Pedagogical development and technical research in the area of geothermal power production

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ABSTRACT

This work describes the types of power plants used for geothermal power generation in the world; the dry steam power plant, the flash steam power plant and the binary cycle power plant. The objective of the MSc work was the development of learning content on the subject of geothermal power generation for the CompEdu platform in the energy department at KTH. The power plants are described from a system perspective followed by an explanation of the operation of major components. Examples and calculations are given with the aim of illustrating which parameters are most important to the operation of each plant from a performance perspective. An important point is that the report does not go into detail for auxiliary components such as piping and valves. These components are essential from the point of view of fluid handling, however are less important in terms of understanding the mode of operation of the power plant. The power plants must consider the fact that geothermal fluid is corrosive and contains non-condensable gases. The choice of the type of geothermal power plant depends on the temperature and state of the geothermal fluid being utilised (liquid or vapour dominated). The research shows that each power plant has its own significant optimisation criteria, to summarise these: the dry steam power plant uses the selected wellhead pressure for extraction of geothermal fluid to optimise power output, the flash steam power plant uses the operating conditions in the steam separator to optimise power output and the binary cycle uses the required heat exchanger area per unit of power generated to select the optimal working fluid for power generation. Finally, innovative alternatives for power generation from geothermal resources that are on the horizon are introduced.
PREFACE
ACKNOWLEDGEMENT
I would like to thank my supervisor, Anneli Carlqvist, who has guided me patiently and given me honest feedback about my work. I would also like to thank my family and close friends who have supported me throughout.
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<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>Bar, a</td>
<td>Absolute bars of pressure</td>
</tr>
<tr>
<td>CC</td>
<td>Combined Cycle</td>
</tr>
<tr>
<td>CCS</td>
<td>Carbon Capture and Sequestration</td>
</tr>
<tr>
<td>GW_{e}</td>
<td>Gigawatts of electricity</td>
</tr>
<tr>
<td>KTH</td>
<td>Kungliga Tekniska Högskolan</td>
</tr>
<tr>
<td>MW_{e}</td>
<td>Megawatts of electricity</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine Cycle</td>
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1 BACKGROUND / INTRODUCTION

The Energy Department of KTH has since the late -90’s continuously been in the process of developing its online learning resource called CompEdu and would like to increase the amount of information in the area of geothermal power production. The description of CompEdu from the website, (CompEdu, 2011) quote: “The CompEdu platform is an inexpensive, low cost gateway to self-study on-line and/or on-campus learning, education and training in heat and power technology, including performance, gas turbines, steam turbines, district heating, cogeneration, combined cycles, renewable energy systems, aero- and thermodynamics of turbomachinery, both from a technical and economical side.”. Furthermore the website also outlines the aim and structure of the different chapters “The different chapters in the platform start on a basic, non-engineering level explaining the fundamental nature of the processes or components in the system, and end in some learning objects at a Master or PhD level.”

Currently, an objective of the Energy Department at KTH is to complete a book on the subject of geothermal energy, which is a renewable and sustainable energy source from which (if utilised correctly) heating, cooling and base-load electricity can be generated. In addition to this it should be mentioned that geothermal power generation has a high level of unrealised potential in the world today, especially with the advent of binary geothermal power plant technology which can generate electricity from temperatures as low as 57°C (Erkan et al., 2008). In recent times energy and the climate have been in the spotlight, considering the geopolitical issues, lack of energy security and climate change associated with conventional fossil fuels. For this reason, research, development and knowledge of renewable, sustainable energy sources, especially sources with a high level of unrealised potential, is in great demand. The energy department at KTH has already begun writing the chapters introducing geothermal energy, such as geothermal energy resources, exploration, drilling and extraction. Therefore there is a necessity for a chapter about geothermal power generation as it is one major way in which the geothermal energy underground can be converted into useful energy, i.e. electricity. This thesis work seeks to develop the appropriate chapters with respect to geothermal power production in a pedagogical fashion.
1.1 Literature Resources

There are many resources that explain how geothermal power plants operate, however most of these resources describe their operation in a very general way. There may be several reasons for this, one being that geothermal power production is at a small scale, especially when compared to installed capacities of wind, hydropower and nuclear power plants. Another reason is the fact that geothermal power generation uses most of the equipment used in and has similar processes as the conventional, fossil-fuel fired power plants (for example the Rankine cycle, a hydrothermal working fluid, turbines and heat exchangers) and because of this, information that is specific to geothermal power generation is often overlooked.
2 SCOPE

The aim of this thesis is to facilitate the thermodynamic and technical understanding of the major types of power plants involved in geothermal power generation. Also it seeks to give an appreciation for the considerations specific to the operation of geothermal power plants, especially when compared to fossil-fuel power generation. In order to accurately illustrate possible geothermal power plant considerations, (such as those concerned with optimisation and showing the necessity of special components and processes) calculations and examples are given. The scope for implementing future, more advanced technology in the field of geothermal power generation is further discussed. It should be noted that the level of detail present in the description of the systems is more focused on the primary areas/components, for example turbines and heat exchangers, as opposed to auxiliary components such as piping and valves.
3 OBJECTIVES AND GOALS

The general objectives of the MSc thesis project are as follows:

- To identify the defining characteristics of electricity generated from geothermal resources and compare geothermal electricity with electricity from other sources in terms of cost and land use requirement
- To introduce the different technologies used for geothermal power generation and highlight the situation(s) in which each different technology is preferred over the others
- To facilitate the understanding of the processes and components present in the operation of geothermal power plants by using examples and simulations.
- To provide exercises and simulations that reinforce the learning material
- To highlight limitations in current technology and introduce advanced concepts in the area of geothermal power generation that are interesting and encourage new thinking and possible areas for future research
4 METHOD OF ATTACK

The method of attack involves doing background research in the form of a literature review, where the different systems are identified and described. After the preliminary study, each system is examined further by analysing the flow chart and also the thermodynamics of the system, power cycles as well as the main individual components, such as turbines, heat exchangers and steam separators. Individual components are analysed mainly in terms of thermodynamics, however special mechanical and technical considerations that arise with geothermal power plants (such as for example corrosion) are also discussed. Also special considerations will be highlighted and explained. The method of thermodynamic and technical analysis includes examples of modelling and simulation, using Engineering Equation Solver (EES) and Microsoft Excel for assistance in calculation of thermodynamic properties. The examples are meant to be understood and executed by students.
5 PRESENT SITUATION

5.1 Geothermal Electricity Introduction

Electricity generation from geothermal energy, in its most simple form, is not much different from power generation in a conventional steam turbine operating on the Rankine Cycle. The major difference lies in the fact that the heat source is in the Earth’s crust as opposed to combustion of fossil fuels in conventionally fired power plants (see Figure 5-1). According to (Dickson and Fanelli, 2004) “A geothermal system is made up of three main elements: a heat source, a reservoir and a fluid, which is the carrier that transfers the heat.”. Geothermal systems must have a certain temperature before they can be considered adequate for power generation, and depending on the nature, conditions (temperature, pressure, and enthalpy) and chemical composition of the geothermal fluid, a different type of energy conversion system may be optimal. Other factors such as techno-economical feasibility and capital costs also determine which method of energy conversion is best suited for each situation.

Figure 5-1 Illustration of how a geothermal system is used for power generation
5.2 Geothermal Resource

For geothermal electricity to be produced in a feasible and renewable way, the resource must be above a certain temperature and enthalpy. The lowest temperature of a geothermal resource used for power generation currently is 57°C, this is being achieved at the Chena Hot Springs, Alaska geothermal system (Erkan et al., 2008). This project is a commercial one and is a part of the community’s vision to be self sufficient when it comes to energy, food and heating (Holdmann, 2007). Geothermal resources are classified based on the thermodynamic state (temperature, pressure, and enthalpy) of the resource. Commercial geothermal resources that are being utilized today use a hydrothermal resource (normally between 100°C and 300°C), meaning that geothermal energy comes from a resource containing mostly water, whether it exists in the form of a liquid or a vapour or a mixture of both. The geothermal resources with the most potential for power generation tend to be located in regions with a high geothermal temperature gradient and high tectonic activity. A high geothermal temperature gradient means that the rate at which the temperature changes with depth below the Earth’s surface is relatively high when compared with the global average.

5.3 Characteristics of Geothermal Electricity

There are some characteristics of geothermal electricity and geothermal power plants that can be recognised instantly, characteristics include:

- Renewable: if it is utilised in a sustainable way. Geothermal energy is, by definition, renewable as it is the heat from the centre of the Earth, which is essentially inexhaustible when compared on a timescale (geothermal heat lifetime: billions of years) to the human lifetime. However the geothermal resource must be managed such that the rate of utilisation of the geothermal fluid matches the heat rate. If the geothermal fluid utilisation is over-exploited it may result in the geothermal reservoir having a significantly lower temperature with time. If the resource has a lower temperature, it will not be as productive, compromising the ability of future generations to meet their energy need from this resource, and therefore not being sustainable.
• High availability: the geothermal energy source does not change much over time (years) especially when compared to other renewable sources such as wind or solar energy (which can vary by the minute and are sensitive to weather conditions). This characteristic makes geothermal electricity suitable for base-load power generation. Base load is the minimum amount of power that an electricity utility must provide to customers, given a reasonable estimate of the demand. Power plants that provide base load are usually those power plants which have cheap operating costs, and provide reliable power, since this is the portion of the power demand that is always present.

• High geothermal potential is highly geographically specific.

• The operating and maintenance costs associated with geothermal power plants are different from conventionally fired power plants, since there is no “fuel cycle”. A fuel cycle refers to the different required processes and stages that a fuel must go through in its life-cycle including extraction, preparation for utilisation, transportation, utilisation and also disposal or treatment along the way. Geothermal power plants have no fuel cycle because they do not combust fuel; the geothermal fluid is heated underground as a result of the heat radiating from the Earth’s centre.
### 5.3.1 Comparison of geothermal with other electricity sources

<table>
<thead>
<tr>
<th>Plant Type</th>
<th>Capacity Factor (%)</th>
<th>U.S. Average Levelised Costs (USD in year 2009)/Megawatthour for Plants Entering Service in 2016</th>
</tr>
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<tbody>
<tr>
<td></td>
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<td>Levelised Capital Cost</td>
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<tr>
<td>Natural Gas (Conventional, Advanced CC, CCS, Combustion Turbine)</td>
<td>30-87</td>
<td>17.5 - 45.8</td>
</tr>
<tr>
<td>Hydro</td>
<td>52</td>
<td>74.5</td>
</tr>
<tr>
<td>Wind</td>
<td>34</td>
<td>83.9</td>
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<td>Geothermal</td>
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<td>Coal (Conventional, Advanced, CCS)</td>
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<td>Solar Thermal</td>
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<td>259.4</td>
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* Costs are expressed in terms of net AC power available to the grid for the installed capacity
Table 2 Description of land use intensity requirement for different sources of electricity (McDonald et al., 2009)

<table>
<thead>
<tr>
<th>Source</th>
<th>Land use intensity km²/TWh/yr</th>
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<tr>
<td>Nuclear</td>
<td>2.4</td>
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<tr>
<td><strong>Geothermal</strong></td>
<td><strong>7.5</strong></td>
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<tr>
<td>Coal</td>
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<td>Solar Thermal</td>
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<td>Solar Photovoltaic</td>
<td>36.9</td>
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<tr>
<td>Hydropower</td>
<td>54.0</td>
</tr>
<tr>
<td>Wind</td>
<td>72.1</td>
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</table>

**Strengths**

When comparing geothermal electricity to electricity from other sources mentioned above the main strength lies in the fact that it can be used for base-load generation, while being unobtrusive to the surroundings, having a relatively low land use intensity (see Table 2), moderate levelised system costs (see Table 1), no “fuel cycle” and being relatively benign when it comes to the environment. Electricity generated from wind and solar resources is not suitable for base-load power generation because the nature of the resource is intermittent; sunlight is not always available and the wind is not always present. Capital investment and land requirement in solar electricity is very high, while the conversion efficiency is low. Biomass power plants require a continuous supply of biomass as a fuel input and this adds a significant operation cost for growth, extraction, processing and transport before it can be utilised for power generation. Large hydropower plants tend to be huge projects which can disturb the natural flow of rivers and the wildlife. Nuclear power plants have high investment costs and require strict safety regulations for normal operation; also there is an issue the nuclear fuel cycle, where the safe disposal of nuclear waste is a challenge. Nuclear power plants also have far-reaching consequences in the event that there is a failure, hence the heavy emphasis on safety.

**Limitations**

The potential for geothermal power generation is limited to areas of high tectonic activity, more specifically those areas with a high geothermal temperature gradient and also some kind of geothermal reservoir (with the exception of Enhanced Geothermal Systems which
are covered as a separate chapter in the CompEdu platform). This fact also means that there must be significant amounts of research and planning done on a potential resource site before the construction of a plant and also geothermal power generation requires drilling deep beneath the surface of the Earth into regions of high temperatures and pressures which is very expensive. Another limitation for geothermal power generation is the low electrical conversion efficiency when compared to other thermal power plants, due to the fact that most geothermal plants utilise a saturated liquid resource at a moderate pressure and enthalpy in the initial stage as opposed to saturated or superheated steam at a high temperature, pressure and enthalpy in the case of conventional fossil fuel fired thermal power plants. To summarise the main limitations of geothermal electricity when compared to the mentioned sources of electricity, they are that there is a large amount of planning and initial expense required for a potential resource, there is low electrical conversion efficiency and it is highly geographically specific.

5.4 Geothermal Electricity in the World Today

Geothermal power plants are operating in many places across the globe as seen in Table 3 which shows the top ten countries with the highest installed capacity of geothermal power in the world today. For an extended list showing the top twenty four geothermal power generating countries see Table 10 in the appendix. Since geothermal energy potential is highly geographically specific (at the moment), its utilisation tends to be limited to these regions. Table 4 compares the amount of electrical energy generated by various sources in 2008 where geothermal represents a small fraction of global electricity generation.

<table>
<thead>
<tr>
<th>Country</th>
<th>Installed Capacity (MWₑ)</th>
<th>Rank</th>
</tr>
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<tbody>
<tr>
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<tr>
<td>Philippines</td>
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</tr>
<tr>
<td>Indonesia</td>
<td>1197</td>
<td>3</td>
</tr>
<tr>
<td>Mexico</td>
<td>958</td>
<td>4</td>
</tr>
<tr>
<td>Italy</td>
<td>843</td>
<td>5</td>
</tr>
<tr>
<td>New Zealand</td>
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<tr>
<td>Iceland</td>
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<tr>
<td>Japan</td>
<td>536</td>
<td>8</td>
</tr>
<tr>
<td>Source of Electricity</td>
<td>Energy generated by these power plants in 2008 (TWh)</td>
<td></td>
</tr>
<tr>
<td>--------------------------------------------</td>
<td>---------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>Geothermal Power plants (IEA, 2008)</td>
<td>58</td>
<td></td>
</tr>
<tr>
<td>Wind Turbines (GWEC, 2008)</td>
<td>260</td>
<td></td>
</tr>
<tr>
<td>Hydropower plants</td>
<td>3288</td>
<td></td>
</tr>
<tr>
<td>Nuclear power plants</td>
<td>2731</td>
<td></td>
</tr>
<tr>
<td>Coal/Peat</td>
<td>8263</td>
<td></td>
</tr>
<tr>
<td>Oil</td>
<td>1111</td>
<td></td>
</tr>
<tr>
<td>Gas</td>
<td>4301</td>
<td></td>
</tr>
<tr>
<td>Other</td>
<td>169</td>
<td></td>
</tr>
<tr>
<td>Total electricity generated globally</td>
<td>20181</td>
<td></td>
</tr>
</tbody>
</table>

There is much unrealized potential for geothermal electricity in certain countries. For example in Ethiopia the installed capacity is just 7.3 MWₑ, however the estimated potential is between 640 and 1710 MWₑ (Holm et al., 2010). Another example is that of Chile where, as of 2010, there were no installed geothermal power plants, however the potential is estimated to be between 780 and 1630 MWₑ (Holm et al., 2010). The opposite situation also exists, where countries which do not have particularly high geothermal gradient or tectonic activity such as Germany have considerable installed geothermal power capacity. Figure 5-2 below shows the trend of installed capacity of geothermal electricity plants around the world between 1916 and 2007 and shows that over the years geothermal power plants have become increasingly popular with an estimated 10 GWₑ of installed capacity globally in 2007. The total figure is quite low when compared to other energy sources and to put it into perspective, in 2008 there were 924 GWₑ of installed hydropower capacity and 372 GWₑ of installed nuclear power capacity (IEA, 2010).
5.4.1 The different roles geothermal power generation play around the world

Since geothermal power plants are geographically specific, geothermal power generation plays different roles in countries’ energy mixes. For example, in the United States in California, the Geysers geothermal field is one of the only systems which utilises dry steam resources and provides 2.5 GW of electricity, however this only represents 4.5% of the electricity generation in the state. On the other hand, Iceland’s geothermal heat and power plants provide 25% of the electricity load and 90% of the heat requirements, with a total of 62% of total energy coming from geothermal resources (Bertani, 2010). In New Zealand, one of the first nations to utilise geothermal electricity commercially, geothermal electricity accounts for 13% of total power generation in the country. Indonesia is currently the third largest geothermal power producer after the USA and the Philippines, however Indonesia has the highest geothermal power generation potential in the world. Current power generation stands at 1197MW or about 3.7% of electricity supply, however by the year 2025, Indonesia aims to have installed over 9000MW of geothermal power capacity.
5.4.2 Types of power plants for geothermal power generation

There are three main technologies used for energy conversion in geothermal power plants today. The names of these types of power plants are the dry steam power plant, flash steam power plant and the binary cycle power plant.
6 GENERAL ASPECTS

This chapter is dedicated to explain general aspects of components that are used in most thermal power plants and which are also utilised in both the dry steam and flash steam geothermal power plants.

