

Analysis of Lifecycle Water Requirements of Energy & Transportation Fuels: Electricity from Geothermal Resources

- Model Description

By

Gouri Shankar Mishra (gouri.mishra@gmail.com)

William Glassley (geobg@nf.au.dk)

Sonia Yeh (slyeh@ucdavis.edu)

December, 2010

Version 1.1

Report Number: UCD-ITS-RR-10-20A

**Institute of Transportation Studies
University of California, Davis**

1.	ABSTRACT	1
2.	ACKNOWLEDGEMENTS	2
3.	NOTATIONS	3
4.	MODEL OBJECTIVES.....	4
5.	DEFINING THE SYSTEM BOUNDARY	5
5.1.	WATER REQUIREMENTS CONSIDERED	5
5.1.1.	<i>Freshwater requirements</i>	5
5.1.2.	<i>Degraded water requirements.....</i>	7
5.1.3.	<i>Geothermal fluid requirements</i>	8
5.2.	FUNCTIONAL UNITS	9
6.	TYPES OF GEOTHERMAL RESOURCES AND POWER PLANTS.....	10
6.1.	GEOTHERMAL RESOURCES	10
6.1.1.	<i>Hydrothermal Resources.....</i>	10
6.1.2.	<i>EGS Resources.....</i>	11
6.2.	POWER PLANTS	12
6.3.	COOLING TECHNOLOGIES	13
7.	SUMMARY OF WATER REQUIREMENTS	15
7.1.	I. ELECTRICITY FROM HYDROTHERMAL RESOURCE USING ORC PLANT	15
7.2.	II. ELECTRICITY FROM EGS RESOURCE USING ORC PLANT	16
7.3.	III. ELECTRICITY FROM HYDROTHERMAL RESOURCE USING SINGLE FLASH POWER PLANT.....	17
7.4.	IV. ELECTRICITY FROM EGS RESOURCE USING SINGLE FLASH POWER PLANT	18
8.	WATER REQUIRED BY BINARY POWER PLANTS.....	19
8.1.	THERMODYNAMICS BEHIND ORC.....	19
8.2.	METHODOLOGY	24
8.2.1.	<i>Thermal Efficiency of OC plants.....</i>	26
8.2.2.	<i>Impact on efficiency – cooling system.....</i>	32
8.2.2.1.	<i>Dry Cooling.....</i>	32
8.2.3.	<i>Estimating parasitic power requirements</i>	40
9.	WATER REQUIRED BY FLASH POWER PLANTS.....	41
9.1.	THERMODYNAMICS BEHIND A SINGLE FLASH POWER PLANT.....	41
9.2.	METHODOLOGY	44
9.2.1.	<i>Exergy efficiency of flash power plants.....</i>	45
9.2.2.	<i>Parasitic power consumption.....</i>	46
10.	GEOTHERMAL RESOURCES	47
10.1.	WATER REQUIRED TO RECHARGE HYDROTHERMAL RESOURCES	47
10.2.	WATER REQUIRED FOR HEAT MINING OF EGS RESOURCES	47
10.3.	POWER REQUIREMENTS FOR GEOTHERMAL FLUID PRODUCTION AND INJECTION.....	48
11.	REFERENCES	50
12.	APPENDIX A: SOME ADVANCED ORC CONFIGURATIONS	54
13.	APPENDIX B: ORC POWER PLANTS	56
14.	APPENDIX C: T&D LOSSES	58

Abstract

The document describes the methodology and data sources for the model “Analysis of lifecycle water requirements of energy and transportation fuels: electricity from geothermal resources”. The model estimates water requirements for electricity from various forms of geothermal resources. It considers two types of hydrothermal resources – wet steam and hot water; as well as enhanced geothermal systems (EGS). Electricity can be generated using flash or binary (organic Rankine cycle) technology depending upon the temperature and pressure of geothermal fluid. Power plants can use three different types of cooling technologies – wet re-circulating, dry systems, and hybrid cooling systems. Requirements are calculated separately for freshwater, degraded water and geothermal fluid.

The spreadsheet-based model (Report Number: UCD-ITS-RR-10-20B) is available at <http://pubs.its.ucdavis.edu/>

The model is part of a series exploring the water footprint of future transportation fuels including bio-fuels and electricity. Other models currently under development examine the lifecycle water requirements of ethanol from corn grain and crop residue, and electricity from concentrated solar power, and biodiesel from soybean.

Acknowledgements

We thank all those who have offered ideas, data, information, and comments on the model including John Maulbetsch, Maulbetsch Consulting; Ron DiPippo, Professor Emeritus, University of Massachusetts Dartmouth, Alissa Kendall, University of California, Davis, and Chuck Kutscher, National Renewable Energy Laboratory.

The research effort is funded by the California Air Resources Board, the Energy Foundation and the David & Lucile Packard Foundation. Funding was also provided by Chevron Corporation Graduate Fellowship.

Notations

ACC	Air cooled condensers (sometimes also referred to as direct dry cooling systems)
BEV	Battery electric vehicle
EGS	Enhanced geothermal systems (sometimes also referred to as engineered geothermal systems)
IHE	Internal heat exchanger
NCG	Non-condensable gases
ORC	Organic Rankine Cycle (also called Binary cycle)
OTC	Once through cooling system
PHEV	Plug-in hybrid electric vehicle
TDS	Total dissolved solids (in mg/L)
T&D	Transmission and Distribution
ZLD	Zero liquid discharge

1. Model objectives

The model estimates water requirements for electricity from various forms of geothermal resources. It considers two types of hydrothermal resources – wet steam and hot water; as well as enhanced geothermal resources (EGS). The model considers multiple power generation technologies including single flash and various configurations of binary or organic Rankine cycle (ORC). Requirements for freshwater, degraded water and geothermal fluid are reported separately.

Based on inputs regarding temperature and form of a geothermal resource, configuration of the power plant and technology of the cooling system, the model estimates “average” water intensity for the electricity produced. The model depends upon an extensive literature review to determine various relationships necessary to determine water usage – for example the relationship between thermal efficiency of a binary power plant and temperature of inlet geothermal fluid, differences in efficiency between various ORC configurations, or different efficiency of a plant with a wet re-circulating cooling system versus one with a dry cooling system.

Since there are wide variations in the environmental conditions, input variations, plant design, and operation conditions, the relationships identified in this study were specified under specific parametric conditions but were generalized here to a broader set of conditions or scenarios. Further, water consumption of power plants will depend upon a large number of factors that the model currently does not consider – mineral content of the geothermal fluid, ambient temperature and humidity, specific design parameters of the power plant, and dissolved solids and chemical composition of freshwater withdrawn from ground or surface sources. Even for a specific plant, water intensity will vary over time due to fluctuations in ambient temperature and humidity, and in temperature of the geothermal fluid.

For these reasons, the model’s water usage estimates should be treated as first-order estimates.

2. Defining the system boundary

Establishing a quantitative model of a system requires that the boundaries of the system be rigorously established. For the purposes of this study, the model's boundary includes the geothermal field, the power generation unit, and the transmission and distribution (T&D) infrastructure to get electricity to the end user. The geothermal field consists of those systems used for geothermal fluid production and transfer to the energy conversion system, and then reinjection back to the ground. The geothermal field and power generation unit are nearly always collocated and jointly called the geothermal power plant.

Figure 1: System boundary for lifecycle analysis of electricity from geothermal



In our model, analysis has been undertaken for two types of end user – electricity end user and plug-in hybrid (PHEV) or battery electric vehicle (BEV) end user. Water requirements are represented using different functional units for each of the two types of end users. Further, to assess water requirements of electricity for charging of PHEV or BEV, two additional factors need to be considered – battery charger efficiency and battery efficiency.

2.1. Water requirements considered

The model considers withdrawal and consumption of three types of water – freshwater, degraded water, and geothermal fluids.

2.1.1. Freshwater requirements

The key focus of this study is freshwater use – water with low concentrations of dissolved solids. Per US Geological Survey (USGS 2010), freshwater has concentration of total dissolved solids (TDS) of less than 1,000 mg/L. Water with increasing levels of dissolved solids are classified as lightly saline (1,000 - 2,000 mg/L), medium saline (3,000 - 10,000 mg/L), and highly saline (10,000 – 35,000 mg/L). Our focus on freshwater is due to its scarcity and lack of substitutability at reasonable economic costs; saline water on the other hand is relatively abundant (Gleick 1996).

We adopt the following indicators of freshwater usage – withdrawal, consumptive use and degradative use (Owens 2001; NETL 2008; Bayart, Bulle et al. 2010). *Freshwater withdrawal* is the removal from a natural water body or groundwater aquifer for industrial, agricultural or domestic usage. *Freshwater consumptive use* denotes the use of freshwater when it is not released into the same watershed because of evaporation, product integration, or evapotranspiration by crops. Discharge into different watersheds or the sea, and sinking to a deep salt sink is also counted under consumptive use. *Water released* is the difference between withdrawal and

consumptive use. If the quality of water is “significantly” altered within the system, water released is referred to as *degradative use*.

Table 1: Water requirements considered for geothermal electricity

Freshwater	Degraded Water	Geothermal Fluid
Requirements considered		
- Power plant cooling	- Injection to maintain pressure of hydrothermal plants - Injection for heat mining of EGS resources	- Withdrawal for power generation - Steam condensate used for power plant cooling
Requirements not considered		
- Resource exploration & development (hydraulic fracturing) - Energy inputs for water supply and treatment - Power plant construction - Power plant cleaning	- Resource exploration & development (hydraulic fracturing) - Energy inputs for water supply and treatment - Power plant cooling using degraded water	

Freshwater is required by geothermal power plants primarily for cooling and dissipating waste thermal energy. There are multiple types of cooling systems and their impact on plant efficiency and water consumption differ. Freshwater is also required for a plant’s “hotel” load i.e. for cleaning, drinking and sanitary use.

While the key differentiator for freshwater is the overall concentration of dissolved solids, concentration of any individual component affects its usability. For example, EPRI (2003) lists the maximum allowable concentration of various constituents in the freshwater used for cooling systems. Similarly, the potable water standards by US Environmental Protection Agency (EPA) has the maximum allowable limits for various components like inorganic and organic chemicals, radionuclides, disinfection byproducts, and microorganisms. In this model, we do not analyze the concentration of individual constituents. When we refer to freshwater, we simply imply a low TDS and an “acceptable” level of concentration of individual chemical constituents.

In the current version of the model, we do not consider the energy consumption and corresponding “embodied water” requirements for supply and conveyance, treatment and distribution of freshwater to geothermal plants. In California around 5% of the state’s electricity is consumed to supply treated freshwater for industrial, agricultural and residential purposes (Klein, Krebs et al. 2005). Nationally, the average energy intensity of freshwater supply is 1.94 kJ/L (Klein, Krebs et al. 2005). Given the average water intensity of electricity produced in the US (Mishra and Yeh 2010), the embodied freshwater requirements to supply freshwater to a geothermal plant would be around 0.033 liters of fresh water withdrawn and 0.001 liters of fresh water consumed per liter of freshwater supplied. Since these numbers are small, we ignore the embodied water from energy use for water treatment and supply in our analysis.

The energy consumption for water supply, treatment and distribution varies significantly by locations. For example, the value is around 1.38 kJ/L in Northern California and around 9.7 kJ/L in Southern California (Klein, Krebs et al. 2005). The differences are largely due to conveyance distances and the need for extensive pumping to transport water over mountain ranges.

2.1.2. Degraded water requirements

“Degraded” water usually refers to water that cannot be referred to as freshwater and includes contaminated groundwater, treated municipal effluent, industrial process water or wastewater, irrigation return water, brackish water, and other types of water impacted by humans or naturally-occurring minerals (EPRI 2003). This category does not include ocean water (TDS of around 35,000 mg/L).

In the context of power generation from geothermal resources, degraded water may be used for injection and heat mining. Under normal operating conditions, geothermal fluids that are obtained from a hydrothermal system are re-injected into the geothermal reservoir. This is done to prevent declines in pressure and, hence, production. However, loss of fluid during power generation and cooling commonly results in a net reduction of fluid that is re-injected into the reservoir. For this reason additional water is often required from an external source. In the dry steam geothermal resources at Geysers, California, storm water runoff from power plant sites and treated municipal effluent are injected to maintain pressure (EPA 1999; City of Santa Rosa 2007). Sea water is injected for enhancement of dry steam resources at Larderello, Italy (Kaltschmitt 2007). Blowdown from cooling towers of power plants may also be injected to the geothermal resource.

For EGS resources, which do not have sufficient naturally occurring geothermal fluids, water is required to draw thermal energy to the surface for conversion to electrical energy. Degraded water may also be used for such purposes. Degraded water is used at the EGS resources at Cooper Basin in Australia (based on personal communications with Chris Mathews, Australian Geothermal Energy Association, May 2010).

Thus a wide range of degraded water sources, with significantly different chemical constituents both in type and quantity, may be used to maintain pressure in hydrothermal resources and mine heat from EGS resources. The only stipulation about the water quality is that it should not result in deposition of minerals that would reduce permeability, or result in excessive dissolution of rock that would ultimately result in mineral deposits on turbines or other infrastructure. During injection, the water will be heated as it passes through the thermal reservoir, so the only deposition that might happen would be related to minerals that have retrograde solubility - minerals like carbonates for which the solubility goes down as the temperature goes up. This implies that water with high levels of calcium, magnesium or carbon dioxide need to be treated before use.

Even though degraded water may also be used for cooling purposes, we do not consider the use of water with such high levels of TDS for cooling purposes given paucity of data¹. Maulbetsch and DiFilippo (2006) analyzed the use of brackish water from agricultural return flows or low-TDS oil-field produced water (2,000 to 5,000 mg/L), and saline water from high-TDS agricultural return or high-TDS oil-field produced water (5,000 mg/L) for cooling purposes in power plants. Power plants with degraded water supply for re-circulating cooling tower need water treatment facilities with high capital and operating costs, primarily parasitic power consumption and chemicals (EPRI 2003).

As in case of freshwater, we do not consider the energy consumption and corresponding “embodied water” requirements for supply and conveyance, treatment and distribution of degraded water to the geothermal fields. These requirements can be significant – for example 9 MW of power is required to convey 42 million liters of water daily for recharging of the Geysers Steamfield in California (City of Santa Rosa 2010); this represents 1% of the total operating capacity of around 1,000 MW.

It should be noted that although significant differences in quality exist between the various sources of degraded water, the model treats all under a single category.

2.1.3. Geothermal fluid requirements

Power generation from geothermal resources involves withdrawal of large volumes of geothermal fluids in form of steam or hot water or some combination of both. Power plant technologies convert thermal energy from these fluids to electrical energy, and re-inject most of the fluids through an injection well. Steam condensate at the outlet of the turbine is used for cooling water requirements and cooling tower blowdown is usually injected back to the ground. The volume of geothermal fluids withdrawn per unit of electricity generated primarily depends upon the temperature of the fluids, and to a small extent on the ambient temperature and efficiency of the power plant technology.

The quality of geothermal fluids can vary significantly from one location to another; and over time from the same location. In geothermal resources with naturally occurring geothermal fluids (hydrothermal resources), concentration of solids especially sodium chloride in the water can be very high. Chloride-bearing fluid is commonly discharged from hot springs and from most geysers (Barbier 2002). High concentrations of sulfate and bicarbonate may also exist making the geothermal fluid highly corrosive. Additionally, dissolved gases like ammonia, hydrogen sulfide, and methane may also be present. Concentration of TDS in the hydrothermal resources in the Imperial Valley of California can range from 240,000 mg/L in Salton Sea to 7,600 mg/L in East Mesa (Layton and Morris 1980).

