DEVELOPMENT OF A LOW TEMPERATURE GEOTHERMAL ORGANIC RANKINE CYCLE STANDARD

A THESIS
SUBMITTED TO THE UNIVERSITY OF CANTERBURY IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF MECHANICAL ENGINEERING

by

Leighton John Taylor

Department of Mechanical Engineering
University of Canterbury
Christchurch New Zealand

February 2015

Table of Contents

Acknowledgements

A special thanks to my senior supervisor Professor Susan Krumdieck for giving me the opportunity to follow my passion in geothermal energy. I would also like to thank my wider supervisory team, Dr. Mark Jermy, Dr. Mathieu Sellier, Dr. Hyung-Chul Jung, and the semester with Dr. Nick Baines who were all helpful during our regular Friday morning meetings.

I would also like to thank my fellow students in the office and Advanced Energy and Materials Systems Laboratory: Michael Southon, Choon Seng Wong, Sandeep Siwach, David Meyer, Denny Budisulistyo, Sungjin Choi, James Hewett, Hari Murthy, Stacy Randel, Miraz Fulh, Darryl Lee, FrithJof Engel, and Alex Stefan. I have enjoyed being part of the team and will always have fond memories of the group trips, hikes, and conferences we attended together.

Special thanks to the industry members: Kim Hardwood, Kevin Koorey, Nigel Taylor, Chris Morris, Greg Thompson, and Kevin Turner, who either helped with specific geothermal questions, arranged informative site visits, or took time to look over my standard. I would also like to specially thank HERA who provided financial funding for my fees while I studied.

Finally, I would like to thank my parents, Paul and Carol, my brothers Ashley and Ben, my sister Jordena, and my girlfriend Lisa who all kept me motivated throughout my studies and always there to support me; furthermore, I would like to specially thank my father Paul who helped me improve my standard with his expert knowledge and vast connections in the geothermal industry. Thanks also to my extended family and friends for their encouragement and support during this process.

DEVELOPMENT OF A LOW TEMPERATURE GEOTHERMAL ORGANIC RANKINE CYCLE STANDARD

Abstract

The growth in renewable electricity generation is forecast to continue as fossil fuel levels decrease and carbon dioxide emissions are penalized. The growth in geothermal is becoming constrained as conventional high-temperature sources are fully exploited. Geothermal can be a cost competitive base load power source. Governments and utilities are looking at the potential of electricity generation from low temperature geothermal resources for future development. This technology, unlike the high and medium temperature, is not mature and there are a number of companies looking at entering the Organic Rankine Cycle (ORC) market.

This thesis aims to provide a necessary step for reliable commercial develop this technology by developing the first draft of a low temperature geothermal ORC standard. The standard outlines the critical stages of a geothermal ORC project as the Prospecting stage; Pre-Feasibility stage, Feasibility stage, and the Detailed Design stage. The standard is unlike other standards that are used to design one component; this standard guides the engineers though the various critical steps of the ORC design to correctly assess the geothermal resource and to inform design and investment decisions.

The standard provides particular guidance on critical factors in ORC design, primarily the working fluid selection and component selection limitations. Experienced industry engineers have provided advice and insight regarding the critical design points and processes. The draft standard was reviewed by a number of geothermal industry engineers who have worked with large scale, conventional ORCs. They each commented on the standard from their prospective in the industry and gave general feedback was that it is a technically relevant standard that can be used as a potential start point to develop a new standard for the low temperature binary ORC industry. The final draft standard has been submitted to the ISO for consideration.

This thesis first sets out the general background on the state of the art and the industry for low-temperature binary ORC power plants, and provides the review assessment of the standard draft. However, the bulk of the thesis is the standard itself. The standard represents a substantial contribution to the mechanical and thermal systems engineering field.

Glossary of Terms

Binary Plant: A geothermal ORC is sometimes called a binary plant

Brine: Hot geothermal liquid typically with dissolved chemicals

Flash Plant: A geothermal power plant that utilizes wet steam

Kalina Cycle: A ammonia water mixture fluid that has a similar cycle to the ORC

Low Temperature Geothermal: A liquid dominated geothermal resource below 150°C

Owner: The geothermal owner owns the geothermal resource and power plant, such as Contact or Mighty River Power in New Zealand

Vendor: The vendor supplies the technology to the owner, such as Ormat

Volumetric Expander: An expander that uses volume displacement, such as a piston or scroll

Wet Steam: Steam that has a portion of brine and steam

Working Fluid: The fluid within the ORC

Zeotropic mixture: Mixture of two working fluids that never has the same vapour phase and liquid phase composition at the vapour – liquid equilibrium state.

List of Acronyms

ACC	Air Cooled Condenser
AISC	American Institute of Steel Construction
AGGAT	Above Ground Geothermal and Allied Technologies
ANSI	American National Standards Institute
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BS	British Standard
COP	Coefficient of Performance
EES	Engineering Equation Solver
EPC	Engineer Procure Construct
GNS	Geology and Nuclear Sciences
IRENA	International Renewable Energy Agency
IRR	Internal Rate of Return
ISO	International Standard Organization
LCOE	Levelized Cost of Energy
LMTD	Log Mean Temperature Difference
NCG	Non Condensable gas
NPSH	Net Pressure Suction Head
NPSHA	Net Pressure Suction Head Available
NPSHR	Net Pressure Suction Head Required
NPV	Net Present Value
NZGW	New Zealand Geothermal Workshop
ORC	Organic Rankine Cycle
PPT	Pinch Point Temperature
PR	Pressure Ratio
TEMA	Tubular Exchanger Manufactures Association
T-S	Temperature – Entropy
WCC	Water Cooled Condenser
WF	Working Fluid

Key Variables

These are the main variables in the ORC standard. Each equation used in the standard will define the variables as required by the ISO format.

Variables	Description	Units
$\boldsymbol{Q}$	A measure of the available heat in the fluid	kW
$\dot{\boldsymbol{m}}$	Mass flow rate	Kg/s
$\boldsymbol{C_p}$	Specific heat of a fluid	kJ/kgK
$\boldsymbol{\Delta T}$	Temperature difference either between two fluids or of one fluid across a process	K or °C
$\boldsymbol{h}$	Specific enthalpy of a system – Can be the inlet or outlet or state enthalpy depending on the subscript	kJ/kg
$\boldsymbol{h_{fg}}$	Latent heat of vaporization	kJ/kg
$\boldsymbol{x}$	Fluid Quality	-
$\boldsymbol{W}$	Potential work	kW
$\boldsymbol{T}$	Temperature	K or °C
$\boldsymbol{\eta}$	Efficiency	-
$\boldsymbol{f_d}$	Darcy Friction Factor	-
$\boldsymbol{L}$	Length	m
$\boldsymbol{D}$	Hydraulic or normal Diameter (will be specified)	m
$\boldsymbol{V}$	Average Velocity	m/s
$\boldsymbol{g}$	Acceleration due to gravity	(9.81) m/s^2
$\boldsymbol{PB}$	Payback Period	Years
$\boldsymbol{CC}$	Capital Cost	$
$\boldsymbol{P_r}$	Sale Price of Electricity	$/kWh
$\boldsymbol{C}$	Capacity Factor or Cost	- or $
$\boldsymbol{\Delta P}$	Change in Pressure	kPa
$\boldsymbol{S}$	Entropy	kJ/kgK
$\boldsymbol{N_s}$	Specific Speed	-

-Continued on next page-

Variables	Description	Units
$\boldsymbol{\omega}$	Rotational Speed	Rad/s
$\boldsymbol{\rho}$	Density	m^3/kg
$\boldsymbol{U}$	Overall Heat Transfer Coefficient	W/m^2K
$\boldsymbol{A}$	Area	m^2
$\boldsymbol{I_f}$	Installation factor	-
$\boldsymbol{t}$	Time of cash flow	Years
$\boldsymbol{i}$	Discount rate	-
$\boldsymbol{\sigma}$	Plant Reduction Cost	%
$\boldsymbol{N}$	Rotational Speed of a turbine	RPM
$\boldsymbol{R_{Foul}}$	Fouling Resistance	m^2K/W

List of Figures

List of Tables

Introduction

I. Introduction

Renewable energy has gained interest over the past decade with the installed capacity for renewable energy, excluding hydro, increasing from 2.2% to 9.7% of the global production [2]. In this time there was a 550% increase in annual investment into renewable energy. Geothermal annual growth capacity also increased by 4% in 2013, which is greater than the average 3%[2]. This growth with conventional geothermal is not sustainable and both industry and governments are looking at ways of maintaining this growth by improving plant performance, cycle efficiency, and the use of low temperature fluids.

Typical renewable energy develops begin with the easiest opportunities to maximise the profits and minimise project risks[3]. As the technology matures and expands there are less obvious developments. To continue growth in renewable energy field developers look at more difficult resources previously considered too risky or unfeasible.

This is specifically accurate for the geothermal industry. The early geothermal projects utilized known resources with high enthalpy vapour dominated resources to operate a steam power plant. As these high temperature resources became less common developers began to explore lower enthalpy resources, which required a different technology approach to capitalize on the available heat. Initially the flash separator geothermal technology was developed to utilize wet steam. As the geothermal field matured there were less high enthalpy opportunities were available; therefore, the Organic Rankine Cycle (ORC) or Binary technology was developed to utilize lower enthalpy liquid dominated resources.

II. Motivation

The Above Ground Geothermal and Allied Technologies (AGGAT) initiative sponsored by the New Zealand Ministry of Business, Innovation and Employment (MBIE) and industry members was setup to face the issue that New Zealand, a leader in expertise in the global geothermal industry, does not engineer or manufacture their own geothermal ORCs. The large ORCs used in New Zealand's geothermal sector have all been procured from Ormat, an overseas supplier. The AGGAT initiative aimed to explore the potential to manufacture ORCs in New Zealand by focus initially on low temperature geothermal resources which is a developing technology.

The AGGAT team understood the importance of a design tool to assist an engineer through the stages of ORC design. The first attempt of a design tool was completed by a final year project team that highlighted the 4 main stages in an ORC development as the prospecting, feasibility, detailed design, and construction stages.

The motivation of this thesis was to develop the first draft of a standard for Low Temperature Geothermal ORC design that can be adopted by industry and used for designing and checking design ORC developments in New Zealand. The standard can also be used by a company wanting to enter the ORC market as the standard will supply them with the process to follow for a successful development.

III. Geothermal Background

a. Origins

The majority of the heat in the earth comes from the decay of radioactive elements in rocks; this heat is continuously conducted to the surface with a typical temperature gradient of 15-30°C/km[4]. Areas where there are interactions between tectonic plates, such as the Taupo Volcanic Zone where two plates have collided and one plate has sub ducted under the other, can cause volcanic activity that increases the heat flux to the surface. These volcanic zones are more likely to contain a geothermal resource, which is defined as 'volumes of rock where heat is stored as either liquid or vapour within the pores and fractures in the rock'[5]. The amount of heat in a geothermal resource depends on the type and porosity of the rock in the reservoir.

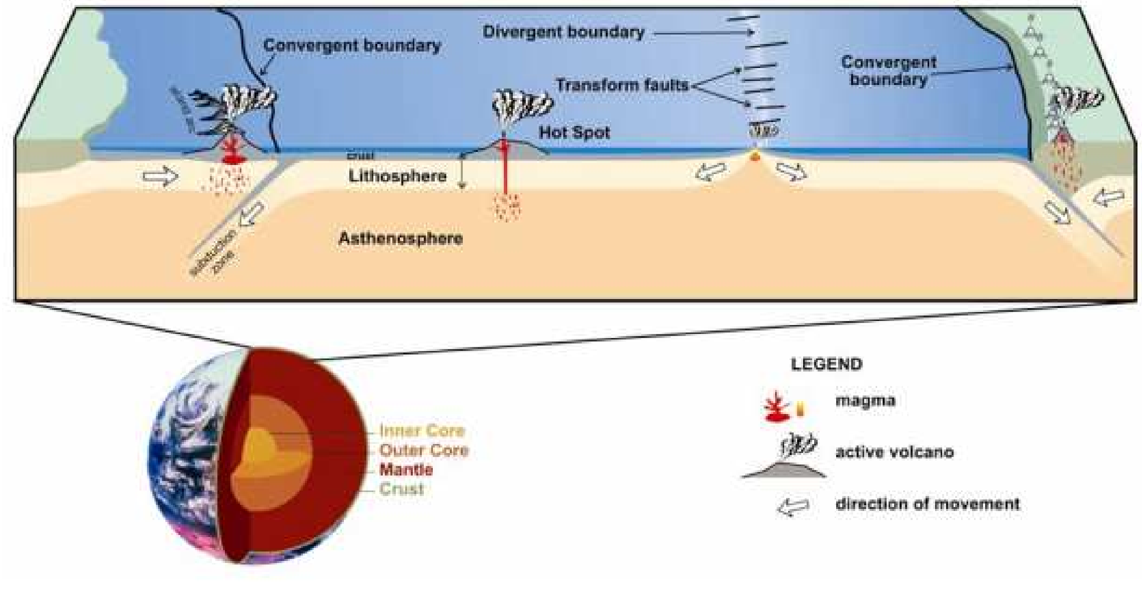


Figure 1 – Layers of the Earth and typical tectonic settings for geothermal systems, Credit GNS [4]

b. Usage

Geothermal fluids have been a part of New Zealand's history with uses in direct heat and electrical generation. In the early years geothermal hot springs were used for heating, cooking, preserving food, and for their preserved health benefits [6]. The increased demand for thermal baths by the first settlers led to the first of shallow wells being drilled in Rotorua.

Rotorua has used geothermal heating for many years to heat homes, businesses and institutions. The dry years in the 1950s reduced the national hydro power reserves resulting in electrical shortages and more of

Rotorua's residents turning to geothermal heat for district heating [7]. The impact of this was evident in the 1970s when surface manifestations in Rotorua started to disappear from the excessive use of geothermal fluids. To prevent further loss of the geothermal surface manifestations, which are significant to both New Zealand culture and tourism, regulations were put in place to manage the geothermal use and protect the geothermal features.

The 1950s also saw the Tasman pulp and paper mill built in Kawerau to take advantage of the geothermal resource for thermal processes in the plant. This project still remains the largest single use of geothermal direct use in the world, using some 8 PJ annually[6]. Other significant direct uses for geothermal heat in New Zealand are the Miraka milk drying plant and green house that use fluid from the Mokai geothermal field. The Huka Prawn farm is an example of cascade geothermal use. It uses discharged brine from the heat exchangers of the Wairakei ORC. The prawn farm has been a very successful operation and is capable of harvesting 400 tons of Giant Malaysian Fresh-Water Prawns per year which sell for 17-27USD/kg [8].

The total amount of direct heat usages in New Zealand is estimated to be around 10 PJ per annum[6]. The New Zealand Energy Efficiency and Conservation Strategy wants to increase this by at least 20% by 2025 [9].

Geothermal fluid has been used to generate power since 1913 in Larderello Italy. The original power plant in Larderello was a dry steam system that used vapour from a vapour dominated geothermal resource in a traditional steam turbine to generate electricity [7]. The power plant was destroyed during WW2 and rebuilt shortly after. New Zealand engineers went to Larderello to visit the 140MW geothermal power plant; this coincided with dry years in New Zealand and the national hydro plants struggled to meet the demand. The New Zealand government was looking for alternative energy sources and exploratory drilling started at Wairakei in 1949 to develop New Zealand's first geothermal power plant. Wairakei was a favourable choice because of the low exploration resource risk. Geothermal steam discovered at Wairakei when the hotel drilled just 170m down. The Waikato River would also provide the required cooling fluid for the power station. The power station at Wairakei was built between 1958 and 1963 and because of the wet fluid used New Zealand engineers developed the first separators for geothermal fluids to separate brine and geothermal steam. The Wairakei power station was the first geothermal power station in the world to utilize wet geothermal steam [10].

Despite the success with the Wairakei power station geothermal development was stagnant after the discovery of the Maui gas field[7]. In the 1980s renewed interest in geothermal power resulted in the Ohaki and Kawerau stations being built. Furthermore, the deregulation of electricity supply and generation in 1993 resulted in rapid growth in geothermal power sector between 1995 and 2000. Currently there are 14 geothermal power stations in New Zealand with an installed capacity of about 750MW, which is 13% of New Zealand's electricity supply[11]. Nine of these plants either use an ORC as the main method of power generation or as a bottom cycle on the power plant. There is a further 770 MW of potential geothermal development which could fulfil a third of New Zealand's energy demand[11].

Despite the potential geothermal expansion there are a number of barriers to geothermal development. Environmental considerations can significantly impact further developments. Geothermal tourism also contribute up to 310 million New Zealand dollars annual and the risk large geothermal develops pose to these geothermal manifestations can potential outweigh the benefits[11]. The Waikato Regional Council administrating the RMA has a classification system for potential resources that are likely to damage geothermal manifestations. Low temperature geothermal develops are less likely to cause the same problem as the large high temperature resource.

c. Power Conversion Options

i. Flash Plant

The flash plant technology is commonly used for wet geothermal systems where there is a risk that liquid droplets would damage the turbine[12] or not enough steam in the system. A wet geothermal plant uses the fluid from the geothermal well that has decreases from reservoir pressure to well head pressure to stimulate flow and generate more geothermal steam for the power plant. This two phase mixture used in a separator to separate the geothermal steam from the brine; figure 2 illustrates the basic flash plant components. This process can be repeated a number of times to separate more steam at lower pressures. Nga Awa Parua uses a triple flash separation process, flashing the brine to lower pressures, to have three steam streams: high pressure, intermediate pressure, and low pressure steam. The streams are then used in the appropriate stages in the steam turbine. All the low pressure steam at the back end of the turbine is then discharged directly into a direct contact condenser to condense the steam at the outlet pressure. The condensate is then pumped to the cooling tower which uses evaporative cooling to cool the condensate which is then used in the condenser once more. Make up water for the condenser can come from ground water or the unused brine stream. The brine not used in the plant is then re-injected back into the geothermal reservoir or used for a downstream operation either an ORC or direct heat project.

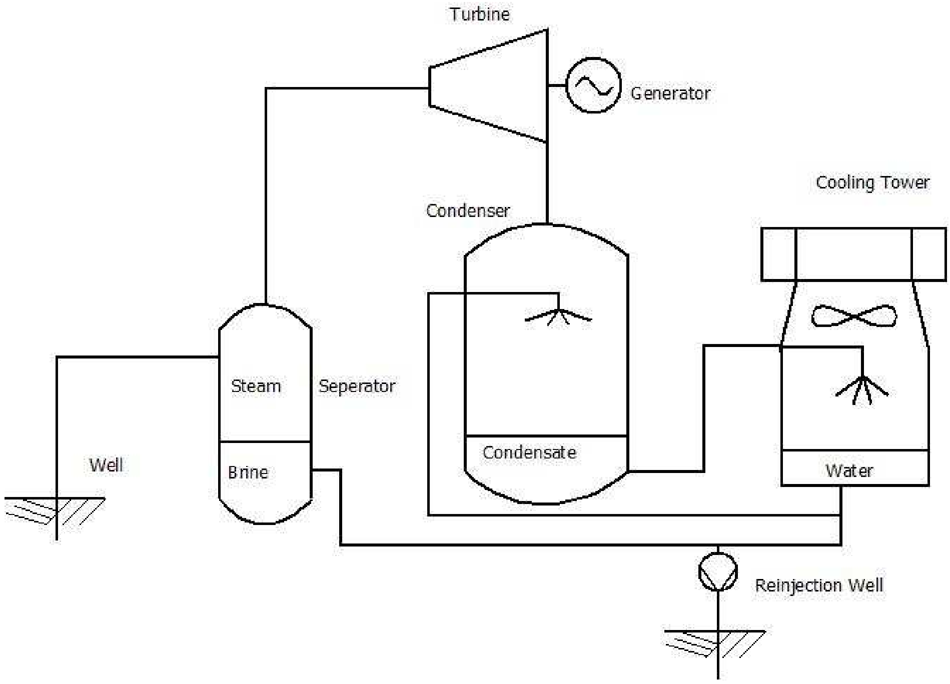


Figure 2 - Basic Flash Geothermal Power Plant

ii. ORC

A geothermal ORC can use steam, brine, or both in the heat exchangers to heat a secondary working fluid to be used in the Rankine cycle. The basic ORC is designed for a geothermal system that has a liquid dominated resource or a geothermal system that requires 100% fluid reinjection. The brine either flows freely or is pumped from the well to the heat exchanger in the ORC. The heat exchangers for the brine working fluid heat transfer are the evaporator and pre-heaters, shown in figure 3. The brine first enters evaporator at its hottest state; the working fluid at this point is at the saturation temperature following the heat transfer in the preheater. The heat transferred here boils the working fluid to a saturated vapour. The fluid typically then goes through a super heater to superheat the vapour this ensures no liquid droplets will enter the turbine. The brine is then used in the pre-heater to preheat the working fluid form a cool high pressure liquid to near saturation.

The working fluid vapour from the evaporator is then expanded in a turbine and condensed in either shell and tube or air cooled condenser. Air cooled condensers are commonly used when there is no reliable source of cooling water. The recuperator is also used in number of ORCs where there is a high limitation on the geothermal reinjection temperature. The addition of a recuperator can reduce the amount of pre-heating required by the brine and improve the efficiency of the cycle [13]. The later chapters in the standard will go over the ORC technology in much further detail.

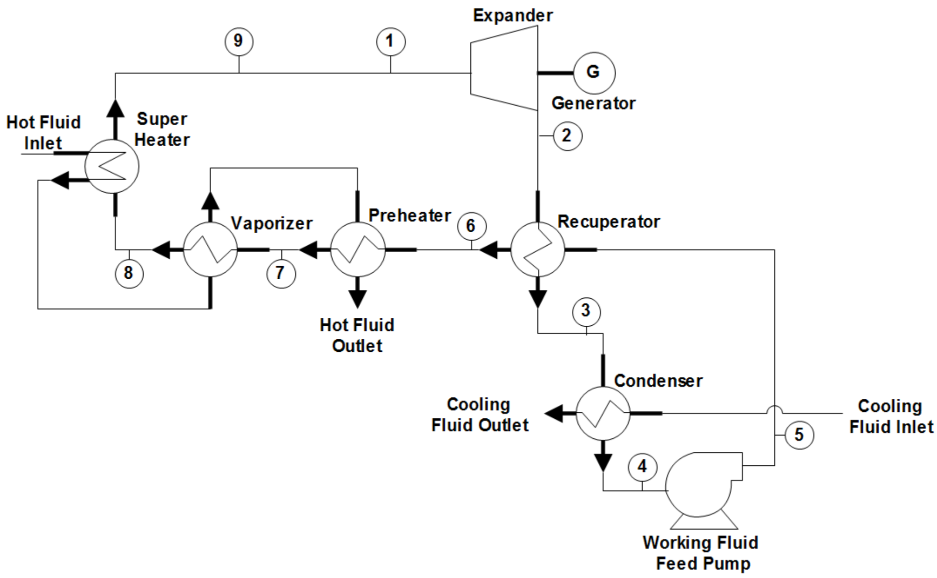


Figure 3 - Process Flow diagram of the typical Recuperative ORC

iii. Kalina

The Kalina cycle is another potential system used for waste heat and geothermal heat utilization. However, the Kalina cycle patents are still held by Wasabi Energy and the second generation Kalina cycle developed by Dr. Kalina is held by his company Kalex technology[13].

The Kalina cycle uses a mixture of ammonia and water in a modified Rankine cycle that can be designed to operate with a low or high temperature resource. The mixture of ammonia and water allows the phase change processes to happen over a range of temperatures. Ammonia and water has been used for refrigeration for a number of years and is well known that the concentration of ammonia changes the impact of the phase change temperature glide and latent heat of the mixture. The typical concentration of ammonia and water for power generation is 75%[13].

The defining features of the Kalina cycle are the number of internal heat exchangers and separators used in the system. Figure 4 illustrates the basic Kalina cycle. The Kalina cycle receives the ammonia water mixture at stage 3 at a near saturated state. The evaporator uses geothermal brine or waste heat to partially boil the mixture. The vapour boiling from the saturated mixture is mostly ammonia. The two phase fluid is then separated at stage 4. The ammonia rich vapour then passes to the turbine to generate the power from the system. The unused lean mixture goes to the high temperature regenerator where it heats the liquid mixture before the evaporator. The liquid is then expanded through a throttle and mixed with the discharged vapour form the turbine. The fluid mixture is now at the average concentration state again and is sent to the low temperature regenerator to recover the heat from the hot vapour. The fluid mixture is then condensed in the condenser back to a liquid state. The ammonia in the liquid mixture changes throughout the condensation process resulting in the temperature change during condensation. The fluid at stage 11 is then pumped through the circulation pump and passes through both regenerators to recover the unused heat in the system before entering the evaporator once more.

One advantage of the Kalina cycle is the ability to use a traditional steam turbine opposed to designing a turbine specifically for the ammonia water mixture. The temperature change during phase in the heat exchangers higher thermal efficiency compared to other cycles; however, it is also a challenge to determine the exact pinch point of the heat exchange process. A cycle using a pure fluid such as IsoPentane will have the pinch point at the dew point of the fluid.

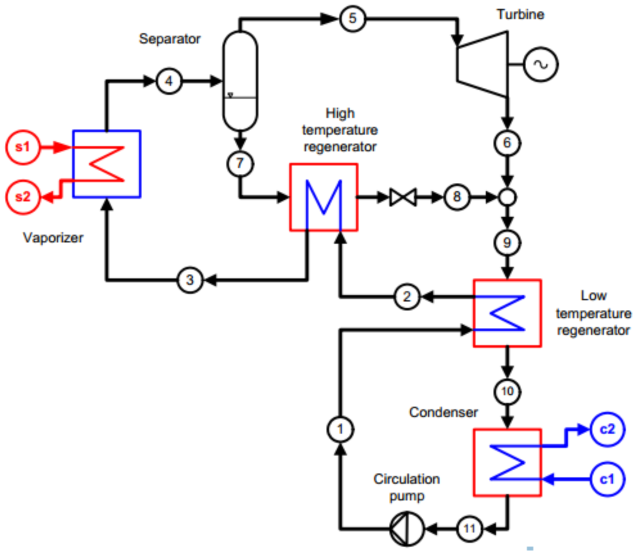


Figure 4 - Basic Kalina Cycle Components - Credit Valdimarsson [13]

IV. Literature Review

The ORC has been studied extensively by researches and their potential for affordable low temperature heat recovery. The main ORC research areas are the working fluid selection, cycle configuration, optimization strategies, component consideration, case study analysis, and the design and build of ORCs. This literature review was carried out to highlight the most important aspects of the ORC design to assist with developing the standard.

a. Geothermal Resource

There is a significant risk associated with exploring a geothermal reservoir. These risks should not be associated with the power plant development but instead taken on by the exploration and drill phase of the geothermal development.

Every geothermal project requires a detailed understanding of the reservoir. The geochemistry can be determined from initial test drilling and this study can determine the minimum allowable temperature of the geothermal fluid as well as the material compatibility, which is important for turnkey contract negotiations. Geothermal fluids have a silica deposition risk which can cause pressures losses in geothermal pipework and reduced performance in the heat exchangers. Low temperature geothermal fluids have less likely be have of an issue with silica [12]; however, other depositions can significantly impact the plants performance. Stibnite can form in low temperature geothermal fluids and impact heat exchanger performance and maintenance schedules[14]. This can be predicated and avoided with a robust geochemical analysis.

The amount of fluid available for a power plant is not confirmed until the production wells are drilled and tested. This puts increased pressure on the resource risk as the cost of drilling is a large portion of the overall cost and if the wells do not flow this can bankrupt the owner and stop project. A detailed reservoir analysis can help minimise this risk and determine the best location for production wells; however, there is still uncertainty that the production well will be suitable before a well flow test.

A geothermal resource will also change throughout the life of the power plant. It is necessary to maintain a good reservoir understanding throughout the power plants life and prepare drilling phases for make up production wells to maintain the temperature and flow rates for the power plant.

b. Geothermal ORC Overview

Recent studies have shown that the ORC market is growing; however, most of this growth is in the MW range Quoilin [15]. There have been a number of case studies recently exploring industrial ORC project and potential improvements. Jalilinasrabady [16] compared different geothermal conversion systems with the ORC system performing the best. The condensing steam turbine can achieve better power output; however, the parasitic loads can also increase greatly with condenser changes. Dipippo [17, 18] has investigated a number of different operational ORCs and their performance. These studies have shown that alternative cycles to those used at the plant might achieve a better performance; however, this is regardless of a cost analysis which is most likely the deciding factor. Dipippo [17] also suggests the small ORC market is developing with recent improvements in 'off the shelf technology' making small geothermal plants more economically competitive with cheap natural gas prices.

Dipippo has researched geothermal binary plants in operation around the world and suggest that a better measurement of their efficiency should be used [19, 20]. The 1st law efficiency is poor for ORCs, which can lead to misleading interpretation of performance, and the 2nd law efficiency or triangular efficiency is better representation of a geothermal ORCs utilization. He also covers a simple analysis for a preliminary silica deposition analysis of the different types of geothermal operations and concluded that silica is less of an issue in ORCs; however, it is still highly recommended that a detailed geochemical analysis be done to determine the reinjection temperature. Dipippo and Kestin [21] have also developed the source book for electricity form geothermal. This covers the design approach of the heat exchangers, turbine, and pipework of an ORC and was used to guide some sections of the standard.

Agahi [22] is another researcher who is attempting to improve the ORC market. He published a paper regarding Atlas Copco's new variable inlet guide vane turbine that can potential be more flexible with a changing geothermal resource. As technology improvements mature the low temperature ORC market should become more economical for an owner.

c. Geothermal Costs

Conventional high temperature geothermal technology is a mature technology and provides a competitive base load generation source. However, it does depend greatly on the type of geothermal project and the levelzied cost of energy (LCOE) from geothermal can vary widely. A green field, a geothermal field without any development, is typical less economic with a LCOE of around 0.14 USD/kWh. Meanwhile, a second stage development on a well-known resource can achieve an LCOE around 0.04 USD/kWh making it a very economically competitive base load solution when compared to fossil fuel power plants[23].

Figure 5 shows how the costs in a large geothermal development are distributed. A significant cost of the development is the power plant cost. This figure is for a flash geothermal plant and this distribution will change depending on the geothermal field and technology used.

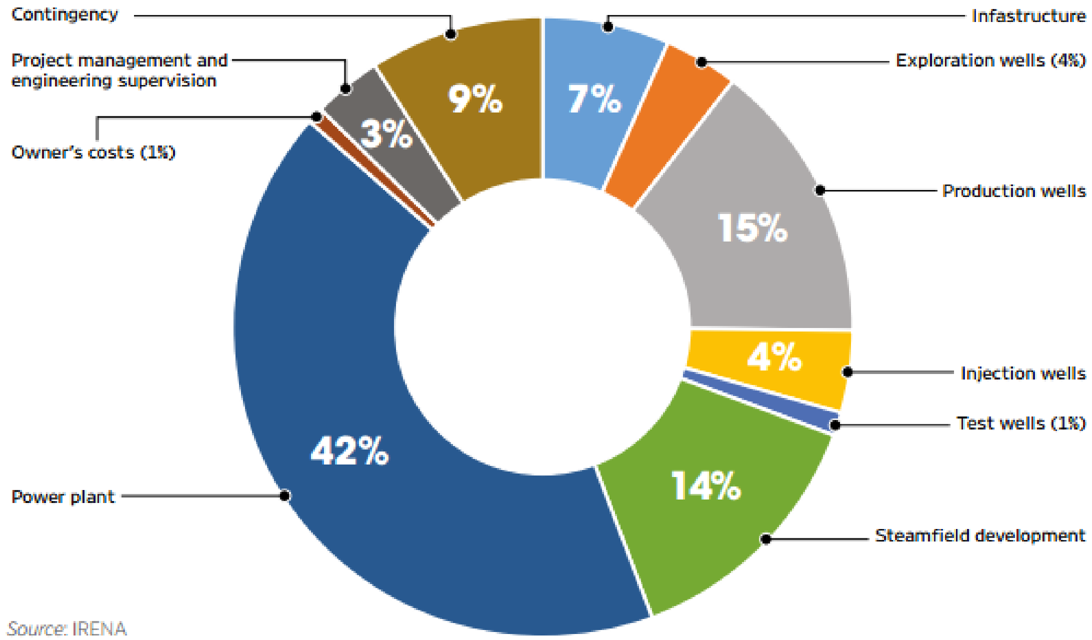


Figure 5 - Geothermal development cost – Credit IRENA [23]

Geothermal costs have been increasing with the EPC cost increasing as a result of rising commodity prices. ORC geothermal power plants are generally more expensive than a conventional steam geothermal plant [24]. Figure 6 shows the price of the two types of technology over the years and figure 5 shows that power plant cost is a significant portion of the cost of the development and because binary plants are more expensive it is critical that their design is economical optimized.

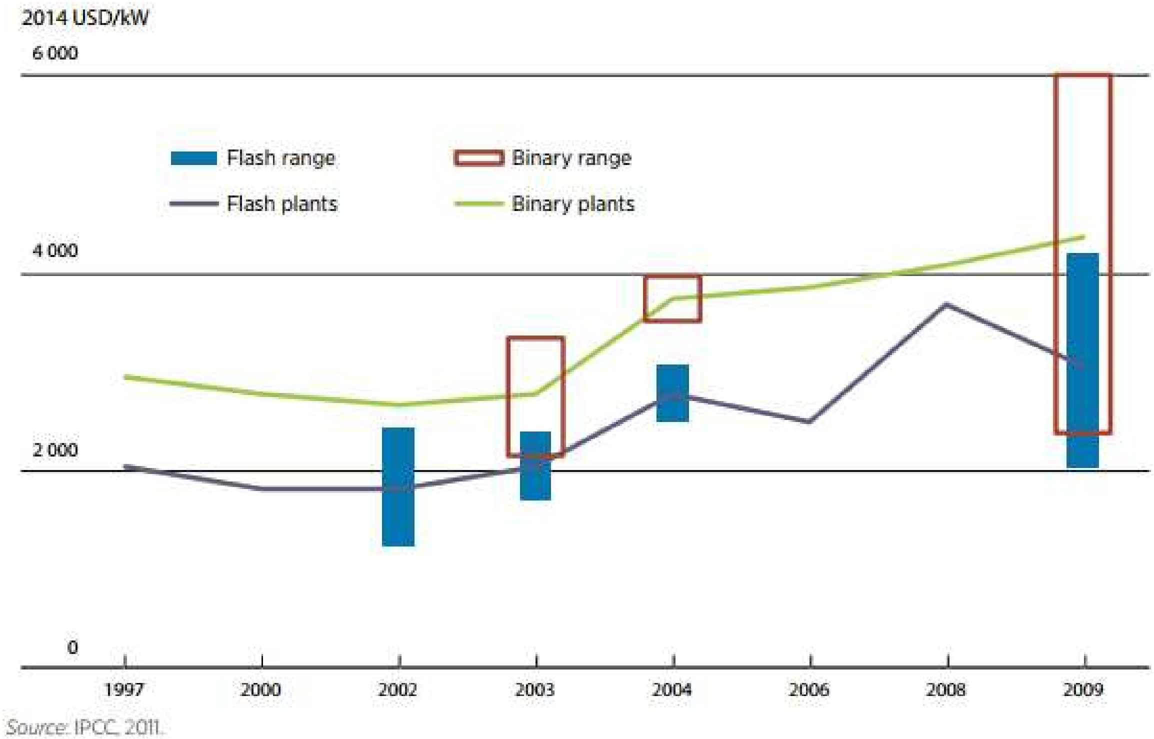


Figure 6 - Geothermal Power plant costs - Credit IRENA [23]

The geothermal market is also exploring new areas with developers looking at ORC plants for low temperature applications. These initial low temperature developments are costing between 5000 and 10000 USD/kW [23].

d. Working fluid

The choice for working fluid in an ORC is the most important step for an ORC as it will determine the cycle points and the component sizes[25]. Papers investigating the impact working fluids can be categorized into two types, the pure fluids and the zeotrpoic mixtures.

The majority of the papers investigating working fluid selection use thermodynamic modelling software such as, Engineering Equation Solver (EES) or RefProp to explore how different fluids behave in their ORC. The papers considered in this literature review only explore waste heat recovery or geothermal ORCs because the working fluid selection must match a given heat source opposed to a biomass ORC which uses a fuel source.

Quoilin [26], Chen [27], Maizza [28], Zhai [29], Papadopoulos [30], Astolfi [31], and Saleh [32]have all explored the properties of working fluids that make them favourable for ORCs. These papers highlight that isentropic and dry fluids are more favourable as they require small amount of superheat. High vapour density and reasonable evaporator pressure also reduces the component size and cost. There is some disparity between Chen [27] and Zhai [29] on whether high or low latent heat is more beneficial. Chen suggests that high latent heat is more critical as this leads to more heat absorbed in the evaporator; however, Zhai [29] suggests low latent heat is better for geothermal applications as it is more appropriate for heat source utilization. Saleh [32]and Astolfi [31] say the critical temperature of the fluid should be considered in relation to the heat source. Astolfi [31] uses the critical temperature and heat source temperature ratio as a preliminary design tool to highlight likely fluids the optimum being between 0.8-0.9. All papers suggest that the working fluid should also be non-ozone depleting and lowest global warming potential to avoid potential phase out in the future. Quoilin [26]and Maizza [28] also suggest that the working fluid should have a positive gauge pressure during condensation to reduce the risk of air leaking into the system and cost of large air extraction components.

The method used to judge which working fluid is best for the ORC was different for a number of researchers. Wang [33] considered the working fluid with the highest thermal efficiency for a cycle using a screw expander concluding that R245fa was the best when the cycle and fluid safety are considered. Wang also concluded that condenser design was more critical to cycle performance than the evaporator temperature and should be carefully considered for the design of the cycle. Shengjun [34], Roy [35], Zhai [29], Liu [36], however, suggest that only considering the maximum thermal efficiency can be misleading and result in inappropriate fluid selection for the cycle. Shengjun [34]and Stijepovic [37] put more emphasis on the economic implications of the fluids because maximizing thermal efficiency can increase the cost. Roy [35] and Quoilin [26] both investigate the fluid implication on the expander; Roy [35]suggested that the vapour quality at the exit of the expander is a critical factor. Quoilin [26] understood the importance of the expander consideration for working fluid selection as this approach can lead to ruling out unrealistic fluid early in the design process.

A large number of papers investigating working fluids for ORCs also consider zeotropic Fluids. A Zeotropic mixture is a working fluid mixture that can achieve small temperature changes during phase change. Angelino [38] considered the benefits zeotrpic mixture fluid could have on an ORC but could not make confident ORC designs due to the lack of robust thermodynamic data. Recently Wang [39] , J.L Wang [40], Chys [41], and Heberle [42] have all explored zeoptropic mixtures in ORCs. The models all agree that a zeotropic fluid can achieve higher thermal efficiencies compared to pure fluids. Wang [39] suggests that a zeotropic mixtures can make more wet fluids with good thermodynamic properties suitable for ORCs. Heberle [42] identified that a mixture can better match a resource and decrease the amount of exergy lost in the heat exchangers, which can be a better solution for geothermal ORCs. Chys [41] suggests that a zeotropic mixture will have a greater impact on the low temperature utilization of ORCs. The only actual experimental results were from J.L Wang [40] and this rig used a throttle to simulate the expander on an ORC. The results supported the fact that as zeotropic mixture can achieve higher thermal efficient.

The number of research papers that focus on the working fluid selection show that it is a critical design decision for an ORC.

e. Cycle Design

The cycle design is another aspect of the ORC design process that can significantly impact outcome. There are a number of papers that focus on cycle design propose new methods to optimize an ORC for a given heat source. Frick [43], Quoilin [44], Madhawa [45], Astolfi [31, 46] each propose different methods to optimise the ORC. Quoilin [44] and Madhawa [45] both use objective functions to achieve the optimum economic ORC. Frick [43]suggests that a holistic approach must be used to optimise an ORC as this can improve the revenue and capital investment. Astolfi [31, 46] goes through the optimization process from both a thermodynamic an economic analysis and the difference the two approaches can achieve.

Papers by Shengjun [34], Astolfi [31], Schuster [47], and Gabbrielli [48] have each modelled supercritical cycles and the potential with geothermal fluids. The results indicate that supercritical cycles with geothermal fluids can achieve better resource matching and utilization; however, supercritical ORCs have not been widely used in industry and their economic implications are less well known. Shengjun [34] suggests that a supercritical cycle can be more economical for a low temperature resource; however, the high pressures in the system can be a drawback for the operation and component cost. Astolfi [31] indicates the potential for a supercritical system to utilize a geothermal resource and the issues with the size of heat exchanger required may reduce its economic feasibility.

Papers exploring cycle optimization also comment on the impact of cycle parameters on the outcome of the design. Roy [35], Dai [49], Wei [50], Franco [51], and Frick [43] highlight some of the most controlling parameters when modelling an ORC . The amount of superheat on a subcritical ORC is explored by Roy [35]and Dai [49]. These researchers both concluded that superheat is necessary for the system to avoid any droplets during the expansion process; however, increasing the amount of superheat reduces the system performance for both isentropic and dry fluids which are the preferred fluids for an ORC. Wei [50] and Franco [51] both look at impact of condensing conditions in an ORC. The condensing temperature has a greater impact on the ORC performance compared to the evaporation temperature. Wei [50] suggested that avoiding excessive sub cooling will improve system performance; however, it is also important to have a small amount of sub cooling to safely operate the pump; the suggested sub cooling amount was 0.5-0.6K. Franco [51] recommends that the design condenser temperature is one of the critical aspects for a successful ORC design.

Cycle design is important in the ORC industry as well and Chris Taylor [52] discussed the benefit to an ORC development that can develop rapid conceptual cycle designs that highlight the main points of the ORC.

f. Components

A significant challenge with ORCs is the component size and selection. This issue is not as challenging for large ORC systems as there are providers for expanders and heat exchangers; however, for a small system the expander market is not developed and component costs can quickly increase the capital cost of the system.

Research on small ORC components is focused on the expander and the prospect of using a volumetric expander, either a scroll or screw expander, or a small turbo expander. Quoilin [15, 26, 53] has done an extensive investigation into volumetric expander usage in ORCs to reduce the overall cost of the system making them more feasible for the low power range. There are limitations associated with volumetric expanders that Quoilin has highlighted and that an ORC predicated to produce more than 1MW will use a turbine instead [26]. The working fluid also impacts the type of expander selected. Volumetric expanders converted from mass produced compressor technology can achieve isentropic efficiencies as high as 70% and as low as 30%. This technology can help make the kilowatt sized ORC more affordable with potential specific cost of 2136 Euros per kilowatt [44]; this would be a big advantage as other small ORCs cost between 5000 and 10000 USD/kW. The green machine [54] is an example of a commercial ORC that has adopted the use of volumetric expander technology. Zhou [55] also explored the potential for a scroll expander in a ORC utilizing a waste heat resource. The maximum efficiency achieved was 57% and the evaporator pressure had the biggest impact on its performance.

The other option for a small ORC is a turbine and researches have done detailed models and have tested the potential for a small turbine in an ORC. Macchi [56], Yamamoto [57] , Pei [58], Kang [59], and Quoilin [26] have all explored the benefits of a small turbine for an ORC system. Quoilin [26] highlighted the working fluid, resource temperature, and power output necessary to utilize a small turbine. Macchi [56] has done work on the preliminary investigation of axial turbines in ORCs and the impact that high pressure ratio lowers the efficiency of a turbine. Macchi's [56] work provides good tools for the preliminary investigation of a turbine in an ORC. Pei [58] and Kang [59] have designed, built, and tested their own turbines in an ORC; both of these turbines required high rotational speed to achieve the maximum efficiency of 65% and 78% respectively and because of lower than expected efficiencies it was critical to avoid other unnecessary losses in the cycle, such as pressure loss in the condenser and cavitation in the feed pump. Yamamoto [57] also tested a turbine and discovered that the optimum working condition for an ORC turbine is with the minimum amount of super heat which is expected from a cycle analysis. Yamaoto also discovered that the optimum speed of the turbine was dependent on the inlet temperature of the fluid .

A big component decision for an ORC is the addition of a recuperator. A recuperator is only necessary for an ORC if there are strict limitations on the reinjection fluid temperature [13]. An ORC that has no restriction on the reinjection temperature cannot improve the power output with the addition of a recuperator.

V. Standard

a. Steps in the Standard

The standard was proposed to help New Zealand industry fast track ORC projects. A publication from ENEX [60] supported the approach taken in this standard. The geothermal handbook [3] also outlines the same major steps for a geothermal project; therefore, the prospecting, pre-feasibility, feasibility, and detailed design stages were used as the main chapters for the standard and this thesis.

Very few case studies are published that cover all the steps of a geothermal project that can be used to verify the standard. The best published geothermal case study is the Chena ORC in Alaska [61-63]. This project further validated the correct approach for the standard as they went through the similar stages in their project.

The most critical component of a commercial geothermal project appeared to be the feasibility study. The feasibility study is where a number of the main cycle design decisions are made. This is typically intellectual property for the companies involved and not published in open literature. The only report that discusses the critical information for a successful ORC feasibility study are from two Electric Power Research Institute (EPRI) reports[64, 65] that discuss the decisions made during an early geothermal ORC project at Heber. The project was one of the early attempts at a large ORC power plant and the main reason it failed was that the engineers underestimated the resource available.

b. Purpose

There are no other standards similar to what the ORC standard is trying to achieve. Typically a standard is set of rules to design a single piece of equipment. This ORC standard gives the tools method for an engineer to make decisions on the potential of power generation from a given geothermal resource.

This standard is the first draft of the ORC standard that will be submitted to the International Standard Organization (ISO). The ISO will review the standard and determine its global relevance before establishing a technical committee of experts to continue its development. This standard was developed from available literature and input from some members of the geothermal industry. Typically standards are produced by a technical committee than have worked in the field for a number of years.

c. Audience

This standard was designed to be adopted by a wide audience. Each section of the standards increases in complexity and technical detail. Each involved party in a geothermal project should find a chapter or chapters that are useful to them.

The prospecting stage is designed for an owner of a geothermal resource with limited technical knowledge and background in geothermal and ORCs. The prospecting stage has a number of simple graphs and calculations to determine how viable the resource is and whether an ORC is the only option.

The pre-feasibility stage is the initial ORC model and cost analysis. This is thought to be used by an engineer with some thermodynamic experience or an owner looking to do some of the engineering designs themselves if they have the capability. The pre-feasibility stage is also a good starting point for an engineer wanting to model an ORC and has limited experience with thermo cycles. This stage will also provide a detailed cost estimate for the components required for the ORC. The outcome of the pre-feasibility stage is an understanding of the potential basic ORC cycle that could be used on the geothermal resource.

The feasibility study is for an engineer who has thermodynamic experience and wants to determine the most efficiency ORC technology for the available resource. The feasibility stage looks into more the complex cycle analysis and component design. The feasibility study also covers a significant the risks that should be considered before making an investment decision on the project. The outcome of the feasibility study should determine the best cycle for the geothermal resource which is either the most economical or thermodynamic efficient solution. The feasibility study should provide the owner with the knowledge to make the final investment decision on the ORC development.

The detailed design section is the most technical section of the standard. Here the reader is expected to know each aspect of the ORC thermodynamic design and can use this section to do the detailed design of the main components of the ORC. The outcome of this section is sufficient component information to begin manufacture and construction of the system

d. Method

The initial approach proposed to develop this standard was to do a thematic analysis of geothermal projects. This would identify the critical information in each step of the development and the main points to cover in the standard. However, because a significant amount of the development process in a geothermal power plant is intellectual property of the owner and the contracting engineers involved finding suitable reports was difficult. The best projects that documented a significant amount of their development were the Chena[62] and Heber projects[64, 65]. These reports helped identify the critical design issues with ORC.

There are several research publications looking into ORC developments. These helped develop the key stages in the standard. The main research areas highlighted the important design steps for an ORC and what needed to be included in the standard. This required understanding of detailed multivariable thermo modelling in Engineer Equation Solver to correctly model a suitable ORC design. Component consideration required understanding into heat exchanger design practices and limitations. The expander is a very technical piece of equipment and a course offered by Nick Baines on turbine design helped with understanding the preliminary turbine design.

The initial stages proposed by the early attempted of a standard were changed slightly match the stages in a typical geothermal development [3]. Communication with members of industry confirmed the important outcomes of each of the stages and supplied a sample of some of the outcomes.

Communication with industry also re-emphasised the important design decision in a successful ORC and the main points to cover in the standard [25].

There is no other standard available that is used as a guide to develop a whole energy system. This meant there was no similar standard to follow and the typical structure of the standard was changed slightly to better suit the purpose of this standard. Opposed to detailing the limits and requires of standard equipment this standard aimed to help the reader though all the steps required to confidently design and ORC.

A conference paper at the New Zealand Geothermal Workshop (NZGW) in 2013 went over the basics of the standard and what its relevance. The feedback from the presentation and discussion with industry members improved the path of the standard and its industrial relevance.

At the NZGW in 2014 a draft copy of the standard was displayed to a number of industry members with several taking a copy for review. Once again the feedback on the goals of the standard was positive with several industry members interested with its prospects.

The following chapters are each a section from the standard in the ISO standard format.

Prospecting

Introduction

This standard will assist with technology selection for a given geothermal resource. The resource assessment is outside the scope of this standard; however, optimal low temperature limits are considered at times.

The prospecting stage is for owners of a geothermal resource that have a limited background and technical skills needed to analyse a geothermal resource. This section consists of simple formulas, graphs, and tables that help understand the potential for electricity generation in a geothermal resource.