6.1 Turbine

6.1.1 Expansion in the turbine

Expansion in the turbine is the process primarily responsible for power generation. The thermodynamics behind the power generated in a dry steam power plant are such that the power is given by the product of the mass flow, the enthalpy change across the turbine and the turbine efficiency.

\[ \text{Power output}_{\text{turbine}} = \dot{m} \times \eta_{\text{turbine}} \times (h_1 - h_2) \]  
\[ \text{Eq. 1} \]

Where,
\[ \dot{m} \] - Mass flow of fluid through the turbine
\[ \eta_{\text{turbine}} \] - Isentropic efficiency of the turbine
\[ h_1 \] - Enthalpy at the entry of the turbine
\[ h_2 \] - Enthalpy at the exit of the turbine

Turbine efficiency

The isentropic efficiency is given by the ratio of the actual enthalpy change across the turbine and the ideal enthalpy change (which occurs at constant entropy). This isentropic efficiency can be expressed mathematically while referring to Figure 7-2 for the different stages of the process of expansion:

\[ \eta_{\text{turbine}} = \frac{h_1 - h_2}{h_1 - h_{2s}} \]  
\[ \text{Eq. 2} \]

Where,
\[ \eta_{\text{turbine}} \] - Isentropic efficiency of the turbine
\[ h_1 \] - Enthalpy at the entry of the turbine
\[ h_2 \] - Enthalpy at the exit of the turbine
\[ h_{2s} \] - Enthalpy at the exit of the turbine if the process was isentropic (ideal)

Conventional (subcritical) fossil fuel fired power plants operate using the steam conditions: Temperature - 538-566°C, Pressure – 16-20 MPa and significant superheat and reheat of
the steam which is used in the cycle, which means that the steam has a high enthalpy and the turbine has a higher efficiency since it will be operating utilising steam of high enthalpy and low moisture. Geothermal power plants use steam resources with pressures normally below 7MPa, temperatures around 170°C and utilising an essentially saturated steam resource (no superheat or reheat of the steam). Utilising a relatively low enthalpy (low pressure and temperature), saturated steam resource, results in considerable moisture formation during expansion in the steam turbine and relatively high mass flows are required for a moderately sized power plant (McCloskey, 2003)

6.1.2 **Turbine losses**

It helps to understand generally where and how losses occur in steam turbines used in thermal power plants. The effect of moisture is discussed in the section titled ‘Moisture and turbine efficiency’ on page 43. By understanding the mechanics behind losses in a turbine, the measures taken to improve the geothermal steam turbine performance can be appreciated.
Figure 6-1 Illustration showing the different losses occurring in a turbine stage, the different turbine components are labelled in black font and the different losses are labelled in red font and the locations of where specific losses occur are shown by the red lines (Toshiba, 2011).

Figure 6-1 shows the different losses that occur in a conventional steam turbine stage, these losses are normally a result of some form of leakage or non-ideal flow through the turbine. These losses occur because of several things, the turbine is not perfectly airtight, the turbine operates at high pressures and temperatures and also there are losses associated with friction and irreversibilities associated with changing the steam flow direction. The losses are briefly described:

Profile Loss – Profile loss is associated with the pressure loss and frictional loss which occurs over the nozzle and rotor blades due to their shape and how they change the direction and velocity of flow
Blade tip leakage loss – This loss occurs as there is steam which passes in the space between the rotor and turbine casing, normally there are blade fins which help to minimise this loss.

Labyrinth leakage loss – Occurs as a result of steam leaking through the labyrinth seal between the nozzle row and turbine rotor blade row.

Blade root loss – a turbine blade root is designed with structural integrity as the priority and not aerodynamic efficiency, because of this less than optimal aerodynamic design there is a loss.

Secondary flow loss – secondary flow occurs at the boundary layer of the flow and results in vortices being created that disturb the main flow and extract energy from it.

6.1.3 Turbine design

Conventional turbine technology is used in geothermal power plants utilising both impulse and reaction types of turbines. The rotor in an impulse turbine is driven mainly by the kinetic energy of the steam. The majority of the pressure drop, kinetic energy increase (and temperature drop) occurs over the stator or nozzle with very little occurring over the rotor. In a reaction turbine a considerable portion of the pressure drop occurs over the rotor (at least 50%) as a result the rotor is driven by the difference in pressure between the convex and concave side of each blade (Figure 6-2).
In conventional steam power plants an impulse turbine is used as the initial high pressure stage because it offers a great temperature drop, and as a result controls the temperature to which the rotor is exposed (McCloskey, 2003) (See section ‘13.2.6 Control of impulse turbine stage on temperature’ for explanation). Normally impulse turbine stages are used for high pressure sections of the turbine while reaction turbines are used for low pressure sections of the turbine.

6.2 Cooling and condensation

6.2.1 In the condenser

The condenser is a heat exchanger that cools and condenses the exhaust steam to water. There are two categories of condenser; the direct contact condenser and the surface type condenser. The surface type condenser uses a surface, such as the outside of a tube, to facilitate heat exchange between a cooling water circuit and the steam turbine exhaust, such that both the cooling water and the steam turbine exhaust are physically separated.
and heat exchange occurs across the surface (Figure 6-3). The cooling water passes through tubes in the condenser while steam passes over these tubes, directed by baffles. The cooling water condenses the steam on the tubes and the steam condensate exits the condenser. The cooling water (which is now warm) exits as the water outlet where it goes to a cooling tower. There is also an outlet to the ejector vacuum system, which mechanically maintains the sub-atmospheric pressure in the condenser.

![Figure 6-3 Illustration of a surface condenser showing the flow of cooling water in blue and the steam flow in red.](image)

Direct contact condensers are also called sometimes called “Injection condensers” or “Barometric condensers”. They do not use a surface to facilitate heat exchange, and for this reason, both the cooling medium and the turbine exhaust are in direct contact. Normally for direct contact condensers the cooling medium is sprayed into the turbine exhaust and condensation is achieved (Figure 6-4).
Figure 6-4 Illustration of a direct contact, or barometric condenser, the steam enters the condenser where water is sprayed into the flow stream causing the steam to condense to water, there is an outlet for air and non-condensable gas extraction and the condensate and warm cooling water are collected at the bottom of the apparatus.

The thermodynamics of the condenser described will be discussed further in Eq. 3. The energy balance in the condenser is given by the realisation that the heat transferred to the cooling water results from the condensation of the steam from the turbine outlet.
This equation can be reformulated by substituting the letters at each corresponding stage from Figure 6-5, to get Eq. 4, conservation of energy and Eq. 5, conservation of mass.

\[ m_a \times h_a + m_c \times h_c = m_b \times h_b + m_d \times h_d \]  \hspace{1cm} \text{Eq. 4}

\[ m_a + m_c = m_b + m_d \]  \hspace{1cm} \text{Eq. 5}

Ideally, the condenser should convert 100% of the steam to liquid in order to maximise efficiency, however “conversions of 80-95% would still provide a sufficiently large pressure change” (Glassley, 2010). This points out that the pressure change associated with the change of steam to liquid water is more important than lowering the temperature of that water, therefore in practice the temperature of the water leaving the condenser can in some cases be relatively high.
As indicated by the name, dry steam power plants utilise dry steam resources emerging from production wells. Dry steam is steam that is either saturated or superheated, with no liquid present. Energy conversion is relatively straightforward when compared to other types of geothermal power generation because of the nature of the resource. All that is really necessary is a turbine to extract work from the steam and a condenser to condense it to facilitate reinjection (see Figure 7-1 and Figure 7-2). The main challenge with operation of dry steam plants is to maintain the dry steam resource. Dry steam plants account for 12% of geothermal plants by number and generate 26% of worldwide geothermal power capacity, however only an estimated 5% of hydrothermal resources are dry steam resources (DiPippo, 2008).

Figure 7-1 Flow chart describing the basic processes involved in a dry steam power plant
7.1 System

Even though the process of energy conversion is straightforward in a dry steam power plant, important decisions must be made concerning the flow rate and desired power output from the power plant. The power available from the geothermal well to a power plant is
primarily dependent on the mass flow and enthalpy of the resource. The mass flow is determined by the pressure selected for extraction of steam from the wellhead, selection of a lower pressure (greater pressure difference between the reservoir and wellhead) will yield a greater mass flow. However the potential enthalpy change across the steam turbine also varies with the pressure selected, the lower the pressure selected, the smaller the potential enthalpy change across the turbine. Therefore a compromise must be made in order to get the desired power output; a feasible mass flow at a reasonable enthalpy change.

7.1.1 Selecting the wellhead pressure

Imagine a geothermal steam resource in a reservoir with a temperature of 235°C and saturated steam conditions, meaning that it is at a pressure of 3060 kPa. In addition to these reservoir conditions, it is assumed that the condenser in the system is operating at 50°C and a sub-atmospheric pressure of 12.34 kPa, which are nominal conditions for a condenser in a geothermal power plant (Figure 7-3).

![Figure 7-3 T-s diagram showing the reservoir conditions and condenser conditions in a geothermal reservoir](image)

The reservoir has a mass flow profile that is representative of a typical geothermal reservoir; mass flow initially increases rapidly, followed by a more gradual increase as the
pressure difference between reservoir pressure and the selected wellhead pressure increases (Figure 7-4).

It can be found in steam tables that the enthalpy of the steam before the wellhead extraction will be 2803 kJ/kg and the exact pressure will be 3060 kPa. A compromise between mass flow and enthalpy change must be made to determine where the optimal power output occurs. An h-s diagram in Figure 7-5 shows how the enthalpy change (Δh) varies with a decreasing selected wellhead pressure.
Figure 7-5 h-s diagram showing how isentropic enthalpy change across a turbine varies depending on the wellhead pressure selected for a given condenser pressure

The potential enthalpy change ($\Delta h$) occurs between the wellhead (before the turbine inlet) and the turbine exhaust/inlet to the condenser, the enthalpy change is depicted by $\Delta h$ and the double arrowed blue lines in Figure 7-5. Decreasing selected wellhead pressure is shown in Figure 7-5 by the enthalpy change arrows moving to the right of the figure. This case has several simplifying assumptions. Enthalpy change is assumed to be isentropic in nature, meaning that conversion in the steam turbine occurs with no losses, in reality this is impossible; however the assumption is included for the sake of showing that potential enthalpy change varies depending on the wellhead pressure selected. The assumption also avoids the calculation of turbine isentropic efficiency since this value changes depending on the level of moisture present in the steam flow stream. Another assumption is that even though the pressure at the wellhead is lower than the pressure in the reservoir, the enthalpy in the reservoir and the enthalpy at the wellhead are the same; the process of lowering the pressure is isenthalpic.

In Figure 7-5 the enthalpy change decreases with a lower wellhead pressure for a given set of condenser conditions. By plotting the variations of both the enthalpy change and mass
flow the problem can be more easily understood and the optimal wellhead pressure can be selected in order to maximise power output (see Figure 7-6 below). Figure 7-6 shows how the enthalpy change ($\Delta h$) and mass flow rate across the turbine vary with wellhead pressure.

![Variation of Enthalpy Change and Flow Rate with Pressure](image)

*Figure 7-6 Variation of Enthalpy Change and Mass Flow Rate with Wellhead Pressure at a geothermal wellhead (Glassley, 2010)*
Table 11 in the appendix displays the data used to create Figure 7-6. It also contains information about the theoretical power output if turbine isentropic efficiency was equal to 1.0. Since the enthalpy change decreases with the pressure selected at the wellhead and the mass flow increases with decreasing wellhead pressure, the power must be calculated at different pressures and the optimal point of operation can be found in this way. Figure 7-7 shows the variation of power output with wellhead pressure for the stated geothermal production well with the assumption that the turbine isentropic efficiency is 1.0. It shows that power peaks at a value that is not far from half of the reservoir pressure, in order to get an optimal power output there must be a compromise; a moderate mass flow and enthalpy change.

![Variation of power output to wellhead pressure](image)

*Figure 7-7 Variation of power output with wellhead pressure for the specific geothermal reservoir*
7.2 Processes and components

The dry-steam power plant has several processes and requires different components in order to operate effectively. In Figure 7-8, there are a few additional components shown when compared to Figure 7-1 which are important for normal operation of the power plant.

Where in the above figure:
1-Production well
2-Particle remover
3-Moisture remover
4-Steam turbine
5-Electrical generator
6-Condenser
7-Cooling tower
8-Injection well

The processes which occur in the power plant can be described by referring to the different points which represent components in Figure 7-8. As the geothermal fluid is extracted from the production well (Point 1) at a specific pressure, it passes through the particle and moisture removers (Points 2 and 3 respectively). These are important components which protect the turbine from damage from entrained particles and moisture, however not as important in flash steam plants where the geothermal resource is more “liquid-dominated”. Particle removal and moisture removal in geothermal power plants are similar to steam separation and the principle is described in section 8.2.2 (Separator). After the steam exits
the moisture and particle removers it is ready for admission to the steam turbine. In the steam turbine, the thermal energy of the steam (some of the thermal energy is converted to kinetic energy in the nozzle or stator of the turbine) is converted to mechanical energy in the rotation of the turbine shaft which turns an electrical generator shaft (Electrical generator is at Point 5) which converts this mechanical rotation into electrical energy. At the exit of the steam turbine the steam is cooled and condensed by the condenser (Point 6) and cooling tower (Point 7) in tandem. The geothermal fluid is often reinjected (see section 13.2.5 ‘Reinjection’ on page 130) to the geothermal reservoir underground via the injection well (Point 8) in order to maintain the resource for sustainable power generation.

7.2.1 Moisture and turbine efficiency

Turbine performance is negatively affected by increased moisture content in the steam in addition to different types of losses. The different losses include pressure losses, leakage losses and losses due to friction. Pressure losses from the steam occur as it flows through pipes and valves, leakage losses occur as the flow is not completely sealed and leakages occur at the tip of the rotor, between turbine stages and at various openings throughout the turbine. Losses also occur due to roughness of the turbine blades which cause frictional losses, surface roughness can occur as a result of imperfect machining of the blades as well as erosion.

Moisture begins to appear when the steam expands into the mixed flow region of the Mollier Diagram. There are two main ways in which moisture negatively affects turbine efficiency; by disturbing the flow pattern of the steam and by inducing a resistive force against the moving turbine blade through collision of droplets with the blade (CANDU, 1994).

The first way that turbine efficiency is affected by moisture arises from the fact that a steam turbine is designed to operate with a specific steam flow pattern for maximum efficiency, one where the flow of steam into the turbine is uniform and at a particular speed. If moisture begins to condense in the flow, energy from the steam provides a driving force in order to move the entrained, relatively dense water droplet. Some of the kinetic energy in the steam will be used to carry the water droplet, slowing the average speed of the steam flow. To visualise this, the concept of a strong wind carrying raindrops can be used. Since the water droplet is denser than the steam, the droplet moves more slowly and whenever the steam flow direction is changed the droplet move centrifugally outward due to its higher density.
(see Figure 7-9, Figure 7-10). The larger the droplet is, the slower it will move and the greater the disturbance to the steam flow pattern.

*Figure 7-9 Diagram showing how steam with entrained water flow through a stator (nozzle)*
Figure 7-10 Diagram showing the paths that steam and entrained water droplets take through a turbine stage

The second way in which moisture degrades turbine efficiency is related to the slow movement of the droplets. Since there are water droplets in the steam flow, the flow stream becomes heterogeneous or non-uniform. The fast moving turbine blades collide with these droplets which results in the blades moving slower. This can be visualised by looking at Figure 7-11. The left half of the diagram shows the relative motion of steam and water droplets, the steam follows the blade profile, while the water droplets are centrifuged and move at a lower velocity. The result of this low speed is that the fast moving turbine blade collides with it after a specific time.
Figure 7-11 Diagram showing the mechanism behind collision of water droplets with turbine rotor blades

The resistive force can also be visualised by walking at a given velocity into the wind (which is blowing at a specific velocity) without rain versus trying to walk at the same velocity into wind while it is raining heavily; either it will expend more energy to walk at the same velocity as the first case or if the same amount of energy is used during walking as in the first case the end result will be a lower velocity in the rainy case.

When a part of the expansion occurs in the superheated steam region, with the rest of expansion occurring in the mixed liquid-vapour region, there will be two different turbine efficiencies; one applicable to each region (see Figure 7-12). As expansion occurs between point 1 in the superheated region and point a on the saturated vapour line a straight line which indicates a constant turbine efficiency, where the steam flow contains no moisture. Between point a on the saturated vapour line and point 2 in the mixed liquid vapour region of the T-s diagram, a curved line shows that the expansion occurs and the turbine efficiency (indicated by the gradient of the curve) is decreasing due to increasing quantity of performance-degrading moisture in the steam flow.
Figure 7-12 T-s diagram illustrating turbine expansion from the superheated region into the mixed liquid-vapour region.

In a steam turbine there are normally several rows of stator-rotor stages all having different, individual efficiencies depending on the conditions under which they operate. In this example of determining the effect of moisture on turbine efficiency, it is an average turbine efficiency that is found for expansion in the superheated and mixed liquid-vapour region. The efficiency found is not the efficiency of a specific turbine stage; it is an average of the turbine efficiencies for individual turbine stages for the superheated region and mixed liquid vapour region of expansion.

