FEASIBILITY DESIGN OF AN INTEGRATED SINGLE-FLASH BINARY PILOT POWER PLANT IN NW-RWANDA

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ABSTRACT

Having 60% of its total electricity production dependent on diesel generators, Rwanda has adopted a strategy to produce its energy from other sources, especially from renewable ones. Geothermal energy is one of the most promising resources and its development is ongoing. The purpose of this report is to pre-design a pilot geothermal power plant intended to be used in the country. In this study, an integrated single-flash binary power plant has been proposed and the modelling was done using the Engineering Equation Solver (EES) software. The results show that a back-pressure turbine for the pilot power plant is best option, both economically and with regard to technical aspects. About 23 kJ per 1 kg of geothermal fluid can be produced, and later an Organic Rankine Cycle (ORC) can be added to reach about 86 kJ per 1 kg of fluid. At this stage, the overall first law of utilisation efficiency will be 10.4% and the second law efficiency will be about 47.5% which is good compared with other geothermal plant cycles.

1. INTRODUCTION

Geothermal energy stored within the interior of the earth amounts to a vast quantity which, if fully exploited, could supply the energy needs of mankind for millennia. Over the course of a year, about $4x10^{17}$ kJ or 100 billion megawatt-hours of heat energy is conducted to the surface from the interior of the earth (Dipippo, 1979). As the total primary energy consumption in the world is about 400 EJ/year (exajoules per year) (Fridleifsson, 2006), this emitted energy represents ten times the primary energy used by the entire world. Of the total world energy consumption, fossil fuels occupy 79%.

The significant fluctuations in oil prices should encourage governments to focus more on indigenous energy sources to meet their basic energy requirements. These oil price fluctuations have started to affect some countries that don't produce oil, such as Rwanda. In the recent past, electricity production in Rwanda was totally dominated by hydro power, and diesel generators were used only as backup systems. In the year 2002, the country faced a serious energy crisis because of an increase in electricity demand and a reduction in precipitation. The power plants were producing about 40% of their installed capacity which was not sufficient to meet the demand resulting in electrical power shortages. To overcome these power shortages, the government integrated additional diesel generators that now produce 60% of the total electricity needed in Rwanda. This can only be regarded as a short-term solution, because depending on the importation of fossil fuels poses huge problems for a landlocked developing country.

In order to reduce this dependency, one sustainable solution the government adopted is the utilisation of geothermal energy. Available data indicates a significant potential in this resource that could end the energy crisis and provide 100 percent of the electricity needs in Rwanda (Watkins and Verma, 2008).

This report concerns a tentative design of an integrated single-flash - ORC cycles pilot power plant for the northwest geothermal field in Rwanda by optimising the net output power from both turbines for constant well head enthalpy. As necessary data for this study such as fluid mass flow and the chemistry of the reservoir are not yet collected, a calculation model was made by assuming some parameters such as a flow rate of 1 kg/s and reservoir enthalpy of 900 kJ/kg that corresponds to 210°C, according to available geochemical data from recent geothermometer analysis (Newell et al., 2007). Scaling during reinjection and in heat exchangers was assumed to occur at 110°C.

2. GEOTHERMAL ENERGY

2.1 Principles of geothermal energy

Geothermal energy is the natural heat of the earth generated and stored in the earth's core, mantle and crust. It is a reliable energy source which can be used to produce power with low emissions compared to fossil fuels. It is a relatively clean electrical power source as it emits little or no greenhouse gases, reliable and home grown which makes countries less dependent on foreign oil.

This energy is economically exploitable in areas where the crust is fractured, and hot fluid flows through or it is trapped within hot rock formations. The rocks have to be fairly permeable in order to allow water to circulate through and transfer heat from hot rock formations to reservoirs at depths manageable for commercial drilling. Geothermal reservoirs must be isolated from cold groundwater, which lies like a blanket in surface formation (Fridleifsson and Freeston, 1994). This heat is comes to the surface through the so-called hot springs or fumaroles as explained in Figure 1.
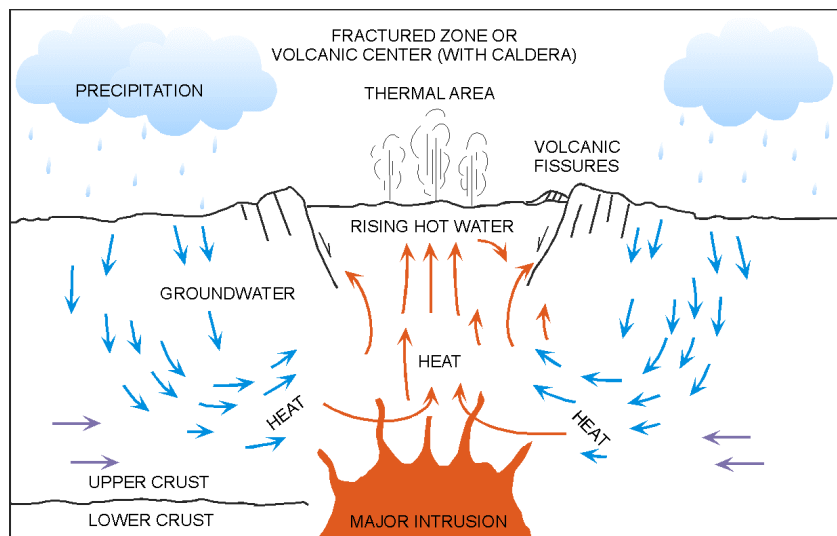

FIGURE 1: Formation of high-temperature geothermal areas (Hjartarson and Sigurdsson, 2008)

Exploitable geothermal systems are divided into two groups depending on whether they are related to young volcanoes and magmatic activity or not. High-temperature fields utilised for conventional power production (with temperatures above 150°C) are confined to the former group, but geothermal fields exploited for direct applications can be found in both groups. Volcanic activity mainly occurs along the so-called plate boundaries. According to plate tectonic theory, the earth's crust is divided into a few large rigid plates which float on the mantle and move relative to each other at average rates counted in centimetres per year. The plate boundaries are characterised by intense faulting and seismic activity and in many cases fractured and permeable rocks intersect a readily available heat source (Fridleifsson and Freeston, 1994).

2.2 Geothermal power plants

Depending upon the resource characteristics, a variety of different technologies can be applied to generate electricity from a geothermal system. In general, the key factor that dictates the choice of technology to be used is the reservoir temperature.

For low- to moderate-temperature systems (less than 150°C), electricity is often generated using "binary or ORC technology". With this technology, the geothermal fluids are transported to the surface in production wells and the produced water is then piped to a power plant. The geothermal water flows through a heat exchanger, which causes the vaporisation of a hydrocarbon fluid which is then sent through a turbine to generate power. As this cycle runs in a closed loop system, nothing has to be emitted to the atmosphere. The cooled geothermal brine can be returned to the geothermal system through injection wells, while the hydrocarbon is condensed and circulated back through the heat exchanger (US Department of Energy, 2008).