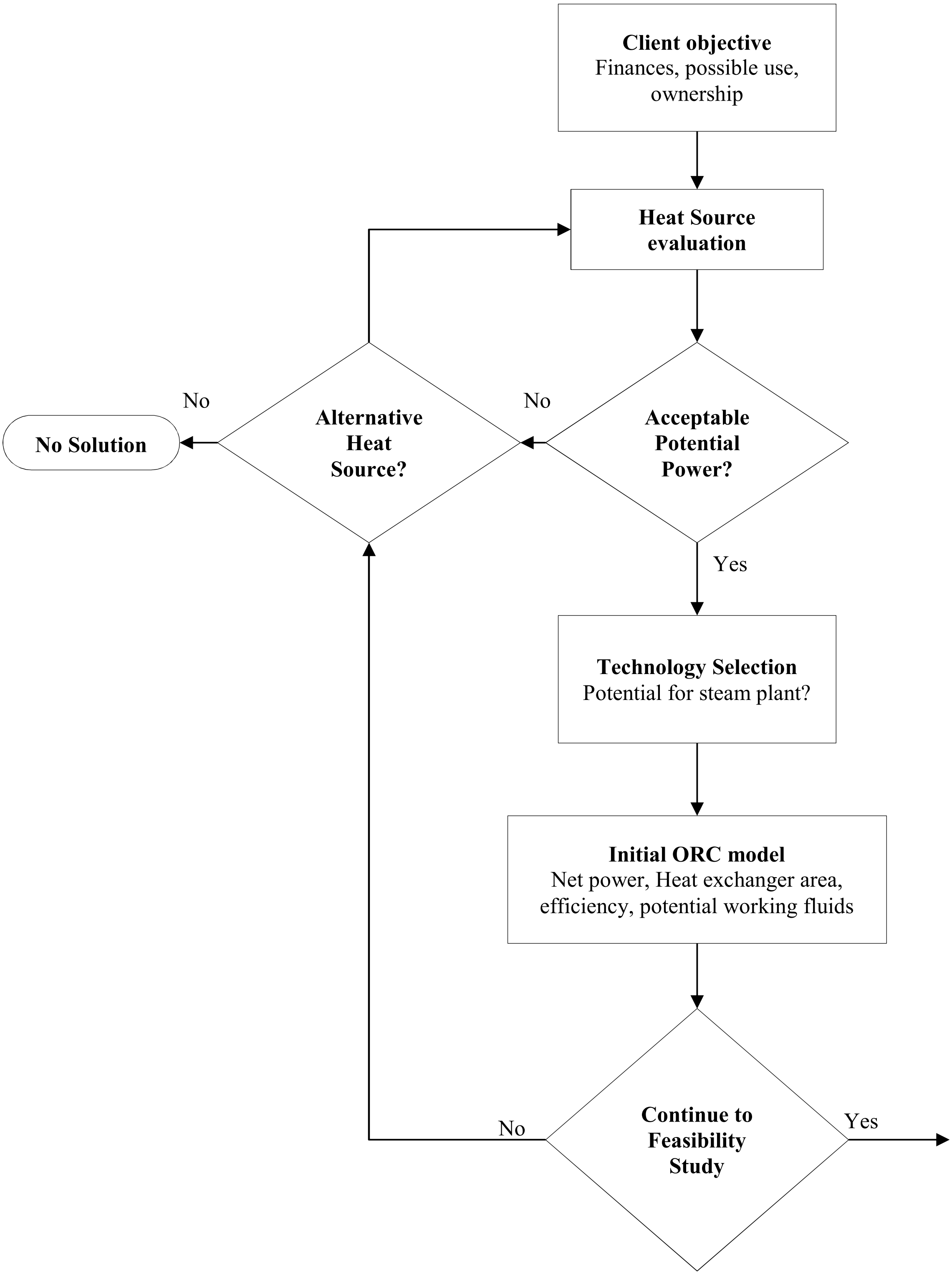
The outcome of this stage is a basic cost estimate and potential power output. There are also a number of graphs to give the owner an idea of the working fluid and component feasibility this for discussing the options with technology vendors.

Section Overview

The flow chart on the flowing page is an overview of the process of the prospecting section.

1. Client Objective - The start point of the project determines the objective of the owner. It is important to communicate to the owner that a geothermal resource can be used for direct heat applications or power generation.
2. Heat Source - The heat source analysis determines the availed heat in the geothermal fluid and gives the owner a better understanding of potential uses.
    a. There is a decision loop at this stage in the standard. This standard is for ORC development and without sufficient heat an ORC development is no feasible. However, the owner can use multiple resources.
3. Technology Selection – A flash plant and ORC are both considered. The low temperature fluid will favour the ORC.
4. ORC model: The ORC size output and cost are estimated from basic plots and tables. This section also looks into the details of the ORC components to help the owner understand the sizes of components when approaching a technology vendor. The owner then must make a decision whether or not to continue with the ORC design process.

Prospecting Process



1 Prospecting

1.1 Scope

This section focuses on the possible uses and technology selection. The ORC is the best option if power generation is possible; however, if the fluid is hot enough a flash plant is better.

An ORC system is unique to each resource this requires knowledge of thermodynamics cycles to best match an ORC to the available resource. Inappropriate sizing will impact the performance of the ORC and life of the project [3].

A Data sheet is provided at the end of this chapter that should be filled out with the key information from each section.

1.2 General

The prospecting stage evaluates the client's needs and the potential resource. The prospecting stage is a quick analysis that does not require an in-depth understanding of ORCs and thermodynamics. The outcome this chapter will be a rough estimate of power, capital cost, simple payback, and possible working fluids.

1.3 Client Objective

Communication with the resource owner will determine their plans. The following sections are some of the necessary topics covered.

1.3.1 Site

The physical location of the resource will restrict the resource usages[64]. Furthermore, the location will determine the condenser used. Remote locations will pay more in transmissions fees to connect the system to the grid and off grid power is a better use.

1.3.2 Finances

An ORC is a capital cost intensive investment which puts extra financial risk on the owner. A low temperature geothermal system may not require a production well to be drilled; however, in the case a production well is drilled geothermal drill is expensive and a larger power plant is needed to overcome the costs [3].

1.3.3 Use

Low temperature geothermal resources can be used for a number of applications. This depends on the temperature and amount of the resource; therefore, power generation is not always the best use. A resource below 80°C is too low for an ORC and direct use is a better option.

1.3.3.1 Direct Use

Direct use can be the best application for the geothermal resource. Heat from a geothermal resource has been used for a number of direct heating processes in the past. The Lindal diagram in figure 7 illustrates the potential direct uses for a geothermal. The location of the geothermal well must be close to the direct use potential.

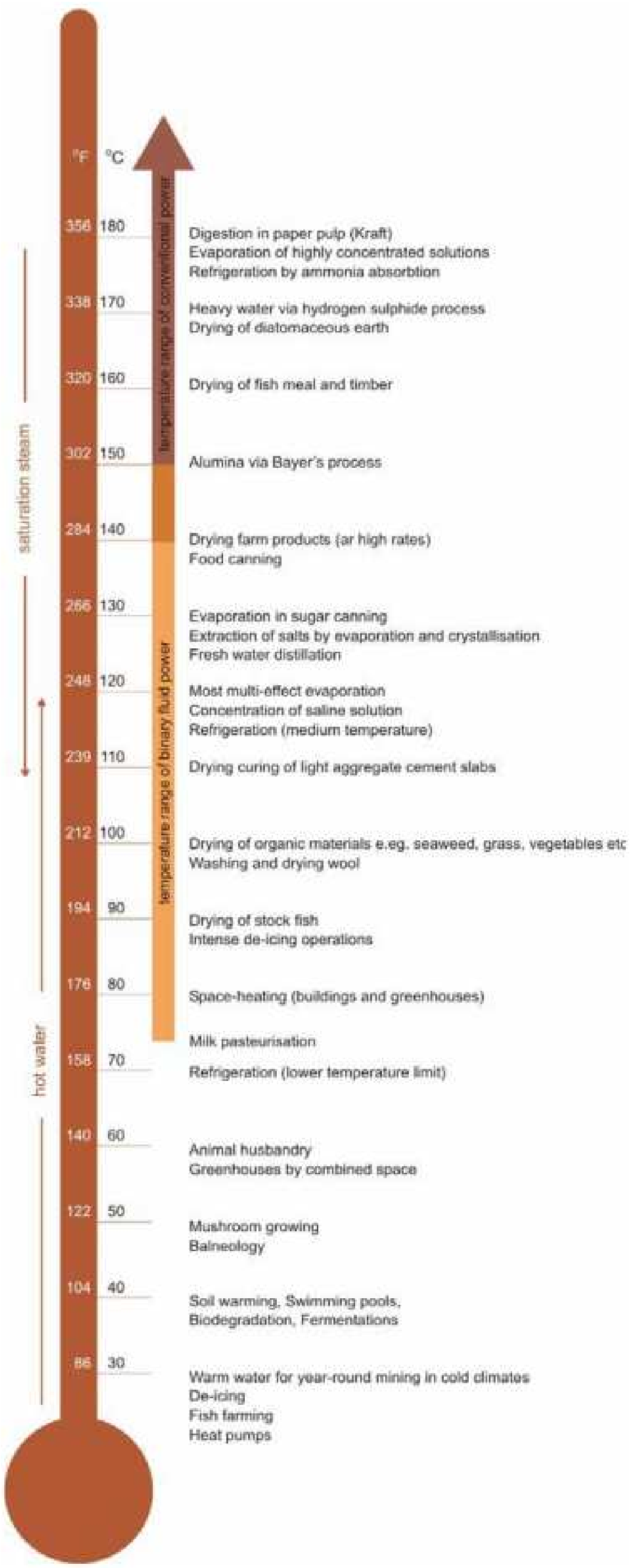


Figure 7 - Lindal Diagram for Geothermal uses – Credit GNS [5]

1.3.3.2 Power Generation

Geothermal fluid can be used in either an ORC or a traditional steam power plant; however, as shown in the Lindal diagram the low temperature range ORCs the common choice. An ORC can either produce power with a generator or be directly coupled to other shaft work[66]. Figure 8 shows the temperature and heat range where an ORC is possible.

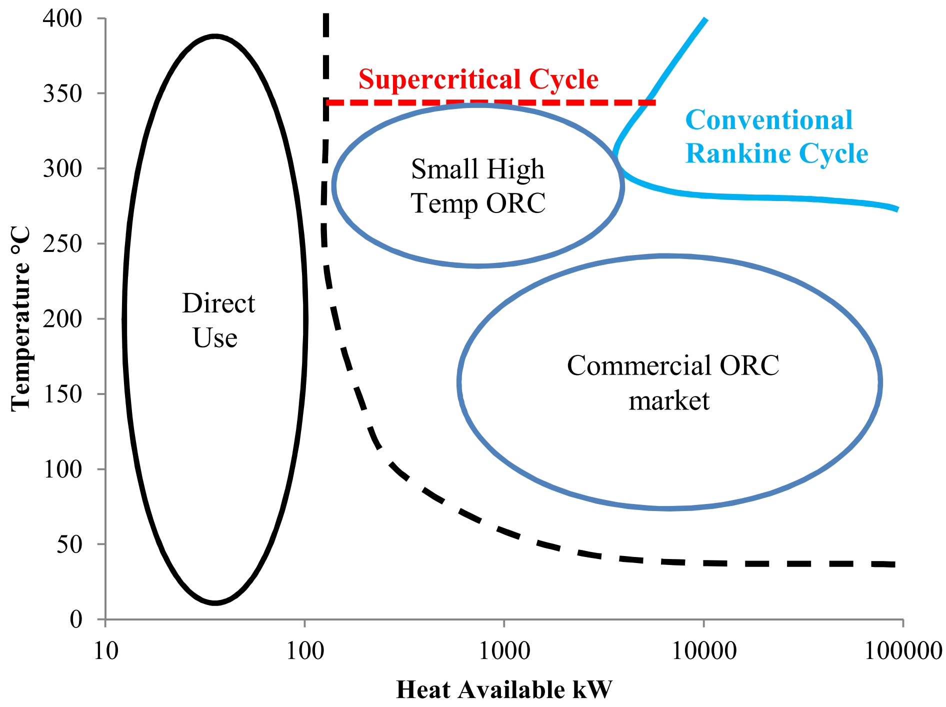


Figure 8 – Direct heat or power map – Credit Gaia [1]

1.4 Evaluation of Heat source and cooling source

Understanding the geothermal resource is critical for a robust analysis of the potential power[67]. Once the resource limitations are known there is less risk of excessive fouling in the ORC.

1.4.1 Geothermal potential

Low temperature geothermal resources are typically water dominated with a temperatures between 80-150°C[4]. It is possible to use cooler geothermal fluid; however, this will require cool fluid temperature below 5°C year round.

The minimum allowable outlet temperature is critical as it will determine the available heat in the geothermal resource. Some factors than can limit the minimum allowable temperature are: downstream uses, resource analysis, or the geochemistry of the fluid. A geochemical analysis of the fluid is critical as it will determine a safe outlet temperature and material selection[67].

1.4.2 Available Heat

The heat in the system is simply the change in enthalpy between the inlet and outlet states. The following sections provide the equations for both a single and two phase system.

1.4.2.1 Liquid Resource

The available heat in a single phase liquid resource is simply calculated with the following equation.

$$Q = \dot{m}C_p(\Delta T)$$

Q Is the available heat in the flowing geothermal liquid kW

$\dot{m}$ Is the mass flow rate of the geothermal resource kg/s

C_p Is the specific heat of water 4.2 kJ/kgK

ΔT is the temperature difference of the geothermal fluid, measured at the inlet and outlet temperatures (K)

1.4.2.2 Two Phase Resource

A two phase resource is possible in this temperature range. A two phase system is mostly liquid with some fraction vapour.

Steam tables are used to look up the enthalpy properties of the fluid which is determined by the saturation temperature and the quality of the fluid.

$$Q = \dot{m}(h_{in} - h_{out})$$

Q Is the available heat in the flowing geothermal liquid kW

h_{in} Is the specific enthalpy of the fluid before use kJ/kg

h_{out}Is the specific enthalpy of the fluid after use kJ/kg

The fluid is a two phase mixture so the dryness faction needs to be taken into account.

$$h_{tp} = h_f + xh_{fg}$$

h_{tp}Enthalpy of a two phase fluid kJ/kg

h_f Enthalpy of saturated liquid kJ/kg

h_{fg} Enthalpy of evaporation kJ/kg

x The dryness faction of a two phase mixture

1.4.3 Initial Power Estimate

A rough power estimate can be made using the available heat and an estimate of thermal efficiency. This a preliminary rough estimate and will change with more investigation. A common thermal efficiency assumption for an ORC is 10% (0.1)[45]. Figure 16 can provide a better assumption of the thermal efficiency relative to the geothermal fluid temperature. If the available power is less than 50kW there are limited commercial options.

$$W = Q(\eta_{th})$$

W Is the potential work output from the heat source kW

Q Is the available heat in the heat source kW

η_{th} Is the estimated thermal efficiency of the ORC

1.4.4 Carnot Efficiency

The Carnot cycle is a theoretical thermodynamic cycle that operates at the maximum theoretical thermal efficiency between the heat source and sink. This efficiency is theoretical but gives the engineer an understanding of the limitations of the cycle.

$$\eta_{carnot} = 1 - \frac{T_C}{T_H}$$

η_{carnot} is the Carnot efficiency of heat source

T_C Is the sink temperature of the cold source K

T_C Is source temperature of the hot source K

1.4.5 Initial Cost estimate

The initial power estimate and the geothermal fluid temperature is used to estimate the rough cost of the power plant. Table 1 consists of the potential specific costs of an ORC; this price estimate is the cost of the power plant equipment and does not include the geothermal collection system. If the available flow rate of the power plant is unknown Table 1 is used to estimate this.

Table 1 - Specific Plant Costs – Credit Elliot [68]

Power Plant Capital Cost and Brine Flow Requirements

Power Plant Net Size kW W	Inlet Temperature T_H					
	100°C		120°C		140°C≤	
	\$/kW	$\dot{m}$ kg/s	\$/kW	$\dot{m}$ kg/s	\$/kW	$\dot{m}$ kg/s
≤100	2535	13.9	2210	6.7	2015	3.9
200	2340	27.8	2040	13.1	1860	7.5
500	2145	63.1	1870	30.8	1705	18.1
1000≤	1950	126.7	1700	63.1	1550	36.1

1.4.6 Acceptable Power

It is the owner's decision whether the power output is sufficient to continue the investigation. . A plant producing less than 1MW is more challenging and less economic as there is a limited market for small ORC components[69]. A ORC producing more than 250kW will still be feasible; however anything below this becomes more of a challenge and the lower limit should be 50kW. Alternatively, an ORC for remote power generation is a niche market and is more feasible [62].

1.4.6.1 Commercial ORC options

Commercial ORCS range from the low kW range to the multiple MW size. If the owners plan on using a commercial ORC they must look to vendors in the market and the owner's checklist is provided at the end of this standard.

1.4.7 WF impact

The working fluid will impact the design of the cycle and components in the ORC. The working fluid safety is critical decision. Either use a hydrocarbon that has a low global warming potential but is flammable or a refrigerant that is non-flammable but has a high global warming potential.

1.5 Other Technology Options

The ORC may not be the best solution for a geothermal resource and other technology that could be suitable is a flash plant, a dry steam power plant, or for very small power productions thermoelectric material. The Kalina cycle is not considered in this section.

1.5.1 Steam Power plant

A geothermal steam plant is mature technology for power generation. The steam cycle uses the steam from the geothermal resource in a conventional steam turbine. Figure 9 is a schematic of a flash steam power plant. A low temperature geothermal resource is unlikely to use this process because it requires excessively large pipework that will increase the cost of the power plant[52].

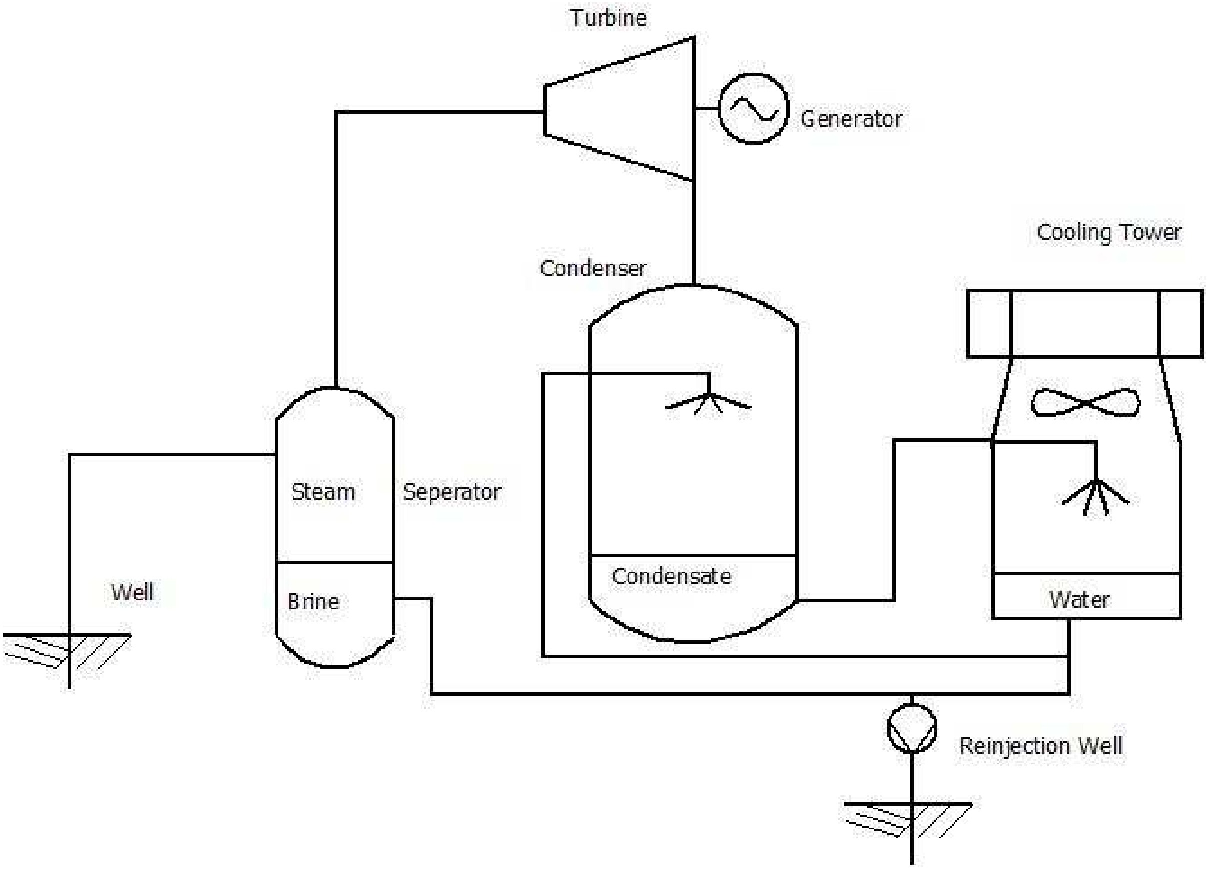


Figure 9 - Basic Flash Steam power Plant

1.5.1.1 Dry Steam power plant

The dry steam geothermal power plant is the simplest method of generating power from geothermal steam[12]; however, it is also very unusual to have enough dry steam in a geothermal reservoir. It does not require equipment to flash geothermal liquid to vapour. The dry steam from the well can be used directly in the steam turbine.

1.5.1.1.1 Dry steam power plant power estimate

A source of dry geothermal steam is rare; however, the first geothermal power plant used dry steam. The power available is determine from the enthalpy drop across the turbine.

$$W = \dot{m}(h_{in} - h_{out})$$

W is the power output of the turbine kW

$\dot{m}$ is the geothermal steam mass flow rate kg/s

h_{in} is the specific enthalpy of the geothermal steam at the turbine inlet kJ/kg

h_{out} is the specific enthalpy of the geothermal steam at the turbine outlet kJ/kg

The outlet enthalpy is calculated using the assumed efficiency of the turbine this is assumed to be 85%.

$$\eta_s = \frac{\Delta h}{\Delta h_s}$$

η_s Is the isentropic enthalpy of the turbine, which can be assumed to be 0.85

Δh is the actual enthalpy drop across the turbine in kJ/kg

Δh_s is the isentropic enthalpy drop across the turbine in kJ/kg

The isentropic enthalpy drop is calculated with the condensing temperature that is assumed to be 45°C and using either steam tables or a Mollier diagram the isentropic outlet enthalpy is calculated.

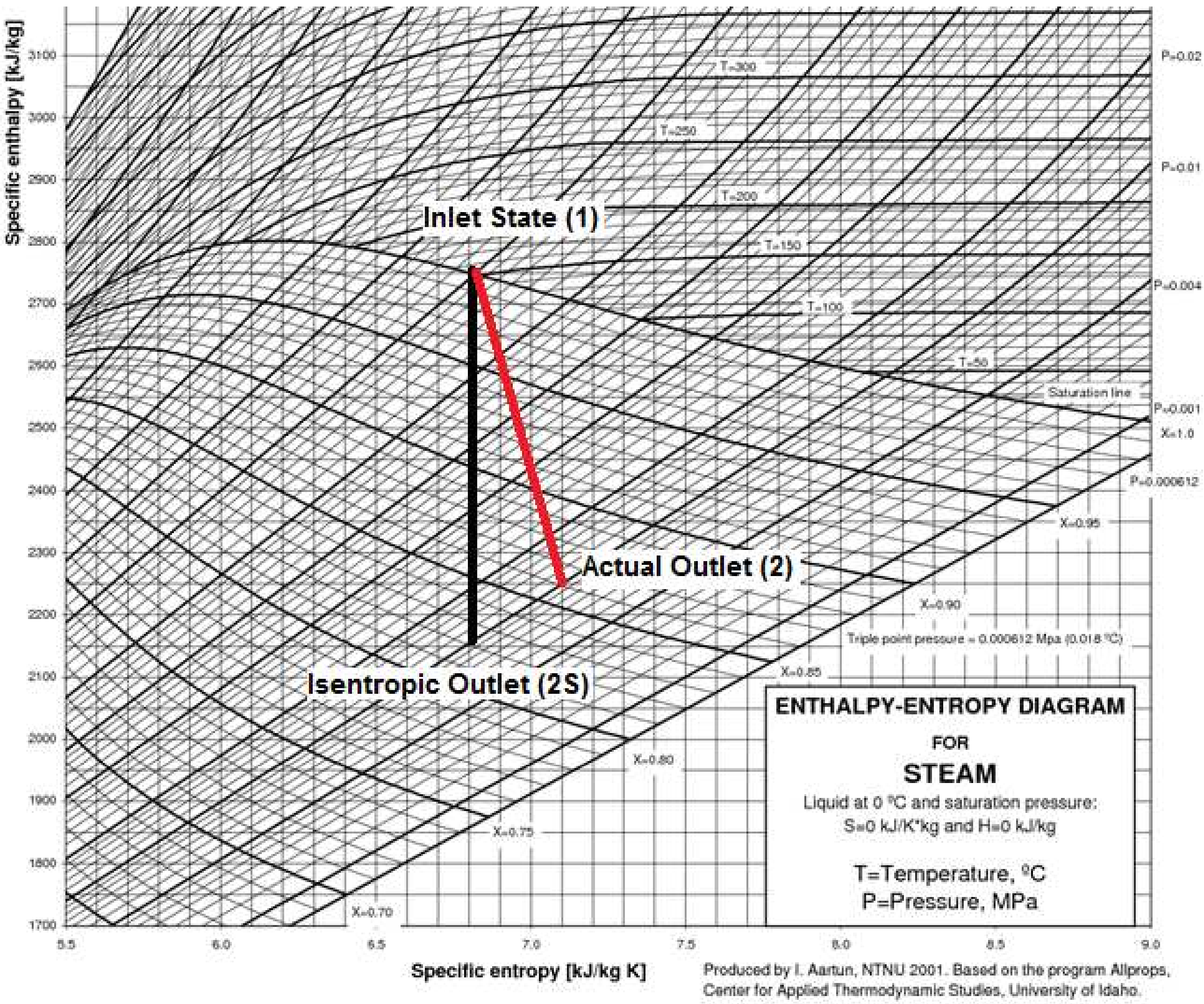


Figure 10 - Mollier Diagram – Credit University of Idaho [70]

The isentropic outlet enthalpy is the enthalpy of the steam at the condensing temperature of the system after an ideal turbine.

1.5.1.2 Single Flash Power Plant

A single flash geothermal power plant uses a water dominated geothermal resource. The flash process takes the high temperature geothermal fluid from the reservoir and expands the fluid in an isenthalpic process to a lower temperature and a portion of the fluid is flashed to steam. The T-S diagram in figure 11 illustrates this process.

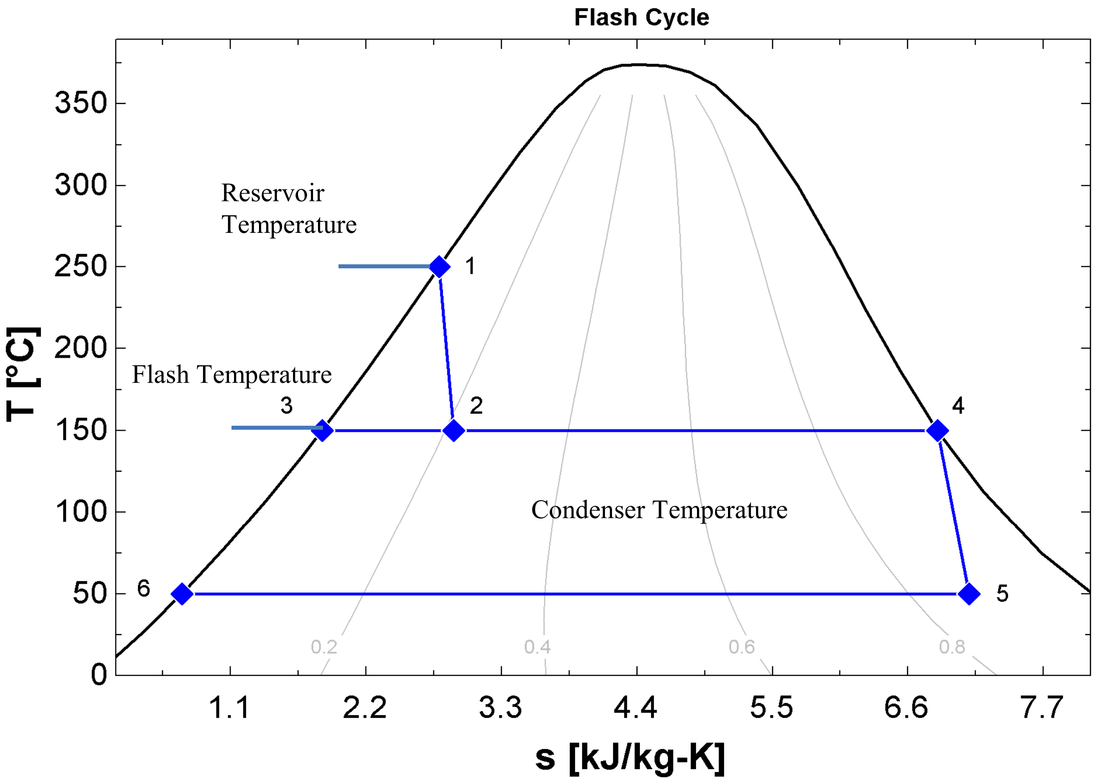


Figure 11 - T-S Diagram of a single Flash Steam Plant – Created in EES

Stage 1 : State of fluid in the reservoir

Stage 1-2: Isenthalpic expansion of the fluid to a lower temperature

Stage 3: Liquid resource from the separator

Stage 4: Steam resource from the separator for the turbine

Stage 4-5: Turbine expansion

Stage 5: Turbine outlet / condenser inlet

Stage 5-6: Condensation

State 6: Steam condensate from the condenser. This is mixed with the brine from the separator and sent for reinjection.

1.5.1.2.1 Flash plant Power estimate

The first step to estimate the power output of a flashing geothermal power plant is to estimate the separator temperature. The optimum separator temperature is related to well head pressure and flow rate; this is an approximate formula that is suitable for early power estimates[12].

$$T_{sep} = \frac{T_{res} + T_{con}}{2}$$

T_{sep} Is the optimum temperature to separate the fluid °C

T_{res} Is the temperature of the geothermal reservoir °C

T_{con} Is the temperature of the condenser of the system, which can be assumed to be between 40-50°C

Once the optimum flash temperature is calculated the steam quality is calculated with use of steam tables and the enthalpy of each state.

$$x_{sep} = \frac{h_{sep} - h_f}{h_g - h_f}$$

x_{sep} Is the quality of the steam after the flash process %

h_{sep}Is the enthalpy of the fluid entering the separator, which is the same as the enthalpy of the reservoir kJ/kg

h_f Is the enthalpy of saturated liquid at the separator temperature kJ/kg

h_gIs the enthalpy of saturated vapour at the separator temperature kJ/kg

An alternative method to determine the vapour quality at the separator temperature is to use the T-S diagram and read the entropy values of each state to determine the vapour quality.

$$x_{sep} = \frac{s_{sep} - s_f}{s_g - s_f}$$

s_{sep}Is the entropy of the fluid entering the separator, which is not equal to the entropy of the reservoir kJ/kg

s_f Is the entorpy of saturated liquid at the separator temperature kJ/kg

s_gIs the entropy of saturated vapor at the separator temperature kJ/kg

The steam mass flow rate is calculated with the steam quality and is used to calculate the power output of the turbine.

$\dot{m}_s = x_{sep}(\dot{m}_{geo})$

$\dot{m}_s$ is the steam mass flow rate in kg/s

x_{sep} Is the quality of the geothermal fluid once separated

$\dot{m}_{geo}$ Is the total mass flow rate of the geothermal fluid from the reservoir in kg/s

The steam mass flow rate is used in the general turbine equation below

$W = \dot{m}(h_{in} - h_{out})$

1.5.1.3 Double flash Power Plant

In some cases multiple separators are used in series to flash the geothermal fluid two times to produce more steam for the turbine. The brine from the first separator is flashed a second time and this steam is used in the lower stages of the turbine.

Figure 12 shows the T-S diagram of a double flash geothermal process; the blue line is the high pressure flash process and the red line is the low pressure flash.

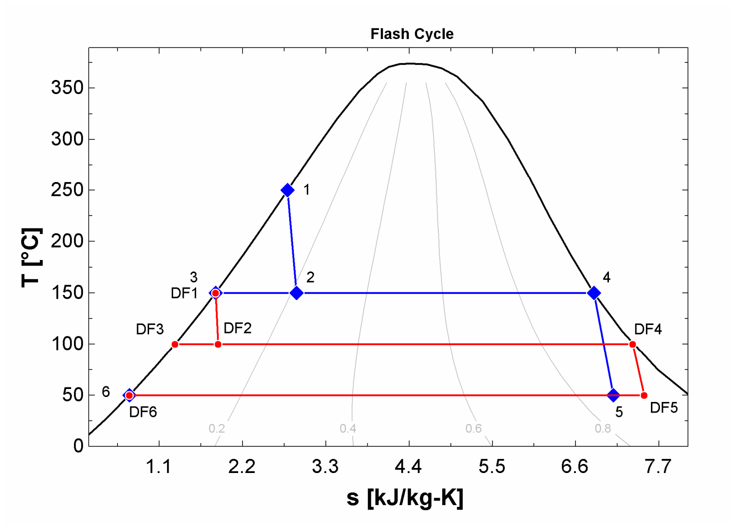


Figure 12 - T-S Diagram of a double flash steam plant – Created in EES

1.5.1.3.1 Flash plant Power estimate

The power output of a dual flash power plant uses the same principles of a single flash power plant and the secondary separator flashes the geothermal brine discharged from the first separator.

The total geothermal flow rate into the second separator is the brine flow rate from the first separator.

$\dot{m}_{geo,2} = \dot{m}_{geo} - \dot{m}_s$

$\dot{m}_{geo,2}$ Is the mass flow rate of the geothermal fluid entering the second seperator kg/s

$\dot{m}_{geo}$ Is the mass flow rate of the geothermal fluid entering the first separator kg/s

$\dot{m}_{s}$ Is the mass flow rate of the steam leaving the first separator kg/s

1.6 Initial ORC Performance and Component size

The following section determines the performance of the ORC and rough component sizes. This section uses a series of graphs from a simple ORC that uses a number of typical assumptions. The results from this section are not exact and will change with a proper thermodynamic model.

1.6.1 The ORC model

The ORC modelled used in these graphs is valid for geothermal fluids between 90 and 160°C. A geothermal fluid below 90°C will require very low cooling fluid temperatures and a fluid greater than 160°C is more likely suitable for a flash steam power plant or an ORC system that utilizes both steam and brine[5].

The geothermal fluid in this model is all brine and the cooling temperature is 20°C ; this maintains similar conditions for each working fluid. The heat exchangers used in this model are optimal heat exchangers with a 5°C pinch point this is the smallest allowable pinch point temperature for a phase change heat exchanger[71].

This model provides an accurate estimate of ORC power and a short list of the most likely fluids for the ORC. The owner will then decide whether to continue with their own modelling or approach a technology vendor.

The model uses two types of working fluids for the ORC: hydrocarbons; N-Pentane, Iso-Pentane, N-Butane, Iso-Butane, and refrigerants; HFE7000, R245fa, and R134a. There is evidence of each of these fluids use in an actual ORC.

The ORC pressure was limited to 25 bar to maintain a subcritical cycle for R134a and Iso-Butane while also reducing the potential component cost for high pressure equipment[72]. The minimum geothermal outlet temperature is limited to 70°C avoid excessive fouling in the heat exchangers and possible silica scaling.

The turbine and the pump are both assumed to have an 85% isentropic efficiency.

The graphs in this section each use the legend in figure 13.

Figure 13 - Working fluid legend

1.6.2 ORC Performance

The following sections are related to the performance of an ORC at different geothermal temperature with various working fluids. Each performance parameter unless otherwise stated is specific to the brine mass flow rate, for example kW ORC power / kg/s brine flow rate.

1.6.2.1 Specific work output

The potential work from an ORC at each resource temperature is shown in figure 14. The work is specific to the brine mass flow rate. Additional lines are provided to show the ORC in a hot or cold climate. Actual ORCs are also plotted as black crosses to show that the results are within reasonable bounds. This power output does not consider condenser loads.

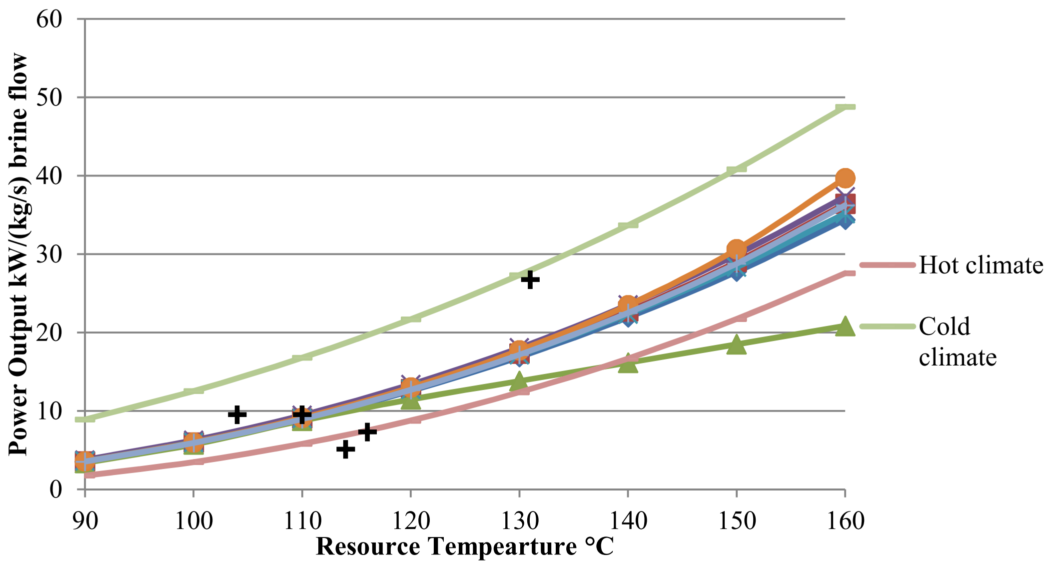


Figure 14 - Output of a basic ORC relative to brine flow rate

1.6.2.2 Geothermal outlet temperature

The reinjection temperature is critical for any geothermal system because excessive cooling will impact the heat exchangers and potentially the reservoir [12]. The temperature of the fluid leaving the ORC should be restricted by the geochemistry of the fluid to avoid significant fouling within the heat exchangers and reinjection well. Figure 15 shows that the working fluid selection will impact the reinjection temperature. R134a is restricted by the 25 bar pressure limit which results in more heat extracted from the geothermal fluid.

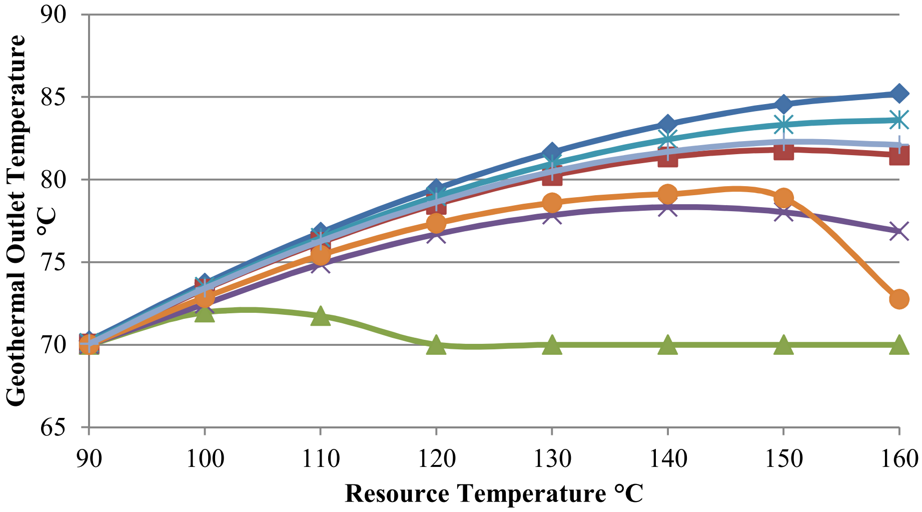


Figure 15 - Optimum geothermal reinjection temperature of a basic ORC

1.6.2.3 Thermal Efficiency

The thermal efficiency of the basic system shows how well the heat delivered to the system is converted to useful power from the turbine. Each of the working fluids in figure 16 have similar thermal efficiencies. One exception is R134a which has a significantly worse thermal efficiency resulting from the 25 bar pressure limit.

$$\eta_{th} = \frac{W_{out}}{Q_{in}}$$

η_{th} Thermal effieciency of the ORC %

W_{out} The work produced by the ORC kW

Q_{in} The heat delivered to the system from the geothermal fluid kW

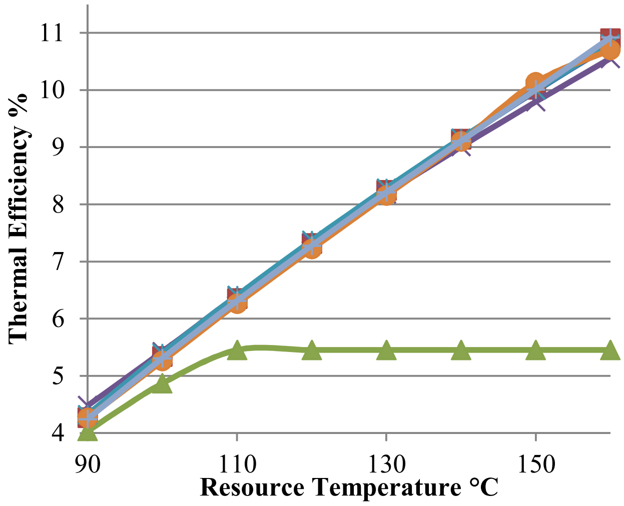


Figure 16 - Thermal Efficiency of a basic ORC

1.6.2.4 Parasitic Loads associated with the condenser

The condenser is a significant parasitic load on the system. An air cooled condenser uses fan banks to cool the fluid and a water cooled condenser requires pumps to maintain the cooling fluid flow. The motors for the pumps and the fans are both 75% efficiency[71]. Figure 17 estimates the parasitic loads associated with the two condensing systems **(left hand axis is a water cooled system, right hand axis is the air cooler).**

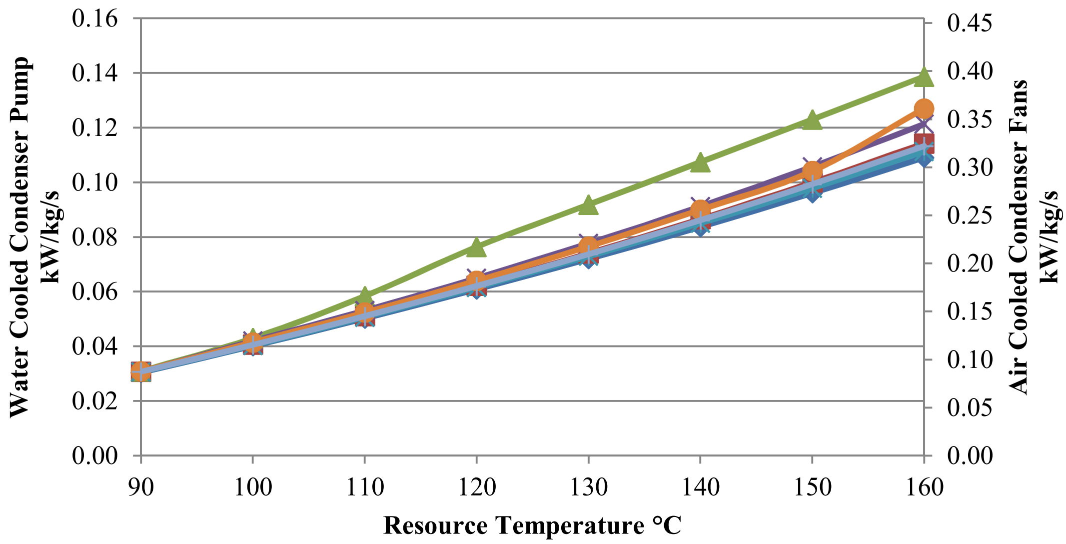


Figure 17 - Required cooling fluid power

1.6.2.5 Mass flow rate ratio

The required working fluid mass flow rate will impact the pipeline design. The equation below is the head loss equation which highlights the impact the fluid velocity has the pressure losses in a system.

$$h_f = f_d\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right)$$

h_f Head loss due to friction m

f_d Darcy friction factor

L Lenth m

D Hydraulic Diameter m

V Average Velocity m/s

g Gravity m/s^2

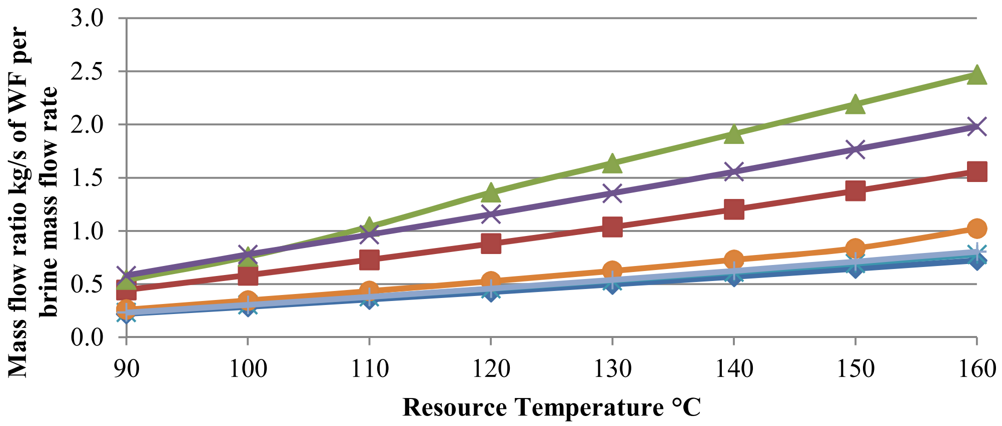


Figure 18 - Working fluid mass flow rate relative to brine flow rate

1.6.2.6 Volumetric Expander Performance Limits

Voluemtric expanders are more suitable to small ORCs because the there is no market for small turbines. A custom designed small turbine is uneconomical unless the design company uses the ORC as a marketing opportunity and offers a competitive solution. Screw expanders and scroll expanders have been used in small ORCs but their isentropic efficiencies are between 35-70%[53]. The maximum possible power output of these expanders is determined by the largest commercial compressors.

The exhaust flow rate from a volumetric expander limits their power output. The exhaust flow rate is equivalent to the maximum inlet flow rate of the original compressor design [26]. Figure 19 predicts the potential maximum power output of volumetric expanders **(the left hand axis is for a screw expander and the right hand axis is for a scroll expander).**

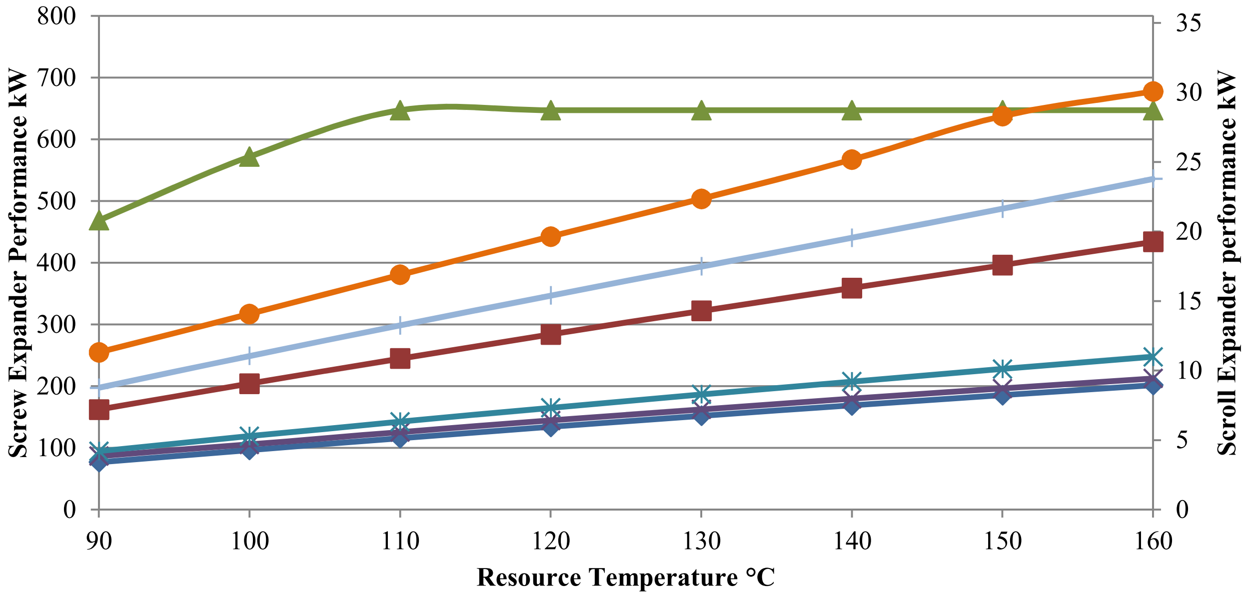


Figure 19 - Potential Maximum work of a volumetric expander

1.6.3 ORC components

The following sections investigate impact the fluid temperature and working fluid has on the components. The increase in component size is an unavoidable cost to a system[26].

1.6.3.1 Heat Exchangers

Heat exchangers in large ORC systems contribute at least 20% of the capital cost[21] this higher in a low temperature ORC. The heat exchanger sizes in the following sections are determined by basic overall heat transfer values.

