intermediate pressure into two streams of different pressures whereby one stream is of high pressure while the other is of a relatively lower pressure. According to Liberman (2012), the use of pressure exchangers can greatly reduce the operating cost for processes that require a high-pressured feed stream such as in the seawater desalination system also known as seawater reverse osmosis (SWRO) system.

1.1.2 Development History of Pressure Exchanger

According to Hauge (1989,1994,2009), the first prototype of a non-rotary pressure exchanger was made in Norway in the year 1896 which then led to the idea of a rotary-type pressure exchanger which was equipped with a rotor. Then in the year 1989, the first US Patent of rotary pressure exchanger for liquids was filed (Hauge, 1999). The design was based on a self-induced rotation through a particular end cover profile. Self-induced rotation is the ability for an object to rotate without the need for an external force such as a motor. However, there were some limitations to the existing design due to its low efficiency which was primarily caused by restriction or resistance to the rotational motion of the rotor as well as uncontrolled level of mixing between fluid streams. Realizing these problems, the Kuwait Institute for Scientific Research (KISR) embarked on a prototype that used hydrostatic bearing system that enabled a smooth and constant rotational motion which allowed them to control the level of fluid mixing by manipulating the rotational speed of the rotor. The best application for this prototype seemed to be for Seawater Reverse Osmosis (SWRO) and the device was integrated into a test facility known as Doha Reverse Osmosis Plant (DROP). The research and development (R&D) program for pressure exchangers by KISR was continued with the exclusion of booster pumps in the SWRO system (Polizos, Babcock, Hauge, & Hermanstad, 2003).

Booster pump are minor or auxiliary pumps also known as circulation pumps that are used to compensate for minor pressure losses in liquids. A development program to change materials from metallic super alloys to ceramic is started. The manufacturing technology for in-house machining of alumina green ware was developed for rotor, end covers and housing. The study and researches conducted in relation to enhancing the efficiency of a pressure exchanger is a continual process and the prove for this can be seen thorough various design modifications such as having rotor with automatic axial alignment (Hauge, 1999) by and pressure exchangers with anti-cavitations pressure relief system in end covers as explained by Polizos et al. (2003).

1.2 Problem Statement

1.2.1 Problem Identification

As explained in the background section of this report, the use of pressure exchangers can greatly benefit the energy recovery technology of various industries. However, the current applications of this device are very limited only to certain fields specifically in the SWRO technology. This is primarily due to the lack of designs for different applications.

1.2.2 Significance of the Project

The absence of proper mathematical models available to represent the dynamic flow of fluids in pressure exchangers is the primary contributor to the aforementioned problem. Therefore, this serves as the main barrier to design pressure exchangers for a particular desired application. Hence, the motivation for the modelling of available pressure exchanger designs.

1.3 Objectives and Scope of Study

1.3.1 Objectives

- To analyze different types of pressure exchanger designs
- To compare and classify the pressure exchangers based on their designs
- To develop appropriate mathematical modelling for a self-rotating pressure exchanger to determine its overall efficiency by taking into account the following factors:-
    - Fluid properties
    - Design of inlet and outlet channels at the end covers
    - Speed of rotor
    - Design of transfer chamber

1.3.2 Scope of Study

For the purpose of this study, a comprehensive literature review was performed on existing pressure exchanger inventions. Based on the review, different types of pressure exchanger designs were classified according to certain criteria that directly influence the pressure transfer and energy consumption of the device. The development of mathematical model will be based on the aforementioned factors. All these factors are then taken into consideration to determine the overall efficiency of the system.

1.4 The Relevancy of the Project

- *Chemical Engineering* - Development of mathematical models requires the application of knowledge from courses such as physical chemistry, fluid mechanics, material and energy balance and transport phenomena.
- *High energy demand* - The generation of energy to meet the increasing energy demand is often insufficient and this inhibits certain high energy processes from being carried out. Hence, with the development of an energy recovery device such as a pressure exchanger, we will be able to reduce the energy required by recycling the high output energy to be reused as input energy.

1.5 Feasibility of the Project within the Scope and Time frame

- *Scope of study* - This project was carried out within the scope of chemical engineering course as it encompasses various aspects of this field of study.
- *Time allocation (2 semesters)* - The time frame is sufficient for a complete study on the literatures available on this topic as well as to develop mathematical models for a basic rotary pressure exchanger design and to determine its overall efficiency.

CHAPTER 2

LITERATURE REVIEW AND/OR THEORY

2. LITERATURE REVIEW AND/OR THEORY

The review on the existing articles or literatures available for pressure exchangers was done based on the development history of pressure exchangers, primary components and working principles of rotary pressure exchanger as well as other designs and configurations of the device. As such, the designs with similar criteria were categorized together and explained based on its operations, advantages and disadvantages. As per the current analysis conducted, some of these criteria include, type of fluid phase used for both streams, operating principles, the use of separators and finally the incorporation of additional features in the design.

2.1 Components and Working Principles of a Rotary Pressure Exchanger

2.1.1 Primary/basic components

A pressure exchanger device may be designed uniquely to serve a particular purpose. As such, the components that are incorporated within them also vary from one design to the other. So, for the purpose of describing the concepts and theory of a pressure exchanger, this study will focus on the most common and widely used design which is a rotary pressure exchanger. However, the review section of this paper will encompass a complete range of pressure exchanger designs. First of all, as explained by Gardiner (1964), a pressure exchanger differs from a pressure transformer. In his findings, he concluded that a pressure transformer utilizes the compression and expansion of waves that were created when fluids at different energy levels are brought into contact to obtain a varying combination and magnitude of pressure of the output fluid. On the other hand, in a pressure exchanger, the only combination and magnitude of pressure obtained at the output are those which are available at the input.

Jendrassik (1954) defines a rotary pressure exchanger as a rotary machine having two inlets and two outlets for high and low pressure fluids respectively. The low pressure fluid will undergo compression while the high pressure fluid will be expanded before both the fluids are expelled out of the system. In general, the device is an assembly of three main components. The first component is the housing or body of the device which acts as an external barrier or protective layer for the smaller components within it. So, it is used to accommodate a rotating rotor where pressure transmission takes place. Thus, the shape of the housing depends on the rotor’s design which is usually cylindrical to allow rotation in its axial plane. Although the rotor is allowed to rotate, the housing itself is an immobile component. The next structure is the end cover which is placed at each end of the housing. Each of these end covers has an inlet and an outlet channel that allow fluids to enter and exit the rotor. The end cover at any one end of the rotor is in contact with only one of the fluids which enter and exit through the channels. These channels are separated by a sealed region to prevent mixing of the inlet and outlet fluids that are of different pressures although they belong to the same fluid type. The end covers are also immovable components that have a fixed configuration. Thus, the channels are arranged in such a way that the inlet port on one end cover faces the outlet port of the other end cover and vice versa.

The final component of a rotary pressure exchanger is a rotor which was discussed in detail by Williamson (1970). In summary the rotor is the mobile section of a pressure exchanger that rotates within the housing at a certain speed determined by either a self-induced rotational force or an external drive force. The rotor is usually cylindrical in shape with multiple chambers or bores within it that extend in the axial direction parallel to the rotational axis of the rotor that forms concentric circles (Hauge, 1989; Gardiner, 1964; Collin & Favrin, 1959). The purpose of the chambers is to allow the two working fluids to come into contact with each other, either directly or indirectly, and transmit pressure from a high pressure stream to a lower pressure stream. Since the quantity and shape of the transfer chambers have a direct influence on the contact time and pressure transmission efficiency of the device, these parameters are subjective according to the designated requirements of the pressure exchanger. The illustration of the components of a pressure exchanger can be seen in Figure 1 (ERI Pressure Exchanger, 2007).

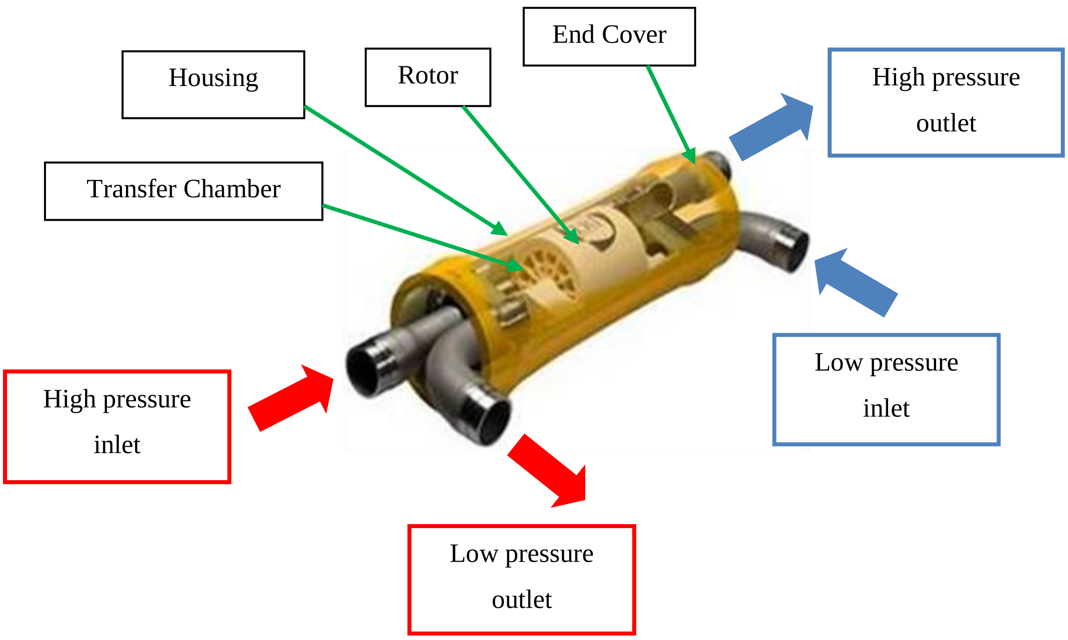

Figure 1: Components of a rotary pressure exchanger

2.1.2 Working Principles

As mentioned previously, a pressure exchanger operates with two fluid streams of varying pressures whereby the stream with a higher pressure transmits its pressure to a lower pressured stream in the transfer chambers of the device. As such, the design of a pressure exchanger should incorporate an inlet and an exit channel for each of these fluids. These channels as described in the preceding section are aligned in a fixed manner in their respective end covers such that the inlet channel of one fluid stream faces the outlet channel of the other fluid stream. For the purpose of this study, fluid 1 refers to the high pressure fluid that pressurizes the second fluid stream whereby 1A is the high pressure inlet stream and 1B is the low pressure outlet stream. Similarly, fluid 2 refers to the low pressure fluid that is pressurized by the fluid 1 in which 2A is the low pressure inlet stream while 2B is the high pressure outlet stream. Hence, when stream 1A enters the transfer chamber through the inlet channel of fluid 1, it forces fluid 2 that occupied the transfer chamber from the previous cycle out as stream 2B. As the cycle continues, the high pressure stream 1A is now trapped in the transfer chamber before it is depressurized and forced out by the incoming stream 2A. These two processes are depicted in Figures, 2(a) and 2(b) respectively. Hence, the combination of these two steps enables the rotor to make a complete turn which marks one cycle of the pressure exchanger as shown in Figure 3.

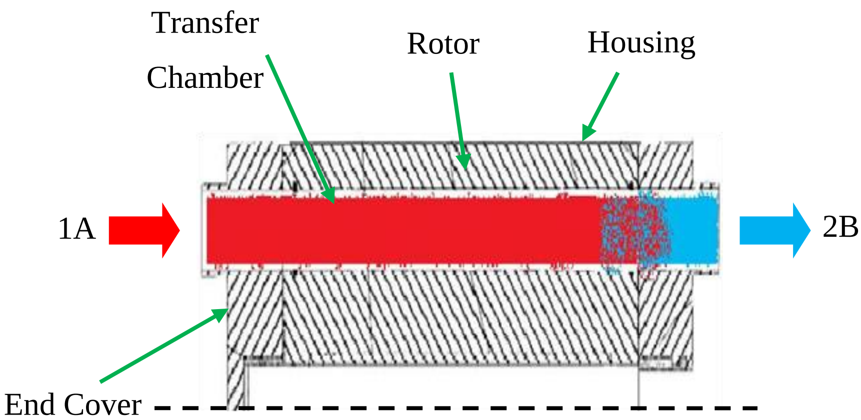

Figure 2(a): Pressurization cycle in the first half of the pressure exchanger

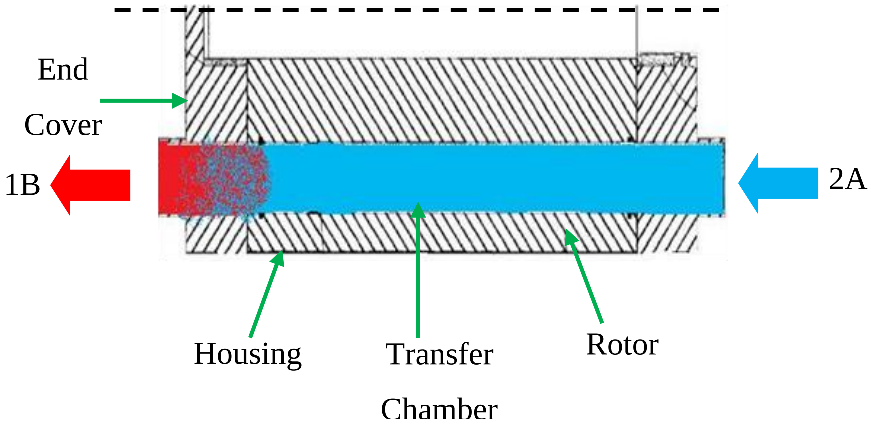

Figure 2(b): Depressurization cycle in the second half of the cycle

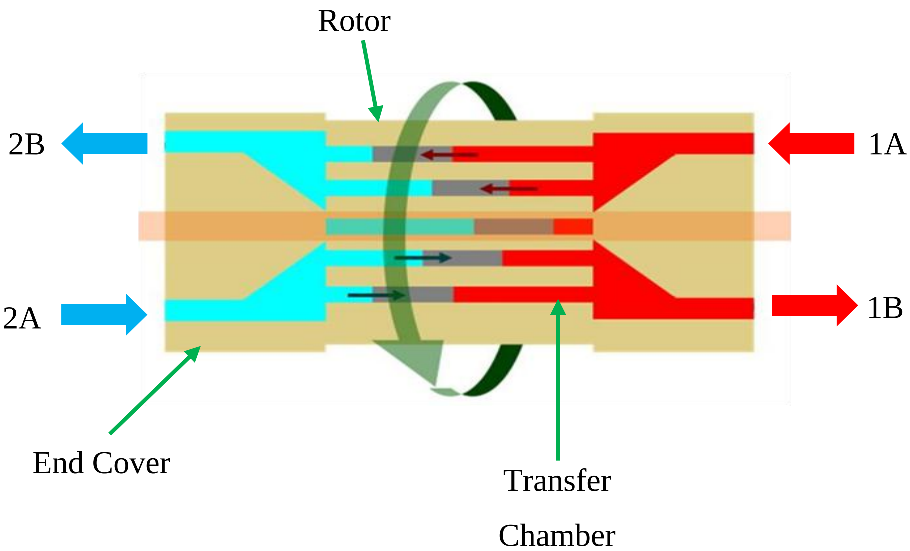

Figure 3: A complete cycle of a rotary pressure exchanger

2.2 Classification of Pressure Exchangers

2.2.1 Fluid Phase

Pressure exchangers operate on the principle of pressure energy transfer from a high pressure fluid stream that enters and exits from one end of the device to another significantly low pressure fluid stream that enters and exits the device from an opposite end. As mentioned previously, this transfer of pressure between streams may occur through either direct or indirect contact between both streams in the transfer chamber. To prevent direct contact between streams which may result in mixing or undesired chemical reactions due to diffusion of one fluid into another, we can install certain component in the chambers called separators which will be discussed in the later section of this chapter. In terms for the fluid phases that can be fed into this system are:-

Pressure Transfer from a High Pressure Liquid Stream to a Low Pressure Liquid Stream

<u>Seawater Reverse Osmosis</u>

Seawater reverse osmosis (SWRO) desalination involves the process of obtaining fresh water from seawater in which the salt concentration is between 35000 to 40000 parts per million (ppm) (Pressure Exchanger, 2012; Went & Anhalt, 2013; Taylor, 1974; Tonner, 1994; Permar, 1995; Childs & Dabiri, 2000). Any pressure applied in addition to the osmotic pressure will cause the process of osmosis to reverse. Hence, the higher the salt concentration, the higher the pressure required for the process which ranges from 800 psi to 1000 psi. The recovery is no greater than 30% before the brine concentration exceed the solubility of certain components. Since the recovery of fresh water accounts for only 30%, Andeen (1987) highlighted that the remaining 70% is lost in the form of pressure energy contained in the brine that is disposed. In order to counter this issue, an energy recovery device in the form of a pressure exchanger is incorporated into the system to minimize the energy loss through the recovery of pressure energy. The device uses this high pressure concentrate stream to transfer pressure to a low pressure seawater stream which becomes pressurized. As illustrated in Figure 4, high-pressure concentrate is then directed to the pressure exchanger from the membranes. These are semi-permeable membrane that is used to extract pure water from the high pressure seawater feed stream. Finally, the brine that leaves the pressure exchanger at low pressure is expelled by the incoming feed seawater flow. Brine is basically concentrated seawater that has a much higher salt and impurities content due to loss in pure water due to reverse osmosis (Al-Mayahi & Sharif, 2012). However, the high pressure brine discharged from the membrane is incapable of transmitting 100% of the pressure it contained and the output pressure of the brine is lower than the pressure required from the feed stream. These minor drawbacks can be resolved with the use some auxiliary booster pump.

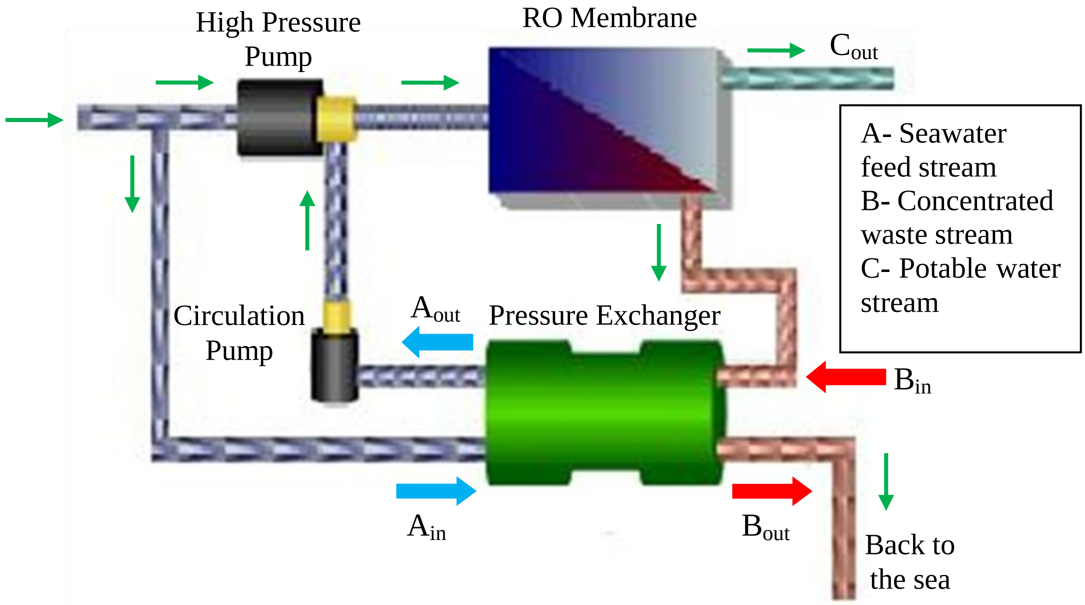

Figure 4: SWRO System

Brackish Water Reverse Osmosis

A similar application was illustrated through the use of SWRO system in treating brackish water as shown in Figure 5 (Went & Anhalt, 2013; Al-Hawaj, 2004). In this case, the water has a higher salt content than fresh water but not as much as seawater. The source of brackish water is primarily from an estuary which is an enclosed coastal body of brackish water with one or more rivers flowing into it with a free connection to the open sea and there is a significant mixing of salt water and fresh water. MacHarg and McClellan (2004) discussed the function of pressure exchanger in this system in relation to that of a SWRO system.