The magnitude of the effect of moisture on turbine efficiency can be discussed through an example. The two efficiencies are the dry-steam turbine efficiency ($\eta_{turbine\ dry}$), which is applicable to the superheated (dry) region and a wet-steam turbine efficiency ($\eta_{turbine\ wet}$), which is applicable to the mixed liquid-vapour region. The dry-steam turbine efficiency is usually known through information provided by the turbine manufacturers; however the wet-steam turbine efficiency depends on the condition of the steam exiting the turbine. The wet-steam turbine efficiency can be estimated using the empirical Baumann rule. According to (Baumann, 1912) it is stated that: “Tests have shown that when the steam is wet the
efficiency is reduced. Assuming the efficiency follows a continuous curve for superheated and wet steam when plotted with entropy as a basis, the efficiency will change by 1 percent for each 1 percent variation in wetness.”. This rule of thumb is empirical, however has remained a mainstay in the area of steam turbines. Using this rule for this example, the average wet turbine efficiency over the mixed liquid-vapour region can be found using the dry-steam turbine efficiency and moisture content of the steam exiting the turbine (DiPippo, 2008).

\[
\eta_{\text{turbine wet}} = \eta_{\text{turbine dry}} \times \left( \frac{x_a + x_2}{2} \right) \quad \text{Eq. 6}
\]

Where,

\(\eta_{\text{turbine wet}}\) - The wet steam turbine efficiency

\(\eta_{\text{turbine dry}}\) - The dry steam turbine efficiency

\(x_a\) - The steam quality at point a on the saturated vapour line \((x_a = 1)\), steam quality indicates the mass of steam present compared to the total mass of the steam-water mixture, when \(x = 1\) the fluid is a saturated vapour and when \(x = 0\) the fluid is a saturated liquid.

\(x_2\) - The steam quality at the exit of the turbine. In Eq. 6 the steam quality at point a and point 2 are used in order to find an average turbine efficiency for expansion in the mixed liquid vapour region (which is the expansion between point a and point 2).

Improving turbine efficiency

Since it is known where and how the losses occur throughout a turbine stage (see section 6.1.2 Turbine losses on page 28), design processes which are implemented in practice to minimise losses can be explored. A few interesting performance increasing practices are discussed:

- Low pressure blades

Geothermal power plants generate power at lower temperatures when compared to fossil fuel fired power plants, therefore it is even more important to extract power from the low pressure section of the turbine. This creates a challenge since the low pressure section of the turbine requires the longest blades for power extraction and also is exposed to the most corrosive conditions (entrained moisture in the steam flow). The low pressure section requires extremely long blades because of the large volume of steam which exists at the low pressure. By looking at Table 5, the volume occupied by 1 kg of steam at 50°C is more than 30 times greater than the volume occupied by steam at
150°C. If the last stage blades are not long enough to accommodate the large volume flow, the enthalpy of the steam will not contribute to moving the turbine blades and rotating the turbine shaft; the energy would be wasted.

**Table 5 showing the specific volume and pressure of saturated steam at 150, 100 and 50°C**

<table>
<thead>
<tr>
<th>Point</th>
<th>Temperature (°C)</th>
<th>Pressure (kPa)</th>
<th>Specific Volume (m³/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>150</td>
<td>476</td>
<td>0.3929</td>
</tr>
<tr>
<td>2</td>
<td>100</td>
<td>101</td>
<td>1.674</td>
</tr>
<tr>
<td>3</td>
<td>50</td>
<td>12.3</td>
<td>12.04</td>
</tr>
</tbody>
</table>

Long blades also imply that there is exposure to high centrifugal forces at the blade roots. One answer to this challenge is to use special alloys containing, for example titanium, which are lightweight and resistant enough to be used for the longer last-stage blades (Yamada, 2001). Titanium has about half the density of 12Cr steels which allows for longer last-stage blades without an increase in centrifugal stresses. The practical limitation for blades constructed of 12% Cr martensitic stainless steel was reached with 840 mm (33.5 in.) blades operating in 3600 rpm machines and 1200mm (48 in.) blades operating in 3000 rpm machines. In contrast, titanium offers an opportunity to go to 1000 mm (40 in.) and 1350 mm (54 in.) blades for 3600 rpm and 3000 rpm machines, respectively.(McCloskey, 2003)

- **Integral shroud blades**
  Integral shroud blades are blades which have an individual shroud at the blade tip of each blade. They can dampen vibratory stresses by more than 90% when compared to conventional grouped blades. The integral shroud blades operate by untwisting during increased centrifugal forces resulting in the shroud of each blade contacting those of adjacent blades, creating a solid ring for the blade stage. Because there is a solid ring around the blade stage there are decreased blade tip leakage losses (MPS, 2011) (Sakai et al., 2009).
In Figure 7-13 the untwisting of the turbine blade at the blade tip is illustrated, the turbine blade is exposed to an outward force which tends to make the blade straight and no longer twisted (by straight it is meant that the blade will have the same profile and direction of the blade at the blade root). This untwisting facilitates the use of integrated shroud blades because it allows the blades to be able to untwist at the rated speed, causing a solid ring to be formed at the shroud. In Figure 7-14 the shape and interaction of the blade shrouds are shown.
Axial exhaust from the turbine to the condenser

This method of improving performance is associated with the exhaust of the turbine, there are several possible configurations for the turbine relative to the condenser and ideally, the one that creates the lowest pressure drop between them is the most optimal. A low pressure drop between the turbine exhaust and the condenser means that the condenser can achieve lower pressures which increase the pressure ratio and furthermore increase power output. The different configurations include the top exhaust, down exhaust and axial exhaust. The Hellisheidi Geothermal Power Plant uses a single-cylinder axial exhaust which minimises pressure loss. Single-cylinder axial exhaust does not change the direction of the flow, the flow from the turbine exits axially outward and directly to the condenser (Works, 2007).

Material considerations

In geothermal power plants there is a major consideration regarding the material with which the turbine should be constructed. Geothermal fluids have varying degrees of
corrosiveness, depending on the chemical composition, and it is recommended that different materials be tested to ensure adequate performance under conditions similar to those occurring during operation (DiPippo, 2008). The geothermal fluid undergoes several processes as described in 7.2, these processes serve to condition and prepare the steam for expansion in the steam turbine. However, in geothermal power plants corrosive substances existing in the steam are between 100 and 1000 times more concentrated than the quantities of these substances in the steam used in fossil fuel fired power plants. This fact make geothermal steam turbines susceptible to different types of deterioration corrosion; Stress corrosion cracking (SCC), corrosion fatigue, erosion corrosion (Sakai et al., 2009).

Stress corrosion cracking according to (Cottis, 2000) is cracking due to a process involving conjoint corrosion and straining of a metal due to residual or applied stresses. It is characterised by a significant loss of mechanical strength with little loss of metal, meaning that it is not easily observed during inspection and can be the cause of sudden failures. It is believed that the cracking occurs initially due to the corrosive particles in the environment and that stress accelerates this process. Corrosion fatigue is the process of deterioration of mechanical properties due to cyclic loading in a corrosive environment. The turbine is a rotating piece of equipment and hence undergoes fatigue based on the nature of its operation. Erosion corrosion is the combined action of corrosion and erosion and occurs in the turbine where small corrosive droplets and particles impinge the turbine blades. The turbine blades are most susceptible in the low pressure stages where the flow contains higher quantities of fluid droplets. Low pressure turbine stages with long blades are subjected to extreme centrifugal forces and high tip speeds deform the blades.

The Fuji Electric Review (Sakai et al., 2009, Yamada, 2001) has made publications about how Fuji approaches corrosion resistance and turbine efficiency described in following list.

To improve corrosion resistance, actions are taken including:

- **Thermal spray coating**
  A special process is used to cover the vulnerable areas with a corrosion resistant material

- **Treating areas which will be exposed to high stress using shot peening**
  Shot peening process impacts the target area with steel balls and as a result induces a compressive stress. This process enhances the resistance to cracking from corrosion
phenomena and is normally applied to areas of the rotor and stator such as the blade root and the rotor grooves which are susceptible

- Choice of turbine material

Depending on the properties of the geothermal fluid in the region a certain amount of lab testing and simulation is done to determine which material is best suited for the specific conditions present at the location of the geothermal power plant.

*Table 6 Table showing the standard materials used for geothermal steam turbines (Sakai et al., 2009)*

<table>
<thead>
<tr>
<th>Part</th>
<th>Standard material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stator Blade material</td>
<td>13%Cr steel</td>
</tr>
<tr>
<td></td>
<td>• 16%Cr-4% Ni steel</td>
</tr>
<tr>
<td></td>
<td>• Ti-6% Al-4% V alloy</td>
</tr>
<tr>
<td>Rotor material</td>
<td>1% Cr-MoNiV steel</td>
</tr>
<tr>
<td></td>
<td>• 2% Cr-MoNiWV steel</td>
</tr>
</tbody>
</table>

*Table 7 Description of common materials used in conventional steam turbines (McCloskey, 2003)*

<table>
<thead>
<tr>
<th>Part</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>High pressure and intermediate pressure blades</td>
<td>12 Cr martensitic* stainless steel</td>
</tr>
</tbody>
</table>

*Martensitic stainless steel – Small category of stainless steel characterized by the use of heat treatment for hardening and strengthening. Martensitic stainless steels are plain chromium steels with no significant nickel content. They are utilized in equipment for the chemical and oil industries and in surgical instruments. The most popular martensitic stainless steel is type 410 (a grade appropriate for non-severe corrosion environments requiring high strength) (AISI, 2011)*
7.2.2 Cooling and condensation

It is important to understand the advantage of having condensation and cooling in a geothermal power plant. In the past, a simple type of dry steam plant was used, where steam is expanded in a turbine and exhausted to the environment at atmospheric pressure without condensation. This type of power plant is called a non-condensing power plant (Figure 7-15). The advantage of this system is its simplicity; only a steam turbine is placed at the wellhead for power generation. The disadvantages of this kind of system are that there is no reinjection to the geothermal well and also that the expansion of steam in the turbine is limited to atmospheric pressure.

Figure 7-15 Flow diagram of a non-condensing geothermal power plant (the top diagram) compared with a condensing/reinjection geothermal power plant (the bottom diagram)
When a condenser is used, expansion is possible to pressures below atmospheric pressure increasing the pressure ratio across the turbine, increasing the possible work output as well as facilitating reinjection to the well. The characteristics of the non-condensing power plant make it favourable for use as way of initially measuring the potential power output from a geothermal well-head.

Flash and dry-steam geothermal power plants generally have a similar cooling/condensation processes; steam from the outlet of the steam turbine is condensed. The condensation is accommodated by cooling water and a cooling tower that transfer the heat from the cycle to the environment. This process can be explained in thermodynamic terms with heat and mass balances which will be illustrated in Figure 7-16.

![Figure 7-16 Flow diagram illustrating the condensation and cooling processes occurring in dry-steam and flash steam power plants](image)

The points in Figure 7-16 will be briefly described below:

- **a** - Moist steam from the outlet of the steam turbine is admitted to the condenser
- **b** - Warm cooling water for removal from the condenser to the cooling tower, the cooling water is warm because there is heat transfer between the turbine exhaust flow and the cooling water.
- **c** - Cooling water which is used to condense the steam from the turbine outlet
- **d** - Steam condensate, warm water for removal from the condenser to the cooling tower, this condensate is comprised of a relatively small portion of moisture that existed in the
turbine exhaust because the expansion happens into the vapour-liquid region of the Mollier diagram. The majority of the condensate exists because the heat from the turbine exhaust (which is mostly comprised of steam) is transferred to the cooling water and as a result condensation occurs.

- **e** - Steam condensate from the condenser which is cooled by allowing it to fall through the cooling tower
- **f** - Warm and moistened air removed from the cooling tower as a result of heat transfer from the steam condensate by the cooling tower fan and evaporation of a portion of the steam condensate.
- **g** - Ambient air for use in the cooling tower
- **h** - Blowdown, which is a portion of the cooling water that is used to control the concentration of minerals in the cooling water
- **i** - Cooled water which is returned to the geothermal reservoir where it is reheated, eventually returning to the production well
- **j** - Cooling water which is prepared for readmission to the condenser
- **k** - Make-up water used to replace water removed from the system

Generally condensers used in flash and dry-steam geothermal power plants tend to be of the surface type, in order to keep the circuits separate. The change from a heavily vapour-dominated steam-liquid mixture to liquid water also lowers the pressure to sub-atmospheric levels, this improves the pressure ratio across the turbine, and by extension increases the power output. The reduction of pressure to sub-atmospheric pressure levels can be explained in diagrams and providing information about the properties of the vapour and liquid phases of water in the condenser. Using the example of a condenser operating at 50°C the vacuum (sub-atmospheric pressure) is achieved because 1 kg of steam at 50°C occupies 12.04 m³ while 1 kg of water occupies 0.001 m³. This means that during the condensation of steam for every kilogram of steam that is condensed, the volume that the steam occupies initially decreases by 12 m³. This reduction gives rise to a partial vacuum since, initially, there is no matter to occupy the space made available through the condensation of the water. However because of the sudden creation of this partial vacuum, the air in the atmosphere will tend to rush in to fill the empty space because of the pressure difference between the condenser (at sub-atmospheric pressure) and the atmospheric pressure. For the vacuum to be maintained, atmospheric air must be kept out of the condenser. Geothermal fluids contain dissolved non-condensable gases which must also
be removed from the condenser. There are two main pieces of equipment that are used to maintain the partial vacuum in the condenser, Steam air ejectors and vacuum pumps (CANDU, 1990). If the air and non-condensable gases are not removed, there will be an undesirable lower pressure ratio, since the condenser will be operating at a higher pressure. Also the presence of gases in the condenser can result in corrosion problems arising from having air and possibly dissolved oxygen (oxygen dissolved in the water) in the condenser. The methods of air extraction are described in more detail.

Steam air ejectors

Steam air ejectors operate by using steam flowing through a nozzle and diffuser to evacuate the air in the condenser. In Figure 7-17 the high pressure steam enters a nozzle where a portion of its thermal energy is converted to kinetic energy. The increase in velocity decreases the dynamic pressure of the steam. The steam enters an area that acts as a diffuser (because of its larger area compared to the area of the steam nozzle exit) the ejector is connected to the condenser and the extremely low pressure that is created (lower than the condenser pressure), draws the air from the condenser into the ejector and the mixture of air and steam exits the ejector via another diffuser to a higher pressure. Air extraction systems must be located such that the steam condenses first (after some number of tubes in a surface condenser). If it was placed at the condenser entrance, extraction would result in mainly steam being extracted and not the air which infiltrates the condenser. Extraction is not limited to air but also to non-condensable gases that may exist in the turbine exhaust.
Figure 7-17 An illustration of a steam ejector which functions to remove unwanted air from the condenser and maintain the condenser (sub-atmospheric) pressure

Depending on the operation of a power plant there will be different requirements for air extraction. For example there is a high requirement for air extraction when a power plant is starting up versus during regular operation. For this reason there are usually two ejectors used, one that can extract large volumes of air during start-up called a “Hogging ejector” and one for regular operation called a “Holding ejector”. The disadvantage of using a Hogging ejector is that large volumes of steam are used and it is inefficient for meeting the air removal requirement for regular operation. From a system perspective, steam air ejectors are normally used such that there are several stages to the air removal. Normally there is a first stage ejector, then the air-steam mixture passes to an intermediate condenser where most of the steam is condensed. The air continues to a second stage ejector which removes more of the air and then the air-steam mixture continues to final condenser where the steam is condensed, while the air is removed to the atmosphere (see Figure 7-18).
Figure 7-18 Showing a common configuration of 2 stage air extraction using steam ejectors and condensers. Steam is used to extract air from the condenser and then passed through smaller condensers to recover the steam used in the extraction process

Vacuum pumps

Vacuum pumps are normally rotating positive displacement pumps which go through a cycle where a volume of gas is captured and sealed in a cavity by increasing the volume of the cavity, and then exhausted to the atmosphere (see Figure 7-19). There are several different designs of vacuum pumps such as the Roots blower, Screw pump and Piston pump. To operate in a power plant system, these pumps are normally used in tandem. For example 3 pumps are operated simultaneously during power plant start up to remove the large volume of air while during full operation only one may be used to maintain the vacuum.
The cooling tower

The purpose of the cooling tower is to transfer heat from the cooling water exiting the condenser to the atmosphere. The cooling tower operates by spraying water into or allowing water to fall through a flowing airstream, cooling it in the process. The cooling tower explained in Figure 7-20 is a direct contact heat exchanger. Heat transfer occurs in two ways; direct cooling and direct evaporative cooling. Direct cooling of the water is due to the temperature difference between the water and the moving air. Direct evaporative cooling occurs, when a portion of the water evaporates into vapour and this vaporisation cools the remaining, liquid water. The water is used in three different ways after it has been cooled:

- Injection via the injection well to the geothermal reservoir
- Recirculation of the cooling water to the condenser
- Treatment of minerals via blowdown. Blowdown is an important aspect of the operation of the cooling tower and is used to manage mineral build-up.
Similarly to the condenser the cooling tower process can be further explained in thermodynamic terms with an energy and mass balance, taking into account the heat transfer as well as the mass transfer involved due to evaporation. There must be a conservation of energy and mass (both mass of water and mass of dry air).