¹ Ocean water is used in power plants with once-through cooling systems. Such power plants account for around 30% of total electricity generated in the United States. Ocean water requirements for such cooling systems are well researched. However, information is limited about requirements of saline water and other forms of degraded water for use in wet re-circulating cooling towers and water-augmented dry cooling towers.

The model reports the amount of geothermal fluid that needs to be withdrawn for power generation. In flash power plants (which will be elaborated later), some of the fluid in vapor state condenses to form high quality water with very low mineral content, which is then used for cooling purposes. The evaporated portion of this condensate is counted as geothermal fluid consumed by the model. It is not considered freshwater consumption because it is not sourced from a body of freshwater. This portion also represents the amount of supplemental degraded water that needs to be injected to the geothermal resource to maintain pressure.

2.2. Functional units

Estimate of water requirements of electricity end user is represented in form of liters of water (withdrawn and/or consumed) per kJ of power consumed by the end user.

For PHEV and BEV end users, the model also presents estimates of water intensity in form of liters of water (withdrawn and/or consumed) per vehicle kilometer traveled (VKT). This representation takes into account energy efficiency of electric vehicles, battery charger efficiency, and battery efficiency. Energy efficiency of electric vehicles is assumed to be 937.3 MJ/VKT or 1,429.7 BTU/mile (GREET 2010)². The model defaults to a battery charger efficiency of 87% (for a 240 V charging system), and a battery efficiency of 85% (King and Webber 2008).

The model assumes transmission and distribution (T&D) losses of 6.14% which is the national average for 2008 (EIA 2010). However, there is significant regional variability in T&D losses; it ranges from less than 1% for Rhode Island to 21% for Montana (Appendix C).

Inclusion of T&D losses increases water intensity of electricity consumed by an electric end user by around 6.2% relative to water intensity when measured at the power plant level. Subsequent inclusion of battery charger efficiency and battery efficiency increases water intensity of electricity consumed by an electric vehicle end user by around 44% relative to water intensity when measured at the power plant level, and 35% relative to water intensity of electricity consumed by an electricity end user.

² Energy efficiency of conventional gasoline vehicles is 3280.45 MJ/VKT or 5003.88 BTU/VMT (GREET 2010).

3. Types of geothermal resources and power plants

The following table summarizes the various geothermal resources, and types of power plants and cooling technologies considered in the current version of the model.

Table 2: Resource types and power plant technologies considered

Geothermal resources	Power plants technologies	Cooling technologies
Within scope		
Hydrothermal – Wet steam	Organic Rankine Cycle (ORC, various configurations).	Dry cooling tower / Air cooled condenser
Hydrothermal – Hot water	Flash power plant (single flash only)	Wet re-circulating cooling tower (with and without zero liquid discharge (ZLD))
Enhanced Geothermal Systems (EGS)		Hybrid cooling system (with and without ZLD)
Outside scope		
Hydrothermal – Dry steam or Vapor	Double flash power plant	Once-through cooling system
Geo-pressured resources	Flash-Binary combined plant	Cooling ponds
Magmatic resources		

3.1. Geothermal Resources

There are four types of geothermal resources: hydrothermal, enhanced geothermal systems (EGS), geopressured, and magmatic. Nearly all geothermal power generated today is from hydrothermal resources. EGS creation and exploitation has been demonstrated extensively at various experimental sites in the US, Europe, and Japan (EPRI 1997; Tester, Anderson et al. 2006; DiPippo 2008); hence it is included in the current version of the model. There have been a few pilot projects to demonstrate the technical viability of geopressured and magmatic resources (EPRI 1997; Tester, Anderson et al. 2006; DiPippo 2008); however significant technological developments are necessary before these can be exploited on a commercial basis. Further, due to lack of data on water use by these resources, we have excluded these resources from further analysis.

3.1.1. Hydrothermal Resources

Hydrothermal resources can be classified into dry steam (or vapor) dominated fields and water-dominated fields (Barbier 2002). Water dominated fields are further classified into hot water fields and wet steam fields.

Vapor-dominated reserves play a very significant role in geothermal power generation today; about half of geothermal electricity in the world today comes from six steam fields including the Geysers in California (Barbier 2002). Such reservoirs are either at a temperature in excess of the critical point (373.946 °C), or at a high enough pressure such that the fluid will not

intersect the two-phase region of “Liquid + Vapor” on its rise to the energy conversion facility. From a thermodynamic point of view, dry steam systems provide the greatest amount of energy per kilogram of fluid extracted (Glassley 2010). This results from the fact that there is much less separation of liquid from the steam and the resulting partitioning of the enthalpy between those two phases. Instead, most of the enthalpy of the fluid remains with the steam as it enters the turbine, and becomes available for energy conversion (Glassley 2010).

Wet steam fields contain pressurized water at temperatures exceeding 100 °C and small quantities of steam. An impermeable cap-rock generally exists to prevent the fluid from escaping to the surface, thus keeping it under pressure. When the fluid is brought to the surface, as in case of producing well, and its pressure decreases, a fraction of the fluid flashes to steam while the greater part remains as boiling water. The water-steam ratio varies from field to field, and even from one well to the next in the same field (Barbier 2002). Hot water fields produce hot water at the surface at temperatures of around 100 °C indicating that the heat source is not sufficient to generate steam.

Vapor and hot steam fields manifest at the surface in form of geysers and hot springs, while hot water fields appear only as hot springs. Today, hot water fields play a very limited role in electricity generation but a significant role in direct-use heat applications - for example state of Oregon uses geothermal energy in the form of direct use applications to the extent of around 500 million BTUs, which is equivalent to around 20 MWt (thermal megawatts of power).

Coproduced hot water from oil and gas operations also represents a potential source of geothermal power; Potential electricity generation from such fluids in the US has been estimated at 6,000 MW (Tester, Anderson et al. 2006). Some of these resources exist at pressures greater than hydrostatic, thus making them "geopressured". Such resources often also contain significant methane associated with the geothermal fluid. Hence, the combination of thermal, chemical and kinetic energy inherent in such systems makes them potentially attractive geothermal energy resources.

3.1.2. EGS Resources

EGS resources have high temperature but contain little or no geothermal fluid, and are not very permeable. To exploit such resources, a permeable reservoir must be created by hydraulic fracturing, and fluid from the surface must be pumped through the fractures to extract heat from the rocks. A pair of wells is drilled into the rocks terminating several hundred feet apart. Fluid, which is usually water, is injected under high pressure through the injection well which creates an artificial reservoir. The fluid then returns to the surface through the production well, and thus transfers the heat to the surface as steam or hot water.

The injection pump provides the sole motive force for moving the water continuously around the loop to mine energy from the reservoir and deliver it to a power plant on the surface.

EGS resources are sometimes referred to as engineered geothermal systems or hot dry rock (HDR). Over the long term, it is anticipated that most of the future growth in electricity from geothermal resources may come from EGS resources (Tester, Anderson et al. 2006; Chandrasekharam and Bundschuh 2008).

3.2. Power plants

There are three broad geothermal power plant technologies used to convert thermal energy in geothermal fluids to electricity: dry steam, flash, and binary or Organic Rankine Cycle (ORC). The type of conversion technology used depends on the state of the geothermal fluid (whether vapor or liquid) and its temperature.

Dry steam systems are best suited for producing electricity from vapor-dominated hydrothermal resources. Flash steam plants are appropriate when geothermal fluids are about 200C. Fluid is sprayed into a tank held at a much lower pressure than the fluid, causing some of the fluid to rapidly vaporize, or "flash." The vapor then drives a turbine, which drives a generator. If any liquid remains in the tank, it can be flashed again in a second tank to extract even more energy and thus increase conversion efficiency.

ORC, also called binary cycle, geothermal plants are the closest in thermodynamic principle to conventional Rankine cycle fossil power plant in that the heat transfer fluid undergoes an actual closed cycle. Hot geothermal fluid and a secondary (binary) fluid with a much lower boiling point than water pass through a heat exchanger. Heat from the geothermal fluid causes the secondary fluid to flash to vapor, which then drives the turbines. The binary fluid³ is condensed at the other side of turbine and returned to the evaporator by means of a feedpump. Since no steam condensate is formed, the entire geothermal fluid can be injected back into the reservoir thereby doing away with the need for external reservoir recharge.

Although binary power plants constitute 32% of all geothermal power plants in operation globally, they generate only 4% of total geothermal power indicating that the average power rating is quite low at 2.3 MW per unit (DiPippo 2008). DiPippo (2008) also indicates that binary cycle power plants are most appropriate for generating power from low temperature (enthalpy) geothermal resources.

³ Organic liquids such as n-butane (Normal boiling point NBP: -0.5C), isobutane (NBP: -11C) and toluene (NBP: 110C) are most commonly used binary fluids (Chandrasekharam and Bundschuh 2008)

The following table maps the preferred energy conversion systems to geothermal fluid temperature.

Table 3: Mapping of power plant technology to resource temperature

Geofluid temperature, °C	Options for Energy conversion system
100	Basic ORC cycle or Kalina cycle
150	Advanced ORC for e.g. regenerative ORC, ORC with IHE, regenerative ORC with IHE, Dual pressure ORC; Kalina Cycle
200	Advanced ORC as above; or Single Flash systems
250	Multiple-flash; Flash-Binary combined plants
300	Multiple-flash; Flash-Binary combined plants

Source: DiPippo (2008); Franco and Villani (2009); Tester, Anderson et al. (2006)

3.3. Cooling Technologies

In both flash and ORC power plants, the vapor exiting the turbine (steam for flash plants and binary fluid vapor for ORC plants) has to be condensed in a condenser. The following section briefly introduces the four types of cooling systems considered in our model.

In *wet re-circulating systems*, the vapor exiting the turbine is condensed in a shell-and-tube condenser. The vapor condenses on the shell side by transfer of heat to cooling water flowing through tubes in the condenser (NETL 2008). The warm cooling water is pumped from the steam condenser to a cooling tower, where the heat from the warm water is transferred to ambient air flowing through the cooling tower. In the process, a portion of the warm water evaporates from the cooling tower – this is the principal method of heat dissipation and is referred to as latent heat transfer. The cooled water is then recycled back to the condenser. Because of evaporative losses, a portion of the cooling water needs to be discharged from the system – known as blowdown – to prevent the buildup of minerals and sediment in the water. Make-up water is required to compensate for evaporative losses and blowdown.

Dry cooling systems use air for condensing the vapor and dissipating heat. The type of dry cooling system considered in our model is also referred as “direct” dry cooling system or alternatively air-cooled condenser (ACC) system. In such systems, the vapor exiting the turbine flows through tubes of an air-cooled condenser; heat is dissipated via conductive heat transfer to ambient air blown by fans across the outside surface of the tubes (NETL 2008). We do not consider “indirect” dry cooling system where the vapor is condensed by flow of cooling water across the outside surface of the tubes. Subsequently, a dry cooling tower is used to conductively transfer the heat from the water to the ambient air. As a result, there is no evaporative loss of cooling water with an indirect dry cooling system (NETL 2008). Such systems are usually considered for retrofitting of power plants with once through cooling system.

In addition to the above two cooling systems, we consider two other systems which reduce the energy penalty associated with dry cooling systems and high water consumption associated with wet re-circulating cooling systems. In *water augmented dry cooling systems* like inlet air cooling, water is introduced into the inlet air stream of the air-cooled condenser to lower the temperature of the ambient air leading to more effective cooling. In *hybrid cooling systems*, heat is rejected through two separate cooling systems — a dry system which carries the cooling load during most of the year and the wet system picking up a portion of the load during the hotter periods when the performance of the dry system is limited.

4. Summary of water requirements

4.1. I. Electricity from hydrothermal resource using ORC plant

Figure 2.1: Water required for cooling

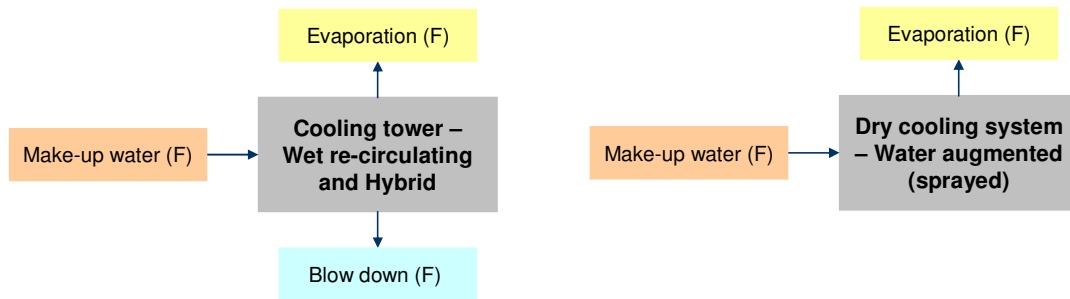
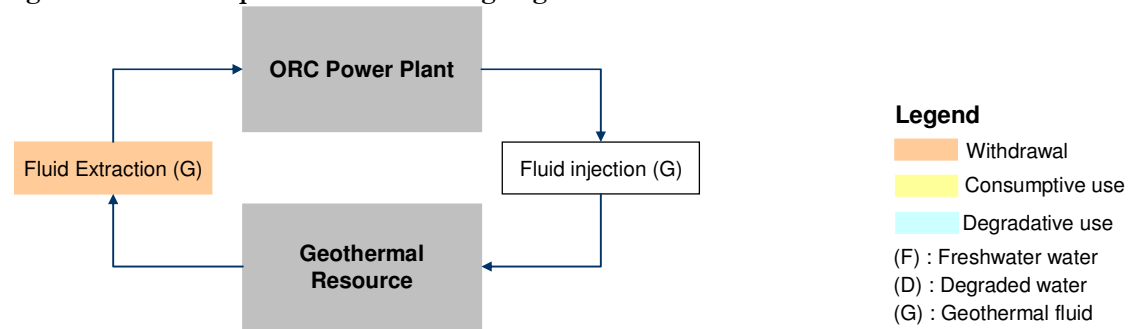


Figure 2.2: Water required for heat mining of geothermal resource



- Freshwater: Cooling water makeup accounts for freshwater withdrawal. Evaporation from cooling towers and blowdown account for freshwater consumptive and degradative use respectively. In water augmented dry cooling system, the entire water is assumed to evaporate and hence there is no blowdown.
- Degraded water: Degraded water is not required for electricity produced from hydrothermal resources using ORC plants.
- Geothermal Fluid: All fluid withdrawn for heat mining is injected back. No consumptive use of geothermal fluid

4.2. II. Electricity from EGS resource using ORC plant

Figure 3.1: Water required for cooling

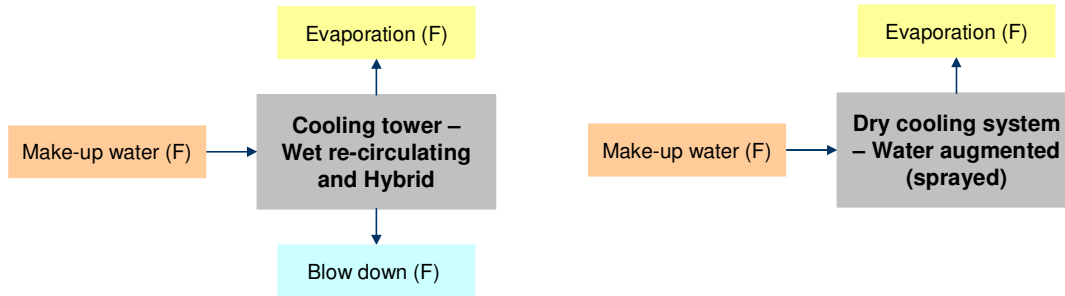
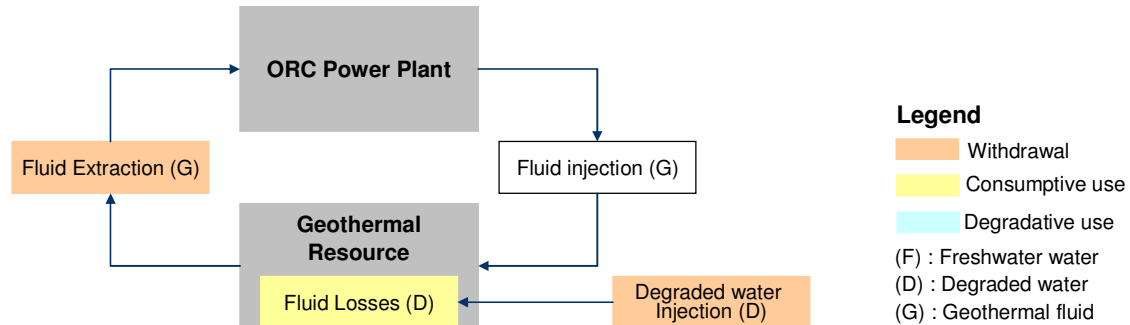


Figure 3.2: Water required for heat mining of geothermal resource



- Freshwater: Same as before.
- Degraded water: Degraded water is required to make up for fluid losses from the geothermal resource resulting from leakage from the fracture system to the surrounding rocks. We do not consider the initial injection of water to set up a loop for heat mining; only incremental water injections during the operational phase of the project are considered. The initial water injection requirement is likely to be small when it is normalized to the electricity generated over the lifetime of the project.
- Geothermal Fluid: All fluid withdrawn for heat mining is injected back. No consumptive use of geothermal fluid.