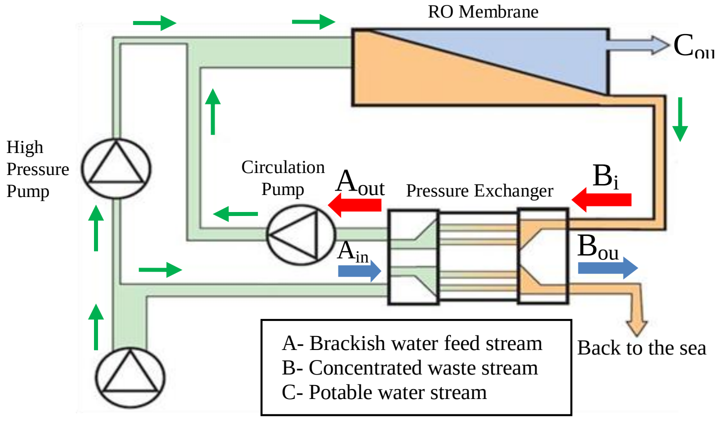

Figure 5: Brackish Water RO system

<u>Transfer of Water to an Elevated Position</u>

Hauge (1989) describes a similar invention that uses liquid-liquid pressure transfer as shown in Figure 6 where hot water from a hot water reservoir tank at ground level is pressurized to be supplied to an elevated position like the roof of a building. Hence, high pressure cold water stream from the elevated position acts as the pressurant that enters the transfer chamber of the pressure exchanger and pressurizes the low pressure hot water stream so that it will have sufficient energy to reach the desired height above ground level.

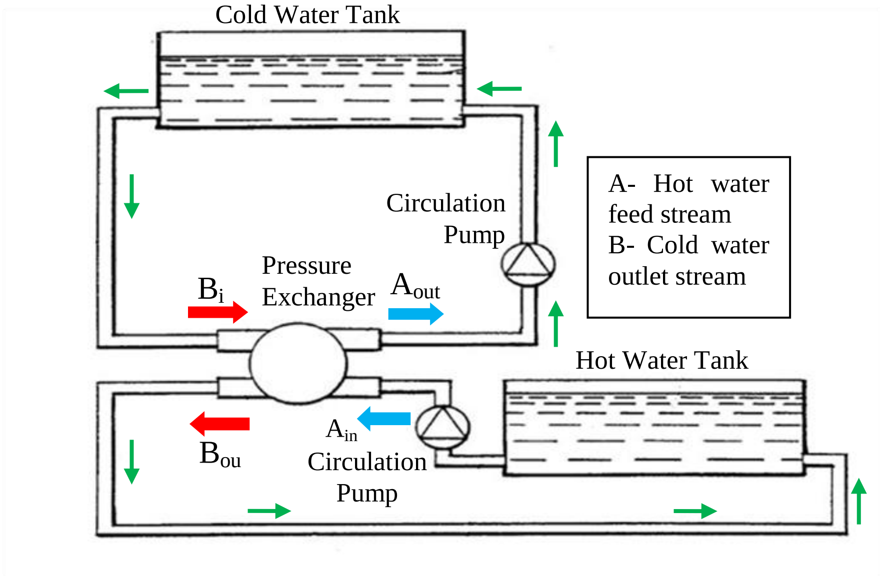

Figure 6: Pressurizing hot water to be supplied to an elevated position

Pressure Transfer from a High Pressure Vapour Stream to a Low Pressure Liquid Stream

According to the inventions (McKinley, 1961), the inventors used high pressure vapour as a pressurant. A pressurant is any high pressure fluid stream that is used to pressurize another fluid stream of a much lower pressure. The application of such an invention can be seen in a rocket engine as shown in Figure 7 (Skinner, 1950). Hence, Knight (Knight, 2002), suggests the use of nitrogen vapour as the pressurant for this system. The vapour is produced from continuous heating of liquid nitrogen in the pressurant tank with its vapour pressure maintained at approximately 2000psi. The pressurant is channelled through a conduit and enters the pressure exchanger where it pressurizes the propellant that enters the pressure exchanger through an opposing conduit. Then, the exhaust pressurant vapour is expelled out of the system while the pressurized propellant is supplied to the rocket engine. In another related embodiment, Alyanak and Eiszner (Alyanak & Eiszner, 1963) suggested an alternative pressurizing mechanism in which the high pressure exhaust gas is used to compress the low pressure propellant.

The invention by Mitchell (Mitchell, 1955) relates to an improved means of pressurizing the propellants in a jet propulsion system. In prior inventions, one of the suggested ways is to use propellant pumps (Baum, 1972; Apfel, 1989). However, this is not feasible as the weight and space occupied by the pump and prime mover is too large and would compromise the weight and space allocated for the propellant itself. This is especially true in the case of small missiles. Apart from that, being mechanical in nature, the pumping system has a tendency of having mechanical failures.

Another manner of pressurizing the propellant by Beveridge and Knuth (1990) is through the use of an inert gas such as nitrogen which is stored in a separate high pressure tank. The high pressure inert gas is connected to the propellant tank via valves so that it can exert pressure and force the propellant into the combustion chamber. The drawback of this method is that, the high pressure tank requires additional space and its walls have to be sufficiently thick to withstand the high pressure of the inert gas contained in it. This will again consume space and weight allocated for the propellant. As such, the present invention aims to solve these problems in a simple and practical manner. In accomplishing this, a portion of liquid oxygen from the propellant tank is channelled through a heat exchanger where it is pressurized to generate high pressure gaseous oxygen. The oxygen gas stream is used to pressurize the remaining liquid in the propellant tank (Mitchell, 1955). According to Pahl (1995), another advantage of this invention is that, the compressed high vapour pressure fluid has a constant pressure at a given temperature and can be used as a reference to control the feed of propellant into the engine's combustion chamber. This reduces the number of electronic sensors and controllers required to operate the engine, reducing the cost and complexity of the engine as well as reducing the likelihood of failure. Some of the suggested safety features that can be incorporated in this design are the use of pressure relief valve at regular intervals to relieve excessive pressure from the gaseous oxygen stream. Apart from that, a piston fitted propellant tank may be used to avoid direct contact between the gaseous oxygen and liquid oxygen during pressure transmission.

In a similar invention, Kayser (1979) suggested the use of pressurized helium gas or air in addition to nitrogen gas. He also stated that 20% to 50% of the fuel can be channelled into a smaller tank where the gas can be easily pressurized. Once again, this reduces the weight and space consumption. The pressurized gas is then directed back into the main fuel storage tank to exert sufficient force on the fuel so that it flows into the combustion chamber at an increased pressure as desired.

Apart from being used in a rocket engine system, pressure exchanger can also be incorporated in a gas turbine system whereby the device acts as a combustion chamber (Jendrassik, 1957; Nalim, 2002). In a conventional gas turbine, there are three distinct sections called compressor, combustion chamber and turbine. The pressure exchanger introduced in this system by Spalding (1966) is a rotary-type pressure exchanger. The system operates based on three consecutive stages. The first stage is when the inlet ports are open to the incoming high pressure air from the compressor which will be directed into smaller cells within the pressure exchanger. Since the pressure exchanger is continuously rotating, these ports will eventually close. In the second phase, as the high pressure air is trapped, fuel is simultaneously injected into these cells to allow both fluids to mix. In addition, each cell is also equipped with an ignition source to ignite the mixture of fuel and air. By the time the cells are opened to the outlet ports, a tremendous amount of pressure builds up in these cells. So, in the third stage when the outlet ports are opened, the high pressure exhaust gas is released into the turbine section to rotate the blades of the turbine.

Since the aim of this system is to provide a high pressure outlet stream through the process of combustion, Jendrassik (1961) proposed that heating of the inlet stream can be done externally before it is channelled into the pressure exchanger. This means that the combustion of fuel takes place in a separate combustion chamber. Nonetheless, he concluded that regardless of whether heating is performed internally or externally, the objective is to produce sufficient gas displacement in a pressure exchanger without the use of mechanically driven devices such as compressors, fans or impellers. Jendrassik (1958) also stated that, the use of pressure exchangers for supercharging internal combustion engines is based upon the principles of expansion and compression of gas in the cells. These two phenomena are produced by shock and rarefaction waves respectively due to the sudden opening and closing of the end of the cells. A shock wave occurs when a cell is opened to a gas of higher pressure than the gas that is already present in the cell. In contrary, a rarefaction wave is produced when a cell is opened to a lower pressure gas as compared to the gas in the cell. Hence, the magnitude of shock and rarefaction waves that are produced highly depend on the circumferential width of cell openings, the opening and closing timing of the cells at opposite ends, the speed of rotation of the rotor and finally the local speed of sound.

In another invention, Hellat and Keller (1987) proposed the use of a pressure exchanger as a high pressure compressor for gas turbines installation instead of the conventional axial or centrifugal compressors. Furthermore, the combustion chamber is located within the rotor cells of the pressure wave machine. Hence, extremely short and low-loss air and gas ducts can be achieved.

Generally, a centrifugal pump coupled with a thermal engine is used to spray liquid. However, the drawback of such pumps is that, they require a very high power. For example, a centrifugal pump with an output of 1300 L/min at a pressure of 12 bar require a power of 120 horsepower (HP) from the thermal engine. Hence, Grouyellec (Grouyellec, 2001) proposed a liquid pumping system which considerably reduces the power required for spraying liquid at a particular flow rate and pressure. The present invention uses a pressurized gas to increase the pressure of liquid to be sprayed. Hence, this mechanism involves two phases operating alternatively and controlled by valves. During the first phase, the chambers are filled with low pressure liquid while the pressurizing gas inlet valve is closed. Then, in the second phase, the liquid inlet valve is closed while the gas inlet valve is opened to allow the high pressure gas to enter the chamber, thus expelling the liquid that was initially contained in the chamber through a set of discharge orifices. The opening and closing of valves are manipulated by a control unit by detecting the liquid levels in the chambers.

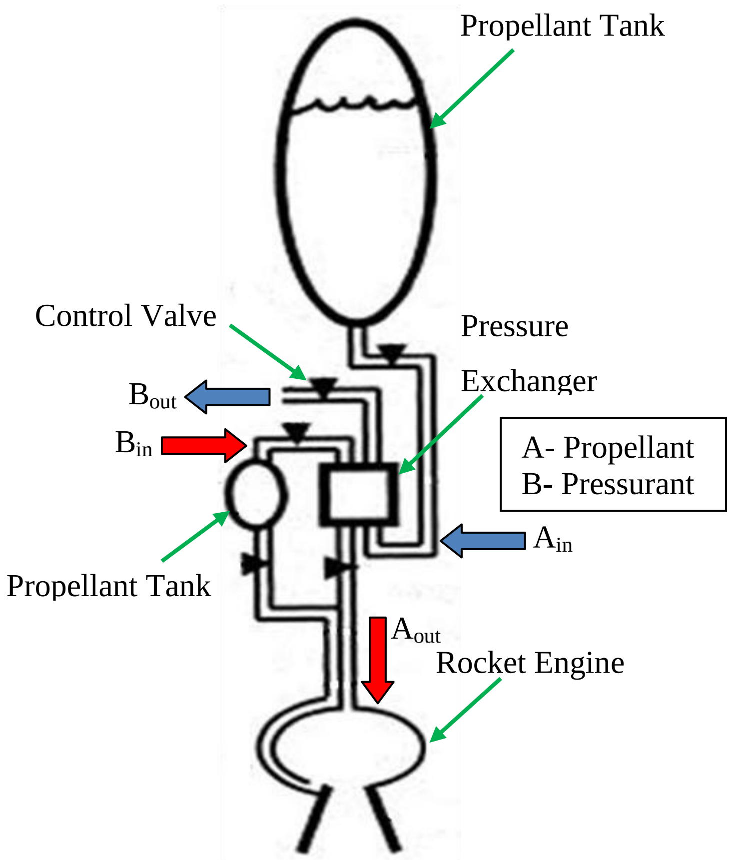

Figure 7: Pressurizer for a rocket engine

Pressure Transfer from a High Pressure Gas Stream to a Low Pressure Gas Stream

Through the embodiment in Figure 8, Herbst et al. (1997) shows one of the configurations in the application of pressure exchanger in an air-cycle air conditioner system. In this application, a pressure exchanger is operated in parallel with a compressor whereby, low pressure hot air from a room or enclosed space enters the pressure exchanger and the compressor simultaneously. Both these streams are pressurized separately by the devices and their output high pressure air streams are combined and channelled into the heat exchanger where the high pressure hot air is cooled. The high pressure cool stream then enters the pressure exchanger as the pressurant and finally exits the system after pressure transfer as low pressure cool air that goes back into the room or enclosed space.

High pressure exhaust gas can also be used to pressurize a relatively lower pressure air stream using a wave compressor supercharger that serves a similar function as a pressure exchanger (Rao, 1981a, 1981b; Coleman Jr., 1958; Vallance & Rahnke, 1982; Berchtold, 1961). This concept was adapted in an application proposed by Reatherford (1998) in an internal combustion engine. In this case, a pressure-wave supercharger uses the high pressure exhaust gas produced from the engine to compress the inlet air that is fed into the engine. Aoki (1986) describes this type of supercharger as a 'comprex' supercharger which is adapted to compress air directly using exhaust gas and supplying the compressed air into an internal combustion engine particularly a diesel engine. The aim of this 'comprex' supercharger is to obtain optimum supercharging pressure with regards to the operating conditions of the engine. This is achieved through the incorporation of a control system to regulate the rotational speed of the rotor. As such, a sensor detects the operating condition in the engine and sends signals to the control element which then manipulates the rotor drive mechanism which either increases or decreases the rotor's speed. This prevents unnecessary high speed rotation in order to prolong the life expectancy of the rotor while improving the device's supercharging efficiency.

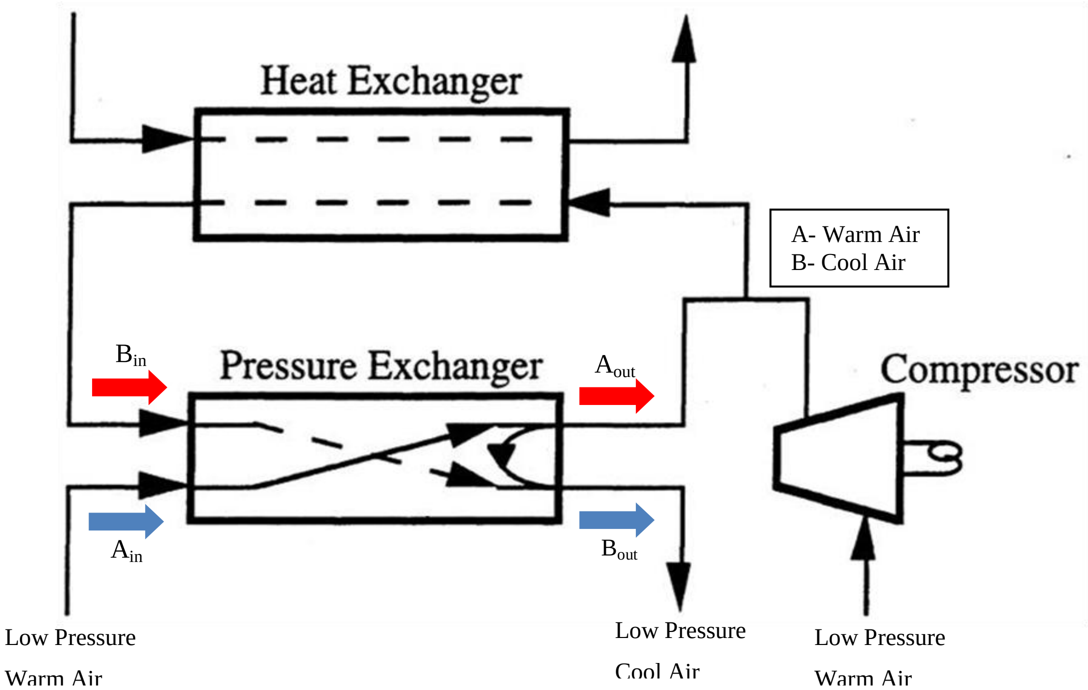

Figure 8: Air-cycle of an air conditioner system using pressure exchanger

Pressure Transfer between Solid-Liquid Slurries

The invention by Hashemi and Lott (1969) relates to the apparatus for exchanging pressure energy between a fluids having high energy content with a relatively lower pressure fluid. The difference that is highlighted in this invention is the application of a pressure transfer mechanism using solid-liquid or pumpable slurries such as those used in freeze crystallization systems.

2.2.2 Operational Principle

As per the study conducted for this project, there are two types of pressure exchangers with one of the designs most commonly used among them. These are rotary-type pressure exchanger and non-rotational pressure exchanger.

Rotational

The rotary-type pressure exchanger is said to be more favourable due to its efficiency in pressure transfer applications. This device uses a cylindrical rotor with longitudinal ducts parallel to its rotational axis. The rotor spins inside a sleeve between two end covers. Pressure energy is transferred directly from the high pressure stream to the low pressure stream in the ducts (transfer chambers) of the rotor. In theory, this rotational action is similar to that of an old fashioned machine gun firing high pressure bullets and it is continuously refilled with new fluid cartridges (Pressure Exchanger, 2012). However, in this case the flow of fluid is continuous to allow constant fluid flow at any point of time as compared to the instantaneous firing of a machine gun. The ducts of the rotor charge (pressurize) and discharge (depressurize) as the pressure transfer process repeats itself. This was clearly shown previously through a schematic illustration in Figure 3.

Non-rotational

The SWRO system as described by Brueckmann et al. (2005) uses a pressure exchanger that has an unconventional operational principle in which it does not involve the rotational motion of a pressure exchanger's rotor during pressure transmission. The system works such that, the inlet and outlet streams of the brine or concentrate is determined by reversing a set of valves. The transfer chambers are equipped with sensors that detect the fluid flow in the chambers and send electrical signals to a control unit which receives and interprets the signals before sending a response signal to the actuator motor. The motor then reverses the valves accordingly to allow fluid flowing into and out of chambers. Hence, in the case of two transfer chambers, at least one chamber is filled with high pressure fluid at all times (Baumgarten, et al., 2006). This ensures continuous supply of high pressure seawater for the RO process. Figure 9 shows the operational layout of this system. During operations, Baumgarten et al. (2006) discovered that the peripheral region of the control element or valve is exposed to different pressure conditions. Hence, in order to facilitate the switch-over motion of the valves, multiple pressure-relief valves can be provided on their external periphery specifically in regions that are subjected to back pressure. Back pressure is defined as a pressure opposing the desired flow of fluid in a confined space such as a pipe (Back Pressure, 2013).

In another embodiment by Andrews (2009, 2012), the reversible valve is referred to a rotary valve element that has fixed exchange ducts which enables the machine to be scaled up in size to accommodate very high flows. Similarly, Shumway (1998) uses a valve system called the linear spool valve device that controls fluid flow in two work (pressure) exchangers. This device ensures that at least one work exchanger is at high pressure at all times thus enabling a continuous supply of high pressure feed stream for a seawater reverse osmosis (SWRO) application.