1.6.3.1.1 Evaporator

The evaporator facilitates the heat transfer from the geothermal fluid to the working fluid. The evaporator here acts as both a pre-heater and an evaporator. The working fluid enters the evaporator as a sub cooled liquid and leaves as a saturated vapour. Figure 20 estimates the size of the heat exchanger for each working fluid relative to mass flow rate.

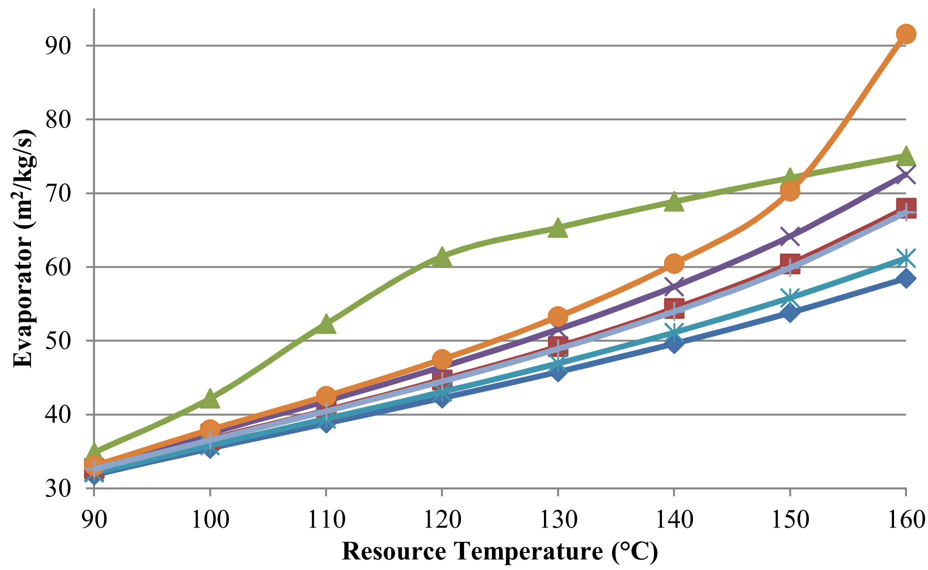


Figure 20 - Brine heat exchanger size

1.6.3.1.2 Condenser

The condensers commonly used in ORCs are the air cooled condenser and the shell and tube water cooled condenser. For all fluids tested expect R134a the fluid enters as a superheated vapour and leaves as a saturated liquid. R134a the fluid enters as a wet vapour and leaves as a saturated liquid. **(left hand axis is for a water cooled condenser and the right hand axis is for an air cooled condenser)**

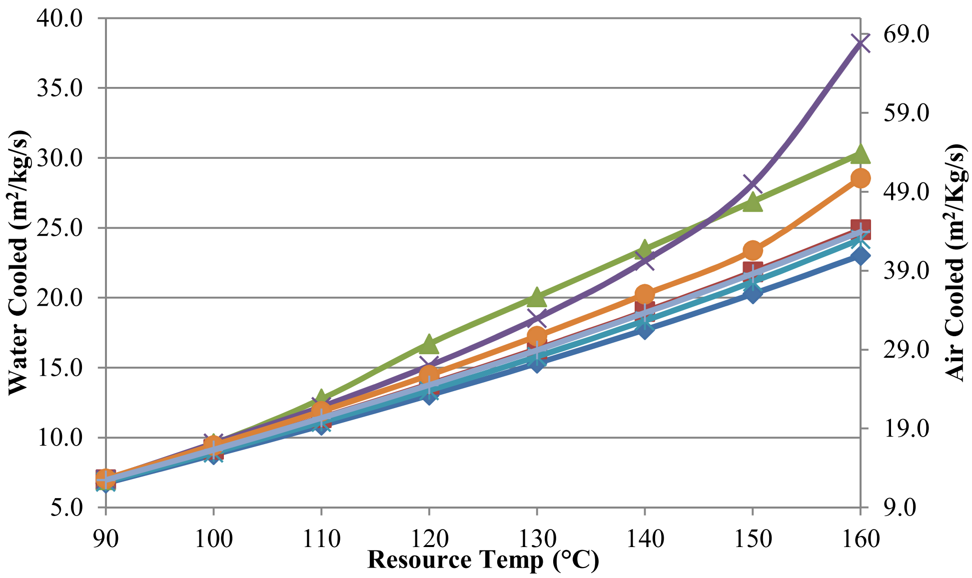


Figure 21 - Basic Condenser size

1.6.3.2 Expander Conditions

The expander is a critical component for a successful ORC as it extracts useful work from the working fluid. A custom made turbine will achieve high efficiencies but will also be very costly. Early understanding of the preliminary design features of a turbine will show the owner complications with a small turbine.

1.6.3.2.1 Specific Speed

Figure 22 can be used to calculate the specific speed of a simple single stage turbine, with the square rooted mass flow rate of the geothermal fluid as specific speed is relative to the square root of the flow rate. The specific speed for this turbine is calculated for a direct drive synchronous turbine spinning at 3000 RPM. Figure 23 is a basic specific speed chart for expanding gas turbines and is used to estimate the turbine efficiency. Therefore, determining the specific speed of the turbine from figure 22 and looking up the isentropic efficiency of the turbine on figure 23 will give the owner an idea of the turbines potential. A more detailed turbine analysis is left for the pre-feasibility and feasibility stages which look into ways to improve the efficiency.

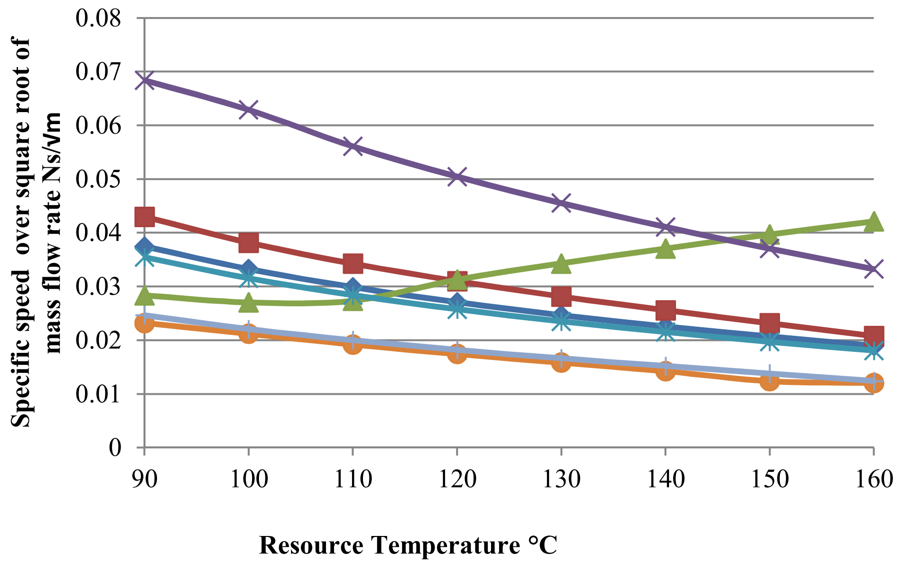


Figure 22 - Specific speed of the turbine relative to the square root of the brine flow rate

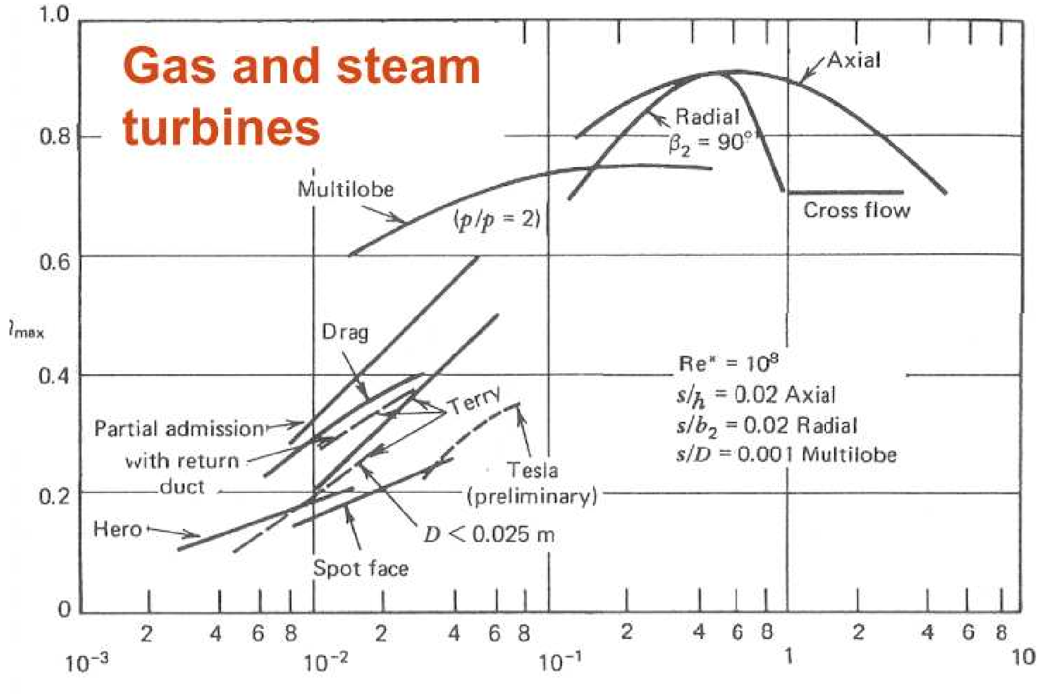


Figure 23 - Specific Speed Chart – Credit Balje [73]

1.6.3.2.2 Expansion Ratio

A fluid with a high pressure ratio will require more stages to accommodate the larger pressure drop across the turbine. A generally rules is that an expansion ratio above 4 will require more than one stage[31]. Alternatively, volumetric expanders can only withstand an expansion of around 4 as well[26].

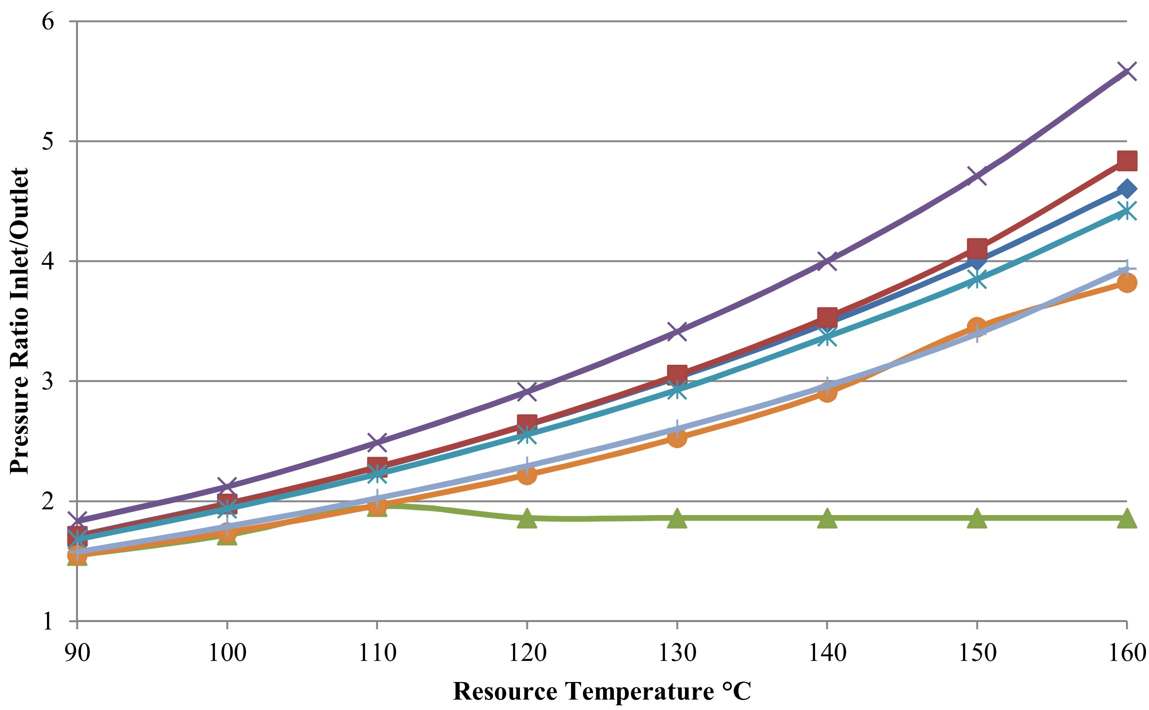


Figure 24 - Expander Pressure Ratio

1.6.3.3 Other aspects that effect Component Size

Figure 25 shows the vapour density of the other fluids. Low vapour densities impact increase the size of ORC components and velocities in piping. Low vapour densities increase the overall system costs.

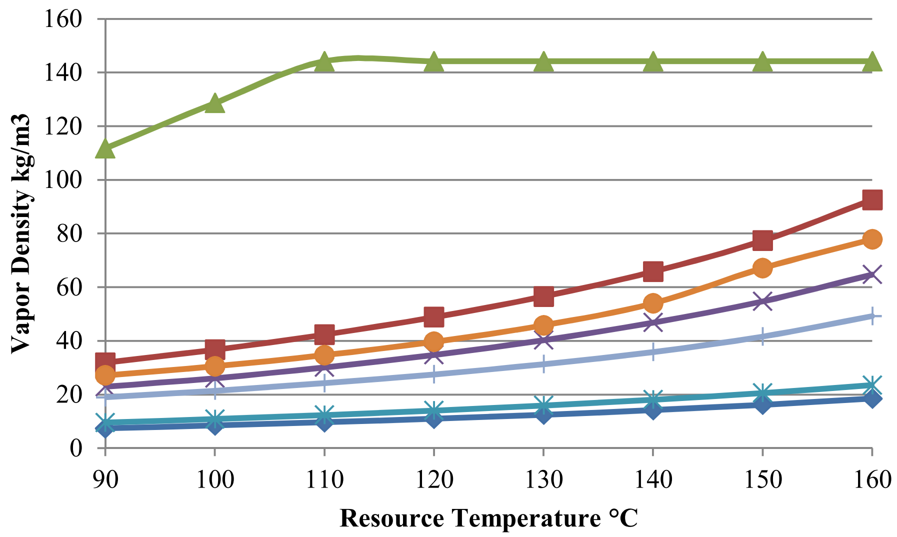


Figure 25 - Vapour Density at the Expander Inlet

The condenser pressure should remain above atmospheric pressure to limit auxiliary components for the ORC and the size of the condenser[46]. The condenser pressure can drop below atmospheric but the ORC will require equipment to purge the air from the system. Figure 26 shows the condenser pressure for each of the fluids investigated. Fluids close to the atmospheric level will need to accommodate for below atmospheric pressure in cold conditions. The difference between the condensing pressure and the atmospheric pressure reduces the allowable pressure drop in the condenser.

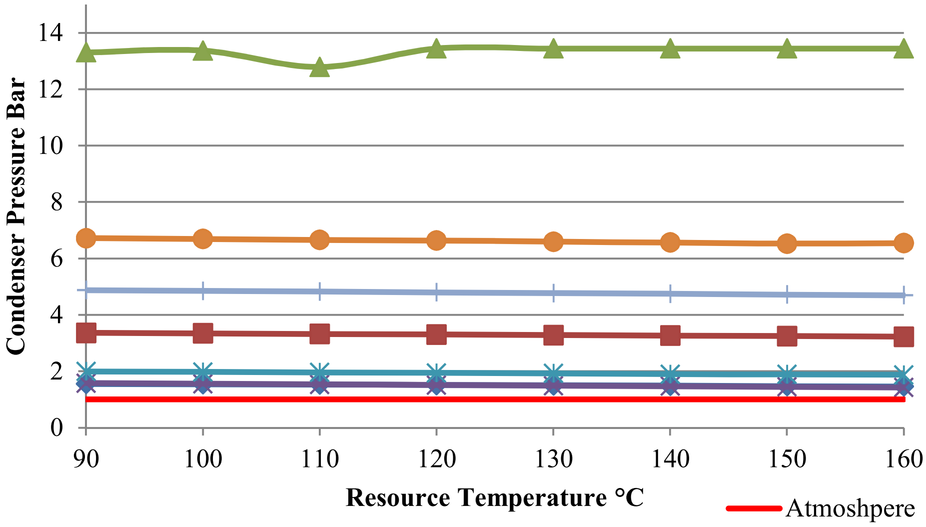


Figure 26 - Condenser Pressure

1.6.3.4 Pump Considerations

The feed pump load impacts the ORCs performance. Figure 27 can be used to estimate how each working fluid impacts the pump performance. This feed pump is assumed to be 85% efficient.

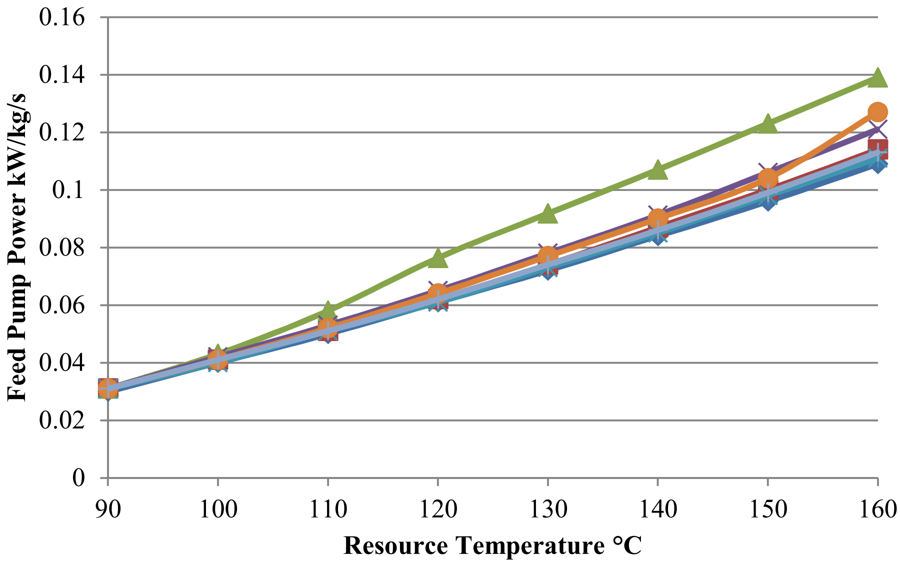


Figure 27 - Feed Pump parasitic loads

1.6.4 Second ORC cost estimate

The ORC performance and component section has provided sufficient information for more detailed cost estimate.

1.6.4.1 Preliminary Cost Table

Use table 2 to estimate the capital cost of all the equipment for the ORC.

Table 2- Component Cost estimates for an ORC – Credit Nazif [24]

Equipment	Units	$/Unit USD
Evaporator	m^2	500
Condenser	m^2	600
Pump	kW	600
Turbine	kW	720
Overheads – piping, controls, instrumentation, construction, civil works, engineering and supervision, contingencies, project management (UNU Nazif [24])	75% of equipment cost	

1.6.4.2 Operational and Maintenance Costs of an ORC

Table 3 uses basic values for small ORCs to estimate the life time maintenance cost of the ORC.

The operational costs for power plants vary depending on the amount of operation required. A number of small ORCs have remote operation that is part of a wider ORC network. It is recommended that full time operation is uneconomic if the power plant is less than 50MW[74]. Small ORCs will not be this large and remote monitoring or significant automation is required.

Table 3 – Annual Maintenance costs of an ORC – Credit Elliot [68]

Net Power, kW	Annual Maintenance Costs $/ year
100	$19,100
200	$24,650
500	$30,405
1000	$44,000

1.6.4.3 Basic payback period

The simple payback period is estimated wth the net revenue of the plant per year and the capital cost.

$$PB = \frac{CC}{NetRevenue}$$

PB is the payback period in years

CC is the capital cost of the plant $

$NetRevenue$ is the annual revenue of the ORC after the operational and maintenance costs

The net revenue is the net power produced after the parasitic loads of the plant sold to the grid.

Net revenue equation

$$NetRevenue = 8766(W_{net})(P_r)(C) - P_{OM}$$

W_{net} is the net power of the ORC kW

P_r is the sale price of electricity

C is the capactiy factor of the plant – 0.92

P_{OM} is the annual maintenance costs of the ORC

1.6.4.4 Land use

The relative land use is estimated with by Dipippo as 1415m^2/MW or 1.42 m^2/kW[75].

1.7 Data Sheet

The data sheet can be filled out with the important information covered in the prospecting stage and used to

Compare other possible geothermal locations.

Prospecting Data Sheet					
Evaluation of heat source					
Geo Brine			**Available Heat**		
Tin		°C	Resource Type		
Tout		°C	Q		kW
P		Bar	W		kW
$\dot{m}$		kg/s	Carnot Eff		%

Cooling Resource			**Initial Costs estimate**	
Type			Cost ($)	
Temperature (°C)				

Rough ORC Estimate			**Steam Plant**		
Min		kW	T_{sep}		°C
Max		kW	H_{sep}		kJ/kg
mid		kW	X_{sep}		%
Commercial ORC			s_{team}		kg/s
Likely Option			S_{team}		kW

Initial ORC					
Working Fluid Choice					
Work estimate (kW)					
T_{out} (°C)					
$\eta_{thermal}$(%)					

Parasitic Load					
WCC (kW)					
ACC (kW)					
Net Power (kW)					
Heat Exchanger Sizes					

Evaporator (m^2)					
ACC (m^2)					
WCC (m^2)					

Turbine					
Specific Speed					
Efficiency (%)					
Size					

Pump Power (kW)					

Costs ($)					
Evaporator					
ACC					
WCC					
Expander					
Pump					

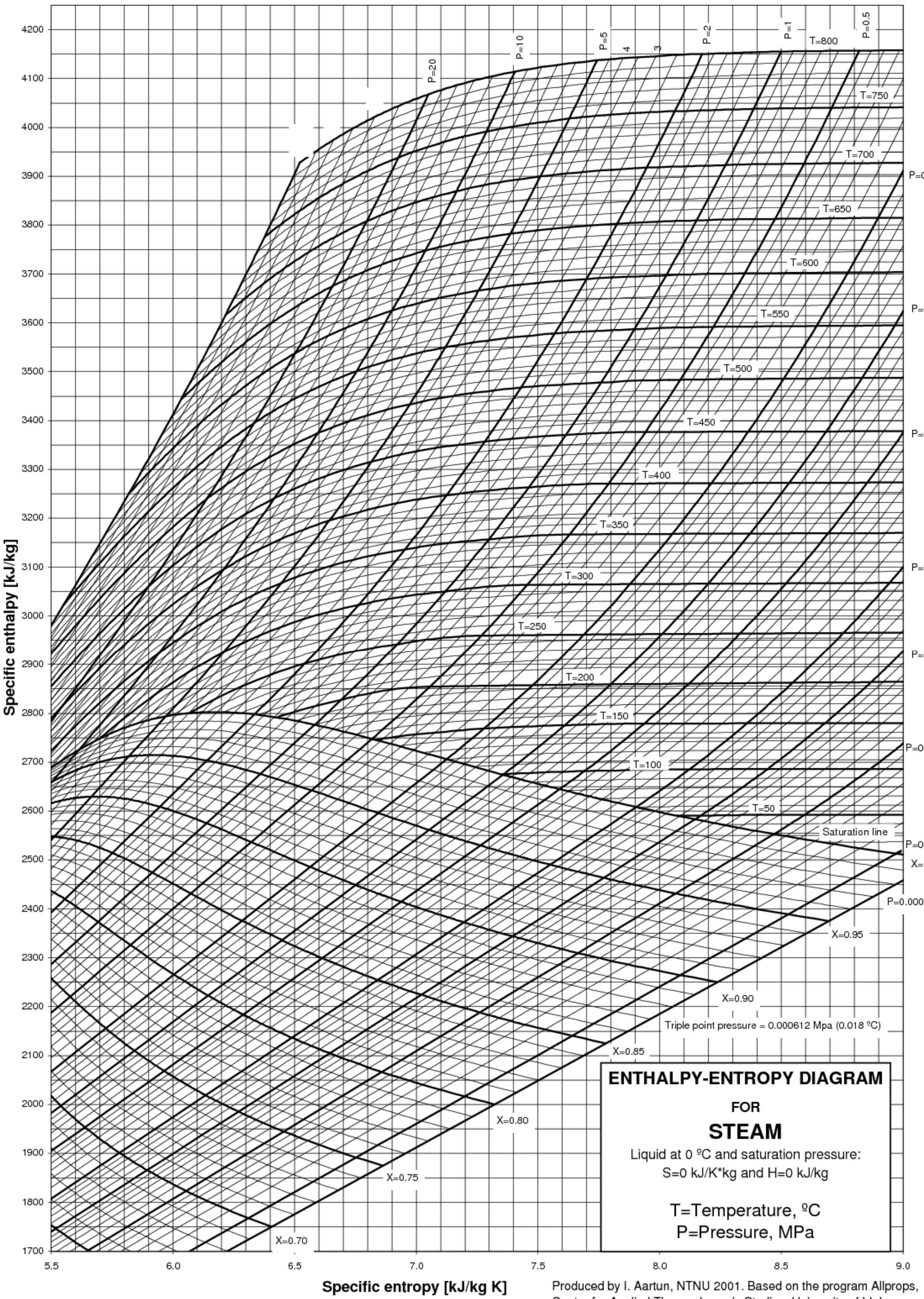
Total ACC					
Total WCC					

Operation and Maintenance					

Net Revenue					

Payback (Years)					
ACC					
WCC					

Land Use (m^2)					



Produced by I. Aartun, NTNU 2001. Based on the program Allprops, Center for Applied Thermodynamic Studies, University of Idaho.

Pre-Feasibility

Introduction - Pre - feasibility

This is the second section of the standard for low temperature geothermal Organic Rankine cycles. The previous section developed an understanding of the resource potential to continue with the prefeasibility investigation. This investigation explores the next level of thermo-cycle design and preliminary component selection and risk assessment.

The results of this section will provide the engineer with a robust understanding of the geothermal resource and the potential for an ORC. The concepts used in this study are also used in the next section for a more detailed feasibility analysis of the system. To complete this section thermo analysis tools are required.

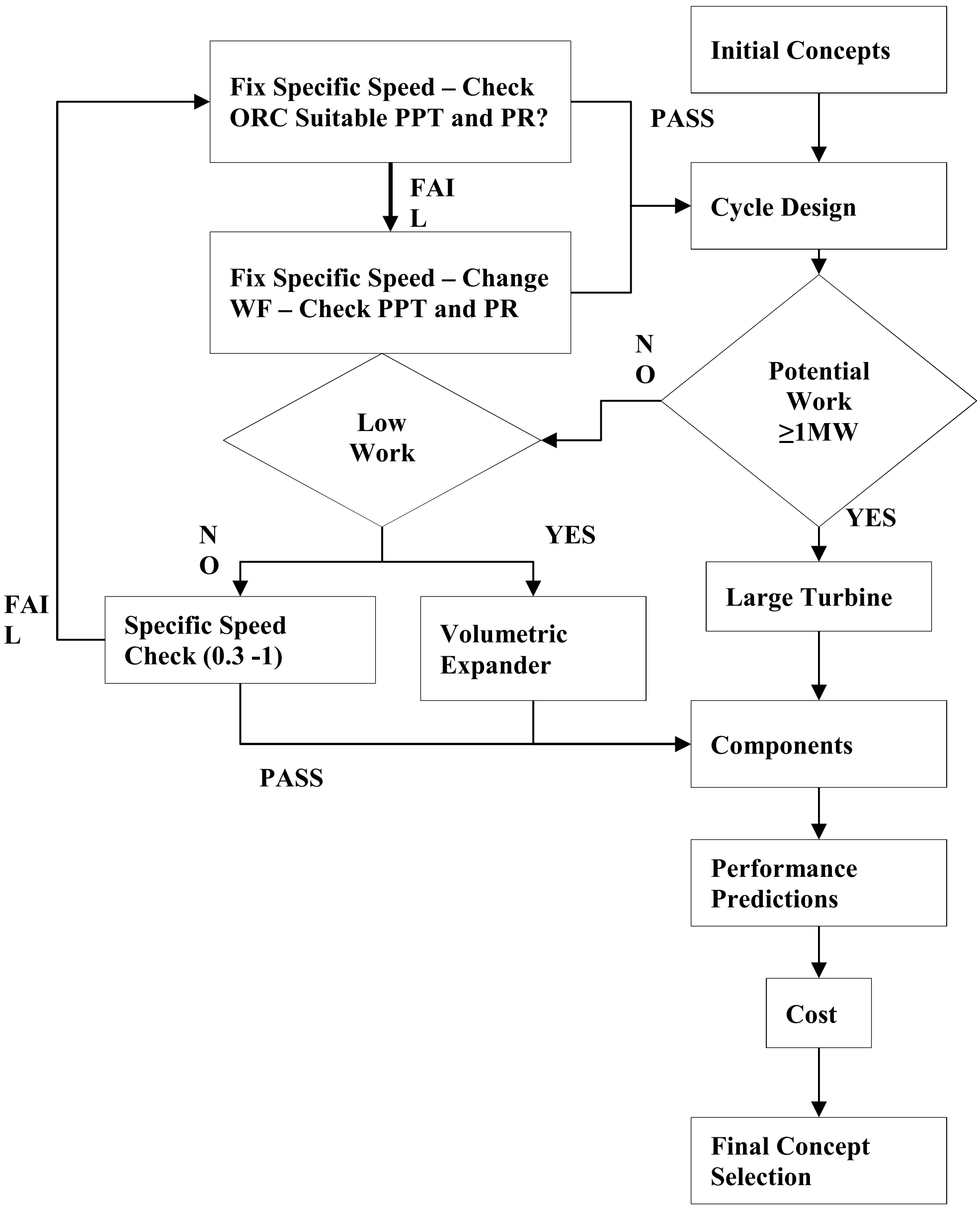
The intended audience for owners with a technical understanding of thermodynamics but have not modelled ORCs before. The other potential user for this section is an engineering wanting to learn ORC thermo cycle design. Once this section has been worked through the reader will have an understanding of the technical details of the ORC and detailed capital cost estimate.

Pre-feasibility Overview

The flow chart on the following page shows the main stages of the pre-feasibility study.

1. Initial Concepts – Determines what concept cycles will be modelled. These concepts will either be a basic cycle or a recuperative cycle. Different working fluids are also modelled.
    a. A small section of resource limitations is included if a geochemical analysis has not been done. It covers the basic silica problem in a geothermal system and should not be considered a robust geochemical study.
2. Cycle Design – This section guides the reader through the process for good thermodynamic cycle design for the different ORC concepts.
    a. The output of this section is all the critical information needed to make decisions on components and losses in the system.
3. The potential work decision gateway is the initial expander selection tool. Expanders are a critical component in an ORC and the feasibility of the ORC can depend on the available expander.
    a. Commercial turbines are readily available for an ORC in the MW range.
    b. A system between 250kW and 1MW can use a turbine; however, it is likely to be a custom design. A volumetric expander can still be considered in this range.
    c. Any system below 250kW is will use a volumetric expander. A custom turbine can still be considered but will be more expensive.
4. The range between 250kW and 1MW explores the turbine feasibility
    a. This uses basic specific speed equations and charts – if the specific speed is in the optimum range then a small turbine is feasible. Otherwise a volumetric expander is better suited. However, the cycle adjustments can improve the turbine feasibility but with significant penalties.
5. Components – This section goes over some of the selection and limitations of components in an ORC highlighting some of the losses associated with the components to give a better perspective of actual performance.
6. Performance Prediction – This remodels the ORC to include some of the losses in the system from the component selection.
    a. The main changes to the system are the new expander efficiencies and the assumed pressure losses in heat exchangers.
7. Costs – This sections looks at the detailed cost analysis of the system and the payback period.
8. Final Concept Selection – This will determine the concept design that is used for the development

Pre-Feasibility Process



2 Pre-feasibility

2.1 Scope

This section evaluates a possible concept ORCs and their performance. The best option from the pre-feasibility section is considered in a detailed feasibility study. The pre-feasibility study will first determine the ORC configurations, such as the working fluid choice, whether a recuperator is used, type of condenser, and the possible geothermal outlet temperature. Each concept is modelled to determine the benefits of each cycle. The components of these concepts are then explored in more detail to understand the performance impacts of the component selection.

A data sheet is provided at the end of this chapter that should be used to fill in the critical information of the study.

2.2 Concept

The concept selection uses the two most common ORC configurations the simple 5 component OR:, pre heater, evaporator, expander, condenser, and pump or the addition of a recuperator to do some pre heating;.

The reader will have to decide on a number of different concepts to compare. Generally all of these concepts will have similar cycle configurations but will use different fluids. A recuperator is only considered if there are strict limitations on the reinjection temperature.

2.2.1 Resource Considerations

The resource analysis is not the scope of the standard and all resource information should be known; this information impacts component design and a poor understanding of the geochemistry will impact the performance of the heat exchangers[21].

A geothermal pump is recommend if the well cannot flow independently [76]. Pumped fluid will ensure the fluid stays a liquid and non-condensable gases (NCGs) remain in solution. This will allow all NCGs to be pumped back into the reservoir, avoiding any gas emissions.

Geothermal power plants have failed in the past because of limited understanding of the geothermal reservoir [77]. There is always a risk when dealing with a geothermal resource as the nature of the fluid can change once operation begins[78]. However, the more investment into understanding the potential and risks associated with the resource should be beneficial.

2.2.1.1 Silica

The simple silica saturation curves are provided if no geochemical analysis has been done [67]. However, this is a basic analysis and the pre-feasibility study should not continue without a proper resource analysis.

Silica is not the only issue with geothermal brine; stibnite, which can deposit around hot springs[79] also causes problems with low temperature geothermal ORCs.

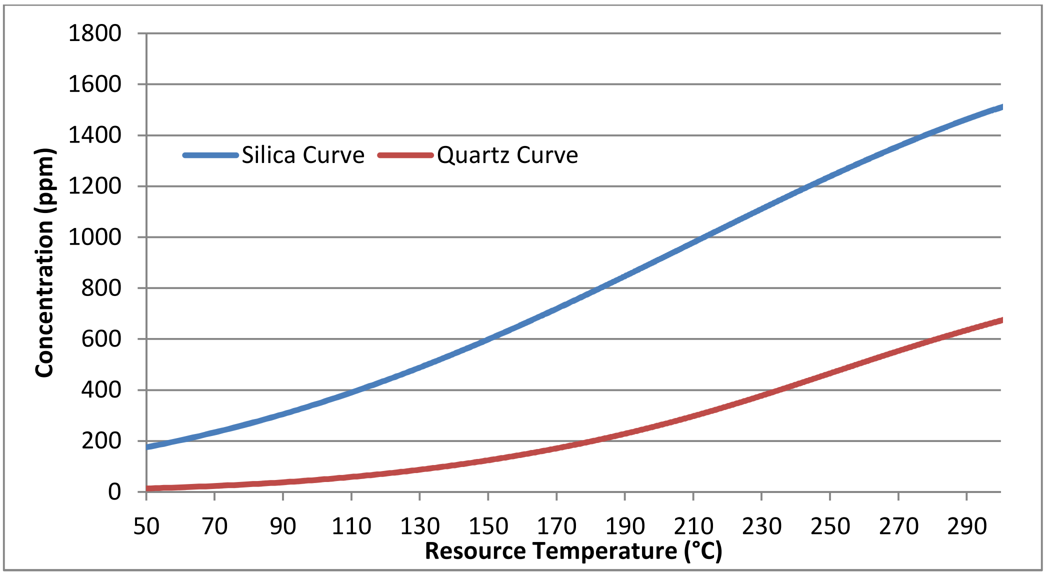


Figure 28 - Silica Scale Potential in a brine only solution

Figure 28 is the quartz and silica saturation curves[67]. The downhole temperature determines the concentration of quartz in the brine. The final reinjection temperature must remain above the corresponding concentration on the silica concentration curve.

A two phase resource that uses a separator increases the concentration of quartz according to the dryness of the steam. The flowing cases show the two how the resource condition impacts the reinjection temperature.

Example Case 1 – Brine only

The downhole temperature is 240°C and the brine is not flashed to steam at any point. Therefore the minimum reinjection temperature to avoid silica saturation is 110°C. Silica can be a minor issue on some low temperature ORCs [12].

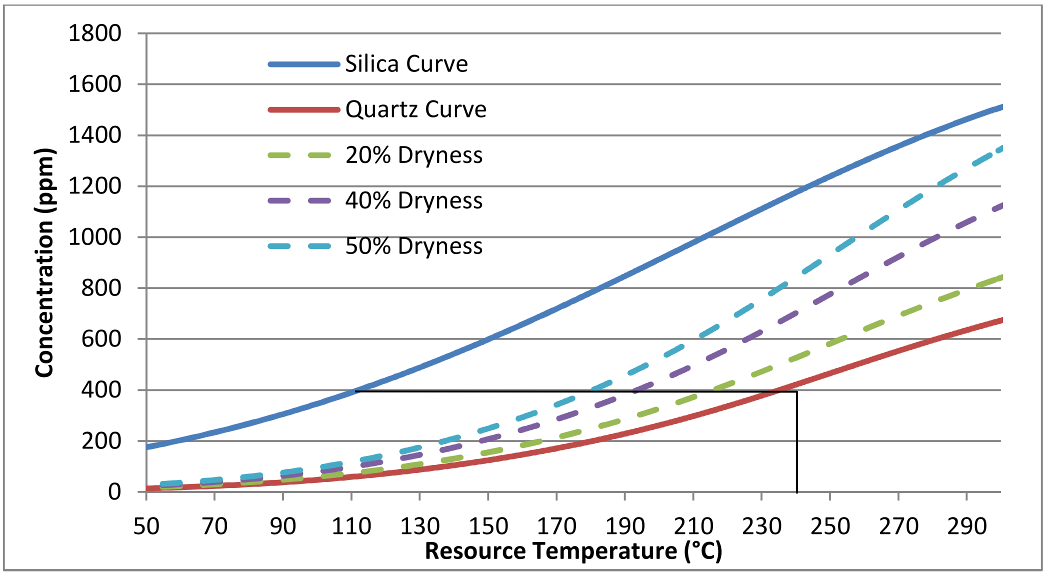


Figure 29 - Separated Fluid – Dashed lines represent different steam amounts

Example Case 2 – Flashed steam

The downhole temperature is 240°C the fluid is flashed and separated at 200°C, which results in a steam quality of 10%. This increases the quartz concentration in the brine to 500ppm opposed to 400 ppm such as case 1. Therefore; this brine must be re-injected above 130°C to avoid silica saturation.

.

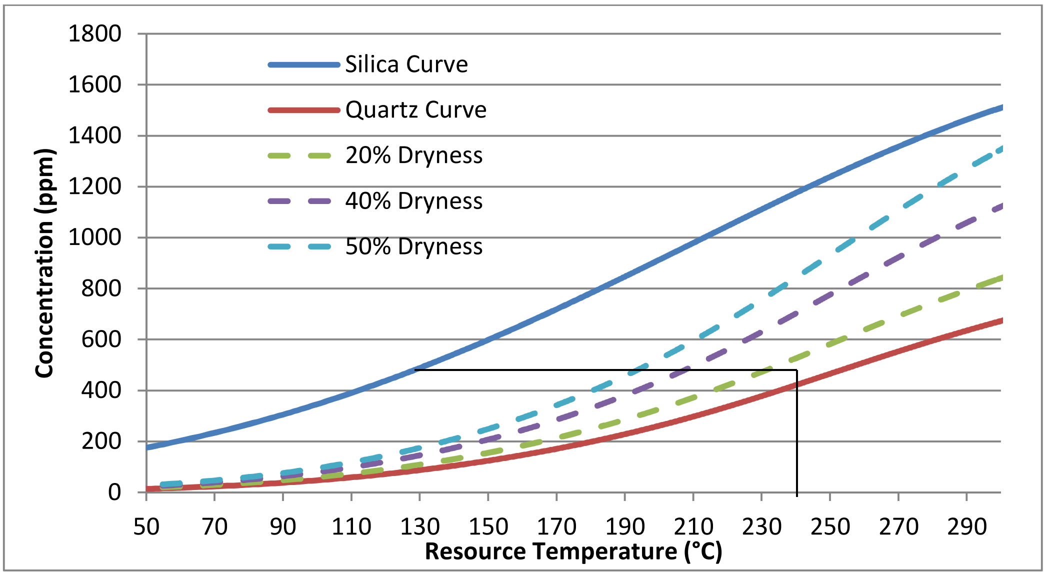


Figure 30 - Silica Example

2.2.2 The General Cycle

The concepts that will be considered for the thermo analysis are decided upon. Any number of different concepts can be selected; however, it will take more time to compare the trade-offs of each concept.

2.2.2.1 Rule of thumb assumptions

There are a number of assumptions that can be made at this point to model the cycle [71].

- 3°C superheat
- 5°C sub cooling
- 15°C pinch point in evaporator
- 5°C pinch point in the recuperator
- 14°C approach temperature in the condenser
- Top 1% ambient temperature condition for an air cooled condenser
- 85% isentropic efficiency in the pump
- No heat loss in the system

2.2.3 Expander Considerations

The expander is a critical component of the ORC and the thermodynamic model with use either a turbine or a volumetric expander efficiency assumption. An ORC with an output potential greater than 250 kW uses a turbine and isentropic efficiency of 85% for the cycle analysis[26]. A system less than 250kW will use a volumetric expander; both the high 70% and low 35% isentropic efficiencies are considered for the volumetric expander[53]. The range between 250kW and 1MW is an overlap zone where either expander can be used. At the point of writing this document it is suggested that below 1MW is uneconomic for a turbine; however, there are some examples of ORCs in this range using turbines.

2.2.4 Concepts

A number of concepts must be considered to determine the best configuration. The working fluids used in the prospecting stage are common fluids used in commercial ORCs. The reader can use other fluids for the models. Other fluids should fulfil the property list below - credit from Quolin [26], Chen [27], Maizza[28, 80], Zhai[29].

- Recommended Fluid Properties

- o High Latent Heat
- o Low specific volume of vapour (high vapour density)
- o Low Viscosity
- o Reasonably low operating pressures
- o High thermal conductivity
- o Good stability
- o Minimal environmental impact
- o Non-Flammable
- o Dry or isentropic
  - ▪ Figure 31 is dry working fluid – best for turbine safety.
  - ▪ Figures 32 and 33 are isentropic and wet working fluids. These require careful expander design considerations.
- o Leaks easily detectable.

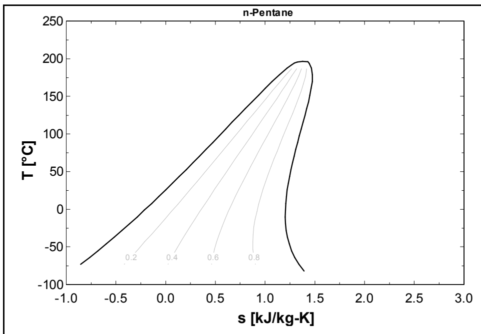


Figure 31 - Dry Working fluid – Created in EES

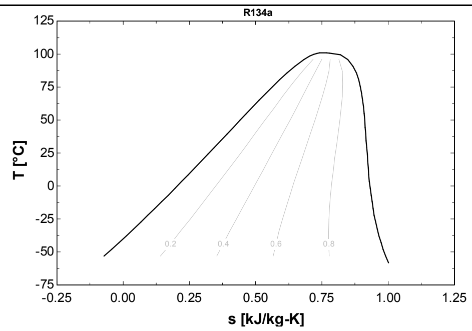


Figure 32 - Isentropic Working Fluid - Created in EES

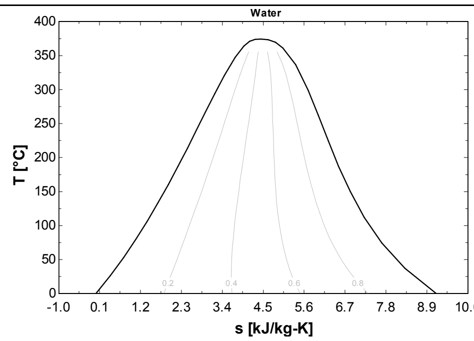


Figure 33 - Wet Working Fluid - Created in EES

2.3 Cycle Design

The following sections require thermodynamic software that can calculate the thermo properties of the fluids at each of the state points in the ORC.

Possible software

- Engineering Equation Solver – Extensive Thermo Library (Software used by the author)
- Thermoflow
- Aspentech
- Cycle-Tempo

- AxCycle

Alternatively thermo property packages such as cool-prop (open source) or Refprop can be used in conjunction with excel to complete the cycle analysis.

2.3.1 Modelling the ORC

An ORC thermo model is an energy balance between the hot source fluid, the working fluid, and the cooling fluid. Figure 35 shows the critical state points of the ORC and table 4 lists each of the energy exchange equations. To determine the energy balances at each of these state points the temperature, pressure, and or the quality of the fluid must be known.

2.3.2 Guideline

This is the outline of the general steps taken to model an ORC. Each of these steps will be discussed in more detail in the following sections

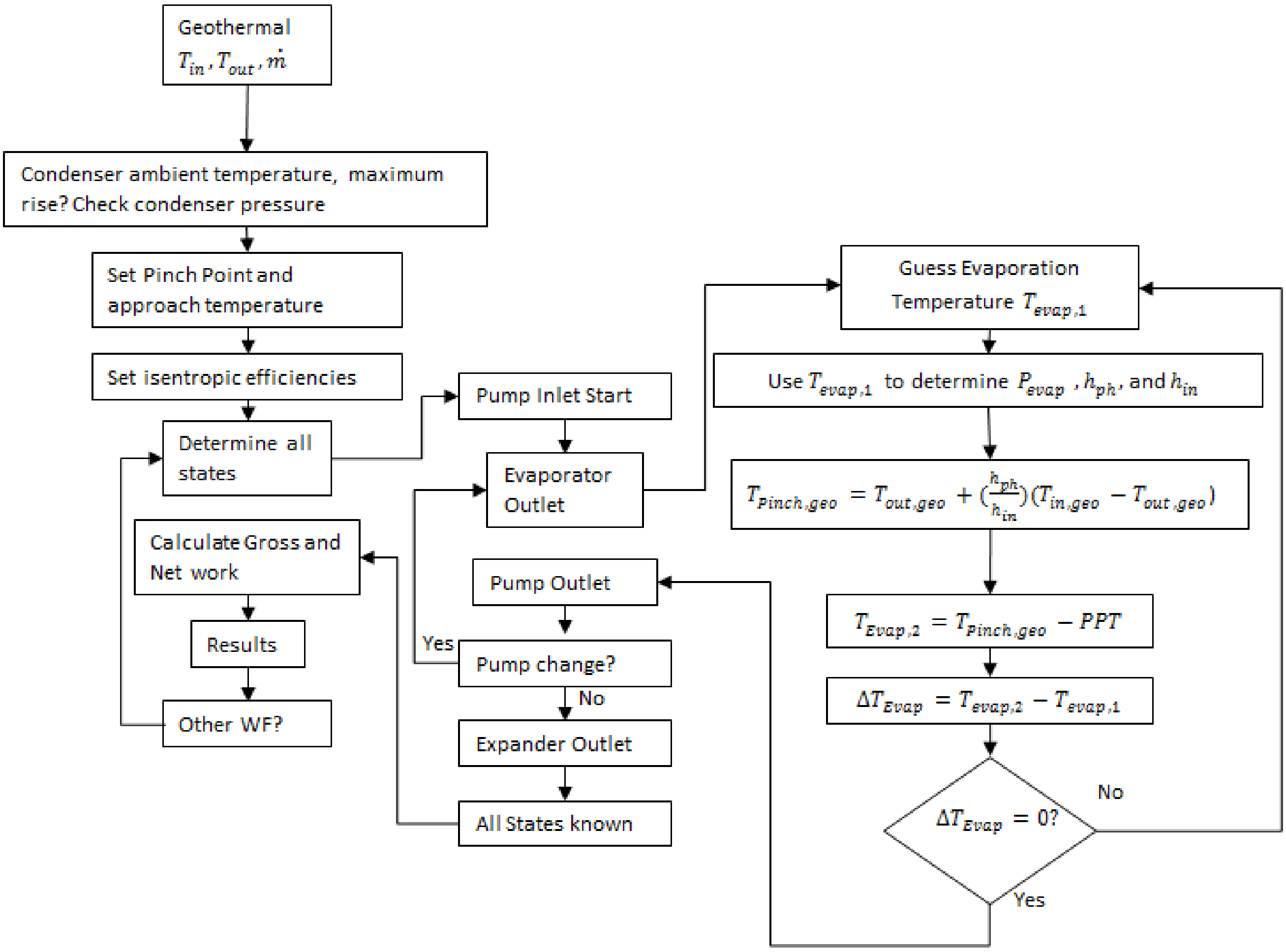


Figure 34 - ORC model approach

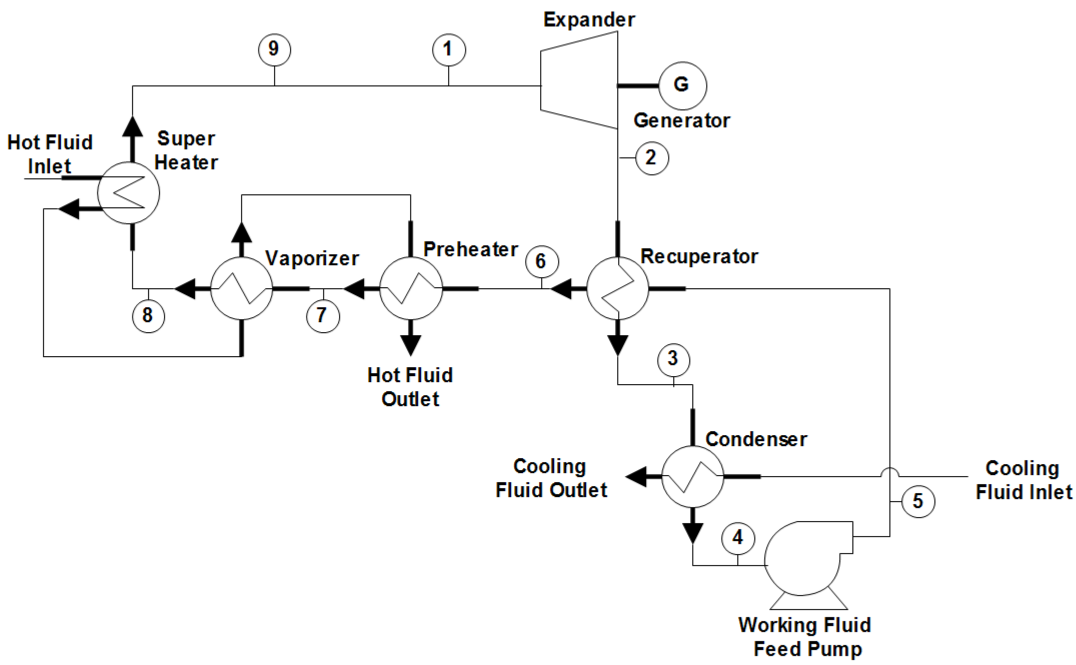


Figure 35 - Process flow diagram of the ORC

Table 4 - State points of an ORC and Energy Balance Equations

Component	Inlet State	Outlet State	Energy Transfer [kW]	Energy Balance	Isentropic or Transfer Efficiency
Expander	1 Superheated Vapour	2 Superheated Vapour	W_{12}	$W_{12} = \dot{m}(h_1 - h_2)$	$\eta_e = \frac{h_1 - h_2}{h_1 - h_{2s}}$
Recuperator – Desuperheat*	2 Superheated Vapour	3 Superheated Vapour	Q_{23}	$Q_{23} = \dot{m}(h_2 - h_3)$	$Q_{23} = U_r A_r \Delta T_{lm23}$
Condenser	3 Superheated Vapour	4 Liquid	Q_{34}	$Q_{34} = \dot{m}(h_3 - h_4)$	$Q_{34} = U_c A_c \Delta T_{lm34}$
Pump	4 Liquid	5 Sub cooled Liquid	W_{45}	$W_{45} = \dot{m}(h_5 - h_4)$	$\eta_p = \frac{h_{5s} - h_4}{h_5 - h_4}$
Recuperator*	5 Sub cooled Liquid	6 Sub cooled liquid	Q_{56}	$Q_{56} = \dot{m}(h_6 - h_5)$ $Q_{56} = Q_{23}$	$Q_{56} = U_r A_r \Delta T_{lm56}$ $Q_{56} = Q_{23}$
Pre-Heater**	6 Sub cooled Liquid	7 Saturated Liquid	Q_{67}	$Q_{67} = \dot{m}(h_7 - h_6)$	$Q_{67} = U_{ph} A_{ph} \Delta T_{lm67}$
Evaporator	7 Saturated Liquid	8 Saturated Vapour	Q_{78}	$Q_{78} = \dot{m}(h_8 - h_7)$	$Q_{78} = U_v A_v \Delta T_{lm78}$
Superheater**	8 Saturated Vapour	9 Superheated Vapour	Q_{89}	$Q_{89} = \dot{m}(h_9 - h_8)$	$Q_{89} = U_{sh} A_{sh} \Delta T_{lm89}$
Net Power			$W_{12} = W_{45} + Q_{56} + Q_{67} + Q_{78} + Q_{89} - Q_{23} - Q_{34}$		

Notes:

* In the scenario that the ORC does not require a recuperator these two sections are removed and the expander exits to the condenser and the pump exits to the preheater

** In the scenario that the ORC does not have a separate preheater and super heater these sections are removed from the table and the pump exits directly into the evaporator which works as a pre-heater and evaporator and then this goes directly to the turbine. This is less effective but will reduce the number of pressure vessels.