4.3. III. Electricity from hydrothermal resource using single flash power plant

Figure 4.1: Water required for cooling

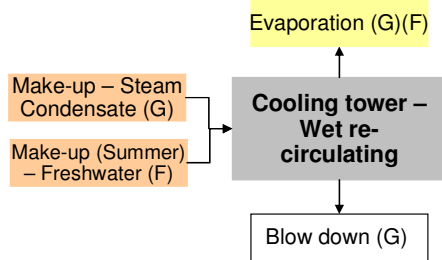
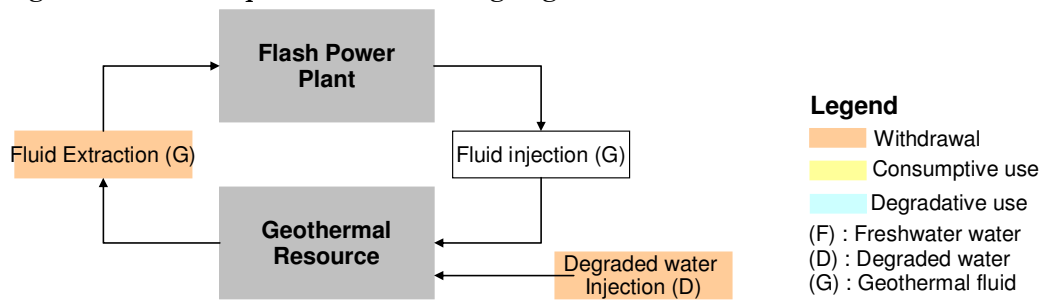


Figure 4.2: Water required for heat mining of geothermal resource



- **Geothermal Fluid:** Geothermal fluid is withdrawn for electricity generation. Steam condensate evaporated in the cooling tower is accounted for under geothermal fluid consumption. The liquid portion of the fluid after flashing and the cooling tower blowdown is injected back.
- **Degraded water:** Degraded water has to be sourced from external sources and injected to compensate for loss of geothermal fluid through evaporation of steam condensate in the cooling tower. This represents degraded water withdrawal and consumption.
- **Freshwater:** As mentioned before, most or nearly all the makeup water for cooling is provided by steam condensate. Additional water may be withdrawn during summer due to increase in water evaporation to dissipate heat to an environment with higher ambient temperature. If freshwater is withdrawn, then it is assumed that it is completely evaporated (consumed).

4.4. IV. Electricity from EGS resource using single flash power plant

Figure 5.1: Water required for cooling

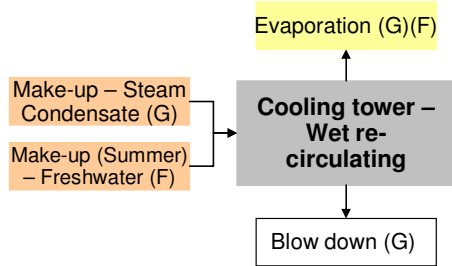
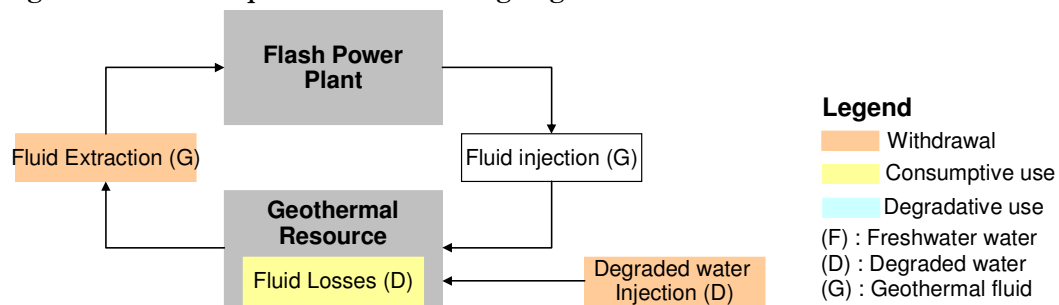


Figure 5.2: Water required for heat mining of geothermal resource



- Geothermal Fluid: Same as before.
- Degraded water: Degraded water is required to be injected not only to compensate for loss of geothermal fluid through evaporation of steam condensate in the cooling tower, but also to account for fluid losses from the geothermal resource resulting from leakage from the fracture system to the surrounding rocks. This represents degraded water withdrawal and consumption. We do not consider the initial injection of water to set up a loop for heat mining; only incremental water injections during the operational phase of the project are considered.
- Freshwater: Same as before.

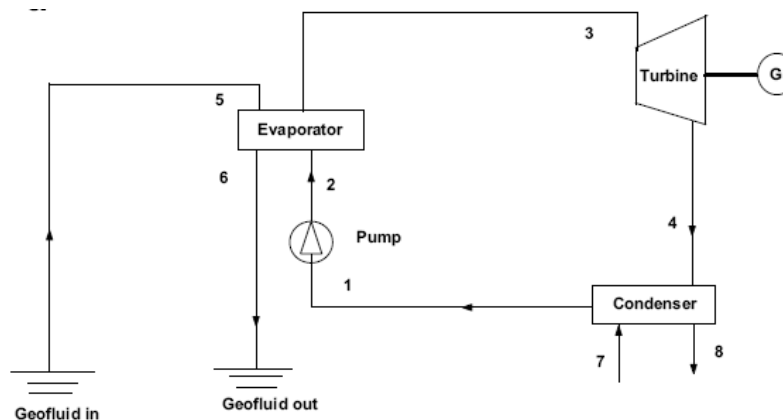
5. Water required by binary power plants

In this chapter, we study the water requirements of binary power plants. These include water requirements indicated in Figures 2.1 and 3.1.

5.1. Thermodynamics behind ORC

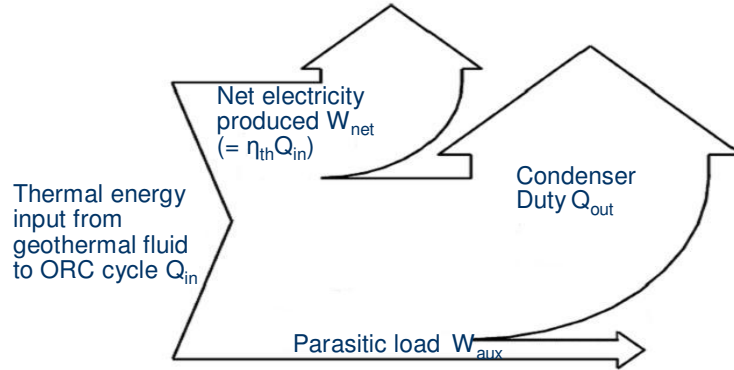
The following section describes the thermodynamics behind a basic ORC plant. Condenser duty is determined by first estimating the thermal input from the geothermal fluid, from which the gross power output of the plant (equal to the sum of net power output and parasitic power load) is subtracted to determine the heat load on the condenser and hence cooling tower. The quantity of water required to dissipate heat from a wet cooling tower system can be determined by taking into account the heat capacity and latent heat of vaporization of water.

Figure 6: Schematic diagram of a basic ORC cycle



Note: Figure reproduced with permission from Yari (2010)

Figure 7: Heat flow in a Binary cycle power plant



▪ **Thermal energy input**

The thermal input to the ORC cycle (or the heat transfer in the evaporator) can be calculated by the enthalpy drop for the geothermal fluid. This enthalpy loss for geothermal fluid is equal to the enthalpy gain by the working fluid (assuming no losses associated with the heat transfer process).

$$\dot{Q}_{in} = \dot{m}_{geo} (h_5 - h_6) = \dot{m}_{wf} (h_3 - h_2) \quad \text{(Equation 1)}$$

where,

\dot{Q}_{in} is the heat input in terms of kJ/s

\dot{m}_{geo} & \dot{m}_{wf} are the mass flow rates in kg/s of geothermal and working fluid respectively

h_5 & h_6 are enthalpies in kJ/kg of the geofluid entering and exiting the evaporator respectively

h_2 & h_1 are enthalpies in kJ/kg of the working fluid entering and exiting the evaporator respectively

The thermal input can also be represented as:

$$\dot{Q}_{in} = \dot{m}_{geo} C_p (T_{geo,in} - T_{geo,out}) \quad \text{(Equation 2)}$$

where,

C_p is the specific heat capacity in kJ/kg-°C of the geothermal fluid

$T_{geo,in}$ and $T_{geo,out}$ are the geothermal fluid temperatures in °C at the inlet and outlet of the evaporator respectively.

Frick, Kaltschmitt et al. (2010) suggested a range of 3.5 to 4.2 kJ/kg-°C for the specific heat capacity of geothermal fluid - lower and upper values correspond to a very high and a very low mineral content of the fluid. The model assumes a default value of 3.85 kJ/(kg °C) and can be changed by the user.

A lower outlet or rejection temperature of the geothermal fluid will increase the thermal energy extracted and power produced. However, the rejection temperature should be high enough to avoid silica oversaturation, which could lead to silica scaling and fouling problems in the evaporators, and in mineral deposition in pipes and valves. The model defaults to a value of 70 °C for $T_{geo,out}^1$ (Franco and Villani 2009). Silica deposition is not a universal problem - hence, this approach provides a conservative estimate.

▪ Power Output

The net power output is represented as

$$\dot{W}_{net} = \dot{W}_{gross} - \dot{W}_{aux,pl} - \dot{W}_{aux,res} \quad (\text{Equation 3})$$

where,

\dot{W}_{net} and \dot{W}_{gross} are the net & gross electricity produced by the plant respectively in kJ/s or kW

$\dot{W}_{aux,pl}$ is the total parasitic power consumed by various equipment within the plant - working fluid feed pump; circulating water pump and cooling tower fans for wet cooling tower; and ACC fans for dry cooling system in kJ/s or kW.

$\dot{W}_{aux,res}$ is the total parasitic power consumed at the reservoir - primarily geothermal fluid pump in kJ/s or kW

We define the first law net thermal efficiency as

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{gross} - \dot{W}_{aux,pl} - \dot{W}_{aux,res}}{\dot{Q}_{in}} \quad (\text{Equation 4})$$

Further detail regarding the thermal efficiency of an ORC plant will be provided later.

▪ Condenser Duty

The heat load on the condenser is calculated by the model as follows

$$\dot{Q}_{out} = \dot{Q}_{in} - \dot{W}_{net} - \dot{W}_{aux,pl} - \dot{W}_{aux,res} \quad (\text{Equation 5})$$

where,

\dot{Q}_{out} is the heat load on the condenser; and equivalently represents the heat that needs to be rejected by the cooling tower

In terms of enthalpy, the heat rejected by the condenser is given by:

$$\dot{Q}_{out} = \dot{m}_{geo} (h_4 - h_1) = \dot{m}_{coolant} (h_7 - h_8) \quad (\text{Equation 6})$$

\dot{Q}_{out} is the heat rejected by the condenser in kJ/s

$\dot{m}_{coolant}$ is the mass flow rate of either air or water used to cool the working fluid in kg/s

▪ Cooling Tower Analysis

Wet re-circulating cooling towers rely on the latent heat of water evaporation to exchange heat between the system and the air passing through the cooling tower. The rate of evaporation of water from the tower is related to the heat load on the tower, and is given by:

$$\dot{w}_{evp} = \dot{Q}_{out} \times f_{latent} / h_{vap} \quad (\text{Equation 7})$$

where,

\dot{w}_{evp} is the evaporation rate of water in kg/s (or liter/s assuming water density to be 1 kg/liter), and sometimes referred to as water consumption

f_{latent} is the fraction of total heat rejected by latent heat transfer i.e. evaporation of water

h_{vap} is the latent heat of vaporization of water and equal to 2,270 kJ/kg

f_{latent} depends largely upon ambient conditions and to some extent on design choice. The fraction can range from 0.65 to 0.9 (based on personal communication with Dr. John Maulbetsch, Maulbetsch Consulting May 2010). The fraction is higher for higher wet bulb temperatures. EPRI (2002) assumed a f_{latent} value of 0.9 to analyze cooling tower performance in California. Our model defaults to a value of 0.8. The remaining heat is dissipated through sensible heat i.e. increase in temperature of water which is a function of the specific heat capacity and mass flow rate of water.

Operation of wet cooling towers necessitate regular discharge of water (blowdown) from the cooling system in order to control the buildup of dissolved and suspended materials that concentrate in the system as a result of the evaporation cycles. Technical considerations may impose limits on the maximum allowable concentration of any particular chemical constituent either in the circulating water inside the tower or in the blowdown water; and this constituent will usually define the concentration limit for the cooling system. Regulatory criteria may also apply, e.g. limits are set on concentration of certain constituents such as copper or ammonia in the discharged water. EPRI (2003) has summarized limits of 21 key chemical constituents or constituent pairs. For each constituent of concern, cycles of concentration (N) can be calculated based on the constituents' concentration in intake or make-up water.

$$N_i = \frac{C_{limit,i}}{C_{MU,i}} \quad (\text{Equation 8})$$

where,

N_i is the cycles of concentration applicable for constituent i

$C_{Limit,i}$ is the water quality limit for constituent i in mg/liter

$C_{MU,i}$ is the concentration of constituent i in the make-up water in mg/liter

Thus, N_i needs to be calculated for each of the constituents of concern, and the constituent with the lowest calculated N_i value will be the limiting parameter for that source of water. This value of N will be the maximum cycles of concentration achievable. In addition to defining the maximum concentration for a limiting chemical constituent, N is also used to determine the blowdown rate, and consequently the make-up rate.

$$w_{BD} = \frac{w_{evp}}{(N - 1)}, \quad \text{(Equation 9)}$$

and

$$w_{MU} = w_{evp} + w_{BD} \quad \text{(Equation 10)}$$

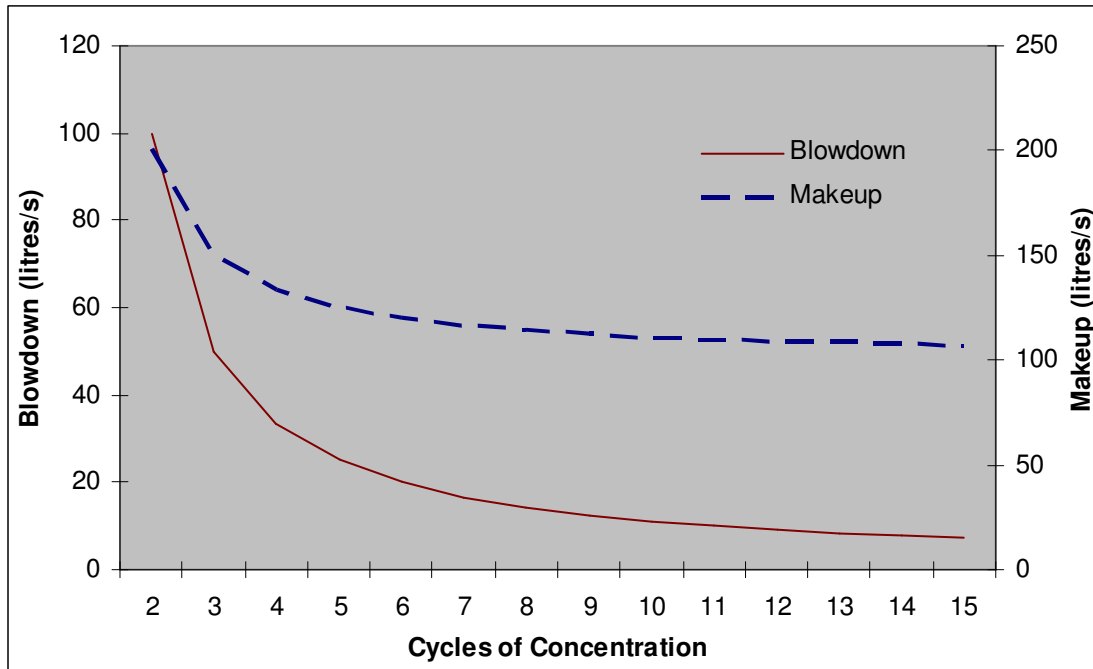
where,

w_{BD} is the blowdown rate in kg/s (or liter/s)

w_{MU} is the make-up rate in kg/s (or liter/s) and sometimes referred to as the withdrawal rate

The above equations indicate that smaller the value of N , the larger will be the blowdown rate. The following graph shows the impact of cycles of concentration on makeup and blowdown. It assumes an evaporation rate of 100 liters/second.