Conservation of energy

\[
\dot{m}_c \times h_c + \dot{m}_g \times h_g + \dot{m}_k \times h_k = \dot{m}_j \times h_j + \dot{m}_h \times h_h + \dot{m}_i \times h_i + \dot{m}_f \times h_f \tag{Eq. 7}
\]

Conservation of water

\[
\dot{m}_c + \dot{m}_{\text{water} - g} + \dot{m}_k = \dot{m}_{\text{water} - f} + \dot{m}_j + \dot{m}_h + \dot{m}_i \tag{Eq. 8}
\]

Conservation of dry air

\[
\dot{m}_{\text{dry air} - g} = \dot{m}_{\text{dry air} - f} \tag{Eq. 9}
\]

The cooling tower “consumes” a significant amount of water during operation, between one half and three quarters of the mass of fluid admitted to the turbine is transferred to the atmosphere in the cooling tower (Glassley, 2010). This shows that water must be accounted for as a resource that is used in the system for energy conversion.
Blowdown

A clear explanation of the process of, and necessity for blowdown is given in an example from (Aquaprox, 2009). The process of blowdown is used to manage the concentration of dissolved minerals in the water used in the power plant, because the water used (especially in geothermal power plants) is not pure and contains minerals which must be removed in some way. To quote an example (also represented graphically in Figure 7-21) from Aquaprox [2009]:

“Let us take, as example, a hypothetical cooling system having a water volume of 1,000 m³, and a recirculation rate of 2,000 m³ per hour. Its evaporation rate is of 40 m³ per hour (about 10°C of temperature drop through the cooling tower).

If we suppose that the make-up water contains 400ppm of dissolved solid matters, we have to start with 1,000 m³ containing 400kg of solid matters.

During the first twelve hours of operation, about 480 m³ of water is evaporated, leaving the dissolved solid matters in the cooling system. And during this time, 480 m³ of make-up water was added, which contains an additional 192 kg of dissolved solid matters. The amount of solid matters in the system now reaches 592 kg. The concentration of dissolved salts has increased by 50%, and after 24 hours of the similar operation, there will be about 800ppm of dissolved salts; after 72 hours, there would be 1600 ppm of dissolved salts.

This increase in concentration cannot be maintained, as the amount of (dissolved) salts will inevitably exceed the amount related to their solubility limit, thus causing deposits in the circuit, as well as the risks of corrosion under these deposits.

With the aim of avoiding such a concentration, a part of the water is continually removed from the circuit, in the form of blowdown. A small quantity, less than 0.1% of the recirculation water is carried over into the air.” (Aquaprox, 2009)

INITIALLY

COOLING TOWER

VOLUME=1000m³

Mass of dissolved solids=400kg
Figure 7-21 Illustration showing why blowdown is necessary, water used in cooling towers is not 100% pure and contains dissolved solids, the concentration of which must be managed in the system.

The Aquaprox example shows the fundamental necessity for blowdown in the general operation of a conventional cooling tower. The make-up water contains dissolved solids which means that the concentration of dissolved solids will increase until at some point the cooling system will be saturated with dissolved solids and then the solids will begin to precipitate and cause problems such as scaling and corrosion without some form of mineral management. In a geothermal power plant the importance of blowdown is increased because the geothermal fluid contains higher concentrations of dissolved salts.
8 FLASH POWER PLANTS

Flash steam power plants are the most commonly used geothermal power plant that utilise water-dominated geothermal resources. They require that the geothermal fluid be “flashed” to a steam and water mixture and that the steam be separated from the water. By “flashing” it is meant that the geothermal fluid is taken to a lower pressure by expansion causing it to flash into steam which can be seen from Figure 8-2. The steam is then expanded in a steam turbine where the thermal energy is first converted to mechanical energy in the rotating turbine shaft and then the mechanical energy is converted to electrical energy through an electrical generator. Following the steam turbine the exhaust from the turbine is taken to a condenser where it is condensed to a liquid before reinjection (see Figure 8-1, Figure 8-3). The water that was initially separated may be reinjected or be flashed again as in the case of double-flash (or triple flash) power plants. According to (DiPippo, 2008) single-flash and double-flash power plants account for 32% and 14% of geothermal plants by number respectively and when grouped together flash power plants generate over 65% of worldwide geothermal power capacity. Flash steam power plants have similar structure and components to dry steam power plants, the major differences are that the flash steam power plant utilises a liquid dominated resource above 140 degrees Celsius (°C) and the flash steam power plant requires a steam separator (and provide a way for the liquid to be flashed to steam).

Figure 8-1 Flow diagram describing the basic processes involved in a flash steam power plant
In Figure 8-2, the geothermal fluid starts out as a saturated liquid in the geothermal resource (top of the green line). The green line depicts the flash process where the pressure that the initial liquid is exposed to is lowered, while maintaining the same enthalpy. The result of this isenthalpic pressure drop is that the liquid “flashes” into a mixture of water and steam (bottom of the green line). The mixture is then separated by using a cyclone separator into a saturated liquid (intersection of the blue line and the saturation curve) and a saturated vapour (intersection of the red line and the saturation curve). The saturated vapour (steam) is used in a steam turbine.
Figure 8-3 A T-s diagram showing the characteristics of the geothermal fluid (assumed to be water) as it goes through the flash geothermal power plant.

The processes in Figure 8-3 are described in more detail. From point 1 the compressed liquid in the well is flashed (isenthalpic process) to a liquid vapour mixture at point 2. The liquid vapour mixture at point 2 enters a separator which separates it into saturated steam (point 4) and saturated liquid (point 3). The saturated steam enters a steam turbine (point 4) which facilitates energy conversion from thermal energy eventually into mechanical rotation of the turbine shaft. After exiting the turbine at point 5 the fluid mixture passes through a condenser which condenses the turbine exhaust. At point 6 and point 3 the utilised geothermal liquid is often reinjected into the geothermal reservoir in order for disposal of the liquid and increasing the useful life of the geothermal resource. Point 5s represents the ideal case of turbine expansion, where expansion is isentropic (without losses). Point 7 is the thermodynamic state if the turbine exhaust to the condenser was saturated vapour (at the same condenser temperature and pressure)
8.1 System

The flash steam plant is the major type of geothermal power plant which uses water-dominated or hydrothermal resources. Since energy conversion with two-phase flow is complicated and inefficient, the main focus of the flash steam plant is to address this challenge. A major portion of the flash steam plant is similar to a conventional steam power plant however the differences mentioned above give rise to other, different challenges that will be explained in more detail.

8.2 Processes and Components

8.2.1 Flashing

Hydrothermal resources originate as compressed saturated liquid in the geothermal reservoir and as the liquid moves toward the production well, the pressure falls and at some point it flashes to steam. The actual flashing process can occur in a number of different places relative to the power plant. Flashing may occur naturally in the reservoir, the production well or induced just before entering the separator by the addition of some kind of expansion valve. The calculation of the depth at which flashing occurs is complex and depends mainly on the reservoir conditions, the mass flow of the fluid and also the diameter of piping used in the production well. The depth at which flashing occurs, is given by (Glassley, 2010):

\[
X_f = \frac{X_f}{P_R - P_f - C_D \times m} \left[ (\rho \times g) + (\Gamma \times m^2) \right] \left[ (\rho \times g) + (\Gamma \times m^2) \right] \quad \text{Eq. 10}
\]

Where

\(X_f\) - distance from the top of the reservoir to the depth at which flashing occurs

\(P_R\) - Pressure at the top of the reservoir

\(P_f\) - Pressure that flashing occurs at for the given conditions

\(C_D\) - Draw down coefficient, empirically determined and accounts for the change in pressure at the bottom of the well due to pumping of fluid from the reservoir, given by: \(C_D = (P_R - P_B) / m_v\), where \(P_R\), \(P_B\) (bottom hole pressure) and \(m_v\) are found through testing on site

\(m\) - mass flow rate of the geothermal fluid

\(\rho\) - density of the geothermal fluid

\(g\) - acceleration due to gravity

\(\Gamma\) - factor accounting for how pipe diameter affects the depth at which flashing occurs, it is given by the function, \(\Gamma = (32 \times f) / (\rho \times \pi^2 \times D^2)\), where \(f\) is the friction factor of the pipe
Flow velocity and flash depth

In the Figure 8-4 it can be seen how the flash depth, the depth below the surface where flashing occurs, changes with the parameters of pipe diameter and flow velocity. The explanation of this graph derives from the fact that each geothermal fluid has a specific pressure at which it begins to ‘flash’ (transition from the liquid phase to the vapour phase) depending on its enthalpy. There is a pressure loss as the fluid moves up the pipe from the reservoir and depending on the characteristics of the flow and/or the piping the profile of this pressure loss will be different. For example the trend seen above, where increased flow velocity leads to a deeper flash depth can be explained by using the Bernoulli equation below where a portion of the pressure energy is converted to kinetic energy (dynamic pressure) and causes a pressure drop which results in the fluid reaching the flashing pressure at a deeper depth than if the flow rate had a lower value.
Figure 8-5 Illustration of a geothermal reservoir and a production well pipe being used to extract geothermal fluid, the diagram helps to explain the variation of pressure and velocity with pipe depth.

\[
p_1 + \frac{\rho v_1^2}{2} + g \times z_1 = p_2 + \frac{\rho v_2^2}{2} + g \times z_2
\]

**Eq. 11**

As it can be seen in Figure 8-5 and Eq. 11, there are two points along a pipe from a geothermal reservoir, point 1 at the point at which the reservoir meets the pipe and point 2 above the reservoir with the flow travelling at a speed \(v_2\). The velocity at point 1 is assumed to be negligible since it is at the entrance of the pipe and the pressure \((p_1)\) and elevation \((z_1)\) at point 1 are known values. At point 2 the velocity is increased, and also the elevation is increased since point 2 is vertically above point 1, and in order for there to be conservation of energy, \(p_2\) decreases; the higher the velocity, \(v_2\), the lower the pressure, \(p_2\). In addition to the above explanation, head loss due to friction should also be considered. The head loss due to friction is given by the Darcy equation which is well known in the realm of fluid dynamics where:

\[
h_f = \frac{f \times L \times v^2}{2 \times D \times g}
\]

**Eq. 12**

\(h_f\) - head loss due to friction
\[ f \text{- friction factor depending on the pipe roughness and the Reynolds number which gives an indication about the level of turbulence in the flow (factor obtained from Moody diagram)} \]

\[ L \text{- The length of the pipe} \]

\[ v \text{- average velocity of the flow} \]

\[ D \text{- diameter of the pipe} \]

\[ g \text{- gravitational constant, } 9.81 \, m/s^2 \]

From the above equation it can be seen that the head loss due to friction increases such that it is proportional to the velocity squared (if all other parameters remain constant). Both of the above mechanisms contribute to the fact that pressure loss is greater at higher flow velocity and support the graph obtained.

**Pipe diameter and flash depth**

From the curve it is seen that a smaller pipe diameter results in flashing occurring at a lower depth. Some of the same mechanisms are responsible for the pressure drop in the case of changing the pipe diameter, as the fluid moves toward the surface the pressure decreases as the elevation decreases. There is also a pressure loss due to friction in the piping and the magnitude of the pressure drop is affected by the pipe diameter. The pressure drop is given by the same equation above:

\[
h_f = \frac{f \times L \times v^2}{2 \times D \times g}
\]
Figure 8-6 Moody Diagram (Beck and Collins, 2008)

Figure 8-6 illustrates the Moody diagram, the friction factor is determined from this diagram and depends on the Reynold’s Number for the pipe flow as well as the pipe roughness, which are given by the following equations:

\[
\text{Reynold’s Number} = \frac{\rho v d}{\mu} \quad \text{Eq. 13}
\]

Where
\( \rho \)-fluid density
\( v \)-average flow velocity
\( d \)-pipe diameter
\( \mu \)-dynamic viscosity

\[
\text{Pipe Roughness} = \frac{\varepsilon}{d}
\]

Where
\( \varepsilon \)-The average height of roughness in the pipe
\( d \)-pipe diameter

If the pipe diameter is increased, the Reynold’s number increases and the pipe roughness decreases which yields a lower friction factor in the Moody diagram. In addition, frictional
pressure loss is decreased since it is given by $h_f = \frac{f \times L \times v^2}{2 \times D \times g}$; if the length and velocity remain constant the pressure loss due to friction decreases with increased pipe diameter, and vice versa. If it is desired for flashing to occur at a shallower depth (closer to the surface) it is possible to increase the pipe diameter to achieve this.

Two-phase flow

After the flash process has begun there will be a two-phase flow mixture of steam and water. Two phase flow is a complex phenomenon. The flow has different “regimes”, or defining characteristics, depending on the pressure and the ratio of steam to water. Also because the flow characteristics are physically different, the mechanisms behind pressure drop are also different. Understanding of two-phase flow is highly empirical; however the different changes the two-phase flow undergoes can be illustrated (see Figure 8-7). The two-phase flow transitions through the following regimes, in the following order:

- Bubbly flow (flashing begins with this flow regime)
- Slug flow
- Annular flow
- Annular flow with entrainment
- Drop flow
Figure 8-7 Flow patterns observed under normal gravity two phase flows in a vertical pipe. Gravity direction is downwards (Balasubramaniam et al., 2006)
8.2.2 Separator

The separation process in geothermal flash steam power plant typically occurs in a cyclone separator. A cyclone separator utilises the large difference in density between steam and water to effectively separate the two phases of the fluid. The design of the cyclone separator is such that the two phase mixture enters the cylinder in a radial direction, centrifugal forces allow entrained liquid to be deposited on the walls of the separator and flow down to where the liquid is collected and then removed from the separator. The separated steam flows through a pipe with a high entry point where it is extracted in a downward direction for expansion in the steam turbine (Figure 8-8).

![Separator Illustration](image)

*Figure 8-8 Illustration showing a cross-section of a steam separator, two phase fluid enters the separator tangentially and steam rises and is extracted from the pipe at the top of the separator, while separated liquid exits the separator at the bottom of the separator.*

Separator design

There are general guidelines for the design of steam separators and moisture removers used in geothermal power plants. The geothermal industry has settled on the Webre-type separator (DiPippo, 2008) depicted in Figure 8-8. According to (Lazalde-Crabtree, 1984) cyclone steam separators and moisture removers should have the specifications outlined in the chapter in Appendix Table 19, Figure 13-11 and Figure 13-10 for efficient separation.

Separator operation

There are several issues that must be considered during separation.

- The temperature and pressure at which the separator should operate for optimal operation of the power plant.
- Depending on the conditions selected there will be a different pressure, temperature, mass flow and quality of steam, which affects the potential power available.

- There are two kinds of flow which can emerge from the well head:
  - choked
  - non-choked flow.

The flow is characterised by how the mass flow changes with the pressure of the well-head.

![Figure 8-9 Showing the shape of a choked mass flow profile on the left compared with the shape of a non-choked mass flow profile on the right (DiPippo, 2008)](image)

**Choked flow**

In a choked flow system as the wellhead pressure is lowered the mass flow of geothermal fluid rapidly increases to a certain value where it tapers off and any further decrease in the pressure does not yield significant increase in the mass flow rate, hence the name choked flow (see Figure 8-9). Typically production wells that have a choked flow profile are said to be more productive wells as they yield high mass flows when compared to non-choked flow systems at the same pressure. The descriptions of flow presented here are the flows from the production well, and that after the flow is utilised in the system and condensed, it is reinjected to the geothermal reservoir underground where it is reheated and eventually re-emerges at the production well (see Appendix 13.2.5 Reinjection)
Non-choked flow

As opposed to the choked flow, non-choked flow occurs when the decrease in well head pressure leads to maximum mass flow only at very low pressures. Non-choked flow has a linear increase of mass flow with decreased wellhead pressure (see Figure 8-9).

Optimisation

The graphs in Figure 8-9 describe how the mass flow changes with the wellhead pressure depending on the conditions of the geothermal resource. (DiPippo, 2008) uses a method where the specific power output possible at different pressure and temperature levels in the steam separator is calculated using steam tables. Figure 8-9 is prepared using data for a hydrothermal (liquid-dominated) geothermal resource with a temperature of 240°C. During the process several simplifying assumptions have been made to arrive at an optimisation starting point;

- Flashing is an isenthalpic process
- Pressure loss between the wellhead and the separator is negligible
- The condenser operates at 0.123 bar or condenses steam at 50°C.
- The steam turbine being considered is isentropic in nature, this assumption avoids the need to calculate the turbine efficiency before arriving at an optimisation starting point since the steam is expanded into the mixed liquid-vapour region of the saturation curve.

The optimisation process makes a compromise. When a high temperature is selected, the amount of steam obtained by flashing is low, even though the enthalpy drop across the turbine is large. For a medium separator temperature the proportion of steam obtained through flashing versus the amount of liquid initially extracted is moderate and also the enthalpy drop is moderate. When a low separator temperature is selected the proportion of flashed steam is relatively large, however the enthalpy change across the turbine is low (see Figure 8-10). Figure 8-10 shows how the power cycle will look depending on what conditions are selected in the separator and has the same shape as Figure 8-3, however the quality of steam in the separator and enthalpy change across the turbine vary with high, medium and low separator temperatures.
Figure 8-10 T-s diagram showing the characteristics of the power cycle depending on the temperature selected for the separator.

By plotting the specific work output at different separator temperatures (or pressures) and multiplying the results by the mass flow associated with each temperature or pressure the point of maximum total power output can be found and give an idea of the optimal power output possible from the power plant. A derivation of the specific work output in a geothermal flash steam plant is done for reference in the appendix section 13.1.1 on page 115.