NOMENCLATURE

W	Specific Work output or input
h	Specific enthalpy
$\dot{m}$	Mass flow rate
Q	Heat rejected or absorbed
U	Overall heat transfer coefficient
A	Heat transfer area
ΔT_{lm}	Log Mean Temperature Difference
h_f	Head loss due to friction

Greek Symbols

η	Efficiency

2.3.2.1 Geothermal Parameters

The first step is to set the known geothermal resource condition: inlet temperature, outlet temperature and flow rate. However if the outlet temperature is not known set it to 70°C. This will be adjusted to find the optimum outlet temperature for the working fluid. The allowable geothermal flow will also be required.

$T_{in,Geo}$ Geothermal fluid temperature available for the ORC °C

$T_{out,Geo}$ the geothermal outlet temperature from the ORC °C

$\dot{m}_{geo}$ the geothermal mass flow rate kg/s

2.3.2.2 Condensing Condition

The condensing conditions depend on the type of condenser, either a water cooled condenser or air cooled condenser. The inlet temperature of the condenser, ambient conditions for an air cooled condenser, sets the outlet state of the expander. The design condenser conditions is critical for the ORC design[33]. This section uses the 5% maximum temperature of the cooling fluid[81]. This is the ORCs worst case scenario. A floating condenser conditions is explored in the feasibility study.

$T_{in,CF}$ condenser cooling fluid inlet temperature °C

2.3.2.3 Set the pinch point and approach temperature

The approach temperature is the difference between the outlet temperature of the working fluid in the condenser and the inlet temperature of the cooling fluid. The approach temperature is typically between 8-14°C[71].The approach cooling fluid inlet state and approach temperature set the condenser outlet condition.

Phase change heat exchangers commonly have a pinch point of 15°C and the minimum recommended pinch point is 5°C[71]. The condenser pinch point is set to 5°C and the evaporator pinch point is set to 15°C. The condenser can have a lower pinch point because the condensing condition has a larger impact on the system.

$PPT_{approach}$ the approach temperature is the difference between the condenser working fluid outlet and the cooling fluid inlet – Use 14°C

$PPT_{Condenser}$ The pinch point between the hot working fluid leaving the expander and the cooling fluid – Use 5°C

$PPT_{Vaporizer}$ The pinch point temperature between the saturated working fluid and the hot brine – Use 15°C

2.3.2.4 Isentropic Efficiency

The turbine efficiency and pump efficiency are required to calculate the net power of the system. The efficiency of a turbine is set to 85% while a volumetric expander can have efficiency will need to test the best and worst case scenario, 70% and 30% respectively. The actual efficiencies of the equipment is impact by the flow conditions and will change.

η_T isentropic efficiency of the turbine – Use 85%

η_{FP} isentropic efficiency of the feed pump - Use 85%

η_{VE} isentropic efficiency of a volumetric expander – Use a minimum 30% and a maximum 70%

2.3.2.5 Determine the states of the ORC

The thermodynamic properties of the fluid at each state are calculated to understand the system and design the components.

2.3.2.6 Pump Inlet State

The pump inlet is has sufficient information to determine the thermodynamic properties. Here the working fluid is a sub cooled liquid. The fluid temperature is calculated by the cooling fluid inlet temperature and approach temperature. 5°C sub cooling is required to avoid cavitation in the feed pump. Therefore, saturation conditions are 5°C above the inlet condition of the pump; this sets the pressure of the condenser and outlet of the expander.

T_4 is the pump inlet temperature °C

P_4 is the pressure at the pump inlet, which is equal to the saturation pressure at $T_4 + 5$

2.3.2.7 Evaporator Outlet Conditions

The evaporation temperature calculation is an iterative process that requires an evaporator temperature estimate and an assumption that the temperature does not change in the feed pump. This will be reassessed once all initial state points are known.

$T_5 \approx T_4$ The preheater inlet is approximated to be equal to the pump inlet temperature °C

$T_6 = T_{evap,guess}$ the evaporator temperature is guessed simple iteration is used to find optimum evaporation temperature °C

2.3.2.8 Iterative Evaporator Calculation

Figure 34 is the process followed to determine the evaporation temperature of the fluid. The guess evaporator temperature is used to estimate the heat required for the pre-heating, evaporating, and superheating the fluid processes. The geothermal pinch point temperature is then calculated from a temperature heat analysis.

$T_{evap,guess}$ sets the pressure in the evaporator $P_{evap,guess}$ this is used to calculate the amount of heat required to pre-heat h_{ph}, vaporize h_{evap},, and superheat h_{sh}, the working fluid. This whole heating process is the heat input to the system h_{in}. The corresponding geothermal pinch point temperature is calculated with the equation below.

$$h_{ph} = h_6 - h_5$$

$$h_{evap} = h_7 - h_6$$

$$h_{sh} = h_8 - h_7$$

$$T_{Pinch,geo} = T_{out,geo} + (\frac{h_{ph}}{h_{in}})(T_{in,geo} - T_{out,geo})$$

If the geothermal fluid has no restrictions on the outlet temperature 70°C is used. The optimum outlet temperature for the fluid can be determine once the states in the cycle are known.

The new evaporation temperature is 15°C less than the pinch point temperature of the geothermal fluid.

$$T_{Evap,2} = T_{Pinch,geo} - PPT_{evaporator}$$

If there is no difference between the guessed value for the evaporator and the new value from the pinch point analysis the evaporation temperature is appropriate for that geothermal condition. However, if there is a significant difference between the guess value and the new value; that new value is then used in the process again until there is no difference between the guessed and calculated evaporation temperature.

The evaporation temperature sets the high pressure of the ORC this is used to determine the temperature and enthalpy change in the pump. This might change the evaporation temperature because the pump increases the enthalpy of the fluid.

With the evaporation outlet conditions known the isentropic efficiency equation of the expander is used to set the outlet enthalpy of the expander. At this point all the states of the system are known.

The pressure of the expander outlet is assumed to be equal to the pressure of the pump inlet. The turbine efficiency equation is used to calculate the enthalpy at the expander outlet and the specific work output of the ORC.

$$\eta_e = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

h_1 is the turbine inlet enthalpy kJ/kg

h_{2s} is the turbine outlet isentropic enthalpy kJ/kg

h_2 is the actual outlet enthalpy of the expander kJ/kg

2.3.2.9 Pump Outlet

With the high pressure side of the ORC known the pump isentropic efficiency data will determine the outlet conditions of the pump. This will slightly change the evaporator conditions and this should be checked once more.

$$\eta_{FP} = \frac{h_{5s} - h_4}{h_5 - h_4}$$

h_{5s} is the isentropic enthalpy after the pump kj/kg

h_5 is the actual enthalpy of the state kJ/kg

2.3.2.10 Recuperator Approximation

If a recuperator is used in the concept then the evaporator calculations are repeated with heat added by the recuperator. The recuperator pinch temperature is the difference between the outlet of the hot fluid from the recuperator and the outlet of the cool working fluid from the pump. Assuming no heat is lost and the recuperator is 100% effective the enthalpy lost by the gas after the turbine to the fluid after the pump should be equal. However, it is common that recuperator are considered less than 100% effective with general literature suggesting that 80% effectiveness should be used when investigating recuperator benefits[82].

$T_{out,R,GAS} - PPT_{Recuporator} = T_{in,R,Fluid}$ the outlet temperature of the recuperator gases and the inlet cooling fluid is the pinch point of the recuperator . This pinch point is assumed to be 5°C[49]

$T_{out,R,Fluid} + PPT_{Recuporator} = T_{in,R,Gas}$ Alternatively the outlet of the cooling fluid is same temperature difference as the inlet of the gas.

$h_{out,R,Gas} = h_{in,R,Fluoid}$ the heat is transferred from the hot gas to the cool fluid.

This reduces the heat required in the pre-heater and increases the ORC thermal efficiency. However, a recuperator is only worthwhile if there is a strict limitation on the reinjection temperature[13].Therefore, a recuperator is most likely not economical for a low temperature ORC.

2.3.2.11Working Fluid Flow rate

Calculate the working fluid mass flow rate with the energy balance between the geothermal fluid and the working fluid. The heat transferred from the geothermal fluid is equal the heat gained by the working fluid.

$$\dot{m}_{geo}(\Delta h_{geo}) = \dot{m}_{wf}(\Delta h_{wf})$$

$\dot{m}_{geo}$ is the geothermal mass flow rate, which is known kg/s

Δh_{geo} is the specific enthalpy change of the geothermal from the outlet of the well to outlet of the ORC, which is also known kJ/kg

$\dot{m}_{wf}$ is the working fluid mass flow rate, unknown kg/s

$\Delta h_{wf}$is the change in specific enthalpy of the working fluid between the inlet of the pre-heater and the outlet of the super-heater, which is known from the ORC analysis kJ/kg

2.3.2.12Condenser Parasitic Loads

The condenser mass flow rate must be sufficient so that all the heat in the condenser is expelled from the working fluid to the condensing fluid.

2.3.2.12.1 Mass flow rates

First the condenser outlet temperature is required to calculate the mass flow rate. The pinch point and condenser inlet temperature is known. These are used in the following equation.

$$T_{out,cf} = \left(\frac{h_{out}}{h_{cond}}\right)\left(T_{Pinch,cf} - T_{in,cf}\right) + T_{in,cf}$$

$T_{in,cf}$ is the inlet temperature of the cooling fluid

$T_{Pinch,cf}$ is temperature of the air at the pinch point (assumed to be 5°C cooler than the working fluid saturation temperature)

h_{out} the total specific enthalpy rejected from the system turbine outlet to pump inlet kJ/kg

h_{cond} the enthalpy rejected from the saturated vapour state to the inlet of the pump kJ/kg (From condenser inlet to pump inlet)

With the temperature rise of the fluid across the condenser the required flow arte is calculated.

$$\dot{m}_{cw\ or\ air} = \dot{Q}_{out,wf}/(c_p \Delta T)$$

$\dot{m}_{cw\ or\ air}$ is the mass flow rate of the cooling fluid kg/s

$\dot{Q}_{out,wf} = h_{out}(\dot{m}_{wf})$ is the heat rejected by the working fluid kW

c_p is the specific heat of the cooling fluid at standard conditions 4.18 kJ/kgK for water and 1.007 kJ/kgK for air.

ΔT is the temperature raise of the cooling fluid across the condenser

2.3.2.12.2 Parasitic Loads

To calculate the parasitic load of the cooling water pump the following assumptions are made, density is 1000kg/m^3, pressure increase of 100 kPa, and pump efficiency of 70%. These assumptions are used in the following equation.

$$\dot{W}_{Cw} = \frac{\dot{m}_{cw}}{\rho_{cw}}(\Delta P_{cw})\frac{1}{\eta_{cw}}$$

$\dot{W}_{Cw}$ is the parasitic load of a water cooled condenser kW

$\dot{m}_{cw}$ is the mass flow rate of cooling water through the condenser

ρ_{cw} is the density of the cooling water – 1000kg/m^3 is assumed

ΔP_{cw} is the assumed pressure increase of the cooling water – 100 kPa is assumed

η_{cw} is the efficiency of the pumps – 70% is assumed

The parasitic load of an air cooled condenser is calculated with the same equation but using the properties of air and the fan efficiency. The assumptions for an air cooled condenser are that the air density is 1.18kg/m^3, a pressure increase of 0.15 kPa, and a fan efficiency of 70%.

$$\dot{W}_{fan} = \frac{\dot{m}_{air}}{\rho_{air}}(\Delta P_{fan})\frac{1}{\eta_{fan}}$$

$\dot{W}_{fan}$ fan parasitic loads kW

$\dot{m}_{air}$ is the mass flow rate of the air through the condenser kg/s

ρ_{air} is the density of air – 1.18kg/m^3 is assumed

ΔP_{fan} is the pressure increase by the fans - of 0.15 kPa is assumed

η_{fan} is the efficiency of the fans – 70% is assumed

2.3.2.13Gross and Net Work

With all details of the ORC known the gross and net work power outputs are calculated.

The gross work output is the total work produced by the expander.

$$\dot{W}_{gross} = \Delta h_e \dot{m}_{wf}$$

$\dot{W}_{gross}$ is the gross work in kW

Δh_e is the enthalpy drop across the expander kJ/kg

$\dot{m}_{wf}$ is the working fluid mass flow rate kg/s

A shaft and generator efficiency of 98% can also be considered at this point; therefore, the actual electrical work produced by the generator is 98% of the gross work.

The net work of the ORC is the sum of the power output and the parasitic loads to operate the ORC.

$$\dot{W}_{net} = \dot{W}_{gross} - \dot{W}_{FP} - \dot{W}_{Cond} - \dot{W}_{geo} - \dot{W}_{other}$$

$\dot{W}_{net}$ Net output of the ORC – power available for sale kW

$\dot{W}_{FP} = \Delta h_{fp} \dot{m}_{wf}$ The work necessary to operate the feed pump – kW

Δh_{fp} is the change in specific enthalpy across the feed pump kJ/kg

$\dot{W}_{Cond}$ is the work necessary to operate either the air cooled condenser or water cooled condenser kW

$\dot{W}_{geo}$ the work necessary to pump the geothermal fluid kW

$\dot{W}_{other}$ other systems in place to operate the ORC kW

2.3.2.14Results

A T-S diagram illustrates all the processes in the ORC and ensures no pinch points are violated. The T-S diagram can also help with understanding the potential for a recuperator in the cycle if there is excessive superheat after the turbine a recuperator is more feasible.

The important information for the next sections are the net work output, heat exchanger duties and temperatures, and working fluid flow rate

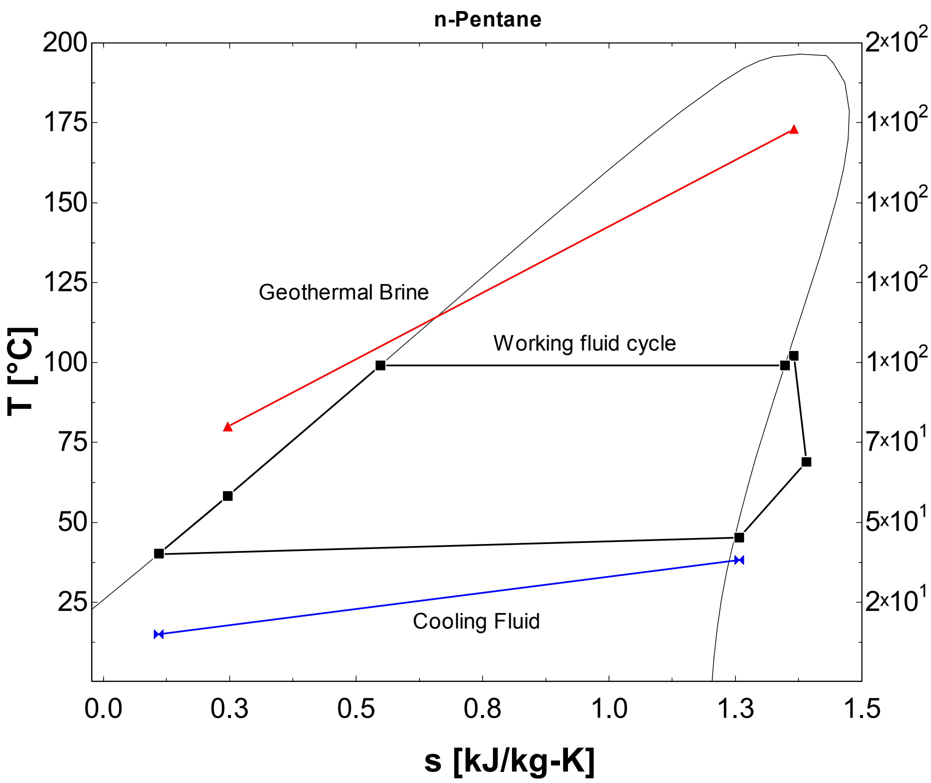


Figure 36 - General T-S diagram- Created in EES

2.3.2.15 Working Fluid

If the main difference between the concepts is the choice of working fluid then the working fluid can be changed simply in the cycle to see if there is a significant difference between the fluids for the same resource and cycle conditions.

2.4 Potential Work and Expander Recommendations

The success of a small ORC greatly depends on the expander efficiency. An expander that is not suitable for the working fluid or flow conditions will no perform at the desired efficiency.

If the potential work is greater than 1MW a turbine is the best option for the ORC[69]. A project with a potential greater than 1MW can approach large turbine manufactures for their turbine procurement and the following turbine analysis is unnecessary.

A custom designed turbine can achieve high isentropic efficiencies to maximize the power available in the low temperature heat source; however, there will be some development risk that the turbine manufacture and owner must negotiate.

Figure 37 shows the process to the efficiency of the potential expander in an ORC.

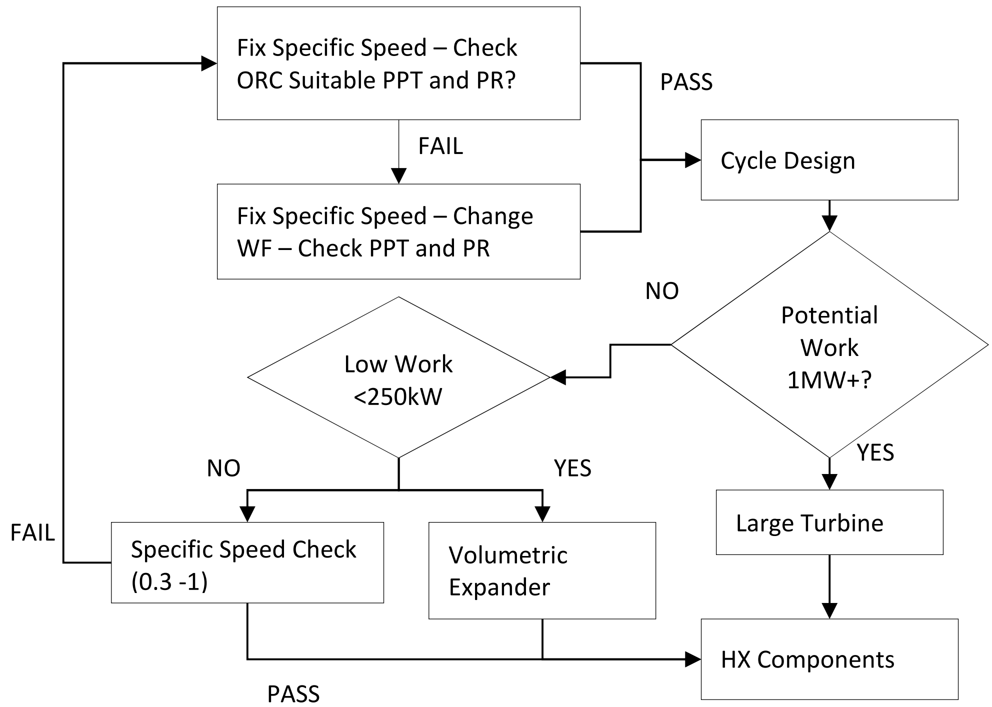


Figure 37 - Expander Consideration

2.4.1 Small Work <250kW

A low temperature resource generally has a lower work output. It has been mentioned that there are two options for a small ORC, either a volumetric expander or a small turbine. A volumetric expander should be used with ORCs that are producing up to 250kW as they are more available and economical[26]. However, if desired the feasibility of a small turbine can be investigated. The market for small turbines for power generation at the time of writing this document was small.

2.4.2 Turbine Feasibility

The best check to understand the efficiency of a turbine in the ORC is the specific speed analysis. Both radial and axial turbines achieve the best performance when the specific speed is between 0.3 and 1[83]. It is preferred that the speed of the turbine is designed to operate at the synchronous speed of the generator – 3000 rpm in New Zealand's case.

This case is for a single stage turbine.

Specific speed formula

$$N_s = \frac{\omega\sqrt{m/\rho_{01}}}{(\Delta h_0)^{3/4}}$$

N_s is the specific speed of the turbine

ω is the rotational speed of the turbine rad/s

m is the mass flow rate into the turbine kg/s

ρ_{01} is the density of vapour into the turbine m^3/kg

Δh_0 is the isentropic enthalpy drop across the turbine stage kJ/kg

The specific speed chart shows the impact on the maximum efficiency.

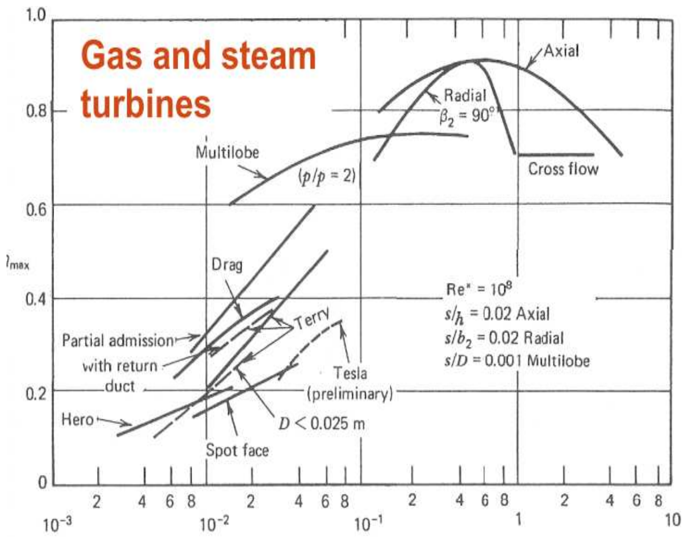


Figure 38 - Specific Speed Chart – Credit Balje [73]

2.4.3 Desired Specific Speed Check

Because of the low temperature and flow rates in the ORC the turbine is typically not within the best efficiency zone. The speed of the turbine can be changed to overcome this; however, this increases the complexity and cost of the system. Alternatively, the cycle can be adjusted to better match the optimum specific speed. Figure 39 outlines the process to follow to recalculate the specific speed and efficiency of the turbine. If matching the specific speed to the cycle results in a losing more than 30% of the original work, a more complex expander is needed. This will either operate at higher speed with a reduction gearbox or be a multiple stage turbine. This will be covered in the feasibility study and for the rest of this investigation use the 85% isentropic efficiency.

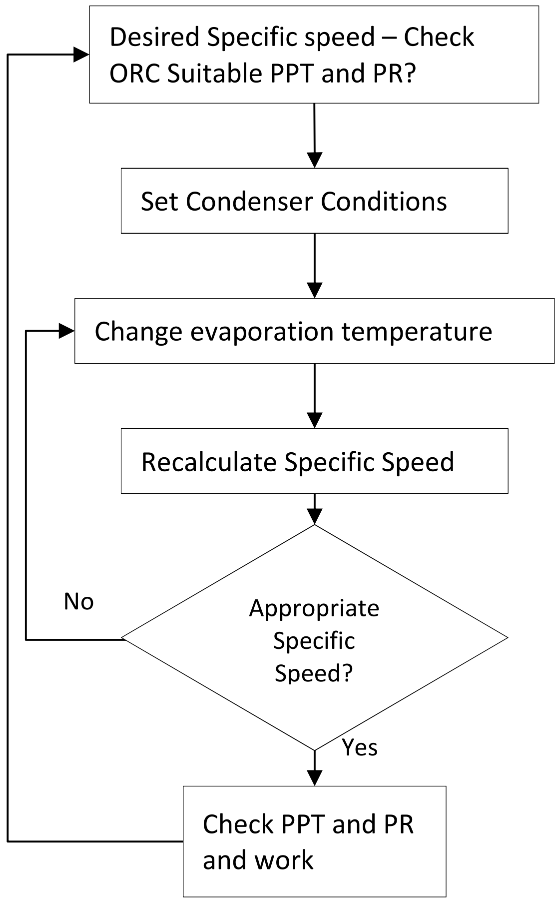


Figure 39 - Desired Specific Speed Analysis

2.4.3.1 Set Condenser Conditions

The two main variables that impact the specific speed of the turbine are the evaporation temperature and condensing temperature. These two variables impact the mass flow rate, vapour density, and enthalpy drop. The condenser conditions can be less flexible than the evaporator conditions so this is the first variable set before analysing the specific speed.

2.4.3.2 Change Evaporation Temperature

In order to increase the specific speed, normally the issue with small ORCs, it is necessary to decrease the evaporation temperature. This decreased temperature increases the volumetric flow rate of the fluid and decreases the isentropic enthalpy drop. All of these conditions contribute to an increased specific speed.

If the new specific speed is appropriate then the evaporation temperature does not need to be changed again. Once the specific speed is appropriate the pinch point temperatures and pressure ratio should be checked and are not unfeasible.

The results of the ORC should be checked once more to understand the implications of adjusting the cycle for a better specific speed.

If either the PPT is less than 5 or the PR greater than 5 it is unreasonable and the condenser temperature is changed and process redone. Finally if it appears there is no feasible single stage turbine operating at synchronous speed the working fluid can be changed. If there are no solutions this turbine will be explored in more detail in the feasibility study; otherwise, a volumetric expander is a simpler option.

The remainder of the pre-feasibility study can be completed with the new turbine and cycle or use the assumed 85% isentropic efficiency in the original cycle analysis.

2.5 ORC Component Considerations

The main heat exchangers in ORCs are the pre-heater, evaporator, super heater, recuperator, and condenser. The two types of heat exchanger that can be considered for each of these applications are plate type heat exchangers and shell and tube heat exchangers. Furthermore, an air cooled condenser should also be considered for the condenser.

Shell and tube heat exchangers are the industry norm for the pre-heater, evaporator, super heater, and recuperator. The choice between an air cooled condenser and either a shell and tube condenser depends on location and resources available to cool the working fluid.

Plate heat exchangers have been used in some geothermal applications; however, they have issues with fouling and are not used regularly for two phase applications. A plate heat exchanger is only recommended for a clean geothermal fluid.

Table 5 - Heat Exchanger Comparison Credit [84-86]

Plate Type Heat Exchangers		Shell and Tube Type Heat Exchanger	
Advantages	**Disadvantages**	**Advantages**	**Disadvantages**
Simple and Compact	Capital Cost – expensive plates	Less expensive to plate?	Less effective heat transfer
High Heat Transfer	Leak detection	Higher T and P	Cleaning is difficult
Easier Cleaning	Temperature and pressure limits	Less Pressure Drop	Cannot increase capacity
Flexible (add or take away plates)	Higher Pressure Drop	Easy leak detection	More space required
Maintenance (remove leaky plates)	Careful dismantling and assembly	Can act as receivers	Designed for a single duty
Turbulence can reduce fouling	Over tightening increase Pressure drop	Corrosion protection is easier	Require insulation
Counter flow heat transfer	Long gaskets	Any State	Vibration issues
More resistant to fouling	Narrow flow between plates		
	No appropriate for vaporizing fluids		
	Gasket free versions are impossible to open		

Table 6 - Air cooled condenser Credit - [76, 87]

Air Cooled Heat Exchanger	
Advantages	**Disadvantages**
Attractive if no cooling water available	High initial cost
High temperature processes	Large Footprint
	Higher process outlet temperatures (approach)

2.5.1 Heat Exchangers

Heat exchangers are a significant portion of the ORC cost as they require large amounts of material and skilled manufacture for high pressure operation. In typical geothermal ORCs they contribute to at least 20% of the system cost [21] and this higher in more in a small ORC. There are potentially five different heat exchangers for an ORC: a pre-heater, evaporator, super heater, recuperator, and condenser. A small ORC can design an evaporator that also superheats the fluid before it leaves the heat exchanger. There is also evidence of small ORCs having the pre-heater and evaporator within the same unit; this reduces the number of vessels and improves the overall cost of the system.

The three heat exchangers that have been used in ORCs are listed in table 5 and 6 these tables highlight the advantages and disadvantages of each heat exchanger. Shell and tube heat exchangers have normally used for geothermal ORCs as the tubes are easier to clean once fouling and they can withstand higher pressures. However, if the fluid is clean a gasket plate heat exchanger can be used and will require care to avoid leaks.

Shell and tube heat exchangers design rule of thumb pressure drop is 10PSI (68.9 kPa), while some design allows up to 1 Bar (100 kPa) to facilitate further optimization of the heat exchanger [88].

Plate heat exchangers typically have higher pressure drops compared to a shell and tube heat exchanger. The pressure drop of a plate heat exchanger increases as the volumetric flow rate increases. A general rule of thumb for a plate heat exchanger is to assume a 1 bar (100kPa) pressure drop for heat exchanger [89].

This should be the maximum pressure drop in the heat exchangers and when the detailed design is complete this value will decrease.

2.5.1.1 Air Cooled Condenser

Pressure drop in the condenser is less of a concern as the pump is directly after the condenser. The biggest concern for a condenser pressure drop is to ensure the condenser pressure does not result in a sub atmospheric condition. A sub atmospheric condenser has the risk of air leaking into the system and reducing the performance of the ORC.

An air purge system can be installed to release the NCG from the system; however, it is recommended to choose a working fluid that will not have this risk. If the atmospheric condensing pressure is significantly above atmospheric pressure the other concern are the reduced approach temperature and the reduction of sub cooling as this will increase the cavitation risk in the pump.

2.5.2 Pump

The highest efficiency feed pump is required to reduce parasitic loads in the system. Improper pump selection will result in an inefficient pump that wills reduce the power sales of the ORC. If the parasitic loads of the pump are greater than 10% of the system the pump must be changed.

The three most important considerations for a pump are the flow, required head, and efficiency. The net pressure suction head (NPSH) of the is also required because the pump operates close to the fluids saturation temperature.

2.5.2.1 Pump selection guide

The centrifugal pump is the most likely pump for the ORC. However, for very small systems it is possible that other pump choices are better.

2.5.2.2 Pump Limitations and Predictions

Table 7 and 8 highlight the limitations and recommended efficiencies estimates for the pumps considered.

Table 7 - Available Rotary Pumps – Credit [90]

Pump Type	Max Flows L/s	Corresponding Max Pressure (bar)
Gear Pump	• 10 • 12 • 15	• 12 • 8 • 5
Three Screw Pump	• 12 • 40 • 90	• 150 • 120 • 3
Two Screw Pump	• 10 • 60 • 100	• 8 • 6 • 6

Table 8 - Pump Isentropic Efficiency – Credit [90]

Pump Type	
Plunger Pump	85 – 90%
Rotary Pump	65 – 90%
Single Stage Centrifugal	60-90 (90% for unique cases at optimum conditions)

2.6 Performance Prediction

This system performance prediction takes into account losses that were previously assumed negligible, such as heat exchanger pressure loss, adjusted pump efficiency, and new turbine efficiency. This is a more accurate performance prediction of the ORC; however, there is still room for improved performance when the component design is completed.

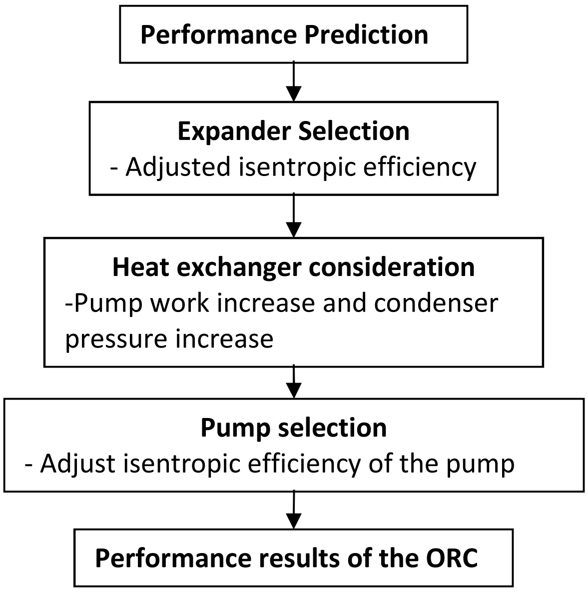


Figure 40 - Performance Prediction

2.6.1 Heat Exchanger Pressure Drop

The type of heat exchanger will impact the pressure drop of the working fluid. To overcome this pressure drop the feed pump will have to increase the pressure of working fluid further so that the working fluid still evaporates at the same design pressure. This increases the work input on the pump affecting the net power of the ORC. The evaporator pressure drop is still negligible.

Another heat exchanger area to consider is the pressure of the condenser. If it is desirable to operate the condenser above atmospheric pressure to maintain a positive pressure at all stages in the ORC this will influence the condensing temperature and the performance of the expander. The allowable pressure drop in the condenser should ensure the pressure in the condenser does not become sub atmospheric ideally. This could result in a higher pressure condenser that will impact the turbine performance.

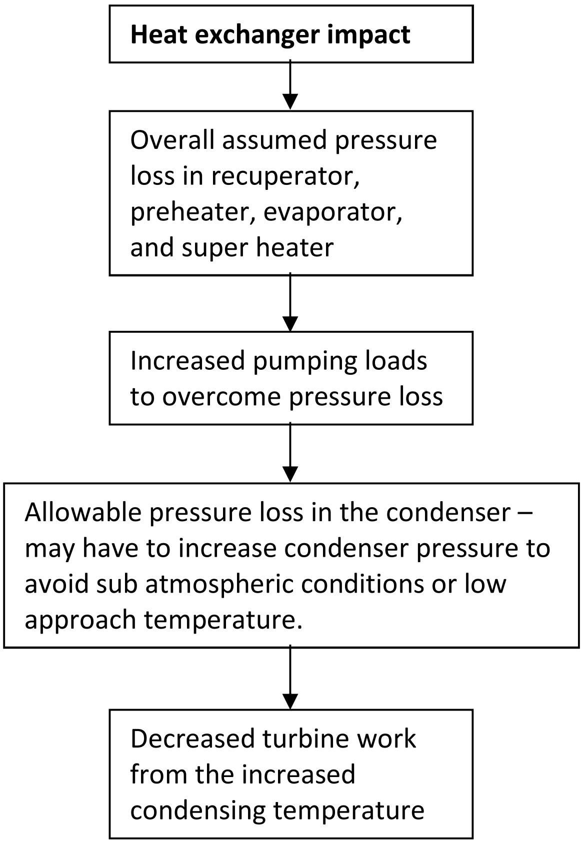


Figure 41 - Heat Excnhagner impact on performance

2.6.2 New Efficiencies considerations

The expander analysis will improve the accuracy of the isentropic efficiency assumption. If the specific speed analysis determined a new turbine efficiency this is used instead. Otherwise the 85% efficiency still stands.

The pump choice for the system will also affect the system performance as different types of pumps have different assumed efficiencies.

2.6.3 Efficiencies

The thermal and exergetic efficiencies are calculated to understand how well the geothermal resource has being utilized [19]. Each of the following efficiencies are used to characterise the ORC performance. Thermal efficiency is the most used; however, the utilization and functional efficiencies are a better way of understanding how the geothermal ORC is designed.

2.6.3.1 Thermal Efficiency

Thermal efficiency is the ratio of the net work produced by the system and the amount of heat supplied to the system.

$$\eta_{thermal} = \frac{W_{net}}{Q_{in}}$$

$\eta_{thermal}$ Thermal efficeicny

W_{net} net work output of the ORC kW

Q_{in} heat supplied to the ORC kW

2.6.3.2 Utilization Efficiency

The utilization efficiency uses the second law to determine the ORC efficiency. The utilization efficiency is the ratio of the net work output and the maximum theoretical power obtainable from the geothermal fluid in its reservoir state. [12]

$$\eta_u = \frac{W_{net}}{m_{geo}[h_{res} - h_0 - T_0(s_{res} - s_0)]}$$

η_u Utilizational efficiency

m_{geo} geothermal mass flow rate kg/s

h_{res} geothermal reservoir enthalpy kJ/kg

h_0 dead state enthalpy kJ/kg

T_0 Dead state temperature K (ambient temp)

s_{res} entropy of the reservoir kJ/kgK

s_0 entropy of the dead state kJ/kgK

2.6.3.3 Functional Efficiency

The functional efficiency is the ratio of the net work out of the system and the exergy available for the ORC.[12]

$\eta_f = \frac{\dot{W}_e}{\dot{m}_{geo}(h_{res} - h_{out,geo} - T_0[s_{res} - s_{out,geo}])}$ η_f Functional Efficiency

$h_{out,geo}$ The enthalpy of the geothermal fluid at the ORC outlet

$s_{out,geo}$ the entropy of the geothermal fluid at the ORC outlet

2.7 Costs

The deciding factor for the concept selection will be the overall cost opposed to performance. The following cost section estimates the cost of each component. The costing estimate uses general cost per unit of the component for a large system. However, for the smaller systems cost equations are used as these systems typically do not follow the same cost curves as large systems. Furthermore, a trade-off optimization will be carried out in the feasibility study to optimize the cost of the ORC.

These costs are all representative because costs change over time and it is likely at the point of writing this document the costs of these components would have changed.

2.7.1 Component Costs

The component costs are adapted from Gerrard [91] and Couper [92]. Component cost estimates are for the installed price.

2.7.1.1 Heat Exchangers

The basic requirement for a heat transfer cost estimate is the rough surface area required. The heat transfer equation below is used to estimate the rough size of heat exchangers, no LMTD correction factors are considered at this point of the study.

$Q = U(LMTD)A$

Q is the heat transfer required for the heat exchanger W (ensure that this value is in W not kW)

U is the overall heat transfer coefficient W/m^{2}K

LMTD is the log mean temperature difference K

A is the heat transfer area m^2

The LMTD is the average temperature between the two fluids in the heat exchangers. The following equation is used to calculate the LMTD between the pre-heater, super-heater, and recuperator.

$$LMTD = \frac{\Delta T_A - \Delta T_B}{\ln(\frac{\Delta T_A}{\Delta T_B})}$$

$$\Delta T_A = Inlet\ Fluid\ A - Outlet\ Fluid\ B$$

$$\Delta T_B = Inlet\ Fluid\ B - Outlet\ Fluid\ A$$

For the condenser or evaporator a slightly different LMTD is used to account for the phase change [93].

$$LMTD_{evap} = \frac{T_{Geo,in} - T_{Geo,out}}{\ln(\frac{T_{Geo,in} - T_{evap}}{T_{Geo,out} - T_{evap}})}$$

$$LMTD_{cond} = \frac{T_{cool,out} - T_{cool,in}}{\ln(\frac{T_{cond} - T_{cool,in}}{T_{Cond} - T_{cool,out}})}$$

2.7.1.1.1 Shell and Tube Heat Exchanger

Shell and tube heat exchangers are commonly used with geothermal ORCs for pre-heaters, evaporators, superheaters, and water cooled condensers. A shell and tube heat exchanger can withstand high pressures commonly found in geothermal systems and the tubes can be cleaned with water blasting or chemical cleaning.

Common overall heat transfer coefficients used for shell and tube heat exchanger calculations are shown in table 9.

Table 9 - Common U values for Shell and Tube Heat Exchangers – From [24, 71, 94]

Use	**Tube Side**	**Shell**	**Overall Heat Transfer Coefficient W/m^{2}K**
Pre-Heating 6-7	Liquid	Liquid	150-1200
Evaporator 7-8	Liquid	Vaporizing fluid	600-1500
Super Heating 8-9 / Recuperator 5-6	Liquid	Gas	100-300
Condensing 3-4	Water	Condensing Vapour	300-1200

Costing formula

$C_{S\&T}\ (Q2\ 2014\ NZD) = 2140\ \times A^{0.578}\ \times\ m_f\ \times I_f$ $4\text{m}^2 < \text{A} < 900\ \text{m}^2$

If the shell and tube heat exchanger requires more than 900m^2 the basic linear cost of 500NZD for a preheater and 620NZD for an evaporator or recuperator per m^2.

The purchase cost of a heat exchanger depends largely on the material it contains. In order to account for the material used, multiply the result by a material factor. An illustrative list of material factors is below.

Table 10 - Cost Factors – From [91, 92]

Shell material	**Tube Material**	**Factor m_f**
Carbon Steel	Carbon Steel	1.0
Carbon Steel	316 Stainless	2.2
Nickel 200	Nickel 200	5.5
Inconel	Inconel	4.2
Titanium	Titanium	4.0

The installation cost factor I_f for a shell and tube heat exchanger is typically between 1.4 – 2. Larger units or units with less costly materials tend to have installation factors at the lower end of that range.

2.7.1.1.2 Plate Heat Exchanger

Plate heat exchangers have the potential to be a very cost competitive solution for a low temperature geothermal ORC. However; their use with geothermal brine is a high risk because they are more susceptible to fouling, corrosion, and blockage. A robust understanding of the geochemistry of the fluid and scaling potential should be known before a plate heat exchanger is installed with an ORC.

The plate heat exchangers for the ORC should also be gasket type heat exchangers to allow for cleaning of the plates. However; gasket heat exchanger use with phase change operations is an issue as they are more likely to leak and the uncertainty between the working fluid at the gasket material.

The typical overall heat transfer coefficients used for various plate heat exchanger operations is given in table 11.

Table 11 - Typical Overall Heat transfer Coefficients for Plate Heat Exchangers – Credit [95-97]

Use	**Plate**	**Plate**	**Overall Heat Transfer Coefficient W/m²K**
Pre-Heating 6-7	Liquid	Liquid	1000-4000
Evaporator 7-8	Liquid	Vaporizing fluid	1000-4000
Super Heating 8-9 / Recuperator 5-6	Liquid	Gas	70-800
Condensing 3-4	Water	Condensing Vapour	900-1200

$C_{PL}\ (Q2\ 2014\ NZD) = 3370\ \times A^{0.489}\ \times I_f$ A < 20 m², 316 Stainless Steel

$C_{PL}\ (Q2\ 2014\ NZD) = 5140\ \times A^{0.463}\ \times I_f$ A < 20 m², Titanium

$C_{PL}\ (Q2\ 2014\ NZD) = 1140\ \times A^{0.691}\ \times I_f$ A > 20 m², 316 Stainless Steel

$C_{PL}\ (Q2\ 2014\ NZD) = 1100\ \times A^{0.751}\ \times I_f$ A > 20 m², Titanium

These costing equations are for pressures below 12 bar. Plate heat exchangers can operate up to 25 bar but it is more likely that it will cost significantly more. The installation cost factor is typically between 1.5 – 2.0.

Air Cooled Condenser

Air cooled condensers are the standard condenser for ORC that do not have a readily available source of cooling water; generally a water cooled condenser is cheaper.

Table 12 - Typical Overall Heat transfer Coefficient for an Air Cooled Condenser – Credit [71]

Use	Outside	Tube side	Overall Heat Transfer Coefficient W/m^2K
Condenser	Air	Condensing Vapour	350-500

$$C_{ACC}\ (Q2\ 2014\ NZD) = 10950\ \times A^{0.40}\ \times I_f$$

The installation cost factor is typically around 2.5 for air cooled condensers.

2.7.1.2 Expander and Generator

The two different types of expander are explored in this cost estimation.

2.7.1.2.1 Turbines

The cost estimates for this component vary greatly especially at smaller sizes, as the market for ORC turbines is very young. A general rule for a rough price estimates is to take a standard turbine price estimate and multiply it by 1.5 [44]. This can be considered an optimistic price estimate for favourable flow conditions. Turbine analysis will quickly indicate whether a turbine above this price estimate may be necessary. The following equation has been made for an initial price estimate for ORC turbines up to 4 MW. Above 4MW a linear relationship of $750/kW is commonly assumed [24].

$$C_{Tu}\ (Q2\ 2014\ NZD) = 1360\ \times P^{0.81} \times I_f \qquad 15\text{ kW} < P < 4000\text{ kW}$$

The installation factor for turbine is typically 1.5.

2.7.1.2.2 Volumetric Expanders

The market for volumetric expanders is still developing and there are limited examples of ORCs using volumetric expanders for commercial operations. The cost of these expanders is not well known and an estimate from experience is $2000/kW for a volumetric expander, which comes from specific costs for comparable compressors.

The turbine appears to be a cheaper option for small expanders at this point; however, there is not market for small turbines in ORCs and the cost of a custom designed turbine would significantly increase this.

2.7.1.2.3 Generators

In general, the generator price is small compared to the other major components, and is roughly accounted for by the electrical installation factor. For small systems however, the generator cost is more significant. For generators up to 100 kW, the following equation can be used to find the generator price.

$$C_G\ (Q2\ 2014\ NZD) = 225\ \times P\ (kW)\ + 875 \qquad \text{P < 100 kW}$$

2.7.1.3 Pumps

The cost of installing pumps is relatively insignificant for large plants if low cost materials are used. For smaller plants, the following equation can be used as a price estimate for centrifugal pumps.

$$C_{Pu}\ (Q2\ 2014\ NZD) = m_f\ \times\ I_f\ (450 \times V + 2236) \qquad 0.3\ \text{l/s} < V < 6\ \text{l/s}$$

The material and installation factor vary greatly with the material required. Some examples are listed in the table 13 below.

Table 13 - Pump Material Cost Factors – From [91, 92]

Material	Installation Factor	Material Factor
Cast Iron	2.0	1
Cast Iron with SS fittings	1.9	1.15
Carbon Steel	1.9	1.35
Stainless Steel	1.5	2.00
Nickel	1.35	3.50
Monel	1.35	3.30
Titanium	1.2	9.70

Large pumping requirements should use the linear relationship of $500/kW. While positive displacement pumps are similar costs per kW as the volumetric expanders, which is $2000/kW

2.7.2 Annual Costs

2.7.2.1 Operating Costs

A full time operator is unlikely for any development below 50MW [74]. However, it is possible that a small ORC can be remotely operated by the manufacture or by the owner in a central control room overlooking a number of ORCs. A small ORC will need to be automated with minimal operator reliance and have welled designed fail safes.

2.7.2.2 Maintenance Costs

The maintenance cost is also highly variable. Experience from refrigeration chiller systems and existing ORCs indicate an average maintenance cost of between 0.5% - 7% of the initial investment outlay annually. Large systems tend to have a proportionally smaller maintenance cost than small systems. For the purposes of initial estimation table 14 can be used.

Table 14 - Average maintenance costs for power plants

Size (MW)	Annual maintenance cost as % of capital cost
0.05 – 0.25	7
0.25 – 0.5	6
0.5 – 0.75	5
0.75 – 1.0	4
1.0 – 5.0	3
5.0 – 10	2
10 – 20	1
20+	0.5

2.7.3 Estimating Return

The return depends heavily on the situation in which the unit is to be employed. The following formula can be used to provide a rough estimation.

$$R_t = 8766 \times P_{net} \times P_r \times C$$

Where;

R_t is the annual financial benefit from the plant in dollars.