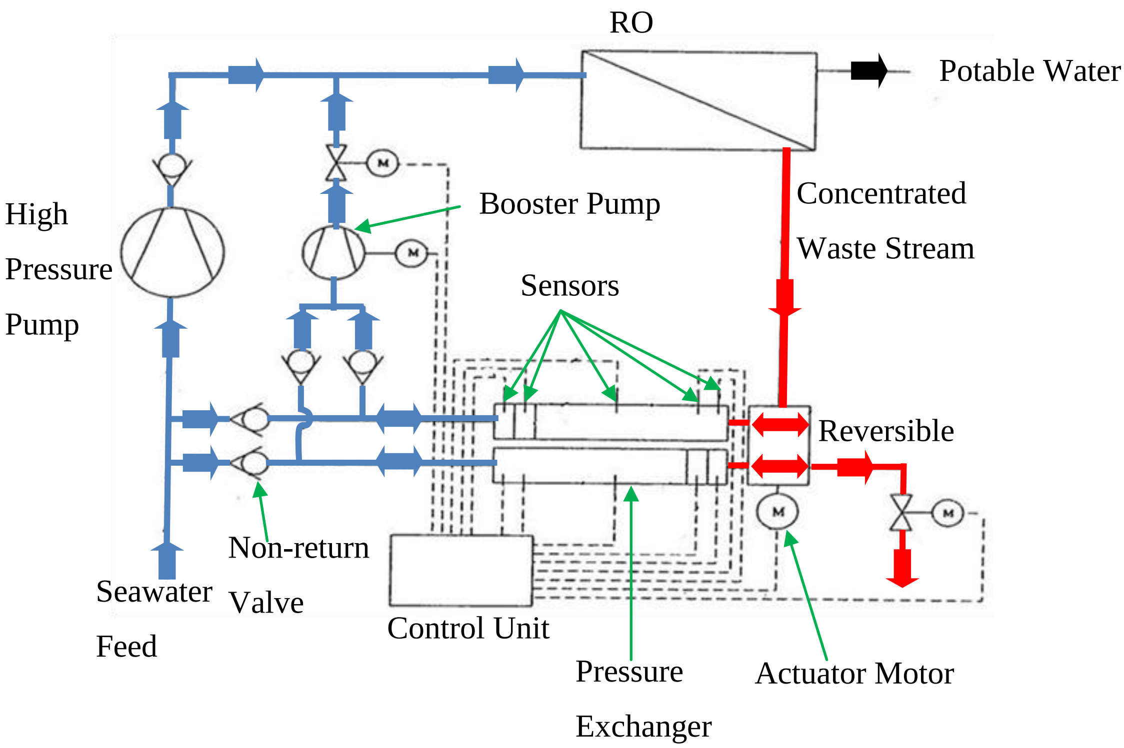

Figure 9: Pressure exchanger in a SWRO system

2.2.3 External Drive/Force

Pressure Exchanger with External Motor

Since the most common type of pressure exchanger among the two designs discussed in the previous section is the rotary pressure exchanger, there are a number of ways to induce this rotational motion. One of which is through the use of an external source of force or power and this can be achieve using an electrical motor. Based on Knight's (2002) explanation, Figure 10 shows one of the ways in which an external motor can be attached to a shaft which is then connected to the rotor and causes it to rotate.

Bross and Kochanowski (2007) explained a similar design in an external drive drives the rotor via a shaft. According to them, the cavities at the end covers have a shape that makes the velocity of flow more uniform in the flow opening area of the housing. In addition, a regulator is provided as a speed regulating device for the external drive, and thus the rotor speed can be regulated at a rotational speed suitable for essentially shock-free admission of flow into the rotor channels.

Garris (2009) mentioned that in certain applications, a designer may wish to include a motor to facilitate overcoming bearing friction and to manipulate the rotational speed to be greater or lesser in accordance with the operating conditions. This is not possible for pressure exchangers that depend on ideal free-spinning speeds.

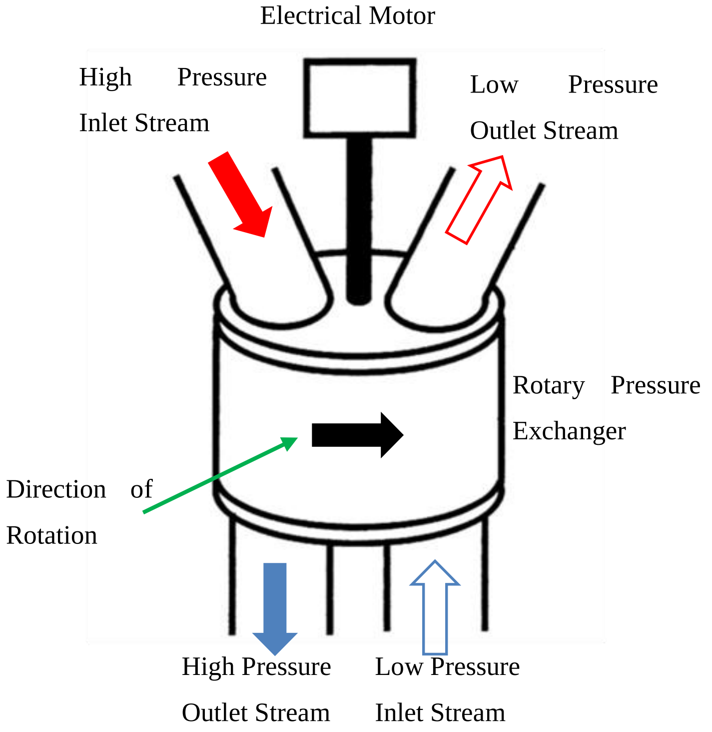

Figure 10: Pressure exchanger with an electrical motor

Pressure Exchanger without External Motor

<u>Inclined Rotor</u>

The mechanism that used is the insertion of a rotor into the housing of the pressure exchanger at an inclined plane instead of the conventional parallel position with the housing wall. Figure 11(a) depicts this configuration.

The operational principle behind this concept was thoroughly explained by Hauge (2006) and can be seen through Figure 11(b) which is the force vector diagram that illustrates the impulse momentum principle for self-rotation. The first and second stream in the tangential cross-section of the rotor duct has a tangential velocity in the plane of rotation similar to the tangential inlet velocity component of the first incoming low pressure stream. Tangential velocity is the velocity in the y-direction while axial velocity is the velocity in the x-direction of the Cartesian plane or generally in the vertical and horizontal directions respectively. The overall relationships between the tangential and axial velocity components are shown below:-

i) Relationship between the tangential inlet velocity ($v_{y_{in}}$) tangential velocity (v_r) and tangential outlet velocity ($v_{y_{out}}$)

$$(v_{y_{in}} = v_r \neq v_{y_{out}})$$

ii) Relationship between the axial inlet velocity ($v_{x_{in}}$), axial velocity (v_d) and axial outlet velocity ($v_{x_{out}}$)

$$(v_{x_{in}} = v_d = v_{x_{out}})$$

Since velocity is a vector quantity, the opposing direction of ($v_{y_{in}}$) and ($v_{y_{out}}$) creates a resultant tangential force that induces the rotational motion in the rotor. The force vector diagram illustrates this concept as well as proves the existence of the force in the tangential (y) direction through a derived formula using input and output momentum.

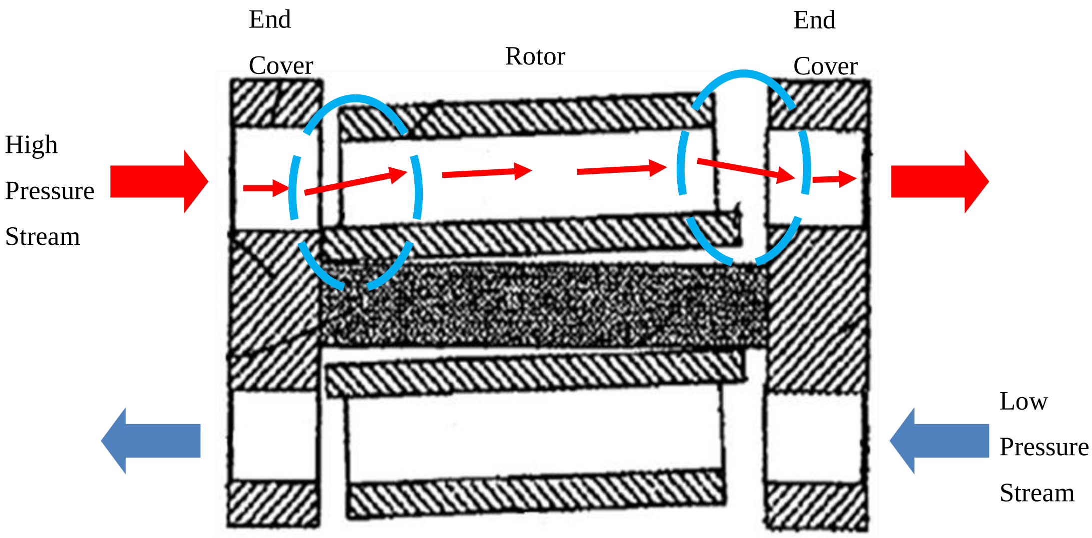

Figure 11(a): Inclined channel in the rotor

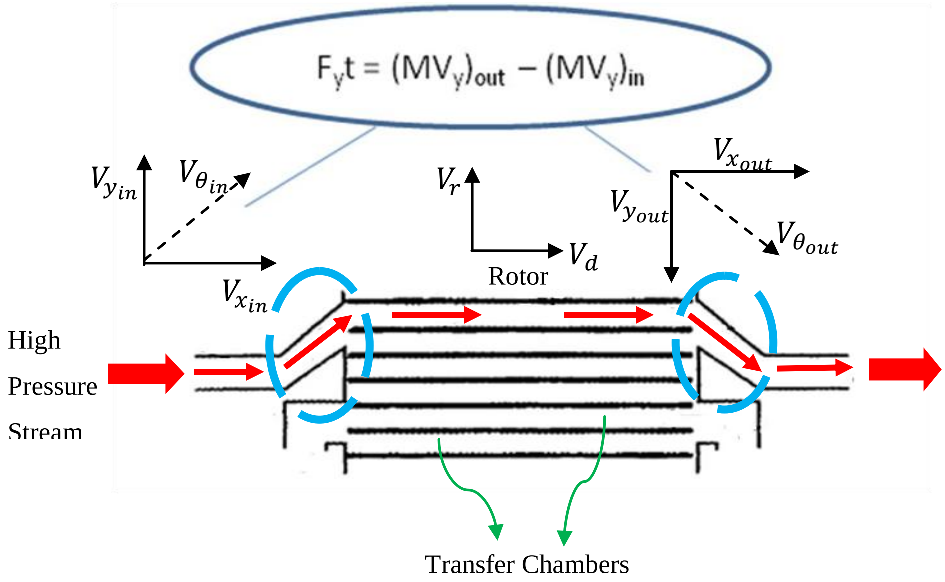

Figure 11(b): Principle behind the self-inducing rotation

<u>Jet Hole</u>

The concept used is this invention is using a particular action to generate an impulsive reaction that induces rotational motion in the rotor (Al-Hawaj, 2004; Hashemi, 1972). For the purpose of explanation, Knight (2002) uses the term spindle that refers to a rotor and a spindle housing that refers to rotor housing. The spindle housing has only one housing jet hole that penetrates the housing and spindle at an angle that is not perpendicular to their walls that is pointed in a direction that is opposite to the rotational direction. On contrary, there are multiple spindle jet holes on the spindle which are cut similarly to the housing jet hole whereby these holes also point in the same direction. The difference between the spindle jet holes and the single housing jet hole is that, the spindle jet holes corresponds to its transfer chambers whereby each transfer chamber has a spindle jet hole incorporated within it. The configurations of the jet holes are clearly illustrated in Figures 12(a) and 12(b).

In terms of operational principles, the single housing jet hole is positioned after the high pressure inlet and outlet chambers with reference to the rotational direction. This is to ensure that the chambers are filled with high pressure inlet stream. Once the chambers enter the sealed area between the housing and the spindle, the stream will not be able to escape until the individual spindle jet hole comes into contact or aligns with the housing jet hole. When this happens, the fluid is able to escape from the transfer chamber at a high pressure in the direction denoted by the “black arrow”. Hence, this generates an equal but opposite reaction that causes the spindle to rotate.

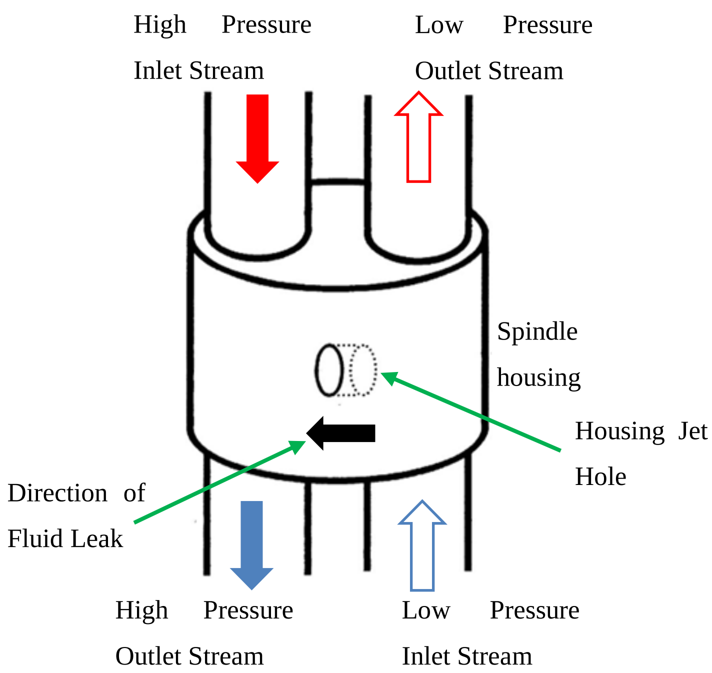

Figure 12(a): Jet hole on the spindle housing

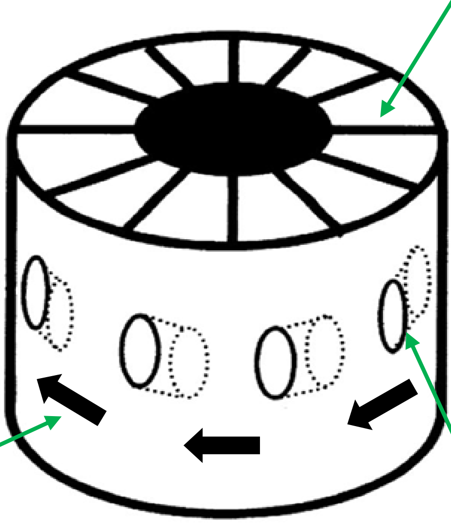

Figure 12(b): Jet holes on the spindle at every channel

Manipulating the Shapes of the Inlet and/or Outlet Channels

Bross and Kochanowski (2007, 2010) proposed an alternative method to drive the rotor wherein a flow guiding shape in the form of a channel contour that deflects the flow is arranged in the inlet area of the rotor channels starting from the channel openings. This flow guiding shape ensures impact-free oncoming flow to the rotor channels. Hence, the driving torque for the rotor is achieved by a direct transfer of momentum from the incoming flow to the rotor end face through the impact-free flow deflection in the channel opening region. In another invention by the same inventors, this system was further enhanced with the use of sensor elements arranged in the liquid systems to monitor the operating states and a regulating device connected to the sensor elements that adjusts the rotor speed in accordance to the altered operating states when deviations occur.

Pique et al. (2011) suggested the use of oblique ramps at the inlet and discharge channels of the high pressure streams to induce rotational motion of the rotor in a particular angular direction. At the same time such modifications are devoid for low pressure streams to prevent them from retarding the revolution of the rotor from its initial angular direction.

Spalding (1963) discovered an alternative means of inducing self-rotation by manipulating the end plate which incorporates a port that has a non-uniform radial dimension and preferably in the form of an arcuate slot. Since the ports in the end plates can either be an inlet or an outlet port, their configuration may be varied. For example, if it is an outlet port, the major part of the port length is uniform while its trailing edge region is tapered. On the other hand, if the port functions as an inlet port, the major part is again kept uniform while its leading edge region is tapered. These non-uniform sections of the end plate port can be varied by a cam mounted on the edge of the port so that the rotational movement of the cam alters the port's configuration. A cam is basically a profile that is shaped to convert circular into reciprocal or variable motion. According to Barnes and Vickery (1965), the arcuate adjustment of the end plates may be controlled relative to the operating requirements of the pressure exchanger. These include temperature, pressure, flow rate or speed of rotation.

Deviation of Sidewalls in Transfer Chambers

Since pressure exchangers involve high pressure fluids in motion, Coleman and Weber (1977) developed a design that utilizes the high pressure fluid stream to perform useful work particularly in inducing rotational motion in the rotor. The design mainly involves modifications to the sidewalls of the transfer chambers. Although the major part of the chamber has substantially parallel sidewalls it also incorporates a nozzle section which has a sharp cross-sectional constriction before it ends at the outlet opening. Hence, when these transfer chambers come into communication with a high pressure fluid, shock waves are initiated at the inlet. As the shock waves pass through the nozzle, they are reflected causing the high pressure fluid to expand. This expansion is the primary drive that causes the rotor to rotate.

2.2.4 Separators

Another key element in studying the dynamics of fluid flow and pressure transfer between fluids is the presence of a separator between the fluids in the transfer chamber. The use of separators depends on a number of factors primarily based on the type of fluids that are in contact. Usually, the need for a separator arises when we would like to minimize or eliminate any form of diffusion or mixing between the two fluids in order to maintain the purity of both or even one of the fluid. Another reason for this could be to prevent any undesired chemical reaction between the two fluids which may change the properties of both fluids. However, there are instances when direct contact is permitted between fluids of the same or almost similar kind. For example, in the SWRO system both the seawater and concentrate come into direct contact with each other during the transfer of pressure. The reason for this will be further explained in the later section. Hence, we will look at both instances whereby a separator is present and absent in the transfer chamber of a pressure exchanger.

Pressure Exchangers with Separators

Different types of separator play vital roles in the pressure transfer mechanism of a pressure exchanger in relation to the type of fluid used as well as the degree of mixing that is permissible. Although the use of separators seems to be beneficial in terms of prevention of fluid diffusion and maintaining uncontaminated fluid streams, it has its disadvantages too. The use of separators mainly compromises the pressure transfer efficiency of a pressure exchanger. This is because some of the pressure energy from the high pressure fluid stream will be lost in the process of moving the separators from one end of the transfer chamber to the other end. As a result, we may not be able to obtain an approximate 100% pressure transfer to the low pressure fluid. However, this disadvantage can be minimized by using low friction separators in the transfer chambers and also by maintaining a high pressure difference between the two streams.

Piston (Solid)

The separation method used in this invention equips every transfer chamber with its own solid piston that moves simultaneously as they are pushed from a high pressure fluid region to a relatively lower pressure fluid region (Andeen, 1987; Al-Hawaj, 2004; Brueckmann, Knoebl, Bruhns, & Kochanowski, 2005; El-Sayed, Abdel, & Al-Odwani, 2010). For the purpose of illustration, Figure 13(a) only shows two transfer chambers where one chamber is in fluid communication with the high pressure inlet and outlet streams while the other chamber is in communication with the low pressure inlet and outlet stream (Shumway, 2003).

MacHarg (2010) proposed an alternative design in which the pistons in both transfer chambers are connected to each other and operate alternately by allowing fluid to enter and exit the chambers. For example, when a high pressure fluid stream enters the chamber through, it exerts pressure on one of the pistons which in return pushes the low pressure fluid stream out. The rotor than rotates and the aforementioned piston assumes the initial position of the other piston. Now, the higher pressure fluid stream enters the transfer chamber and exerts pressure on the first piston which then forces the low pressure fluid stream out. The schematic diagram of the pressure exchanger is shown below in Figure 13(b).