For high-temperature systems (of above 180°C), electricity is most often produced by passing the geothermal steam directly through a condensing steam turbine. The production wells produce hot water which is then sprayed into a separator held at a pressure much lower than that of the geothermal fluid, causing some of the fluid to boil, or flash. The steam is then piped directly to a turbine to produce power, while the residual water is returned to the reservoir through injection wells. This type of plant is called a "single-flash power plant" (US Department of Energy, 2008).

Instead of directly injecting the brine, there is a possibility of flashing it again in a second separator, kept at a lower pressure than the first one, to extract more energy; this second plant is called a "double-flash power plant". For both power plants (single- and double-flash), the steam that flows through the turbine is condensed and returned to the reservoir through dedicated condensate injection wells. If the fluid from the production well is primarily steam, the steam goes directly to a turbine which drives a generator that produces electricity; this plant is known as a "dry-steam power plant" (US Department of Energy, 2008).

Besides the above types of power plants, different combinations of cycles can be made in order to maximise the output power from the system. Among these are: ORC cycle in parallel with a single-flash cycle, modified double-flash cycle by adding a recuperator between the geothermal brine and the steam at the high-pressure turbine outlet, a Kalina binary cycle power plant that uses a mixture of water and ammonia as working fluid, etc. (US Department of Energy, 2008).

The production wells that produce hot water and steam are generally sited over the higher temperature parts of the resource, while the injection wells for separated water and steam condensate are normally located in the distant, cooler part of the reservoir and preferably downhill from the production locations. The injection needs to be properly distributed in the reservoir formation to avoid premature returns of the cool injected water to the producing wells (US Department of Energy, 2008).

3. GEOTHERMAL FIELDS IN RWANDA

3.1 Geological and structural settings

Rwanda hosts two prospective areas: the National Volcanoes Park, and faults associated with the East African Rift near Lake Kivu (Figures 2 and 3). The Volcanoes National Park was identified as a potential host for large, high-temperature geothermal systems, while the rift

FIGURE 2: Western and eastern branches of the East African Rift (Newell et al., 2007)
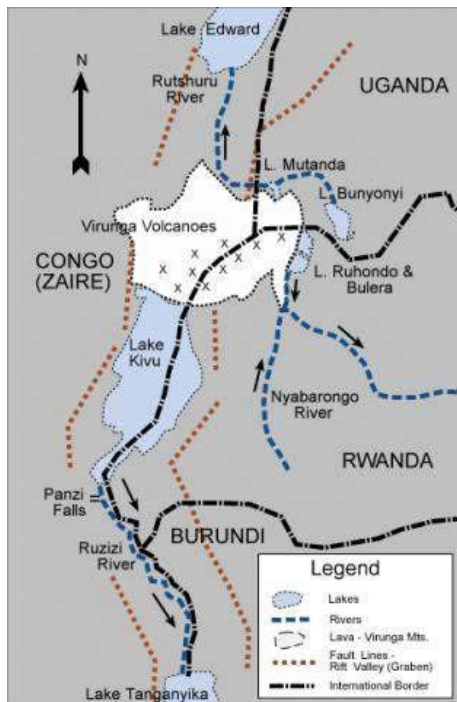
FIGURE 3: The prospective volcanic
geothermal area (Newell et al., 2007)

provides an environment for small, low- to moderate-temperature resources. The current study involves the northwest part of Rwanda that includes the Volcanoes National Part. The main structural trends of this area are controlled by the older basement structures. The Nyiragongo volcano to the west of the area has been erupting periodically. This signifies that a heat source for a geothermal system could exist around the major volcanoes. The area of exploration is between Nyiragongo volcano to the west, Muhabura volcano to the east and Gisenyi to the south (Figure 4).

The major structural trends, as summarised by the German Institute of Geosciences and Natural Resources (BGR), show that the older rift border faults have a predominantly northwesterly trend while the younger eastern rift border faults have a northeasterly trend (Figure 5). Other important structures include an accommodation zone which marks the boundary between the basement rocks and the volcanic rocks. It is anticipated that this structural pattern is probably buried below the younger volcanic rocks (Onacha, 2008).
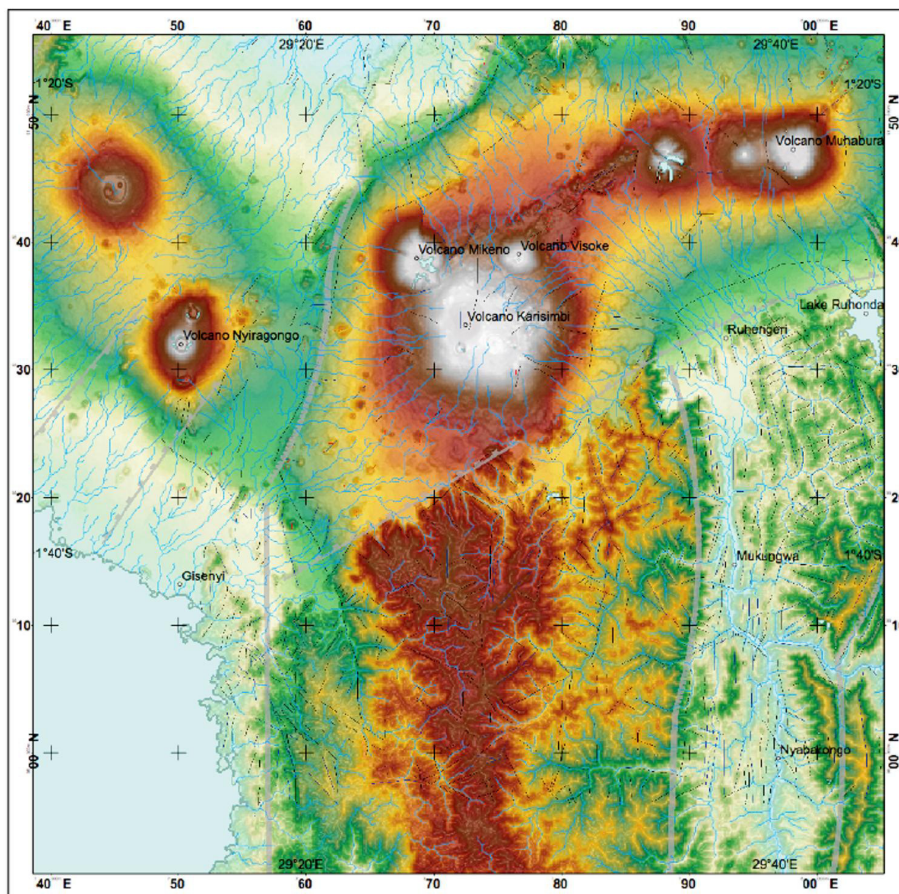