Optimal choked flow operating conditions in the separator

Figure 8-11 tells us the value of the separator pressure that yields the highest specific work as well as the fact that the mass flow remains constant in the region of maximum specific work. Since the mass flow does not change in the maximum specific mass flow region, the point of maximum specific work is the same as the point of maximum total power and corresponds to a separator pressure of around 3.6 bar or a separator temperature of 140°C (see appendix Figure 13-1 on page 114). Furthermore, information about the performance of the plant at different operating conditions is documented in Table 12 on page 114 in the appendix.
Optimal non-choked flow operating conditions in the separator

Figure 8-13 shows how the mass flow and specific work output change with separator pressure. The flow is not choked and the mass flow continues to increase with decreasing separator pressure. This difference when compared to choked flow creates a profile for the total work output where the point of maximum total power output is different from the point of maximum specific work (see Figure 8-14 and Table 13 in the appendix). The point of
maximum specific work corresponds to a separator pressure of 3.6 bar, a and separator temperature of 140°C. However, the point of maximum total work output occurs at a separator pressure of 2.7 bar, a and separator temperature of 130°C. The difference between the point of maximum specific work and maximum total work shows the effect which a non-choked mass flow profile has on power output.

**Figure 8-13** Graph showing the variation of mass flow and specific work with separator pressure conditions in a non-choked flow system
Figure 8-14 Variation of Total power output and Specific power output with separator temperature in a non-choked flow system

There is a rule of thumb about the optimisation of a single flash plant in choked geothermal wells, where the optimal separator temperature is approximately half-way between the reservoir temperature and the condenser temperature. In the case titled “Optimisation” on page 76 the reservoir temperature was 240°C, and the condenser temperature was 50°C, using the rule of thumb the approximation would be that the optimal separator temperature would be close to 145°C. The actual optimal separator temperature for choked flow was in fact 140°C, suggesting that the rule of thumb holds some validity.

8.3 Double Flash power plants

Double flash geothermal power plants utilise the separated geothermal fluid for a second flash and separation process. The advantage of the double flash geothermal power plant is that it uses a larger portion of the resource extracted for power generation when compared to single flash geothermal power plants which translates to a greater efficiency.
Double flash power plants use the same technology as the single flash power plants, however the efficiency is increased since the initially separated water is utilised by flashing to a lower pressure (see Figure 8-15).

Figure 8-15 Illustration of the major components involved in the double flash geothermal power plant
Figure 8-16 A T-s diagram showing the characteristics of a double flash geothermal power plant,
The process begins in a similar way to the single flash geothermal power plant. Referring to Figure 8-16, compressed geothermal liquid (point 1) is flashed (point 2) and then separated into saturated steam (point 4) and saturated water (point 3). The saturated steam passes through a high pressure steam turbine, while the saturated liquid at point 3 is flashed a second time (point 6) and then separated into saturated water (point 7) and saturated steam (point 8). From this point the saturated steam at point 8 as well as the high pressure turbine exhaust at point 5 are combined (point 9) and pass through the low pressure turbine, with the turbine exhaust being point 10. The low pressure turbine exhaust is condensed in the condenser and saturated water at points 7 and 11 are removed, most times for reinjection to the geothermal reservoir.

Even though the same technology is used, additional equipment is required in the power plant, in order to carry out the second flash process. It is also more difficult to determine which temperatures and pressures should be used in the steam separators for maximum efficiency. A rule of thumb exists where the temperatures of the separators should be equally spaced between each other and the temperature of the fluid in the reservoir and the temperature in the condenser.

8.3.1 Comparison between Single Flash and Double Flash specific work

The increase in efficiency associated with the use of a double flash power plant related to single flash power plants is estimated at between 20 and 30%. If the cost of additional components and equipment is justified then the double flash power plant is recommended since the efficiency is increased. By using the example given in (13.1.1 Derivation of specific work associated with a steam separator conditions on page 115) and using the same initial conditions of geothermal liquid, a calculation is made in section 13.1.2 titled “Sample calculation of specific power output from a double flash power plant with a reservoir condition of 240°C saturated liquid” on page 120 for a double flash power plant and the maximum efficiency of energy conversion is compared to that of the single flash power plant.

The maximum specific work obtained in the single flash power plant section for a saturated liquid resource at 240°C is 85.86 kJ/kg, while the maximum specific work obtained from a double flash plant while utilising the same resource is 110.48 kJ/kg. This represents a
28.7% increase in specific power output for the double flash plant when compared to the single flash power plant

8.3.2 Components

Most components and processes used in the flash steam power plant are no different from those used in the dry steam power plant. The steam turbine, condenser, cooling tower and reinjection are all similar in both power plants and therefore the information will not be repeated. The reader is referred to section 7.2 ‘Processes and components’ starting on page 42, which discusses the components in dry steam power plants. Reinjection is covered in section 13.2.5 ‘Reinjection’ on page 130.
9 BINARY POWER PLANTS

Geothermal binary cycle power plants utilize geothermal fluid as an external heat source to power a thermodynamic cycle which uses a separate working fluid. Both the geothermal fluid and energy conversion system are in their own closed loop with a heat exchanger between them (see Figure 9-1). The closed circuit of geothermal fluid is represented as the loop including points A and B in Figure 9-1, where geothermal fluid is extracted from the production well at point A and passes through a heat exchanger before being reinjected at point B. The closed circuit for working fluid in the power generation cycle is represented by points 1-4 in Figure 9-1, where between point 1 and 2 the working fluid passes through the heat exchanger which acts as an evaporator for the working fluid. Between points 2 and 3 the working fluid is expanded in a turbine before passing through a condenser and (3-4) and a pump (4-1). After going through the pump the working fluid completes the cycle and begins another by entering the heat exchanger. Binary cycle plants are generally suited for lower geothermal resource temperatures and cases where the geothermal brine is difficult to handle due to high corrosiveness.

Figure 9-1 Flow diagram describing the basic processes involved in a binary cycle power plant.
There are several types of binary cycles with the two most popular being the organic Rankine cycle and the Kalina cycle. The organic Rankine cycle is very similar to the conventional steam Rankine cycle, however an organic fluid is used as the working fluid. The Kalina cycle is a thermodynamic cycle that uses a water ammonia mixture as a working fluid to more closely match the temperature profile of the heat source which is beneficial to performance. Cooling may present a challenge to binary cycle power plants since a cooling system with a separate fluid is required, for example using a condenser cooled by fresh water or air. Binary cycles are the most flexible type of geothermal power plant as they depend on the properties of the working fluid selected; they can be used for energy conversion of very low temperature resources, as low as 57°C. (Erkan et al., 2008). They can be used as bottoming cycles for traditional flash power plants, hybrid power plants and have found use in industrial waste heat applications. Binary cycle power plants constitute a big share of the number of geothermal power plants in operation worldwide, 32%; however generate only 4% of the global generation capacity (DiPippo, 2008). This data implies that individual plants have a relatively small capacity.

### 9.1 Organic Rankine Cycle

The Organic Rankine Cycle (ORC) is almost identical to the Rankine cycle, except it uses an organic working fluid instead of water in a Rankine cycle; it is a closed cycle where an organic working fluid is taken through a closed cycle where the working fluid is heated in an evaporator/heat exchanger then expanded in a turbine and then cooled and condensed before being pressurised and returned to the evaporator. Organic Rankine cycle is used in waste heat power plants and has been discussed as a topic in the CompEdu platform (see section S1-B5-C5) where the principle of operation, advantages/disadvantages as well as applications are explained. Regarding the use of the ORC in a geothermal power plant, the major considerations are the working fluid to use as well as the heat exchange between the geothermal source and the working fluid and also the heat exchange involved in the cooling process of the working fluid. Seeing that the general topic of the ORC has already been outlined in CompEdu, its specific considerations in relation to geothermal power plants are explained. The Organic Rankine cycle utilises liquid-dominated resources normally below 140 degrees Celsius. When utilising this resource it is not favourable to use water because the boiling point is close to the resource temperature which may result in complications in heat exchange (such as incomplete boiling). An organic fluid is used because it has a lower boiling point and more favourable heat transfer characteristics than water for these low temperatures.
9.1.1 Optimisation and selection of working fluid

![T-s diagram](image)

Figure 9-2 T-s diagram of an organic Rankine cycle, showing the power cycle as well as the thermal characteristics of the geothermal water and cooling water.

Where in Figure 9-2:
- Point 1 – Entry point to turbine/Evaporator exit
- Point 2 – Entry to the condenser/turbine exit
- Point 3 – Entry to the working fluid pump/Condenser exit
- Point 4 – Entry to the evaporator/Pump exit
- \( Q_E \) – Heat transferred from the geothermal water to the working fluid in the evaporator
- \( Q_C \) – Heat transferred from the working fluid to the cooling water in the condenser
- \( T_{HWI} \) – Temperature of the hot water inlet (geothermal water inlet temperature)
- \( T_{CW1} \) – Temperature of the cold water inlet (cooling water inlet temperature)

Selection of the working fluid is an important decision because correctly matching the ORC working fluid to the geothermal heat source is significant in determining the efficiency and operating characteristics of the power plant. The boiling point, heat transfer coefficient, heat of vaporisation play a part in the pressure required in the evaporator as well as the areas required for heat transfer in the condenser and evaporator. A high pressure requirement in the evaporator will increase the pumping power required, which is a parasitic load of the
power cycle. Having a poor heat transfer coefficient increases the area required for heat transfer which increases the cost and complexity of the heat exchanger. A good method of illustrating the challenges faced in this stage is through an example from research. In the following example adopted from (Madhawa Hettiarchchi et al., 2007) the selection of working fluid and optimisation process are outlined.

9.1.2 Fluid selection and optimisation example

Before the actual process of fluid selection and power plant optimisation can begin, it must be decided what is the most suitable property for optimisation. It is helpful to look at the nature of the power plant and the resources it utilises. Typically the organic Rankine cycle (and by extension binary power plants) is used to utilise geothermal resources at medium-low temperatures, from about 150°C to temperatures at 70°C and lower. The low temperature resource utilisation means that for conversion in a thermal power plant practical efficiencies are limited to below 20% in accordance with the Carnot Efficiency. For such low temperature utilisation another factor that arises is the heat exchanger area required to effectively use the heat provided by the geothermal resource. A practical property in the case of a power plant is the cost of the power plant in relation to how much power is generated. Since in practice, the cost of a power plant depends on when it is constructed as well as where and who is constructing it, a more suitable assumption is that a great deal of the cost is derived from the amount of heat exchanger area required. Therefore the property to be optimised is chosen as the ratio between the heat exchanger area and the power generated; in the ideal situation the smallest heat exchanger area required to generate a unit of power is the optimal choice since heat exchanger area is related to the cost of the power plant.

Working fluid selection

For the example the geothermal resource has a temperature of 90°C and the cooling water temperature is 30°C, based on these assumptions as well as how well the properties of the working fluid match these temperatures in the T-s diagram (see Figure 9-2) the following four working fluids were selected (PF 5050, HCFC 123, Ammonia, n-Pentane). A brief explanation of the desirable working fluid properties is presented:

- Small specific volume and low specific heat
A small specific volume means that the fluid is dense enough for machinery used in the power plant to be relatively small and efficient, while a low specific heat means that a low amount of heat is required to raise the fluid temperature (advantageous to heat transfer)

- Dry or Isentropic working fluid
  A dry working fluid suggests that after expansion in the turbine the working fluid will be superheated (no liquid present); the saturated vapour line has a positive gradient. For a wet working fluid such as water expansion occurs into the mixed liquid-vapour zone; the gradient of the saturated vapour line is negative. Isentropic working fluids have an indefinite gradient (ideally between wet and dry working fluids. If the working fluid is dry or isentropic, the expansion in the turbine is simplified since there is no liquid present which can damage turbines and degrade performance.

- High Thermal conductivity
  A high thermal conductivity decreases the required area for heat exchange.

- High auto-ignition temperature and suitable thermal stability
  Thermal stability and a high auto-ignition temperature ensures that the power plant operates safely.

- Reasonable cost
  In order to improve the economics of the power plant, ideally the cost of the working fluid should not be excessive unless the additional cost is justified by the performance increase

- Non-corrosive and non-toxic
  For the plant to operate safely and have low maintenance costs the working fluid should not be corrosive to power plant equipment and not toxic in order for it to be relatively safe in the event of a leak.

**Optimisation Process example**

The optimisation process begins with assumptions and reasonable values for the performance of components in the cycle and a few operating conditions.

**Initial Assumptions**

<table>
<thead>
<tr>
<th>Working fluids</th>
<th>Ammonia, n-Pentane, HCFC-123, PF5050</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross power $W_{\text{Gross}}$</td>
<td>10 MW_e</td>
</tr>
<tr>
<td>Condenser and Evaporator</td>
<td>Shell and plate type</td>
</tr>
</tbody>
</table>
Heat transfer plate material is titanium
\( l = 1460 \text{ mm}, w = 550 \text{ mm}, t = 0.6 \text{ mm}, \delta x = 5 \text{ mm}, \delta y = 5 \text{ mm} \)

<table>
<thead>
<tr>
<th></th>
<th>Heat transfer plate material is titanium</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal water temperature, ( T_{HWI} )</td>
<td>90°C</td>
</tr>
<tr>
<td>Geothermal water and cooling water pump efficiency, ( \eta_{WP} )</td>
<td>0.80</td>
</tr>
<tr>
<td>Working fluid pump efficiency, ( \eta_{WFP} )</td>
<td>0.75</td>
</tr>
<tr>
<td>Cooling water temperature, ( T_{CW1} )</td>
<td>30°C</td>
</tr>
<tr>
<td>Turbine efficiency, ( \eta_T )</td>
<td>0.85</td>
</tr>
<tr>
<td>Generator efficiency, ( \eta_G )</td>
<td>0.96</td>
</tr>
</tbody>
</table>

Another assumption is that this organic Rankine cycle is a basic one which follows the shape of that in Figure 9-2, meaning that there is no reheat, regeneration or superheat in the evaporator.

- Optimisation process (see Figure 9-3 for reference)

  After the initial plant conditions and assumptions, the temperatures of the evaporator and condensers are assumed in addition to the velocity of the cooling water flow as well as the velocity of the geothermal fluid.

From the temperatures and velocities, the state points can be plotted on the T-s diagram.

After the state points are plotted, the mass flow of the working fluid, heat transfer across the evaporator and condenser as well as the Rankine efficiency can be calculated.

The optimisation now moves into more detail in the condenser and evaporator, where the temperatures of the plate of the heat exchanger, the working fluid and the water are assumed and the heat transfer characteristics are determined based on the type and model of heat exchanger selected as well as the characteristics of the working fluid. The heat transferred to the working fluid, the water and across the plate are calculated, changing the values of temperatures of the plate, working fluid and water until there is a convergence such that the heat transferred across the plate is equal to the heat transferred by the working fluid and the heat transferred to the water.
After there is convergence in the heat transfer characteristics, the overall heat transfer coefficient, required heat transfer area, Power output, mass flow of water as well as required pump work can all be calculated.

Finally the optimisation criteria (heat transfer area per unit of power generated can be calculated, giving an indication of how well suited a working fluid is for the chosen geothermal fluid characteristics.