Figure 8: Impact of cycles of concentration (N) on cooling tower make-up & blowdown



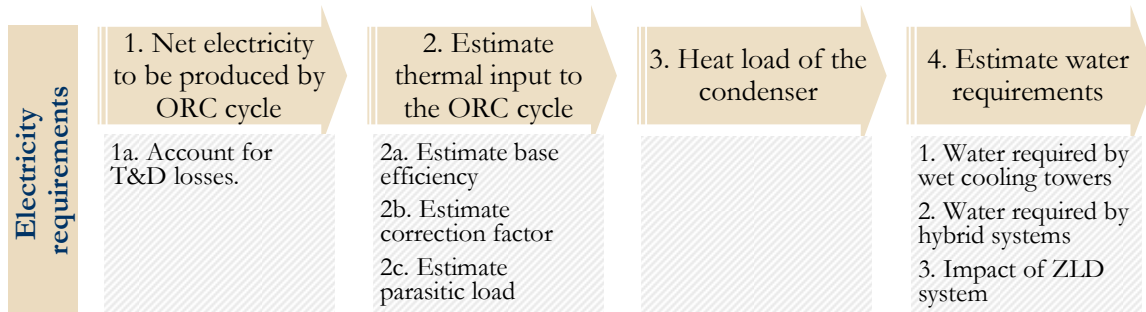
The graph indicates that below 5-6 cycles of concentration, blowdown rates increase dramatically. Water treatment systems allow plants to de-mineralize water so that higher cycles of concentration could be adopted and waste water blowdown and freshwater makeup rates reduced.

The model assumes freshwater supply and defaults to N=8.

5.2. Methodology

The model follows a four-step process to determine water requirements of an ORC plant.

Figure 9: Methodology to determine water requirements of ORC cycle



▪ Step 1

The starting point of the model is the electricity consumed by the end customer, which in turn gives the electricity required to be produced by the ORC plant after accounting for transmission and distribution losses.

$$\dot{W}_{net} = \dot{W}_{consumed} (1 + l_{T\&D}) \quad (\text{Equation 11})$$

where,

$\dot{W}_{consumed}$ is the electricity in kJ/s or kW required by end consumer
 $l_{T\&D}$ represents transmission and distribution losses.

▪ Step 2

To estimate the required thermal input from geothermal field, the net efficiency of the ORC plant is estimated in steps (2a) and (2b). In (2a), we first estimate the “base” net thermal efficiency level of a particular ORC plant configuration (e.g. ORC with Internal Heat Exchanger and wet cooling towers) based on a linear relationship between temperature of the geothermal fluid and thermal efficiency. The particular ORC configuration was chosen because the relationship between efficiency and temperature of geothermal fluid for this configuration was available in the literature.

To account for more advanced ORC configurations or different cooling systems, correct factors could be used.

$$\eta_{th} = \eta_{th,base} (1 + cf) \quad (\text{Equation 12})$$

where

η_{th} is the net efficiency of the ORC configuration being examined
 $\eta_{th,base}$ is the net efficiency of base ORC configuration for the given inlet geothermal fluid temperature
 cf is the correct factor

In Step (2c) we determine parasitic load. The total thermal input required by the ORC plant given $\dot{W}_{consumed}$ is:

$$\dot{Q}_{in} = \frac{\dot{W}_{net}}{\eta_{th}} \quad (\text{Equation 13})$$

In Step (2c) we determine parasitic load which is the sum of parasitic load at the plant and at the geothermal field.

▪ Step 3

The heat load on the condenser is given by equation 5 which can be rewritten based on above equation as

$$\dot{Q}_{out} = \frac{(1 - \eta_{th})}{\eta_{th}} \dot{W}_{net} - \dot{W}_{aux} \quad (\text{Equation 14})$$

▪ **Step 4**

The water consumption for a wet re-circulating cooling system can be estimated based on equations 7 to 10. Hybrid cooling systems will consume a certain percentage of the water consumed by wet re-circulating cooling systems as detailed in later sections.

5.2.1. Thermal Efficiency of OC plants

▪ **Ideal thermal efficiency for geothermal binary plants**

DiPippo (2007) argues that the ideal thermodynamic cycle appropriate for binary power plants is the triangular (or trilateral) cycle. The triangular efficiency imposes a lower upper-bound on the thermal efficiency in comparison to the Carnot cycle which is used to define the upper bound of traditional Rankine cycle power plants fueled by coal and nuclear. The efficiency of the ideal Carnot cycle can be expressed as⁴:

$$\eta_{th}^C = \frac{T_H - T_L}{T_H} \quad (\text{Equation 15})$$

where,

η_{th}^C is the thermal efficiency of the ideal Carnot cycle

T_H is absolute temperature in Kelvin of the heat source. In this case, it is the temperature of the inlet geothermal fluid

T_L is absolute temperature in Kelvin of the heat sink (condensing temperature)

DiPippo (2007) indicates that since the heating medium in a geothermal binary plant is not an isothermal source, but rather a fluid that cools as it transfers heat to the cycle working fluid, the Carnot cycle is not appropriate. Rather, the triangular cycle with the following thermal efficiency is more applicable:

$$\eta_{th}^{TRI} = \frac{T_H - T_L}{T_H + T_L} \quad (\text{Equation 16})$$

where,

η_{th}^{TRI} is the thermal efficiency of the triangular cycle

If the geothermal fluid could be cooled down to the dead-state temperature, T_0 , thus allowing all of the waste heat to be discharged at the lowest possible temperature, then the maximum thermal efficiency is defined as

⁴ In equations 15, 16 and 17, the temperatures are in Kelvin. In all other equations pertaining to the ORC cycle, the temperatures are in °C.

$$\eta_{th,max}^{TRI} = \frac{T_H - T_0}{T_H + T_0} \quad (\text{Equation } 17)$$

where,

$\eta_{th,max}^{TRI}$ is the thermal efficiency of the triangular cycle
 T_0 is the dead state or ambient temperature in Kelvin

DiPippo (2007) analyzed six ORCs, and finds that on average the thermal efficiency is around 55% of the maximum triangular cycle thermal efficiency $\eta_{th,max}^{TRI}$.

- **Base-case thermal efficiency of ORCs**

Heberle and Brüggemann (2010) simulated the impact of inlet geothermal fluid temperature, and selection of working fluid on the thermal efficiency of ORCs. They configured the ORC to include an internal heat exchanger⁵ (IHE) and wet cooling system; and made parametric assumptions regarding pinch point temperature (5 °C), turbine and feed pump isentropic efficiency (0.75) and cooling water temperature (15 °C). The following equations summarize the linear relationship between the geothermal fluid's inlet temperature and thermal efficiency:

$$100\eta_{th} = 0.0925T_{geo, in} - 2.0178 \quad (\text{fluid: iso-pentane}) \quad (\text{Equation 18})$$

$$100\eta_{th} = 0.1002T_{geo, in} - 2.88 \quad (\text{fluid: R245fa}) \quad (\text{Equation 19})$$

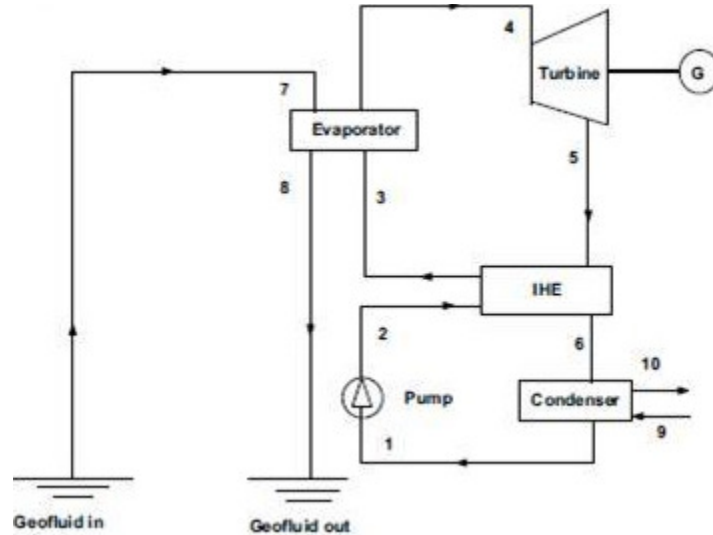
where,

η_{th} is the thermal efficiency of the ORC

$T_{geo, in}$ is the inlet geothermal fluid temperature in °C

The relationship is shown to hold for geothermal fluid inlet temperatures ranging from around 75 to 180 °C.

Figure 10: Schematic diagram of an ORC cycle with IHE (recuperated cycle)



Note: Figure reproduced with permission from Yari (2010)

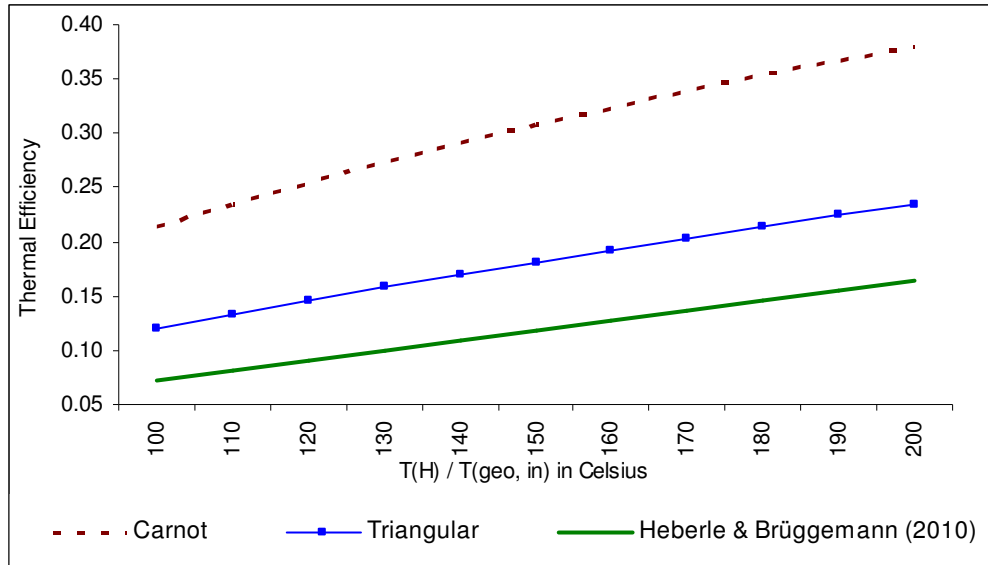
The MIT study (Tester, Anderson et al. 2006) carried out a statistical analysis based on data from ten ORCs across the world, and came up with the following relationship

$$100\eta_{th} = 0.0935T_{geo, in} - 2.3266 \quad (\text{Equation 20})$$

⁵ Impact of an internal heat exchanger on efficiency relative to a basic ORC will be discussed later. Such a configuration is also referred to as “recuperated cycle”. However, we will use the term “ORC with IHE” to distinguish it from the “regenerative ORC” which is defined later.

The dataset included plants with inlet fluid temperatures ranging from 103 to 166 °C. It includes ORCs with both wet and dry cooling; and both basic and advanced ORC configurations – for example the Húsavík plant in Iceland is a Kalina cycle ORC

Figure 11: Comparison of thermal efficiencies of Carnot and Triangular cycles



Assumptions: (i) Heat sink or condensing temperature (T_L) is equal to the dead state or ambient temperature (T_0) of 20 °C

For our analysis, we have adopted the thermal efficiency estimates for iso-pentane by Heberle and Brüggemann (2010). These estimates give us a base case scenario – impact of dry cooling and advanced ORC configurations like Kalina cycle or regenerative ORC will be included in the correction factors discussed in the next section. We considered iso-pentane instead of R245fa because organic liquids are the most commonly used working fluids (Chandrasekharam and Bundschuh 2008).

- **Impact on efficiency – ORC configurations**

Yari (2010) analyzed and compared four different ORC configurations – the basic ORC, ORC with an Internal Heat Exchanger (IHE), regenerative ORC, and finally regenerative ORC with an IHE⁶.

Both the IHE and regenerative process aim to increase the temperature of the working fluid on its way from the condenser to the boiler. This increases the average temperature at which heat is added to the working fluid from the geothermal fluid in the evaporator (boiler); and thus increases the thermal efficiency.

The IHE relies on the fact that the expansion in the turbine ends for most organic fluids not in the wet steam regime (as in traditional Rankine cycle), but in the gas phase above the condenser temperature. An IHE is used to capture remaining enthalpy and thus improve efficiency. In regenerative ORC, a “bleed” vapor is extracted from a suitable intermediate point in the turbine to pre-heat the working fluid on its way from for the condenser to the boiler. Based on the schematic diagrams, it can be inferred that an open working fluid heater system has been modeled where the extracted vapor and the condensate are physically mixed.

Yari (2010) assumed an inlet geothermal fluid temperature of 180 °C; turbine and feed pump isentropic efficiencies of 0.8 and 0.9 respectively; and air cooled condensers. Efficiencies were calculated for three different working fluids - R113, R123, and n-Pentane.

Table 4: Thermal efficiencies of various ORC configurations

	Basic ORC	ORC with IHE (recuperator cycles)	Regenerative (bleed feed heaters) ORC	Regenerative ORC with IHE
R113	0.131	0.145	0.143	0.153
R123	0.133	0.142	0.145	0.154
n-Pentane	0.126	0.141	0.141	0.150
Average	0.130	0.143	0.143	0.152

Source: Based on Yari (2010)

From the above results, our model defaults to 8.8% fall in efficiency of a basic ORC relative to an ORC with IHE, and 0.5% and 6.9% rise in efficiency of Regenerative ORC and Regenerative ORC with IHE respectively relative to an ORC with IHE configuration.