FIGURE 4: Northwest part of Rwanda showing the
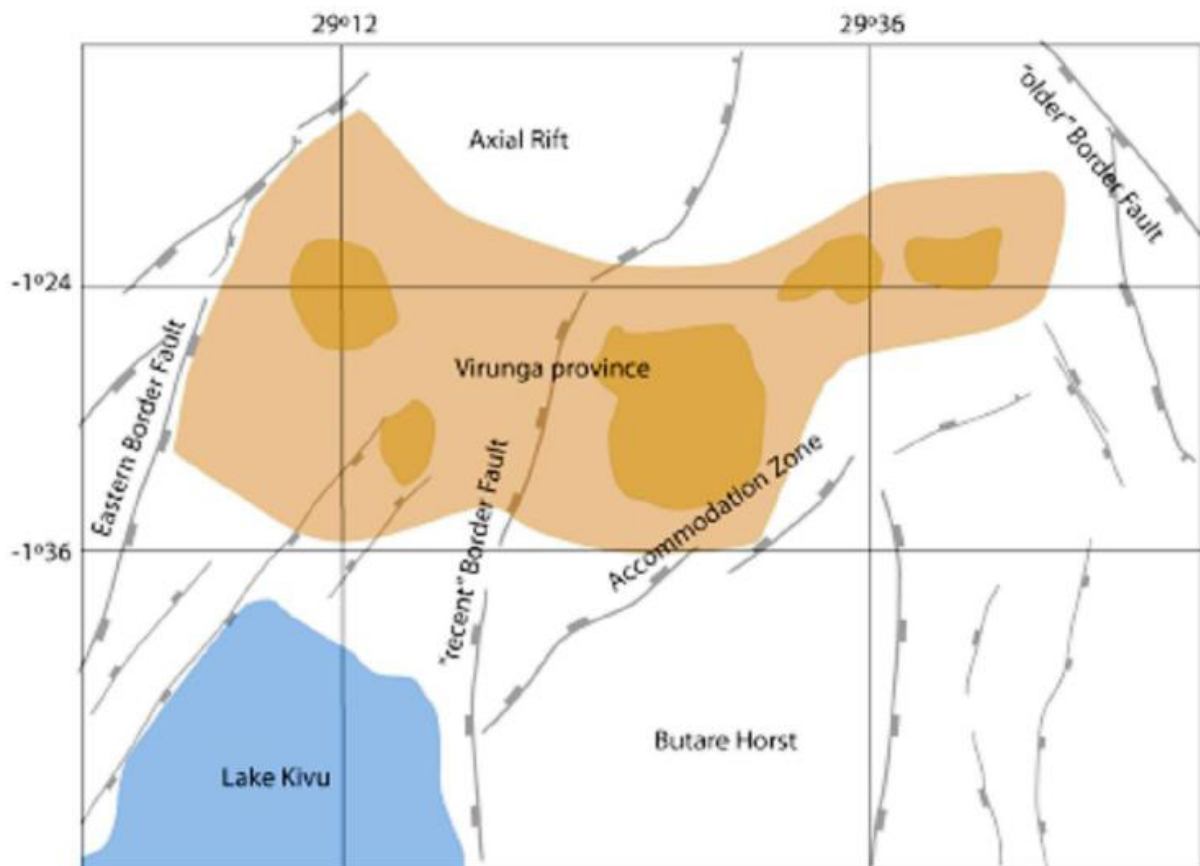area under exploration (Onacha, 2008)

FIGURE 5: Major structural trends in NW-Rwanda (Onacha, 2008)

3.2 Geochemistry of the field

Two different geochemical surveys were conducted in 1982 and 2006, respectively; two sites have proven promising for further evaluation for electric generation. One of them, the Gisenyi prospect in the area under exploration, consists of a single hot spring with several small vents located at the eastern shore of Lake Kivu. The hot springs, which issue from a brecciated and silicified quartzite, produce sodium bicarbonate ($Na-HCO_3$) waters with temperatures between 70 and 75°C. The combined flow rate from all vents is estimated to be 2-5 kg/s, although there are likely additional unseen vents below the lake surface. The highest temperature vents deposit a slight coating of reddish-brown iron oxide. The springs produce very little gas, probably CO_2, and there is a very faint odour of hydrogen sulphide. No travertine is being deposited by the hot spring, although travertine deposits could be in the near vicinity. This spring is considered to be $Na-HCO_3$ in nature; it contains lower concentrations of magnesium and higher concentration of silica, indicating that the waters likely originate from a higher temperature source (Newell et al., 2007).

A number of chemical geothermometers have been applied to the Gisenyi fluids in order to estimate the reservoir temperature (Table 1). Based upon these geothermometers, the reservoir temperature is estimated to be between 150 and 210°C. Included is the quartz geothermometer, which is based on the silica concentrations in the fluids, and the N-K-Ca and N-K-Ca-Mg geothermometers. The quartz geothermometer may reflect cooling and dilution along the flow path from the reservoir. If reservoir temperatures of 150-210°C are confirmed through exploration drilling, then the Gisenyi resource could possibly be economically exploited with binary technology (Newell et al., 2007).

TABLE 1: Geochemical analyses of samples taken in 1982 and 2006 from
Gisenyi, Mashyuza, and Lake Kivu (Newell et al., 2007)

Prospect	SampleID	Sample Type	Date	Temp., C	pH (field)	pH (lab)	Na	K	Ca	Mg	Li
Gisenyi	G (1982)	Water	1982	70.6		6.47	528.8	40.7	37.8	11.1	0.41
Gisenyi	G (2006)	Water	2006	69	7.0	7.03	518.8	39.8	36.4	11.1	0.42
Lake Kivu	LK (1982)	Water	1982	24.6		8.89	116.1	92.7	9.0	82.9	0.04
Lake Kivu	LK (2006)	Water	2006		8.0	9.05	110.9	87.5	8.1	80.1	0.05
Mashyuza	M-1 (1982)	Water	1982	41.8		6.45	287.4	45.4	77.0	51.8	0.90
Mashyuza	M-1 (2006)	Water	2006	33	6.5-7.0	6.76	291.2	45.3	89.5	53.1	0.93
Mashyuza	M-2 (1982)	Water	1982	54.2		6.26	298.9	47.3	72.9	54.0	0.95
Mashyuza	M-2 (2006)	Water	2006	47	6.5	6.72	307.8	48.0	76.0	55.0	0.96