Figure 9-3 Diagram illustrating the optimisation process in a organic Rankine cycle
**Where in Figure 9-3**

\( T_E \) – Temperature of the evaporator
\( T_C \) – Temperature of the condenser
\( V_{CWI} \) – Velocity of the cold water inlet
\( V_{HWI} \) – Velocity of the hot water inlet

State points 1,2,3,4 – Points in the power cycle in accordance with Figure 9-2

\( m_{WF} \) – mass flow rate of the working fluid
\( Q_E \) – Heat transferred across the evaporator
\( Q_C \) – Heat transferred across the condenser
\( \eta_R \) – Rankine cycle efficiency
\( T_P \) – Temperature of the plate in the heat exchanger
\( T_{WF} \) – Temperature of the working fluid
\( T_W \) – Temperature of the cooling water
\( \alpha_{WF} \) – Working fluid heat transfer coefficient
\( Q_{WF} \) – Heat transferred by the working fluid
\( \alpha_W \) – Cooling water heat transfer coefficient
\( Q_W \) – Heat transferred to cooling water
\( Q_{AP} \) – Heat transferred across heat exchanger plate
\( U \) – Overall heat transfer coefficient
\( A_T \) – Total area required for heat transfer
\( W_N \) – Net Work output from the cycle
\( P_{WW} \) – Cooling water pump work
\( m_W \) – mass flow rate of cooling water
\( \zeta \) – The ratio between the heat exchanger area and the power generated, this is the property that is optimised in this example

Using the properties of the working fluids and the system model, the organic Rankine cycle parameters are numerically calculated (shown in Table 8). The objective of the calculation was to optimise the ratio between the heat exchanger area required and the net power generated, which means that the values in the table represent the system parameters when the ratio of heat transfer area to net power generated are lowest. Table 8 highlights some very important points, for example it is shown that the cycle using Ammonia as the working fluid has only the 3rd highest Rankine efficiency, however utilises the smallest area for heat exchange (for more detailed information about the other calculated parameters see Table 18 on page 125 in the appendix). If the ratio of heat transfer area to net power generation is
the property to be optimised then ammonia would be the most suitable choice for a working fluid using these geothermal power plant conditions. The property for the ratio of heat exchange area to net power generated, $\zeta$, is important and since heat exchanger area gives an indication of the cost and complexity of the power plant, when the values are compared it can be seen that it Ammonia requires half the area for heat transfer per unit of power generated ($\zeta = 0.35$) when compared to HCFC 123 ($\zeta = 0.70$) and less than half for PF5050 ($\zeta = 1.26$) and n-Pentane ($\zeta = 0.89$).
Table 8 Numerical calculation of selected parameters of an organic Rankine cycle for a geothermal resource with a temperature of 90 °C and a gross power output of 10MWₑ, showing the characteristics of the cycle for four different working fluids (Madhawa Hettiarachchi et al., 2007)

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>PF5050</th>
<th>HCFC 123</th>
<th>Ammonia</th>
<th>n-Pentane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross Power W_{Gross} (MWₑ)</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Geothermal water outlet temperature (°C)</td>
<td>85.7</td>
<td>85.0</td>
<td>81.45</td>
<td>86.0</td>
</tr>
<tr>
<td>Geothermal water flow rate (kg/s)</td>
<td>8496</td>
<td>5972</td>
<td>3693</td>
<td>7236</td>
</tr>
<tr>
<td>Cooling water flow rate (kg/s)</td>
<td>6598</td>
<td>4864</td>
<td>3898</td>
<td>4640</td>
</tr>
<tr>
<td>Working fluid flow rate (kg/s)</td>
<td>1243</td>
<td>669</td>
<td>494</td>
<td>294</td>
</tr>
<tr>
<td>Evaporator area Aₑ (m²)</td>
<td>7059</td>
<td>3766</td>
<td>1364</td>
<td>5660</td>
</tr>
<tr>
<td>Condenser area Aᶜ (m²)</td>
<td>3961</td>
<td>2773</td>
<td>1377</td>
<td>2317</td>
</tr>
<tr>
<td>Ratio of Heat exchanger area to net power generation ζ (m²/kW)</td>
<td>1.26</td>
<td>0.70</td>
<td>0.35</td>
<td>0.89</td>
</tr>
<tr>
<td>Rankine cycle efficiency ηₑ</td>
<td>7.8</td>
<td>9.8</td>
<td>8.9</td>
<td>9.9</td>
</tr>
</tbody>
</table>

This example shows that the process of determining which properties are most important to the performance and operation of the organic Rankine cycle is a long one and is specific to the geothermal resource temperature as well as the types of working fluids and equipment used in the cycle. In order to arrive at a suitable conclusion about optimisation, the optimisation criteria must be clear. Accurate models of the system and its components must be used in addition to correct data about the characteristics of the geothermal heat source and the cooling system. This example represents a simple organic Rankine cycle with a geothermal resource at a temperature of 90°C, no superheat or reheat or regeneration, plate heat exchangers and a cooling water system. However for different cases this model would need to be adjusted in order to optimise the system, in other cases there may be a different geothermal source temperature, a different type of cooling system or heat exchanger available and also the cycle may use regeneration or superheat. With this in mind, the point of this example was to highlight the approach to optimisation, not the specific details since for other ORC’s the specific steps in the process of optimisation may be very different.
Cooling of the Organic Rankine Cycle

The ORC, as stated before, is most suitable for use of geothermal resources that are at a temperature too low for feasible utilisation in a flash steam power plant. The ORC utilises resources at a relatively low temperature, with low efficiency and also separates the geothermal fluid from the power plant. From the point of view of the cooling and heat exchange the properties of the ORC (low efficiency, separate power cycle circuit) make it necessary to transfer a high amount of heat to the environment in the ORC. Unlike in the dry-steam and flash power plants, the ORC does not utilise the geothermal resource as a working fluid in the cycle and this means that steam condensate cannot be used as make-up water in the cooling system. Several alternatives for cooling exist, however considering that there is low efficiency and no steam condensate available for utilisation the cost and complexity of the required cooling system will be increased. The major types of cooling systems are to use a wet cooling tower or air cooling.

- **Wet cooling tower**
  The wet cooling tower has been described in (The cooling tower on page 60) however the use of this type of cooling system depends on the availability of cooling water at the site of the power plant as well as the economics of the power plant

- **Air cooling**
  The air cooling method uses ambient air to transfer heat from the cycle to the environment. The disadvantage of using an air cooling heat exchanger is the parasitic power required to drive cooling fans as well as the fact that variation in ambient temperature of the environment throughout the year can significantly affect the power output and efficiency of the plant in a negative way. The advantage of using air cooling is that it can be used in areas where water cooling is not feasible (low availability/high cost). Air cooling requires a very large volume of air for cooling when compared with the water requirement in water cooling system. This difference in required volume is derived mainly from the large difference in density and heat transfer characteristics between water and air. In Table 9 the properties of density, specific heat capacity and conductivity for water and air are compared at the temperature of 50°C and the pressure of 100 kPa.
Table 9 Comparison of the density and heat transfer characteristics of water and air at a temperature and pressure of 50°C and 100 kPa

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Density (kg/m³)</th>
<th>Specific Heat capacity (kJ/kg-K)</th>
<th>Conductivity (W/m-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>1.078</td>
<td>1.01</td>
<td>0.027</td>
</tr>
<tr>
<td>Water</td>
<td>988</td>
<td>4.18</td>
<td>0.63</td>
</tr>
</tbody>
</table>

This difference in characteristics (particularly density where the density of water differs from that of air by an order of three) means that the volume of air required for cooling is greatly increased. The fact that air used for cooling is in a gaseous phase while cooling water is in the liquid phase means that instead of pumps, fans must be used.
10 ADVANCED AND FUTURE ASPECTS

Currently there are three main types of power plants for utilising geothermal resources (which were previously described), however with technological advancements as well as concerns about energy and the environment, there is an abundance of research into innovative and more efficient ways of utilising geothermal resources and a few of these interesting concepts will be introduced in this chapter.

10.1 Using Supercritical Resources

Currently, commercial geothermal resources in use are hydrothermal and either exist as steam, water or a mixture of the two with all the resources existing in the sub-critical state. This innovative idea seeks to utilise supercritical geothermal fluid to act as the geothermal resource. The concept is heavily in the research stage which is led by the Iceland Deep Drilling Project (IDDP). This project has several research interests, not just for geothermal power generation. Although it is in the research phase many potential challenges and also possible advantages have already been identified.

10.1.1 Challenges

Supercritical geothermal fluids have not been used for practical purposes and therefore require research into handling and utilising these fluids. Supercritical geothermal fluid exist under extreme temperatures (450-600°C) and pressures (<200 bar, a) they also contain dissolved volcanic gases and induce phenomenon such as thermal creep due to the high temperature. Special drilling techniques, materials and safety measures must be employed to ensure that it is indeed feasible and safe to harness power from supercritical geothermal fluids(Stefansson et al., 2008). Although the characteristics of the fluid will ultimately determine the most suitable type of energy conversion technology it is suggested that if supercritical geothermal fluid is used as a geothermal resource it would be used in an indirect, binary cycle where the heat from the supercritical geothermal fluid is transferred to the power generation circuit; the hot fluid will be used as just the heat source and not the working fluid (unlike in flash or dry steam plants). The main reason for this suggestion is that the conditions of geothermal supercritical fluid would prove harmful to equipment in the cycle and also require, for example, treatment for dissolved gases in the condenser.
10.1.2 Possible Advantages

A comparison between a potential power cycle using supercritical geothermal fluid and conventional geothermal flash cycle was done by (Albertsson et al., 2003) and the potential of supercritical geothermal fluid utilisation is very encouraging. From those results which will be explained further below, for the same volumetric flow, the conventional flash power plant had a turbine output of 4.98 MW\(_e\) while the supercritical geothermal fluid power plant turbine output was 49.07 MW\(_e\). The two plants used in the comparison will be briefly described and represented in figures. Firstly, the conventional plant being used is most similar to a dry steam power plant where steam emerges from the production well with a volumetric flow rate of 0.67 m\(^3\)/s. The steam is then passed through a steam turbine and then condensed and re-injected, the resulting power output was 4.98MW (see Figure 10-1 for representation of this power plant). The power plant utilizing supercritical geothermal fluid uses a binary cycle where supercritical geothermal fluid is passed through a heat exchanger where its heat is transferred to a separate circuit with water. The geothermal fluid has the same volumetric flow rate as in the conventional dry steam power plant case, 0.67 m\(^3\)/s. The water that is used in the power cycle is heated in the heat exchanger to a superheated state, expanded in the steam turbine and then it is condensed, cooled and pumped, before it re-enters the heat exchanger to complete the cycle (Figure 10-2).
Figure 10-1 Conventional power plant adopted from feasibility study investigating the potential of a geothermal power plant utilising supercritical geothermal fluid (Albertsson et al., 2003).

Point 1
\[ m_1 = 10 \, kg/s, \quad T_1 = 235^\circ C, \quad P_1 = 30 \, bar, \quad h_1 = 2802 \, kJ/kg, \quad Q = 0.67 \, m^3/s \]

Point 2
\[ m_2 = 10 \, kg/s, \quad T_2 = 183.2^\circ C, \quad P_2 = 7.7 \, bar, \quad h_2 = 2802 \, kJ/kg, \quad x_2 = 100\% \]

Point 3
\[ \eta_{total} = 0.75, \quad \Delta h_{isentropic} = 663.4 \, kJ/kg \quad Power = 4.98 \, MW \]

Point 4
\[ P_4 = 0.1 \, bar, \quad x_4 = 87\% \]

Point 5 (Cooling Water)
\[ T_5 = 42^\circ C, \quad m_5 = 254 \, kg/s, \]

Point 6 (from Cooling Tower)
\[ Power_{PUMP} = 42.5 \, kW, \quad T_6 = 22^\circ C \]

Point 7
\[ Power_{PUMP} = 13.3 \, kW \]

Points 8 and 9 do not have great significance in this study
Figure 10-2 Proposed geothermal power plant utilising supercritical geothermal fluid adopted from the feasibility study investigating the potential of a geothermal power plant utilising supercritical geothermal fluid (Albertsson et al., 2003)

Point 1
\[ m_1 = 55.16 \text{ kg/s}, \quad T_1 = 503^\circ\text{C}, \quad P_1 = 195 \text{ bar}_a, \quad h_1 = 3260 \text{ kJ/kg}, \quad Q = 0.67 \text{ m}^3/\text{s} \]

Point 2
\[ m_2 = 53.21 \text{ kg/s}, T_2 = 495^\circ\text{C}, P_2 = 170 \text{ bar}_a \]

Point 3
\[ \eta_{total} = 0.70, \quad \text{Power} = 49.07 \text{ MW} \]

Point 4
\[ P_4 = 0.08 \text{ bar}_a \quad x_4 = 88\%, \quad T_4 = 42^\circ\text{C} \]

Point 5 (Cooling Water)
\[ T_5 = 38^\circ\text{C} \]

Point 6 (from Cooling Tower)
\[ \text{Power}_{PUMP} = 249 \text{ kW} \]
\[ T_6 = 22^\circ\text{C}, \quad m_5 = 1740 \text{ kg/s}, \quad \text{Power}_{PUMP} = 1301 \text{ kW} \]

Point 8
\[ T_8 = 39^\circ\text{C} \]

Point 9
\[ T_9 = 62^\circ\text{C} \]
There are different benefits that a power cycle utilising supercritical resources would have when compared to conventional geothermal technology. The most evident advantages can be seen when looking at the main differences in the thermodynamics of the processes. These will be listed:

- **Enthalpy of resources**
  - The supercritical geothermal fluid exists at a higher temperature and pressure when compared to the conventional dry steam power plant and for this reason has a higher enthalpy.
  - Supercritical geothermal fluid at the production well: \( T=503^\circ C, P=195 \text{ bar,a} \) and \( h=3260 \text{ kJ/kg} \)
  - Conventional geothermal fluid at the production well: \( T=235^\circ C, P=30 \text{ bar,a} \) and \( h=2802 \text{ kJ/kg} \)

- **Mass flow**
  - Supercritical resources exist under extreme pressures and the high pressure greatly increases the density of the fluid, allowing a greater mass to be contained in the same volume when compared to the subcritical steam resource used in the dry steam plant.
  - For a volume flow rate of \( 0.67 \text{ m}^3/\text{s} \) in the supercritical power plant the corresponding mass flow is \( 55.16 \text{ kg/s} \)
  - For a volume flow rate of \( 0.67 \text{ m}^3/\text{s} \) in the conventional dry steam power plant the corresponding mass flow is \( 10 \text{ kg/s} \)

From the two major differences above the main advantages of the supercritical power plant are yielding a higher efficiency by utilising higher temperature resources and also improving power plant economics by having to dig fewer wells to get the same mass flow as before.

### 10.2 Total flow systems

The conventional flash steam power plant is the most common geothermal power plant in use today, unfortunately the flash steam power plant utilises only a fraction of the geothermal fluid mass flow (normally between 20 and 30\% for single flash power plants). The portion of the geothermal fluid that is not utilised for power generation in the conventional single flash power plant is the separated liquid. A total flow geothermal power plant performs energy conversion by utilising the saturated geothermal liquid and expanding it into the two phase flow zone (see Figure 10-3).
This expansion is a challenge because the two phase flow is complex, fast-moving entrained droplets create a hostile environment where a high rate of erosion (due to impingement) is common. This challenge is evident in current axial steam turbines which expand steam deep into the two-phase zone (greater than 10% liquid) where entrained water droplets erode the turbine blades. If this challenge to total flow systems can be overcome then there will be several opportunities to increase efficiency and utilisation of geothermal resources in several ways.

10.2.1 Developments
The most promising type of technology for two phase flow energy conversion is a pure reaction, radial outflow style turbine, which uses nozzles and the kinetic energy of the flow to drive a turbine, similar to an Aeolipile or Hero Engine (see Figure 13-4 in the appendix on page 126).
Radial Outflow Reaction Turbine

Research at the Lawrence Livermore Laboratory in the 1970’s achieved an isentropic efficiency of 32.7% (House, 1978b) using a “Radial Outflow Reaction Turbine” (RORT); the basic mechanical design of this device is illustrated below in Figure 10-4. The nozzles function to drive the turbine by using the kinetic energy of the two phase flow.

Since then there have been further advances with adaptations of the radial outflow reaction turbine the more successful ones are called the Velocity Pump Reaction Turbine (VPRT) and a machine developed by Dr. Gracio Fabris under the Energy Innovations Small Grant (EISG) Program called the two phase flow turbine.

Velocity Pump Reaction Turbine

The velocity pump reaction turbine operates by adopting the same basic principle as the radial outflow reaction turbine, however there is an internal rotor which accelerates and changes the direction of the flow. This additional rotor enhances the efficiency of the turbine at harnessing the energy of the pressurised liquid. The inner rotor requires an external power source to be driven and changes the direction and speed of the flow and also increases the pressure such that the fluid remains as a single phase liquid. After the inner rotor there is a gap and the outer rotor operates in the same way as a radial outflow reaction turbine. From the research of (Demuth, 1984) it was calculated that the potential isentropic efficiency of the VPRT has an upper limit of 0.77, however this was just a preliminary investigation, without the building of a prototype.
Figure 10-5 Illustration of the VPRT and its general concept, showing the inner rotor and velocity pump while the outer rotor accelerates the flow and rotates as a result of this acceleration.

Two Phase Flow Turbine
The two phase flow turbine is also similar to the radial outflow reaction turbine, however a curved reaction nozzle is used. This alteration eliminates certain losses that were present in the earlier models and an actual prototype was built and has achieved 50% turbine efficiency, while pointing out that there was room for improvement (Fabris, 2005). The phenomenon of erosion due to the two phase flow was not considered in the research of Fabris, the main objective of Fabris’ research was to determine the possible efficiency of conversion when utilising pressurised, saturated liquid as an input to a turbine.
10.2.2 Potential Advantages

The major drawbacks of total flow systems are complexity of two-phase flow and design of a machine capable of efficiently performing the challenge of energy conversion while being resistant to damage. If these challenges are overcome the possible applications in geothermal power plants are:

- Use in a topping cycle for a single flash power plant
- Use in a bottoming cycle for a single flash power plant
- Use in a binary cycle power plant.

The potential advantage of using a total flow system in geothermal power plants is highlighted by (House, 1978a), where a RORT is modelled as a part of different geothermal power plants and the efficiency is compared to that of conventional geothermal power plants. From a system perspective the advantage of using a RORT in a bottoming system is that separated geothermal liquid can be utilised for power generation, instead of just being reinjected. In the case of a single flash power plant, it may be possible for the RORT to replace both the separator and conventional steam turbine, decreasing the amount of equipment in the cycle.
Figure 10-7, Figure 10-8 show different possible configurations of geothermal power plants. Figure 10-7 represents a power plant which uses a RORT total flow turbine with an assumed turbine efficiency of 50% to extract work from a saturated geothermal resource. Figure 10-8 is a geothermal hybrid plant similar to a single flash plant, the major difference is that it uses a RORT instead of a valve before the steam separator, the RORT turbine extracts useful energy before the geothermal fluid enters the separator. The power plants with the RORT can be compared to conventional single and double flash power cycles utilising a resource of the same temperature (148.9°C) and with cycle efficiencies of 34.7%.
and 47.7% respectively (more detailed diagrams of the single and double flash plants can be found in the appendix in Figure 13-5 and Figure 13-6). From the calculated efficiencies by (House, 1978b, House, 1978a) which are also indicated in the figures there are results which are listed below, it should be noted that in this case the cycle efficiency is the ratio of the turbine work to the isentropic work:

- Total flow geothermal system utilising a RORT turbine has a cycle efficiency of 46.6% while the conventional double flash system has an efficiency of 47.7%. Even though the double flash system requires more equipment to operate (two turbines, two steam separators, two scrubbers) it only translates to a 1.1% greater effective efficiency than the total flow system.
- The hybrid system with the RORT used before the steam separator achieves a very high effective efficiency of 53.0%.
- The conventional single flash plant has a effective efficiency of only 37.4%.