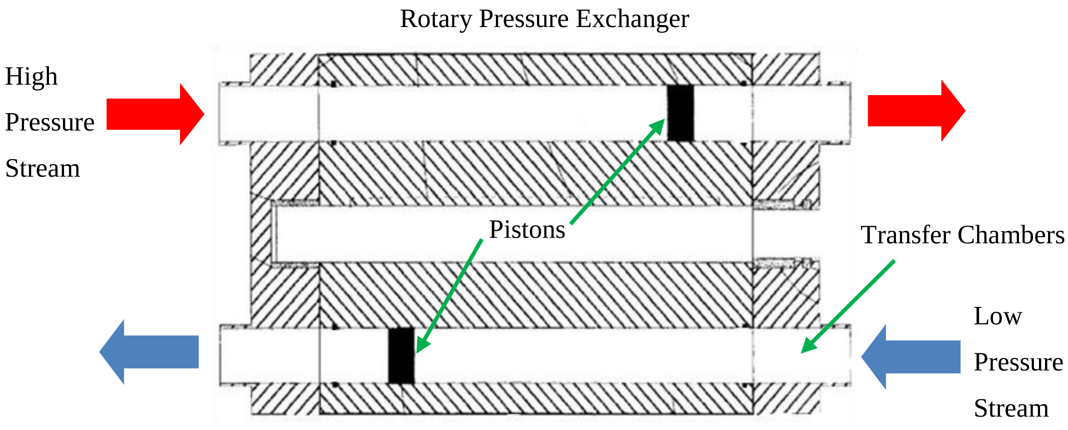

Figure 13(a): Pressure exchanger with individual piston separators

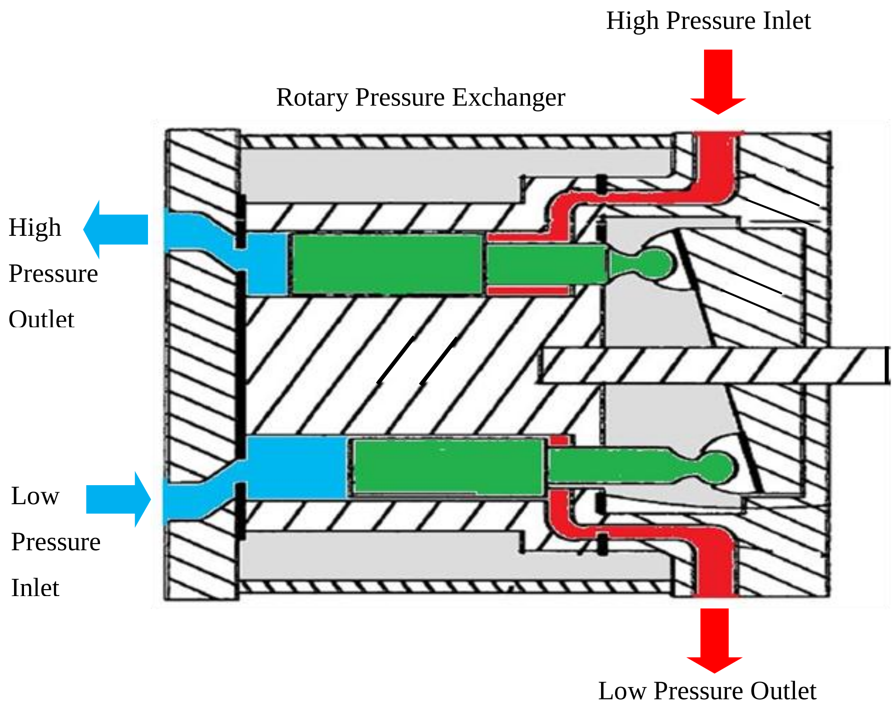

Figure 13(b): Pressure exchanger with connected piston separators

Piston (Liquid)

The main concept behind this mechanism is to use a liquid barrier instead of a solid one like a piston as mentioned in the previous section. The advantage of using a liquid barrier can greatly reduce the pressure energy loss due to friction and additional work done to displace a solid piston. The liquid used is usually of an inert nature whereby it does not allow both the working fluids to diffuse through it and also does not participate in any chemical reactions with the said fluids. Another method of creating such barrier without the use of any foreign or inert chemical is by allowing the two streams to diffuse and a small volume of the mixture of these streams will create a natural liquid piston/barrier which will be trapped permanently in the transfer chamber (MacHarg & McClellan, Pressure Exchanger Helps Reduce Costs in Brackish Water RO System: Ocean Reef Brackish PX, 2004; Al-Hawaj, 2004). Figure 14 illustrates the formation of a liquid piston in the transfer chamber of a pressure exchanger used in brackish water RO system.

Myran et al. (2012) mentioned that a common problem with pressure exchangers using physical septum (separator) is that it must be longer than the diameter of the pressure vessel to prevent binding or sticking in the vessel. Therefore, this limits the diameter of the vessel which can only be overcome by having a longer vessel to meet the required volume. Apart from that, while it is necessary to separate both fluids using a septum, it is equally important to incorporate a method of passing fluids through the septum in both directions to prevent stalling of the process and slamming of the septum against the ends of the vessel. This controlled process of mixing results in additional manufacturing costs and potential malfunctions. Hence, the inventors came up with a solution to address these issues by creating an inherently reliable virtual septum instead of a physical one. This can be achieved through the mixing interface that forms when two fluids are in contact. Ultimately, this modification allows flexibility in the designs such as having a shorter vessel with bigger diameter.

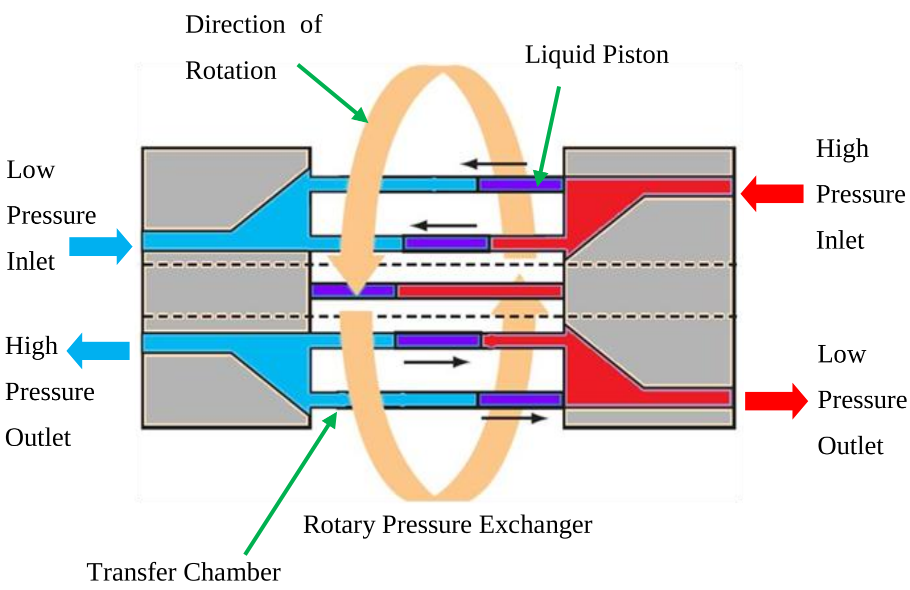

Figure 14: Liquid separator as a barrier between the two working fluids

<u>Membrane Separator</u>

According to Knight (2002), the use of membrane separator is more efficient in terms of reducing pressure energy loss as compared to a piston because it does not require to be moved from one end of the transfer chamber to the other end. Hence, the transfer chamber has to be as short as possible to enable efficient pressure transmission as depicted in Figure 15(a). Due to its elasticity, the membrane is attached in the centre of the chamber where only the peripherals of the membrane is fixed to the wall of the chamber while the centre portion of the membrane is free to move axially as it comes into contact with high pressure fluid streams. The cross-section of the transfer chamber is shown in Figure 15(b).

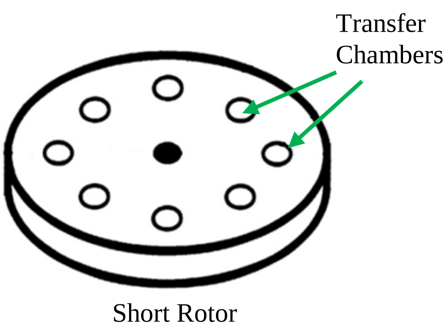

Figure 15(a): Rotor with multiple short transfer chambers

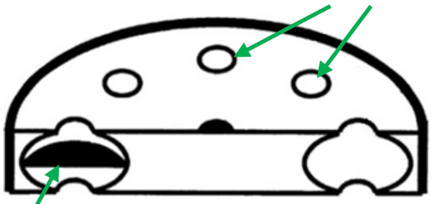

Figure 15(b): Cross-section of transfer chamber with an elastic membrane separator

Ball Separator

In the previous sections that were discussed under the topic of separators, the primary objective of the technologies used was to completely prevent the mixing or diffusion of two fluid streams. However, in this invention, the separators used are in the form of balls that partially function as pistons due to the fact that they are incapable of completely separating the two streams as shown in Figure 16 (Al-Hawaj, 2004; Aoki, 1986). The reason for using this mechanism has an advantage too. According to MacHarg (2010), by using a ball that has a diameter smaller than the diameter of transfer chambers, it is capable of moving freely in the chambers without as much friction or restriction as compared to a piston. Hence, this will be able to aid in a more efficient pressure transfer as lesser energy will be lost in moving the separator. On the other hand, this principle is only applicable if a small amount of diffusion of the two streams is permissible since complete separation cannot be achieved.

One of the problems with ball separators as highlighted by Bross et al. (2005) is that it causes significant energy losses to move the ball from one end of the passage to the other. Additionally, the rapid pressure changes may lead to sudden impact of the ball on the walls of the pressure exchanger resulting in cavitation.

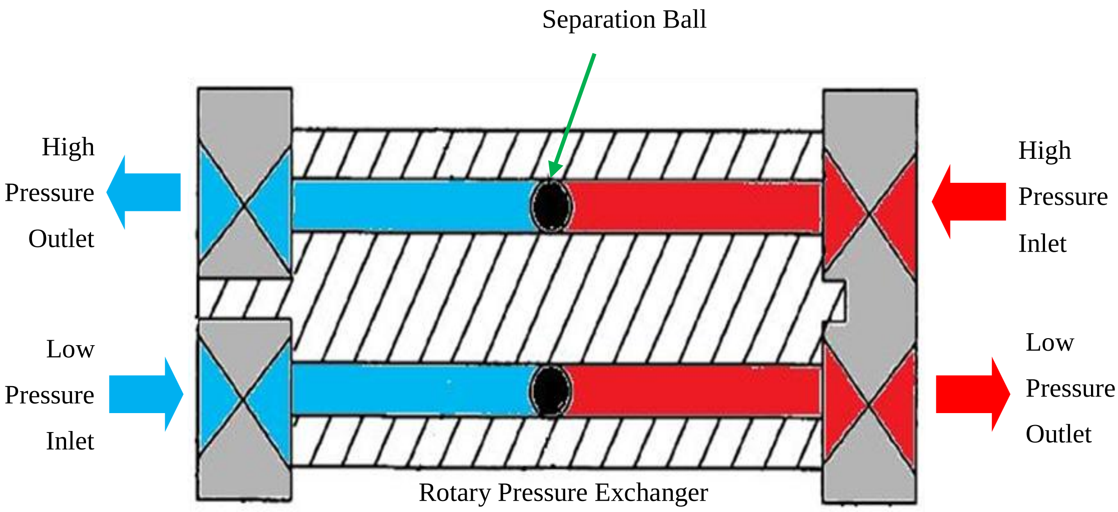

Figure 16: Ball separator in a pressure exchanger used in SWRO system

Pressure Exchangers without Separators

As mentioned previously, the application of pressure exchangers may not involve the use of separators where there are no obstructions in the transfer chambers that allow maximum or complete pressure transfer with a controlled diffusion of fluid from one stream to the other. Although mixing is acceptable in certain pressure exchanger, their designs are established to achieve a minimum amount of mixing. For example, Bross and Kochanowski (2007, 2010) mentioned that a flow guiding shape ensures impact-free oncoming flow to the rotor channels. As a result of this, uniform velocity distribution is established in the rotor channels. This prevents flow components from running transversely and lead to the development of eddies within a flowing column of liquid which ultimately results in mixing of fluids within the rotor channels. In a system, particularly the SWRO system, in which production of a pure liquid is the goal, mixing is a deleterious aspect.

Friedrichsen et al. (2013) mentioned that in an effort to minimize mixing of fluids in a SWRO system, he used a concentrate sensor to detect the concentration of the outlet high pressure supply seawater exiting the pressure exchanger. If the concentration is beyond a certain limit, it shows an unfavourable amount of mixing. Subsequently, the volume of low pressure seawater supplied into the pressure exchanger is reduced until the desired concentration is achieved.

2.2.5 Incorporation of Additional Features

Among the various literatures available on the existing pressure exchanger designs, some of which do stress on the importance of safety features that should be incorporated in the designs to allow safe and reliable operations of the device.

Reducing the Effects of Cavitations

One of the issues that may arise in pressure exchangers when dealing with high pressures is cavitations which may lead to wear and tear of the affected parts of the device. According to an online article, cavitation is a phenomenon that occurs when there is a rapid pressure change either due to pressurizing or depressurizing of fluids (Cavitation, 2013). Therefore Hauge (2006) proposed the incorporation of a depressurization groove to the design in order to bleed off some of the pressure from the high pressure stream before it goes into communication with a low pressure stream and also to pre-pressurize the low pressure stream before it goes into communication with a high pressure stream. This groove is an indentation built on the inner surface of the pressure exchanger's end cover that is exposed to the high pressure inlet stream. In Figure 17(a), we can see that graph shows a desired pressure exchanger that has minimal pressure fluctuation as a rotary-type pressure exchanger completes a full cycle. When compared with the graph in Figure 17(b), the pressure change in the second graph occurs very rapidly. This is an ideal condition to induce cavitations.

Hence, with reference to Figure 17(c), the depressurization groove allows some of the high pressure stream to be bled off the transfer chamber (red) before it is depressurized when exposed to the outlet channel in the end cover which leads to a significant pressure loss. The leaked high pressure fluid then re-enters the transfer chamber (blue) once depressurization is complete and just before pressurization takes place in the channel on the opposite side of the end cover.

In another invention, Friedrichsen et al. (2013) suggested the use of a throttle valve on the high pressure concentrate inlet stream to prevent the risk of cavitation.

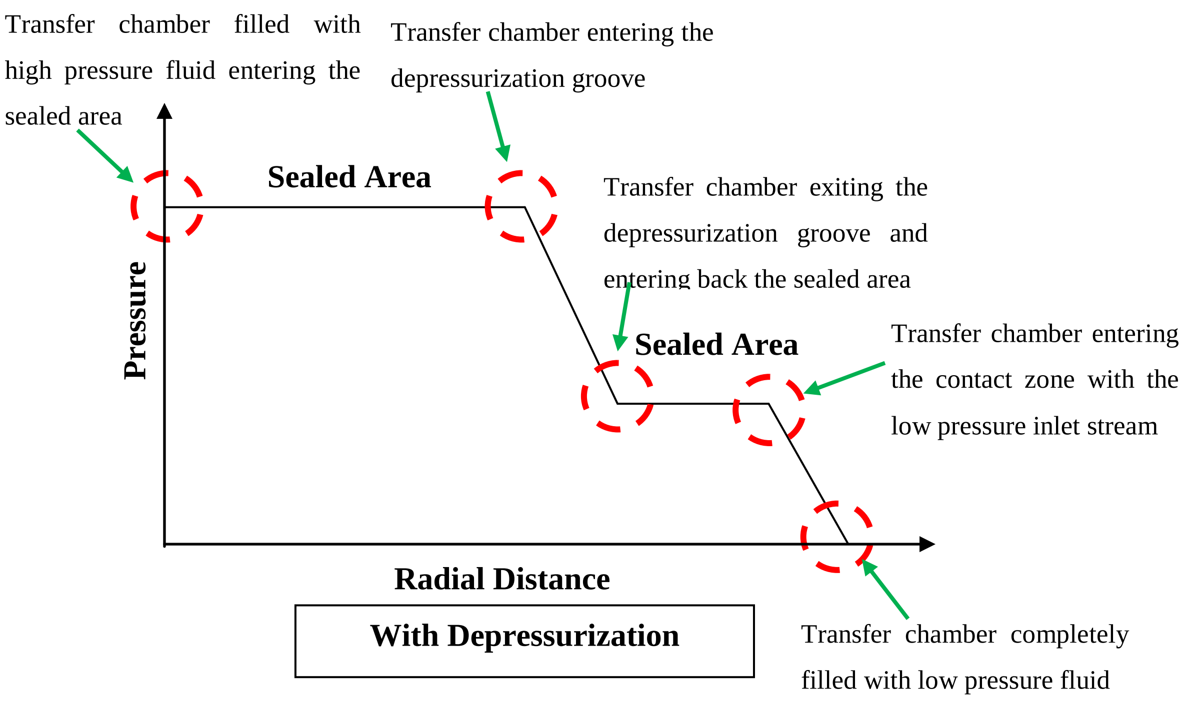

Figure 17(a): Pressure change profile in channels for a complete rotation with depressurization groove

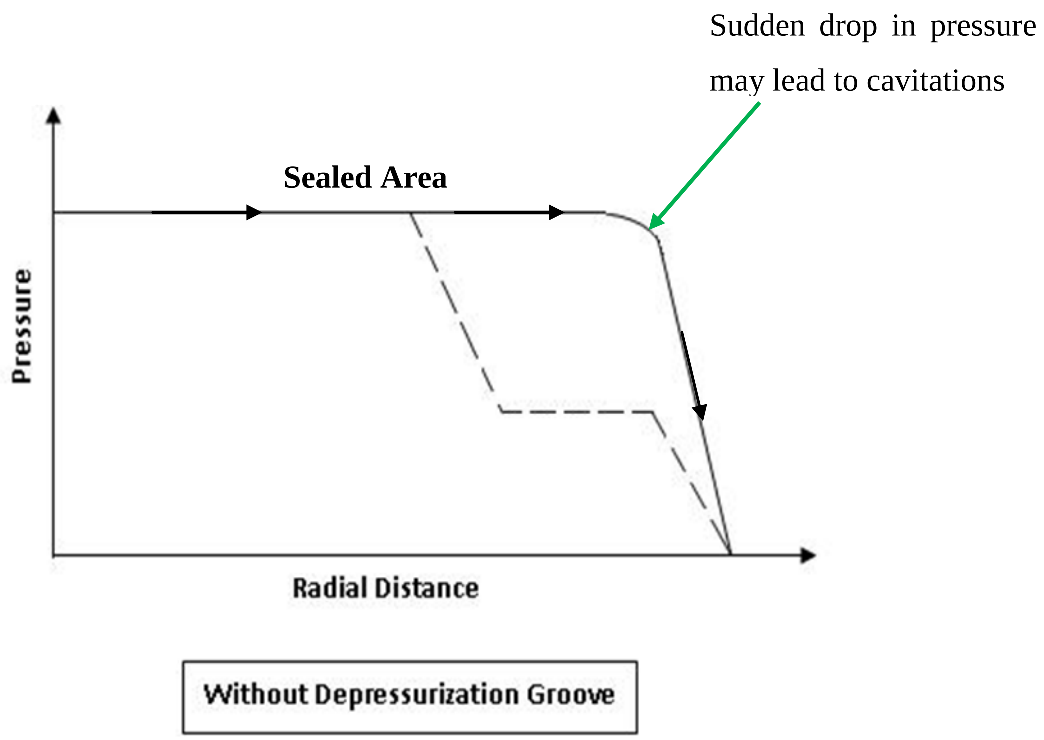

Figure 17(b): Pressure change profile in channels for a complete rotation without depressurization groove

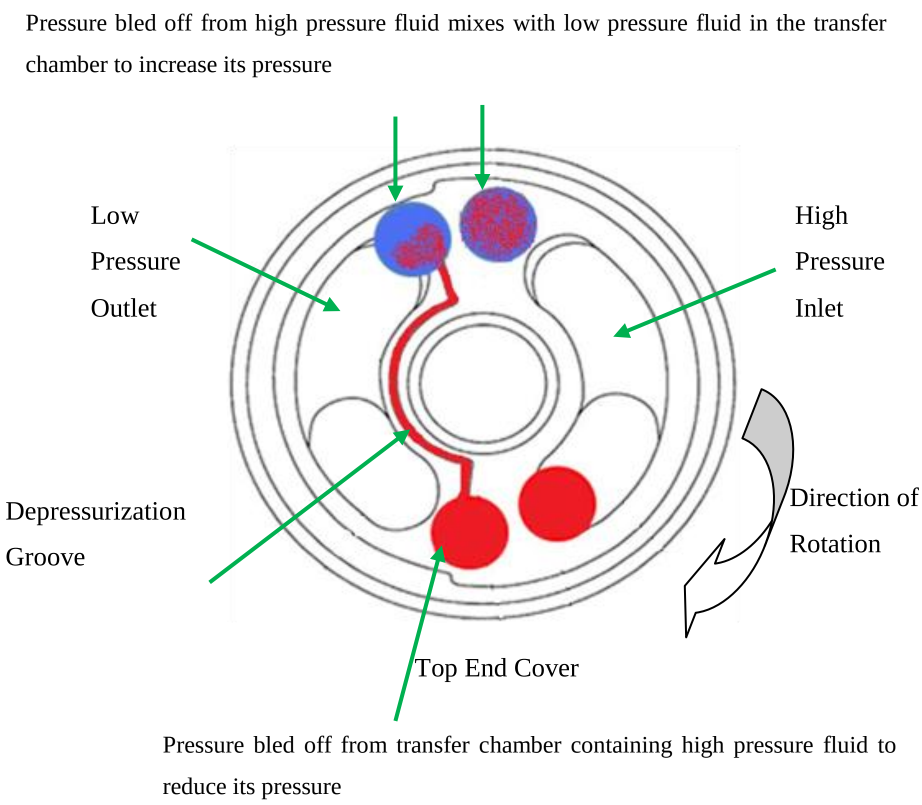

Figure 17(c): End cover with depressurization groove

Providing Dimensional Stability

From the description of the basic components that make up a pressure exchanger, it is known that the end covers have flat inward end faces interface with flat end faces of the rotor. These are important components of a rotary pressure exchanger because during high pressure operation such as in the seawater reverse osmosis (SWRO) system, the pressure of the incoming brine stream is approximately 700-1200 psi greater than that of the incoming seawater stream. Hence, it is vital to provide dimensional stability to these components, especially the rotor which is mounted between end covers in a housing which is subject to full compression stress (Hauge, 2003).