Kalina Cycles are those binary cycle plants where a mixture of ammonia and water is used as the working fluid. Chandrasekharam and Bundschuh (2008) indicate that these plants have 20-40% higher efficiency than ORC largely because of more efficient heat transfer from

⁶ As mentioned before, the ORC with IHE are also referred to as the “recuperated cycle” or “ORC with a recuperator”. Such configurations are fairly common. However, no binary geothermal plants use the regenerative cycles (bleed feed heaters); they are impractical given the small size of the turbines in geobinary plants (based on personal communications with Dr. Ron DiPippo, August 2010).

geothermal fluid to the secondary fluid. However, DiPippo (2004) believes such conclusions are based on insufficient data; superior performance of Húsavík Kalina plant in Iceland is largely due to use of a very cold Icelandic stream (5 °C) in a once-through cooling arrangement. He concludes that the difference in performance is about 3% in favor of a Kalina cycle. In absence of any further literature on the subject, our model defaults to a 10% difference in performance in favor of Kalina cycle relative to ORC with IHE configuration.

A dual pressure ORC cycle has a two-stage heating/boiling process – this helps to reduce the average temperature difference between the hotter geothermal fluid and cooler working fluid and thus increases the thermodynamic efficiency. Franco and Villani (2009) analyzed thermal efficiencies for six different working fluids and compared thermal efficiencies of various advanced configurations with basic ORC configuration. We took those scenarios where the optimized solution was the dual pressure ORC cycle and compared the average increase in efficiency relative to that of a basic ORC. The average efficiency difference was 18% in favor of dual pressure ORC. Given that ORC with IHE is 8.8% more efficient than basic ORC (Table 4 above, based on Yari 2010); Franco and Villani’s (2009) result translates to a 8.2% efficiency difference between ORC with IHE and dual pressure ORC in favor of the later.

Our model defaults to the following differences in thermal efficiencies relative to the base case scenario of ORC with IHE

Table 5: Thermal efficiencies of ORC configurations relative to ORC with IHE

ORC Configuration	Change in thermal efficiency relative to an ORC with IHE	Based on
Basic ORC (single pressure)	-8.8%	Yari (2010)
Regenerative ORC (single pressure)	0.5%	Yari (2010)
Regenerative ORC with IHE (single pressure)	6.9%	Yari (2010)
Dual pressure ORC	8.8%	Franco and Villani (2009), Yari (2010)
Kalina Cycle (single pressure)	10.0%	Chandrasekharam and Bundschuh (2008), DiPippo (2004)

Franco and Villani (2009) note that the advantages related to the use of complex technical solutions (e.g. dual-pressure level cycles or regenerative cycles) may be important for high geothermal fluid inlet temperature (140–160 °C). At lower inlet temperatures (120–130 °C), efficiency gains are negligible. They also indicate that advanced ORC configurations are sensitive to variations in operating conditions (e.g. a decrease in geothermal fluid inlet temperature during the lifecycle of the plant); and hence not always desirable.

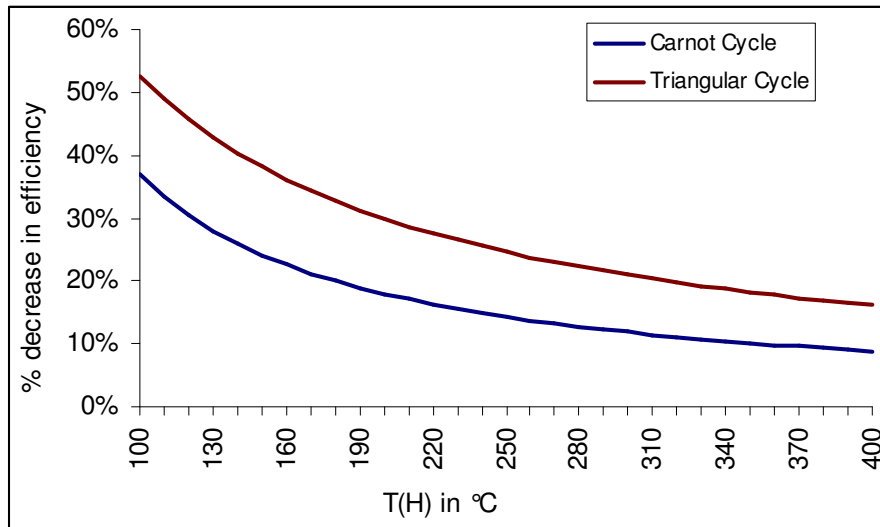
For simplicity, our model defaults to differences in efficiencies summarized in the above table over the entire range of inlet geothermal fluid temperature. Users should choose advanced configurations only for high inlet temperatures.

5.2.2. Impact on efficiency – cooling system

The model considers three types of cooling systems: wet re-circulating cooling towers, dry cooling systems, and hybrid cooling systems. We also consider the impact of Zero Liquid Discharge (ZLD) systems on volume of water withdrawals. We do not consider once-through cooling (OTC) systems – US EPA's Section 316(b) strongly discourages OTC systems and NETL (2008) expects that most new power plants will have to use closed-loop, re-circulating systems or dry air-cooled systems.

Geothermal plants are sensitive to temperature of the heat sink. This is evident if one considers Carnot cycle efficiency (Equation 15). If the heat source temperature is 500 °C, as is the case with thermoelectric Rankine cycle power plants, an increase in heat sink temperature from 5 °C (around 40 °F) to 40 °C (around 105 °F) will reduce the Carnot efficiency by around 9%. However, for a heat source temperature of 200 °C, a similar increase in heat sink temperature decreases efficiency by 18%. For a low enthalpy geothermal resource at 100 °C, the fall in efficiency is as high as 37%.

Figure 12: Decrease in efficiency if heat sink temperature increases from 5 to 40°C



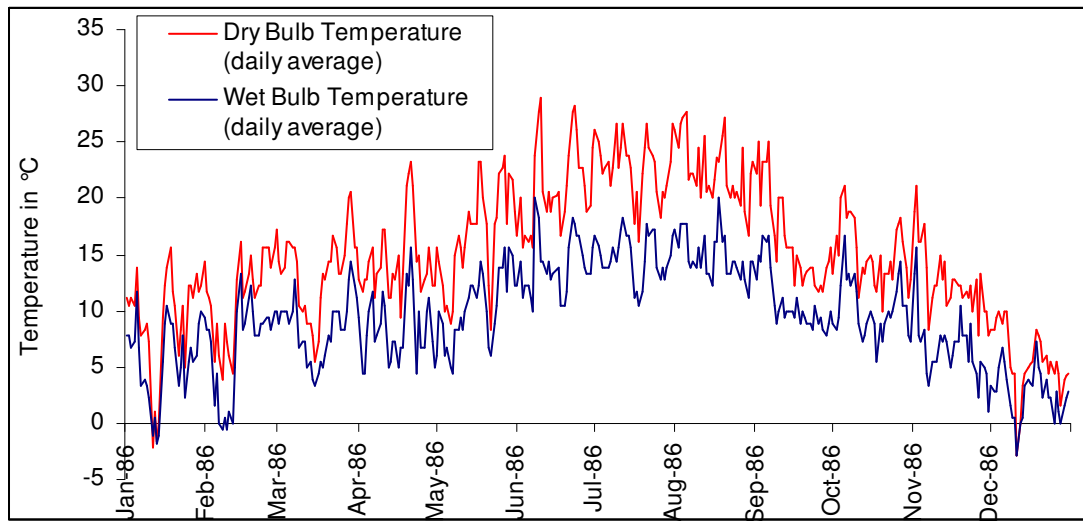
As the above graph shows, Triangular cycles are even more sensitive to the temperature of the heat sink. In case of geothermal resources, efficiency is more sensitive to heat sink temperature (T_L) than to the heat source temperature (T_H) which is the geothermal resource temperature $T_{geo, in}$.

5.2.2.1. Dry Cooling

Dry cooling systems cool by transferring heat to the ambient air without evaporation (sensible cooling). As a result they can cool the working fluid vapor only to a temperature that approaches the dry-bulb temperature or the ambient air temperature. On the other hand, wet re-circulating cooling systems cool primarily through evaporation; thus the temperature of the cooled working fluid approaches wet-bulb temperature.

Wet-bulb temperature is always lower than dry bulb or ambient air temperature except at 100% humidity. Further, the dry bulb temperature is more variable. The following graph shows the daily average wet and dry bulb temperatures measured at Brownsville, Nevada County in California over an entire year. The graph indicates that the difference between the two temperatures is higher in summer and that the dry bulb temperature is more variable.

Figure 13: Daily average wet and dry bulb temperature at Brownsville, California.



Source: UC IPM (2010)

As a result, the following may be stated about the difference in performance between a wet cooled and dry cooled ORC plant:

- *Lower efficiency of dry cooled plants*

The performance of a power plant is directly related to the pressure drop across a turbine; which in turn is related to the performance of the cooling system. As a result, dry cooling systems lead to a greater reduction in power plant's net energy production than a wet re-circulating system. In other words, efficiency of power plants with wet re-circulating systems is always higher.

Since the parasitic energy consumption of both wet and dry cooling systems are the same (EC 2001; Maulbetsch and DiFilippo 2006), the efficiency penalty can be attributable entirely to lower plant operating efficiency resulting from lesser cooling.

- *Higher variability in operations of dry cooled plants*

Power output for an air cooled geothermal plant can decrease by up to 50% from winter to summer (Michaelides and Ryder 1992; Kanoglu and Cengel 1999). However, air-cooled thermoelectric plants suffer a much lower drop in performance in summer (DOE 2002; Maulbetsch and DiFilippo 2006); usually around 10% or less. Diurnal fluctuations may also be significant; DiPippo (2004) reports that power output of the air cooled bottoming binary cycle in Brady was 33% lower at 6PM than at 6AM in the morning, based on observations

Observed relationship between power output and heat sink temperature

For a 27MW dry-cooled binary power plant in Reno, Northern Nevada, Konaglu (1999) found that the gross power output was related to the ambient dry bulb temperature by the following polynomial equation:

$$\dot{W}_{gross} = 22,089.20 - 119.7841T - 4.171015T^2$$

where \dot{W}_{gross} is the gross power output measured in kW
 T is the ambient air temperature measured in °C.

We analyzed the relationship between gross power output and first-law thermal efficiency of the bottoming binary cycle of the Brady power plant in Nevada based on observations made by DiPippo (2004) during August and September 2002. The following relationships were established:

$$\dot{W}_{gross} = 7,384.55 - 122.66T, \text{ and}$$

$$\eta_{th} = 0.14163 - 0.0028T$$

The above relationships confirm the negative relationship between power output and ambient temperature.

over 10 days in September 2002. Average ambient temperatures were 30.1 °C and 16.8 °C at 6PM and 6AM respectively.

To estimate an annual average energy penalty of implementing a dry cooling system relative to a wet re-circulating cooling tower, we adopted the following three steps:

1. **Establish an upper lower-bound** by estimating the energy penalty for thermoelectric and solar thermal power plants

The following table summarizes the potential efficiency penalty (equal to energy penalty) of a power plant with dry cooled systems relative to one with wet re-circulating cooling system.

Table 6: Efficiency penalty of dry cooling systems relative to wet re-circulating cooling systems

Source	Plant details	Site condition & location	Efficiency penalty-(1)(2)	Notes
WorleyParsons (2008)	250MW parabolic trough solar Rankine Cycle	Hot & arid (Mojave Desert)	5.90%	Study was undertaken for Beacon Solar Energy Project
Maulbetsch and DiFilippo (2006)	500 MW NG Combined Cycle	Hot & arid (Desert; Riverside, CA)	9.12%	Change in gross efficiency of Rankine cycle only (design). Change in overall plant gross efficiency is 2.98%
	350 MW Brayton	Hot & humid (Valley;	4.81%	Change in overall plant gross

	cycle (gas turbine)	Bakersfield CA)		efficiency is 1.20% (design)
	& 150 MW Rankine cycle (steam turbine)	Cool & humid (Coast; San Francisco, CA)	1.80%	Change in overall plant gross efficiency is 0.3% (design)
		Variable (Mountain; Redding, CA)	8.58%	Change in overall plant gross efficiency is 1.9% (design)
DOE (2008)	274MW parabolic trough solar (Rankine)	Hot & arid (Desert)	4.90%	
DOE (2002)	400 MW supercritical coal plant	Delaware River Basin	3.57% (6.53%)	Retrofit of a once-through plant to either wet- or a (indirect) dry-cooling tower
		Michigan / Great Lakes	3.29% (7.20%)	
		Ohio River Valley	3.39% (6.60%)	
		South (Georgia)	4.41% (7.30%)	
		Southwest	6.96% (10.45%)	

(1) Annual average efficiency penalty

(2) Figures in brackets gives efficiency penalty during summer - One percent highest temperature conditions

The DOE study (2002) considered indirect dry cooling systems instead of direct dry cooling (also referred to as Air cooled Condensers or ACC). In ACC, the heat from the working fluid vapor, after exiting the turbine, is directly transferred to the surrounding. In an indirect dry cooling system, the working fluid vapor condenses in the surface condenser, which utilizes a secondary cooling water loop to reject the heat from the cooling water to the ambient air via the cooling towers. Indirect dry cooling was considered by DOE (2002) because the scope included retrofitting of existing coal plants with OTC. GAO (2009) indicates that the DOE (2002) energy penalty estimates are higher than when a dry cooled system is designed according to the unique specifications of a newly built plant.

Maulbetsch and DiFilippo (2006) compared the impacts of dry cooling and wet re-circulating cooling systems on energy efficiency of a combined cycle power plant. In the table above, we have summarized the impact on the Rankine cycle only. It should be noted that power plants' designs have been optimized in the study for total cost or total levelized cost of electricity and not for lowest water use or most efficient power production. Thus, for example, a dry cooled plant could be designed to operate at a higher design ambient temperature and equipped with a larger air cooled condenser (ACC). This will reduce the energy penalty and improve hot day capacity, but might lead to higher capital and operating costs and lower financial viability. The implication of this is that energy penalties of dry cooling system have been over estimated in this study.

2. Estimate **change in ideal Triangular cycle efficiency** when the heat sink temperature (T_1) changes from wet bulb to dry bulb ambient temperature.

Kutscher (2002) provides the dry bulb (DB) temperature and wet bulb (WB) temperature for each hour of a typical day of each month at Empire, NV (hot and dry climate). Based on this data, we calculated the monthly and annual average penalty of dry cooling relative to wet

cooling for a range of geothermal resource inlet temperatures. For the calculations we assumed an approach temperature of 10 °F.

Table 7: Change in ideal Triangular cycle efficiency when T_L changes from WB to DB temperature – monthly and annual average

Month	Energy Penalty			
	$T(H) = 100^\circ\text{C}$	$T(H) = 125^\circ\text{C}$	$T(H) = 150^\circ\text{C}$	$T(H) = 175^\circ\text{C}$
January	6.3%	5.2%	4.3%	3.9%
February	4.9%	4.0%	3.3%	3.0%
March	5.6%	4.5%	3.7%	3.3%
April	9.2%	7.4%	6.1%	5.4%
May	12.2%	9.7%	8.0%	7.0%
June	14.5%	11.5%	9.4%	8.3%
July	15.2%	12.0%	9.8%	8.6%
August	15.2%	12.0%	9.8%	8.6%
September	13.5%	10.7%	8.8%	7.7%
October	9.8%	7.8%	6.4%	5.7%
November	7.9%	6.4%	5.3%	4.7%
December	6.2%	5.1%	4.2%	3.8%
Average	10.1%	8.1%	6.6%	5.9%

Notes: (1) Based on weather data from Kutscher (2002)

EPRI (2004) provide typical distribution of WB and DB temperature over a year based on data from the National Climate Data Centre, National Oceanic and Atmospheric Administration. The data are based on observations made between the years 1972 and 1996 and are available for four stations. As before, we assumed an approach temperature of 10 °F.