The above points show that even though total flow power plants do not exist today in practice that they can have a high level of performance in the future; if setbacks to implementation such as efficient design and resistance to damage can be overcome, there is a high possibility that this concept will be used in future geothermal power plants.
11 DISCUSSION AND CONCLUSIONS

The aim of the MSc work is to explain the operation of the different types of geothermal power plants in use around the world today. The power plants are described from a system perspective then the operation of major components is explained. In addition to these things, there are also explanations which parameters are most important to the operation of each plant from a performance perspective.

Geothermal electricity is responsible for a very small fraction of global power generation. Also it has a potential for growth and development and uses technology similar to fossil-fuel fired power plants and even nuclear power plants (for example steam turbine, condenser and cooling tower). A possible consequence of the characteristics mentioned above is that specific information about the vital components used in geothermal power plants is not widely publicised, for example in most of the information concerning specifics about turbines used in geothermal plants was obtained from technical journals from only a handful of turbine manufacturers. For the organic Rankine cycle the optimisation methodology and choice of working fluid were emphasised over the components and processes of the ORC. The reason for doing this is that the major difference between the organic Rankine and the conventional Rankine cycle is the working fluid which affects the operating conditions; components and processes in the organic Rankine cycle are similar to the conventional Rankine cycle.

This report also includes innovative ways in which heat from geothermal resources can be used to generate electricity. Only a few major types are covered in this report and these represent the most interesting ones with great potential for use in the future. It should be noted that Enhanced Geothermal Systems are not covered in this report because there will be a special chapter on the subject in CompEdu.

In conclusion it can be said that the major types of geothermal power plants have been described and explained in a way that facilitates understanding into how each type works and for which situation(s) each is most suited. More specific components such as piping and valves can be considered for future work. Pipes and valves handle and transport the geothermal fluid, and piping must be constructed to minimise unwanted pressure variations. Pipes are used throughout the power plant and have different requirements for different sections of the plant. Also a recommendation for further work is to do more research into the economics of geothermal power generation. Operating costs of geothermal power
Plants are of significant interest since geothermal power plants are thermal power plants where combustion of a fuel does not occur; this means there are no costs associated with fuel.
12 REFERENCES

The Stainless Steel Family [Online]. Available:


COMPEDU 2011.


http://www.iga.1it.pl/314,what_is_geothermal_energy.html.


HOUSE, P. A. 1978b. Performance tests of the radial outflow reaction turbine for geothermal applications.


## 13 APPENDICES

### 13.1 Appendix A

**Table 10 Top twenty four geothermal power generating countries**

<table>
<thead>
<tr>
<th>Country</th>
<th>Installed Capacity (MWₑ)</th>
<th>Rank</th>
</tr>
</thead>
<tbody>
<tr>
<td>United States</td>
<td>3086</td>
<td>1</td>
</tr>
<tr>
<td>Philippines</td>
<td>1904</td>
<td>2</td>
</tr>
<tr>
<td>Indonesia</td>
<td>1197</td>
<td>3</td>
</tr>
<tr>
<td>Mexico</td>
<td>958</td>
<td>4</td>
</tr>
<tr>
<td>Italy</td>
<td>843</td>
<td>5</td>
</tr>
<tr>
<td>New Zealand</td>
<td>628</td>
<td>6</td>
</tr>
<tr>
<td>Iceland</td>
<td>575</td>
<td>7</td>
</tr>
<tr>
<td>Japan</td>
<td>536</td>
<td>8</td>
</tr>
<tr>
<td>El Salvador</td>
<td>204</td>
<td>9</td>
</tr>
<tr>
<td>Kenya</td>
<td>167</td>
<td>10</td>
</tr>
<tr>
<td>Costa Rica</td>
<td>166</td>
<td>11</td>
</tr>
<tr>
<td>Nicaragua</td>
<td>88</td>
<td>12</td>
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<tr>
<td>Russia</td>
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<td>13</td>
</tr>
<tr>
<td>Turkey</td>
<td>82</td>
<td>14</td>
</tr>
<tr>
<td>Papua New Guinea</td>
<td>56</td>
<td>15</td>
</tr>
<tr>
<td>Guatemala</td>
<td>52</td>
<td>16</td>
</tr>
<tr>
<td>Portugal</td>
<td>29</td>
<td>17</td>
</tr>
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<td>China</td>
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<td>18</td>
</tr>
<tr>
<td>France</td>
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<td>Ethiopia</td>
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</tr>
<tr>
<td>Germany</td>
<td>6.6</td>
<td>21</td>
</tr>
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<td>Austria</td>
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<td>Australia</td>
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<td>23</td>
</tr>
<tr>
<td>Thailand</td>
<td>0.3</td>
<td>24</td>
</tr>
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</table>
Table 11 Relationship between Flow Rate through the geothermal wellhead, Pressure Ratio of the wellhead pressure in relation to the reservoir pressure and potential power Generation in a dry steam power plant (Glassley, 2010)

<table>
<thead>
<tr>
<th>Pressure (bars)</th>
<th>Pressure Ratio</th>
<th>Flow Rate (kg/s)</th>
<th>Enthalpy change (kJ/kg)</th>
<th>Power (kW)</th>
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</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.03</td>
<td>5.00</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>2.0</td>
<td>0.07</td>
<td>4.99</td>
<td>124.2</td>
<td>618</td>
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<tr>
<td>4.0</td>
<td>0.13</td>
<td>4.96</td>
<td>244.2</td>
<td>1199</td>
</tr>
<tr>
<td>6.0</td>
<td>0.20</td>
<td>4.90</td>
<td>314.2</td>
<td>1508</td>
</tr>
<tr>
<td>8.0</td>
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<td>1668</td>
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<td>10.0</td>
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<td>399.2</td>
<td>1774</td>
</tr>
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<td>0.40</td>
<td>4.58</td>
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</tr>
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<td>0.47</td>
<td>4.42</td>
<td>454.2</td>
<td>1776</td>
</tr>
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<td>16.0</td>
<td>0.53</td>
<td>4.23</td>
<td>469.2</td>
<td>1678</td>
</tr>
<tr>
<td>18.0</td>
<td>0.60</td>
<td>4.00</td>
<td>499.2</td>
<td>1597</td>
</tr>
<tr>
<td>20.0</td>
<td>0.67</td>
<td>3.73</td>
<td>509.2</td>
<td>1414</td>
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<td>22.0</td>
<td>0.73</td>
<td>3.40</td>
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<td>1188</td>
</tr>
<tr>
<td>24.0</td>
<td>0.80</td>
<td>3.00</td>
<td>534.2</td>
<td>961</td>
</tr>
<tr>
<td>26.0</td>
<td>0.87</td>
<td>2.49</td>
<td>544.2</td>
<td>677</td>
</tr>
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<td>28.0</td>
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<td>1.80</td>
<td>559.2</td>
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<td>30.0</td>
<td>1.00</td>
<td>0.00</td>
<td>566.2</td>
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</table>
Figure 13-1 Graph plotting Total power output against Separator pressure (graph to the left) and Separator temperature (graph to the right) for a choked flow system

Table 12 Data calculated for different separator conditions in a choked flow system

<table>
<thead>
<tr>
<th>Pressure Separator (bar, a)</th>
<th>Temperature Separator (°C)</th>
<th>Specific Work (kJ/kg)</th>
<th>Total Work (kW)</th>
<th>Mass flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.321</td>
<td>125</td>
<td>84.05</td>
<td>8068.8</td>
<td>96</td>
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<tr>
<td>2.701</td>
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<td>8046.72</td>
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<tr>
<td>6.178</td>
<td>160</td>
<td>82.25</td>
<td>7896</td>
<td>96</td>
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</table>
Table 13 Data calculated for different separator conditions in a non-choked flow system

<table>
<thead>
<tr>
<th>Pressure Separator (bar, a)</th>
<th>Temperature Separator (°C)</th>
<th>Specific Work (kJ/kg)</th>
<th>Total Work (kW)</th>
<th>Mass flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.321</td>
<td>125</td>
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<td>5.431</td>
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<td>83.82</td>
<td>3227</td>
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<tr>
<td>6.178</td>
<td>160</td>
<td>82.25</td>
<td>3049</td>
<td>37.07</td>
</tr>
</tbody>
</table>

13.1.1 Derivation of specific work associated with a steam separator conditions

![Figure 13-2 T-s diagram showing the path that geothermal fluid takes from the reservoir through to the separator and the steam turbine](image)

In the above diagram the processes and important points will be explained:
a-b – Flashing of the geothermal liquid to a vapour-liquid mixture at a lower pressure and temperature, this process is ideally isenthalpic.
b-c – Separation of the liquid portion of the mixture in the steam separator
b-d – Separation of the vapour portion of the mixture in the steam separator
d-e – Real expansion of the steam in a steam turbine
d-f – Ideal expansion of steam in a steam turbine (isenthalpic)

Another important parameter is the ratio of the length of c-b to c-d, this gives the value of the quality of the steam (the percentage of steam in the mixture, by mass).

The figure is essential to understanding how the specific work is calculated from the steam tables since the steam enthalpy, steam quality (both in the separator and after the turbine) turbine efficiency are all changing depending on the temperature or pressure selected for separation. By using fundamental principles and steam table information from the Engineering Equation Solver (EES), the results achieved by (DiPippo, 2008) will be derived.

Sample calculation of system parameters at a separator temperature of 125°C

In the reservoir, Point a
In the reservoir the temperature is 240°C (assumed to be saturated liquid), this gives an enthalpy at point a:

\[ T_a = 240°C, \text{ saturated liquid } (P_a = 33.45 \text{ bar}, a) \]
\[ h_a = 1037 \text{ kJ/kg} \]

After Flashing Process, Point b
The pressurised liquid is flashed to a vapour-liquid mixture at 125°C (Point b) in a process assumed to be isenthalpic. The pressure and steam quality of the mixture can be determined since the temperature and enthalpy are known

\[ h_b = h_a \]
\[ T_b = 125°C \]
\[ P_b = 2.32 \text{ bar}, a \]
\[ x_b = 0.234 \]

After Separation, Point c and Point d
Point c represents the liquid portion that is separated from the mixture and Point d represents the steam that is separated and removed for expansion in the steam turbine. The enthalpies at points c and d can be determined since the temperature is known and it is known that at point c, there is saturated liquid and at point d there is saturated vapour. The enthalpy at point c however is not important to the specific work available to the turbine.

\[ T_d = 125^\circ C \]
\[ P_d = 2.32 \text{ bar, } a \]
\[ x_d = 1 \]
\[ h_d = 2713 \text{ kJ/kg} \]

After expansion in the turbine, Point e
Point e is at the outlet of the steam turbine, the state of the working fluid at this point depends on the isentropic efficiency of the turbine and also the conditions at the inlet of the turbine. The isentropic efficiency of the turbine however changes depending on the amount of liquid in the mixture, the Baumann rule gives a good approximation for the deterioration of the performance of the turbine in moist-vapour conditions.

The isentropic efficiency of the turbine is given by

\[ \eta_{\text{isentropic wet}} = \frac{h_d - h_e}{h_d - h_f} \]

Rearranging the equation, we get

\[ h_e = h_d - [\eta_{\text{isentropic wet}} \times (h_d - h_f)] \]

According to the Baumann rule

\[ \eta_{\text{isentropic wet}} = \eta_{\text{isentropic dry}} \times \frac{(1 + x_e)}{2} \]

A reasonable dry isentropic efficiency of the turbine is given as 0.85 (DiPippo, 2008). In the above two equations, the wet isentropic efficiency is an unknown, and it affects the values of enthalpy and quality at point e. However by using the EES program, values were substituted into the above equations until there was convergence in the values of wet isentropic efficiency. The results of the calculation are in Table 14. The results indicate that the average turbine isentropic efficiency is still reasonably high for a power plant operating with steam at moderate temperatures and pressures. The average turbine isentropic efficiency drops to about 80% in the wet steam condition from 85% in a dry steam condition.
Calculation of Specific work

Now that all the relevant enthalpies have been determined, the specific work can be calculated. A straightforward approach is to imagine 1 kg/s of water extracted from the geothermal reservoir. When the water is flashed a portion of it will remain in the liquid form and the rest will exist as steam ($x_b \times 1\ kg/s$). The mixture then enters the steam separator where the mass flow of $x_b\ kg/s$ of steam is separated and taken to the steam turbine where work is extracted. The specific work available is the product of the mass flow of steam and the enthalpy difference across the turbine.

$$Specific\ Work = x_b \times (h_d - h_e)$$

$$Specific\ Work = 0.234 \times (2713 - 2354)$$

$$Specific\ Work = 84.01\ kW/kg$$

The value determined by (DiPippo, 2008) for the same separator operating conditions was 84.05 kW/kg. In the table below the result of calculation are compared with those determined by (DiPippo, 2008).

<table>
<thead>
<tr>
<th>Temperature conditions in the separator (°C)</th>
<th>Calculated Specific Work (kJ/kg)</th>
<th>Specific Work determined by (DiPippo, 2008) (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>83.99</td>
<td>84.05</td>
</tr>
<tr>
<td>130</td>
<td>85.08</td>
<td>85.15</td>
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<tr>
<td>135</td>
<td>85.69</td>
<td>85.80</td>
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<td>85.86</td>
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</tr>
<tr>
<td>160</td>
<td>82.13</td>
<td>82.25</td>
</tr>
</tbody>
</table>
As can be seen from the table the calculated results are similar to those obtained through the literature, the small difference (less than 0.16%) could be attributed to different Steam tables being used and also rounding error.

This method of calculating specific work in the separator is logical and provides a good starting point for determining the best separator conditions for operation. However in order to obtain more accurate results, pressure and energy losses that were previously neglected should be accounted for in the calculation.
Sample calculation of specific power output from a double flash power plant with a reservoir condition of 240°C saturated liquid

Fig. 13-3 a T-s diagram representing the double flash power cycle being used for this sample calculation

Referring to Figure 13-3, compressed geothermal liquid (point a) is flashed (point b) and then separated into saturated steam (point d) and saturated water (point c). The saturated steam passes through a high pressure steam turbine, while the saturated liquid at point c is flashed a second time (point f) and then separated into saturated water (point g) and saturated steam (point e). From this point the saturated steam at point e as well as the high pressure turbine exhaust from point d are combined and pass through the low pressure turbine, with the turbine exhaust being point h. The low pressure turbine exhaust at point h is condensed in the condenser (point i). Saturated water at points g and i are removed, most times for reinjection to the geothermal reservoir.

The following example assumes that 1kg/s of fluid is extracted from the production well in order to calculate the specific work from the cycle, and subsequent mass flows calculated in the turbine and the separator are fractions of the initially extracted 1kg/s
In the reservoir, Point a

In the reservoir the temperature is 240°C (assumed to be saturated liquid), this gives an enthalpy at point a:

\[ T_a = 240°C, \text{ saturated liquid (} P_a = 33.45 \text{ bar, } a \) \]
\[ h_a = 1037 \text{ kJ/kg} \]
\[ \dot{m}_a = 1 \text{ kg/s} \]

After Flashing Process, Point b

The pressurised liquid is flashed to a vapour-liquid mixture at 125°C (Point b) in a process assumed to be isenthalpic. The pressure and steam quality of the mixture can be determined since the temperature and enthalpy are known

\[ h_b = h_a \]
\[ T_b = 170°C \]
\[ P_b = 7.91 \text{ bar, } a \]
\[ x_b = 0.1552 \]
\[ \dot{m}_b = 1 \text{ kg/s} \]

After Separation, Point c and Point d

Point c represents the liquid portion that is separated (from first separator) from the mixture and Point d represents the steam that is separated and removed for expansion in the steam turbine. The enthalpies at points c and d can be determined since the temperature is known and it is known that at point c, there is saturated liquid and at point d there is saturated vapour. The enthalpy at point c is important to the specific work available to the turbine, since the saturated liquid will be used in the second flash process.

Point c
\[ T_c = 170°C \]
\[ P_c = 7.91 \text{ bar, } a \]
\[ x_c = 0 \]
\[ h_c = 719.3 \text{ kJ/kg} \]
\[ \dot{m}_c = 0.8448 \text{ kg/s} \]
Point d

\[ T_d = 170^\circ\text{C} \]
\[ P_d = 7.91 \text{ bar, a} \]
\[ x_d = 1 \]
\[ h_d = 2768 \text{ kJ/kg} \]
\[ \dot{m}_d = 0.1552 \text{ kg/s} \]

After expansion in the high pressure turbine, Point e

Point e is at the outlet of the steam turbine, the state of the working fluid at this point depends on the isentropic efficiency of the turbine and also the conditions at the inlet of the turbine. The isentropic efficiency of the turbine however changes depending on the amount of liquid in the mixture, the Baumann rule gives a good approximation for the deterioration of the performance of the turbine in moist-vapour conditions.