Prospect	SampleID	Sample Type	Date	B	SiO_2	Cl⁻	SO_4	HCO_3	TDS
Gisenyi	G (1982)	Water	1982	5.01	105.8	234.0	44.0	1122.7	
Gisenyi	G (2006)	Water	2006	0.55	58.5	236.8	62.1	1137.3	2101.90
Lake Kivu	LK (1982)	Water	1982	4.00	19.0	36.2	18.0	799.3	
Lake Kivu	LK (2006)	Water	2006	0.10	7.9	32.0	22.3	796.1	1145.09
Mashyuza	M-1 (1982)	Water	1982	1.50	84.7	120.9	48.0	1049.5	
Mashyuza	M-1 (2006)	Water	2006	1.18	50.2	141.0	50.5	1115.8	1836.78
Mashyuza	M-2 (1982)	Water	1982	4.50	75.1	128.0	46.0	1061.7	
Mashyuza	M-2 (2006)	Water	2006	1.07	48.3	137.9	55.3	1122.6	1855.00

3.3 Geophysical exploration

Geophysical measurements started in August 2008 and 40 sites (Figure 6) were identified for both Magneto-telluric (MT) and Transient Electromagnetic (TEM) measurements. Initial results from geochemistry, surface manifestations and extensive volcanic activity in the remote area suggested that here could be a heat source for a viable geothermal system. The detailed surface geophysical investi-

FIGURE 6: Location of the proposed sites for MT and TEM measurements (Onacha, 2008)

gations should, therefore, be instrumental in evaluating the existence of a geothermal system, its

extent, size and potential. The MT and TEM data will be acquired both on a profile and on a grid so as to evaluate the dimensions of the geothermal resource if it exists in the study area. The combined results of the detailed surface exploration studies will be used to locate the best sites for drilling exploration wells (Onacha, 2008). When this report is written, the results of the combined TEM and MT survey were not yet available.

4. INTEGRATED SINGLE-FLASH ORC POWER PLANT

In order to extract from the geothermal fluid as much energy as possible, an integrated single-flash ORC cycle has been proposed for this study. The overall circuit for such a plant is shown in Figure 7 (for terminology see Nomenclature). The geothermal steam first drives a back-pressure steam turbine and is then condensed in the pre-heater, P. Thus, the exit steam from the back-pressure turbine is used as a preheating source and the separated brine is used as a vaporising source. The non-condensable gases (NCGs) are extracted and compressed by the vacuum pump and then mixed with condensate and brine before reinjection. There is a trade-off involved in this arrangement in that the steam expansion process is cut short relative to the combined flash-binary plant. The work that would be obtained from the lower-pressure stage of the turbine is forfeited in favour of using the heat of condensation (from process 3-4) to vaporise the working fluid in the binary loop (Dipippo, 2008). The modelling of the back-pressure cycle is done in Section 4.1 and that of ORC in Section 4.2.

4.1 Single-flash cycle

In a single-flash power plant operation, geothermal fluid is transported from the wells to the separation station. After being separated from the brine, the steam passes through a mist eliminator and is then

FIGURE 7: Integrated single-flash ORC power plant

piped to the turbine to generate electricity. After turning the turbine, the steam is either condensed in a condensing turbine power plant or exhausted to the environment by a back-pressure turbine.

4.1.1 Separator

A geothermal separator is placed between the wellhead and the turbine in order to separate the steam from the brine. A two-phase mixture of geothermal water and steam is sprayed into the separator. As water is heavier than steam, the separated water falls down and the steam flows on to the top of the separator. Poor separating results in scales, clogging, and the erosion of various components that reduce power plant efficiency; they also significantly increase maintenance expenditure and can even destroy turbines. High pressure drop across separators reduces a depleting reservoir's ability to deliver steam to the turbine. Better separators and mist eliminators can help reducing the number of wells drilled to support the plant, enhance resource recovery, and reduce operating and maintenance cost (US Department of Energy, 2008). Figure 8 shows a typical vertical separator.
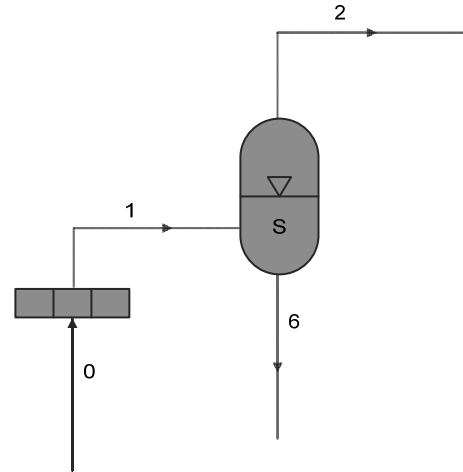
FIGURE 8: Vertical steam separator

The flashing process is assumed isenthalpic as no work is extracted during this process, and the separation process is assumed to be isobaric. The flashing process is justified by the following equation:

$$h_0 = h_1 \tag{1}$$

The steam fraction (x_1) in the two-phase flow to the separator is given by the following relationship:

$$x_1 = \frac{h_1 - h_6}{h_2 - h_6} \tag{2}$$

where h_0 = Enthalpy of the reservoir (kJ/kg);
        h_1 = Wellhead enthalpy (kJ/kg);
        h_2 = Enthalpy of the saturated steam (kJ/kg);
        h_6 = Enthalpy of the saturated brine (kJ/kg);
        x_1 = Steam fraction.

The mass of the steam and the brine are, respectively, given by the following equations:

$$\dot{m}_2 = x_1 \dot{m}_1 \tag{3}$$

$$\dot{m}_6 = (1 - x_1)\dot{m}_1 \tag{4}$$

where $\dot{m}_1$ = Total mass flow rate from the well (kg/s);
        $\dot{m}_2$ = Mass flow rate of the steam (kg/s);
        $\dot{m}_6$ = Mass flow rate of the brine (kg/s).

4.1.2 Mist eliminator

The mist eliminator (steam dryer or steam scrubber) is a tank consisting of filters that catch the remaining condensed water drops in the steam and any other solid dust that could travel together with the steam.

4.1.3 Steam turbine

A steam turbine is a mechanical device that converts thermal energy from pressurised steam into useful mechanical work. In this study, a back-pressure (or an atmospheric exhaust) steam turbine has been chosen. Back-pressure turbines are simplest, and they have the lowest capital cost of all geothermal cycles. With this type of plant, steam is separated from the geothermal discharge and fed through a conventional axial flow steam turbine that exhausts directly into the atmosphere. Such machines consume about twice the steam per kW output (for the same inlet pressure) as condensing plants. Nevertheless, they have their uses as pilot plants with the added advantage of being started without the need for an external power supply (Hudson, 1995). A simplified schematic diagram of a back-pressure plant is shown in Figure 9.
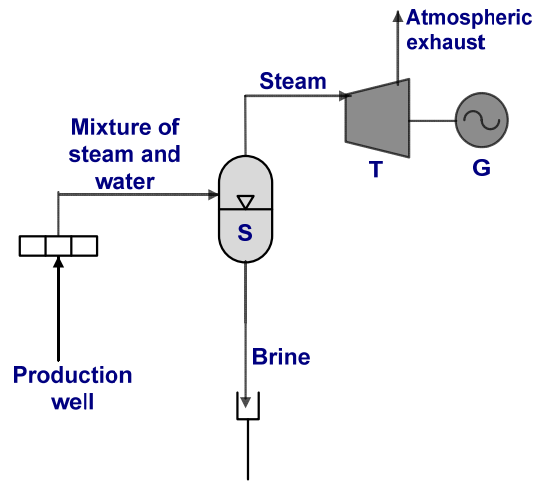
FIGURE 9: Simplified schematic diagram of an atmospheric exhaust cycle

As shown in Figure 7, instead of letting the turbine exhaust to the environment, the exit turbine has been connected to the pre-heater of the ORC cycle; this has given a successful result as will be discussed in Section 5 of this study.