8766 is the average number of hours in a year.

P_{net} is the estimated plant net generation capacity

P_r is the current average price of electricity paid for / sold.

C is the expected capacity factor of the plant. 0.92 is commonly used from experience, but this may be lower for higher rick resources.

The capacity factor for the first year of operation is usually assumed to be 0.5 as the system is regular checked and the geothermal discharge must also be checked. After the first year of operation with no issues then the capacity factor is assumed to be 0.92.

2.7.4 Pay Back Period

If real electricity prices are assumed to remain stagnant for the payback period of the project, then the following formula can be used to estimate the discounted payback period. The discounted payback period is a more realistic payback check.

2.7.4.1 Simple Payback Period (discount rate of 0)

$$PP = \frac{C}{R_t}$$

Where;

C is the initial investment outlay

R_t is the net annual cash flow after the investment outlay, assumed to be constant.

2.7.4.2 Discounted Payback Period (Geometric Series)

$$DPP = \frac{Ln\left(\frac{1}{1 - C \times \frac{i}{R_t}}\right)}{ln(1 + i)}$$

R_t is the net cash inflow after the initial investment (Assumed to have the same simple value each year)

C is the Initial investment outlay.

t is the time of the cash flow (in years)

i the chosen discount rate. Electricity price inflation can be accounted for if desired by taking away the inflation rate from the discount rate. A rule of thumb in industry is 10%; however, new values for renewable energy have been considered (6.6 USFED) with -0.4% considered for projects between (years 1-10) and then after that 1.1% (10-30).

2.8 Concept Selection

At this stage the number of concepts should be reduced before progressing further to the feasibility study. The more concepts that are taken further into the feasibility stage will increase the costs of the feasibility and time it takes to complete the feasibility study.

2.9 Data Sheets

The data sheets highlights the key information collected during the pre-feasibility study.

<u>**ORC Data Sheet Pre-Feasibility**</u>	**Concept**				
Geothermal Data	1	2	3	4	5
Geothermal Inlet Temperature (°C)					
Geothermal Outlet Temperature (°C)					
Mass Flow Rate (kg/s)					

Working Fluid Choice					
Cycle Components (Yes-No)					
Pre-Heater					
Evaporator					
Super-Heater					
Recuperator					
Condenser					
Turbine					
Feed Pump					

Design Conditions					
Pinch Point (°C)					
Approach Temperature (°C)					
Ambient Temperature (°C)					
Turbine Efficiency (%)					
Pump Efficiency (%)					

Cycle States T/P	°C / Bar	°C / Bar	°C / Bar	°C / Bar	°C / Bar
Before Turbine 1	/	/	/	/	/
After Turbine 2	/	/	/	/	/
After Recuperator 3	/	/	/	/	/
After Condenser 4	/	/	/	/	/
After Pump 5	/	/	/	/	/
After Recuperator Liq 6	/	/	/	/	/
After Pre Heater 7	/	/	/	/	/
After Evaporator 8	/	/	/	/	/
WF Flow Rate kg/s					
Net Power (kW)					

Expander - Ideal ORC					
Type					
Isentropic Efficiency (%)					
Required NS					
Turbine Speed (RPM)					
Power Output (kW)					

Expander - Feasible Turbine					
Isentropic Efficiency (%)					
Required NS					
Turbine Speed (RPM)					
Turbine Output (kW)					
Net Power (kW)					
Evaporator PPT (°C)					
PR					
Turbine Choice for continued exploration					

Efficiencies					
Thermal Efficiency					
Overall Exegetic Efficiency					

Heat Exchangers					

Condenser Conditions					
Type					
Ambient Temperature (°C)					
Maximum Allowable Raise (°C)					
Approach Temperature (°C)					
Pinch Point temperature (°C)					
Outlet Air Temp (°C)					
Required Air Flow Rate (kg/s)					
Allowable Pressure Drop - to Atmosphere (Bar)					
Cond Temperature (°C)					
Fan/Pump power requirements (kW)					

Recuperator					
Type					
Cold Liquid Inlet (°C)					
Cold Liquid Outlet (°C)					
Allowable Pressure Drop (Bar)					
Hot Vapour Inlet (°C)					
Hot Vapour Outlet (°C)					
Pinch Point Temperature (°C)					
Allowable Pressure Drop (Bar)					

Pre Heater					
Type					
Geothermal Inlet (°C)					
Geothermal Outlet (°C)					
Working Fluid Inlet (°C)					
Working Fluid Outlet (°C)					
Pinch Point Temperature (°C)					
Allowable Pressure Drop (Bar)					

Evaporator					
Type					
Geothermal Inlet (°C)					
Geothermal Outlet (°C)					
Working Fluid Inlet (°C)					
Working Fluid Outlet (°C)					
Pinch Point Temperature (°C)					
Allowable Pressure Drop (Bar)					

Super Heater					
Type					
Geothermal Inlet (°C)					
Geothermal Outlet (°C)					
Working Fluid Inlet (°C)					
Working Fluid Outlet (°C)					
Pinch Point Temperature (°C)					
Allowable Pressure Drop (Bar)					

Pump considerations					
Increased Pumping Pressure (Bar)					
Pump work (kW)					
Overall Net Power (kW)					
Volumetric Flow Rate (L/s)					
Basic NPSHA (Bar)					

Costs					
Heat Exchangers					
Pre - Heater					
Q (kW)					
U (kW/m^2K)					
LMTD (K)					
A (m^2)					
Cost ($)					

Evaporator					
Q (kW)					
U (kW/m^2K)					
LMTD (K)					
A (m^2)					
Cost ($)					

Super-Heater					
Q (kW)					
U (kW/m^2K)					
LMTD (K)					
A (m^2)					
Cost ($)					

Recuperator					
Q (kW)					
U (kW/m^2K)					
LMTD (K)					
A (m^2)					
Cost ($)					

Condenser					
Q (kW)					
U (kW/m^2K)					
LMTD (K)					
A (m^2)					
Cost ($)					

Expander and Generator					
Turbine (kW)					
Generator (kW)					

Pump					
Capital Cost Equipment ($)					
Overhead and other components costs ($)					
Total Capital of ORC ($)					

Operation and Maintenance					
Estimated Operation cost					
Insurance					
Annual Maintenance Cost					

Estimated Return					
Sale Price					
Net Output					
Capacity factor					
Return					
Net Return					

Simple Payback					
Years +- a year					
Discount Rate %					
Discounted payback period					

Ranking of most likely concepts					

Feasibility

Introduction – Feasibility

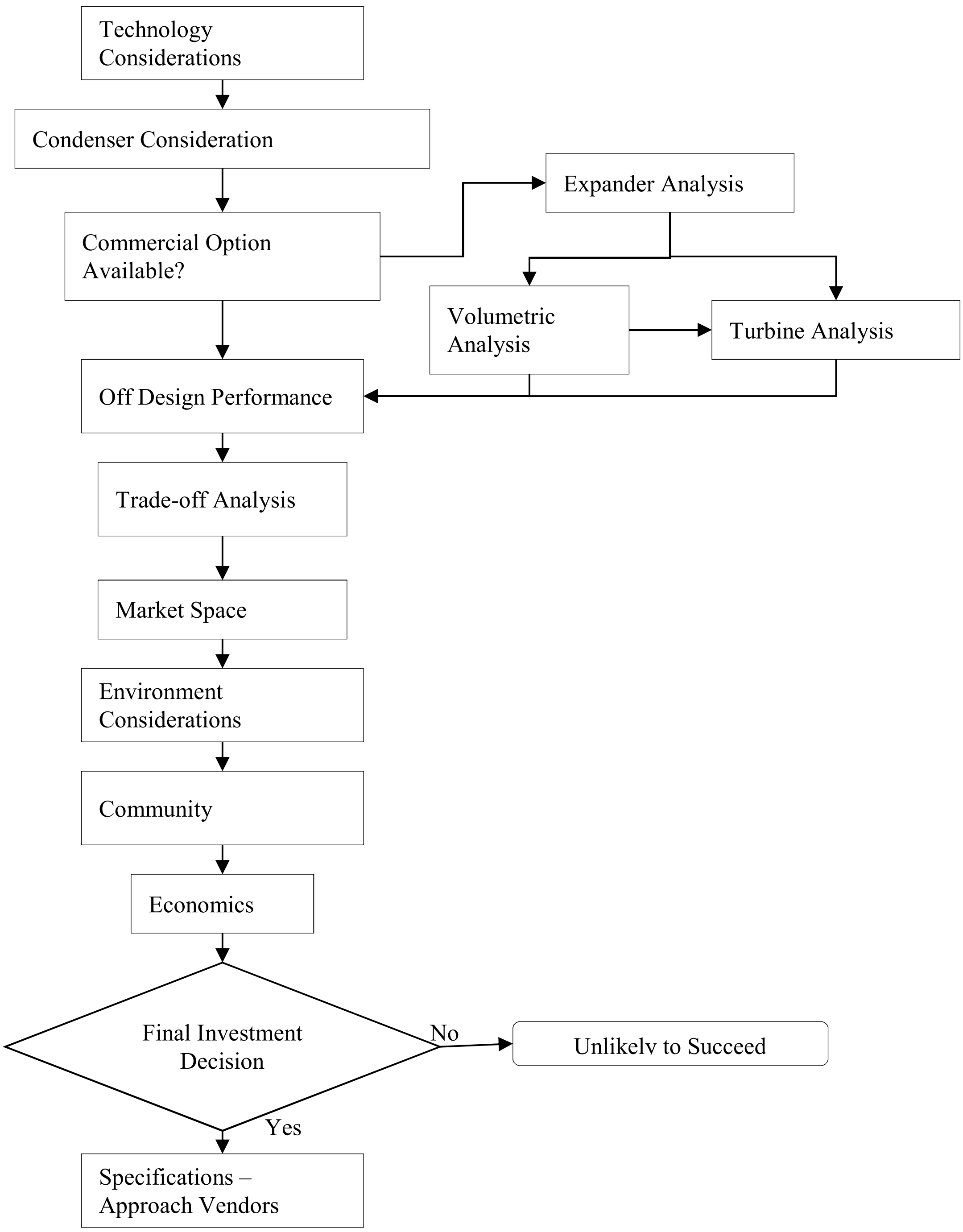
The feasibility study is the final section before the detailed design it will finish the thermo cycle design. The feasibility study also explores the risks associated with a geothermal ORC as well as final costs and rate of return for the development. Once this is all complete the final investment decision is made to do the detail design or stop the project. Specification sheets are supplied if the owner wants to approach component vendors instead of doing the detailed design themselves once the final investment decision is made.

The intended audience for this section must have a technical background in engineering or thermodynamics. The outcome of this section will supply the owner with the information needed to make the final investment decision on the project.

Feasibility Overview

1. Technology Considerations – This ensures that the current cycle configuration is the best possible solution because only the basic and recuperative cycles were looked into for the pre-feasibility.
2. Condenser consideration – The design condenser temperature will impact the ORC performance. This section guides the engineer through the benefits and drawbacks of different condenser conditions.
3. Commercial Options – With a better understanding of the potential ORC commercial options are reconsidered.
4. Expander Analysis – The expander analysis is revised to understand the complexity of the potential turbine. A multistage custom turbine will have more development risk than a commercial option.
    a. A volumetric expander and turbine analysis are provided
5. Off Design Performance – A geothermal resource can change throughout the projects life. It is important to understand the implications of these changes on the ORC design.
6. Trade-off analysis – The thermodynamic analysis to this point has used rules of thumb and common parameters to model the cycle. This section provides potential changes in the cycle design to either optimize the design from either an economic perspective or a thermodynamic perspective
7. The market space, Environmental, and Community sections are all cover certain risks associated with the ORC development
8. Economics – The capital cost, net present value, and internal rate of return of the final design are calculated. This is the best information to understand the economic performance of the ORC.
9. The outcome of this section is a final investment decision on the ORC and is continued onto the detailed design.

Feasiblity Process



3 Feasibility

3.1 Scope

The feasibility study looks into the technical, economic, environmental, and social feasibility[64] of the low temperature geothermal ORC.

The feasibility study is a continuation of the pre-feasibility section and at this point the user must have a good understanding of the potential geothermal ORC. A number of basic concepts would have been investigated in the pre-feasibility study constrained by rule of thumb assumptions and common practice. This section will continue the investigation and explore other possible alternatives not considered in the pre-feasibility stage.

The outcome of the feasibility study is the final investment decision, at which point the owners will have a good understanding of the ORC feasibility.

The feasibility study also explores the other risks and hurdles associated with a geothermal ORC, such as resource consent, market outlook, power sale agreement, environmental considerations, and local opinion [3].

Once the final investment decision is made the final section provides specification sheets for the major components of the ORC to approach vendors.

3.2 Technology Considerations

The technology consideration provides alternative solutions not explored in the pre-feasibility study, mainly more complex ORC configurations. An ORC can be configured in a number of different ways to maximize the energy recovered from a geothermal resource. The pre-feasibility used either the basic or recuperative ORC, which are the most common configurations for ORCs, other options can improve the performance of the ORC; however, they will increase the capital cost.

3.2.1 Alternative Working Fluids

Working fluid selection is considered the most critical design step for an ORC [25] and it has been explored in the pre-feasibly study but should be relooked in the feasibility study. The working fluid is the most critical aspect of the ORC as it impacts each aspect of the design, performance, and cost. Commonly used working fluids from ORCs operating around the world have been suggested in the pre-feasibility study; however, there is the possibility other working fluids are cheaper or more available to the developer. Therefore, these should also be considered for the ORC.

3.2.1.1 Recommended Fluid Properties

There are a significant number of potential working fluids that can be used for an ORC and selecting an appropriate fluid can be time consuming. The following is a list of recommended working fluid properties compiled from a literature published by Quoilin [26], Chen[27], Maizza[28, 80], and Zhai [29].

- Recommended Fluid Properties
    - High Latent Heat of vaporization Δh_{vap}
    - Low specific volume of vapour (high vapour density) ρ_{vap}
    - Low Viscosity μ
    - Reasonably low operating pressures P_{max}, P_{min}
    - High thermal conductivity k
    - Good stability
    - Minimal environmental impact
    - Non Flammable
    - Dry or isentropic
        - Wet fluids pose a greater risk to turbine erosion
        - The saturation dome is indication of this, figures 42, 43, and 44 show the difference between fluid saturation domes.
    - Leaks easily detectable.

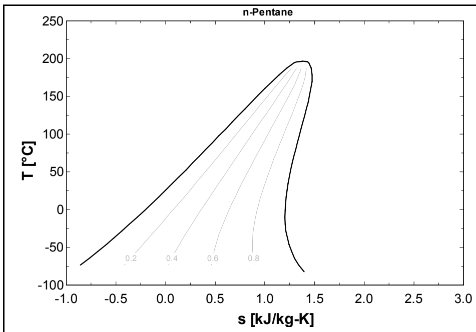


Figure 42 - Dry Working fluid - Created in EES

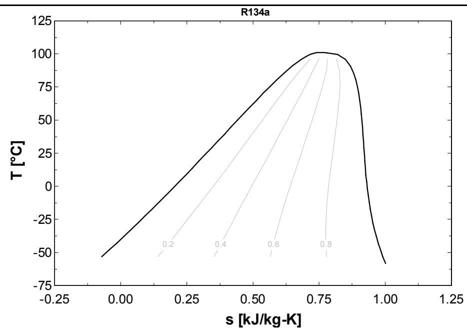


Figure 43 - Isentropic Working Fluid - Created in EES

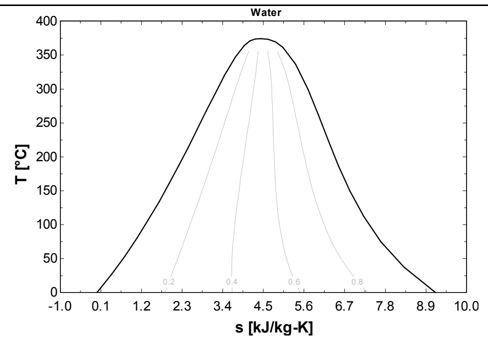


Figure 44 - Wet Working Fluid - Created in EES

3.2.1.2 Zeotropic Fluid Mixture

Another alternative for the working fluid in an ORC is a zeotorpic fluid mixture. This is a working fluid mixture that has a different composition of vapour and liquid during phase change. The difference between the saturation temperatures in the mixture results in a temperature glide during vaporization and condensation. The temperature glide improves the temperature match between the fluids thus improving ORC performance[47]. At the point of writing this document there was limited literature regarding zeotropic mixtures in actual ORCs. Angelino [38] discussed the potential benefits of zeotropic working fluid for ORCs and there have been a number of studies by X. Wang [39], Heberle [42], Chys [41], and J.L Wang[40] that all conclude that zetropic mixtures can improve the thermal efficiency of ORCs; however, actual use of these fluids in ORCs is still an unexplored area.

3.2.2 Cycle Alternatives

At this point in the ORC development the only cycles investigated have been the basic configuration and the potential of a recuperator. The other alternatives that have been used with successful ORC developments have been a Dual in-line cycle and a Dual fluid cycle. The other less common ORC are the super critical ORC and a flash evaporation ORC. There is literature regarding the potential performance for these ORCs; however, there are few examples of their application for power generation.

3.2.2.1 3.1.2.1 Dual in-line Cycle

The Dual in-line cycle consists of two ORCs running in series at two different design points off the same geothermal fluid stream. This arrangement maximizes the potential heat extracted from the geothermal fluid as the second ORC can extract more heat from the fluid at a low evaporation temperature. This arrangement is not recommended if the geothermal fluid has a critical reinjection temperature that risks the performance of the ORC heat exchangers.

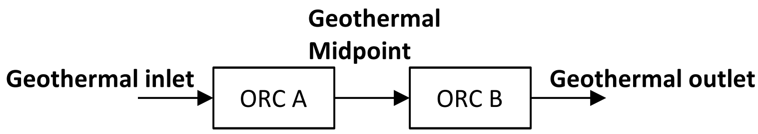


Figure 45 - Dual In-line ORC

3.2.2.2 Dual Fluid Cycle

The Dual fluid cycle separates the geothermal fluid after the evaporator in the primary ORC. At this point half goes to the heat exchangers for the secondary ORC and the other half is used in the pre-heater of the primary ORC. The other configuration possible is to use the geothermal fluid to vaporize the fluid in the primary heat exchanger and then the secondary heat exchanger before separating the fluid to the pre-heaters of each cycle. These ORCs can also use the exhaust fluid of the primary ORC for some heating in the secondary ORC.[12]

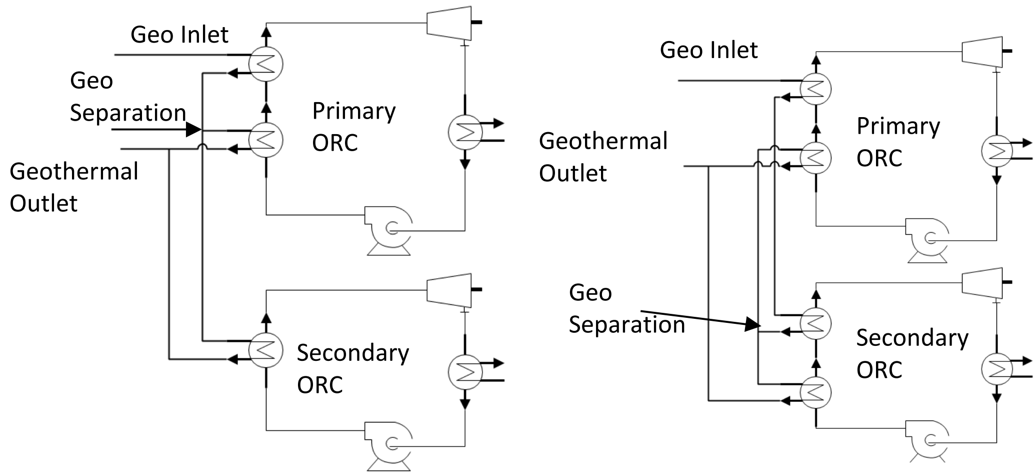


Figure 46 - Examples of Duel Fluid Cycles

3.2.2.3 Geothermal Combined Cycle

Geothermal combined cycles are commonly used for high temperature resources where there is a significant amount of steam and brine. The high pressure steam is first expanded in a back pressure turbine before being used in an ORC along with the separated brine. This configuration is appropriate to a development that plans to expand the steam filed in stages [98].

3.2.2.4 Dual Pressure ORC

A dual pressure ORC is similar to the Dual fluid ORC. The fluid passes through the first heat exchanger then some fluid is pumped to a higher pressure to be used in a high pressure turbine while the other low pressure fluid is used in a lower pressure turbine. Both turbines exhaust the fluid at the same pressure to the condenser [12].

3.2.2.5 Super Critical ORC

The super critical cycle has been discussed in a number of papers. Schuster [47] and Lai [99] investigate the working fluid properties important for a supercritical ORC; however, there is still little experience with super critical ORCs operating in the industry. A super critical ORC pressurises the fluid pass the critical point of the fluid. The fluid is then heated above the vapour dome into the supercritical vapour zone. This process allows for better temperature matching with the heating fluid as there is no phase change pinch point. However; it also requires the heat exchangers to be designed to a higher pressure and the working fluid feed pump often has high parasitic loads to meet the required pressure increase. The working fluids must also be thermally stable at high temperatures. This cycle is not recommended at the point of writing this document.

The current supercritical ORCs advertised requires a fluid temperature over 300°C, which is not in the bounds of a low temperature geothermal ORC [100].

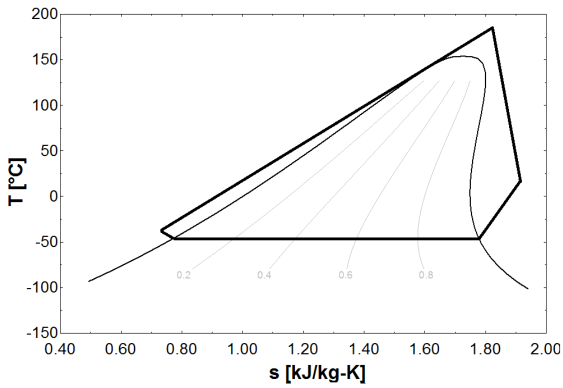


Figure 47 - Super Critical ORC - Created in EES

3.1.2.5 Flash ORC

There has been research recently into an ORC flash evaporator system. The ORC cycle heats the fluid to a saturated liquid state and then is expander into a lower pressure flash vessel where the fluid and vapour are separated[19]. The separated vapour is then used to run the ORC and the fluid can either be flashed another time or added to an intermediate pressure line to be mixed with the cool fluid from the feed pump. Once again this is new unexplored technology and at the point of writing this document is not recommended. There are parallels between this configuration and the Kalina cycle.

3.2.3 Condenser Consideration

The condenser temperature greatly impacts the performance and cost of the ORC [21]. There are two types of condenser used in ORCs, a design point condenser and a floating condenser[81]. A design point condenser operates at a condenser design point regardless of the ambient conditions This condenser simply adjusts the cooling fluid flow rate to maintain the same condensing condition and maintaining the optimum efficiency and work output for the turbine.

A floating condenser follows the ambient temperature and the fluctuations in the cooling fluid medium. This results in a higher work output from the ORC on average. A floating condenser is more effective with a dry cooling system because dry bulb temperature fluctuates more than the wet bulb temperature. A floating condenser maintains the 100% cooling fluid flow rate regardless of the ambient temperature conditions.

3.2.3.1 Condenser Analysis

To investigate the impact of a floating condenser a coefficient of performance ratio is used to compare the design point condenser to the average condensing condition.

The design point condenser is designed to use the 5% upper temperature range of the site and is guaranteed to produce this minimum power output throughout the year.

The average condenser condition is the power output of the same ORC operating at the same turbine inlet condition as the design point but now the condenser pressure is set by the average ambient temperature.

$$COP = \frac{W_{net,Average}}{W_{net,Design}}$$

COP is the Coefficient of Performance

$W_{net,Average}$ Is the average net power output of the system in kW

$W_{net,Design}$ Is the design power output of the system at the design temperature in kW

The coefficient of performances gives an idea how much power can be produced on average as the temperatures fluctuate.

A system with a floating condenser is designed to the design point but the turbine is designed to the average condition as the efficiency will also fluctuate as the condensing condition fluctuates. This ambient temperature matching does add a little more complexity to the system but will improve the ORC performance.

3.2.3.1.1 Humidity Implications

Depending on the location of the condenser the humidity also impacts the condenser. The 5 – 95% humidity range should be considered as well as the mean.

3.2.3.2 Cost Implication

Another consideration for floating condenser design is the cost savings and power reduction if the condenser is designed to operate at the average condenser condition[81].

The plant reduction cost

$$\sigma = \left[1 - \frac{1}{COP}\right] * 100$$

σ Plant reduction cost %

COP coefficient of performance

Designing the plant to this condition will reduce the cost by the given amount. However; this will also result in a power reduction equivalent to percentage gained according to the COP test.

Example:

COP of theoretical power plant is

W_{design} = 442 kW

W_{ave} = 493 kW

COP = 1.11

11% power increase if designing for the design with a floating condenser

Possible cost saving if designing for a floating condenser at ambient conditions; however, this will also result in an 11% power reduction.

σ = 9.1%

3.2.4 Cooling Geothermal Resource

The geothermal resource will change over the course of the ORCs life the of it extent is only known with a good understanding of the reservoir that supplies the geothermal resource.[3]

Exploring how the system behaves as the geothermal temperature decreases highlights what components are most affected by the resource temperature. ORCs are flexible and the components can be overdesigned to compensate for this or a heat exchanger can be added at some point in project life.

The two scenarios for a cooling resource are designing the plant to operate at the cool conditions and examining if those operating points are kept constant for a plant operating off the higher resource. Second is to see the implication of trying to maintain the same plant design point for the cooler resource.

The areas of concern are the net work output, the heat exchanger area, and the pinch point between the geothermal fluid and the working fluid.

3.3 Commercial Option Reassessment

At this point in the ORC investigation the reader has developed a detailed understanding of the potential for an ORC operating from their geothermal resource. The reader will also understand some of the component limitations and development issues with small ORCs.

Commercial options must be reconsidered once again as they range from the kW range to the multiple MW range. A commercial ORC reduces the amount of development risk associated with system and likely to provide better support if the ORC does not perform as expected. The owners checklist provided at the final section of this standard is recommended if approaching commercial ORC vendors.

3.4 Expander Consideration

With the best thermo cycle determined the next step is to re-consider the feasibility of expander design, the most critical component of the ORC.

There are limited supplies of expanders for small ORCS as the design for the expander depends on the working fluid and the cycle conditions. Communications with a turbine company has recommended that the smallest turbine considered for an ORC should be 1MW[69], anything below this would be not economically feasible. However, there have been some unique cases where a small turbine has been successfully designed for an ORC, such as the Chena geothermal development[62]. In this document the smallest recommend turbine for an ORC is 250kW. Anything below this will use a volumetric expander. A system operating between 250kW and 1MW can consider both options for their ORC.

3.4.1 Volumetric Expanders

Volumetric expanders are scroll or screw type volumetric compressor operating in reverse, figure 48 and 49 respectively. There are examples of commercial operations using these types of expanders in their small ORCs [54]. Volumetric expanders have also been used by research institutes for small research ORCs as they are more available in the small scale range compared to turbines[53, 101].

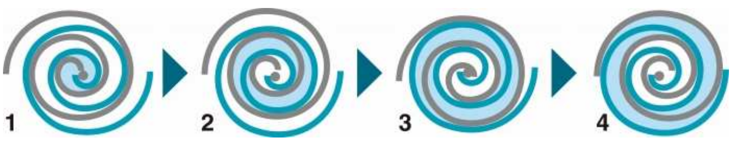


Figure 48 - Scroll Expander – Credit ANEST IWATA [102]

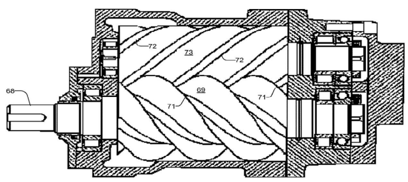


Figure 49 - Screw Expander – Credit Langson [103]

Figure 50 is the volumetric expander selection process. This process should give an understanding of the performance of the ORC operating with a volumetric expander as well as the limitations associated with them.

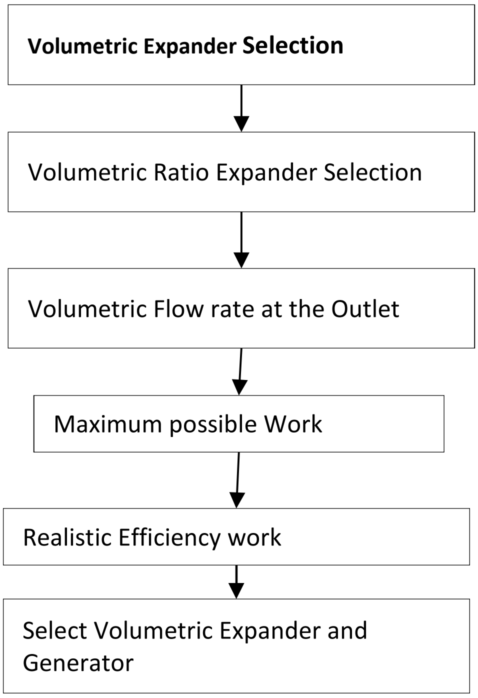


Figure 50 - Volumetric Expander Selection Process

3.4.1.1 Volumetric Expander Selection

At the point of publishing this document there was little guidance when considering a volumetric expander for an ORC. However, there has been literature on the topic to help with the selection process and performance predictions.

Firstly the two types of volumetric expander are a scroll and a screw expander. Regardless of what type of expander selected for the ORC application there is will to be an isentropic efficiency reduction in the expansion process. A turbine can provide an isentropic expansion between 80-95% while a volumetric expander in the best case scenario will have a 75% isentropic efficiency [53].

3.4.1.2 Volumetric ratio Expander

The two volumetric expansions ratios possible with volumetric expanders are 4 and 5 for a scroll and screw respectively. This will be the first point of selection[26]. A system that requires a higher volumetric expansion ratio is be better suited with a turbine.

A volumetric expander can operate outside of this volumetric expansion ratio limitation and some work is extracted in the overexpansion process. However; it is likely that this will result in a lower efficiency compared to a turbine.

3.4.1.3 Volumetric Flow rate

The potential work extracted using a volumetric expander is also limited by the volumetric flow rate. Volumetric expanders, unlike turbines, have a maximum volumetric flow that they can withstand. The limit for a custom designed volumetric expander is unknown at this point; however, the knowledge of flow rate limitations for compressors is used as a guideline.

3.4.1.3.1 Limits

A volumetric expander operates as a compressor in reverse; therefore, the compressor's maximum and minimum allowable flow rate will be equivalent to the expander's maximum and minimum flow rate. This limit for a compressor is at the inlet and so the expander's limit is calculated at the outlet of the expander where volumetric flow rate is greatest.

Table 15 - Volumetric Expander Limitations – Credit Quoilin [26]

Expander Type	Minimum Volumetric Flow	Maximum Volumetric Flow
Scroll	1.1 l/s	49 l/s
Screw	25 l/s	1100 l/s

3.4.1.4 Maximum Possible work

The maximum possible work from a system with a single volumetric expander will be limited by the volumetric ratio and flow rate through the expander. If these are within the limits of a volumetric expander it is suitable for the ORC.

3.4.1.5 Efficiency

At the point of writing this document there are no efficiency prediction maps for volumetric expanders. The literature on volumetric expanders has run experiments with efficiencies between 30-75% [53]. The ORC system output should be remodelled with the maximum and minimum possible isentropic efficiencies to understand the risk associated with a volumetric expander.

3.4.2 Turbine Design

Turbine Design is a complex process that requires a deep skill base and tools to achieve the ideal design. The appropriate specific speed is best preliminary approximation for turbine efficiency.

Figure 51 is the process is used to understand the feasibility of a turbine in the ORC. There are a number of different methods to approach preliminary turbine design and this should not be considered the only possible method. This process is the basic turbine design and is not the final decision for the turbine investigation; however, it will provide a greater understanding of the turbine limitations. The turbine design process can be explored further in the Introduction to Turbomachinery [83].

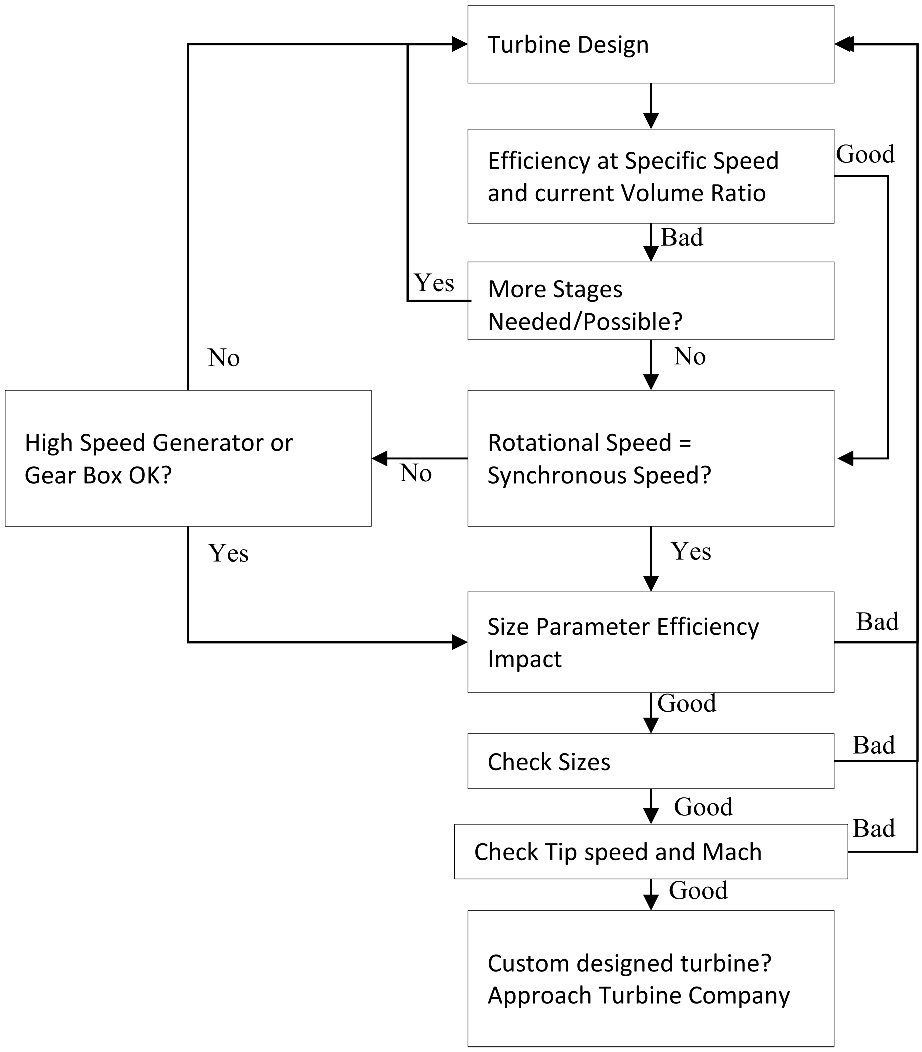


Figure 51 - Turbine Design Process

3.4.2.1 Impact of Specific speed and Volumetric Ratio on Potential Efficiency

Figure 52 and 53 show the impact specific speed and the volumetric expansion ratio have on the efficiency of both an axial and radial turbine. It is clear from these graphs that large volumetric ratios reduce the maximum possible efficiency [56]. The recommended volumetric ratio per stage is 4 and if the stage volumetric ratio increases beyond this will impact the turbine performance.

Once the stage volumetric ratio of the turbine is greater than 4 another stage is required. If more than one the axial turbine is the best solution. Radial turbines can be designed to have multiple stages; however, the fluid flow paths are unfavourable when compared to an axial turbine.

The specific speed calculated in the following plots uses RPS opposed to Rad/s as used in the Pre-Feasibility section. Therefore, the alternative specific speed equation should be used.

Alternative Specific Speed Equation

$$N_s = \frac{\left(\frac{N}{60}\right)\sqrt{m/\rho_{01}}}{(\Delta h_0)^{3/4}}$$

N_s specific speed of the stage

N rotational speed of the turbine in RPM

m mass flow rate of the working fluid in the turbine kg/s

ρ_{01} density of the working fluid entering the turbine stage kg/m^3

Δh_0 isentropic enthalpy drop across the turbine stage kJ/kg

The volumetric ratio of the turbine stage is calculated with the below equation

$$V_{ratio} = \frac{\dot{V}_{out}}{\dot{V}_{in}}$$

V_{ratio} Volumetric Ratio of the turbine stage

$\dot{V}_{out}$ The outlet volumetric flow rate of the turbine stage m^3/s

$\dot{V}_{in}$ The inlet volumetric flow rate of the turbine stage m^3/s

3.4.2.1.1 Axial Turbine

Figure 52 shows the impact both specific speed and the volumetric ratio of a turbine stage have on the efficiency of a turbine. At the optimum specific speed it is still possible to achieve acceptable isentropic efficiency from high volumetric ratio stages. However; as the specific speed changes away from the optimum the volumetric ratio greatly impacts the stage performance.

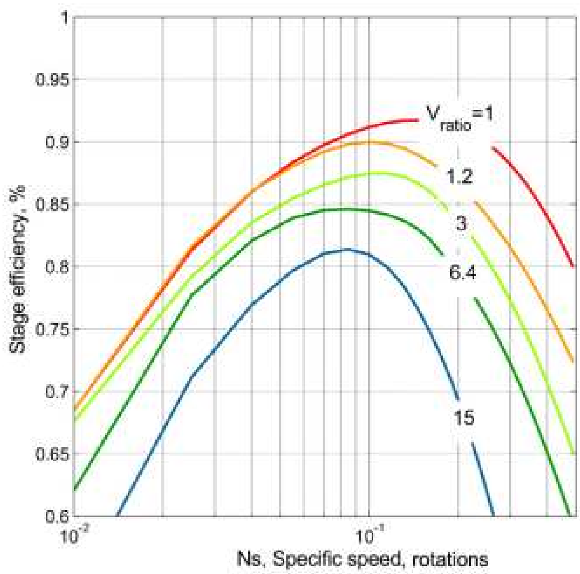


Figure 52 - Axial Turbine Specific Speed- Credit Astolfi [31]

3.4.2.1.2 Radial Turbine

Figure 53 shows the impact specific speed and volumetric ratio has on a radial turbine, another parameter known as size parameter VH is also included but will be explored in the next section. This plot is less detailed compared to the axial turbine plot; however, it is still clear that the volumetric ratio impacts the system performance. Figure 53 also gives an indication of what the blade will look like at the given volumetric ratio and speed.

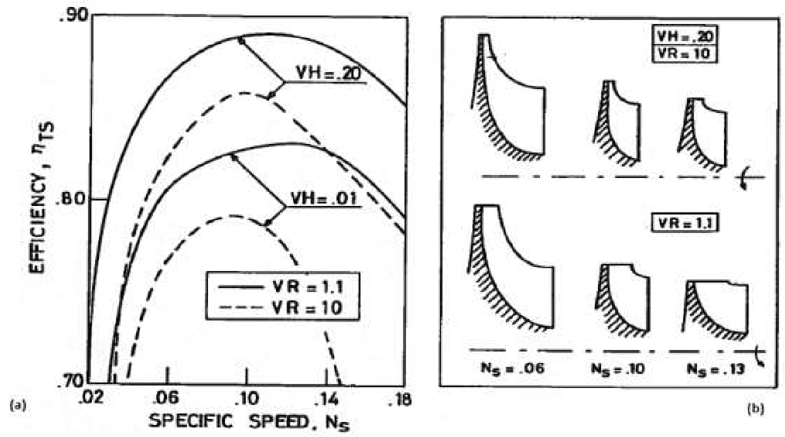


Figure 53 - Radial Turbine Specific Speed- From [104, 105]

3.4.2.2 More stages

There are a number of ways to improve the overall efficiency of a turbine and one way is to add another stage. Dividing the expansion processes across multiple stages will change the isentropic enthalpy drop across each stage and impacting the specific speed of the turbine.

Adding multiple stages will also change the optimum rotational speed to achieve the optimum stage specific speed and adding more stages can be used to better match the rotational velocity of the turbine to a synchronous generator.

3.4.2.3 Rotational Speed

The rotational speed can be the easiest thing to change in order to achieve the required specific speed to improve the cycle performance. However; increasing or decreasing the rotational speed of the turbine away from the synchronous speed of a turbine will require a high speed generator or a reduction gear box.

3.4.2.3.1 Speed Check

The optimum specific speed is used to check the optimum rotational speed for the turbine. Therefore, if there is a large difference between this speed and synchronous speed the turbine will need more stages, a reduction gearbox, or a high speed generator. Alternatively, there is a possibility that the optimum speed is close to the synchronous speed of the generator; therefore, operating the turbine at synchronous speed could only be a slight efficiency penalty.

3.4.2.3.2 High speed generator or Gearbox

If it is unlikely that a turbine will be able to operate at synchronous speed and high isentropic efficiency then a high speed generator or gear box must be considered. In these cases high speed generators are available at a number of different operating speeds. A gear box can also be designed to be coupled with a synchronous generator. Therefore, the efficiency possible is explored with high speed generator speeds, while a gearbox can be designed to suite the optimum rotating speed of the turbine.

3.4.2.4 Size Parameter Impact

Another tool in turbine design is the size parameter and its implications on efficiency. If the size parameter reduces efficiency more stages are needed. With this information an estimate of the actual size of the turbine can be calculated.

$$VH = \frac{\sqrt{V_{out}}}{(\Delta h_{isen})^{1/4}}$$

VH size parameter

V_{out} is the volumetric flow rate at the outlet of the turbine or turbine stage m^3/s

Δh_{isen} is the isentropic enthalpy drop across the turbine stage kJ/kg

3.4.2.4.1 Axial Turbine

Figure 54 highlights the impact the size parameter has on the axial turbine at the optimum specific speed and different volumetric ratios. Small size parameters greatly impact the potential performance of the turbine.

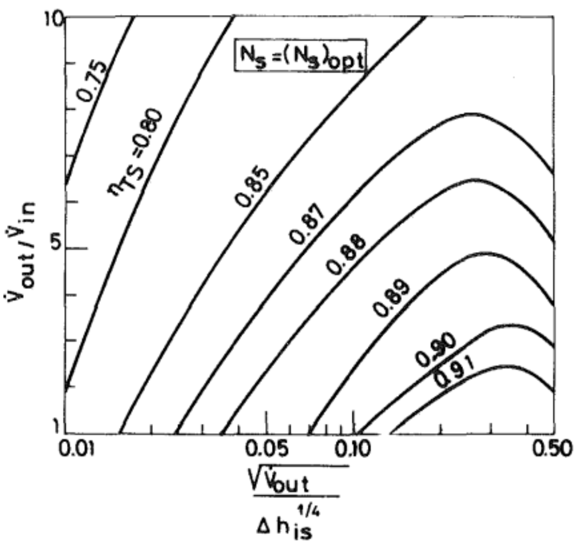


Figure 54 - Axial Turbine size parameter at optimum specific speed – Credit Macchi [56]

3.4.2.4.2 Radial Turbine

Figure 55 shows the impact the size parameter has on the radial turbine. Radial turbines are less affected by the size parameter when compared to axial turbines.

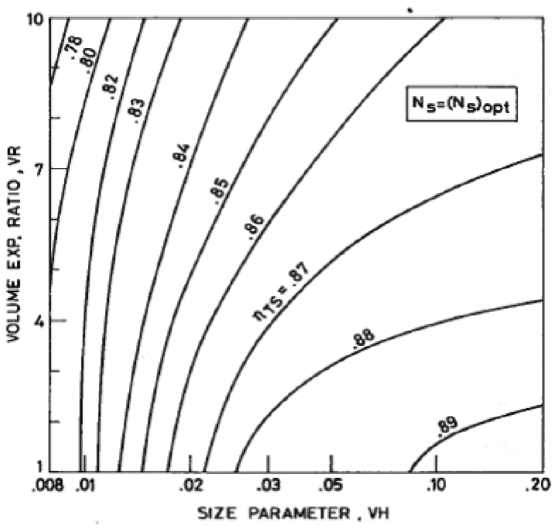


Figure 55 - Radial Turbine size parameter at optimum specific speed- From [104, 105]

3.4.2.5 Size Check

The size of the turbine is calculated with the more detailed efficiency plot and the following specific diameter equation. The efficiency plot highlights the specific diameter necessary for the required efficiency. The specific speed on the x-axis of this chart is determined with the original specific speed equation below.

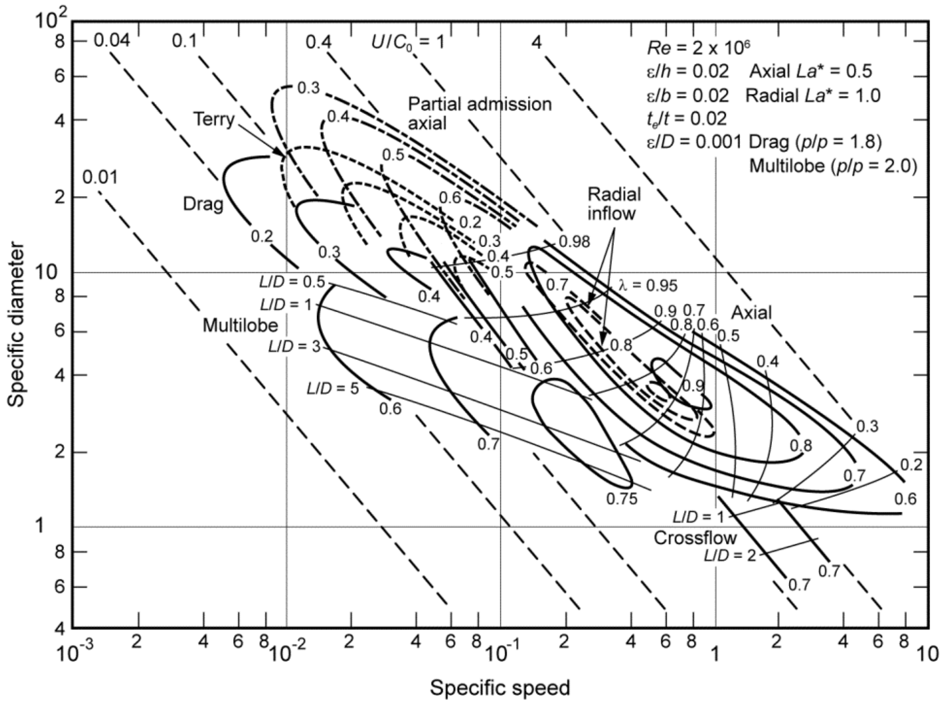


Figure 56 - Detailed Efficiency Plot – Credit Balje [73]

$$N_s = \frac{\omega\sqrt{m/\rho_{01}}}{(\Delta h_0)^{3/4}}$$

ω is the rotational speed of the turbine Rad/s

With the specific speed known there are a number of specific diameters possible to achieve the desired isentropic efficiency. Once the specific diameter is chosen the following equation will calculate actual diameter of the turbine which is the inlet rotor diameter of a radial turbine and the average diameter of an axial turbine.

$$D = \frac{D_s(\sqrt{\frac{m}{\rho_{exit}}})}{(\Delta h_{isen})^{1/4}}$$

D_s Is the specific diameter of the desired turbine

D is the mean diameter of a axial turbine and the inlet tip diameter of a radial turbine m

h_{isen} is the isentropic efficiency drop of the turbine stage – Reminder units should be J/kg

m mass flow rate of the working fluid kg/s

ρ_{exit} density of the working fluid kg/m^3

If the turbine diameter is below 30 mm manufacturing tolerances are hard to achieve and a large turbine is required. A small turbine will also have issues with the required blade heights.

3.4.2.6 Check Tip speed and Relative Mach number of turbine

The two important considerations for turbine design for an ORC are the turbine stresses and the relative Mach number of the fluid exiting the turbine.

A rule of thumb for turbine stress is that the turbine tip speed should be less than 400m/s, thus limiting the centrifugal stresses. The other consideration for turbine design is the relative Mach number. The Mach number should remain below 0.95 to avoid supersonic flow at the turbine exit [73]. The tip speed is a common speed limit for gas and steam turbine and ORC turbines are typically slower.

3.4.2.6.1 Radial Turbine Speed Check

The first step to determine whether a radial turbine violates either of these conditions is to know the diameter of the rotor inlet. This was calculated in section 3.4.2.5

3.4.2.6.1.1 Tip Speed Check

The following equation is used to determine the tip speed of a turbine, U.

$U = r\omega$

U is the velocity of the tip of the rotor m/s

r is the radius of the turbine rotor m

ω is the rotational speed of the turbine rad/s

The maximum allowable tip speed is recommended as 400m/s[83]; this minimizes potential stresses resulting from the centrifugal forces in the rotor.

3.4.2.6.1.2 Relative Mach Number

The next step is to determine the relative Mach number at the stage exit, which is the ratio of exit velocity and Mach velocity. A number of assumptions are made to simplify the process. The velocities across a turbine rotor are calculated with velocity triangles and more information regarding turbine velocity triangles can be found in the following references [73, 83]. The equations in this section are simplified for the results that are required to determine the outlet conditions.