Stover (2011), in his findings stated that improved operation and stability of rotary pressure exchangers can be accomplished by supporting inward facing surfaces of the end covers by balancing the forces to which these end covers are constantly being subjected during operation. For example, in a normal SWRO arrangement, the outward end faces of the two end covers will be respectively subjected either to the pressure of the high pressure incoming stream of brine, or to the pressure of the high pressure outgoing stream of seawater. In both cases, the inward end faces will be supported only peripherally where they contact the sleeve or body of the pressure exchanger. Therefore, to maintain the overall stability of these end covers, central support needs to be provided to balance these pressures on both sides of the end covers. One way to achieve this is by providing a chamber within the rotor that is in communication with either the high pressure incoming brine stream or the pressurized seawater discharge stream. The pressurized chamber will then be able to balance the inward and outward forces on both end covers.

Proper Sealing Mechanism

Pfenning et al. (1979) proposed a design which is mechanically simplified and efficient in sealing against the escape and loss of the fluids with a relatively less power requirement to operate. The key element for this invention is an improved version of seal plates. The rotor seal plates employed at opposite ends of the rotor assembly are disk-shaped and has a sealing surface into which are formed equally spaced ports arranged circumferentially. As contrasted with the conventional rotor-seal plates, the present invention employs floating seal plates which can undergo canting movement during rotation. The seal plates are also maintained at an even contact pressure during all increments of the rotational speed of the rotor. Hence, the advantage of this design is the construction of the apparatus which permits a very significant reduction in the need for O-ring seals employed in the sealing structure associated with a rotor while achieving an improved sealing arrangement.

The problem of axial clearances between the rotor and casing end surfaces also affects the efficiency and operating behaviour of a pressure wave supercharger or a pressure exchanger (Rao, 1981a; Haase & Mayer, 1984; Lott & Hashemi, 1971; Loeffler & Higginbottom, 1975; Gardiner, 1961; Herger & Wunsch, 1966; Alcock, 1954). Satisfactory operation of the pressure wave supercharger and good efficiency can only be obtained using very small axial operating clearances because the leakage losses at the end faces of the rotor can be minimized. However, the rotor must be simultaneously prevented from touching the casing end surfaces which is a challenge by itself due to thermal and the danger of rubbing between the rotor and the casing end surfaces resulting from the rotor vibrations. According to Kirchhofer and Schelling (1985), a solution to avoid rotor damage due to possible heavy contact is the addition of a rubbing layer, for example a graphite/nickel layer can be applied to the casing or rotor end surfaces. However, the rubbing layer is only abraded in the radial region of the relatively sharp-edged cell walls. The layer in the region of the thick hub tube is merely compressed and this may lead to the rotor becoming jammed in the case of severe rubbing. Besides that, due to ageing of the layer, it can flake off and hence lead to poor efficiency of the pressure wave supercharger. Alternatively, rubbing layer applied by flame-spray method is too expensive for mass production of the device. Hence, a gas dynamic pressure wave supercharger was designed to address this issue. In this invention, the supercharger comprises a rotor located between an air casing and a gas casing. On the air casing side, at least one of the two mutually facing end surfaces of the rotor and the air casing is convex. The shape serves to maintain an axial clearance when the motor operates from a cold start-up. Alternatively, on the gas casing side of the rotor at least one of the two mutually facing end surfaces may be concave.

The importance of a proper sealing mechanism at the end caps was also highlighted by Hauge (2012) for an improved manufacturing efficiency. The pressure exchanger according to his embodiment consists of two opposite facing end caps connected mechanically with a seal. Since the end caps are either casted or welded with structurally integrated ports, it is believed that the pressure vessel will not develop external leaks through the seals.

Jendrassik (1957, 1965) mentioned that the clearance spaces between the end plates and the cell rings of the rotor should be made as small as possible in order to restrict working fluid leakage. One way to achieve this is by having labyrinth seals. However, if the fluid is a hot gas or contaminated by soot or dust, the clearance spaces may be clogged or distorted leading to a damaged cell ring and/or end plates. In order to address this problem, sealing gas can be introduced into the clearance spaces whereby it restricts the working gas from flowing into the said spaces. This sealing gas may be extracted from the high pressure passage or cell of the pressure exchanger itself and the gas can be introduced either throughout the entire sealing space or at several equally-spaced positions. Alternatively, the seal gas may also be obtained from an external source such as an air compressor. Furthermore, in addressing the problem of fluid leakage, the clearance between the end covers should be kept as small as possible while taking into account of thermal expansion of the rotor axially when high temperature fluids are used. Hence, the end plates and rotor are arranged adjacent to one another in association with bearings that permit relative rotation of the rotor and provide sufficient axial spacing for thermal expansion. During operation, the end plates can be prevented from moving laterally by support means that keep them rigid.

In another invention, Honma and Nakino (2005) proposed a rotary pressure exchanger that uses gland packing as its sealing means. One of the advantages of this sealing mechanism is that it has a high durability against high temperatures. There is also a suggested method of recovering fluid that has leaked past the sealing means. This minimizes the necessity for replenishing any lost fluid and since it is returned to the downstream side, the performance of the operating part will not be affected.

The seal structure described by Lott and Hashemi (1971) comprises of a rigid seal plate to counter the fluid leakage as well as the shearing forces resulting from the rotational motion of the rotor with respect to the stationary end covers. Some of the important criteria that need to be met by this sealing element are, low coefficient of friction, high shear strength and a monoplanar bearing surface that forms a bearing between the seal plate and the rotor face. Lastly, this sealing mechanism should be economically constructed and not rapidly destroyed or weakened by the passage of fluid across the seal. However, Loeffler and Higginbottom (1975) discovered several drawbacks to the previous invention and developed an improved face seal that seals effectively while exerting a lesser resistance to rotation of the rotating face. As a result, less power is required to turn the rotor while the face seal itself sustains less wear and stress. The face seal is also improvised such that it reduces the magnitude of shock loading when high pressure fluid is suddenly introduced into the pressure exchanger.

Wunsch (1973) noticed that in most applications of pressure exchangers or pressure wave machines, the use of bearing is crucial in allowing the rotor to rotate between the stationary end plates and the housing. The use of bearings, whether plain or rolling type, require lubricating oil to ensure smooth movement of the components during operations. Therefore, apart from reducing fluid leakage from inside the rotor cells, this oil must also be prevented from entering the rotor cells. Hence, it is the aim of this invention to seal and prevent oil escaping from the bearing chamber to the rotor by using stationary rubber sealing rings with elastic sealing edge that is pressed radially on the rotating shaft of the rotor by means of a spring. However, this sealing mechanism is not appropriate to be used in high temperature applications as rubber becomes brittle and cracks, compromising the integrity of the seal. In order to counter this issue, the inventor suggested that the sealing ring is cooled by oil-spray cooling technique and he also recommended the use of oil deflector plates which guides escaped oil into a separate hub.

High Operating Temperatures

According to Rao (1981b), the use of exhaust gas together with wave compression action results in an elevated temperature in the rotor cells. Hence, the rotor was designed using ceramic materials that have capabilities of withstanding high operating temperatures and having relatively low expansion coefficient. The relevancy of using ceramic material in designing the rotor of a pressure exchanger is further supported by Mayer (1989) who stated that ceramic materials have high thermal conductivity. Hence, they are able to reduce thermal stresses generated within the rotor.

Haase and Mayer (1984) explained that in applications involving high temperature fluids such as in gas turbines, the walls of the rotor has to be sufficiently thick to withstand thermal and dynamic stresses. This can be achieved through the use of metal super alloys. However, the challenge arises to maintain a delicate clearance space between the rotor, housing and end covers. This is because, changes in temperature causes expansion and contraction of these components which may result in scrapping and damage to their surfaces. One way to overcome this issue is by having a relatively thin wall to enable rapid heating and cooling. However, this signifies a greater heat loss and therefore compromises the equipment's efficiency. According to this invention, the use of ceramic materials in constructing the rotor, housing and end covers should be adopted to resolve the above issues. The reason for this is because, when hot and cold air is charged into the rotor, a regenerative heat exchange takes place which impairs the rotor's efficiency. However, in the case of a ceramic rotor, the heat exchange between the rotor and the charging air is lesser due to the lower specific heat capacity of ceramic materials. Hence, the loss in efficiency is also minimized. Furthermore, ceramic materials cost lower than metal super alloys for mass production.

Since pressure exchangers or pressure wave machines with integrated combustion chamber such as in a gas turbine are subjected to very high temperatures, Zauner (1996) invented a method to cool the rotor without the use of cooling air compressors or external cooling media. In his invention, cooling air conduits are arranged in the rotor radially inward and outward of the cells. In addition, the rotor is surrounded by at least one metal sheet guide at its inner and outer peripheries to provide further cooling passages. In this cooling mechanism, a small part of entering air is branched off before it mixes with the fuel in the combustion chamber. The air that is branched off is led through the casing to the rotor section via cooling air conduits. This method enables effective cooling of the pressure wave machine without any additional design complication. Furthermore, the magnitude of cooling effect can be manipulated according to the operating conditions by controlling the width of the cooling air inlet and outlet openings.

Eliminating Unwanted Vibrations and Noise

The production of excessive vibrations and noise from any device including a pressure exchanger is unfavourable as it compromises the energy efficiency of the equipment. This is due to the loss of energy in the form of kinetic and sound energies. Hence, Rao's (1981b) invention incorporates an improved elastomeric mounting mechanism so that the vibrations can be isolated from the rotor structure thus preventing damage to its ceramic body while dampening the noise generated from these vibrations. In a similar invention by Rahnke (1982), unwanted noise, particularly high frequency noise is further reduced with unequal sizes of air ports that moderate the siren-like effect that is produced by a conventional multi-cycle supercharger having equal size ports.

According to Kentfield (1963), one of the disadvantages of a pressure exchanger is the noise generated during operations. Conventionally, there are two ways in which this problem can be addressed. One of which is through the use of silencers for gas streams leaving and entering the apparatus as well as sound-insulating lagging around the pressure exchanger. Unfortunately, this has a merely palliative effect and does not have any significant contribution to the reduction of noise from the device. The other alternative method is based on the concept that the frequency of oscillation of the noise emitted by a pressure exchanger corresponds directly to the frequency with which the cells move past a fixed point. Hence, the amount of noise can be reduced if not eliminated by increasing the oscillation above the upper limit of audio frequency. This can be achieved by increasing the number of cells in the cell ring of the rotor. However, such an increase will lead to other complications, primarily the increase in boundary layer fluid in the cells which leads to loss in efficiency.

The time taken for a cell to become fully opened at a port = $\frac{60}{Nn}(\sec onds)$

Where, N – Rotational speed of the cell ring in rpm

n – Number of cells in the cell ring

The time taken for a wave to traverse the entire length of the cell = $\frac{l}{a_0}(\sec onds)$

Where, l – axial length of each cell in feet

a_0 - velocity of sound in feet/second

Ratio, $\delta = \frac{\frac{60}{Nn}}{\frac{l}{a_0}}$ $\qquad \delta = \frac{60a_0}{Nnl}$

Based on the previous equation, the ratio should not be less than 0.3 in order to prevent the loss of efficiency of the pressure exchanger. Therefore, realizing this, Kentfield (1963) suggested that in order to increase the oscillation above the upper limit of audio frequency, the number of cells moving past a fixed point per second has to be equal to or more than 16,000 cells per second (cps). Therefore, $f_{cell} \geq 16{,}000cps$.

$\delta = \frac{60a_0}{Nnl}$ ----- (1) $\qquad f_{cell} = \frac{Nn}{60}$ ----- (2)

Substituting (1) into (2),

$$\delta = \frac{a_0}{l} \cdot \frac{1}{f_{cell}}$$

$$f_{cell} = \frac{a_0}{l\delta}$$

The noise produced by a pressure exchanger results from the periodical impingement of inlet and outlet flow on the cell walls. The individual frequencies cause the noise to be perceived as a shrill whistle which is unpleasant and disturbing even when silencers are used. Hence, Leutwyler and Wunsch (1971) proposed and invention that relates to a design that reduces noise based on the concept that the frequency is determined by the number of cells in the rotor and its speed of rotation. However, they disagreed with the previously discussed invention that aims to increase the number of cells passing a fixed point in order to locate the frequency above audible range. This would require a huge amount of cells on the rotor that may lead to loss in frequency due to heat transfer and friction in the finely subdivided rotor. The solution to this problem resides in that the individual frequencies of the rotational sound are resolved into ranges of frequencies with varying amplitudes. This can be achieved by arranging the cells into groups in which the width of cells in the same group are similar while those of adjacent groups are different.

Increased Efficiency

The efficiency of a pressure exchanger depends on various design-related factors. As such, the invention of this device can be manipulated in a variety of scope and concept. One of which was seen in the previous section whereby the pressure exchanger was designed to generate a self-rotational motion of its rotor instead of depending on an external source of power such as an electric motor. Apart from depending on design modifications and fluid flow to induce such rotational forces, other forms of drive mechanism that does not require any power source can be implemented. In one such application, El-Nashar et al. (1987) and Mayer (1989) proposed the use of high pressure exhaust gas from internal combustion engines to drive the rotor in an application where a pressure wave supercharger is used to supply compressed air into the engine. The use of an electric motor in the context of a pressure exchanger is said to be inefficient due to several disadvantages as discovered by Hashemi (1972). This is primarily due the device's dependency upon an external power source to operate. Secondly, a great power loss will be incurred with the use of gear reduction mechanism to control the rotational speed of the rotor. Apart from that, the motor has to have a higher power rating than that which is required for normal operation of the pressure exchanger during average load requirement. This is due to the variable load generated by a pressure exchanger.

Martin and Stover (2011) have found that it is practical to employ two or more inlets and two or more outlets in each end cover by arranging them so that each channel will charge and discharge at least twice at each end. As a result, such a rotary exchanger will have the potential to increase the pressure of a liquid with a volume that is twice as large during each revolution as compared to a conventional pressure exchanger.

As mentioned previously, pressure exchangers can be incorporated in gas turbine systems whereby it is placed between the turbine and the rotary compressor instead of a combustion chamber. As such, an additional feature was introduced by Coleman (1958) to increase the overall efficiency of the system. He utilized the heat from the turbine's exhaust gases to pre-heat the compressed air before it comes into contact with the fuel in the combustion chamber within the pressure exchanger. This can be achieved via a pressure exchanger unit in an external heat regeneration circuit.

Based on Kentfield (1965) observation, most pressure exchangers have cells that are of constant volume. According to him, this is disadvantageous because it limits each cycle's pressure ratios. Besides that, direct production of shaft power is also difficult with this type of pressure exchangers unless there is an incorporation of a turbine as a separate component or constructed integrally with the pressure exchanger. Hence, the invention aims to provide a pressure exchanger with variable volume cells in order to improve the device's overall efficiency. The cell volume variations can be obtained by eccentrically mounting the cell ring with radially sliding cell walls in relation to the rotor's structure.

As described previously, a wave compression supercharger is a device used to exchange pressure energy between a pressurized hot exhaust gas from an internal combustion engine and ait at atmospheric pressure. The ambient air is compressed to be fed into the engine itself. This greatly reduces the power consumption to compress the air via other mechanical means such as a compressor. In order to further increase the efficiency of a wave compression supercharger used in an internal combustion engine, Vallance et al. (1982) invented a mechanism that produces three compression cycles per revolution of the rotor shaft. Hence, the design of this device includes two end plates at each axial ends of the rotor called the exhaust gas port plate and the air port plate. Each of these plates accommodates three inlet and three outlet ports. This configuration allows three compression cycles to take place during each rotor revolution. One of the advantages of this invention is that the length of the rotor and the overall size of the wave compression supercharger are greatly reduced as compared to a device with only two compression cycles per rotor revolution. In addition to that, the shortened rotor length enables the clearance between the rotor and its axial ends to be maintained at a closer tolerance despite high operating temperatures. It is preferable to keep the end clearance as small as possible to minimize gas leakage.

Reducing Mass and Cycle Time

A typical rocket engine requires propellants to be fed at high pressure. This in return compromises the overall mass of the storage tank and other necessary equipments to maintain a high operating pressure at all times. Conventionally, this has been done using a pump. However, pumps are generally very complex and expensive as they require their own driving means, such as an engine. This method burns a significant percentage of the total propellant. Alternatively, pressurized fluids can be used to pressurize the propellant via a pressure exchanger. This too has a problem whereby the walls of the pressurized fluid's tank must be thick enough to withstand the high pressure contained in it. This contributes to an increased weight which is unfavourable in a rocket engine system (Lanning & Blackmon Jr., 2001).

According to Sobey (1965), the propellant storage tank need not have a thick wall since it contains low pressure fluid. However, the tank in which pressurized fluid exerts force on the propellant needs to have thick walls to withstand the high pressure. So, he suggested the use of a smaller tank for this purpose and concluded that there is a reduction in the total weight of the rocket engine system.

Knight (2006) then discovered that the size reduction of the storage tank causes decrease in the cycle time as well as the opening and closing time of the valves. This indirectly compromises the efficiency of the system as the rocket engine will not be receiving sufficient propellant. Hence, he proposed a pressurizer comprising of at least two storage tanks operating alternately whereby at any one time, one tank is being filled with propellant to be pressurized while the other tank is discharging high pressure propellant into the rocket engine.