The isentropic efficiency of the turbine is given by

\[
\eta_{\text{isentropic wet}} = \frac{h_d - h_e}{h_d - h_f}
\]

Rearranging the equation, we get

\[
h_e = h_d - \left( \eta_{\text{isentropic wet}} \times (h_d - h_f) \right)
\]

According to the Baumann rule

\[
\eta_{\text{isentropic wet}} = \eta_{\text{isentropic dry}} \times \frac{(1 + x_e)}{2}
\]

From the case discussed by (DiPippo, 2008) the dry isentropic efficiency of the turbine was given as 0.85. In the above two equations, the wet isentropic efficiency is an unknown, and it affects the values of enthalpy and quality at point e. However by using the EES program, values were substituted into the above equations until there was convergence in the values of wet isentropic efficiency. The results of the calculation are in the table below.

<table>
<thead>
<tr>
<th>(\eta_{\text{isentropic wet}})</th>
<th>(\eta_{\text{isentropic dry}})</th>
<th>(x_e)</th>
<th>(h_d) (kJ/kg)</th>
<th>(h_e) (kJ/kg)</th>
<th>(\dot{m}_d) (kg/s)</th>
<th>Real Work (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.8162</td>
<td>0.85</td>
<td>0.9205</td>
<td>2768</td>
<td>2505</td>
<td>0.1552</td>
<td>40.82</td>
</tr>
</tbody>
</table>

Second flash process, Point c to Point f

From point c the saturated liquid is flashed a second time this time to a temperature of 105°C.

\[ h_f = h_c \]
\[ T_f = 105^\circ C \]
\[ P_f = 1.208 \text{ bar}, a \]
\[ x_f = 0.1244 \]
\[ \dot{m}_f = 0.8448 \text{ kg/s} \]

Second separation process, Point g and Point h

Point g
\[ T_g = 105^\circ C \]
\[ P_g = 1.208 \text{ bar}, a \]
\[ x_g = 0 \]
\[ h_g = 440.2 \text{ kJ/kg} \]
\[ \dot{m}_g = (0.8448 \text{ kg/s}) \times (1 - 0.1244) \rightarrow 0.7397 \text{ kg/s} \]

Point h
\[ T_h = 105^\circ C \]
\[ P_h = 1.208 \text{ bar}, a \]
\[ x_h = 1 \]
\[ h_h = 2684 \text{ kJ/kg} \]
\[ \dot{m}_g = (0.8448 \text{ kg/s}) \times (0.1244) + (0.1552 \text{ kg/s}) \times (0.9204) \rightarrow 0.2479 \text{ kg/s} \]

Expansion in the low pressure turbine
Repeating the same procedure as in the high pressure turbine, the expansion in the low pressure turbine must take into account that the mass flow entering the low pressure turbine is the combination of the high pressure turbine exhaust and the low pressure steam made available from the second flashing process.

Table 17 Calculated parameters for low pressure steam turbine

<table>
<thead>
<tr>
<th>( \eta_{\text{isentropic wet}} )</th>
<th>( \eta_{\text{isentropic dry}} )</th>
<th>( x_i )</th>
<th>( h_h (\text{kJ/kg}) )</th>
<th>( h_i (\text{kJ/kg}) )</th>
<th>( \dot{m}_h (\text{kg/s}) )</th>
<th>Real Work (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.8164</td>
<td>0.85</td>
<td>0.9208</td>
<td>2684</td>
<td>2403</td>
<td>0.2479</td>
<td>69.66</td>
</tr>
</tbody>
</table>
Calculation of Specific work
Now that all the relevant enthalpies and relative mass flows have been determined, the specific work can be calculated. Since the work output of the high pressure turbine and low pressure turbine are calculated for a 1 kg/s mass flow, the specific work output from the cycle is simply the sum of the work outputs for a 1kg/s mass flow.

\[
Specific \ Work = \frac{HP \ Turbine \ Work}{(kg/s)} + \frac{LP \ Turbine \ Work}{(kg/s)}
\]

\[
Specific \ Work = 40.82 \frac{kJ}{kg} + 69.66 \frac{kJ}{kg}
\]

\[
Specific \ Work = 110.48 \frac{kJ}{kg}
\]
Table 18 Extensive Numerical calculation of parameters of an organic Rankine cycle for a geothermal resource with a temperature of 90 °C and a gross power output of 10MW_e, showing the characteristics of the cycle for four different working fluids (Madhawa Hettiarachchi et al., 2007)

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>PF5050</th>
<th>HCFC 123</th>
<th>Ammonia</th>
<th>n-Pentane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross Power W_{Gross} (MW_e)</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Geothermal water inlet temperature T_{HWI} (°C)</td>
<td>90</td>
<td>90</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td>Cooling water temperature T_{CWl} (°C)</td>
<td>30</td>
<td>30</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Evaporation Temperature (°C)</td>
<td>80.0</td>
<td>79.9</td>
<td>76.9</td>
<td>80.2</td>
</tr>
<tr>
<td>Evaporation Pressure (MPa)</td>
<td>0.48</td>
<td>0.49</td>
<td>3.83</td>
<td>0.37</td>
</tr>
<tr>
<td>Condensation temperature (°C)</td>
<td>41.0</td>
<td>41.6</td>
<td>43.0</td>
<td>40.7</td>
</tr>
<tr>
<td>Condensation pressure (MPa)</td>
<td>0.14</td>
<td>0.16</td>
<td>1.70</td>
<td>0.12</td>
</tr>
<tr>
<td>Geothermal water outlet temperature (°C)</td>
<td>85.7</td>
<td>85.0</td>
<td>81.45</td>
<td>86.0</td>
</tr>
<tr>
<td>Cold water outlet temperature (°C)</td>
<td>35.2</td>
<td>35.5</td>
<td>37.41</td>
<td>35.7</td>
</tr>
<tr>
<td>Geothermal water flow rate (kg/s)</td>
<td>8496</td>
<td>5972</td>
<td>3693</td>
<td>7236</td>
</tr>
<tr>
<td>Cooling water flow rate (kg/s)</td>
<td>6598</td>
<td>4864</td>
<td>3898</td>
<td>4640</td>
</tr>
<tr>
<td>Working fluid flow rate (kg/s)</td>
<td>1243</td>
<td>669</td>
<td>494</td>
<td>294</td>
</tr>
<tr>
<td>Geothermal water pumping power (kW)</td>
<td>395</td>
<td>454</td>
<td>767</td>
<td>374</td>
</tr>
<tr>
<td>Cooling water pumping power (kW)</td>
<td>522</td>
<td>567</td>
<td>842</td>
<td>518</td>
</tr>
<tr>
<td>Working fluid pumping power (kW)</td>
<td>365</td>
<td>212</td>
<td>624</td>
<td>169</td>
</tr>
<tr>
<td>U_C (kW/m²K)</td>
<td>4.4</td>
<td>5.50</td>
<td>9.9</td>
<td>6.3</td>
</tr>
<tr>
<td>U_E (kW/m²K)</td>
<td>2.87</td>
<td>4.46</td>
<td>11.91</td>
<td>2.82</td>
</tr>
<tr>
<td>Evaporator area A_E (m²)</td>
<td>7059</td>
<td>3766</td>
<td>1364</td>
<td>5660</td>
</tr>
<tr>
<td>Condenser area A_C (m²)</td>
<td>3961</td>
<td>2773</td>
<td>1377</td>
<td>2317</td>
</tr>
<tr>
<td>Ratio of Heat exchanger area to net power generation ζ (m²/kW)</td>
<td>1.26</td>
<td>0.70</td>
<td>0.35</td>
<td>0.89</td>
</tr>
<tr>
<td>Rankine cycle efficiency η_R</td>
<td>7.8</td>
<td>9.8</td>
<td>8.9</td>
<td>9.9</td>
</tr>
</tbody>
</table>
Figure 13-4 Illustration of an Aeolipile or Hero Engine which is used for energy conversion (1876)

Cycle Efficiency = 37.4%

Figure 13-5 Illustration of a single flash power plant configuration
Figure 13-6 Illustration of a double flash power plant configuration

Cycle Efficiency = 47.7%
13.2 General concepts

In the different types of geothermal power plants there are several concepts that are repeated several times and the purpose of this chapter is to serve as a reference point for these general concepts.

13.2.1 Enthalpy

According to (Baker, 2002), quote: “Thermal energy changes under constant pressure (neglecting any field effects) are most conveniently expressed in terms of the enthalpy, H, of a system. Enthalpy, also called heat content, is defined by:

\[ H = E + PV \]

Where,

- \( E \) - is the internal energy; The sum of the kinetic energy (energy of motion) and potential energy (stored energy) of a system is called its internal energy, \( E \). Internal energy is characterised solely by the state of the system
- \( P \) - is the pressure
- \( V \) - is the volume

In essence enthalpy is the total measure of the energy in a thermodynamic system, it is the internal energy of the system plus the energy needed to accommodate the system in the environment (pressure-volume term)

13.2.2 Entropy

Simply put, entropy is the measure of disorder in a system (Weisstein, 2007). Entropy represents the energy (per degree of absolute temperature, \( T \)) in a system that is not available for work. In terms of entropy, the Second Law says that “all natural processes tend to occur only with an increase in entropy, and the direction of the process always is such as to lead to an increase in entropy.” (Baker, 2002)

13.2.3 Mollier Diagram

The Mollier Diagram, also called a T-s diagram, describes the state of a substance. Using the example of water, the Mollier Diagram, is used to determine several properties of water (Temperature, Pressure, enthalpy, entropy, specific volume, steam quality). If a few (two or three) of the properties are known, normally the rest of the properties can be found in this diagram (see Figure 13-7)
Figure 13-7 Enthalpy-entropy Mollier Diagram for Water and Steam, it is possible to know temperatures, pressures, specific volumes and steam quality from this diagram.
13.2.4 Dry Steam

Dry Steam seems like a contradictory statement, because after all, steam is the vapour phase of water (so it is inherently wet). However dry steam refers to steam which does not contain any liquid water. It is steam which exists on the vapour-only side of the Mollier curve and the dry steam state is bordered by the saturated vapour curve and the critical point (above which the vapour becomes a supercritical fluid).

13.2.5 Reinjection

Reinjection is the process of returning utilised geothermal water to the geothermal reservoir. In geothermal power plants, reinjection was initially seen as a method to dispose geothermal waste water, since disposal of the fluid above ground is generally frowned upon, however over time it has become a vital tool to geothermal reservoir management and if properly implemented can increase the productive lifetime of geothermal reservoirs. It is a very complex process which requires extensive knowledge of the chemistry of the geothermal fluid, water-rock interaction, geothermal reservoir engineering and mechanical engineering (Stefansson, 1997). There are possible advantages and disadvantages that come with reinjection. Examples of potential benefits include increased productivity from production wells, an extended productive lifetime for the reservoir and maintaining or
boosting pressure at the production well. Potential negative effects are that reinjection can cool the geothermal reservoir by disturbing the heat balance between the geothermal fluid and the heat from the surrounding rocks and that water may be transferred from the reinjection well to the production well without being heated sufficiently. The practice of reinjection is highly individual, meaning that each geothermal field will respond differently and that extensive research and optimisation must be done to determine how best to implement reinjection. It should be noted that not all geothermal fields use the process of reinjection and that some systems may not require reinjection for increased productivity. An example of such a system is a geothermal field with a stable pressure and a high recharge rate of geothermal fluid over time when compared with the amount of fluid extracted from the production well. Reinjection is not necessary for increased performance in these cases as the system already reaps the benefits of reinjection; maintaining reservoir pressure and a high rate of recovery of geothermal fluids.

Reinjection Parameters

The important parameters that must be managed against each other, and optimised for reinjection include the following (Stefansson, 1997):

- **Disposal of waste fluid**
  - From an environmental point of view, the industry’s view of reinjection has shifted from being only as a method of disposal to a tool for reservoir management.

- **Cost**
  - Reinjection requires an initial investment in piping and wells, however it is well worth the cost if the benefits of increased wellhead production, extension of the useful life of the resource and maintenance of reservoir pressure can be realised.

- **Reservoir temperature**
  - A major consideration that is mentioned as a potential disadvantage for reinjection is the risk for thermal breakthrough, which occurs when the cool injected fluid reaches the production wells (Kaya et al., 2011). Thermal breakthrough can have negative effects on the capacity for power generation since the temperature of the resource decreases. When injection wells are relatively close to production wells (infield reinjection) there is a greater risk for thermal breakthrough because of the proximity of the wells. This risk can be managed by doing research into how the reservoir functions, tracing the
movement of fluid from the reinjection well to the production well and also moving reinjection well further away from the production well

- **Reservoir pressure**
  - When geothermal resources are utilised without reinjection, over time pressure in the reservoir will decrease, meaning that less fluid will be available for extraction. By implementing reinjection the pressure in the reservoir tends to increase with the amount of fluid injected. By maintaining a higher pressure the amount of fluid made available is increased.

- **Temperature of injected fluid**
  - The temperature of the fluid to be reinjected has been investigated to determine if injecting the separated geothermal fluid at a lower temperature translates to increased efficiency as it seems it would. However if the resource is injected into the reservoir and fully recovered the injection temperature has no influence on the efficiency.

- **Silica scaling**
  - Silica scaling is a consideration during reinjection because when flashing occurs in a flash geothermal power plant the separated liquid portion becomes super saturated in SiO₂, and this leads to deposits of silica forming in scale form, especially when reinjection occurs at lower temperatures. The process rate at which scaling occurs is highly pH dependent and efforts to control the pH have been successful in controlling the rate of scaling. Other than controlling the pH, higher temperature of the injected fluid reduces the rate of scale deposition.

- **Sandstone**
  - In low enthalpy geothermal fields, where the rock is of the sedimentary kind, reinjection has been unsuccessful, for reasons not fully understood. Currently research is being conducted to understand why reinjection has failed in this environment.

- **Location of reinjection wells**
  - It has been found that for most efficient reinjection that reinjection wells should surround the production wells.

- **Subsidence**
  - When a geothermal resource is exploited without reinjection, many times the pressure changes, can lead to changes in the reservoir structure and in turn cause subsidence of the land. Subsidence means that the land elevation is lowered because of changes due to the pressure in the reservoir. It has been
recorded that reinjection can prevent subsidence from being a major concern by maintaining pressure conditions in the reservoir.

An example of a study in which reinjection was predicted to have a great impact on power production is about the geothermal field in Ahuachapan in El Salvador (Ripperda et al., 1991). In this study it was predicted that, by using injection at 60% and increasing the power output from 50MW to 75MW, the power output could be maintained for 30 years as opposed to the current practice of disposing the geothermal fluid to the sea and having a declining power output during the 30 year time.

13.2.6 Control of impulse turbine stage on temperature

The control of the impulse turbine on temperature can be explained in terms of thermodynamics, where the flow over the stator or nozzle is described. In impulse turbines the static or total state of the flow and the dynamic states are very important in describing the flow. The static state of a property (enthalpy, pressure, temperature) is the state of the flow if its velocity was decreased to zero, that is, if it was stopped. The dynamic state of a property gives the state of the property while the flow is moving with a specified velocity. There is a relation between the static enthalpy, the dynamic enthalpy and the velocity of a flow which is given by:

\[ h_{\text{static}} = h_{\text{dynamic}} + \frac{c^2}{2} \]

\textit{Eq. 14}
Let us have point 0 and point 1 where point 0 is before the stator and point 1 is after the stator. The flow enters with a specific flow velocity, $c_0$, an dynamic enthalpy of $h_0$, and a dynamic pressure, $p_0$, and accordingly the stator increases the flow velocity to $c_1$ while the dynamic pressure is lowered to $p_1$, and the total enthalpy remains the same, since no work is extracted and there is a slight decrease in total pressure since there are frictional losses in the nozzle, the dynamic enthalpy and by extension temperature will decrease (Figure 13-9).
Figure 13-9 h-s diagram describing the flow through a nozzle of an impulse turbine and showing that the nozzle controls the temperature which the rotor is exposed to by increasing the velocity and kinetic energy of the flow.

As seen above in Figure 13-9 the dynamic enthalpy decreases due to the increase in kinetic energy occurring in the nozzle between points 0 and 1 (the most relevant terms of dynamic enthalpy and the kinetic energy term of $c_0^2/2$ are highlighted in red). The dynamic enthalpy decreases and also the dynamic temperature decreases because in general terms the relation between enthalpy and temperature is a proportional one given by:

$$h = c_p \times T$$  \hspace{1cm} \text{Eq. 15}

Where,

$h$ is enthalpy
$c_p$ is the specific heat capacity of the flow which remains relatively constant for these conditions
$T$ is temperature
13.2.7 Separator and Moisture Remover Design Specifications

Table 19 Design parameters (velocities) for efficient separator and moisture remover operation (Lazalde-Crabtree, 1984)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Separator</th>
<th>Moisture remover</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum steam velocity at the 2-phase inlet pipe</td>
<td>45 m/s</td>
<td>60 m/s</td>
</tr>
<tr>
<td>Recommended range of steam velocity at the 2-phase inlet pipe</td>
<td>25-40 m/s</td>
<td>35-50 m/s</td>
</tr>
<tr>
<td>Maximum upward annular steam velocity inside cyclone</td>
<td>4.5 m/s</td>
<td>6.0 m/s</td>
</tr>
<tr>
<td>Recommended range of upward annular steam velocity inside cyclone</td>
<td>2.5-4.0 m/s</td>
<td>1.2-4.0 m/s</td>
</tr>
</tbody>
</table>

Figure 13-10 Illustration showing the recommended dimensions of a steam separator according to (Lazalde-Crabtree, 1984)
Figure 13-11 Illustration showing the recommended dimensions of a moisture remover according to (Lazalde-Crabtree, 1984)