Assumptions

- Zero swirl at the outlet $C_{\theta,exit} = 0$
- Flow coefficient of 0.2 $\phi = 0.2$
- Radius ratio is 1.8 typically in the range of 1.6-1.8 $r_r = 1.8$

The equation below is used to determine the relative Mach number at the turbine outlet. The critical velocity is the speed of sound at the corresponding fluid condition, which can be looked up with fluid property programs.

$$M_{rel,exit} = \frac{W_{exit}}{C_{critical}}$$

$M_{rel,exit}$ Is the relative mach number at the turbine exit

W_{exit} Is the absolute velocity of the fluid at the turbine exit m/s

$C_{critical}$ is the speed of sound in the fluid at the outlet conditions of the turbine m/s

The exit velocity of a radial turbine is determined with the following equation.

$$W_{exit}{}^2 = C_{m,exit}{}^2 + (C_{\theta,exit} - U_{exit})^2$$

The above equation for exit velocity can be simplified to the below equation using assumptions made earlier.

$W_{exit}{}^2 = U_{in}{}^2[\phi^2 + (1/r_r)^2]$

U_{in} is the rotor tip velocity m/s – calculated in the previous section.

ϕ is the flow coefficient

r_r is the radius ratio

Once the exit velocity is calculated the relative Mach velocity can be calculated. If the relative Mach velocity exceeds the 0.95 limit more stages will be needed.

3.4.2.6.2 Axial Turbine Check

The average diameter of the axial turbine is calculated previously from the specific speed and efficiency chart.

3.4.2.6.2.1 Tip speed Check

The same equation is used in section 3.4.2.6.1.1 is used to calculate the rotor speed of the axial turbine. The actually tip speed velocity will be slightly larger than the blade velocity calculation because the given diameter is the average diameter. The chart below gives a rough estimate of blade high for a turbine at optimum specific speed and known size parameters. This can be used to get a better understanding of the actual tip speed of an axial turbine.

$U = r\omega$

U is the velocity of the tip of the rotor m/s

r is the radius of the turbine rotor – which is the average diameter plus half the blade height m if interested. Otherwise the average velocity is an appropriate approximation.

ω is the rotational speed of the turbine rad/s

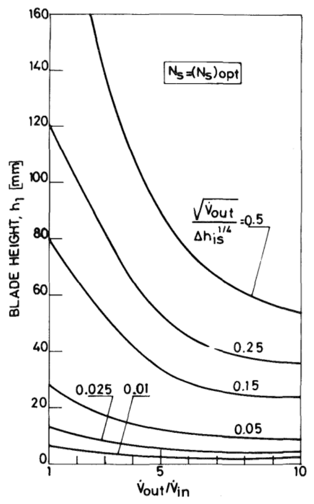


Figure 57 - Blade height estimate [56]

3.4.2.6.2.2 Relative Mach Number

An axial turbine has the same equation for the relative Mach number as a radial turbine. However an axial turbine has the same tip velocity as there is no change in blade diameter across the stage. Below are the assumptions made for an axial turbine; these velocities are calculated with turbine velocity triangles. The basic equations used here are simplified for this guide.

- Meridional Velocity C_m and tip speed U are constant across all stages
- Flow coefficient is set to 0.4 to maximize potential efficiency $\phi = 0.4$

The next step for an axial turbine is to determine the reaction of the axial turbine, which is between 0 and 1. An impulsive turbine, used for small single stage turbines with a high work output, will have a reaction of 0. Turbines are quite often a mix of impulse and reaction and will have a reaction of 0.5.

Reaction

Λ ues both 0 and 0.5

The next step is to calculate the stage loading coefficient of the turbine, which is done with the following equation.

$$\psi = \frac{\eta \Delta h_{isen}}{U^2}$$

ψ loading coefficient

η desired isentropic efficiency (roughly 0.85)

Δh_{isen} isentropic enthalpy drop across the stage – reminder j/kg

U the average blade velocity of the axial turbine (not tip velocity) m/s

The next step is to determine the relative flow angle of the fluid at the stage exit, which is calculated with the following equation.

$$\tan \beta_{exit} = -\frac{\psi + 2\Lambda}{2\phi}$$

β_{exit} is the relative flow angle at the turbine outlet (degrees)

Λ is the reaction of the turbine (between 0 and 1)

ϕ is the flow coefficient 0.4 for this axial turbine

The relative flow angle at the exit is then used to determine the relative flow velocity at the turbine exit.

$$W_{exit} = \phi U / \cos \beta_{exit}$$

With this information the relative Mach number can be calculated and if it is above 0.95 more stages are needed.

3.4.2.7 Results

The results of this turbine analysis will highlight the limitations of the turbine and will be useful when approaching a turbine design company to discuss the specifications of the turbine and the expected efficiencies.

3.5 Trade-off Analysis

Up until this point the thermo cycle has been modelled with a number of assumptions that have been carried through from the pre-feasibility study. This is a chance to revisit these assumptions and review the impact on the performance and economics of the system. For each trade off explored the ORC work output and capital cost are required to understand the trade-offs.

Section 2.7 of the pre-feasibility study should be followed once again to understand the costs of the ORC. A spread sheet is recommended as the calculations will be repeated with each cycle to understand the implications of each trade-off.

The following assumptions have been made at the start of the modelling process. A trade off analysis will show the impact on the performance and costs of the system.

Assumptions and potential impact on the ORC a number of these assumptions are associated with heat exchanger design from [71].

- 3°C superheat

- 5°C sub cooling
- 15°C pinch point in evaporator
- 5°C pinch point in the recuperator
- 14°C approach temperature in the condenser
- Outlet Geothermal Temperature
- Efficiency of pump and turbine

3.5.1 Super Heat

Superheat is assumed in the original system to protect the expander from liquid.

3.5.1.1 Potential Impact on the System

Increasing the super-heat amount will have negligible impact on the ORC. A dry fluid will result in a decrease in the thermal efficiency of the ORC. A wet fluid should have enough superheat to ensure the fluid is dry after the expansion process.

Decreasing the superheat of the system will result in more work from the ORC as more heat is supplied to evaporate the fluid to a higher pressure. Deceasing the amount of superheat will also reduce the size of the required heat exchangers and reducing the overall system cost. However; lowering the superheat is dangerous to the life of the expander and increasing the inherent component risk.

3.5.2 Sub Cooling

Sub cooling is required to reduce the risk of the pump cavitation. The ORC feed pump operates very close to the fluids saturation pressure. Investigating the NPSH-R for possible pumps will indicate the minimum amount of sub cooling.

3.5.2.1 Potential impact on the system

Reducing the sub cooling will improve the performance of the ORC as the condenser can condense to a lower temperature. However, this will impact the available net suction head available and increase the risk of cavitation in the feed pump. This can be overcome with increasing the static heat of the fluid entering the feed pump.

3.5.3 Pinch Point in the Evaporator

The pinch point is a critical design point for the ORC. It impacts the performance of the ORC and the reinjection temperature of the geothermal fluid.

3.5.3.1 Potential impact of the Pinch point in the evaporator

The pinch point in the evaporator is one of the main design factors that determine the evaporation pressure. Decreasing the pinch point temperature will increase the heat transferred in the heat exchanger and might result in a further reduction in geothermal fluid temperature. The pinch point has significant implications on the size of the heat exchanger.

Increasing the pinch point temperature will decrease the work produced by the ORC; however, it will also reduce the required heat exchanger size and cost.

3.5.4 Recuperator Pinch Point

The recuperator is typically only used in ORC system that have a strict limit to the reinjection temperature and the owner wants to maximize the potential output of the system. A system without strict reinjection temperatures will not benefit from a recuperator.

3.5.4.1 Potential Impact of the Recuperator Pinch Point

Reducing the pinch point further will result in more heat recovered from the fluid and more heat available in the geothermal fluid. This can improve the systems performance but at the same point greatly increase the recuperator size and cost.

Increasing the pinch point of the recuperator too much will make the amount of heat recovered insignificant.

3.5.5 Condenser Approach Temperature

The approach temperature determines the condensing temperature of the ORC.

3.5.5.1 Potential Impact of the Condenser approach temperature

Reducing the approach temperature will improve the performance of the ORC but also increase the cost of the size of the condenser. Increasing the approach temperature will decrease the size of the condenser but also reduce the performance of the ORC.

Decreasing the approach temperature also increases the parasitic loads on the condensing fans or pumps. There is an optimum point where if the approach temperature is decreased much further the parasitic loads will outweigh the potential power increase. There will also be an economical optimum point where decreasing the approach anymore will not return the investment for the number of extra fans required [106].

3.5.6 Geothermal Outlet Temperature

The outlet temperature of the geothermal fluid should be restricted by a geochemical analysis; however, in the case that there is no minimum temperature limit then this should also be explored.

3.5.6.1 Potential impact of the geothermal outlet temperature

Generally as the geothermal outlet temperature decreases there is more work available to the system. However; there is an optimum outlet temperature to maximise the work out of a basic ORC. The other alternative if there is no minimum geothermal limit is to use an alternative cycle.

3.5.7 Efficiency

Explore a range of pump efficiencies to understand the minimum allowable pump efficiencies. If the pump reduces the power by 10% another pump is required.

3.6 Off Design Performance

The trade-off analysis will finalise the thermodynamic cycle of the ORC. The off-design performance of the ORC is investigated to decide on the amount of redundancy that can be built into the design to cope with the off design conditions and improve the flexibility of the ORC.

3.6.1 Condenser

The condenser at this point has either been designed as a floating condenser or a maximum temperature condenser. If the condenser has been designed for the maximum possible temperature range then it should be adequately sized to maintain the ORC performance at each of the potential cooling fluid temperatures.

The other case would be if the condenser was chosen to be a floating condenser designed at the average temperature limit to minimize capital cost. This scenario would require exploring the potential performance at the different ambient temperature conditions. At the higher temperature conditions it is also important to check the required surface area to condense the working fluid.

A floating condenser will have a variable output throughout the year and have a fluctuating performance. A plot like figure 58 is required to understand the performance at varying ambient temperatures.

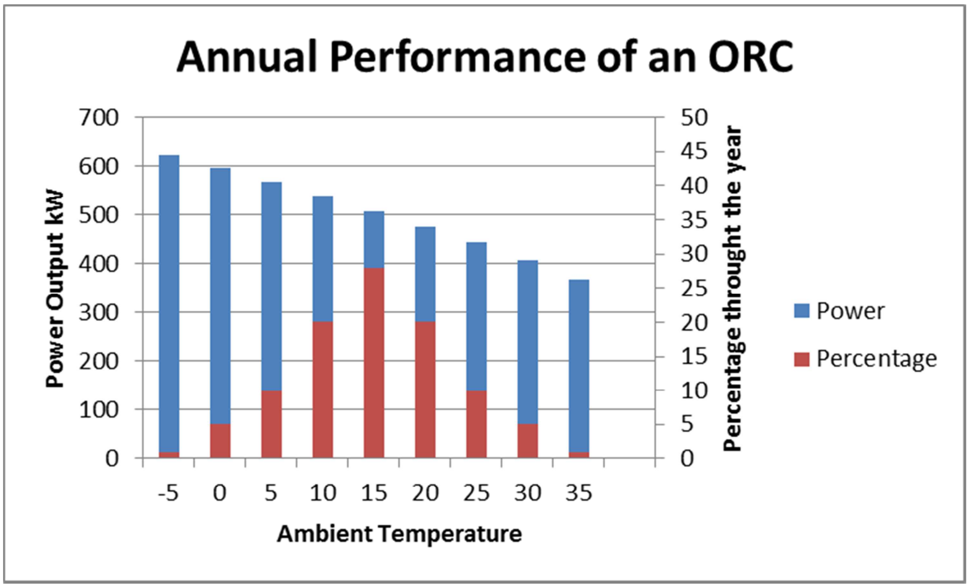


Figure 58 - Ambient Temperature and Power

The condensing temperature also risks lowering the geothermal temperature further as the inlet temperature to the pre-heater changes. If there is no issue with further reducing the geothermal temperature the required heat exchanger area at these different temperatures must be explored. The alternative is to increase the pinch point temperature as the condenser temperature decreases. This will maintain the same reinjection temperature and will require less heat exchanger surface area to facilitate the heat transfer; however, it will require less working fluid flow, which will result in a reduced power output compared to the optimum output at that condensing condition.

The humidity changes in the area also impact the performance of the condenser and the ORC and need to be taken into consideration.

3.6.2 Geothermal Fluid Changes

Geothermal resources change during the life of the project. Changes in the reservoir result in a cooler fluid available for the ORC, reduces brine flow rate, or a higher restriction on the geothermal reinjection temperature. Each of these affects must be considerations for the ORC design.

The most likely change over the life of the ORC will be the reservoir temperature reduction resulting in cooler fluid available for the ORC. As the geothermal fluid cools less heat is available to maintain the same turbine inlet conditions and this can be overcome by decreasing the pinch point temperature. This will result in a larger pre-heater and evaporator to maintain turbine inlet conditions. Another opportunity is to give the plant an additional heat exchanger at some point in the plants life to accommodate the cooling geothermal fluid. This flexible option can benefit project as it reduces the capital cost of the ORC.

If the geothermal fluid flow rate decreases there will be less power produced by the system but it will not require more heat exchanger investment. In the scenario of geothermal fluid increase more heat exchanger area will be required for each of the heat exchangers.

The heat exchanger will also foul throughout the plant life reducing the heat transferred from the geothermal fluid. A minimum heat transfer is required to determine at what point the plant requires maintenance and cleaning. The heat exchanger area can be changed to potential fouling factors. The geochemistry will determine what fouling factor applies to the geothermal fluid; however, below are some guidelines if it is not known.

Fouling Factors

Geothermal Brine [65]

- Clean -1.76E-5 m^2K/W
- Medium 1.94E-4 m^2K/W
- Bad 5.81E-3 m^2K/W

Working Fluid [71]

- Clean – 1.00E-4 m^2K/W
- Oil Traces – 3.50E-5 m^2K/W

The original overall heat used in the original area calculations can be adjusted to accommodate the fouling factor with the following equation.

$$U_{new} = 1/(R_{Foul} + \frac{1}{U_{old}})$$

U_{new} is the new overall heat transfer coefficient with the fouling factor included W/m^2K

R_{Foul} is the fouling factor of the heat exchanger m^2K/W

U_{old} the original overall heat transfer coefficient of the heat exchanger W/m^2K

Good understanding of the reservoir will help with decision making to build redundancy and flexibility into the ORC.

Each of the geothermal potential changes should be investigated to check if the heat exchangers should be oversized and the potential performance risk. The geothermal changes will also impact the condenser load and the condenser size should be checked.

3.7 Risk

A detailed risk assessment is an important section for a feasibly study. A risk analysis highlights the potential pitfalls for a geothermal development to help the owners develop strategies to overcome these issues.

3.7.1 General Geothermal Project Risks

There are a number of risks associated with any geothermal development these examples are from the geothermal handbook [3]

- Resource Risk
- Oversizing
- Financing due to high upfront costs
- Completion/delay risk
- Operation risk
- Off take risk
- Price risk
- Regulatory risk

3.7.1.1 Resource Risk

Detailed resource risk is not in the scope of this standard but still must be understood when designing an ORC. There is generally a significant amount of uncertainty associated with a geothermal resource and this can complicate the development in a number of ways, such as material selection and potential fouling rates in the heat exchangers.

3.7.1.1.1 Geothermal Manifestations

All geothermal developments have the potential to impact the natural geothermal manifestations in the area. Low temperature geothermal power plants can possibly operate off hot springs which might be cooled quicker from the re-injected fluid. This will be a serious problem if there are other hot springs in the area that are fed from the same reservoir.

3.7.1.2 Oversizing

Developed power plants can be the wrong size for the resource. An undersized plant is not a risk but will result in a loss of opportunity. Oversizing a plant has two main risks, one is that the capital cost intensifies and the payback period is extended if the reservoir cannot meet the plant design, second an oversized plant increases the depletion rate of the reservoir resulting in excessive cooling and off design performance. This risk is generally applied to the resource assessment, emphasising the importance of the resource assessment.

3.7.1.3 Financing upfront costs

Financing risks such as large exchange rate swings and finance rates can occur during the projects development before financial closure. This will reduce the financial return but is a common risk with any large project.

Geothermal projects have long lead time, which can be shorter with Low temperature geothermal system not requiring significant well development. Banks may require higher credit risk premium to account for the large debt for the capital requirements.

3.7.1.4 Completion / Delay Risk

Normal delay risks associated with any other large industry project apply to ORC development. However there are further risks associated with completion and delay if the drilling program is included in the ORC development.

The working fluid procurement can delay the completion of an ORC because a number of regulations are in place for environmentally hazardous working fluids. There is also a chance that a working fluid a banned during the project.

3.7.1.5 Operation Risk

An ORC also has operation risks associated with the choice of working fluid. Hydrocarbons increase the risk of fire and explosions and non-flammable refrigerants can be toxic to the environment. Working fluids also have an asphyxiation risk. The failure of any main component can cause 6-12 month downtime.

A working fluid leak has unique environmental risks and can also result in downtime for clean-up and recharging the ORC with fresh working fluid which may take a long time of procure from overseas. A number of commercial ORC have a tank of working fluid on site for storage during shutdown and in case of a leak to top up their ORC working fluid amount.

Operational risks can be minimized by business interruption insurance; however, this is another cost to the owner.

3.7.1.6 Off take Risk

The failure of the buyer to take power due to reasons concerning dispatch, transmission congestion, or transmission line failure. These risks should be no higher than any other type of power generation system; however, a remote power system may be able to minimize these risks if the power is used directly onsite instead of selling the power to the grid. However, remote power developments can cost more if the plant is following a load profile opposed to operating as a base load provider.

3.7.1.7 Price Risk

Price risk is the risk of less than expected revenue resulting from lower than expected off-take prices. This is a serious risk when some or all of the off-take is at market prices opposed to the fixed prices proposed under a power purchase agreement.

3.7.1.8 Regulatory Risk

These are the risks resulting from a government's holding of discretionary power over factors affecting the project developer's or investor's commercial success [3]. Some government policies that impact the success of the geothermal development are related to pricing, taxation, natural resource use, procurement procedures, environmental concerns, and land usage permits. Therefore; understanding the regulatory risks is critical for informing the investor.

3.7.1.9 Other risks

Opposed to risks associated with any other grid-connected power development there are other market wide risks that a geothermal development needs to take into account. These include the foreign exchange risk, interest rate risk, and commodity price risk.

3.7.2 Specific Risks associated with Low Temperature Geothermal ORCs

There are a number of further risks not covered in the geothermal handbook that are more specific to low temperature geothermal developments, which are listed below.

- Environmental
- Electricity Market
- Technology Risk
- Community
- Power Use
- Access
- Operation
- Development

3.7.2.1 Environmental Risks

There are a number of environmental risks associated with geothermal project. The risk to the environment can shut down a geothermal project so precautions to avoid these risks are critical [107]. An environmental impact assessment is required for a geothermal project. However, these risks are much lower than other power generation options and geothermal power is generally considered environmentally friendly.

The carbon footprint is very small for a geothermal power plant and an ORC has the potential to produce no carbon dioxide if all of the reservoir fluid is re-injected and no NCGs are released.

3.7.2.1.1 Working Fluid Impact

The working fluid in an ORC has some associated risk; however, it is easily managed with good design. An appropriate working fluid for the ORC should not deplete the ozone and have a low global warming index; therefore, if there is a leak the environmental impact is still minimal. The direct risk to the environment around the ORC will be included in an environmental assessment.

A water cooled condenser should be equipped with leak detection to avoid the working fluid leaking in to the cooling water.

3.7.2.1.2 Water Quality and Use

A benefit of an ORC is that they can re-inject 100% of the geothermal fluid if designed properly. One exception to this is if a water cooled condenser and evaporative cooling tower are used. This cooling system could use cooled geothermal fluid as make up water the condenser. However, this is unusual as an ORC will either use river water or more commonly an air cooled condenser.

3.7.2.1.3 Air Emissions

ORCs are generally closed loop systems that will re-inject the gases dissolved in geothermal brine. However; some NCGs, typically CO2 and H2S, will be ejected from the brine in the heat exchangers in very low concentrations. It is common to eject these gases at the highest point to disperse them as much as possible and avoid it gather in areas.

3.7.2.2 Electricity market

The electricity market is important to a small geothermal development because the power produced by a small low temperature geothermal ORC can result in long pay back periods. However, a small ORC will be considered for some niche markets.

3.7.2.3 Technology Risk

The small scale ORC can use technology that is not mature in the power generation field. One example unique to the ORC is the expander; a number of these ORCs are small and, therefore, lacks significant knowledge of the longevity of volumetric expanders in a base load ORC. Another risk is small scale turbines costing more than predicted because of the complexities in a one off turbine design

3.7.2.4 Community

There can be increased risk to the project success if the local community does not agree with the development. There is the possibility that land owners will have to agree that pipe lines can cross their land. Also their demands could result in a height restriction if they consider the plant unsightly. Construction and access to the building site will also need to be discussed with community members.

The noise levels will also have to be taken into account if there is a community nearby. One solution can be to put the ORC in a noise insulated building. Inclosing an ORC in a building brings new risk of asphyxiation in the event of a working fluid leak.

The geothermal system could have cultural significances to the local community and their concerns must be addressed before any plans for development take place. This should be covered in the geothermal assessment.

3.7.2.5 Power Use

Where the power plant can be connected is important for a small scale ORC because building long transmission lines to connect the ORC to the grid will increase the capital cost. The other alternative is to use it with nearby industry or as a remote power system. However, a variable load ORC is typically more complicated than the base load design.

3.7.2.6 Site Access

It is likely that a small ORC will be developed for remote power generation and will require significant machinery to access the desired site. A remote location may require infrastructure to accommodate the amount of equipment necessary for the development.

3.7.2.7 Operation

If the plant needs an operator this will reduce the revenue of the system and increase the payback period. Alternatively there could be remote operation of a number of plants. This option will require further investment to setup remote operation systems.

3.7.2.8 Development Risk

An owner designing their own ORC will inherit the design and development risk of the ORC. The main development risk is associated with the turbine design as most turbine providers have been improving their technology for a number of years.

3.8 Site Description

The project site should be reviewed once more to ensure it suitable. The following conditions should be favourable for a successful project:[3]

- Location
- Climate
- Topography
- Geology
- Water Supply
- Access
- Transmission Lines

3.8.1 Location

Identify the physical location of the geothermal site and what important features are nearby, roads, rivers, national parks, and cities.

3.8.2 Climate

The average wet and dry bulb temperatures are important as the maximum and minimum temperatures throughout the year.

3.8.3 Topography

There should be sufficient flat space to facilitate the equipment required to develop the geothermal site; otherwise, significant earthworks would be required. The average 1.42 m^2/kw[25] can be used as a rough guide for this.

3.8.4 Geology

The geology of the site what are the rocks

3.8.5 Water Supply

The water supply available to the site is important for a system using a water cooled condenser.

3.8.6 Access

The site access should be sufficiently large to allow heavy equipment and drill rig transport

3.8.7 Transmission lines

A close site to connect the ORC to the grid

3.9 Other requirements of an Power plant

- Fire
- Fencing
- Possible Flare Stack
- Control rooms

3.10 Economics Revisited

At this point the ORC is finalized and a more in-depth economic analysis can be done to determine the capital cost of the system and makethe final investment decision. Section 2.7 of the pre-feasibility study provides the equation needed for the cost estimate.

3.10.1 Additional Factors

The components selection should be fully decided at this point and now additional factors can be added to improve the accuracy of the cost analysis.

3.10.1.1 Additional Cost Factors

Once the installed component costs have bene estimated the overall plant cost can be estimated with an additional cost factor. This factor includes costs for plant and steam field piping, contingency, controls, consulting, plant house and foundations, electrical connection, instrumentation and financial services.

Six case studies were examined in order to estimate the Lower and Upper bound system costs [108], [109], [110], [111] and [112].

Lower bound systems are characterised as ‘easy’ installations. There are minimal additional infrastructure costs involved, and the resource is readily available from a plant engineering perspective.

Upper bound or ‘hard’ installations require large additional installation costs; this is for a green field. These can by typified by installation in a remote location, significant steam field piping, high engineering requirements for cooling water and reinjection pumping and the necessity for additional treatment systems for one or all of the fluids.

The upper and lower bounds represent the average for the hard and easy projects respectively. Most systems can be expected to cost somewhere between these two bounds, but unusually difficult or easy projects can be expected to fall outside of these bounds.

Adding the additional cost factor to the estimated installed component cost produces an estimate of the final overall project costs.

The additional cost factor for the majority of plants can be expected to lie within the two bounds.

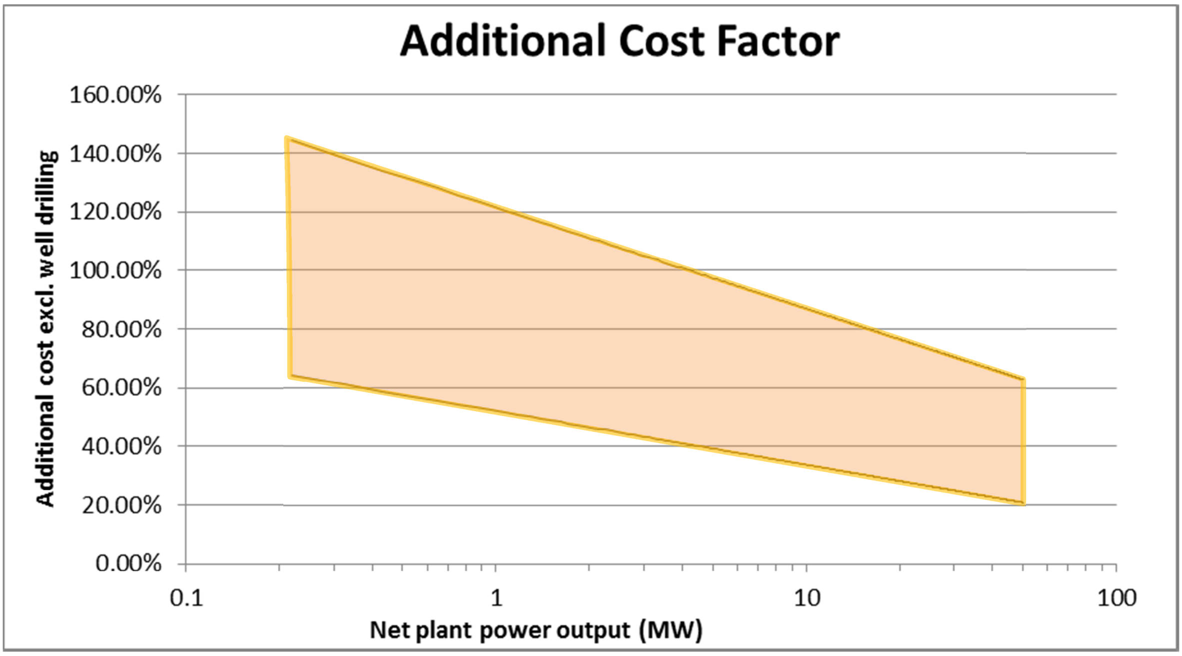


Figure 59 - Plant Project additional installation cost

3.10.1.2 Steam field Costs

Steam field costs

For geothermal projects, exploring and developing the steam field is a high risk stage of the project. There is a large variability in the amount of time and money that must be invested into the steam field. As such, an estimate of the cost by choosing an appropriate cost factor may not be meaningful in the context of a project.

In order to provide a basic guideline, the following ranges can be used for NZ fields, as provided in [112];

Table 16 - Steam Field Costs – Credit SKM [112]

Size	20 MW	50 MW
Lower Cost (NZD / KW)	1100	650
Upper Cost (NZD / KW)	1200	1850

Note, these cost estimates were provided with a 2007 basis, and have been adjusted using the PPI price index (all sectors).

3.10.1.3 Overall Cost Estimate Comparison

Lastly, the overall plant cost estimate (excl. well drilling) can be compared with this overall specific cost chart. This cost chart can is a reality check for the cost of the plant designed here.

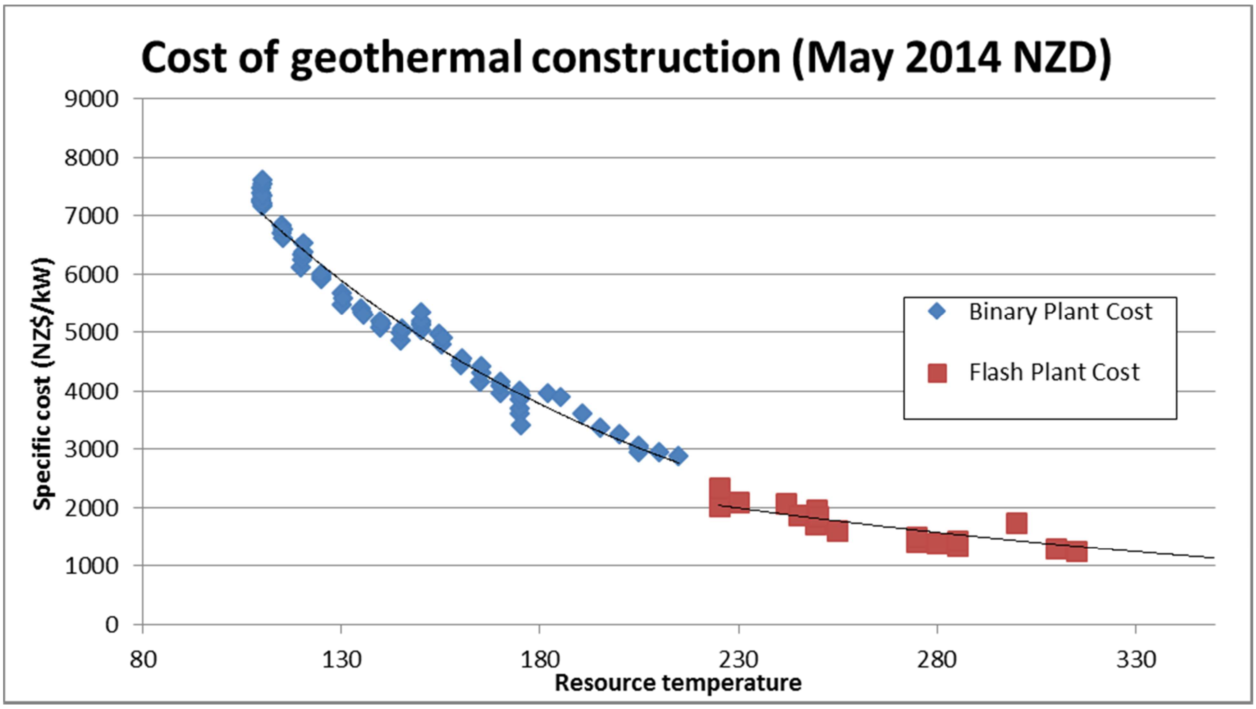


Figure 60 - Specific Costs of geothermal plants – Credit IRENA [113]

3.10.2 Present Value and Internal Rate of Return

The net present value (NPV) and internal rate of return (IRR) are common tools used to determine economic feasibility of a project. The IRR is also used to compare other possibilities offered to the owner.

3.10.2.1 Net present Value

NPV is a good indication whether the investment will add or subtract value from the company during the life of the project. A positive NPV at the end of the project life indicates that the project will return on the investment made; however, a negative NPV is an indicator that the project will not provide return on the investment. A NPV of 0 will neither add not subtract value and so other considerations will determine whether the project is a success.

$$NPV = \sum_{t=o}^{N} \frac{R_t}{(1+i)^t}$$

NPV is the sum of the project costs and returns over the life of the project $

R_t is the net cash flow at that time t $

i is the discount rate %

t is the time of cash flow years

The discount rate used is a critical factor as it will determine the NPV. 10% is the rule of thumb; however, as discussed in the pre-feasibility the discount rate for renewable energy can be smaller.

3.10.2.2 Internal Rate of Return

The internal rate of the return is the discount rate to ensure the NPV at the end of the projects life is equal to 0. Typically a geothermal project life is around 30 years. The higher this value is the better return on investment the project will have. It is also a common tool used to compare a number of different capital intensive projects.

3.10.3 Other Alternative Costs

The owner should also consider other potential power sources if their main goal for the ORC is electricity generation. The general specific costs of other renewables and non-renewables are listed below.

3.10.3.1 Renewables

The renewables considered at solar, wind, and micro hydro. [23]

3.10.3.1.1 Solar

The cheapest possible solar installation is considered to be 1000 USD/kW. However, this is without the auxiliary equipment required for solar making it suitable for grid connection. A more accurate solar estimate is 4000 – 5000 USD/kW.

The solar resource however has a limited capacity compared to a geothermal ORC and is taken into account when looking as specific costs of the system. The capacity factor of solar is roughly 20% and a comparison can be made to geothermal ORC with a much higher capacity factor by increasing the specific costs by 4 times to account for the reduced capacity factor. Therefore, the resulting specific costs of solar are 20,000 – 25,000 USD/kW.

$$\frac{Specific\ Cost}{Capacity\ Factor} = Adjusted\ cost$$

3.10.3.1.2 Wind

Wind power generally has a specific cost of around 2000 USD/kW. Once again the capacity factor for wind is 30%, which is significantly lower than a potential geothermal ORC. The same capacity comparison used for solar can also be used for wind; therefore, the adjusted wind specific cost is 6700 USD/kW.

3.10.3.1.3 Hydro

Small hydro is a very cost competitive solution. A small run of river or generic hydro system will cost between 1000 -8000 USD/kW. The capacity factor is also very similar to geothermal and so this specific cost does not need to be adjusted.

3.10.3.2 Non Renewable

Alternatively if this project is for remote power a diesel generator can be used. Diesel power generation costs between 0.3-0.5 USD/kWh. A renewable system will also be a more favourable option for remote power generation.

3.11 Final Investment Decision

At this point the details of the costs of the system are understood and the final investment decision must be made. If the costs of the system are too high and the payback periods are not favourable then it is unlikely that the ORC will be successful. However; the ground work for the ORC has been done at this point and should the market open up this work can be used to approach an ORC company to develop the detailed design for the ORC.

3.12 Specification Sheets

If the final investment decision is made then the next step for the ORC project is to develop the detailed design of the ORC. This can be done in house and it the next section will highlight the appropriate standards for the components in the ORC. Otherwise specification sheets can be used to approach suppliers of the components to do the detailed design. Blank specification sheets are supplied for each possible component in the ORC.

3.12.1 Heat Exchangers

3.12.1.1 Shell and Tube Heat Exchanger

This is the specification sheet for a shell and tube heat exchanger. Commonly used by all shell and tube heat exchanger manufactures.

				SHELL-AND-TUBE HEAT EXCHANGER		
				CLIENT	*EQUIP. NO*	*PAGE*
REV	PREPARED BY	DATE	APPROVAL	*W.O.*	*REQUISITION NO.*	*SPECIFICATION NO.*
0						
1				*UNIT* *AREA*	*PROCURED BY*	*INSTALLED BY*
2						

1	Size	TEMA Type		Connected in (series/parallel)		
2	Surface per Unit		Shells per Unit	Surface per Shell		m^2
3	**Performance of One Unit**					
4	Fluid Allocation			Shell Side		Tube Side
6	Fluid Name					
7	Flow Total	kg/h				
8	Vapor	kg/h	(in/out)			
9	Liquid	kg/h	(in/out)			
10	Steam	kg/h	(in/out)			
11	Water	kg/h	(in/out)			
12	Noncondensable	kg/h	(in/out)			
13	Temperature (In/Out)	°C	(in/out)			
14	Density	kg/m3				
15	Viscosity	cP				
16	Molecular Weight, vapor					
17	Specific Heat	kJ/kg-°C				
18	Thermal Conductivity	W/m-°C				
19	Latent Heat	kJ/kg				
20	Inlet Pressure	kPa(g)g	(inlet)			
21	Velocity	m/s				
22	Press Drop Allow/Calc	kPa(g)				
23	Fouling Factor	m^2-°C/W				
24	Heat Exchanged	W			LMTD (corrected) °C	
25	Service Coeff.	W/m^2-°C	Dirty		Clean	
26	**Construction Data for One Shell**					
27			Shell Side	Tube Side	Sketch	
28	Design/Test Press	kPa(g)g				
29	Design Temperature	°C				
30	No. Passes per Shell					
31	Corrosion Allowance	mm				
32	Connections	In DN				
33	Size & Rating	Out				
34		Intermediate				
35	**Tubes** No.	OD, mm	Gauge	Length, m	Pitch layout, deg.	
36	Type		Material		Pitch ratio	
37	**Shell**	OD, mm	ID, mm	Material		
38	Channel or Bonnet	OD, mm	Thick	Channel Cover		
39	Tubesheet Type					
40	Floating Heat Cover			Impingement Protection		
41	Baffles Cross (number)		% Cut (d)		Spacing C/C, mm	
42	Baffles Long		Seal Type No			
43	Supports Tube		U-Bend		Type	
44	Bypass Seal Arrangement			Tube-Tubesheet Joint		
45	Expansion Joint No.			Type		
46	Rho-V2-Inlet Nozzle		Bundle Entrance		Bundle Exit	
47	Gaskets - Shell Side			Tube Side		
48	Floating Heat Cover			Supports		
49	Code Requirements			TEMA Class		
50	Weight per shell kg		Filled w/water		Bundle	
51						
52	Notes					
53						
54						

3.12.1.2 Plate type heat exchanger

Listed here are some specifications that should be made for a plate type heat exchanger. It is important that a plate heat exchanger with geothermal fluid is a plate and frame type so that it can be dismantled and cleaned once scale has built up.

Below is a list of specifications from a plate heat exchanger manufacturer [114]

Design Requirements

- Thermal Design
    - To Maintain velocities and reduce fouling, unit shall be sized to provide 100% of the area required
    - Shall be designed for future expansion to accommodate a minimum of 20% extra heat transfer surface area
    - Liquid velocity thought the inlet and outlet ports shall be a maximum of 7.62 m/s to minimize pressure drop and erosion
- Mechanical Design
    - Frame, tie bolts, and supports shall permit the future installation of 20% additional plates
        - This will also allow ease in disassembly and cleaning of unit
    - Plates
        - Minimum thickness shall be 0.5mm
        - Plate nozzles connections shall force each fluid across the plate surface in a diagonal path
            - Insure proper fluid distribution
            - Parallel flow paths shall not be permitted
        - Plate design must withstand hydro test 1.3 times design pressure
            - Each channel pressurized independently
        - Design of plates must permit metal to metal contact between adjacent plates
        - Gasket grooves designed such that in the compressed state, the gasket will interlock between adjacent plates
        - Plates fully supported by carrier bar. Bar in contact with heat transfer plates shall be stainless steel
        - Guide bar designed to maintain plate alignment. Guide bar shall not be used for support, also stainless steel.
    - Gaskets
        - Glued Design to prevent slippage during assembly and cleaning
        - Elastomeric gaskets
        - Gaskets designed so that when plates are tightened gaskets will be compressed a minimum of 20 – 25%
        - Port holes in each plate must be fully gasketed and vented to atmosphere to force any leaks to the outside – this prevents process fluid mixing

3.12.1.3 Air Cooled Condenser

The specification sheet used by air cooled condenser manufactures

Date	Rev.	By
Proposal		Item No.
Job No.		Item No.
Customer		
Address		Plant Location

Service			Weight/Bay
Model		No. of Bays	No. of Fan Cells / Bay
Dimensions L	W	H	

PERFORMANCE	
Customer Item	
Fluid Circulated	
Heat Exchanged	
U External - Bare	
CLMTD	
External Surface	
Bare Tube Surface	

TUBE SIDE				
Total Flow				
	IN		OUT	
Vapor "				
Liquid "				
Steam "				
Non-Condensable "				
Temperature				
Pressure				
Pressure Drop	Allowed:		Calculated:	
Density L|V				
Viscosity L|V				
Vapor Molecular Weight				
Specific Heat L|V				
Thermal Conductivity L|V				
Fouling Resistance IS				

AIR SIDE		
Temperature In - Out		(Min. Amb. =)
Elev	Air Quantity	

CONSTRUCTION	
Design Press. - Temp.	(MDMT =)
Codes	
No. Bundles - Tubes / Bundle	
No. Rows - Passes / Bundle	
Header Type - Corrosion Allowance	
Header Material	
Nozzles In - Out - Series / Bundle	
Tube Material	
Tube O.D. - BWG - Length	
Fin Material - Bond Type	
Fin Height - No. Fins / Inch Pitch	

FAN	MECHANICAL EQUIPMENT		MISCELLANEOUS
Mfr.	SPEED REDUCER	DRIVER	
Model	Type	Type	
No. of Fans	Mfr.	Mfr.	
Dia. No. of Blades	Model No.	/Driver	
RPM Pitch °	Ratio	Enclosure	
Tip Speed ft/min		Service Factor	
		RPM	
Blade Material	Support:	V P C	
Static Pressure			
SPL/Fan @ = dBA			

3.12.2 Pumps

Typical pump specification sheet used by suppliers.

				CENTRIFUGAL PUMP		
				CLIENT	EQUIP. NO	PAGE
REV	PREPARED BY	DATE	APPROVAL	W.O.	REQUISITION NO.	SPECIFICATION NO.
0						
1				UNIT AREA	PROCURED BY	INSTALLED BY
2						

1	**General**				
2	Fluid Service		Pump Manufacturer		
3	Number Required		Model Number		
4	Pump Type		Size		
5	Location				
6	**Process Data**				
7	Fluid Pumped				
8	Normal Flow Rate	L/min	Corrosive or Non-Corrosive		
9	Design Flow Rate	L/min	Corrosive Compounds		
10	Pumping Temperature	deg C	Solids		
11	Vapor Pressure @ P.T.	kPa			
12	Viscosity @ P.T.	cP	Hazards		
13	Specific Gravity @ P.T.	water=1			
14	**Pumping Conditions**				
15			Suction		Discharge
16	Terminal Pressure	kPa			
17	Static Head	m / kPa			
18	Equipment Loss (see sketch)	m / kPa			
19	Line Loss (per 100 equiv ft)	m / kPa			
20	Equiv. Line Length	m			
21	Safety Factor in Line Loss (%)				
22	Total friction Loss	m / kPa			
23	Control Valve	m / kPa			
24	Net Suction Pressure	kPa			
25	NPSH available	m / kPa			
26	Total Discharge Pressure	kPa			
27	**Differential Pressure (TDH)**	m / kPa			
28	**Design Flow Rate**	L/min			
29					
30	**Mechanical Data**				
31		Type	Material	Sketch	
32	Pump Head				
33	Impeller				
34	Shaft				
35	Seal				
36	Baseplate				
37					
38	Connections	Suction	Case Drn.		
39		Discharge			
40	**Electrical Data**				
41	Area Classification	Class	Group	Division	Enclosure
42	Power	Volts	Phases	Cycles	Frame
43	Horsepower	Calculated -	BHP	Nominal	RPM
44		Efficiency			
45	Notes				
46	Motor must be non-overloading at runout				
47	Fabricated baseplate sized to catch drips from pump, seal, or other components				
48	Provide 1-inch lip around baseplate so drips are contained				
49	Provide 3/4" hose bibb at low point of baseplate containment, with cap				
50					
51					
52					

3.12.3 Turbine Specification Requirements

An Example of a steam turbine data sheet by API standards, which can be used to develop an ORC turbine datasheet.

TURBINE DATA SHEET SI UNITS

JOB NO. ITEM NO.
PURCHASE ORDER NO.
SPECIFICATION NO.
REVISION NO. DATE
PAGE OF BY

APPLICABLE TO: PROPOSAL PURCHASE AS-BUILT
FOR UNIT
SITE SERIAL NUMBER
SERVICE NUMBER REQUIRED
MANUFACTURER MODEL DRIVEN EQUIPMENT ITEM NO.
DRIVEN EQUIPMENT TYPE: COMPRESSOR GENERATOR OTHER
NOTE: INFORMATION TO BE COMPLETED BY: PURCHASER MANUFACTURER PURCHASER OR MANUFACTURER

PERFORMANCE

OPERATING POINTS	SHAFT		INLET			INDUCTION/EXTRACTION			EXHAUST		
AS APPLICABLE	POWER kW	SPEED r/min	FLOW kg/h	PRESS kPa	TEMP °C (TT)	FLOW kg/h	PRESS kPa	TEMP °C (TT)	PRESS kPa	TEMP °C (TT)	ENTHALPY kJ / kg
RATED											
NORMAL (3.26)(6.1.4)											
MINIMUM											

STEAM RATE, kg/kW.h (3.44): NORMAL RATED
POTENTIAL MAXIMUM POWER(3.30)
INDUCTION CONTROLLED UNCONTROLLED
EXTRACTION CONTROLLED UNCONTROLLED

STEAM CONDITIONS

		INLET	EXHAUST	EXTRACTION INDUCTION	EXTRACTION INDUCTION	EXTRACTION INDUCTION
FLOW kg/h	MAXIMUM					
	NORMAL					
	MINIMUM					
PRESSURE kPa	MAXIMUM					
	NORMAL					
	MINIMUM					
TEMPERATURE °C (TT)	MAXIMUM					
	NORMAL					
	MINIMUM					

SITE AND UTILITY DATA

LOCATION:
INDOOR HEATED UNDER ROOF OUTDOOR
UNHEATED PARTIAL SIDES GRADE MEZZANINE
OTHER:
WINTERIZATION REQUIRED TROPICALIZATION REQ'D
LOW TEMPERATURE CORROSIVE AGENTS
ELECTRICAL AREA CLASSIFICATION:
CLASS GROUP DIVISION
ZONE GROUP TEMPERATURE RATING:

SITE DATA:
ELEVATION m BAROM. PRESS kPa
WINTER TEMP. °C SUMMER TEMP. °C
REL. HUMIDITY % DESIGN WET BULB °C
UNUSUAL CONDITIONS: DUST FUMES
OTHER

UTILITY CONDITIONS:
AUXILIARY STEAM: MAX NORM MIN
INITIAL PRESS.(kPa))
INITIAL TEMPERATURE, °C (TT)
EXH. PRESS. (kPa)
INST. AIR (kPa): NORM MIN MAX
INSTRUMENT AIR DEW POINT: °C

ELECTRIC:	DRIVERS	HEATING	INSTRUMENT/ CONTROL	ALARM/ SHUTDOWN
VOLTS				
PHASE				
HERTZ				
kW AVAILABLE				

COOLING WATER:
INLET TEMPERATURE: °C MAXIMUM RETURN °C
PRESS. NORM.: kPa DESIGN kPa
MINIMUM RETURN PRESSURE: kPa
MAXIMUM ALLOWABLE PRESS. DROP: kPa
WATER SOURCE
VELOCITY, m/s: MIN MAX
FOULING FACTOR: m^2-K/kW

UTILITY CONSUMPTION:
COOLING WATER: m^3/h INST. AIR m^3/h
AUX. STM: NORMAL kg/h MAXIMUM kg/h
AUX. DRIVERS: ELECTRIC kW STEAM kW
HEATER(S): kW OTHER:

REMARKS:

3.12.3.1 List of other considerations important for a turbine and generator [65]

- Manufacturer
- Overall Dimensions and weights (approx.)
- Length (above Floor)
- Width
- Height Above floor
- Minimum distance required from floor line to CL crane hood to lift largest piece
  - During Erection
  - Specify which piece
  - After Erection
  - Specify which piece
- How Far Forward from front end of turbine below floor does oil tank extend
  - Does purchaser have flexibility in locating oil tank below floor
  - If yes what are the approximate limitations
- What are pulling dimensions from CL of generator for removing generator field
  - Straight pull
  - Canted
- Approximate dimensions of components
  - Oil tank (including oil coolers)
  - Seal Oil control unit (if applicable)
  - H_2 cabinet (if applicable)
  - Excitation cubicle
- Assembled weight of turbine complete without external accessories
  - Shipped Assembled
- Assembled weight of generator complete without external accessories
  - Stator weight
  - Shipped Assembled
- Weight of turbine rotor
- Weight of generator rotor
- Heaviest piece and weight to be handled during erection
- Heaviest piece and weight to be handled after erection

Turbine Generator Features

- Type of Turbine
  - Impulse
  - Reaction
  - Combination Impulse – Reaction
- Number of Stages
- Stop or trip throttle valves
  - Number
  - Where located
- Casing
  - Method of support
  - Material
  - Number of Inner shells
  - Methods of employed in construction and/or operating procedure to reduce thermal stresses
  - Size of openings
    - Inlet
    - Exhaust
- Turbine Rotor
  - Material
  - Built-up wheels, solid, or combination
  - Blading fastening method
  - Blading Material
  - Banding of blading (yes or no)
- Stationary hydrocarbon path
  - Blade rings or diaphragms
    - Nozzle material

    - Method of support
    - Type of interstate seals
- Shaft sealing system
    - Type of system
    - Sealing fluid
- Bearings
    - Number of journal bearings
        - Turbine
        - Generator
    - Type of journal bearings
    - Type of thrust bearing
    - Dummy piston required
    - If dummy piston is required, is there pro-vision to move rotor axially under load
    - Are sight flows provided
- Oil system
    - Capacity of oil tank (litres)
    - Number and type of pumps on oil tank
    - Type of main pump
    - Are pressurized oil feed lines guarded
- Oil coolers
    - Number
    - Total cooling capacity (%)
    - Tubing diameter and thickness
    - Tubing material
    - Required cooling water flow rate at 35°C (max temp)
- Governing
    - Type of system
    - Manufacturer
    - Maximum predicted speed under rejection of maximum load
    - Can emergency governor be tested
- How is the generator core mounted
- Number of generator coolers
    - How mounted
    - % load of one cooler out of service
- Armature winding material
- Armature slot material
- Method of fastening end turns of the armature bards
- Method of balancing generator rotor
- Armature slot wedge material
- Rotor Wedge Material
- Installation data
    - Will turbine be shipped assembled
    - Will generator be shipped assembled
    - Will oil tank and coolers be shipped assembled
    - Are electrical connections made up to junction boxes at the factory or in the field
    - What portion of oil piping is pre-fabricated in the factory

Detailed Design

Introduction – Detailed Design

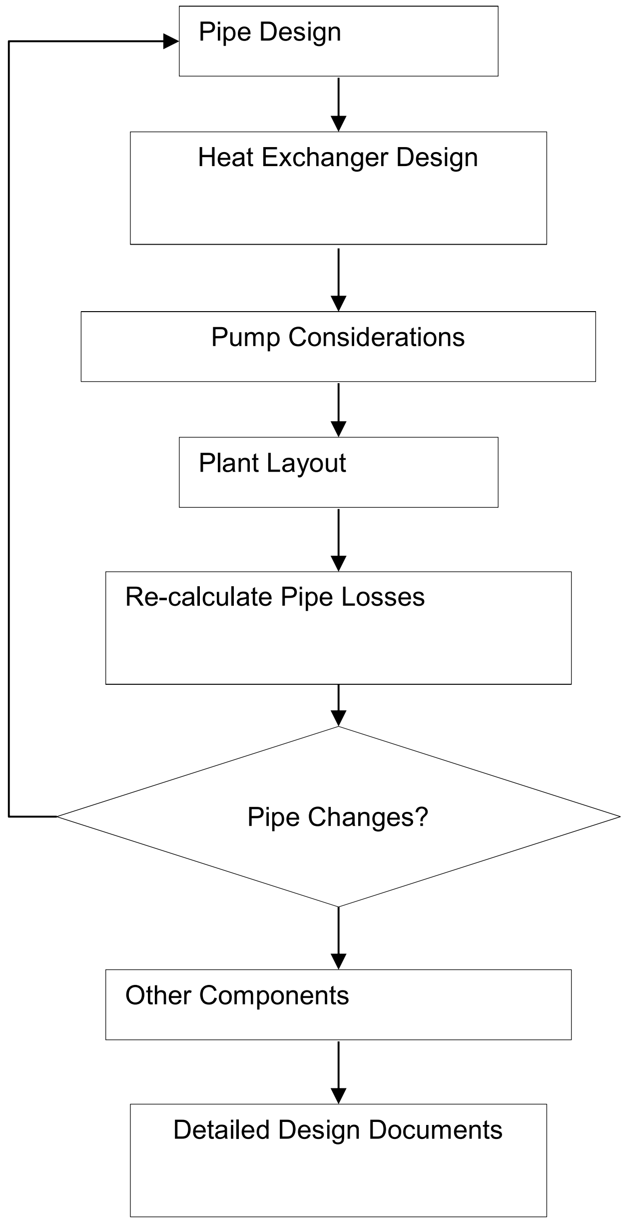
The detailed design section is only commenced once the feasibility section is complete and the cycle conditions are known. The detailed design section will look at the processes and standards used to design the main components of the ORC. The pump and turbine detailed design is not included because it is likely these components will be procured from a supplier. Or for the case of the turbine the detailed design will be done by a turbine design company.