Figure 10 shows a T-s diagram of the single-flash back-pressure turbine where its exit is connected to the pre-heater of an ORC unit. The expansion in the turbine is isentropic for an ideal turbine and (as can be seen in Figure 10) for process increases from 2 to 3_s. But in a real turbine, the process is not isentropic and increases from 3_s to 3 during expansion. The isentropic efficiency of the turbine is given by:
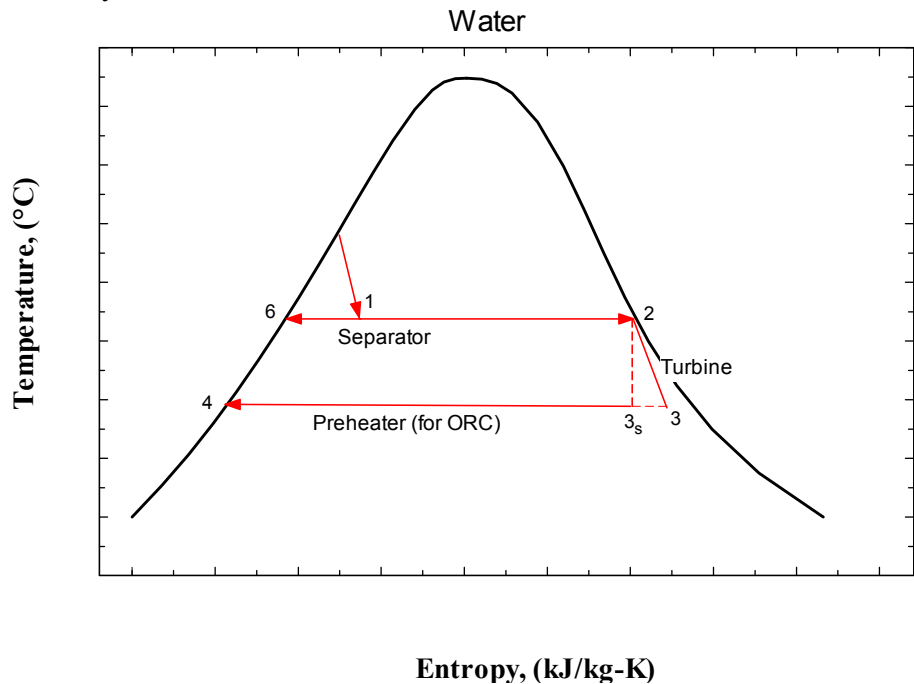

FIGURE 10: T-s diagram for a back-pressure turbine connected to an ORC vaporiser

$$\eta_t = \frac{h_2 - h_3}{h_2 - h_{3s}} \tag{5}$$

In many cases, the isentropic efficiency is assumed to be 85%; Equation 5 is used to calculate the enthalpy at state 3.

The mechanical power output from the turbine is given by:

$$W = \dot{m}_2(h_2 - h_3) \tag{6}$$

As earlier said, the enthalpy at state is not known. The output power is given by Equation 7.

$$W = \eta_t \dot{m}_2 (h_2 - h_{3s})$$

(7)

where W = Mechanical power (kJ/s);
       h_{3s} = Ideal enthalpy at the turbine exit (kJ/kg);
       h_3 = Real enthalpy at the turbine exit (kJ/kg);
       η_t = Isentropic efficiency of the turbine.

4.1.4 Non-condensable gases

Non-condensable gases (NCGs), which are always present in geothermal steam, cause power plant inefficiencies that result in increased steam consumption and higher operation costs. In many steam fields over the world, the NCG content exceeds 5% in the steam, a level at which steam consumption and costs increase rapidly as a function of the NCG concentration. However, it is recommended to expand the steam in a back-pressure turbine if the NCG ratio in the steam exceeds 10%; otherwise, the extraction of the NCGs will not be economically advisable because of two factors: Firstly, the steam jet ejectors and vacuum pumps that evacuate NCGs from the turbine condenser require more steam and electric power for their operation. Secondly, higher gas levels lead to higher condenser pressure (that is, higher turbine outlet pressure), thus yielding a lower power output per unit of steam feeding the turbine. The costs of the cooling water system and parasitic power losses within the circulating pump and tower fans increase. These problems can be alleviated when a re-boiler system is used for steam cleaning, that is the re-boiler system removes NCGs from steam upstream of the turbine. This is accomplished with a simple standard heat exchanger (the re-boiler) within which geothermal steam is condensed and its NCGs components are discharged. The heat of condensation is used to produce NCG-free steam by evaporation of clean water. The source of clean water is the condensate recovered from the condensing steam (Cory et al., 1996).

If a vacuum pump is used for extracting the NCGs, its power consumption is given by the following equation (Siregar, 2004):

$$P_{Vpump} = \left(\frac{\gamma}{\gamma - 1} \right) \frac{m_g R_u T_{cond}}{\eta_{Vpump} M_{gas}} \left[\left(\frac{P_{atm}}{P_{cond}} \right)^{\left(1 - \frac{1}{\gamma}\right)} - 1 \right]$$

(8)

where P_{vpump} = Power of the vacuum pump (kW);
       γ = $C_{p.gas}/C_{v.gas}$;
       m_g = Mass flow rate of the NCGs (kg/s);
       R_u = 8.314 kJ/ (kmol K), universal gas constant
       T_{cond} = Temperature of the condensate (K);
       η_{Vpump} = Efficiency of the pump;
       M_{gas} = Molar mass of the mixture of the NCGs;
       P_{atm} and P_{cond} = Atmospheric and condenser pressures, respectively (bar).