CHAPTER 3

METHODOLOGY

3. METHODOLOGY

3.1 Research methodology

3.1.1 Project Activities

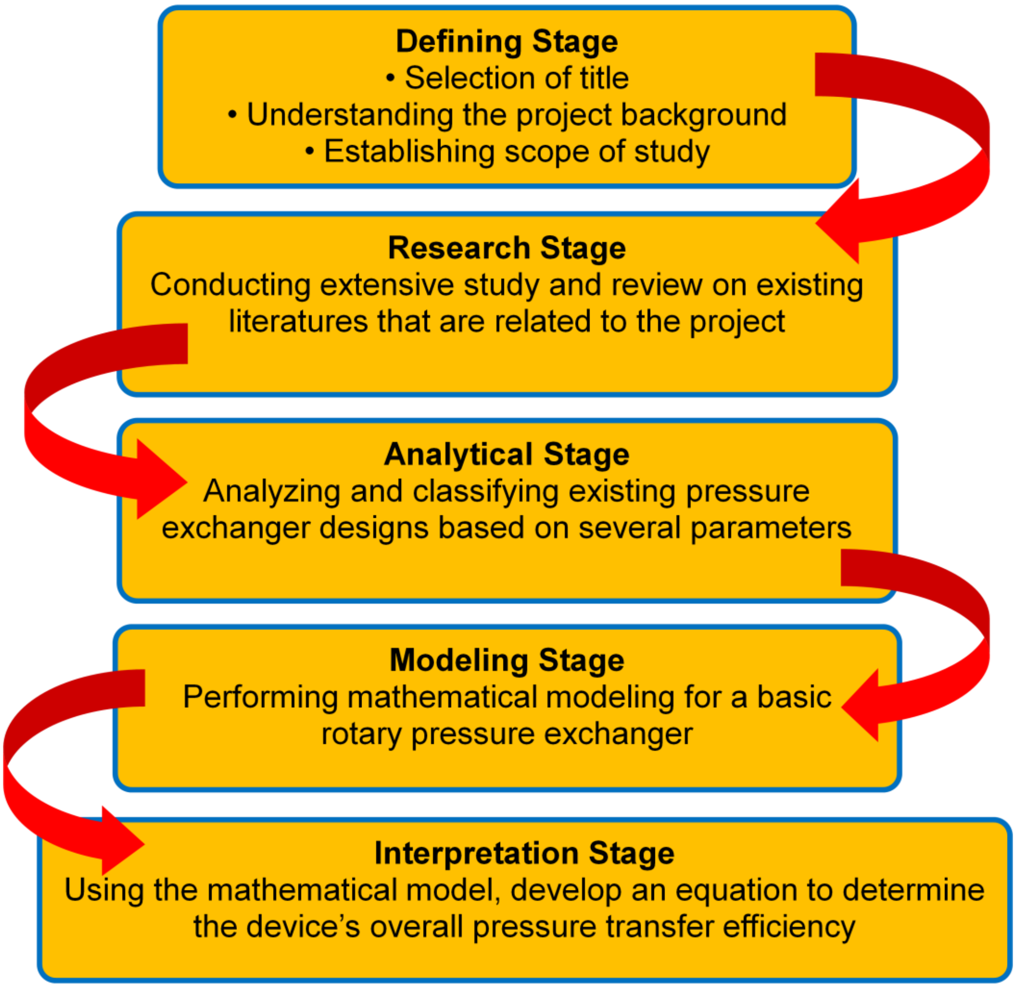

Figure 18: Flow of research procedure

3.2 Key Milestone

Several key milestones for this research project must be achieved in order to meet the combined objectives of Final Year Project I (FYP I) Final Year Project II (FYP II).

3.2.1 Key Milestone for FYP I

- Selection and confirmation of project title
- Completion and submission of Extended Proposal
- Oral presentation of Proposal Defence
- Submission of draft and final Interim Report

3.2.2 Key Milestone for FYP II

- Completion and submission of progress report
- Pre-SEDEX preparation and oral presentation
- Completion and submission of final draft report (soft bound)
- Completion and submission of technical paper
- Oral presentation (VIVA)
- Completion and submission of project dissertation (hard bound)

3.3 Gantt Chart

3.3.1 FYP I

NO	WEEK / DETAIL	1	2	3	4	5	6	7	Mid-semester Break	8	9	10	11	12	13	14
1	Selection of Project Title															
2	Preliminary Research Work and Literature Review															
3	Submission of Extended Proposal							●								
4	Preparation for Oral Proposal Defence															
5	Oral Proposal Defence Presentation															
6	Continuation of Project Work															
7	Preparation of Interim Report															
8	Submission of Interim Draft Report														●	
9	Submission of Interim Final Report															●

3.3.2 FYP II

NO	DETAIL / WEEK	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
1	Continuation of Project Work								**Mid-semester Break**								
2	Submission of Progress Report									●							
3	Continuation of Project Work																
4	Pre-SEDEX Presentation												●				
5	Submission of Final Draft Report (soft bound)													●			
6	Submission of Dissertation (soft bound)														●		
7	Submission of Technical Paper														●		
8	Oral Presentation (VIVA)															●	
9	Submission of Project Dissertation (hard bound)																●

CHAPTER 4

RESULTS AND DISCUSSION

4. RESULTS AND DISCUSSION

4.1 Project Findings

The demand for fresh water has led to major breakthroughs in the field of seawater desalination. One particular process that has been proven effective is the seawater reverse osmosis (SWRO) system. However, the process itself requires a very high pressure feed stream in order to operate. The use of turbines and pumps are some of the conventional energy conservation techniques that were adopted to increase the pressure of the feed stream. However, the inefficiencies of such techniques and the high cost of apparatus have been major disadvantages in the said industry (Cheng & Cheng, 1970). Hence, Moloney (1964) stated that the main drawback in desalinating seawater or brackish water economically is the high cost of power requirement. An online article defines the application of pressure exchangers in industries as a major breakthrough in reducing the overall energy consumption and thus leading to a more sustainable use of energy (Pressure Exchanger, 2012). This is because, the device has a long and trouble-free operating life which is uninterrupted by plugging of valves or binding of pistons which is common in pumps (Shumway, 2007). Although its implementation is currently only limited to the desalination of seawater, the potential that an energy recovery device like a pressure exchanger possess can be clearly seen in terms of its energy and cost reduction capabilities (Andrews, 2009). Table 1 shows some of the findings based on industrial application of a self-rotating pressure exchanger in a SWRO plant. In another news article by KSB AG (2008), the use of pressure exchanger in Malta has been proven to reduce desalination cost.

Friedrichsen et al. (2013) stated that the incorporation of a pressure exchanger is based on the task of keeping the energy consumption in a SWRO system as low as possible. This statement was supported by Myran et al. (2012) who regarded work exchanger (pressure exchanger) as an energy recovery device used to reduce the net energy required for the process of reverse osmosis in which the potential (pressure) energy contained in the concentrate leaving the reverse osmosis (semi-permeable) membrane module is recovered. The amount of potential energy contained in the concentrate stream is typically sixty percent (60%) or more of the total energy required by the reverse osmosis process when applied to a solute such as seawater (Shumway, 1998). In addition, this device has the capacity to increase efficiency by recovering as much as ninety-eight percent (98%) of the potential energy contained in the concentrate stream. In this invention, the work exchanger system utilizes pairs of pressure vessels operating in an appropriate sequence. However, the pressure recovered as mentioned is insufficient to fulfil the pressure required for the SWRO unit. Thus, Shumway (2007) suggested the use of an integrated booster pump that works with the pressure exchanger to meet the remaining pressure requirement. The booster pump is comprised of a high pressure conduit that is in fluid communication with an impeller.

Table 1: Comparison of SWRO systems with and without a pressure exchanger

Characteristics	SWRO System	
	With Pressure Exchanger	Without Pressure Exchanger
Mode of pressurizing the feed stream	• 40% using a high pressure pump • 60% using a pressure exchanger	• 100% using a high pressure pump
Energy requirement	• The pressure exchanger does not require any external force to operate	• Consumes high energy during operation.
Concentrated output brine	• Used to transfer its high pressure energy to the incoming feed seawater	• Channelled directly into the sea as a waste stream without utilising its high pressure energy
Cost of desalination	• \$0.46/m^{3}of seawater-accounts for the 40% usage of high pressure pumps	• \$1.15/m3 of seawater-accounts for the 100% use of high pressure pumps (Pressure Exchanger, 2012) • 2.5 times higher as compared to using a pressure exchanger

4.2 Project Analysis

4.2.1 Establishing Parameters for Mathematical Modelling

Based on the study of various existing literatures on pressure exchangers, I was able to categorize all the important parameters that need to be considered for modelling a self-rotating pressure exchanger. These parameters were selected based on their influence on the dynamics of fluid flow in the different sections of the pressure exchanger. Discussion on the parameters is as the following:-

Fluid characteristics

i) Incoming and outgoing flow rates of both fluids.

ii) Density and viscosity of both fluids.

iii) Diffusion rate of each fluid when in contact with one another specifically for pressure exchangers without separator / where minimal mixing of fluid is permitted.

iv) Length / thickness of fluid plug that is formed in between the two fluids in contact in the transfer chamber due to diffusion / mixing.

End cover channels characteristics

i) Thickness, shape, cross-sectional area and volume of both inlet and outlet channels at the two end covers.

ii) Area of sealed region on the inner surface of the end cover that separates the inlet and outlet fluid flow.

Rotor characteristics

i) Rotor speed / revolution per minute (RPM) that determines the number of transfer chamber that is in fluid contact with the end cover channels at any one time.

ii) Determines the allocation of time for each transfer chamber to be in fluid contact with the inlet and outlet fluid streams at the end cover channels.

iii) Influences the rate of diffusion and thus controls the amount of mixing between the two fluid streams.

Transfer chamber characteristics

i) Length, shape, cross-sectional area and volume of transfer chamber.

ii) Resistance of fluid flow in the transfer chambers.

These factors are the prime factors that will be taken into consideration during the preliminary stages of modelling a self-rotating pressure exchanger. The following table summarizes the aforementioned discussion.

Table 2: Factors to consider for pressure exchanger modelling

Fluid	Inlet/Outlet Channels at End Covers	Rotor	Transfer Chamber
• **Flow** • **Fluid Properties** - Density - Viscosity • **Rate of diffusion** - Volume of fluid lost per cycle due to mixing - Length of fluid ‘plug’ formed in the transfer chamber	• **Dimension of each channel** - Thickness - Shape - Cross-sectional area - Volume • **Area of sealed region**	• **Rotational Speed (RPM)** **-** Duration for each transfer chamber to be in fluid contact with the inlet/outlet channels at end covers - Number of transfer chambers that are exposed to the inlet/outlet channels at any one time	• **Dimension of transfer chamber** - Length - Shape - Cross-sectional area - Volume • **Resistance to fluid flow**

4.3 Modelling Phase

For the purpose of this report, the 4 main streams will be identified as the following:-

Fluid A- Fluid that is used to pressurize

Stream A1- High pressure inlet

Stream A2- Low pressure outlet

Fluid B- Fluid that is pressurized

Stream B1- Low pressure inlet

Stream B2- High pressure outlet

Figure 19: Identification and labelling the main streams of a pressure exchanger

4.3.1 General Efficiency Equation

$\eta =$ (Energy Out / Energy In) x 100%

$$\eta = \frac{\Sigma QP_{out}}{\Sigma QP_{in}} \times 100\%$$

Where Q = Volumetric flow rate (m^3/s)

P = Pressure (kPa)

$$\eta = \frac{(QP)_{A2} + (QP)_{B2}}{(QP)_{A1} + (QP)_{B1}} \times 100\% \text{ --- } (1.1)$$

4.3.2 Conservation of Mass and Energy

The calculations that were performed are all based on SI units.

Conservation of Mass

For accumulation = 0,

$m_1 = m_2$ where, $m = Q\rho$

$Q_1\rho_1 = Q_2\rho_2$

For accumulation $\neq 0$,

$$Q_1\rho_1 - Q_2\rho_2 = V\frac{\delta\rho_{avg}}{\delta t}$$

Steady-state: $\delta t = 0$

Incompressible/ Newtonian fluid: Constant ρ, $\delta\rho_{avg} = 0$

Therefore,

$$Q_1\rho_1 - Q_2\rho_2 = 0$$

$$Q_1\rho_1 = Q_2\rho_2$$

$$U_1A_1\rho_1 = U_2A_2\rho_2 \Rightarrow \text{Continuity equation}$$

Conservation of Energy

Total Energy = $U + mgh + \frac{mv^2}{2} + PV$ where, U - Internal Energy

mgh – Petential Energy

$(mv^2)/2$ – Kinetic Energy

PV – Pressure Energy

Energy In: E_1 – Energy of the high pressure inlet stream

W_i – Work done on the system (Eg: the use of motor to rotate the rotor)

Energy Out: E_2 – Energy of the high pressure outlet stream

W_o – Work done by the system (Eg: to overcome frictional force)

Energy Balance

$$E_1 + W_i = E_2 + W_o$$
$$E_2 = E_1 + W_i - W_o$$

$$Since\ \ E = U + mgh + \frac{mv^2}{2} + PV$$

$$(U_2 + m_2gh_2 + \frac{m_2v_2^2}{2} + P_2V_2) = (U_1 + m_1gh_1 + \frac{m_1v_1^2}{2} + P_1V_1) + W_i - W_o \text{ --- (2.1)}$$

Assumptions

For E_1

$$E_1 = U_1 + m_1gh_1 + \frac{m_1v_1^{\ 2}}{2} + P_1V_1 \text{ --- (2.2)}$$

1. Since the internal energy, U is constant, U_1=U_2

$$E_1 = m_1gh_1 + \frac{m_1v_1^2}{2} + P_1V_1 \text{ --- (2.3)}$$

2. Assuming m_1=1kg

$$E_1 = gh_1 + \frac{v_1^2}{2} + P_1(\frac{m}{\rho})_1$$

$$E_1 = gh_1 + \frac{v_1^2}{2} + \frac{P_1}{\rho_1} \text{ --- } (2.4)$$

3. Assuming horizontal flow from the source (elevation=0m)

$$E_1 = \frac{v_1^2}{2} + \frac{P_1}{\rho_1} \text{ --- } (2.5)$$

Hence, $E_2 = \frac{v_1^2}{2} + \frac{P_1}{\rho_1} + W_i - W_o$ --- (2.6)

For E_2

$$E_2 = U_2 + m_2 gh_2 + \frac{m_2 {v_2}^2}{2} + P_2 V_2 \text{ --- } (2.7)$$

1. Since the internal energy, U is constant, $U_2 = U_1$

$$E_2 = m_2 gh_2 + \frac{m_2 v_2^2}{2} + P_2 V_2 \text{ --- } (2.8)$$

2. Conservation of mass (no accumulation), $m_1 = m_2$=1kg

$$E_2 = gh_2 + \frac{v_2^2}{2} + P_2 \left(\frac{m}{\rho}\right)_2$$

$$E_2 = gh_2 + \frac{v_2^2}{2} + \frac{P_2}{\rho_2} \text{ --- } (2.9)$$

3. Horizontal flow through the pressure exchanger (horizontal configuration)

$$E_2 = \frac{v_2^2}{2} + \frac{P_2}{\rho_2} \text{ --- } (2.10)$$

Basic Equation

$$E_2 = E_1 + W_i - W_o$$

$$E_2 = \left(\frac{v_1^2}{2} + \frac{P_1}{\rho_1}\right) + W_i - W_o$$

$$\left(\frac{v_2^2}{2} + \frac{P_2}{\rho_2}\right) = \left(\frac{v_1^2}{2} + \frac{P_1}{\rho_1}\right) + W_i - W_o$$

$$P_2 = \frac{\rho_2 (v_1^2 - v_2^2)}{2} + P_1 \frac{\rho_2}{\rho_1} + \rho_2 (W_i - W_o) \text{ --- } (2.11)$$

4.3.3 With this general energy equation, the parameters can be manipulated for different conditions.

Condition 1: No work done on the system, no frictional loss, $W_i = 0$, $W_o = 0$

(2.11): $P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + P_1\frac{\rho_2}{\rho_1} + \rho_2(W_i - W_o)$

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + P_1\frac{\rho_2}{\rho_1} + \rho_2(0 - 0)$$

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + P_1\frac{\rho_2}{\rho_1} \text{ --- } (3.1)$$

Condition 2: Work done on the system, no frictional loss, $W_i \neq 0$, $W_o = 0$

Work done by an electric motor

By definition we know that,

$P_w = \frac{(F \times d)}{t}$ --- (4.1) Where, P = Power F = Force d = distance t = time

$\tau = F \times r$ Where, τ = Torque r = radius

$$F = \frac{\tau}{r} \text{ --- } (4.2)$$

Distance per revolution,

$$D = 2\pi \cdot r$$

Distance per minute,

$d = D \times rpm$ Where, rpm = revolutions per minute

$$d = 2\pi \cdot r \cdot rpm \text{ --- } (4.3)$$

Substituting (4.2) and (4.3) into (4.1),

$$P_w = \frac{\tau}{r} \times 2\pi \cdot r \cdot rpm$$

Simplifying,

$$P_w = \tau \times 2\pi \cdot rpm \text{ --- } (4.4)$$

Converting revolution per minute to revolution per second,

$$P_w = \tau \times 2\pi \cdot 0.01667 rpm$$

The work done on the system,

$$w_i = \tau = \frac{9.55P_w}{rpm} \text{ --- } (4.5)$$

Therefore, substituting (4.5) into (3.1),

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + P_1\frac{\rho_2}{\rho_1} + \frac{9.55P_w}{rpm} \text{ --- } (4.6)$$

Condition 3: Work done on the system with frictional loss, $W_i \neq 0$, $W_o \neq 0$

Flow patterns of fluids in cylindrical tubes (pipe) can be represented by Reynold's number:-

$\mathrm{Re} = \frac{(\rho v d_i)}{\mu}$ where Re – Reynolds number

ρ - density

v - velocity

d_i - internal diameter of the cylindrical tube

μ - dynamic viscosity

The Reynold's number can also be determined from the correlation of mass flow rate of the fluid and the cross-sectional flow area in the pipe or tube:-

$$\mathrm{Re} = \frac{(\rho v d_i)}{\mu}$$

$\mathrm{Re} = \frac{(G d_i)}{\mu}$ where $G = \rho v$, $G = \frac{M}{A}$ and M - mass flow rate

A - cross-sectional area

As the velocity of the flowing fluid increases, the Reynold's number increases as well resulting in a change in flow pattern from laminar to turbulent. Hence, for laminar flow, $\mathrm{Re} \leq 2100$ and for turbulent flow, $\mathrm{Re} \geq 2100$. Ideally, in very smooth conduits, laminar flow can exist at a much higher Reynold's number.

However, this is not the case in real situations where shear stresses are present in pipes. Hence, it is important to consider the forces that act on a cylindrical element of the fluid:-

<u>1. Force related to fluid pressure</u>:-

From the equation, $P = \frac{F_1}{A}$

$F_1 = PA$

$F_1 = \Delta P_f(\pi \cdot r^2)$ --- (5.1) where ΔP_f - frictional pressure drop

$\pi \cdot r^2$ - cross-sectional area of a cylinder

2. Force related to shear stress:-

$$\tau = \frac{F_2}{A}$$

$$F_2 = \tau A$$

$F_2 = \tau_w 2\pi \cdot rL$ --- (5.2) where τ_w - shear stress at the wall

$2\pi \cdot rL$ - total surface area of a cylinder

Combining (5.2) and (5.1), gives a force balance equation:-

$$\Delta P_f \pi \cdot r^2 = \tau_w 2\pi \cdot rL$$

$$\Delta P_f \pi \cdot r^2 - \tau_w 2\pi \cdot rL = 0$$

Rearranging the equation enables us to determine the shear stress and frictional pressure drop for laminar flow:-

$$\tau_w = \frac{r_i}{2}\left(\frac{\Delta P_f}{L}\right)$$

$\tau_w = \frac{d_i}{4}\left(\frac{\Delta P_f}{L}\right)$ and $\Delta P_f = \frac{4\tau_w L}{d_i}$ --- (5.3)

However, in turbulent flow, pressure drop is proportional to the density of the fluid. Hence, in order to correlate the frictional pressure drop with density, the kinetic energy per unit volume of the fluid is taken into consideration.