The intended audience for the detailed design section is an engineer with a technical background that can understand the process for each component design and are aware of the appropriate standards for their country.

Detailed Design – Overview

1. The first stage of the detailed design process is the initial piping design to estimate connection sizes for each of the components and potential losses in the system. This design considers losses and estimated cost of the pipework.
2. Heat Exchanger Design - The heat exchanger design is the most critical design section included in this section. This covers the thermal design of the different heat exchangers used in an ORC and the appropriate standards for mechanical design are listed.
3. Pump Consideration - The pump is taken into consideration with better pressure drop knowledge of the heat exchanger and the require head to ensure no cavitation.
4. Re-Calculate pipe losses - With rough sizing of components the plant layout can be determined. This will give a better understanding of the amount of pipework required. The plant layout will also determine the height of the condenser to provide sufficient head for the pump. The pipe size is reconsidered and if there is a change the other component design will also change.
5. Other Components – This section covers some of the important considerations for other components and instruments needed for an ORC.
6. Detailed Design Documents – This section provides some of the available spec sheets for the components in and ORC.

Detailed Design Process



4 Detailed Design

4.1 Scope

This detailed design section looks at certain design aspects of components in an ORC. A significant focus is on the heat exchangers design.

4.2 Pipe Design

The first step for the overall plant layout and design is to guess initial pipe diameters. The chosen pipe diameter impacts the performance and cost of the system. Larger pipe diameters reduce pressure losses between components but also increase the overall cost of the system as they require more material. The pipe diameter will also determine size of the connection for each component.

The pipe diameters will also impact the cost of the valves installed throughout the system.

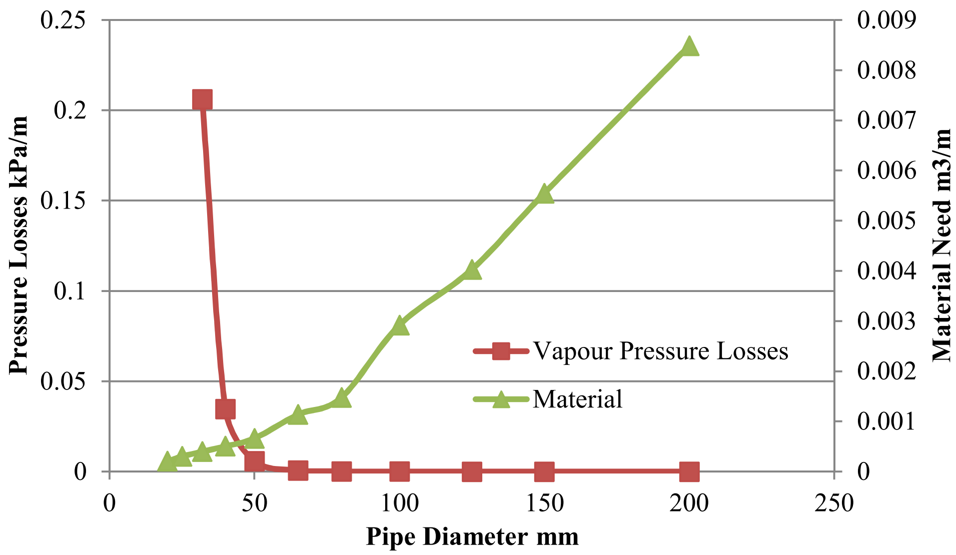


Figure 61 - Pipe Trade off graphic

4.2.1 Pressure Losses

The pressure losses in the pipe work impact the performance of the ORC and is the first step to select the optimum pipe size for the ORC. There velocity estimates to determine appropriate pipe size.

4.2.1.1 Recommended Velocities

4.2.1.1.1 Liquid

The recommended velocities for low viscosity fluids in a pipe line is between 0.4 m/s for narrow pipes and 2.3 m/s for larger diameter pipes (12mm – 600mm) [115]. A boiling fluid's velocity is between 1 and 2 m/s for pipes between 25mm and 300mm[116]. The maximum recommended pipe velocity is 4.5 m/s[71].

4.2.1.1.2 Vapour

Vapours are significantly less dense so will require much larger pipe diameters to achieve the recommend velocities. The recommend velocity entering and exiting a turbine is between 15 – 21 m/s [21]. However, the density will change across the turbine and the outlet will be significantly larger and is economical to minimise the distance between the turbine and the condenser. The minimum recommended gas velocity is between 3 and 4.5 m/s.

4.2.1.2 Major Losses

The Darcy - Weisbach equation is used to determine the major losses through the pipes. The overall length of the pipes for each section is not known at this point; therefore, specific pressure losses can be used to understand the performance of each pipe section.

$$\Delta p_{major} = f_D(\frac{L}{D})(\frac{\rho V^2}{2})$$

f_D darcy friction factor

L Length- Use 1 meter lengths

D Diameter m

ρ density kg/m^3

V Velocity m/s

The Darcy friction factor is determined from the moody diagram.

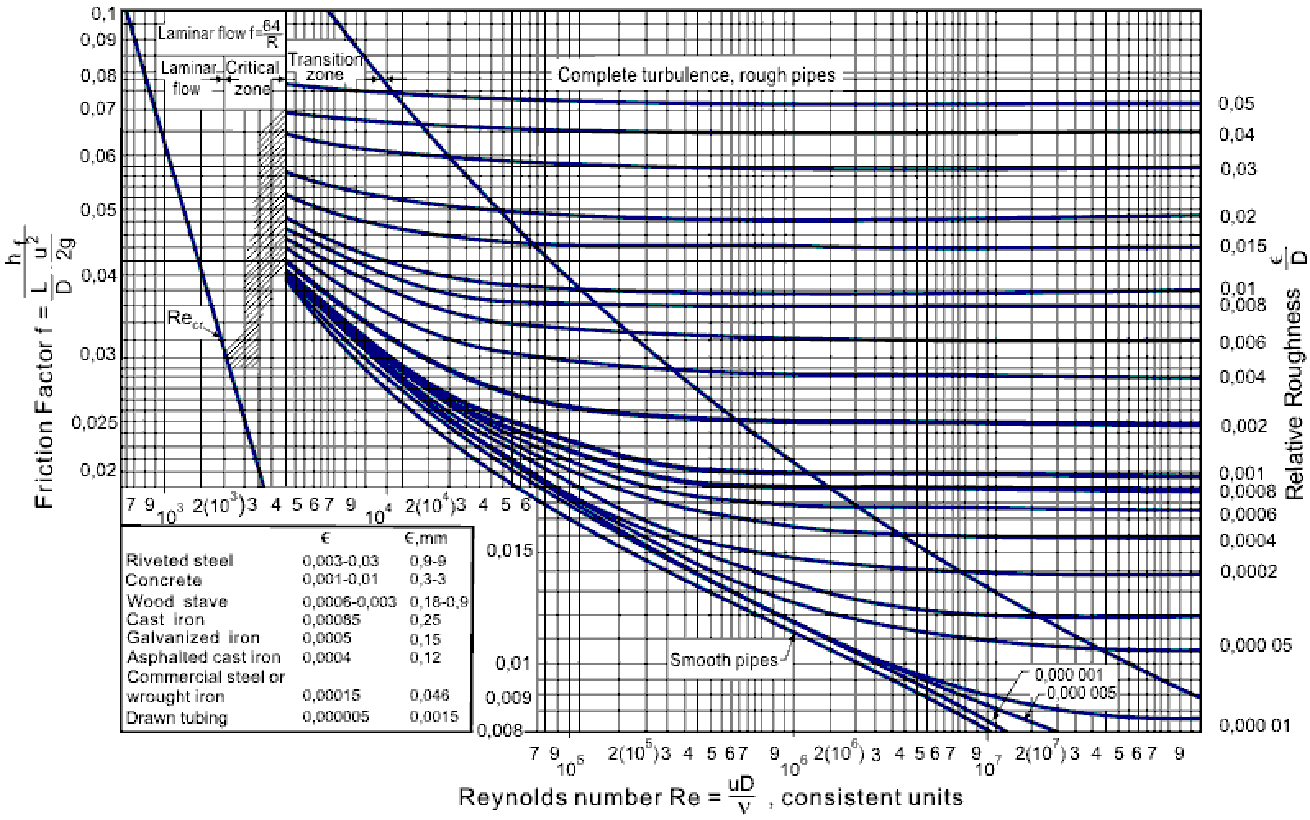


Figure 62 - Moody Diagram - Credit Streeter [117]

The specific pressure losses are determined for each fluid state between components: *after pump before pre-heater, after evaporator before turbine, after the turbine before the condenser, and after the condenser before the pump.*

4.2.1.3 Minor Losses

The minor losses are the losses in the pipe bends, connections, and valves. It is unknown how many bends and valves will be in the system at this point. There should be at least one isolation valve between each component and a control valve before the turbine.

$$\Delta p_{minor} = \xi \frac{\rho V^2}{2}$$

ξ pipe component coefficient – Depnends on the type of component

4.2.2 Pipe Thickness

The second issue that should be explored with piping is investigating the required thickness of the pipe to withstand the pressure.

4.2.2.1 ASME B31.1 Piping Standard – Power Piping

ASME B31.1[118] is used when considering any power piping. It prescribes the minimum requirements for design, materials, fabrication, erection, test, inspection, operation, and maintenance of piping systems typically used for power.

If the reader is only going to use one piping standard this is the recommended standard.

This piping code is the most popular standard for pipeline design. The following sections are other standards including in the ASME B31 series.

4.2.2.2 Basic Pipe Stress Formula

The basic pipe stress formula relies on the allowable hoop stress within the pipe.

$$t = \frac{Pd_o}{2(H_s + P)}$$

H_s is the hoops stress in the pipe wall kPa

t pipe thickness mm

P Design pressure kPa

d_o outside diamter of pipe mm

4.2.2.3 ASME B31.3 Piping Standard – Strict Approach

The ASME B31.3 [119] is a standard that calculates the required pipe thickness for a certain pressure and material. This standard is used for performance equipment that requires accurate pipe thickness estimates.

$$t = t_e + t_{th} + \left[\frac{Pd_o}{2(SE + PY)}\right]\left[\frac{100}{100 - T_{ol}}\right]$$

t is the minimum design wall thickness mm

t_e is the corrosion allowance mm

t_{th} thread or groove depth mm

P allowable internal pressure in pipe kPa

d_o outside diameter mm

S allowable stress for pipe kPa

E longitudinal weld joint factor (1 for seamless, 0.95 electric fusion, double butt m straight or spiral seam. 0.85 electric resistance weld, 0.60 furnace butt weld.

Y derating factor (0.4 for ferrous material's operating below 480°C)

T_{ol} manufactures allowable tolerance %(12.5 pipe up to 508mm 10 for >508)

4.2.2.4 ASME B31.4 Piping Standard – Basic Approach

The ASME B31.4 [120] standard is used to calculate the pipe thickness average equipment without needed a detailed pipe thickness analysis.

$$t = \frac{Pd_o}{2(FES_Y)}$$

S_Y minimum yield stress for pipe kPa

F derating factor, 0.72

4.2.2.5 ASME B31.8 Piping Standard – Moderate Approach

The ASME B31.8 [121] standard is a moderate method for calculating the pipe thickness between the two other approaches.

$$t = \frac{Pd_o}{2(FETS_Y)}$$

F is a design factor

T is the derating factor

Both these factors are taken from tables in the standard.

4.2.3 Other Pipe Considerations

The pipe diameter will impact the cost of each valve in the system. It is recommended that shutoff valves are installed between each component as this reduces downtime during maintenance but will also increase costs for the system.

The pipe work connections will either be welded or flanged connections. Welded pipework is cheaper without the need for flanges; however, maintenance is difficult with welded pipe work.

Flanged connection provides simpler construction and more flexibility during maintenance. However, flanged connections can be more expensive and increase the chance of working fluid leaking from the system.

4.3 Heat Exchanger Design

The two main sections of heat exchanger design are the thermal design and the mechanical design. This section only covers the thermal design of the heat exchanger. The mechanical design is covered by a number of international standards. There is heat exchanger software that can do both the thermal and mechanical design of heat exchanger.

The heat exchangers covered in this section are the shell and tube type, air cooled heat exchangers, and the basics of plate heat exchangers.

4.3.1 Shell and Tube Heat Exchanger Design

The pre-heater and recuperator consider use single phase shell and tube heat exchanger design. The evaporator is a shell and tube boiler and the condenser can also be a shell and tube heat exchanger if cooling water is used as the cold fluid.

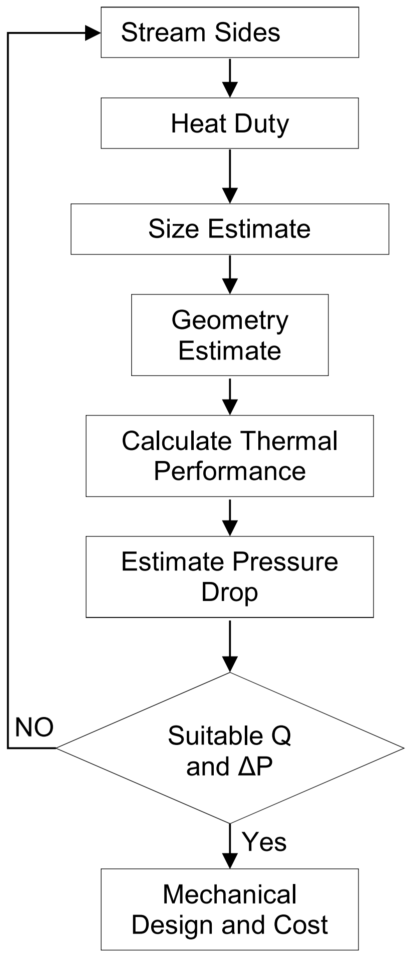


Figure 63 - Shell and Tube Heat Exchanger Design – Credit Heat Exchanger Handbook [76]

4.3.1.1 Single Phase

A single phase shell and tube heat exchanger will follow the Kern or Bell Delaware methods to estimate the overall heat transfer coefficient for given heat exchanger geometry.

4.3.1.1.1 Initial Inputs and Estimates

The first step of the heat exchanger design is to determine the shell side and tube side fluid. A geothermal system will always have the geothermal fluid tube side as it is easier to clean[21]. The other single phase system is the recuperator and the high pressure fluid should be tube side to minimize component cost [76].

The flow rates, inlet temperature, outlet temperature, and heat duty for the heat exchanger must be known. These are used to estimate the area required from the assumed U value from the pre-feasibility study using the standard heat exchanger equation.

$$A = \frac{Q}{LMTD(U)}$$

A is the overall heat transfer area m^2

Q is the heat duty of the system kW

$LMTD$ is the log mean temperature different K

U is the estimated overall heat transfer coefficient kW/m^2K.

The heat transfer area calculated is then used to estimate the geometry of the shell and tube heat exchanger.

4.3.1.1.2 Geometry

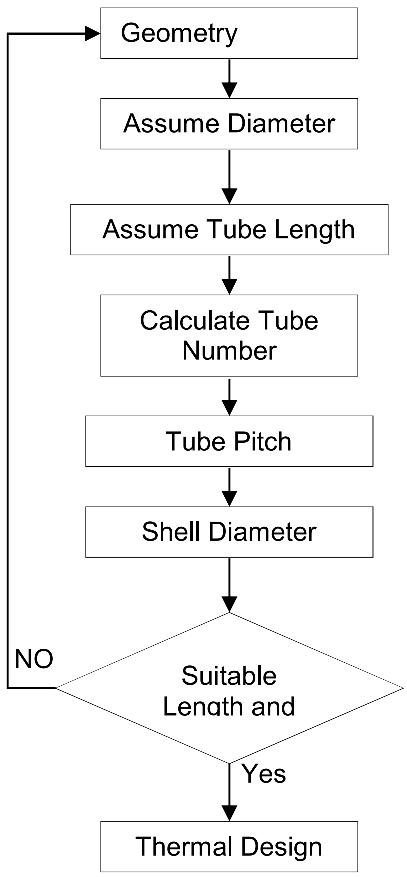


Figure 64 - Method to Calculate Heat Exchanger Geometry

The two most commonly used tubes are 19mm and 25mm OD[71]; however, the other standard OD tubes 6.4, 9.5, 12.7, 15.9, 22.2, 31.8, and 50.8mm can also be considered. The thickness of the pipe will depend on the pressure of the brine.

Once the diameter of the tube is assumed the standard lengths these tubes come in are 2.44, 3.05, 3.66, 4.88, and 6.10m. 6.10m is the most common tube length [71] and should be the first length used to estimate the number of tubes for the heat exchanger as small shell diameters are cheaper heat exchangers. However, if there are length constraints for the ORC a shorter tube length will be used instead.

$$N_t = \frac{A}{dL}$$

N_t number of tubes

A area m^2

d tube diameter m

L tube length m

The shell diameters can be estimated using the following two equations for the two types of tube pitch, triangular and square, with an equal 1.25x pipe diameter pitch[71]. The shell diameter is guessed and typically the optimum design has a shell diameter between 5-10 times less than the tube length.

Triangular

$$N_t = 1298 + 74.86C + 1.283C^2 - 0.0078C^3 - 0.0006C^4$$

$$C = 0.75\left(\frac{D_s}{d}\right) - 36$$

D_s is the guessed bundle diameter – 'initial guess L/7'

Square

$$N_t = 593.6 + 33.36C + 0.3782C^2 - 0.0012C^3 + 0.0001C^4$$

$$C = \left(\frac{D_s}{d}\right) - 36$$

Then the minimum outer tube to shell clearance is 13mm. Finally the inner shell diameter is known and can be selected from the standard shell diameters in table 17.

Table 17 - Standard Shell Diameters

100	125	150	200	250	300	350
400	450	500	600	650	700	750
800	850	900	100	1050	1100	1150<

The final estimate for the shell side calculation is to determine the baffle spacing for the tubes. A rule of thumb for baffle spaces is a maximum of 1/5 the shell diameter and a minimum of 50.8mm. Alternatively, the maximum unsupported tube length for common tubes is in table 18.

Table 18 – Common Unsupported Tube Lengths

Common Tube Diameter (mm)	15.88	19.05	25.40	31.75
Maximum Unsupported Tube Length (mm)	1321	1524	1880	2235

There is now enough information to calculate the thermal performance of this heat exchanger.

4.3.1.1.3 Thermal Design

Detailed thermal design for shell and tube heat exchangers typical uses the Bell-Delaware method which uses a number of factors to develop a reliable heat exchanger performance prediction. References [71, 122] are a reliable source to follow the Bell-Delaware method.

This standard will use the Kern method because it's simpler and still results in a reliable estimate without the use of empirical tables necessary for the Bell-Delaware method.

The three areas of concern for the thermal design are the tube side heat transfer coefficient, shell side coefficient, and the fouling factor.

4.3.1.1.3.1 Tube Side Heat Transfer

The tube side heat transfer equation below is the equation for the heat transfer from a single tube.

$$h_{ts} = 0.023\frac{k^{1-n}j^{0.8}c_p^n}{\mu^{0.8-n}d^{0.2}}$$

h_{ts} tube side heat transfer coefficient W/m^{2}K

d is the hydraulic diameter m

k is the thermal conductivity of the bulk fluid W/m-K

μ viscosity kg/m-s

j mass flux kg/s-m^2

c_p isobaric heat capacity J/kg-K

n 0.4 for heating (wall hotter than bulk fluid) 0.33 for cooling (wall cooler than bulk fluid)

4.3.1.1.3.2 Shell Side Heat Transfer Coefficient – Kern Method

The kern method is in a number of references and this document uses the method from [97, 123].

The first step in the Kern method is to estimate the rough cross flow area of the tube bundle.

$$A_s = \frac{(p_t - d_o)D_s l_B}{p_t}$$

p_t Tube pitch - 1.25 times the tube OD

d_o Tube diameter m

D_s Shell Diameter m

l_B Baffle Spacing m

The rough cross flow area is then used to determine the mass flux of the working fluid across the tube bundle.

$$G_s = \frac{\dot{m}_s}{A_s}$$

G_s is the mass flux kg/s-m^2

$\dot{m}_s$ is the shell side mass flow rate kg/s

The mass flux of the fluid through the tube bundle determines the velocity of the fluid in the shell. The optimum fluid velocity is roughly 1.2 m/s; generally faster velocities improve the heat transfer coefficient but impact the pressure losses. If the velocity is significantly different from this value the shell diameter should be changed.

$$u_s = \frac{G_s}{\rho}$$

u_s is the average velocity m/s – check velocity 1.2 roughly

ρ is the density kg/m^3

The next step is to calculate the shell side hydraulic diameter of the heat exchanger.

Square Pitch

$$d_e = \frac{4\left(\frac{p_t^2 - \pi d_o^2}{4}\right)}{\pi d_o}$$

Triangular Pitch

$$d_e = \frac{4\left(\frac{p_t}{2} \times 0.87 p_t - \frac{1}{2}\pi \frac{d_o^2}{4}\right)}{\frac{\pi d_o}{2}}$$

d_e is the hydraulic diameter m

The Reynolds number is then calculated

$$Re = \frac{G_s d_e}{\mu}$$

Re Reynolds number

Finally a baffle factor j_h is looked up depending on the baffle type used [76]. This factor is used to calculate the shell side heat transfer coefficient.

$$u = \frac{h_s d_e}{k_f}$$

$$\frac{h_s d_e}{k_f} = j_h Re Pr^{\frac{1}{3}} (\frac{\mu}{\mu_w})^{0.14}$$

μ_w viscosity at tube wall kg/m-s

k_f conductivity of the shell fluid W/m-K

Pr is the Prandtl number

Now the tube side and shell side heat transfer coefficients are known the overall heat transfer coefficient is calculated. The geothermal fouling factors are also included.

$$\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_{ts}} + R_f$$

U overall heat transfer coefficient W/m^{2}K

R_f fouling factor m^{2}K/W

The new heat transfer coefficient should be checked to see if more or less area is required for the heat exchanger heat duty. If the heat transfer coefficient does not fulfil the required duty then the heat exchanger needs to be larger.

The final check for the shell and tube heat exchanger is to check the pressure drop.

$$\Delta p_s = 8 j_f \left(\frac{D_s}{d_e}\right) \left(\frac{L}{l_B}\right) \frac{\rho u_s^2}{2} \left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

l_B is baffle spacing m

j_f is the friction factor

If the pressure drop across the heat exchanger is too large then changes must be made to the geometry and the entire heat exchanger calculation must be redone.

4.3.1.2 Phase Change Heat Exchanger

The design of a phase change heat exchanger is not as accurate as the single phase design because calculating boiling and condensing correlations for large heat exchangers is still very empirical and there is no accurate analytical method to confidently calculate phase change heat transfer coefficients. The following sections use recommended methods from the heat exchanger design handbook [76].

4.3.1.2.1 Evaporator

The evaporator method used here is from the heat exchanger handbook [76] for a kettle type reboiler, which is different to the type of evaporator that would be used in an ORC but the correlations used are accurate enough. The process to follow for an evaporator is shown in figure 65, the geometry and tube side heat transfer follows the same process as the previous section. The geometry is of the tubes in contact with the boiling fluid and a further single phase calculation is needed to estimate how many more tubes are required for sufficient super heating.

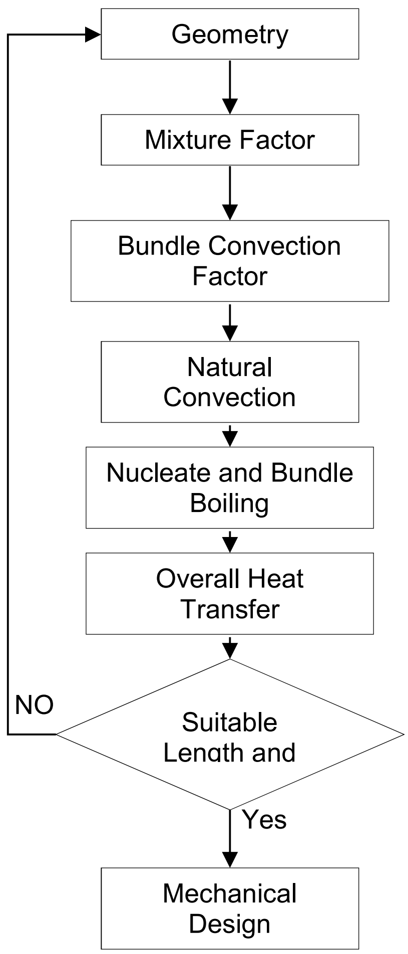


Figure 65 - Evaporator Design Method

4.3.1.2.1.1 Mixture Factor

The mixture factor is only considered when a mixture working fluid is used in the ORC. **A pure fluid will result in a mixture factor of 1.**

$$F_c = \frac{1}{1 + 0.023\dot{q}^{0.15}BR^{0.75}}$$

F_c Mixture factor

$\dot{q}$ Heat flux W/m^2 – ca be estimated using the area assumed and the heat duty Q/A

BR boiling range difference between dew and bubble point. This only applies if a mixture is used otherwise F_c is equal to 1

Bundle Convection Factor

The convection takes into account the bundle geometry's impact on the boiling in the evaporator.

$$F_b = 1 + 0.1\left[\frac{0.758D_b}{C_1(\frac{p_t}{d_o})^2 d_o}\right]$$

$C1$ is 1 for 90 and 45° layouts and 0.866 for 30 and 60° layouts

D_b bundle diameter m

4.3.1.2.1.2 Natural Convection Coefficient

A natural convection factor h_{nc} is assumed to be between 250 for hydrocarbons and a maximum of 1000 for water. The heat exchanger handbook [76] provides more accurate correlations but the natural convection coefficient is not critical for heat exchangers unless the difference between the wall temperature and the bulk flow is less than 4K across the whole heat exchanger.

$$h_{nc} = 250\ W/m^2K$$

4.3.1.2.1.3 Single Tube Nucleate Boiling

The nucleate boiling coefficient is calculated for a single tube initially and the bundle correction factor is used to determine the bundle boiling coefficient.

$$h_{nb1} = 0.00417p_c^{0.69}\dot{q}^{0.7}F_p$$

$$F_p = 1.8\left(\frac{p}{p_c}\right)^{0.17}$$

p is pressure and p_c is the critical pressure

4.3.1.2.1.4 Bundle Average Boiling Heat Transfer Coefficient

The heat transfer coefficient for a bundle fully submersed in the boiling fluid.

$$h_b = h_{nb1} F_b F_c + h_{nc}$$

4.3.1.2.1.5 Overall Heat Transfer Coefficient

$$\frac{1}{U} = \frac{1}{h_b} + \frac{1}{h_{tube}} + R_{Fouling}$$

Boiling side fouling is around 0.00035m^{2}K/W

4.3.1.2.1.6 Check Size

Same as the single phase heat transfer process the new area must be calculated with the new heat transfer coefficient. If more area is required then the geometry must be changed.

4.3.1.2.1.7 Check Heat Flux

The heat flux in the heat exchanger should be checked to determine whether vapour lanes should be designed into the bundle.

$$q_{1,max} = 376 p_c \left(\frac{p}{p_c}\right)^{0.35} (1 - \frac{p}{p_c})^{0.9}$$

$q_{1,max}$ maximum heat flux of 1 tube

4.3.1.2.1.8 Bundle Geometry Factor

$$\psi_b = \frac{\pi D_b L}{A}$$

L bundle length

$$\phi_b = 3.1 \psi_b$$

$\phi_b \leq 1.0$ Limit

4.3.1.2.1.9 Max heat flux

$$\dot{q}_{max} = q_{1,max} \phi_b$$

$$\dot{q}_{design} \leq 0.7 \dot{q}_{max}$$

4.3.1.2.1.10 Vapour Lane Requirements

IF $\phi_b < 0.1$ and $\dot{q}_{design} > 0.5\dot{q}_{max}$

Vapour Lanes are required

4.3.1.2.2 Condenser

The shell and tube type condenser typically used for an ORC will be a cross flow type condenser – TEMA X model. The Heat exchanger handbook [76] had a selection guide and performance predictions of other types of condensers; however, the X type is the most appropriate.

Once again the basic geometry is required for the heat transfer coefficient. Tube side heat transfer coefficient is calculated using the same process as the single phase heat transfer heat exchanger.

4.3.1.2.2.1 Condensing Coefficient

The condensing coefficient is affected by the vapour shear in the heat exchanger. Generally if the condensing vapour is shell side the vapour shear has a minimal impact on the condensing coefficient. However if the velocity in the shell is higher vapour shear has more of an impact. There a different methods to determine heat exchanger coefficients depending on the levels of vapour shear in the heat exchanger.

No Vapour Shear

$$\frac{h_c d_o}{k} = 0.725\left[\frac{d_o^3 \rho_l (\rho_l - \rho_g) g \Delta h}{k\mu(\Delta T_a)}\right]$$

$\Delta T_a = T_{sat} - T_w$

Δh latent heat of condensation kJ/kg

Moderate vapour shear

Average coefficients to account for heat transfer through the bundle

$h_{c,b} = h_c N^{-1/6}$

N is number of tubes in the vertical path

High Vapour Shear

$$\frac{h_c d_o}{k} = 0.3Re^{0.6}Pr^{0.4}\sqrt{\frac{\rho_l}{\rho_g} + 1}$$

$Re = \frac{d_o G_s \rho_l}{\mu \rho_g}$

4.3.1.2.2.2 Overall Heat Transfer

The new overall heat transfer coefficient is calculated

$$\frac{1}{U} = \frac{1}{h_t} + \frac{1}{h_c} + R$$

4.3.1.2.2.3 Check Size

This is used to check the overall size of the heat exchange area and if any adjustments must be made to the geometry.

The other check for the condenser is the difference between the vapour and the wall temperature

$$\Delta T_c = (\frac{U}{h_c})(T_v - T_w).$$

T_v vapour temperature K

T_w wall temperature K

If ΔT_c is not between 0.5 to 2 times larger than ΔT_a the wall temperature assumption temperature must be changed and the calculations repeated.

4.3.1.3 Mechanical Design and Standards

The mechanical design of shell and tube heat exchangers is a critical process that must follow appropriate standards for the county of operation. Standards of Tubular Exchanger Manufactures association (TEMA) [124] is designed to specifically supplement the ASME boiler and pressure vessel code section VII. A large portion of the TEMA standard can also be used to supplement other pressure vessel codes.

Below lists a number of standards associated with pressure vessel and shell and tube heat exchanger design. The reader is responsible for knowing which standard apply to their country.

ASME VII/PED [125]

BS 5500 [126]

ISO Standards 27.060.30

ISO 1129:1980 – Steel tubes for boilers[127]

ISO 6758:1980 welded steel tubes for heat exchanger[128]

ISO 6759:1980 seamless steel tubes for heat exchangers[128]

ISO 16528-1:2007 Boilers and Pressure Vessels – Part 1[129]

ISO 16528-2:2007 Boilers and pressure vessels – Part 2[130]

TEMA is applicable to shell and tube heat exchangers with the following limitations[76]:

Shell diameter is less than 1524mm, Pressure does not exceed 21 MPa, and the product of the shell diameter and the pressure does not exceed 10500 mmMPa

These limitations are to limit the barrel thickness and stud diameter to 50mm and 75mm respectively.

There is a special section in TEMA titled "Recommended Good Practice" which provides designers guidance for shell diameters up to 2540mm.

The three mechanical classes in TEMA are

Class R: Generally sever requirements of petroleum and related processing applications
Class C: generally moderate requirements of commercial and general process applications
Class B: chemical process service

The most common shell types for geothermal applications are E (one pass shell) for the pre-heater and evaporator, and X (Cross Flow) for a water cooled condenser.

4.3.2 Air Cooled Condensers

Air cooled condensers are commonly only used in ORC operations that do not have a source of cooling water and there is no geothermal fluid available for makeup because all fluid for an ORC is reinjected. Air cooled condensers are the only type used in New Zealand ORC industry.

This section will outline the recommended process to size an air cooled condenser. This process is taken from the handbook of heat exchanger design [76].

4.3.2.1 Estimate Tube side Heat transfer

The tube side heat transfer only needs to be estimated for an air cooled condenser because the air side heat transfer is significantly less and is the dominating heat transfer coefficient.

The tube side heat transfer coefficient can be estimated to be between h_t 800-2200 W/m^{2}K this assumption can be calculated more accurately with the heat exchanger handbook; however, the accuracy has minimal impact on the overall result.

4.3.2.2 Tube Selection

The heat transfer coefficient will determine the type of tubes used in the ACC. The Heat exchanger handbook has the appropriate graphs to determine the required area to face area ratio of tubes relative to the heat transfer coefficients. Below are a number of optimum ratios for low, medium, and high heat transfer coefficients.

Low $h_t \approx 200$

$A/_{A_i} = 5.5$

Medium $h_t \approx 1000$

$A/_{A_i} = 14$

High $h_t \approx 5000$

$A/_{A_i} = 25$ (28 for L foot single)

The corresponding standard finned tube can then be selected from a manufacture who will supply the corresponding thermal and hydraulic data for the fin tube relative to air mass flow rate. This data is used to select the air mass velocity.

The air mass velocity is typically between 2-4 m/s.

The corresponding manufacture tube plots for overall heat transfer are used to find unique fin tube characteristics. However, typical U values for air flowing between 2 and 4m/s are 15 and 35 Wm^2K respectively. This is significantly lower than the tube side heat transfer because air side heat transfer is the controlling factor.

4.3.2.3 Calculate the Number of tube Rows

The number of tube rows can be calculated using geometry data from standard finned tube supplies.

$$a = \frac{T_{f,in} - T_{air,in}}{U\left(\frac{A}{S}\right)}$$

A/S is related to the selected tub geometry – 27.8 L foot single
A is area of the HX exposed to air m^2
S is face area of the tube m^2

$T_{f,in}$ working fluid inlet temperature °C

$n_r = C_1 a^{C_2}$

n_r number of rows

C_1 = 24 - Example Value for L Foot single

C_2 = 0.49 - Example Value for L foot single

Constants depend on the type of tube used

4.3.2.4 Calculating the required surface area

The following steps are to calculate the required surface area for the air cooled condenser according to the heat exchanger handbook [76].

4.3.2.4.1 Thermal Numbers

Product Thermal Number

$$\Phi_{prod} = \frac{T_{f,in} - T_{f,out}}{T_{f.in} - T_{air,in}}$$

Coolant Design Number

$$k = \frac{U\ A/S}{u\rho C_p}$$

u is air velocity m/s

ρ is density kg/m^3

NTU Number

$$NTU = n_r k = \frac{\Delta \mathrm{T}_{air}}{EMTD}$$

Optimum design typically has the NTU between 0.8 and 1.8

Coolant Thermal Number

$$\Phi_{air} = \frac{1 - \exp(-\tau(1 - e^{-NTU})}{\tau}$$

$$\tau = \frac{\Delta T_f}{\Delta T_{air}}$$

If $\Delta T_f = 0$

$$\Phi_{air} = \frac{\Delta T_{air}}{T_{f.in} - T_{air,in}}$$

4.3.2.4.2 Adjusted Mean Temperature

The cool and product thermal numbers are then used to estimate the corrected mean temperature difference.

$$\frac{EMTD}{T_{f.in} - T_{air,in}} = \frac{\Phi_{air}}{NTU}$$

4.3.2.4.3 Required Area

Surface Area

This information is then used to calculate required heat transfer surface area as well as a rough face area estimate of the air cooled condenser.

Surface Area

$$A = \frac{\dot{Q}}{U\ EMTD}$$

Face area

$$S = \frac{A}{\left(\frac{A}{S}\right) n_r}$$

4.3.2.5 Mechanical Design

The supplier of the tubes should design the tubes to the required pressure rating for pressure tubing. The relevant standards for air cooled heat exchangers are ISO -13706:2011[131] and AMSE – PTC 30 – 1991[132].

Other ACC Conditions

Other design considerations for air cooled condensers are listed below.
250 pa pressure rise in the fans
Fan tip speed limit – 61 m/s – This will also depend on the noise levels allowed
½ fan diameter between tubes and fans
Max dispersion angle of 45°
Most finned tubes are 25.4mm in diameter, 350-450 fins/m, 16 mm fins, angular pitch of 50.8-63.5, and made from aluminium.
Fan diameters are between 1.8 and 4.1

Typically 3 fans for each bundle of tubes
Tube rows between 3 and 6
Typically triangular pitch
Standard length 6.1-12.2
Height of system must be half the tube length above the ground to ensure inlet velocity is equal to the face velocity
Induced draft more appropriate for ORCs as it is designed to get the product closer to the ambient temperature
AISC standard for steel structure

4.3.3 Plate heat exchanger

Plate heat exchangers are available from a number of reliable manufactures. The required design specifications for a plate heat exchanger are the two fluids and the heat duty required. Alternatively, the heat exchanger handbook provides a more detailed analysis of the plate heat exchanger design. This process relies on empirical correlations to select the appropriate number of plates and passes appropriate for the heat duty and allowable pressure drop of the heat exchanger.

Plate heat exchanges have not been used extensively for geothermal applications because of silica blocking issues at the plate inlet manifold and between plates. Phase change plate heat exchangers are also uncommon.

ISO 15547-1:2005 – Plate type heat exchangers – Part 1: Plate and frame heat exchangers [133]

4.4 Pump Considerations

The most important parameters for selecting a pump are the flow rate, head, and preferred efficiency. The pipe design should also provide the pump supplier with connection sizes. The current Net Positive Suction Heat Available (NPSHA) is known by the different in the vapour pressure at the pump inlet condition and the pressure of the pump inlet. The hydrostatic head of the fluid also is taken into account if an air cooled condenser is used.

$$NPSHA = P(inlet) - P(Cond) + P(static)$$

$P(inlet)$ is the saturation pressure of the fluid at the sub cooled condition kPa

$P(Cond)$ is the pressure in the condenser kPa

$P(static) = h\rho g$ is the hydrostatic head of the fluid Pa

h is the height difference between the condenser and the pump – an air cooled condenser height should be half the length of the condenser pipes.

The pump chosen for the ORC operation must fulfil the required pressure increase and flow rate. This pump selection will provide the required NPSH to operate. There is a risk of cavitation if the NPSHA is less than the NPSHR.

$$NPSHA > NPSHR$$

If the NPSHR is greater than the NPSHA the engineer either must choose a different pump, put the pump in a lower well, increase the height of the condenser, or increase the amount of sub cooling.

A centrifugal pump with a constant speed drive must use a bypass to control the flow rate around the ORC and for low flow operation. The bypass allows the pump to still operate at the maximum efficiency point but delivering smaller amounts of fluid to the ORC while more fluid is circulated around the bypass [134]. The bypass also allows the pump to maintain the minimum recommended flow rate in certain situations. Alternatively variable speed drive pumps can be utilized to accommodate the required changes in the flow rate. Both of these additions will increase the cost of the pump.

A volumetric pump will also require a bypass to reduce load on the pump during start-up.

4.4.1 Pump Standards

The most common pump used in industry is the centrifugal pump. The following are a number of different standards associated with centrifugal pumps.

ANSI/API 610-1995 – Centrifugal pumps for general refinery service [135]

DIN EN ISO 5199 – Technical Specifications for centrifugal pumps[136]

ASME B73.1 – 2001 – Specification of Horizontal end suction centrifugal pumps for chemical process[137]

ASME B73.2 – 2003 – Specifications for vertical in-line centrifugal pumps for chemical process [138]

BS 5257:1975 – Specifications for horizontal end suction centrifugal pumps (16 Bar) [139]

4.5 Other Considerations

4.5.1 Plant Layout

At this point all the components have rough sizes and a basic plant layout will be drawn. Pipe work between components should be minimal to avoid excessive pressure losses in the pipe.

Minimum distance between turbine outlet and condenser / recuperator inlet.

4.5.2 Pipe Losses

The basic plant layout can be used to determine the minor and major pressure losses in the system. The pressures losses should be checked with a number of pipe diameters to ensure that the pipe diameter used is the best option.

4.5.3 Turbine Considerations

The turbine will likely be designed and manufactured by a turbine design company that has the knowledge, tools, and experience required to design a reliable turbine. A turbine should have adequate instrumentation before and after the turbine to measure its performance to ensure it meets the manufacturer's guarantees. A governing valve will also be required to control the flow into the turbine; however, the ORC turbine is expected to be a peak flow piece of equipment and run at full load most of the time.

There are a small number of standards that cover the general requirements and specification sheets of turbine design. The safety and performance of the turbine will be guaranteed by the suppliers.

ISO 14661:2000 - General Requirements [72]

ISO 14661:2000/Amd 1:2002 Turbine Data Sheets [140]

Turbine API standard 617[141] and 614[142]

A bypass is required for the turbine in to divert the flow into the condenser if there is a sudden drop in load which can result in turbine over speed. The bypass will allow the hot fluid to enter the condenser and avoid excessive heating on the high pressure side of the turbine. Isolation of the turbine is also a benefit if any maintenance is required.

4.5.4 Instrumentation and Valves

As mentioned early it is recommended that shutoff valves are installed between each component to help with plant maintenance and shut down times; therefore, each component can be isolated and the working fluid can be kept in specific parts of the system. This could improve the capacity factor of the plant and in the long run improve the power sales.

Temperature and pressure instrumentations are required between each of the main component to check the thermodynamic states of the ORC. Accurate pressure and temperature probes should be at the outlet of the turbine to ensure the quality of the vapour entering the turbine. Both static and dynamic pressures are required for the turbine. Level sensors are also important in the shell and tube evaporator to monitor the liquid level. A precise pressure differential can be used to monitor the liquid level in the evaporator. Alternatively, a number of liquid sensors can also be used in the pressure vessel or a site glass on the outside of the shell. A flow meter is also required to ensure the ORC is operating as expected. There are a number of flow meters available each varying in price and accuracy.

Table 19- Common Flow Instruments – Credit Omega [143]

Flow meter	Range	Pressure Loss	Typical Accuracy % of total scale	Require Upstream Length	Cost
Orifice	4:1	Medium	±4	10 – 30	Low
Venturi Tube	4:1	Low	±1	5 – 20	Medium
Variable Area	10:1	Medium	±1 - ±10	0	Low
Ultra Sonic – Transient Time*	20:1	None	±5	5 – 30	High
Mass (Coriolis)	10:1	Low	±0.4	None	High
Turbine	20:1	High	±0.25 (rate)	5 – 10	High
Vortex	10:1	Medium	±1 (rate)	10 – 20	High

*Note: Ultrasonic transient time has some exceptions if this is used when the fluid is a vapour

4.5.5 Plant Wide Considerations

Copper should not be used in any part of the plant components as the nature of the geothermal fluid commonly reacts with copper. **The heat exchangers cannot use cooper.**

The electrical equipment on site will also be most likely made from copper. It is possible to do much of the electrical work with aluminium; however, this requires more material [21]. Alternatively, precautions and barriers can be installed around copper electrical equipment to prevent expose to H_2S, the main cause of copper reaction.

All electrical components should be designed and built to their appropriate standards.

4.5.5.1 Working Fluid Amounts

The amount of working fluid required in an ORC is not published information. However one approach is to fill the high side system with working fluid such that the pre-heaters are completely filled and the evaporator is 70% full. The ORC may require the removal or addition of some working fluid during operating and the following rules of thumb are from the refrigeration industry for charging refrigeration equipment.

High working Fluid – Pressure higher than expected [144], Excessive sub cooling

Low Working Fluid – Excessive superheating[145]

4.6 Detailed Cost Analysis

Once again the handbook of heat exchanger design should be used for detailed cost analysis as it covers each type of heat exchanger explored in the detailed design section. The important aspects of each heat exchanger are highlighted in this section. All costs determined from the heat exchanger handbook should take inflation into account.

4.6.1 Shell and Tube Cost Analysis

4.6.1.1 Size

The shell diameter and overall length of the heat exchanger are the main aspects of the shell and tube heat exchanger costing analysis. The handbook of heat exchanger design has standard figures for determining the cost of a generic shell and tube heat exchanger and then uses a number of factors to approximate the price for a unique heat exchanger.

4.6.1.2 Factors

Pressure

Shell side and tube side pressures impact the cost of the heat exchanger. Shell side pressures are more significant.

Construction

The type of head on the heat exchanger, baffle pitch, and materials call impact the overall cost with certain factors.

4.6.2 Air Cooled Condensers

4.6.2.1 Bare Area

The main component for air cooled heat exchangers is the surface area of bare tubes (excluding the fin area). This depends on the type of tubes used and whether it is welded or seamless.

4.6.2.2 Factors

The factors that impact the cost of the air cooled heat exchangers are listed below

- Tube rows and material
- Tube lengths
- Fin Pitch
- Type of fins
- Cooler type, forced or induced
- Pressure
- Fan type
- Unit size
- Structure finish

4.6.3 Plate Heat Exchangers

The main cost factor for plate heat exchangers is the working pressure. The equation below is the main equation used to cost plate heat exchangers.

$$Cost = F + Nb + yA$$

F basic frame cost \$

N number of liquid connections

b cost of liquid connection

y cost per unit area of heat transfer surface

A heat transfer area

4.7 Detailed Design Documents

The final outcome of the detailed design stage must be the detailed drawings of all the components that have been designed in the previous section. Mainly the heat exchangers and the pipe work. Other components that will be procured from supplies will require adequate specification sheets to approach possible supplies of the equipment.

A plant layout will also be required to start civil works on the plant site and any geothermal piping from the wells. A pipe and instrumentation diagram should also be made to start the process of purchasing all equipment required for the construction of the ORC.

Feed Back and Revision

5 Feedback and Simple Standard

5.1 Industrial Feedback

This standard was reviewed by Kevin Koorey, Kim Hardwood, and Chris Morris. Each of these reviewers looked at the standard from their prospective in a geothermal project. Kevin Koorey is a consultant engineer working at MB Century. Kim Hardwood is an owner's engineer working at Contact Energy and Chris Morris also works at Contact Energy as a technical engineer specialising in rotating equipment.

The overall comments from their review were that the standard was too technical to be used in the traditional sense of a standard. Kim Hardwood, the owner engineer, suggested ‘dumbing down’ the standard to make it more useable to owners that do not have the technical experience or engineering background, such as Maori Iwi or regional councils.

Chris Morris, a technical engineer, agreed with comments made by Kim and that it is too technical for an owner who would typically use an EPC contact. This would make a large portion of the technical information in the standard unnecessary. Their recommendation was to make a simple bare bones version of the standard that would cover the critical information required for owners when going to market for tenders for technology. This recommendation was used to make the owners checklist for a potential EPC contract.

Kevin Koorey, a contractor, who has done ORC component design agreed with the technical information in the standard; however, also agreed it was unlike a standard he had seen before but understood its intended purpose. Comments from Kevin also suggest that an engineer already designing ORCs would not use this standard because they already know their design practices. However, it would be beneficial for an engineer entering the ORC market or helping an engineer with the methods and tools required for ORC design.

The feedback from industry highlighted that the technical depth of standard was too much for members of the industry already working with ORCs. It would be a valuable tool for a new engineer or starter company entering the ORC market. In response to this feedback a simpler ORC standard was made. This document is aimed to assist an owner in a typical geothermal procurement process. The geothermal process typically does not have the owner doing a significant amount of power plant engineering. Owners today prefer to use the Engineering Procurement and Construction contracts (EPC). The EPC contract shifts the design risk of the power plant to a contractor opposed to the owner. The engineering contractor is then responsible for the design and procurement of the ORC system. The owners in New

Zealand use EPC contractors to install their Ormat ORCs. The simple standard supplied in this section highlights critical information that the owner needs to supply to potential vendors for a geothermal ORC. To make it useful for an owner it has been simplified to a checklist.