4.1.5 Reinjection

Geothermal reinjection involves returning some, or even all, of the water produced from a geothermal reservoir back into the geothermal system after energy has been extracted from the water. In some instances water of a different origin is even injected into geothermal reservoirs. Reinjection is considered an important part of comprehensive geothermal resource management, as well as an essential part of sustainable and environmentally friendly geothermal utilisation. Reinjection provides an additional recharge to geothermal reservoirs and, as such, counteracts pressure draw-down due to production and extracts more of the thermal energy from reservoir rocks than conventional utilisation. Reinjection will therefore, in most cases, increase the production capacity of geothermal reservoirs,

which counteracts the inevitable increase in investment and operation costs associated with reinjection. It is likely to be an economical way of increasing the energy production potential of geothermal systems. Without reinjection, the mass extraction, and hence energy production, would only be a part of what it is now in many geothermal fields. Reinjection is also a key part of all enhanced, or engineered, geothermal system (EGS) operations. Some operational dangers and problems are associated with reinjection. These include the possible cooling of production wells, often because of short-circuiting and cold-front breakthrough, and scaling in surface equipment and injection wells because of the precipitation of chemicals in the water. Injection into sandstone reservoirs has, furthermore, turned out to be problematic. Because of this, extensive testing and research are prerequisites to successful reinjection operations. This includes tracer testing, which is the most powerful tool available to study the connections between reinjection wells and production wells (Axelsson, 2008). To avoid silica scaling, it is recommended to "respect" the silica solubility curve (shown in Figure 11) during the design process of the power plant.
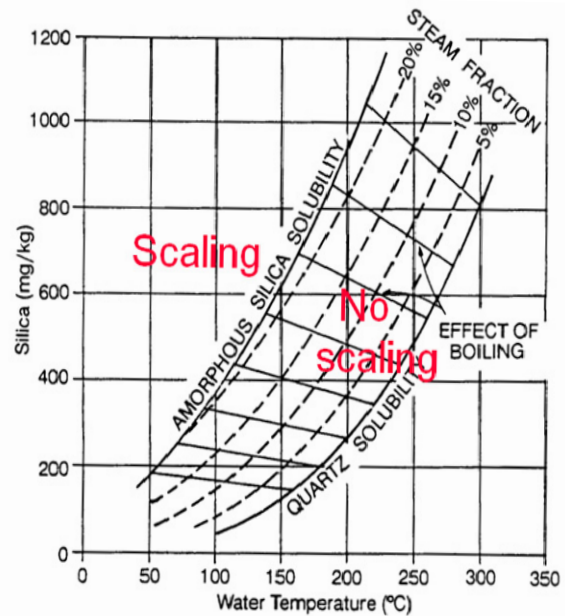
FIGURE 11: Silica solubility curve
(Thórhallsson, 2008)

4.2 Organic Rankine cycle (ORC)

4.2.1 Principle of an ORC power plant

An ORC cycle is an ideal cycle for vapour power plants and includes a boiler, a turbine, a condenser and a pump and it operates in a closed-loop cycle. In many cases, a regenerator is added in order to increase the efficiency of the cycle. The working principle of an ORC power plant is similar to that of the conventional steam cycle; the main difference is the use of organic substances, like hydrocarbons instead of water or steam as the working fluid. Within this process, energy is extracted from geothermal fluid and a secondary fluid with a much lower boiling point than water passes through a heat exchanger. Heat from the geothermal fluid causes the secondary fluid to flash to vapour, which then drives the turbine. The cycle is based on the following series of internally reversible processes as summarised in Figure 12:
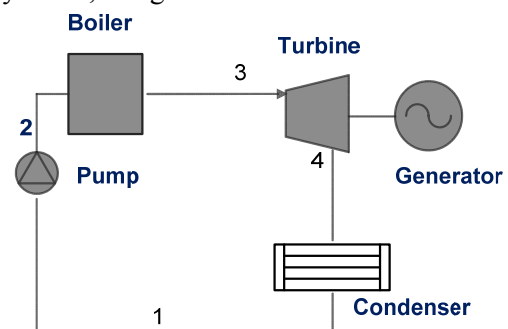
FIGURE 12: Principle of
the ORC cycle

Process 1-2: Isentropic compression in a pump;
Process 2-3: Constant pressure heat addition in a boiler;
Process 3-4: Isentropic expansion in a turbine; and
Process 4-1: Constant pressure heat rejection in a condenser.

Working fluid enters the pump at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The temperature of the fluid increases somewhat during this isentropic compression process due to a slight decrease in the specific volume as shown in Figure 13. Then, the fluid enters the boiler at state 2 and leaves it at state 3 as saturated vapour. The saturated
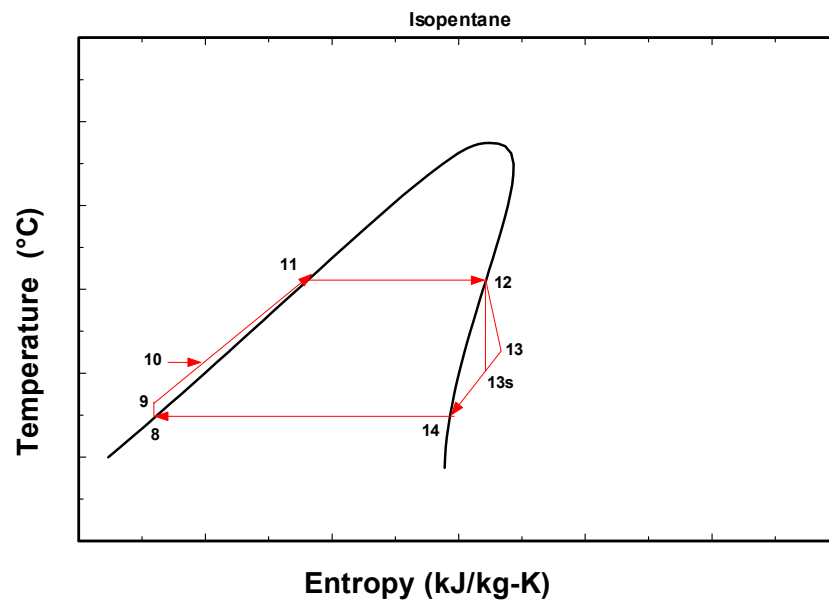
FIGURE 13: T-s diagram for an ORC cycle using
isopentane as working fluid

vapour at state 3 enters the turbine and produces work by rotating the shaft connected to an electric generator. The temperature and the pressure of the steam drop during this process to the values at state 4, where vapour enters the condenser. Then, the vapour is condensed at constant pressure in the condenser, which is basically a large heat exchanger, by rejecting heat to a cooling medium. The heat transferred between the geothermal fluid and the working fluid that takes place in the boiler undergoes two stages: preheating and vaporising; and sometimes superheating. For different types of heating sources (steam and brine for example), these pieces of equipment are independent of one another (Dipippo, 2008). Boiling of the working fluid can commence either in the pre-heater or in the vaporiser.

4.2.2 Working fluid

The selection of a working fluid has great implications for the performance of the binary power plant. While there are many choices available, there are also many constraints on that selection that relate to the thermodynamic properties of the fluids as well as consideration of health, safety and environmental impacts. Table 2 shows properties of some of the possible working fluids.