Substituting $\frac{\rho v^2}{2}$ into (5.3),

$$\Delta P_f = \frac{4L}{d_i}\left(\frac{\tau_w}{\rho v^2/2}\right)\left(\frac{\rho v^2}{2}\right) \text{--- (5.4)}$$

Fanning friction factor,

$$f = \left(\frac{\tau_w}{\rho v^2 / 2} \right) \text{--- (5.5)}$$

Substituting (5.5) into (5.4),

$$\Delta P_f = \frac{4Lf}{d_i} \cdot \frac{\rho v^2}{2}$$

$$\Delta P_f = \frac{2Lf\rho v^2}{d_i} \text{--- (5.6)}$$

The Fanning friction factor can also be applied for laminar flow of a Newtonian fluid in a pipe using the formula:-

$$f = \frac{16}{\mathrm{Re}}$$

Besides that, the value of f varies depending on the relative roughness of the pipe wall.

Relative roughness = $\frac{k}{d_i}$ where k – absolute roughness

The table below indicates the absolute roughness values for some materials

Table 3: Absolute roughness of different materials

Material	Absolute Roughness, m
Drawn tubing	0.0000015
Commercial steel and wrought iron	0.000045
Asphalted cast iron	0.00012
Galvanized iron	0.00015
Cast iron	0.00026
Wood stave	0.00018-0.0009
Concrete	0.0003-0.003
Riveted steel	0.0009-0.009

With the known relative roughness Reynold's number, we can then use the friction factor chart for Newtonian fluids as shown in Figure 20 to determine the Fanning friction factor of a flowing fluid.

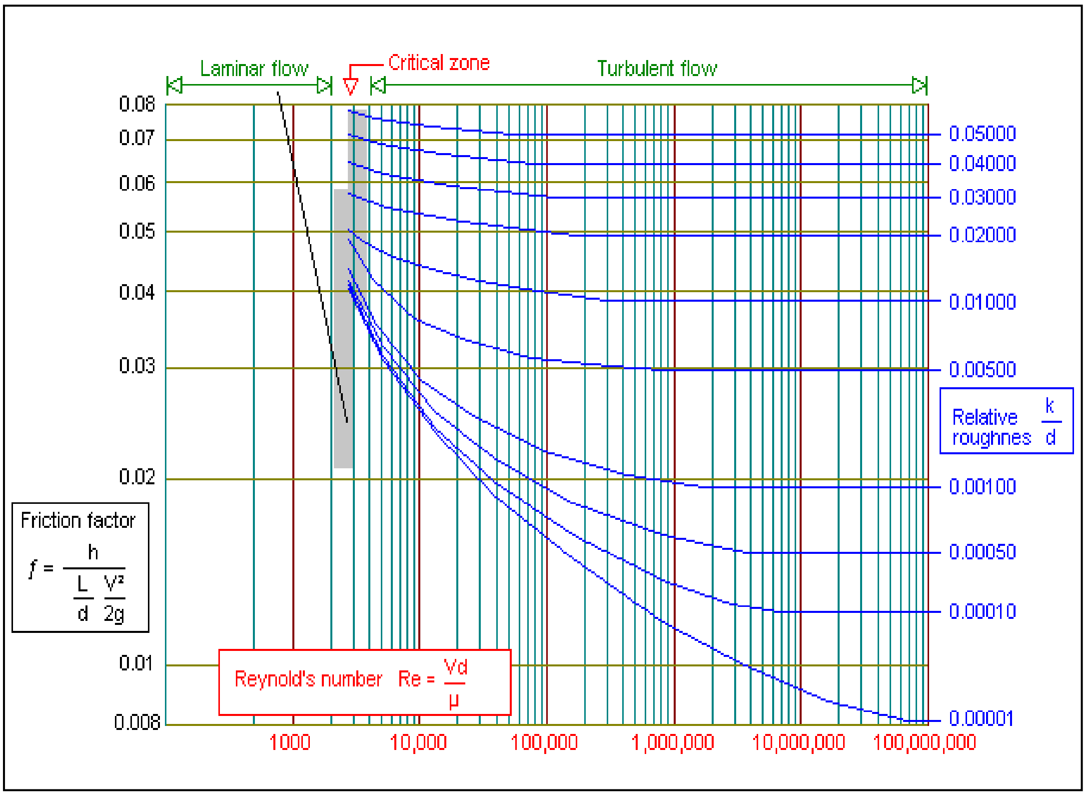

Figure 20: Friction factor chart

Therefore, substituting (5.6) into (4.6)

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}((P_1 - \Delta P_f)) + \frac{9.55P_w}{rpm}$$

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}(P_1 - \frac{2\,fL\rho\,v^2}{d_i}) + \frac{9.55P_w}{rpm} \text{ --- (5.7)}$$

Since we are only considering frictional loss in the high pressure inlet stream,

$$\rho = \rho_1$$

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}(P_1 - \frac{2\,fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm} \text{ --- (5.8)}$$

Condition 4: Mass, m $\neq 1g$

For $m_1 = m_2 = m = 1g$

$$P_2 = \frac{\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}(P_1 - \frac{2fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm}$$

For $m_1 = m_2 = m \neq 1g$

$$P_2(m) = \frac{m\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}(mP_1 - \frac{2fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm}$$

$$P_2 = \frac{[\frac{m\rho_2(v_1^2 - v_2^2)}{2} + \frac{\rho_2}{\rho_1}(mP_1 - \frac{2fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm}]}{m}$$

$$P_2(m) = \frac{\rho_2(v_1^2 - v_2^2)}{2} + P_1(\frac{\rho_2}{\rho_1}) - \frac{2fL\rho_2 v^2}{d_i m} + \frac{9.55P_w}{rpm \cdot m} \text{ --- (6.1)}$$

For $m_1 \neq m_2$,

$$P_2(m_2) = \frac{\rho_2(m_1 v_1^2 - m_2 v_2^2)}{2} + \frac{\rho_2}{\rho_1}(m_1 P_1 - \frac{2fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm}$$

$$P_2 = \frac{[\frac{\rho_2(m_1 v_1^2 - m_2 v_2^2)}{2} + \frac{\rho_2}{\rho_1}(m_1 P_1 - \frac{2fL\rho_1 v^2}{d_i}) + \frac{9.55P_w}{rpm}]}{m_2}$$

$$P_2 = \rho_2(\frac{m_1 v_1^2}{2m_2} - \frac{v_2^2}{2}) + \frac{\rho_2}{\rho_1}(\frac{m_1 P_1}{m_2} - \frac{2fL\rho_1 v^2}{d_i m_2}) + \frac{9.55P_w}{rpm \cdot m_2} \text{ --- (6.2)}$$

Let $m_1/m_2 = m_r$ --- (6.3)

Substituting (6.3) into (6.2),

$$P_2 = \rho_2(\frac{m_r v_1^2 - v_2^2}{2}) + m_r P_1(\frac{\rho_2}{\rho_1}) - \frac{2fL\rho_2 v^2}{d_i m_2} + \frac{9.55P_w}{rpm \cdot m_2} \text{ --- (6.4)}$$

Condition 5: Using seawater and brine

In the process of seawater desalination, the operating fluids used in the seawater reverse osmosis (SWRO) system are seawater and brine. These fluids are basically water with varying dissolved salt concentrations.

Hence, the density of these fluids can be determined using the formula:-

$\rho = b - (a \times T)$ where a and b - constants for different dissolved salt concentrations

T - temperature in ^{o}F

Table 4: Values of constant a and b for different dissolved salt content (Brine, 2013)

Dissolved salt weight percent (%)	**a**	**b**
5	0.043	72.60
10	0.039	73.72
15	0.035	74.86
20	0.032	76.21
25	0.030	77.85

The calculations will be based on ambient temperature which is 25^{o}C.

According to an online article (Beicher, Physics for Scientists and Engineers, 2000); the average dissolved salt content in seawater is 3-5%. Assuming the salt content is maximum (5%) at room temperature, T = 25^{o}C = 77^{o}F

$$\rho_2 = 72.6 - (0.043 \times 77)$$

$$\rho_2 = 69.29 lb / ft^3 = 1.11 g / cm^3$$

In another reference (Brine, 2013), the salt content in brine by weight percent at 0^{o}C and 100^{o}C are given as:-

0^{o}C – 26%

100^{o}C – 28%

Therefore, from table 4 we can assume the maximum weight percent of salt in brine which is 25%,

$$\rho_1 = 77.85 - (0.03 \times 77)$$

$$\rho_1 = 75.54 b / ft^3 = 1.21 g / cm^3$$

Table 5: Density of water at different dissolved salt concentrations

Type of fluid	Density (g/cm^3)
Water (0% dissolved salt)	1.00
Seawater (3-5% dissolved salt $\approx 5\%$)	1.11
Brine ($> 5\% \approx 10\%$)	1.21

$$P_2 = \rho_2 (\frac{m_r v_1^2 - v_2^2}{2}) + m_r P_1 (\frac{\rho_2}{\rho_1}) - \frac{2 fL \rho_2 v^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2} \text{ --- (7.1)}$$

Density of seawater $\rho_1 = 1.21 g/cm^3$ --- (7.2)

Substituting (7.2) into (7.1),

$$P_2 = \rho_2 (\frac{m_r v_1^2 - v_2^2}{2}) + m_r P_1 (\frac{\rho_2}{1.21}) - \frac{2 fL \rho_2 v^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2} \text{ --- (7.3)}$$

Density of brine $\rho_2 = 1.11 g/cm^3$ --- (7.4)

Substituting (7.4) into equation (7.3),

$$P_2 = 1.11 (\frac{m_r v_1^2 - v_2^2}{2}) + m_r P_1 (\frac{1.11}{1.21}) - \frac{2 fL (1.11) v^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2}$$

$$P_2 = 0.555 (m_r v_1^2 - v_2^2) + 0.917 m_r P_1 - \frac{2.22 fL v^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2} \text{ --- (7.5)}$$

Condition 6: With elevation, h $\neq 0$ ***m***

This condition applies for cases in which the reverse osmosis (RO) membrane unit is located above the pressure exchanger unit at height, h. Hence, the potential energy,

PE has to be taken into consideration when determining the total energy of the inlet stream, E_1.

$PE_1 = m_1 gh$ --- (8.1) where h = height from the RO membrane to the pressure exchanger

Substituting (8.1) into (7.5),

$$P_2 = m_1 gh + 0.555 (m_r v_1^2 - v_2^2) + 0.917 m_r P_1 - \frac{2.22 fL v^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2} \text{ --- (8.2)}$$

Condition 7: Include input and output energy by low pressure inlet and outlet streams

For all the previous conditions, only two streams were taken into account:-

a) The high pressure brine which is fed into the pressure exchanger (F_1).

b) The high pressure seawater which is discharged from the pressure exchanger (F_2).

However, there are still two other streams which enter and exit the device that need to be taken into consideration. Although these are low pressure streams, they do contribute to the overall energy balance of the system.

The streams are:-

c) The low pressure seawater stream that enters the pressure exchanger to be pressurized (F_3).

d) The low pressure brine that exits the pressure exchanger after being depressurized (F_4).

Some of the assumptions that can be drawn for these two streams are:-

1. Since the elements of works done on the system as well as by the system has been included, the terms can be discarded from equation (2.1)

$$(U_4 + m_4 g h_4 + \frac{m_4 v_4^2}{2} + P_4 V_4) = (U_3 + m_3 g h_3 + \frac{m_3 v_3^2}{2} + P_3 V_3) \text{ --- } (9.1)$$

2. Since the internal energy, U is constant, $U_3 = U_4$

$$m_4 g h_4 + \frac{m_4 v_4^2}{2} + P_4 V_4 = m_3 g h_3 + \frac{m_3 v_3^2}{2} + P_3 V_3 \text{ --- } (9.2)$$

3. $m_3 \neq m_4$

$$m_4 g h_4 + \frac{m_4 v_4^2}{2} + P_4 (\frac{m}{\rho})_4 = m_3 g h_3 + \frac{m_3 v_3^2}{2} + P_3 (\frac{m}{\rho})_3 \text{ --- } (9.3)$$

Assuming that mass is conserved in which there is no accumulation in the system,

$m_3 = m_2$ and $m_4 = m_1$

$$m_1gh_4 + \frac{m_1v_4^2}{2} + P_4\frac{m_1}{\rho_4} = m_2gh_3 + \frac{m_2v_3^2}{2} + P_3\frac{m_2}{\rho_3} \text{ --- (9.4)}$$

4. Density of fluid is based on brine and seawater

F_3 = seawater and F_4 = brine

$\rho_3 = \rho_2 = 1.11$ g/cm^3 and $\rho_4 = \rho_1 = 1.21$ g/cm^3

$$m_1gh_4 + \frac{m_1v_4^2}{2} + P_4\frac{m_1}{1.21} = m_2gh_3 + \frac{m_2v_3^2}{2} + P_3\frac{m_2}{1.11} \text{ --- (9.5)}$$

5. Inlet and outlet stream elevations, h

Since both fluids are assumed to enter and exit the pressure exchanger horizontally,

$h_3 = h_4 = 0$ m

$$\frac{m_1v_4^2}{2} + P_4\frac{m_1}{1.21} = \frac{m_2v_3^2}{2} + P_3\frac{m_2}{1.11} \text{ --- (9.6)}$$

Rearranging the equation to determine the pressure of the outlet pressure:-

$$P_4 = 0.605\left(\frac{v_3^2}{m_r} - v_4^2\right) + \frac{1.1P_3}{m_r} \text{ --- (9.7)}$$ since $m_1/m_2 = m_r$, $m_2/m_1 = 1/m_r$

Combine equations (32) and (39) to determine the overall pressure output from the system

$$P_2 + P_4 = \left[m_1gh + 0.555(m_rv_1^2 - v_2^2) + 0.917m_rP_1 - \frac{2.22\,fLv^2}{d_im_2} + \frac{9.55P_w}{rpm\cdot m_2}\right] + \left[0.605\left(\frac{v_3^2}{m_r} - v_4^2\right) + \frac{1.1P_3}{m_r}\right]$$
--- (9.8)

Condition 8: Consider mixing between the two working fluids

Since there is no physical or solid barrier between the brine and seawater in the transfer chambers, we have to consider the degree of mixing between these 2 streams which is the function of stream velocity and exposure time. The exposure time depends on the rotational speed of the rotor. For example, when the fluid flow rate increases, the plug velocity increases as well and this may cause the plug to be lost to the bulk fluid once it reaches the opposite end of the transfer chamber. In order to counter this, the rotational speed of the rotor has to be increased so that the exposure time between the end plate ports and the transfer chambers will be reduced (Stover, Development of a Fourth Generation Energy Recovery Device, 2004).

There are a number of criteria that has to be defined when addressing this mechanism:-

1. The diffusion involves movement of salt particles from the more concentrated brine to the relatively less concentrated seawater across a concentration gradient.
2. Once the rate of diffusion becomes constant, there will be a constant loss of salt content from the brine as it comes into fluid communication with the seawater while the seawater will experience a constant increase in salt concentration.
3. The diffusion between both fluids at the interface creates a boundary layer or plug of length, x. Although the value of x may differ in the beginning of the operation, it will eventually become constant once the system has achieved steady-state.
4. In one revolution, the plug moves from one end of the transfer chamber to the other end and back again depending on the direction of force exerted by the flow of high pressure fluid. In order to maintain a fixed quantity of fluid in the plug, the transfer chamber has to be long enough to retain the plug in the system at all times.

Calculating the minimum allowable length of a transfer chamber:-

The following calculations are based on plug movement in relation to inlet flow of high pressure brine. This is because the plug velocity is the maximum during this phase of the rotor's revolution. Figure 21 shows the mixing of both fluids in a transfer chamber that forms the liquid plug.

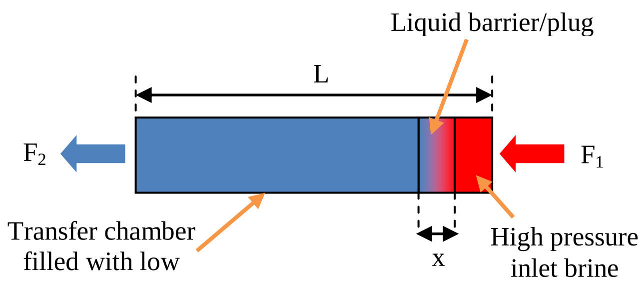

Figure 21: Position and movement of plug in a transfer chamber

Flow rate of stream, F_1, $Q_1 = v_1 A_c$ where v_1 = fluid velocity

A_c = cross-sectional area of transfer chamber

The arrangement of transfer chambers are arranged in concentric circles on the rotor's end surface as shown in Figure 22. Assuming that the transfer chambers are circular, $A_c = \pi \cdot r_c^2$

Since the plug velocity ≈ fluid velocity, $v_1 = \dfrac{Q_1}{\pi \cdot r_c^2}$ --- (10.1)

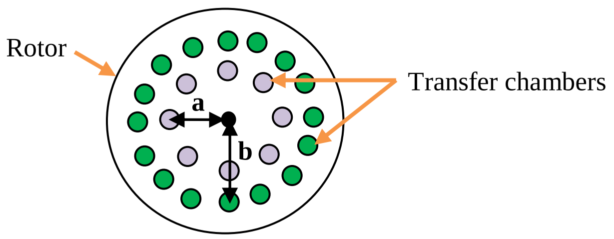

Figure 22: Concentric circular arrangement of transfer chambers on the rotor

Based on Figure 22, the radius of each concentric circle is given as r_a and r_b.

The circumferential distance travelled by the rotor in 1 revolution is given by, $d_r = 2\pi \cdot r_r$ --- (10.2)

The time taken for the rotor to complete one revolution, $t_r = \dfrac{1}{0.01667 rpm}$

$t_r = \dfrac{60}{rpm}$ --- (10.3)

Thus, the rotational speed of the rotor, substituting (10.2) and (10.3) into (10.1):-

$v_r = \dfrac{d_r}{t_r} = \dfrac{2\pi \cdot r_r}{(\dfrac{60}{rpm})} = 0.0333\pi \cdot r_r \cdot rpm$ --- (10.4)

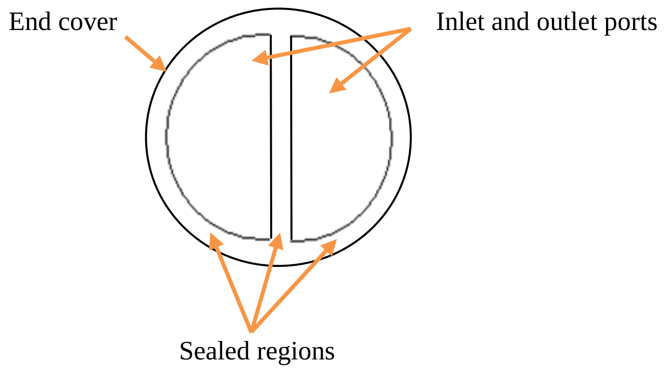

Figure 23 (a): Configuration of end plate of the pressure exchanger

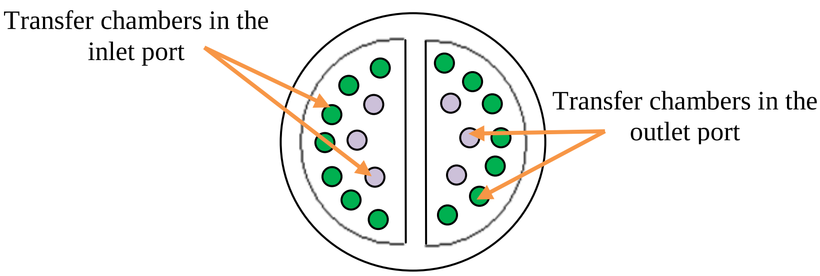

Figure 23 (b): Transfer chambers exposed to the inlet and outlet ports

With reference to Figure 23 (a) and (b), the end plate ports are semi-circular and not all the transfer chambers are exposed to the end plate ports at a particular time due to the sealed regions. Therefore, the circumferential distance travelled by the concentric circle of transfer chambers with radius r_a in the high pressure semi-circle inlet port is,

$$d_c = 2\pi \cdot r_a (\frac{1}{2})$$

$$d_c = \pi \cdot r_a \text{ --- (10.5)}$$

With the known rotational speed of the rotor, $v_r = 0.0333\pi \cdot r_r \cdot rpm$, we can determine the exposure time for each of the transfer chamber:-

$$t_{ea} = \frac{d_c}{v_r} \text{ --- (10.6)}$$

Substituting (10.4) and (10.5) into (10.6)

$$t_{ea} = \frac{\pi \cdot r_a}{0.0333\pi \cdot r_r \cdot rpm} = \frac{30 r_a}{r_r \cdot rpm} \text{ --- (10.7)}$$

Similarly, the exposure time for the concentric circle of transfer chambers with radius r_b,

$$t_{eb} = \frac{30 r_b}{r_r \cdot rpm} \text{ --- (10.8)}$$

Based on the above information, we can now determine the amount of substance being lost in the transfer chamber as the 2 fluids mix with each other. This phenomenon can be represented using Fick's Law of diffusion.