Table 8: Change in ideal Triangular cycle efficiency when T_L changes from WB to DB temperature – annual average for different locations

City	Climate Conditions	Energy Penalty			
		$T(H) = 100^\circ\text{C}$	$T(H) = 125^\circ\text{C}$	$T(H) = 150^\circ\text{C}$	$T(H) = 175^\circ\text{C}$
El Paso, TX	Hot & arid (desert)	10.4%	8.3%	6.9%	5.8%
Jacksonville, FL	Hot & humid	4.0%	3.1%	2.6%	2.2%
Portland, OR	Moderate	7.3%	5.8%	4.8%	4.2%
Bismark, ND	Extreme summers & winters	6.2%	5.0%	4.3%	3.7%

Notes: (1) Based on weather data from EPRI (2004)

The above table indicates that energy penalty varies between sites. In Jacksonville, FL, the difference between WB and DB is small due to high relative humidity. Alternatively, in El Paso, TX, high ambient temperatures and low relative humidity in summer and in the afternoons makes use of evaporative cooling quite attractive.

3. Literature review

There is limited literature on energy penalty associated with dry cooling system for a geothermal binary ORC plant. Two studies reviewed in this section compare dry cooling with water augmented dry cooling – a system that involves spraying a small amount of water into the inlet air stream of an air-cooled condenser (ACC) where it evaporates and cools the air. Spray enhancement are particularly effective when ambient temperatures are greater than 90° F and relative humidity is less than 40 percent (EPRI 2003).

Kutscher (2002) have developed a model to analyze the impact of water augmented dry cooling on performance of a 1 MW dry-cooled ORC plant at Empire, NV (hot and dry desert climate) with an geothermal fluid inlet temperature of around 120 °C. The model results may be used to compare the annual average energy penalty of a dry cooling system relative to four different types of water augmented dry cooling systems. It should be noted that the energy penalty is not with respect to a wet re-circulating cooling system but with respect to various “water enhanced air cooled systems”.

Table 9: Comparative performance of various water enhanced spray cooling system

Type of cooling system	Energy penalty of dry cooling relative to evaporative cooling
Dry cooling	
Spray cooling	11.4%
Munters cooling	2.9%
Deluge cooling	15.3%
Hybrid cooling	7.9%

Source: based on Kutscher (2002).

Notes: Figures in brackets indicate water requirements as a percentage of water required by a wet re-circulating cooling system as calculated by our model.

Imroz Sohel, Sellier et al. (2009) simulated the impact of retrofitting spray cooling at the 35 MW Rotokawa ORC with dry cooling system in Taupo, New Zealand. The authors estimate a 1% increase in power generated over the entire year and 6.8% during summer. The paper does not calculate the amount of water requirements; enough water is used to get 80% of the potential evaporative cooling effect (saturation).

The difference in estimated annual average performance of spray cooling by Kutscher (2002) and Imroz Sohel, Sellier et al. (2009) is partly explained by the differences in climatic conditions – peak summer DB temperatures are around 28 °C and 35 °C at Taupo and Empire respectively, and differences between WB and DB could touch 15 °C at Empire while they are below 10 °C at Taupo. It is also partly explained by differences in assumed saturation level of outgoing air under evaporative cooling – 92% assumed for the Empire study and 80% for the Taupo study.

The above discussion indicates that the user has to consider two key factors while selecting the annual average energy penalty of dry cooling system relative to wet re-circulating cooling system. The system defaults to a level of 10%.

5.2.2.2. Hybrid cooling systems

Many different solutions are available to mitigate the energy penalty imposed by dry cooling systems during the hotter periods. In hybrid or wet/dry systems, heat is rejected through two separate cooling systems—a dry system which carries the cooling load during most of the year and the wet system picking up a portion of the load during the hotter periods when the performance of the dry system is limited. While these systems can achieve significant water conservation and still maintain good hot-day performance, their implementation is limited due to high initial costs as a result of the need for two cooling towers (or a more complex integrated single structure), parallel circulating water loop components, more complex controls, and other requirements associated with providing two, nearly independent cooling systems EPRI (2003). In water augmented dry cooling systems like inlet air cooling, water is introduced into the inlet air stream of the air-cooled condenser to bring the temperature of ambient air closer to the wet bulb temperature.

Maulbetsch and DiFilippo (2006) indicate that most systems in the United States are intended for plume abatement and are essentially all wet systems with a small amount of dry cooling to heat the tower exhaust plume above saturation conditions during cold, high humidity periods when the wet tower plume is likely to be visible and risks causing winter icing in nearby roads.

DOE (2008) simulated and compared the performance of various cooling systems for a 274 MW gross Rankine cycle solar power plant located in a typical Southwest desert site (hot and dry). They modeled a hybrid system that uses an air cooled condenser in parallel with a wet cooling tower. The wet system was put into service during high ambient temperatures when a portion of the steam leaving the turbine is routed to a wet cooling system which is only rejecting a portion of the total waste heat. The simulation of the different hybrid systems were undertaken with a target condenser pressure (turbine backpressure) as indicated in column 2 of the table below.

Table 10: Design and performance of Hybrid cooling systems

Cooling System (1)	Target Condenser / Turbine Back Pressure in KPa (inches HgA)	Power (net efficiency) penalty relative to plant with wet re-circulating cooling system (2)	Water consumption - fraction of wet cooling tower
Hybrid 1 (116%)	8.47 (2.5)	0.8%	44.6%
Hybrid 2 (71%)	13.55 (4.0)	2.2%	13.3%
Hybrid 3 (40%)	20.32 (6.0)	3.6%	0.9%
Hybrid 4 (13%)	27.09 (8.0)	3.9%	0.1%
Dry Cooling		5.7%	0.0%

(1) Wet cooling tower design capacity (condenser duty) as a percentage of the capacity of the tower in wet cooling system (224 MWt)

(2) Annual average

Based on DOE (2008)

The model adjusts the energy penalty for each of the four hybrid systems based on energy penalty assumed for dry cooling tower system.

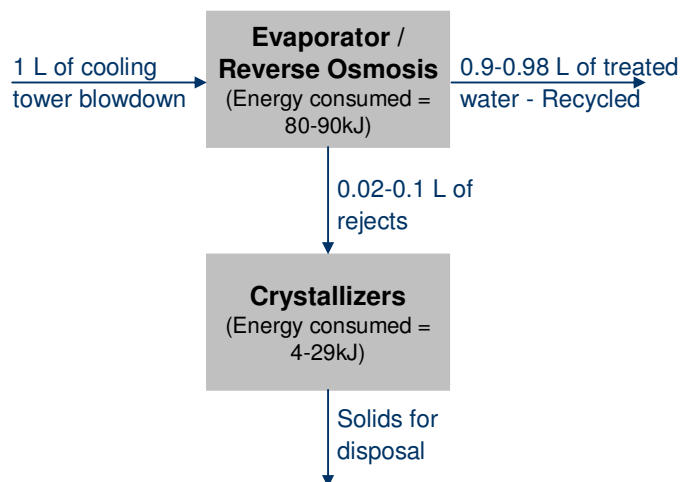
5.2.2.3. ZLD systems

Traditionally power plants have discharged their waste water to a surface water source which is economically very attractive. Weiss (1996) has identified multiple factors that determine the ability of a power plant to discharge to a surface water source - quality and quantity of the waste water, waste water treatment concept applied, the quality and quantity (flow or volume) of the receiving water body, and the environmental sensitivity of the receiving water body. Power plants have to treat waste water to meet various federal, state or local norms before discharge. The waste water may also be sent to the sanitary sewer / municipality treatment plants for treatment and disposal. Other options available to plants are ground water discharge including evaporation ponds and trucking waste to offsite disposal sites. None of these options offer the potential for reuse of water.

Zero Liquid Discharge systems based on mechanical evaporators or reverse osmosis membranes allow power plants to recover some of the waste water. Such systems have historically been applied in areas that are deficient in water supply, remote from suitable receiving streams for wastewater discharge, and/or at projects seeking to streamline their licensing schedule. ZLD systems usually consist of two components: a mechanical evaporator or reverse osmosis membrane that concentrates wastewater into a liquid stream containing concentrated constituents of the wastewater being treated, and a crystallizer to further concentrate wastewater to solids which can then be shipped offsite. Maulbetsch and DiFillipo (2006) indicate that these processes produce high quality water - less than 100-500 mg/L TDS for reverse osmosis and 10mg/L TDS for a mechanical evaporator. 90-98% of the waste water could be recovered depending upon concentration in the input stream.

However such systems consume significant parasitic load – evaporators consume 80-90 kJ to evaporate 1 liter of waste water (blowdown). Subsequently, crystallizers consume 190-285 kJ to crystallize 1 liter of reject to solids for eventual disposal (Maulbetsch and DiFillipo 2006). Thus a 1 liter waste water stream will generate

Figure 14: ZLD system - mass flows & energy consumption



0.9 – 0.98 L of treated water for make-up; and consume 90-120 kJ of energy.

Our model assumes a 90% recovery rate and parasitic power consumption of 110 kJ per 1 liter of waste water treated.

Some studies also include evaporation ponds as part of a ZLD system. Evaporation ponds are preferred when there is sufficient space on site and if local meteorological conditions are favorable for evaporation. However, water cannot be recovered from evaporation ponds. In our model, ZLD systems will not refer to such evaporation ponds.

5.2.3. Estimating parasitic power requirements

Within a power plant – the two key areas of parasitic power consumption are the working fluid feed pump, and the cooling tower equipment.

For the cooling towers, energy consumption depends upon the configuration of the cooling system – approach temperatures and range; local conditions – average temperature and humidity; and seasonal variations – winter versus summer. The major energy users in a cooling system are pumps and fans. Pumps are used for circulating water in wet cooling towers. Their energy use is determined by, *inter alia*, the flow rate or the amount of water that has to be pumped. Fans are used for ventilation in cooling towers and condensers. Their energy use is determined by the number, size and the type of fans; and the amount of air to be ventilated.

Per EC (2001), both dry and wet cooling systems require around 0.02kW per kJ/s of heat to be dissipated (i.e. condenser duty). Maulbetsch and DiFilippo (2006) analysis indicates around 0.01kW per kJ/s of condenser duty.

To account for boiler feedpump and other auxiliary equipment at the power plant, we assume a requirement of 0.01kW per kJ/s of condenser duty. This seems reasonable based on Maulbetsch and DiFilippo (2006).

Overall, for power plant related parasitic power requirements, our model defaults to a value of 0.02 kW per kJ/s of heat to be dissipated (i.e. condenser duty). Assuming a 10% net thermal efficiency, this would imply approximately 20% of the gross capacity of the power plant.

6. Water required by flash power plants

In this chapter, we estimate water requirements indicated in Figures 4.1 and 5.1.

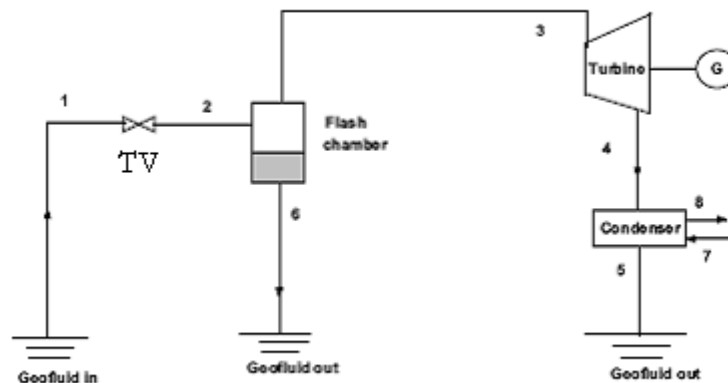
In flash power plants, geothermal fluid above around 180°C (EERE 2010) is sprayed into a tank held at a much lower pressure than the fluid, causing some of the fluid to rapidly vaporize, or "flash." The vapor (steam) then drives a turbine, which drives a generator. The liquid remaining in the tank can be flashed again in a second tank to extract even more energy (double flash power plants). In such plants, the steam is condensed to lower the turbine back pressure and increase overall efficiency. Most or nearly all of the makeup water for cooling towers is provided by the steam condensate.

We model a single flash power plant with a wet re-circulating cooling system. Double flash plants and flash-binary power plants will be modeled in later versions of our model.

6.1. Thermodynamics behind a single flash power plant

The thermodynamic analysis is also important to calculate the required flow rate of geothermal fluid - fluid losses from EGS resources is directly related to the flow rate. This will be addressed in the following chapter.

Figure 15: Schematic diagram of a single flash power plant



Source: Figure reproduced with permission from Yari (2010)

The following thermodynamic analysis of a single flash plant is based largely on DiPippo (2008). Unlike an ORC cycle, the first law thermal efficiency cannot be applied to a flash power plant because it is not a closed cycle. We use the second law efficiency, alternatively referred to as exergetic or utilization efficiency to calculate geothermal fluid flow rates and condenser duty.

- **State 1**

The geothermal fluid will be assumed to be a saturated liquid at an temperature of $T_{geo,in}$ (user input). With the help of tables regarding water saturation properties (steam tables, (NIST 2010), the pressure, entropy, and enthalpy can be obtained. The specific exergy of incoming geothermal fluid can be calculated based on the following:

$$e_1 = h_1 - h_0 - T_0 (s_1 - s_0) \quad (\text{Equation 21})$$

where

e_1 is specific exergy

T_0 is the dead state or ambient temperature

s_0 and h_0 are the specific entropy in kJ/(kg.K) and specific enthalpy in kJ/kg respectively at given ambient conditions T_0 ; and can be obtained from steam tables.

s_1 and h_1 are the entropy in kJ/(kg.K) and enthalpy in kJ/kg respectively in state 1 given temperature $T_{geo,in}$ and corresponding pressure P_1

The total exergy of the incoming geothermal fluid can be calculated by

$$\dot{E}_{in} = \dot{m}_{geo} e_1 \quad (\text{Equation 22})$$

where,

\dot{E}_{in} is total exergy in kJ/s

\dot{m}_{geo} represents the mass flow rate in kg/s of the geothermal fluid extracted

The exergetic efficiency (also called second-law or utilization efficiency) of the flash power plant is defined as

$$\eta_{ex} = \frac{\dot{W}_{net}}{\dot{E}_1} \quad (\text{Equation 23})$$

▪ State 2

The throttle valve (TV) creates a flow restriction which maintains a pressure drop from state 1 (saturated fluid at inlet geothermal fluid temperature $T_{geo,in}$) to state 2 (two-phase mixture of saturated liquid and saturated vapor). After the flash chamber, there is a stream of saturated liquid at state 3 and saturated vapor at state 4.

The mass fraction of the geothermal fluid which flashes to saturated steam can be determined by the “lever rule” which is given as

$$x_2 = \frac{h_2 - h_6}{h_3 - h_6} \quad (\text{Equation 24})$$

where,

h_2 , h_3 , and h_6 are the specific enthalpies in kJ/kg at states 2, 3 and 6 respectively and are obtained from the steam tables.

The enthalpy at state 2 is assumed to be the same as state 1 because there is no heat or work input to the fluid. As a result, the above equation can be written as

$$x_2 = \frac{h_2 - h_6}{h_3 - h_6} = \frac{h_1 - h_6}{h_3 - h_6} \quad (\text{Equation 25})$$

$x_2 \dot{m}_{geo}$ represents the total mass of steam in kg/s that goes to the turbine.

Based on DiPippo (2008), the optimum flash temperature T_2 (and hence pressure P_2) equals the average temperature of the inlet geofluid temperature and the condenser temperature:

$$T_2 = \frac{(T_{geo,in} + T_5)}{2} \quad (\text{Equation 26})$$

where,

T_2 is the temperatures in the flash chamber

T_5 is the temperatures of the liquid exiting the condenser. The model defaults to a value of 50 °C for T_5 .