TABLE 2: Thermodynamic properties of some candidate working fluids (Dipippo, 2008)

Fluid	Formula	Critical temp. T_c (°C)	Critical press. P_c (MPa)	Saturation press. P_s@300K (MPa)	Saturation press. P_s@400K (MPa)
Propane	C_3H_8	96.95	4.236	0.9935	n.a
i-Butane	i- C_4H_{10}	135.92	3.685	0.3727	3.204
n-Butane	C_4H_{10}	150.8	3.718	0.2559	2.488
i-pentane	i-C_5H_{12}	187.8	3.409	0.09759	1.238
n-pentane	C_5H_{12}	193.9	3.240	0.07376	1.036
Ammonia	NH_3	133.65	11.627	1.061	10.3
Water	H_2O	374.14	22.089	0.003536	0.24559

Clearly, all of the fluid candidates have critical temperatures and pressures far lower than water. Furthermore, since the critical pressures are reasonably low, it is reasonable to consider supercritical cycles for the hydrocarbons. This allows a better match between the brine cooling curve and the working fluid heating boiling line, reducing the thermodynamic losses in the heat exchangers. Another important factor of binary candidate fluids is the shape of the saturated vapour curve. For water, this curve has a negative slope everywhere, as shown in Figure 14-A. As shown in Figure 14-B, for certain hydrocarbons and refrigerants, the curve has a certain positive slope for portions of the saturated line. That is, there exist a local minimum in the entropy at some low temperature, T_m, and a local maximum entropy at the high temperature, T_M. Retrograde fluids include normal- and isobutane

and normal- and isopentane. Their T_m are lower than any other temperatures encountered in a geothermal power plant. This has major implications for the ORC cycles.

From Figure 14, it is clear that retrograde fluids allow the expansion from the saturated vapour line into a superheated region, process f-g, avoiding any moisture during the turbine expansion process which is not the case for a normal fluid like water where the turbine is expanded in the two-phase flow region as illustrated by process b-c. In the last case, the quality of steam must be controlled and it is recommended that the quality should not be less than 85% (Pálsson, 2008). For this study, isopentane has been selected as the working fluid.

4.2.3 Recuperator

Having an internal heat exchanger (recuperator) between the super-heated vapour at the turbine outlet and the compressed working fluid after condensation can increase the efficiency of the cycle. The recuperator heats the compressed working fluid before it enters the

FIGURE 14: T-s diagrams of saturated vapour curves;
a) normal fluids; and b) retrograde fluids

boiler to better utilise the geothermal fluid (Karlsdóttir, 2008). The balancing equation for the recuperator is given as:

$$\dot{m}_{wf}(h_{10} - h_9) = \dot{m}_{wf}(h_{13} - h_{14}) \qquad (9)$$

4.2.4 Pre-heater

A pre-heater is simply a heat exchanger that facilitates the exchange of heat between two fluids that are at different temperatures while keeping them from mixing with each other. Heat transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. The rate of heat transfer between two fluids in a heat exchanger depends on the magnitude of the temperature difference between them which varies along the heat exchanger. To calculate the area of a heat exchanger, in most cases logarithmic mean temperature difference (LMTD), which is an equivalent mean temperature difference between the two fluids, is used. However, the LMTD is only applied if both fluids are in fixed state and if condensation or boiling takes place, as is the case in the pre-heater of this current study; the exchanger must be divided into different parts according to when the states change (Figure 15). The place in the heat exchanger where the two fluids experience the minimum temperature difference is called the pinch-point. There are many types of heat exchangers; in this study the counter flow heat exchanger is chosen. Note that in a counter flow heat exchanger,
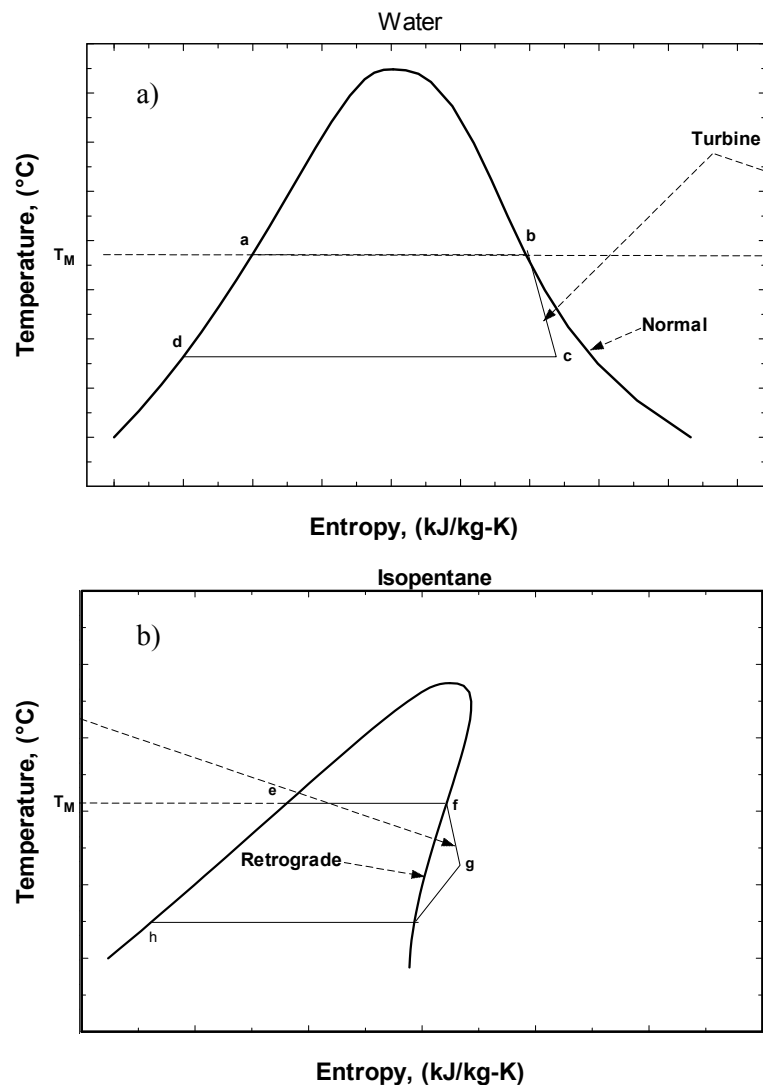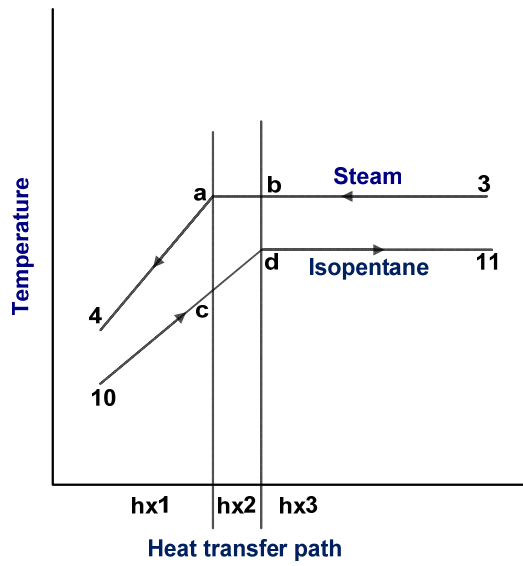
FIGURE 15: Temperature distribution
in a pre-heater

the fluids enter the heat exchanger from opposite ends, and the outlet temperature of the cold fluid in this case may exceed the outlet temperature of the hot fluid.