For the first section, we will consider the loss of salt from the brine into seawater.

Fick's Law (Mass Transfer Equations, 2007):-

$J = -D\frac{\partial c}{\partial x}$ where J = diffusion flux, kg/ (m^2.s)

D = diffusivity coefficient, m^2/s

∂c = change in concentration, kg/m^3

∂x = thickness of plug where diffusion occurs, m

Therefore, in the case of diffusion of salt, we can determine the velocity at which the salt molecules are moving across the plug by:-

$$v_s = J \times \frac{1}{\rho_{salt}} = \frac{kg}{m^2 s} \times \frac{1}{(kg/m^3)} = \frac{m}{s}$$

Using this velocity, we can then estimate the volumetric flow rate of salt that is being lost: - $Q_s = v_s \times A_p$ where A_p = cross sectional area of plug

Assumptions:-

1) Transfer chambers are cylindrical in shape

2) Plug radius, r_p = transfer chamber radius, r_a / r_b

$$A_{p,a} = \pi r_a^2 \qquad A_{p,b} = \pi r_b^2$$

$$Q_{s,a} = v_s A_{p,a} \qquad Q_{s,b} = v_s A_{p,b}$$

Total Q_s in all transfer chambers, $Q_{s,T} = n_a Q_{s,a} + n_b Q_{s,b}$ --- (10.9)

Similarly, there is also diffusion of water molecules from the seawater into the more concentrated brine. This process is called osmosis. Hence, the loss of water must also be taken into consideration.

$$v_w = J \times \frac{1}{\rho_{water}} = \frac{kg}{m^2 s} \times \frac{1}{(kg/m^3)} = \frac{m}{s}$$

$$Q_w = v_w \times A_p$$

$$A_{p,a} = \pi r_a^2 \qquad A_{p,b} = \pi r_b^2$$

$$Q_{w,a} = v_w A_{p,a} \qquad Q_{w,b} = v_w A_{p,b}$$

$$Q_{w,T} = n_a Q_{w,a} + n_b Q_{w,b} \text{ --- (10.10)}$$

4.4 Energy Efficiency Equation

Based on the derivations obtained from section 4.3, we can now generate an equation to calculate the energy efficiency of a pressure exchanger more accurately by modifying the present general efficiency equation (1.1):

$$\eta = \frac{(QP)_{A2} + (QP)_{B2}}{(QP)_{A1} + (QP)_{B1}} \times 100\%$$

With the known inlet values of $(QP)_{A1}$ and $(QP)_{B1}$, we can determine the flow rate and pressure values of the outlet streams A_2 and B_2

For outlet flow rates:-

$Q_{A2} = Q_{A1} - Q_{w,T}$ --- (11.1) where $Q_{w,T} = n_a Q_{w,a} + n_b Q_{w,b}$ --- (10.10)

$Q_{B2} = Q_{B1} - Q_{s,T}$ --- (11.2) where $Q_{s,T} = n_a Q_{s,a} + n_b Q_{s,b}$ --- (10.9)

For outlet pressures:-

$P_{A2} = P_2$ --- (11.3)

$$P_2 = m_1 gh + 0.555(m_r v_1^2 - v_2^2) + 0.917 m_r P_1 - \frac{2.22 fLv^2}{d_i m_2} + \frac{9.55 P_w}{rpm \cdot m_2} \text{ --- (8.2)}$$

$P_{B2} = P_4$ --- (11.4)

$$P_4 = 0.605(\frac{v_3^2}{m_r} - v_4^2) + \frac{1.1P_3}{m_r} \text{ --- (9.7)}$$

Hence, the modified energy efficiency equation can be represented as the following:-

$$\eta = \frac{[Q_{A1} - Q_{w,T}) \times P_2] + [(Q_{B1} - Q_{s,T}) \times P_4]}{(QP)_{A1} + (QP)_{B1}} \times 100\%$$

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

In conclusion, this project revolved around the theoretical research and analysis of an energy recovery device. This device is called a pressure exchanger which has a potential of minimizing the global energy crisis that the world is currently facing. The project was conducted based on an established scope of study which is to address the limitation of a pressure exchanger in terms of industrial applications. As such, the study was conducted in stages based on the relevancy of the objectives to be achieved. In the first phase of the project, a study was conducted to understand the development history, current industrial application, major components and the working principles of a pressure exchanger. Based on these, a comprehensive review was performed on existing literatures in order to classify the different types of pressure exchangers according to their designs and operating principles. The four main criteria that were used in the classification are type of fluids, mode of operation, use of separators and other additional features. In addition, the advantages of using of pressure exchanger in terms of energy and cost saving were clearly portrayed through a comparison made between a conventional SWRO system and the one with the incorporation of a pressure exchanger. The second phase of this project involved mathematical modelling to represent the dynamics of fluid flow in a basic rotary pressure exchanger. The models were developed for eight different conditions and the final equation was used to calculate the overall energy efficiency equation of the device based on its capacity to transfer pressure.

5.2 Recommendations

The suggested future work for this project will be the continuation of pressure exchanger classifications for newly developed designs. Apart from that, the criteria used in the classification for this project can be further divided into more specific parameters such as the potential industrial applicability of a particular design.

Besides that, the mathematical modelling performed for this project can be further developed to obtain a more complete representation of fluid flow in a pressure exchanger. As a result, the accuracy of energy efficiency calculation using the model will be higher. This can be achieved through the establishment of more parameters or criteria such as the inclusion of a solid piston as a separation mechanism in the transfer chambers of a pressure exchanger. This would require consideration in terms of pressure energy losses to move the piston across the chambers. Another such condition would be a pressure exchanger that operates based on self-rotation of its rotor instead of depending on an external source of power like an electric motor. This can be achieved through several modifications to the existing design of the device.

REFERENCE

[1] (2012). *2012 Key World Energy Statistics.* Paris, Frace: International Energy Agency (IEA). Retrieved April 26, 2013, from http://www.iea.org/publications/freepublications/publication/kwes.pdf

[2] *World Energy Consumption*. (2013). Retrieved May 17, 2013, from Wikipedia: http://en.wikipedia.org/wiki/World_energy_consumption

[3] (2012). *World Energy Outlook 2012*. Paris, France: International Energy Agency (IEA). Retrieved June 4, 2013, from http://www.iea.org/publications/freepublications/publication/English.pdf

[4] Homsak, P. G. (1996). Pressure Exchangers in Pinch Technology. *Computers Chemical Engineering, 20*, 711-715. Retrieved May 22, 2013

[5] Sackrison, C. (2001, July). Energy Recovery - Is it the Right Choice for your Application? *Engineering System Solution*. Retrieved May 23, 2013

[6] Cheng, C. Y., & Cheng, S. W. (1970, February 13). *Patent No. US 3489159.* United States.

[7] Garris Jr., C. A. (2009, March 3). *Patent No. US 7497666.* United States.

[8] Andrews, W. T. (2009, July 23). *Patent No. US 2009/0185917 A1*. United States.

[9] Liberman, B. (2012). *The Importance of Energy Recovery Devices in Reverse Osmosis Desalination.* IDE Technology. Retrieved May 19, 2013, from https://www.twdb.texas.gov/publications/reports/numbered_reports/doc/R36C8.pdf

[10] Seipple, C. (1946, April 30). *Patent No. US 2399394.* United States.

[11] Shumway, S. (1998, August 25). *Patent No. US 5797429.* United States.

[12] Shumway, S. (2007, May 8). *Patent No. US 7214315*. United States.

[13] Barnes, J. A., & Vickery, J. P. (1965, January 5). *Patent No. US 3164318.* United States.

[14] Hauge, L. J. (November 2009). *Pressure Exchanger Development History.* Retrieved May 23, 2013, from http://isobarix.com/yahoo_site_admin/assets/docs/Pressure_exchanger_development_ history.330155221.ppt

[15] Hauge, L. J. (1989, December 19). *Patent No. US 4887942*. United States.

[16] Hauge, L. J. (1994, August 16). *Patent No. US 5338158*. United States.

[17] Hauge, L. J. (1999, November 23). *Patent No. US 5988993*. United States.

[18] Polizos, T., Babcock, T., Hauge, L. J., & Hermanstad, R. A. (2003, April 1). *Patent No. US 6540487*. United States.

[19] Gardiner, F. J. (1964, August 25). *Patent No. US 3145909*. United States.

[20] Jendrassik, G. (1954, April 13). *Patent No. US 2675173*. United States.

[21] Williamson, D. H. (1970, October 27). *Patent No. US 3536419*. United States.

[22] Collin, R. J., & Favrin, P. A. (1959, March 10). *Patent No. US 2876704*. United States.

[23] *ERI Pressure Exchanger*. (2007). (Energy Recovery Inc. (ERI)) Retrieved February 11, 2013, from http://www.energyrecovery.com/px-pressure-exchanger-devices

[24] *Pressure Exchanger*. (2012). Retrieved February 2, 2013, from Wikipedia: http://en.wikipedia.org/wiki/Pressure_exchanger

[25] Andeen, G. B. (1987, January 20). *Patent No. US 4637783*. United States.

[26] Went, J., & Anhalt, J. N. (2013, March 28). *Patent No. WO 2013/041547 A1*. Retrieved June 10, 2013, from http://www.google.com/ patents/ WO2013041547A1?cl=en

[27] Taylor, J. S. (1974, July 23). *Patent No. US 3825122*. United States.

[28] Tonner, J. B. (1994, April 26). *Patent No. US 5306428*. United States.

[29] Permar, C. (1995, October 31). *Patent No. US 5462414*. United States.

[30] Childs, W. D., & Dabiri, A. (2000, January 25). *Patent No. US 6017200*. United States.

[31] Al-Mayahi, M., & Sharif, A. (2012, January 24). *Patent No. US 8099958 B2*. United States.

[32] MacHarg, J. P., & McClellan, S. A. (2004, November). Pressure Exchanger Helps Reduce Costs in Brackish Water RO System: Ocean Reef Brackish PX. *AWWA Journal*, 44-47. Retrieved May 12, 2013, from http://oceanpacifictec.com/imagenes/news/11%20AWWA%20Journal%20cean% 20Reef%20Brackish%20PX%2011-04.pdf

[33] Al-Hawaj, O. M. (2004, August 10). *Patent No. US 6773226 B2*. United States.

[34] Knight, A. F. (2002, December 31). *Patent No. US 6499288 B1*. United States.

[35] Mitchell, H. M. (1955, February 8). *Patent No. US 2701441*. United States.

[36] McKinley, C. (1961, November 21). *Patent No. US 3009316*. United States.

[37] Baum, W. (1972, June 27). *Patent No. US 3672165*. United States.

[38] Apfel, S. L. (1989, November 14). *Patent No. US 4880185*. United States.

[39] Beveridge, J. H., & Knuth, W. H. (1990, February 20). *Patent No. US 4901525*. United States.

[40] Pahl, D. A. (1995, December 5). *Patent No. US 5471833*. United States.

[41] Kayser, L. T. (1979, March 23). *Patent No. US 3945203*. United States. Retrieved June 4, 2013, from http:// www.archpatent.com/patents/3945203

[42] Skinner, L. A. (1950, May 2). *Patent No. US 2505798*. United States.

[43] Alyanak, E. J., & Eiszner, W. H. (1963, April 16). *Patent No. US 3085512*. United States.

[44] Spalding, D. B. (1966, February 15). *Patent No. US 3234736*. United States.

[45] Jendrassik, G. (1957, July 23). *Patent No. US 2800120*. United States.

[46] Jendrassik, G. (1961, January 17). *Patent No. US 2968435*. United States.

[47] Jendrassik, G. (1958, May 27). *Patent No. US 2836346*. United States.

[48] Hellat, J., & Keller, J. (1987, July 14). *Patent No. US 4679393 A*. United States.

[49] Nalim, M. R. (2002, October 8). *Patent No. US 6460342 B1*. United States.

[50] Grouyellec, A. L. (2001, April 10). *Patent No. US 6213348 B1*. United States.

[51] Herbst, G. W., Newell, T. A., & Korst, H. H. (August 1997). *Feasibility Study of a Pressure Exchanger in an Air-Cycle Air Conditioner*. Air Conditioning and Refrigeration Center, Mechanical and Industrial Engineering Department. University of Illinois.

[52] Rao, V. D. (1981b, May 26). *Patent No. US 4269570*. United States.

[53] Reatherford, L. V. (1998, September 1). *Patent No. US 5799641*. United States.

[54] Coleman Jr., R. R. (1958, December 16). *Patent No. US 2864237*. United States.

[55] Rao, V. D. (1981a, June 23). *Patent No. US 4272811*. United States.

[56] Vallance, J. K., & Rahnke, C. J. (1982, November 23). *Patent No. US 4360317*. United States.

[57] Berchtold, M. (1961, February 7). *Patent No. US 2970745*. United States.

[58] Aoki, H. (1986, June 14). *Patent No. US 4563997*. United States.

[59] Hashemi, H. T., & Lott, J. L. (1969, March 11). *Patent No. US 3431747*. United States.

[60] Brueckmann, F., Knoebl, W., Bruhns, U., & Kochanowski, W. (2005, August 4). *Patent No. US 2005/0166978 A1*. United States.

[61] Baumgarten, S., Bross, S., Brecht, B., Bruhns, U., Flak, S., Jaeger, C., . . . Ellegaard, M. (2006, March 16). *Patent No. US 2006/0054223*. United States.

[62] *Back Pressure*. (2013). Retrieved June 27, 2013, from Wikipedia: http://en.wikipedia.org/wiki/Back_pressure

[63] Andrews, W. T. (2012, November 13). *Patent No. US 8308444 B2*. United States.

[64] Bross, S., & Kochanowski, W. (2007, June 21). *Patent No. US 2007/0137170 A1*. United States.

[65] Hauge, L. J. (2006, February 16). *Patent No. US 2006/0032808*. United States.

[66] Hashemi, H. T. (1972, February 15). *Patent No. US 3641881*. United States.

[67] Bross, S., & Kochanowski, W. (2010, October 19). *Patent No. US 7815421 B2*. United States.

[68] Pique, G. G., Stover, R. L., Martin, J. G., & Pinto, J. M. (2011, August 16). *Patent No. US 7997853 B2*. United States.

[69] Spalding, D. B. (1963, February 19). *Patent No. US 3078034*. United States.

[70] Coleman, R. R., & Weber, H. E. (1977, January 11). *Patent No. US 4002414*. United States.

[71] El-Sayed, E. E., Abdel, J. M., & Al-Odwani, A. (2010, February 16). *Patent No. US 7661932 B2*. United States.

[72] Shumway, S. (2003, March 25). *Patent No. US 6537035 B2*. United States.

[73] MacHarg, J. P. (2010, September 21). *Patent No. US 7799221 B1*. United States.

[74] Myran, A. F., Perez-Fernandez, F. J., & Lyng, R. (2012, December 4). *Patent No. US 8323483 B2*. United States.

[75] Bross, S., & Kochanowski, W. (2005, December 8). *Patent No. WO 2005/116456 A1*. Retrieved February 12, 2013, from http://www.google.com/patents/ WO2005116456A1?cl=en

[76] Friedrichsen, W., Langmaack, L. N., Olsen, P., & Haugaard, E. (2013, February 14). *Patent No. US 2013/0037464 A1*. United States.

[77] *Cavitation*. (2013). Retrieved May June, 2013, from Wikipedia: https://en.wikipedia.org/wiki/Cavitation

[78] Hauge, L. J. (2003, December 9). *Patent No. US 6659731 B1*. United States.

[79] Stover, R. L. (2011, June 7). *Patent No. US RE42432 E.* United States.

[80] Pfenning, D. B., Higginbottom, L. C., & Sliepcevich, C. M. (1979, November 20). *Patent No. US 4174925.* United States.

[81] Kirchhofer, H., & Schelling, R. (1985, July 16). *Patent No. US 4529360.* United States.

[82] Haase, W., & Mayer, A. (1984, December 11). *Patent No. US 4487552 A.* United States.

[83] Hauge, L. J. (2012, August 9). *Patent No. WO 2012/106712 A1*. Retrieved April 4, 2013, from http://www.google.com/patents/WO2012106712A1?cl=en

[84] Lott, J. L., & Hashemi, H. T. (1971, June 1). *Patent No. US 3582090.* United States.

[85] Loeffler, R. I., & Higginbottom, L. C. (1975, October 7). *Patent No. US 3910587.* United States.

[86] Gardiner, F. J. (1961, March 20). *Patent No. US 3120919.* United States.

[87] Herger, H., & Wunsch, A. (1966, December 13). *Patent No. US 3291379.* United States.

[88] Alcock, J. F. (1954, August 31). *Patent No. US 2687843.* United States.

[89] Jendrassik, G. (1965, October 16). *Patent No. US 2766928.* United States.

[90] Honma, K., & Nakino, H. (2005, October 6). *Patent No. US 2005/0220643 A1*. United States.

[91] Wunsch, A. (1973, July 17). *Patent No. US 3746128.* United States.

[92] Mayer, A. (1989, August 22). *Patent No. US 4859153.* United States.

[93] Zauner, E. (1996, June 4). *Patent No. US 5522217.* United States.

[94] Rahnke, C. J. (1982, November 23). *Patent No. US 4360316.* United States.

[95] Kentfield, J. A. (1963, November 5). *Patent No. US 3109580.* United States.

[96] Leutwyler, P., & Wunsch, A. (1971, January 19). *Patent No. US 3556680.* United States.

[97] El-Nashar, I., Jaussi, F., Kirchoffer, H., Komauer, C., Mayer, A., Perevuznik, J., & Spinnler, F. (1987, January 10). *Patent No. US 4796595*. United States.

[98] Martin, J., & Stover, R. L. (2011, December 13). *Patent No. US 8075281 B2*. United States.

[99] Kentfield, J. A. (1965, October 5). *Patent No. US 3209986*. United States.

[100] Knight, F. (2006, August 1). *Patent No. US 7082750 B2*. United States.

[101] Sobey, A. J. (1965, October 26). *Patent No. US 3213804*. United States.

[102] Lanning, M. E., & Blackmon Jr., J. B. (2001, November 13). *Patent No. US 6314978*. United States.

[103] Moloney, H. J. (1964, December 1). *Patent No. US 3159474*. United States.

[104] Pressure Exchanger Cuts Desalination Costs on Malta. (2008, December). *Journal of Membrane Technology, 2008*(12), 9. Retrieved June 18, 2013, from http://www.sciencedirect.com/science/article/pii/S0958211808702544

[105] *Brine*. (2013). Retrieved July 15, 2013, from Wikipedia: http://en.wikipedia.org/wiki/Brine

[106] Beicher, R. J. (2000). Physics for Scientists and Engineers. Orlando, Florida, United States: Saunders College.

[107] *Mass Transfer Equations: Fick's Law*. (2007). Retrieved July 19, 2013, from The University of Arizona: http://blowers.chee.arizona.edu/cooking/mass/fickslaw.html

[108] Stover, R. L. (2004, April 2). Development of a Fourth Generation Energy Recovery Device. *Desalination , 165*, 313-321.