▪ States 3 and 4

Here the fluid is assumed to be in the saturated vapor state; and at the same pressure and temperature as in state 2. For a double flash power plant, it is possible that the fluid at the inlet of the turbine is wet.

The vapor coming out of the turbine has lower enthalpy (h_4) due to conversion to mechanical/electrical energy. The difference in enthalpy between states 3 and 4 is equal to the gross electric output of the plant after taking into consideration generator efficiency. The vapor is at a lower temperature in 4 than at 3.

$$W_{gross} = \eta_{gen} \eta_{tur} (h_3 - h_4) x_2 \dot{m}_{geo} \quad (\text{Equation 27})$$

where,

W_{gross} is the gross electricity output in kJ/s

η_{gen} is the turbine efficiency. Model defaults to 0.85 based on MIT (2006)

η_{tur} is the generator efficiency. Model defaults to 0.95

▪ State 5

The saturated vapor exiting the turbine is condensed in the condenser. The fluid in state 5 is assumed to be in the saturated liquid state and at an outlet temperature of T_{cond} with a corresponding enthalpy of h_4 . The total condenser duty can be calculated as

$$\dot{Q}_{out} = x_2 \dot{m}_{geo} (h_4 - h_5) \quad (\text{Equation 28})$$

where,

\dot{Q}_{out} is the heat load on the condenser; and equivalently represents the heat that needs to be rejected by the cooling tower

- **State 6**

Here the fluid is in the saturated liquid state; and at the same pressure and temperature as in state 2. The fluid is either injected back, flashed again for additional power generation (double flash), or run through a heat exchanger to generate power in an ORC cycle (flash-binary). In some cases, it may be discharged to the surface as in case of the Denizli-Kızıldere geothermal power plant in Turkey where the geothermal fluid is discharged to the river (Dagdas, Öztürk et al. 2005).

- **Cooling Tower Analysis**

As in case of the ORC cycle, the water required in the cooling tower is given by

$$\dot{w}_{evp} = \dot{Q}_{out} \times f_{latent} / h_{vap} \quad (\text{Equation 29})$$

The blowdown rate can be given as before

$$\dot{w}_{BD} = \frac{\dot{w}_{evp}}{(N-1)}, \text{ and} \quad (\text{Equation 30})$$

The blowdown is assumed to be injected back. Makeup water requirement that needs to be sourced externally is given by

$$\dot{w}_{MU,E} = \dot{w}_{evp} + \dot{w}_{BD} - x_2 \dot{m}_{geo} \quad (\text{Equation 31})$$

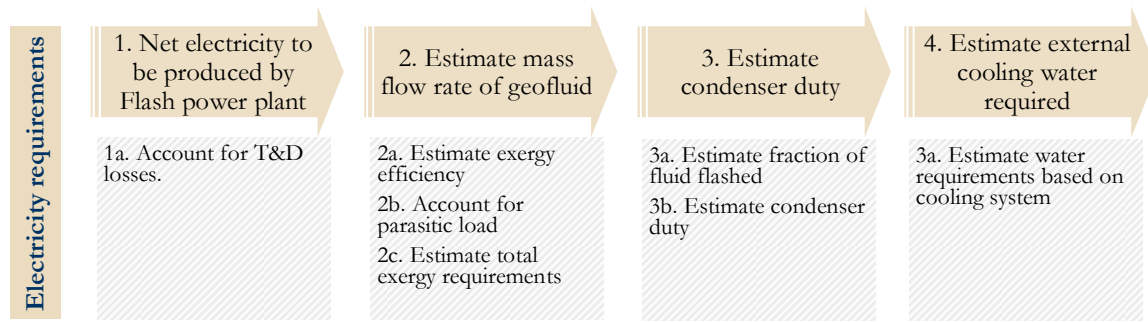
where,

$\dot{w}_{MU,E}$ is the makeup rate of water in kg/s (or liter/s assuming water density to be 1 kg/liter) that needs to be sourced externally.

6.2. Methodology

The model estimates two quantities for flash power plants. First, the geothermal fluid flow rate which is essential to estimate fluid losses for EGS resources. Second, the makeup water requirements for cooling system of the plant.

Figure 16: Methodology adopted for analyzing Flash power plants



▪ **Step 1**

As with ORC plants, the starting point of the model is the electricity consumed by the end customer, which in turn gives the electricity required to be produced by the flash power plant after accounting for transmission and distribution losses (refer to equation 11).

▪ **Step 2**

The model defaults to a net exergetic efficiency of 0.32 for single flash power plants; and a parasitic power consumption of 5.7% of net power generated. An additional modification to parasitic power is made based on non-condensable gases (NCG) content of the steam.

Based on the above relationships, total mass of the incoming geofluid can be calculated based on Equation 22.

▪ **Step 3**

In Step 3, the fraction of fluid flashed is calculated based on Equation 24. The enthalpy of the steam as it exits the turbine can be calculated based on Equation 27, which then allows us to calculate condenser duty based on Equation 28.

▪ **Step 4**

The model assumes that the flash power plant has a wet re-circulating cooling tower. In later versions of the model, we will analyze dry and hybrid cooling systems.

6.2.1. Exergy efficiency of flash power plants

DiPippo (2004) analyzed the relationship between exergetic efficiency and incoming specific exergy of the geofluid (which in turn is directly related to the temperature). The analysis was undertaken for a wide variety of geothermal power plants including plants of flash, binary, and hybrid technology. The author did not find any trend and concluded that exergetic efficiency is determined by the sophistication of the plant design irrespective of the incoming exergy. This is in contrast to the first law thermal efficiency of ORC cycle where there is a positive linear relationship with temperature of the geofluid.

Based on the data given in the following table, the model defaults to net exergetic efficiency of 0.32 for single flash power plant.

Table 11: Exergetic efficiency of various flash power plants

Plant	Brine Inlet Temperature	Net Exergetic Efficiency	Cooling System	Sources
Single Flash				
Miravalles, Unit 1; Costa Rica	240	0.28	WC	DiPippo (2008)
Miravalles, Unit 2	240	0.28	WC	DiPippo (2008)
Miravalles, Unit 3;	240	0.28	WC	DiPippo (2008)
Denizli Kızıldere, Turkey(1)	200	0.20	WC-OT	Dagdas, Öztürk et al. (2005) and Gokcen, Kemal Ozturk et al. (2004)
Simulation	230	0.35	DC	Yari (2010)
Cerro Prieto I Units 1-4, Mexico	250	0.35	WC	DiPippo c
Double Flash				
Cerro Prieto I Unit 5	169.5	0.32	WC	DiPippo (2008)
Cerro Prieto II & III	320	0.49	WC	DiPippo (2008)
Beowawe, NV	215	0.47	WC	DiPippo (1999)

Notes: 1. High fraction of NCG (10–21% mass) leading to high parasitic load.
WC: Wet cooling system; DC: Dry cooling system; OT: Once through cooling

6.2.2. Parasitic power consumption

The average parasitic power consumption for many of the flash plants listed in the above table is 5.1% of gross power generated (5.7% of net power generated). The model defaults to a parasitic power consumption of 5.7% of net power generated.

One of the key determinants of parasitic power consumption is fraction of NCG in the geofluid. NCG concentrations in steam are typically 0.5-1.0% (wt) of steam with CO₂ and H₂S constituting about 95% and 1-2% of the NCG (DiPippo 2008). NCG could significantly reduce turbine power output in a dry- or flash-steam power plant by raising the backpressure on the turbine and thus reducing the enthalpy drop across the turbine. This necessitates the removal of NCG by some means – steam jet ejectors, vacuum pumps, or turbo-compressors which increases parasitic power. However, DiPippo (2008) indicates that the resulting net power output is higher than likely output where NCG were left to accumulate. The need for NCG capture and removal may also be driven by local pollution norms.

At the Denizli Kızıldere power plant in Turkey, NCG constitutes 10-21% weight of the steam. As a result, the parasitic power consumption is almost 23%; much of it accounted for by the compressor (Gokcen, Kemal Ozturk et al. 2004).

The model defaults to a NCG level of 0.5-1.0% (wt) of steam. However, if a higher level of NCG is selected, then additional compressor duty of 100 kW for every 1 %(wt) of NCG is assumed based on Gokcen, Kemal Ozturk et al. (2004) and Yildirim Ozcan and Gokcen (2009).

7. Geothermal Resources

7.1. Water required to recharge hydrothermal resources

In this section, we estimate water requirements indicated in Figures 2.2 and 4.2.

Hydrothermal resources can witness pressure and production declines if the volume of geothermal fluid injected back to the ground is significantly less than volume withdrawn. This is especially true for steam-dominated resources; the steam condensate available for reinjection constitutes only 10-15% of the mass of dry steam withdrawn for power generation (DiPippo 2008). In the Geysers steamfield in California, around 42 million liters of highly treated municipal effluent from the city of Santa Rosa is injected daily (City of Santa Rosa 2007; City of Santa Rosa 2010) to maintain steam pressure.

In case of flash power plants, which are used for energy conversion at wet steam dominated geothermal fields, geothermal fluid and steam condensate available for reinjection constitute around 70-80% of the mass withdrawn (EPRI 1997; DiPippo 2008). The difference of 20-30% is lost primarily through evaporation of the steam condensate in the cooling towers. Our model estimates the steam condensate lost due to evaporation. To compensate for this loss, degraded water has to be sourced externally for reinjection.

In case of binary power plants, which are used for energy conversion of hot water fields, the entire mass of geothermal fluid produced is available for reinjection. Hence, additional water is not required for injection.

7.2. Water required for heat mining of EGS resources

In this section, we estimate water requirements indicated in Figures 3.2 and 5.2.

EGS resources have high temperature but contain little or no geothermal fluid. Water from an external source is injected under high pressure through the injection well. The fluid then returns to the surface through the production well, and thus transfers the heat to the surface as steam or hot water. The injection pump provides the sole motive force for moving the water continuously around the loop to mine energy from the reservoir and deliver it to a power plant on the surface (Duchane 1996).

Ideally, a closed loop is created whereby cold water is pumped down the injection well and returned to the surface through the production well after passing through the hot, artificially fractured formation. However, losses may occur due to permeation and leakage from the fracture system to the surrounding rocks. The extent of losses will depend upon site specific conditions like permeability of rocks, depth of the reservoir, as well as age. Losses are also a function of injection pressure - Tester, Anderson et al. (2006) notes that high-injection pressures extends the fractures and increases permeability which in turn increase flow losses. Murphy, Drake et al. (1985) indicated a loss of 5% of EGS reservoir circulation flow rate

based on experience from the EGS resource at Fenton Hill, New Mexico where losses during start of operation were greater than 10% but decreased after that. Duchane (1992) indicated water losses in the order of 1-2% of EGS reservoir circulation flow rate. EPRI (1997) have calculated water consumption by a EGS resource based on water losses of 5% and 15%.

Fluid losses should be below 10% for long term viability of the resource (Barbier 2002; Kaltschmitt 2007). This is not only true for the very large volumes of freshwater lost in that way, but also for the additional pumping power required for the makeup water. The MIT study (Tester, Anderson et al. 2006) study assumed that fluid losses of 2%. Circulation losses are 1-2% in the EGS project at Soultz, France (personal communication with Susan Petty, January 14, 2011)

Fluid losses in an EGS resource present a trade-off decision to planners. Higher efficiency of energy conversion may require a wet or hybrid cooling tower, but will require lower geothermal mass flow rates. Water required for cooling purposes may be compensated by lower fluid losses during heat mining. Another area of trade-off is between flow losses and electricity generated – higher injection pressures increase flow rates and hence electricity produced but also increases flow losses (Tester, Anderson et al. 2006).

The model defaults to a water loss of 2% of the circulation flow rate.

7.3. Power requirements for geothermal fluid production and injection

In vapor or steam dominated resources, the geothermal fluid originating from the boiling of the geothermal fluid in the production wells was brought to the surface via buoyancy .

Kaltschmitt (2007) lists several reasons for the need to pump geothermal fluid by using centrifugal pumps in production wells: buoyancy may be too small to establish a self-pumping production; or produces an economically justifiable flow rate. Further, a certain overpressure has to be maintained in the geothermal loop to prevent infiltration of oxygen causing corrosion in the surface installations and in the casing of the re-injection well and to avoid plugging of the formation by iron oxide precipitation near the re-injection interval.

Pumping demands are influenced by the above ground horizontal distances over which fluid has to be pumped. Kaltschmitt (2007) indicates that production and injection wells must at least be located at a distance of approximately one km to avoid a fast arrival of the re-injected cold water, and to ensure a long technical lifetime of the system. In sites using directional instead of vertical drilling, the wells could be located in a single well head and this would save pumping demands.

Frick, Kaltschmitt et al. (2010) indicate that pump power requirements range from 1.8 to 10.8 kW/(kg/s). Heidinger, Dornstädter et al. (2006) indicate that parasitic power demand of the pumps consumes a great deal of the electricity produced from EGS resources. Based on a graph given for energy consumption by pumps at the Soultz-sous-Forets EGS reservoir in France, power consumption is in the range of 5-10 kW/(kg/s). However, Heidinger,

Dornstädter et al. (2006) indicate that pump power requirements is more likely to have an exponential, rather than linear, relationship with flow rate.

Our model defaults to a linear relationship of 10.8 kW/(kg/s) for EGS resources and 2.5 kW/(kg/s) for water dominated hydrothermal resources. Steam dominated resources are assumed to not require pumping. These numbers and relationship can be further refined in later versions of the model.

8. References

- Barbier, E. (2002). "Geothermal energy technology and current status: an overview." Renewable and Sustainable Energy Reviews **6**(1-2): 3-65.
- Bayart, J.-B., C. Bulle, et al. (2010). "A framework for assessing off-stream freshwater use in LCA." The International Journal of Life Cycle Assessment **15**(5): 439-453.
- Chandrasekharam, D. and J. Bundschuh (2008). Low-Enthalpy Geothermal Resources for Power Generation Boca Raton, Taylor & Francis.
- City of Santa Rosa (2007). Incremental Recycled Water Program - August 2007 Addendum to Program EIR and Geysers Expansion Project CEQA Checklist. Santa Rosa, CA.
- City of Santa Rosa. (2010). "Geysers Project - Facts and Stats." Retrieved June 2010, from <http://ci.santa-rosa.ca.us/departments/utilities/irwp/geysers/Pages/FactStat.aspx>.
- Dagdas, A., R. Öztürk, et al. (2005). "Thermodynamic evaluation of Denizli KIZildere geothermal power plant and its performance improvement." Energy Conversion and Management **46**(2): 245-256.
- DiPippo, R. (1999). Small geothermal power plants: Design, performance and economics. GHC Bulletin. Klamath Falls, OR, Oregon Institute of Technology: Geo-Heat Center: 8.
- DiPippo, R. (2004). "Second Law assessment of binary plants generating power from low-temperature geothermal fluids." Geothermics **33**(5): 565-586.
- DiPippo, R. (2007). "Ideal thermal efficiency for geothermal binary plants." Geothermics **36**(3): 276-285.
- DiPippo, R. (2008). Geothermal power plants: principles, applications, case studies and environmental impact. Oxford, UK, Butterworth-Heinemann.
- DOE (2002). Energy Penalty Analysis of Possible Cooling Water Intake Structure Requirements on Existing Coal-Fired Power Plants, US Department of Energy - National Energy Technology Laboratory (NETL) and Argonne National Laboratory (ANL).
- DOE (2008). Concentrating Solar Power Commercial Application Study: Reducing Water Consumption of Concentrating Solar Power Electricity Generation - Report to Congress. Washington D.C., U.S. Department of Energy: 35.
- Duchane, D. (1996). Progress in making Hot Dry Rock geothermal energy a viable renewable energy resource for America in the 21st century. Los Alamos, NM, Los Alamos National Laboratory.
- EC (2001). Integrated Pollution Prevention and Control (IPPC): Reference document on the application of best available techniques in industrial cooling systems. , European Commission.