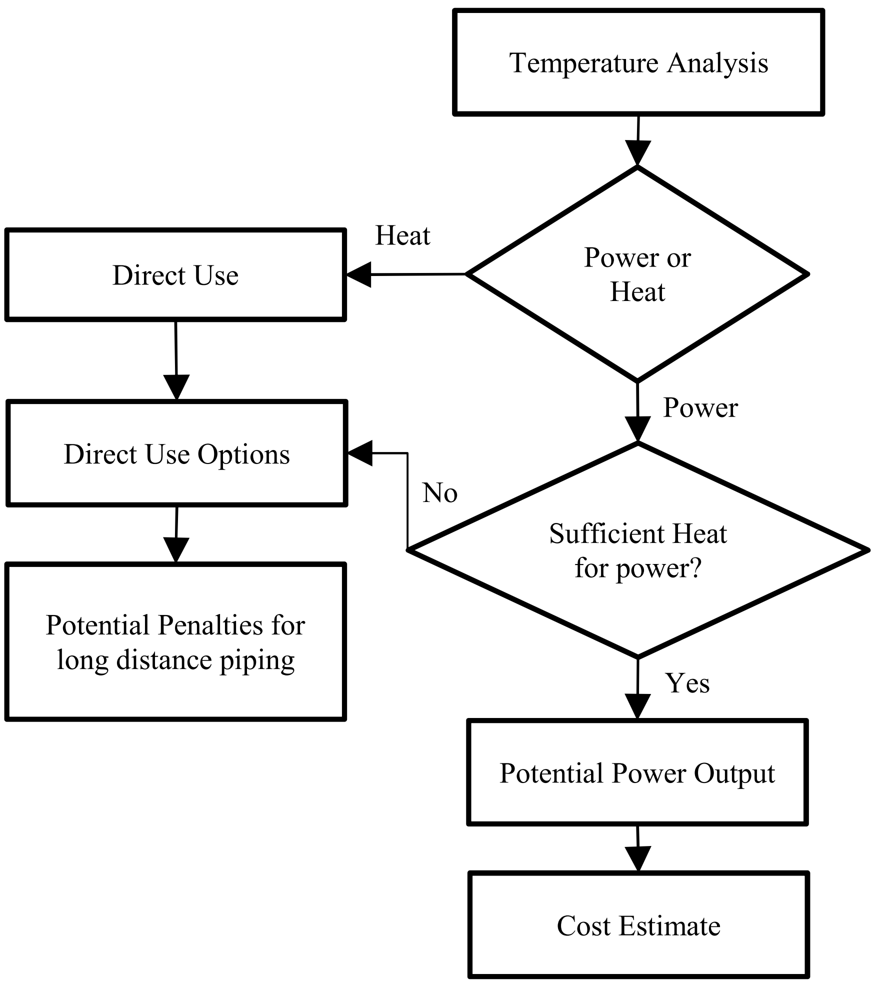
Dr. Keith Alexander at the University of Canterbury has been involved with a number of standards in the past. His opinion was that a new standard needs a document to start from and this can be used by the technical committee to start the process of publishing the standard. The technical level of the standard can be reduced once a technical committee is involved and this current standard can be used as the development tool for the industry standard.

5.2 Simple Standard

This addition is the basic version of the ORC standard highlighting the main aspects of each of the sections for an owner.

5.2.1 Prospecting

The main goal of the prospecting stage is to understand the resource and cover the potential uses.



5.2.1.1 Temperature Analysis

The temperature analysis is the first step to understand the potential uses of the geothermal resource. Figure 66 shows the basic temperature regions and how the geothermal fluid can be used. Direct heat uses are possible at any temperature zone. **An ORC is not possible below 80°C**

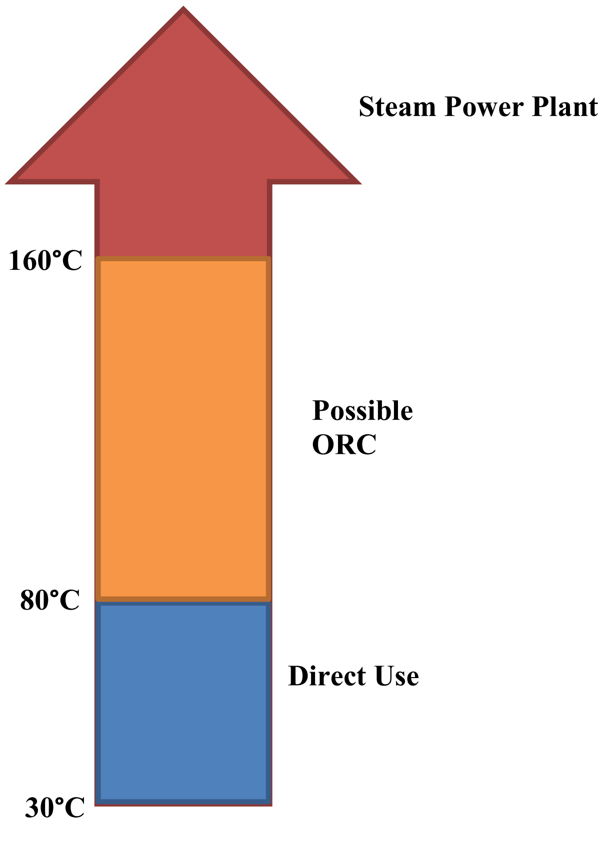


Figure 66 – Potential Temperature Uses

5.2.1.2 Power Potential

Resources above 80°C are capable of producing electricity from the geothermal resource. The next analysis is to determine whether there is substantial heat in the geothermal fluid to feasibility produce electricity. Figure 67 and 68 are used to estimate the amount of heat available in the geothermal fluid relative to climate conditions and then 68 is used to determine whether the amount of heat available in the resource is sufficient to produce power. Figure 68 also highlights the appropriate technology for the resource. If there is insufficient heat and power generation seems unlikely then direct use should be reconsidered.

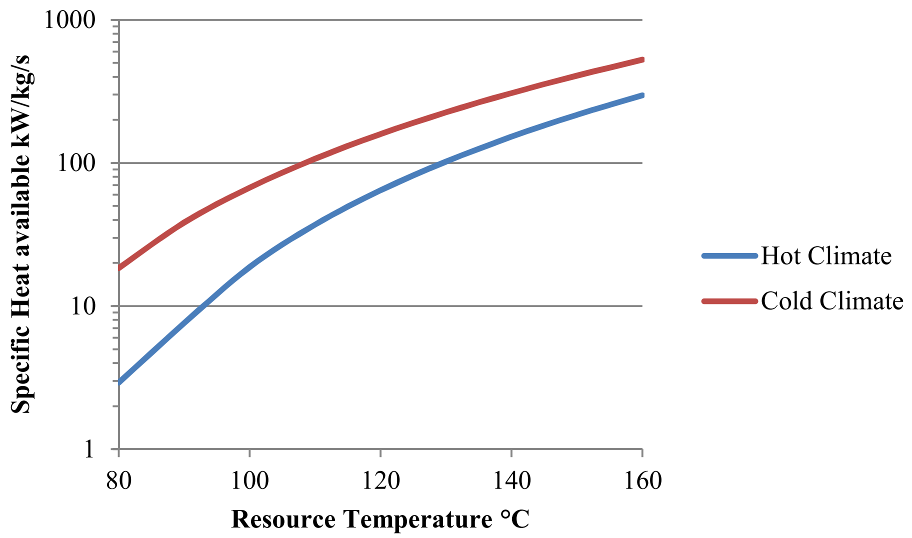


Figure 67 - Specific heat Available

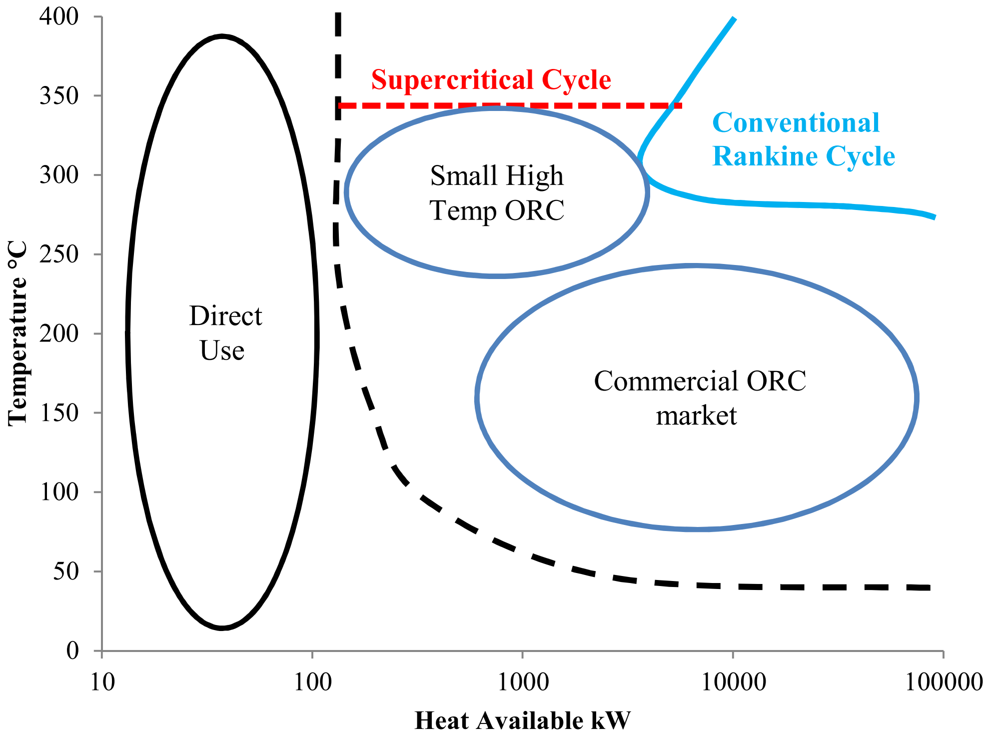


Figure 68 - Power feasibility map – Credit Gaia [1]

5.2.1.3 Potential Power Output

If it is likely that power production is a possible figure 69 is used to estimate the potential output. This power estimate is a rough guess that uses number of assumptions associated with geothermal ORCs and will change with a further investigation.

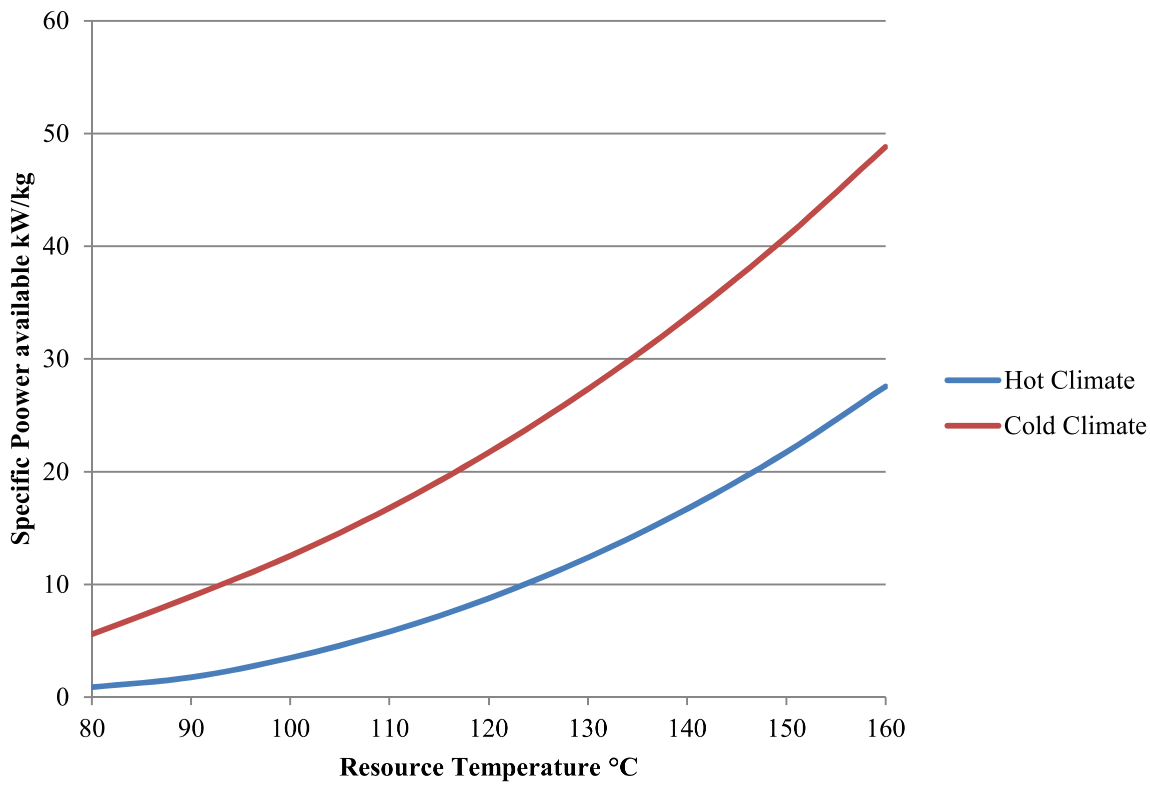


Figure 69 - Specific Power Estimate

5.2.1.4 Rough cost estimate

The total cost of the ORC can be estimated using figure 70. This plot has been developed by IRENA and is an actual plot of plant costs of various geothermal power plants in the world.

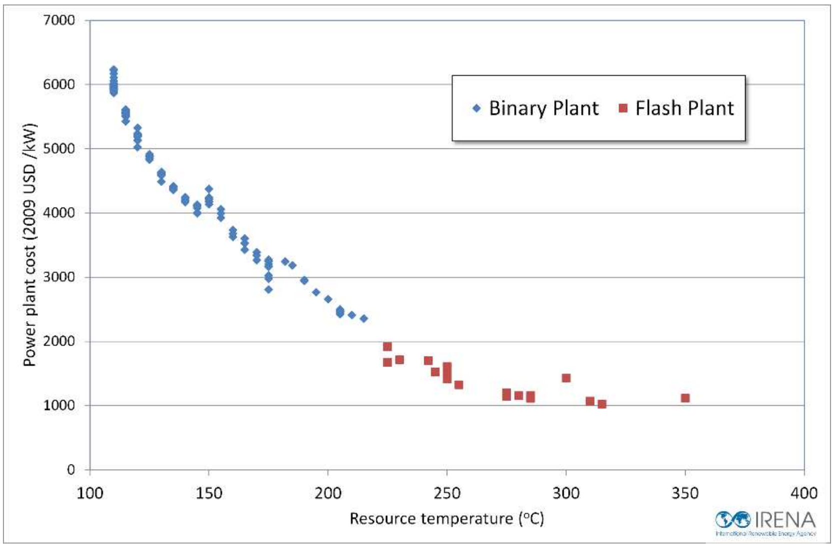


Figure 70 - Specific Costs – Credit IRENA [113]

5.2.1.5 Direct use options

Direct use geothermal heat can be the more favourable option opposed to attempting to develop a small scale ORC. Direct use can be used in a number of ways and figure 71 is a Lindal diagram highlighting some possibilities of direct use.

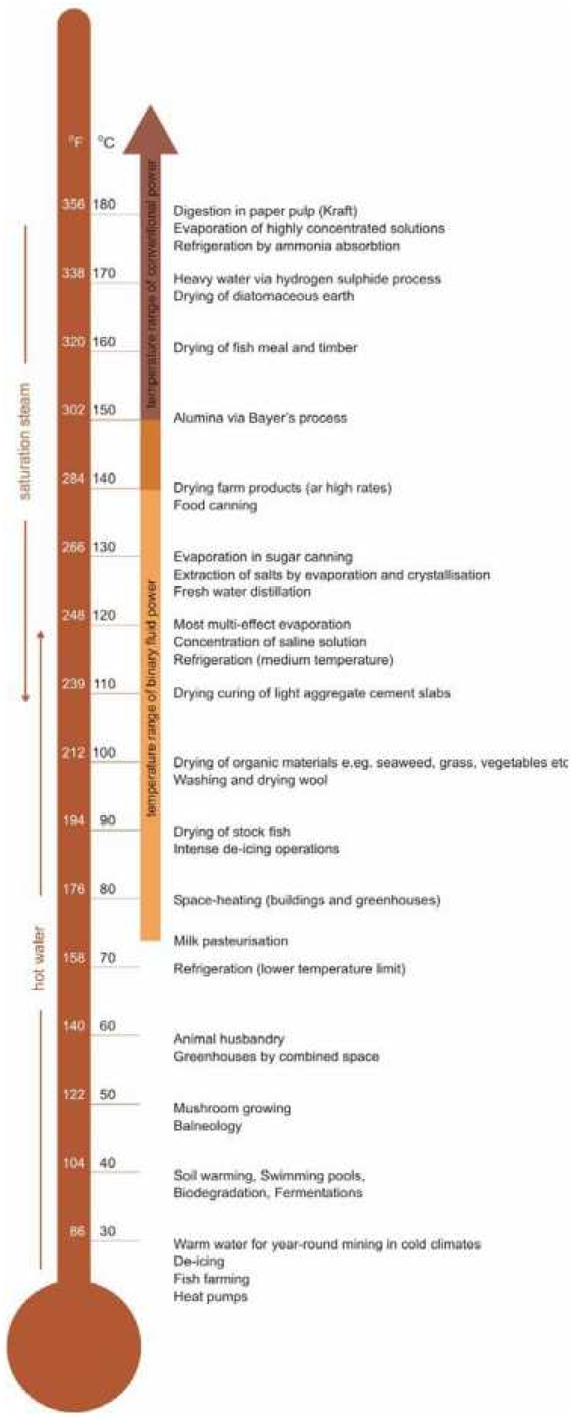


Figure 71 - Lindal Diagram highlighting potential use for geothermal resources – Credit GNS [5]

Pipe Losses

One complication with direct use geothermal systems is the distance from the bore to the location for direct use. The possible losses across the geothermal pipework are shown in figure 72.

The left axis is the potential pressure losses for each meter of pipe required. The right axis is the potential heat loss from insulated pipe for each meter.

Typically geothermal piping is only worthwhile for tens of kilometres. Therefore, the direct use potential will have to relatively close to the geothermal resource. Furthermore, a low temperature resource will most likely need pumping which will have significant parasitic loads.

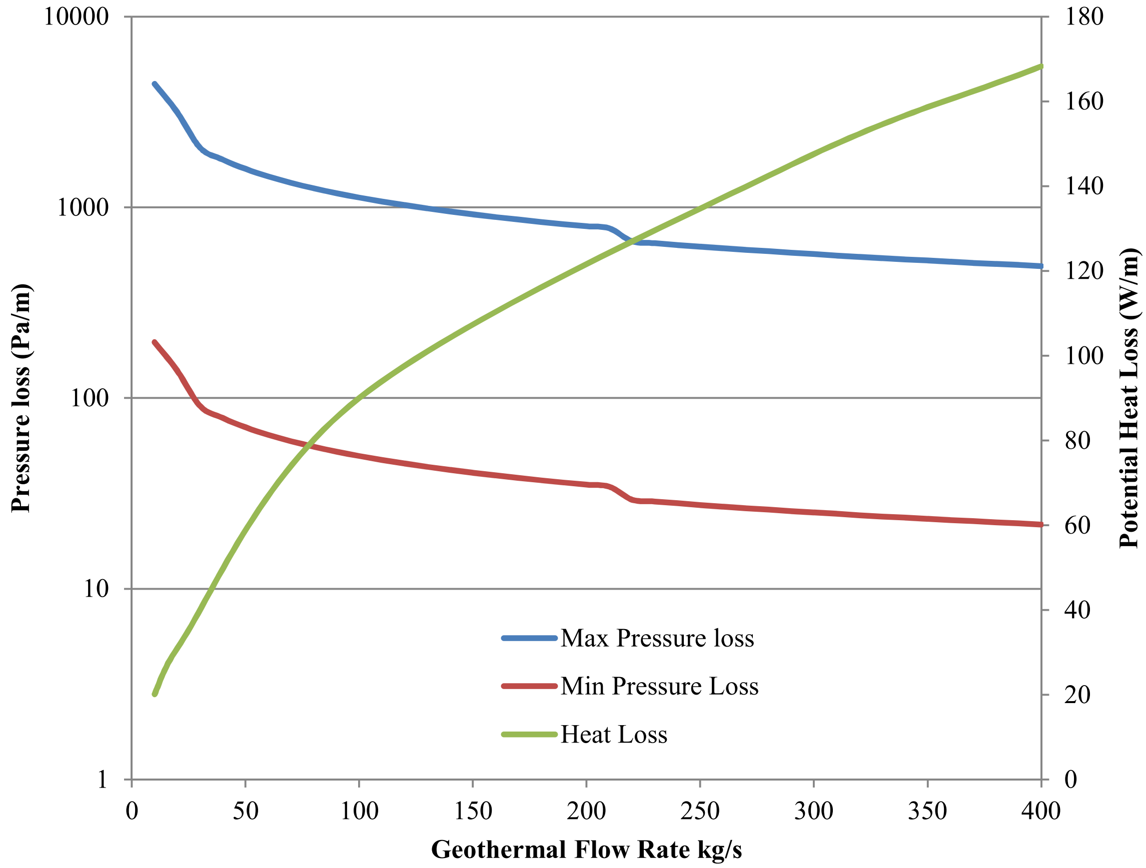


Figure 72 - Geothermal Piping losses

5.2.2 Pre-Feasibility

For a typical geothermal development the owners will have a reservoir team determine the nature of the geothermal fluid before approaching the market for power plant equipment. The following table is a list of critical information the owner must provide when developing an EPC contract for the procurement of power plant equipment. This point onwards the owner will have to compare each technology option provided by potential vendors and determine the best solution for their project.

5.2.2.1 Owners Checklist

Owners Checklist for going to market	
Essential Information to Send to Vendors	
Goal of Development	What is goal of looking at power generation?
Desired Output (kW)	-applicable if the owner has an idea of desired production amount.
Maximum achievable output from resource	- applicable if the owner wants to use all of the available resource
Geothermal Information (Reserve Confirmation)	
Inlet Conditions	The conditions of the fluid supplied to the plant
Temperature (°C)	-Temperature of the fluid
Pressure (kPa)	- Pressure of the fluid
Enthalpy (kJ/kg)	- Enthalpy of the fluid (necessary for a two phase fluid)
Flow Rate (kg/s)	- The flow rate is provided if the owner does not specify the desired power output but instead wants to understand the power available
Fluid Chemistry Comments	- The information regarding the geochemistry should be provided to the vendors so they can make a decision of material requirements and understand outlet conditions.
Outlet Conditions Constraints (if applicable)	- The minimum allowable geothermal conditions
Temperature (°C)	
Location	
Physical Location	- The location of the resource
Description	- important aspects associated with the location (elevation, terrain features, national park concerns)
Accessibility	- Location accessibility
Construction on site or off site	- Does the owner have any issues with construction or will the equipment be prefabricated off site
Water available	- are there any water sources for a cooling water system -Water is also required for utility and firefighting equipment
Meteorological Factors - Relevant factors that impact the design of power plants – mainly ambient temperature	
Mean	- Average conditions at the location of the resource
Wet Bulb Temperature	
Dry Bulb Temperature	
Top 5%	- Highest temperatures reordered at the location of the resource. As this significantly impacts the power production
Wet Bulb Temperature	
Dry Bulb Temperature	
Humidity	

Water Conditions	- Applicable only if there is water available for cooling
Mean Inlet Temperature	
Allowable Outlet Temperature	
Allowable flow rate	
Plant Considerations	
Fully Installed or Equipment Only	- Is the quote for the vendor to do all installation and civil works or just for the equipment
Capacity Factor	- what reliability factor are you considering
Standard or Bespoke Product	- Will the device be similar to vendors previous builds or does the owner want a unique system
Plant Factors	- Any specific factors the owner wants for the plant i.e. non-flammable working fluid, size issues.
Environmental Requirements	Noise, Odour, Emissions, Discharge, Visual impact,
Operations	
Remote or Full time staff	This will depend on the size of the plant and needs of the owner, (Below 50MW unlike to have full time staff)
Control System	Remote monitoring, Alarms, Safe Shutdown
Start-up	Operator Requirements (minimum staff recommended)
Safety	
Standard	Relevant standards for Pressure vessel and piping, construction – Relevant to the country
Fire Safety	All fire safety standards are required.
How System will be evaluated (Select One)	
Maximum Efficiency	- Will potentially use less resource but will likely require more equipment
Minimum Cost	- lowest cost solution but possibly not the most efficient system for the resource
Potential Incentives	- Are there benefits for potential vendors exploring more efficient cycles even it does increase the cost
Desired Feedback from Vendor	
Budget Prices from vendors	- Price estimate from vendors for equipment for resource
Technology used	- what type of technology is used (ORC configuration)
Amount of Fluid Used	- If no fluid amount is given what is the required amount

6 Conclusions

This standard was made with the best knowledge and expert advice available. There are opportunities to improve parts of the standard to make it more accessible. These improvements will only happen once an expert technical committee is established by the ISO to develop the standard. It is presented in a way to make it accessible to a wide audience so all parties in a geothermal project will have a section relevant to them.

The prospecting stage is at a good introductory level for owners with a limited background in the geothermal industry. The next sections increase the technical detail with more in-depth thermodynamic modelling and component design. As the technical detail increases the owner will struggle to follow the standard but engineers starting in the ORC market can use it to improve their ORC design skills.

The sections in this standard have been widely accepted by geothermal industry members and they all have agreed that these are the steps needed in a successful geothermal ORC. The initial feedback from the first industry review was generally positive with the main criticism being the standard is too technical.

The reviewers recommended making a simpler standard which was included in section 5.2. The recommendation was to provide the owners with more information to help them through the initial stages of a geothermal project. The simple standard provides more information understand the potential direct uses of the fluid and the amount of heat required for an ORC. The second recommendation was a comprehensive checklist of what an owner must provide when negotiating an EPC contract. The owners can then use sections of the standard to understand the proposed ORC from the vendors.

The standard will become common use in the industry once there are more developers building and maintaining ORCs. Currently the ORC market only has a few big developers that have cornered the market and are cautious about exposing their ORC design approach. In order to make ORC technology widely available more small ORCs must be built to understand key aspects of the design process. The engineers involved with these small ORCs must follow the standard here and with more project experience can provide the required changes to make the standard more accepted.

The current standard will provide small companies with the ability to address their geothermal resource potential without hiring a consultant. The level of detailed provided also allows them to do the majority of the design and selection themselves. Their experiences will improve the standard and highlight the required steps to design and build an ORC leading to an international accepted standard.

.

References

1. Gaia, M. *30 Years of Organic Rankine Cycle Development* in *First International Seminar on ORC Power Systems* 2011. Delft.
2. REN, *Renewables 2014 Global Status Report*, 2014, REN21.
3. Gehringer, M. and V. Loksha, *Geothermal handbook: Planning and Financing Power Generation Generation*. 2012, Washington DC, USA: THE WORLD BANK GROUP.
4. Thain, I.A., et al., *A practical guide to exploiting low temperature geothermal resources*. Vol. 2006. 2006: GNS Science.
5. Gazo, F. and L. Lind, *Low Enthalpy Geothermal Energy - Technology Review*, 2010.
6. Carey, B., *Low Enthalpy Geothermal Research–New Zealand.*
7. Stewart, C. *Story: Geothermal energy*. 2012 9-11-12 [cited 2015 16-02]; Available from: http://www.teara.govt.nz/en/geothermal-energy.
8. Lund, J.W., *Geothermal direct-heat utilization.* Technika Poszukiwań Geologicznych, 2004. **43**: p. 19-33.
9. EECA, *Annual Report*, 2008: New Zealand.
10. Thain, I.A., *A brief history of the Wairakei geothermal power project.* Geo-Heat Centre Quarterly Bulletin, 1998: p. 1-4.
11. Kelly, G., *History and potential of renewable energy development in New Zealand.* Renewable and Sustainable Energy Reviews, 2011. **15**(5): p. 2501-2509.
12. DiPippo, R., *Geothermal Power Plants : Principles, Applications, Case Studies and Environmental Impact*, 2011, Elsevier Science: Burlington.
13. Valdimarsson, P., *Geothermal power plant cycles and main components.* Short Course on Geothermal Drilling, Resource Development and Power Plants, 2011: p. 16-22.
14. Muller, L., et al., *Stibnite Control in ORMAT Binary Units. Prevention and online cleaning.*, in *New Zealand Geothermal Workshop*2014: Auckland.
15. Quoilin, S. and V. Lemort, *Technological and economical survey of Organic Rankine Cycle systems.* 2009.
16. Jalilinasrabady, S. and R. Itoi. *Flash Cycle and Binary Geothermal Power Plant Optimization*. in *Geothermal Resources Council 2012 Annual Meeting, September*. 2012.
17. DiPippo, R., *Small geothermal power plants: design, performance and economics.* GHC Bulletin, June, 1999. **1**.
18. DiPippo, R. and P. Moya, *Las Pailas geothermal binary power plant, Rincón de la Vieja, Costa Rica: Performance assessment of plant and alternatives.* Geothermics, 2013. **48**(0): p. 1-15.
19. DiPippo, R., *Second Law assessment of binary plants generating power from low-temperature geothermal fluids.* Geothermics, 2004. **33**(5): p. 565-586.
20. DiPippo, R., *Ideal thermal efficiency for geothermal binary plants.* Geothermics, 2007. **36**(3): p. 276-285.
21. Kestin, J., et al., *Sourcebook on the production of electricity from geothermal energy*, 1980, Brown Univ., Providence, RI (USA).
22. Agahi, R., *Pamukuren Geothermal Organic Rankine Cycle Power Plant in Turkey.* Geothermal Resources Council Transactions, 2012. **36**: p. 1057-1060.
23. IRENA, *Renewable Power Generation Costs in 2014*, 2014, IRENA.
24. Nazif, H., *Feasibility of developing binary power plants in the existing geothermal production areas in Indonesia.* United Nations University Geothermal Program, 2011. **Report 29**: p. 709-735.
25. Dipippo, R., *Personal Communication - Binary Plant Design*, 2013.
26. Quoilin, S., et al. *Working fluid selection and operating maps for Organic Rankine Cycle expansion machines*. in *Proceedings of the 21st International Compressor Conference at Purdue*. 2012.
27. Chen, H., D.Y. Goswami, and E.K. Stefanakos, *A review of thermodynamic cycles and working fluids for the conversion of low-grade heat.* Renewable and Sustainable Energy Reviews, 2010. **14**(9): p. 3059-3067.
28. Maizza, V. and A. Maizza, *Working fluids in non-steady flows for waste energy recovery systems.* Applied Thermal Engineering, 1996. **16**(7): p. 579-590.

29. Zhai, H., L. Shi, and Q. An, *Influence of working fluid properties on system performance and screen evaluation indicators for geothermal ORC (organic Rankine cycle) system.* Energy, 2014.
30. Papadopoulos, A.I., M. Stijepovic, and P. Linke, *On the systematic design and selection of optimal working fluids for Organic Rankine Cycles.* Applied Thermal Engineering, 2010. **30**(6): p. 760-769.
31. Astolfi, M., et al., *Binary ORC (Organic Rankine Cycles) power plants for the exploitation of medium–low temperature geothermal sources–Part B: Techno-economic optimization.* Energy, 2014. **66**: p. 435-446.
32. Saleh, B., et al., *Working fluids for low-temperature organic Rankine cycles.* Energy, 2007. **32**(7): p. 1210-1221.
33. Wang, E.H., et al., *Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery.* Energy, 2011. **36**(5): p. 3406-3418.
34. Shengjun, Z., W. Huaixin, and G. Tao, *Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation.* Applied Energy, 2011. **88**(8): p. 2740-2754.
35. Roy, J.P., M.K. Mishra, and A. Misra, *Parametric optimization and performance analysis of a waste heat recovery system using Organic Rankine Cycle.* Energy, 2010. **35**(12): p. 5049-5062.
36. Liu, B.-T., K.-H. Chien, and C.-C. Wang, *Effect of working fluids on organic Rankine cycle for waste heat recovery.* Energy, 2004. **29**(8): p. 1207-1217.
37. Stijepovic, M.Z., et al., *On the role of working fluid properties in Organic Rankine Cycle performance.* Applied Thermal Engineering, 2012. **36**(0): p. 406-413.
38. Angelino, G. and P. Colonna di Paliano, *Multicomponent working fluids for organic Rankine cycles (ORCs).* Energy, 1998. **23**(6): p. 449-463.
39. Wang, X. and L. Zhao, *Analysis of zeotropic mixtures used in low-temperature solar Rankine cycles for power generation.* Solar Energy, 2009. **83**(5): p. 605-613.
40. Wang, J., L. Zhao, and X. Wang, *A comparative study of pure and zeotropic mixtures in low-temperature solar Rankine cycle.* Applied Energy, 2010. **87**(11): p. 3366-3373.
41. Chys, M., et al., *Potential of zeotropic mixtures as working fluids in organic Rankine cycles.* Energy, 2012. **44**(1): p. 623-632.
42. Heberle, F., M. Preißinger, and D. Brüggemann, *Zeotropic mixtures as working fluids in Organic Rankine Cycles for low-enthalpy geothermal resources.* Renewable Energy, 2012. **37**(1): p. 364-370.
43. Frick, S., S. Kranz, and A. Saadat. *Holistic design approach for geothermal binary power plants with optimized net electricity provision*. in *Proceedings World Geothermal Congress*. 2010.
44. Quoilin, S., et al., *Thermo-economic optimization of waste heat recovery Organic Rankine Cycles.* Applied Thermal Engineering, 2011. **31**(14–15): p. 2885-2893.
45. Madhawa Hettiarachchi, H.D., et al., *Optimum design criteria for an Organic Rankine cycle using low-temperature geothermal heat sources.* Energy, 2007. **32**(9): p. 1698-1706.
46. Astolfi, M., et al., *Binary ORC (organic Rankine cycles) power plants for the exploitation of medium–low temperature geothermal sources–Part A: Thermodynamic optimization.* Energy, 2014. **66**: p. 423-434.
47. Schuster, A., S. Karellas, and R. Aumann, *Efficiency optimization potential in supercritical Organic Rankine Cycles.* Energy, 2010. **35**(2): p. 1033-1039.
48. Gabbrielli, R., *A novel design approach for small scale low enthalpy binary geothermal power plants.* Energy Conversion and Management, 2012. **64**(0): p. 263-272.
49. Dai, Y., J. Wang, and L. Gao, *Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery.* Energy Conversion and Management, 2009. **50**(3): p. 576-582.
50. Wei, D., et al., *Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery.* Energy conversion and Management, 2007. **48**(4): p. 1113-1119.

51. Franco, A. and M. Villani, *Optimal design of binary cycle power plants for water-dominated, medium-temperature geothermal fields.* Geothermics, 2009. **38**(4): p. 379-391.
52. Taylor, C. *Selection of Technology Options for Binary Cycle Power Plant*. in *NZGW*. 2013. Rotorua.
53. Quoilin, S., V. Lemort, and J. Lebrun, *Experimental study and modeling of an Organic Rankine Cycle using scroll expander.* Applied Energy, 2010. **87**(4): p. 1260-1268.
54. ElectraTherm. *ElectraTherm's Waste Heat to Power Generation System*. 2012.
55. Zhou, N., et al., *Experimental study on Organic Rankine Cycle for waste heat recovery from low-temperature flue gas.* Energy, 2013. **55**: p. 216-225.
56. Macchi, E. and A. Perdichizzi, *Efficiency prediction for axial-flow turbines operating with nonconventional fluids.* Journal for Engineering for Power, 1981. **103**(4): p. 718-724.
57. Yamamoto, T., et al., *Design and testing of the Organic Rankine Cycle.* Energy, 2001. **26**(3): p. 239-251.
58. Pei, G., et al., *Construction and dynamic test of a small-scale organic rankine cycle.* Energy, 2011. **36**(5): p. 3215-3223.
59. Kang, S.H., *Design and experimental study of ORC (organic Rankine cycle) and radial turbine using R245fa working fluid.* Energy, 2012. **41**(1): p. 514-524.
60. Saemundsdottir, T. and L. Tryggvadottir, *How a Potential Geothermal Project Becomes Reality: The Enex Way.* Geothermal Resources Council Transactions, 2008. **32**.
61. Holdmann, G. and K. List, *The Chena Hot Springs 400kW geothermal power plant: experience gained during the first year of operation.* Chena Geothermal Power Plant Report, Chena Power Plant, Alaska, 2007: p. 1-9.
62. Chena Power, L., *400kW Geothermal Power Plant at Chena Hot Springs, Alaska.* Final Report prepared for Alaska Energy Authority, 2007.
63. Aneke, M., B. Agnew, and C. Underwood, *Performance analysis of the Chena binary geothermal power plant.* Applied Thermal Engineering, 2011. **31**(10): p. 1825-1832.
64. Holt, B. and E.L. Ghormley, *Energy Conversion and Economics for Geothermal Power Generation at Heber, California; Valles Caldera, New Mexico, and Raft River, Idaho - Case Studies*, 1976.
65. Holt, B. and E.L. Ghormley, *Preliminary design manual for a Geothermal Demonstration Plant at Heber, California*, 1978. p. Medium: P; Size: Pages: 149.
66. Tchanche, B.F., et al., *Low-grade heat conversion into power using organic Rankine cycles – A review of various applications.* Renewable and Sustainable Energy Reviews, 2011. **15**(8): p. 3963-3979.
67. Dipippo, R., *A Simplified Method for Estimating the Silica Scaling Potential in Geothermal Power Plants.* Geothermal Resources Council Transactions, 1985. **14**: p. 3-9.
68. Elliott, T.C., *Standard handbook of powerplant engineering.* 1989.
69. King, O.R., *Personal Communication*, 2014, Turboexpanders.
70. Aartun, I., *Mollier Diagram*, 2001, University of Idaho.
71. Perry, R.H., D.W. Green, and J.O. Maloney, *Perry's chemical engineer's handbook*, in *Perry's chemical engineer's handbook*. 1984, McGraw-Hill Book.
72. ISO, *ISO 14661:2000 Thermal Turbines for Industrial applications (Steam turbines, Gas expansion turbines) -- General Requirements*, 2000, ISO: Switzerland.
73. Balje, O., *Turbomachines*. 1981: John Wiley.
74. Taylor, P., *Personal Communication - Geothermal Overview*, L. Taylor, Editor 2015.
75. Lukawski, M., *Design and optimization of standardized organic Rankine cycle power plant for European conditions.* Master's thesis, the School for Renewable Energy Science in affiliation with University of Iceland and the University of Akureyri, 2009.
76. Hewitt, G.F., *Hemisphere handbook of heat exchanger design*. 1990: Hemisphere Publishing Corporation New York.
77. Levy, C.E., *Lessons Learned From Raser Technologies 'Revolutionary' Project*, in *Technology*2011, Breaking Energy
78. Sones, R. and Z. Krieger. *Case history of the binary power plant development at the Heber, California geothermal resource*. in *Proceedings of the World Geothermal Congress 2000, Kyushu-Tohoku, Japan*. 2000.

79. Krupp, R.E., *Solubility of stibnite in hydrogen sulfide solutions, speciation, and equilibrium constants, from 25 to 350 C.* Geochimica et Cosmochimica Acta, 1988. **52**(12): p. 3005-3015.
80. Maizza, V. and A. Maizza, *Unconventional working fluids in organic Rankine-cycles for waste energy recovery systems.* Applied Thermal Engineering, 2001. **21**(3): p. 381-390.
81. Shaffer, C., *Floating power optimization studies for the cooling system of a geothermal power plant.* NASA STI/Recon Technical Report N, 1977. **78**: p. 20646.
82. Delgado-Torres, A.M. and L. García-Rodríguez, *Preliminary assessment of solar organic Rankine cycles for driving a desalination system.* Desalination, 2007. **216**(1–3): p. 252-275.
83. Japikse, D. and N. Baines, *Introduction to turbomachinery*. 1994.
84. ChemKB. *Pros and Cons of Plate Frame Heat Exchangers*. 2013 24/August/2013 [cited 2014 10/10 /14]; Available from: http://www.chemkb.com/equipments/heat-exchangers/plate-frame-heat-exchangers/pros-and-cons-of-plate-frame-heat-exchangers/.
85. Firoz. *Advantages and Disadvantages of Shell and Tube & Plate type Heat Exchangers*. 2013 19/June/2013 [cited 2014 10/10/14]; Available from: http://jabelufiroz.hubpages.com/hub/Advantages-and-Disadvantages-of-Shell-and-Tube-Plate-type-Heat-Exchangers.
86. Hewitt, G.F. and J. Barbosa, *Heat exchanger design handbook*. Vol. 98. 2008: Begell House.
87. Solutions, H.C.H.T. *Heat Exchanger Types and Selection* 2014 [cited 2014 10/10/14]; Available from: http://www.hcheattransfer.com/selection.html.
88. Mukherjee, R., *Effectively Design Shell-and-Tube Heat Exchangers*, 1998: CHEMICAL ENGINEERING PROGRESS.
89. Laval, A., *The theory behind heat transfer - Plate heat exchangers*, A. Laval, Editor 2004, Alfa Laval.
90. Sharma, R.P. *Selection of Pumps for Process Industries*. 2012 [cited 2014 17 Septermber]; Available from: http://chemicaljobs.files.wordpress.com/2012/09/selectionofpumpsforprocessindustries.pdf.
91. Gerrard, A., *Guide to capital cost estimating*. 2000: IChemE.
92. Couper, J.R., W.R. Penney, and J.R. Fair, *Chemical Process Equipment revised 2E: Selection and Design*. 2009: Gulf Professional Publishing.
93. Kharagpur. *Condensers & Evaporators* [cited 2014 15/10/14]; Available from: http://nptel.ac.in/courses/112105129/pdf/R&AC%20Lecture%2022.pdf.
94. EngineeringPage. *Typical Overall Heat Transfer Coefficients (u-Values)*. [cited 2014 16/10/14]; Available from: http://www.engineeringpage.com/technology/thermal/transfer.html.
95. ChemResources. *U in Heat Exchangers*. 2012 [cited 2014 15/10/14]; Available from: http://www.cheresources.com/content/articles/heat-transfer/u-in-heat-exchangers.
96. Roetzel, W. and B. Spang. *Typical Values of Overall Heat Transfer Coefficients*. [cited 2014 15/10/14]; Available from: http://download.springer.com/static/pdf/827/chp%253A10.1007%252F978-3-540-77877-6_6.pdf?auth66=1424747904_beae7851a1cc33b459ea3bd63e647165&ext=.pdf.
97. Coulson, J.M. and R.K. Sinnott, *Chemical Engineering, vol. 6: An Introduction to Chemical Engineering Design*, 1983, Pergamon.
98. Kaplan, U., *Organic Rankine Cycle Configurations.* Proceedings European Geothermal, 2007.
99. Lai, N.A., M. Wendland, and J. Fischer, *Working fluids for high-temperature organic Rankine cycles.* Energy, 2011. **36**(1): p. 199-211.
100. Eckert, K. *Siemens offers new solution for utilization of waste heat using the Organic Rankine Cycle*. Power Generation 2014 [cited 2015 26/01/2015].
101. Meyer, D., et al., *DESIGN AND BUILD OF A 1 KILOWATT ORGANIC RANKINE CYCLE POWER GENERATOR.*
102. ANEST-IWATA. *Compact binary power generator*. 2015 [cited 2015 25/02/15]; Available from: http://www.anest-iwata.co.jp/english/compressor/binary/binary.html.

103. Langson, R., *One and two-stage direct gas and steam screw expander generator system (dsg)*, 2011, Google Patents.
104. Perdichizzi, A. and G. Lozza, *Design Criteria and Efficiency for Radial Inflow Turbines*, in *Gas Turbine Conference and Exhibition*1987: Anaheim CA, USA,.
105. Paltrinieri, A., *A mean-line model to predict the design performance of radial inflow turbines in organic rankine cycles.* 2014.
106. Matthew Proctor, et al., *Optimisation Of A Fin Fan Condenser In A Geothermal ORC Plant*, in *New Zealand Geothermal Workshop*2014: Auckland.
107. Scientists, U.o.C. *Environmental Impacts of Geothermal Energy*. Renewable Energy 2012 [cited 2015 27/01/2015]; Available from: http://www.ucsusa.org/clean_energy/our-energy-choices/renewable-energy/environmental-impacts-geothermal-energy.html#bf-toc-1.
108. White, B., *Waikite Small Geothermal Power Plant Feasibility Study*, 2009, East Harbour Energy. p. 20.
109. Holdmann, G., *400kW Geothermal Power Plant at Chena Hot Springs, Alaska*, 2007, Alaska Energy Authority: Chena Power Company.
110. Reisenhofer, I.O.P.T.E., *Description and Evaluation of the new 1,000 kWel Organic Rankine Cycle process integrated in the biomass CHP plant in Lienz, Austria.* Euroheat & Power, 2002. **10/2002**: p. 17.
111. Nazif, H., *Feasibility of developing binary power plants in the existing geothermal production areas in Indonesia* in *Geothermal training programme*2011, United Nations University: Reykjavik, Iceland.
112. SKM, N., *Asssessment of Current Costs of Geothermal Power Generation in New Zealand (2007 Basis)*, 2009, New Zealand Geothermal Association and SKM. p. 78.
113. IRENA. *Geothermal Cost Charts*. 2014 14/10/2014]; Available from: http://costing.irena.org/charts/geothermal.aspx.
114. GRAHAM. *Gasketed/Brazed Plate Heat Exchangers Data Sheets*. [cited 2014 20/10/14]; Available from: http://www.graham-mfg.com/plate-heat-exchanger-data-sheets.
115. myChemE. *Recommended Pipeline Velocities*. 2013 [cited 2015 29/01]; Available from: http://www.mycheme.com/recommended-pipeline-velocities/.
116. Box, E.T. *Boiling Liquids - Pumping Flow Velocities*. [cited 2015 28/01]; Available from: http://www.engineeringtoolbox.com/pump-delivery-flow-velocity-boiling-fluid-d_234.html.
117. Streeter, V.L., E.B. Wylie, and K.W. Bedford, *Fluid mechanics, WCB*, 1998, McGraw-Hill.
118. ASME, *B31.1 Power Piping*, 2012, ASME.
119. ASME, *Process Piping*, 2012, ASME. p. 540.
120. ASME, *Pipeline Transportation Systems for Liquids and Slurries*, 2012, ASME. p. 140.
121. ASME, *Gas Transmission & Distribution Piping Systems*, 2014, ASME. p. 220.
122. Hewitt, G.F., G.L. Shires, and T.R. Bott, *Process heat transfer.* 1994.
123. Puwar, V. *DESIGN OF SHELL & TUBE HEAT EXCHANGER USING SPERADSHEET*. 2012 [cited 2015 30/01]; Available from: http://www.scribd.com/doc/93755266/DESIGN-OF-SHELL-TUBE-HEAT-EXCHANGER-USING-SPERADSHEET#scribd.
124. TEMA, *Standards of the Tubular Exchanger Manufacturers Association*, 1988, TEMA Inc: New York.
125. Boiler, A. and P.V. Code, *American Society of Mechanical Engineers.* New York, 1998. **10017**.
126. BSI, *Specification for unfired fusion welded pressure vessels*, 2015, BSI.
127. ISO, *Steel tubes for boilers, superheaters and heat exchangers -- Dimensions, tolerances and conventional masses per unit length*, 1980, ISO.
128. ISO, *Welded steel tubes for heat exchangers*, 1980, ISO.
129. ISO, *Boilers and pressure vessels -- Part 1: Performance requirements*, 2007, ISO.
130. ISO, *Boilers and pressure vessels -- Part 2: Procedures for fulfilling the requirements of ISO 16528-1*, 2007, ISO.
131. ISO, *Petroleum, petrochemical and natural gas industries -- Air-cooled heat exchangers*, 2011, ISO.
132. ASME, *Air Cooled Heat Exchangers (ACHE)*, 1991, ASME.

133. ISO, *Petroleum, petrochemical and natural gas industries -- Plate-type heat exchangers -- Part 1: Plate-and-frame heat exchangers*, 2005, ISO.
134. Karassik, I.J., et al., *Pump handbook*. Vol. 3. 2001: McGraw-Hill New York.
135. ANSI/API, *Centrifugal pumps for general refinery service* 1995, ANSI/API.
136. ISO, *Technical Specifications for centrifugal pumps*, 2002, ISO.
137. ASME, *Specification of Horizontal end suction centrifugal pumps for chemical process*, 2001, ASME.
138. ASME, *Specifications for vertical in-line centrifugal pumps for chemical process*, 2003, ASME.
139. BSI, *Specifications for horizontal end suction centrifugal pumps (16 Bar)*, 1975, BSI.
140. ISO, *Data sheets for thermal turbines for industrial applications*, 2002, ISO.
141. API, *Axial and Centrifugal Compressors and Expander-compressors*, 2014, API.
142. API, *Lubrication Shaft-Sealing and Control-Oil Systems for Special-Purpose Applications*, 1999, API.
143. Omega. *Flow Meter Selection Guide*. [cited 2015 30/01]; Available from: http://www.omega.com/techref/table1.html.
144. Inspectapedia. *Air Conditioner / Heat Pump Refrigerant Pressure Too High - Overcharged*. 2014 [cited 2015 30/01]; Available from: http://inspectapedia.com/aircond/Refrigerant_Overcharge.htm.
145. Wulfinghoff, D., *Energy efficiency manual*. Vol. 3936. 1999: Energy Institute Press Maryland.