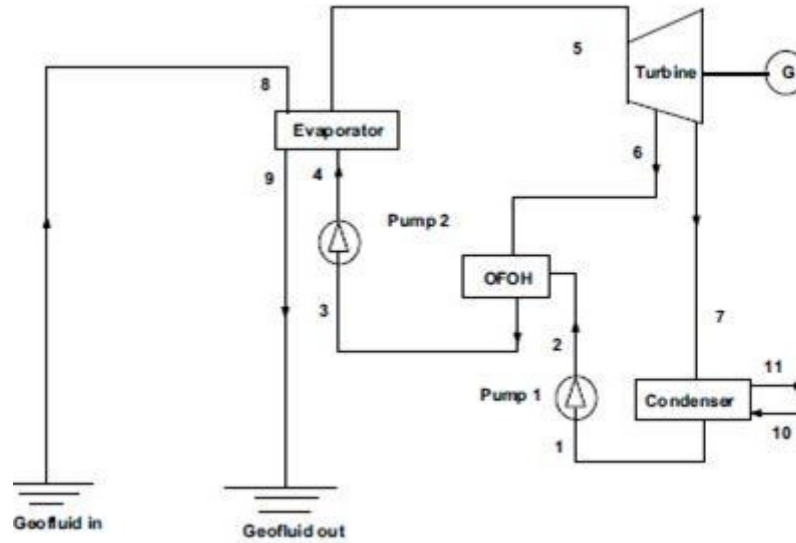
- EERE. (2010). "Hydrothermal Power Systems." June 2010, from <http://www1.eere.energy.gov/geothermal/powerplants.html>.
- EIA (2010). Annual Energy Outlook 2010. Washington DC, US Department of Energy - Energy Information Administration.
- EPA (1999). Electric Power Geothermal Injection Wells. The Class V Underground Injection Control Study, US Environmental Protection Agency. 17.
- EPRI (1997). Renewable Energy Technology Characterizations, Electric Power Research Institute (EPRI) and US Department of Energy (DOE).
- EPRI (2002). Comparison of Alternate Cooling Technologies for California Power Plants - Economic, Environmental and Other Tradeoffs. Public Interest Energy Research Program (PIER). Sacramento, CA, Electric Power Research Institute and California Energy Commission.
- EPRI (2003). Spray Enhancement of Air Cooled Condensers. Public Interest Energy Research Program (PIER). Sacramento, CA, California Energy Commission, Electric Power Research Institute and Crockett Cogeneration.
- EPRI (2003). Use of Degraded Water Sources as Cooling Water in Power Plants. Public Interest Energy Research Program (PIER). Sacramento, CA, Electric Power Research Institute and California Energy Commission.
- EPRI (2004). Comparison of Alternate Cooling Technologies for U. S. Power Plants: Economic, Environmental, and Other Tradeoffs. Palo Alto, CA, Electric Power Research Institute (EPRI).
- Franco, A. and M. Villani (2009). "Optimal design of binary cycle power plants for water-dominated, medium-temperature geothermal fields." Geothermics 38(4): 379-391.
- Frick, S., M. Kaltschmitt, et al. (2010). "Life cycle assessment of geothermal binary power plants using enhanced low-temperature reservoirs." Energy 35(5): 2281-2294.
- GAO (2009). Energy-Water Nexus: Improvements to Federal Water Use Data Would Increase Understanding of Trends in Power Plant Water Use. Washington D.C., United States Government Accountability Office: 78.
- Glassley, W. E. (2010). Geothermal Energy: Renewable Energy and the Environment. Boca Raton, Taylor and Francis.
- Gokcen, G., H. Kemal Ozturk, et al. (2004). "Overview of Kizildere Geothermal Power Plant in Turkey." Energy Conversion and Management 45(1): 83-98.
- GREET (2010). Greenhouse Gases, Regulated Emissions, and Energy Use in Transportation (GREET) Model. Argonne, IL, Argonne National Laboratory.
- Heberle, F. and D. Brüggemann (2010). "Exergy based fluid selection for a geothermal Organic Rankine Cycle for combined heat and power generation." Applied Thermal Engineering 30(11-12): 1326-1332.

- Heidinger, P., J. Dornstädter, et al. (2006). "HDR economic modelling: HDRRec software." Geothermics **35**(5-6): 683-710.
- Imroz Sohel, M., M. Sellier, et al. (2009). "Efficiency improvement for geothermal power generation to meet summer peak demand." Energy Policy **37**(9): 3370-3376.
- Kaltschmitt, M. (2007). Utilisation of Geothermal Energy. Renewable Energy - Technology, Economics and Environment M. Kaltschmitt, W. Streicher and A. Wiese, Springer: 437-496.
- Kanoglu, M. and Y. A. Cengel (1999). "Improving the performance of an existing binary geothermal power plant: a case study. Transactions of the ASME. ." Journal of Energy Resources Technology **121**(3): 196-202.
- King, C. W. and M. E. Webber (2008). "The Water Intensity of the Plugged-In Automotive Economy." Environmental Science & Technology **42**(12): 4305-4311.
- Klein, G., M. Krebs, et al. (2005). California's Water - Energy Relationship. Sacramento, CA, California Energy Commission (CEC).
- Kutscher, C. and D. Costenaro (2002). Assessment of Evaporative Cooling Enhancement Methods for Air-Cooled Geothermal Power Plants Geothermal Resources Council (GRC) Annual Meeting Reno, Nevada, National Renewable Energy Laboratory
- Layton, D. and W. Morris (1980). Geothermal Power Production: Accidental Fluid Releases, Waste Disposal, And Water Use. Livermore, CA, Lawrence Livermore Laboratory.
- Maulbetsch, J. S. and M. N. DiFilippo (2006). Cost and Value of Water Use at Combined-Cycle Power Plants. Public Interest Energy Research (PIER) Program. Sacramento, CA, California Energy Commission.
- Michaelides, E. E. and J. K. Ryder (1992). "The influence of seasonal and daily temperature fluctuations on the work produced by geothermal power plants." International Journal of Energy Systems **12**(2): 68-72.
- Mishra, G. S. and S. Yeh (2010). Analysis of Lifecycle Water Requirements of Transportation Fuels: Corn-based Ethanol. Davis, Institute of Transportation Studies, University of California.
- Murphy, H., R. Drake, et al. (1985). "Economics of a conceptual 75 MW hot dry rock geothermal electric power-station." Geothermics **14**(2-3): 459-474.
- NETL (2008). Estimating Freshwater Needs to Meet Future Thermoelectric Generation Requirements - 2008 Update, National Energy Technology Laboratory (NETL), Department of Energy.
- Owens, J. W. (2001). "Water Resources in Life-Cycle Impact Assessment: Considerations in Choosing Category Indicators." Journal of Industrial Ecology **5**(2): 37-54.

- Tester, J. W., B. J. Anderson, et al. (2006). The Future of Geothermal Energy - Impact of Enhanced Geothermal Systems (EGS) on the United States in the 21st Century, Massachusetts Institute of Technology (MIT).
- UC IPM. (2010). "California Weather Data." Retrieved August, 2010, from <http://www.ipm.ucdavis.edu/WEATHER/wxretrieve.html>.
- USGS. (2010). "The Water Cycle: Freshwater Storage." Retrieved May, 2010, from <http://ga.water.usgs.gov/edu/watercyclefreshstorage.html>.
- Weiss, K. (1996). Liquid and Solid Waste Treatment and Disposal. Power plant engineering - Black & Veatch. L. F. Drbal, P. G. Boston and K. L. Westra, Springer.
- WorleyParsons (2008). FPLE - Beacon Solar Energy Project Dry Cooling Evaluation. Folsom, CA, Submitted to California Energy Commission.
- Yari, M. (2010). "Exergetic analysis of various types of geothermal power plants." Renewable Energy **35**(1): 112-121.
- Yildirim Ozcan, N. and G. Gokcen (2009). "Thermodynamic assessment of gas removal systems for single-flash geothermal power plants." Applied Thermal Engineering **29**(14-15): 3246-3253.

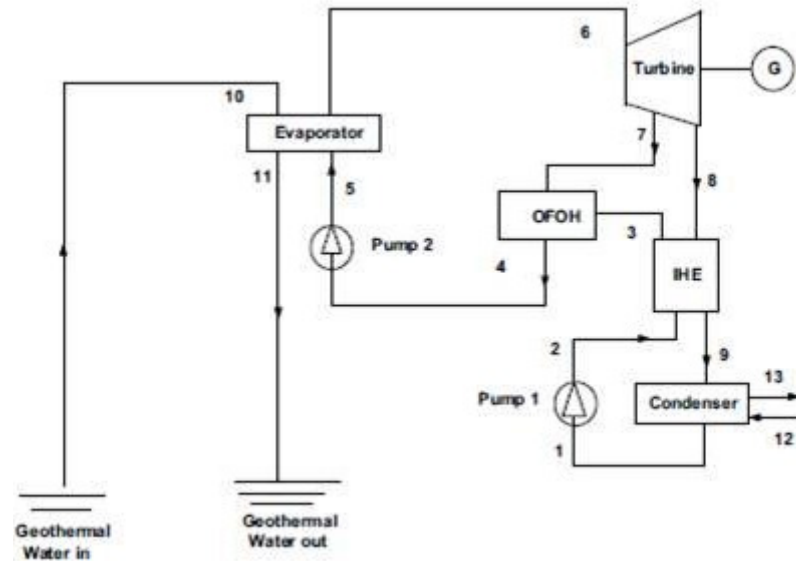
9. Appendix A: Some advanced ORC configurations

Figure A.1: Schematic diagram of a regenerative ORC cycle



Source: Reproduced with permission from Yari (2010)

Figure A.2: Schematic diagram of a regenerative ORC with IHE



Source: Reproduced with permission from Yari (2010)

10. Appendix B: ORC power plants

<i>Plant Name</i>	<i>Brine Inlet Temp (°C)</i>	<i>Outlet Temp (°C)</i>	<i>Efficiency</i>	<i>Cooling System</i>	<i>Net Output (MW)</i>	<i>Parasitic Power (MW)</i>	<i>WF</i>	<i>Notes</i>
Amedee, CA (1)	104	71	5.6%	WC-CP	1.6	0.4	R114	Geothermal fluid is not injected but discharged to surface because of low salinity (Lund 1999)
Wabaska, NV	105		8.0%	WC-CP			Iso-pentane	
Simav-Kutahya, Turkey (Proposed)(2)	145	90	10.6%	WC			HCFC-124	The power plant is proposed and does not exist currently. Kose (2007) assumed a basic ORC.
Húsavík, Iceland (3) (4)	122	80	10.6%	WC-OT	1.696	0.127	82% NH3 and 18% H2O	Kalina Cycle
Miravalles Unit 5, Costa Rica (5)	166	136	12.8%	WC-CT	13.702	2.113	Iso-pentane	Miravalles uses the gravity-fed outflow of waste brine from a set of flash plants - hence lower parasitic power. Performance results for 2006
Nigorikawa, Japan (6)	140		9.8%	WC-CT				Retired
Heber - Second, Imperial Valley CA (1)	168	71	13.2%	WC	31	1.88	Iso-pentane	Dual Pressure binary power plant
Chena Hot Springs, AK (7)	73.5	57	8.2%	WC	0.21	0.04	R134a	
Mammoth Pacific - Unit I, CA (1)	169	77	8.1%	DC	7	2.47	Iso-butane	Basic ORC
Stillwater, NV (8)	163	66	8.89%	DC	1.769	0.467	Iso-pentane	Dual Pressure ORC
Reno, NV (9)	160	90	4.4%	DC	16.396	5.348	Iso-butane	
Brady, NV (3)	108	82.15	6.9%	DC	3.615	0.88		
Steamboat, NV (5)	152		7.9%	DC				
Otake, Japan (6)	130	50	12.9%	DC	1		Iso-butane	Receives brine from adjacent flash-stream plant.

Notes

- WC: Wet cooling; DC: Dry cooling; CP: Cooling pond; OT: Once through; CT: Cooling tower.

- Sources: (1) DiPippo (1999), (2) Kose (2007) (3) DiPippo (2004), (4) (Tester, Anderson et al. (2006), (5) DiPippo (2007), (6) DiPippo (2008), (7) Holdmann (2007), (8) Kanoglu (2002), (9) Kanoglu and Bolatturk (2008)

11. Appendix C: T&D losses

Table C.1: T&D losses for 2008

	Total Disposition (million kWh)	Direct Use (million kWh)(1)	Estimated Losses (million kWh)	T&D losses % (2)
Alabama	100,350	4,771	5,872	6.14%
Alaska	7,106	329	453	6.68%
Arizona	82,672	505	5,549	6.75%
Arkansas	51,726	1,987	3,604	7.25%
California	310,311	21,916	19,681	6.82%
Colorado	57,047	911	3,991	7.11%
Connecticut	34,569	949	2,527	7.52%
Delaware	13,250	751	751	6.01%
Florida	248,202	6,756	15,273	6.33%
Georgia	152,742	5,894	11,675	7.95%
Hawaii	11,389	396	603	5.49%
Idaho	26,706	613	2,103	8.06%
Illinois	157,473	4,993	7,849	5.15%
Indiana	120,806	7,963	5,757	5.10%
Iowa	48,957	1,413	2,056	4.32%
Kansas	43,095		3,579	8.30%
Kentucky	99,304	366	5,510	5.57%
Louisiana	107,654	23,878	5,054	6.03%
Maine	16,375	3,651	332	2.61%
Maryland	69,515	1,204	4,985	7.30%
Massachusetts	62,964	4,080	2,881	4.89%
Michigan	121,047	3,400	7,865	6.69%
Minnesota	74,944	1,348	3,794	5.16%
Mississippi	54,696	3,375	3,600	7.01%
Missouri	90,593	311	5,885	6.52%
Montana	20,541	287	4,439	21.92%
Nebraska	31,317	18	2,488	7.95%
Nevada	36,409	386	764	2.12%
New Hampshire	12,676	952	636	5.42%
New Jersey	89,823	4,823	4,480	5.27%
New Mexico	24,019	272	1,591	6.70%
New York	157,019	2,544	7,060	4.57%
North Carolina	141,798	2,939	8,805	6.34%
North Dakota	14,664	208	1,435	9.93%
Ohio	170,895	1,228	10,278	6.06%
Oklahoma	63,232	2,169	4,784	7.83%
Oregon	54,104	1,460	3,184	6.05%
Pennsylvania	164,693	4,647	9,289	5.80%
Rhode Island	8,048	59	66	0.83%
South Carolina	87,727	1,978	5,099	5.95%
South Dakota	11,904	1	929	7.80%
Tennessee	111,188	2,472	4,546	4.18%
Texas	405,995	34,524	23,399	6.30%

	Total Disposition (million kWh)	Direct Use (million kWh)(1)	Estimated Losses (million kWh)	T&D losses % (2)
Utah	31,505	943	2,316	7.58%
Vermont	7,014	797	373	6.00%
Virginia	117,804	2,760	4,937	4.29%
Washington	103,147	650	4,917	4.80%
West Virginia	37,679	522	2,936	7.90%
Wisconsin	77,992	4,079	3,791	5.13%
Wyoming	19,060	1,001	1,304	7.22%
US TOTAL (AVG)	4,163,746	173,479	245,075	6.14%

(1) Direct Use electricity is electricity that is generated at facilities that is not put onto the electricity transmission and distribution grid, and therefore does not contribute to T&D losses.

(2) To calculate T&D losses as a percentage, Estimated Losses is divided by the result of Total Disposition minus Direct Use.