The energy conservation law allows for determining the status of the heat exchanger by applying the following:

$$\dot{m}_{wf}(h_{11} - h_{10}) = \dot{m}_2(h_3 - h_4) \tag{10}$$

where $\dot{m}_{wf}$ = Mass of working fluid (kg/s);
$\dot{m}_2$ = Mass of steam (kg/s);
h_3 = Enthalpy of the exit steam when entering the pre-heater (kJ/kg);
h_4 = Enthalpy of the steam when exiting the pre-heater (kJ/kg);
h_{10} = Enthalpy of working fluid when entering the pre-heater (kJ/kg);
h_{11} = Enthalpy of working fluid when exiting the pre-heater (kJ/kg).

Because there are changes of state in the pre-heater, the heat exchanger is divided into three parts and the respective areas are calculated as follows:

$$\dot{Q}_{hx1} = U_{hx1}A_{hx1}(LMTD)_{hx1} \tag{11}$$

$$\dot{Q}_{hx2} = U_{hx2}A_{hx2}(LMTD)_{hx2} \tag{12}$$

$$\dot{Q}_{hx3} = U_{hx3}A_{hx3}(LMTD)_{hx3} \tag{13}$$

where $U_{hx1,hx2,hx3}$ = Overall heat transfer coeff. for heat exchangers 1, 2, 3, respectively (kJ/m² °C);
$A_{hx1,hx2,hx3}$ = Area of the heat exchangers 1, 2, 3, respectively (m²);
$LMTD_{hx1,hx2,hx3}$ = LMTD of the heat exchangers 1, 2, 3, respectively (°C).

The logarithmic mean temperature difference, *LMTD*, is calculated as follows:

$$LMTD = \frac{GTTD - LTTD}{\ln\left(\frac{GTTD}{LTTD}\right)} \tag{14}$$

where *GTTD* = Greater terminal temperature difference of the heat exchanger;
LTTD = Lower terminal temperature difference of the heat exchanger.

When *LTTD* equals *GTTD*, the *LMTD* is taken to be equal to *LTTD* or *GTTD* (Páll Valdimarsson, pers. comm.)

4.2.5 Evaporator

The evaporator is a heat exchanger in which vaporisation takes place. The energy conservation law allows for determining the energy exchanged in the evaporator as follows:

$$\dot{m}_{wf}(h_{12} - h_{11}) = \dot{m}_6(h_6 - h_7) \tag{15}$$

4.2.6 ORC turbine analysis

The analysis of the turbine in the ORC cycle is the same as for the steam turbine. With the usual assumption of negligible potential and kinetic energy terms together with steady, adiabatic operation, the power is found from:

$$W = \dot{m}_{wf}(h_{12} - h_{13s}) \tag{16}$$

As in the case of a steam turbine, the binary turbine is assumed to have an isentropic efficiency of 85% and then Equation 16 becomes:

$$W = \eta \dot{m}_{wf}(h_{12} - h_{13s}) \tag{17}$$

with W = Mechanical power output from ORC turbine (kJ/s);
h_{13s} = Ideal enthalpy of the working fluid at turbine exit (kJ/kg);
h_{13} = Real enthalpy of the working fluid at turbine exit (kJ/kg);
h_{12} = Enthalpy of the working fluid at turbine inlet (kJ/kg);
η = Isentropic efficiency of the turbine.

4.2.7 Condenser analysis

A condenser is a heat exchanger which condenses a substance from its gaseous state to its liquid state. In this process, the latent heat is given up by the gaseous substance and is transferred to the condenser coolant. The primary purpose of a condenser is to condense the exhaust steam (or vapour) from a turbine to obtain maximum efficiency, and also to convert the turbine exhaust steam (or vapour) into pure liquid so that it may be reused or re-injected. By condensing the exhaust steam (or vapour) from a turbine, the fluid pressure drop increases the amount of heat available for conversion to mechanical power. Most of the heat liberated due to condensation of the exhaust steam is carried away by the cooling medium (water or air) used by the surface condenser. In this report, the cooling medium is water and the cooling process is shown in Figure 16. Point 13 has though higher temperature than shown, which can be misleading. The reason is that the isopentane must be cooled before it starts to condense.
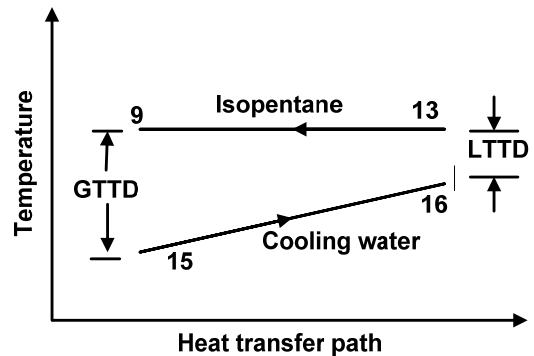

FIGURE 16: Condensing process of the working fluid

4.2.8 Cooling tower

The cooling tower must be designed to accommodate the heat load from the condensed working fluid (isopentane in this study). The cooling water is pumped from the condenser and sprayed into the tower where it falls through an air stream drawn into the tower by motor-driven fans at the top of the tower. The ambient air enters with a certain amount of water vapour determined by the relative humidity. The internal process involves the exchange of both heat and mass between the air and the water (Dipippo, 2008).

From Figure 7, the flow rate of the cooling water is given by:

$$\dot{m}_{wf}(h_{13} - h_9) = \dot{m}_{cw}(h_{16} - h_{15}) \tag{18}$$

where $\dot{m}_{cw}$ = Mass flow of the cooling water (kg/s);
h_{15} = Enthalpy of the cooling water at the condenser inlet (kJ/kg);
h_{16} = Enthalpy of the cooling water at the condenser exit (kJ/kg);
h_9 = Enthalpy of the working fluid at the exit of the condenser (kJ/kg).

4.2.9 Pumps analysis

In the circuit shown in Figure 7, three pumps are envisaged, P_1, P_2 and P_3, for pumping the condensate, the working fluid and the cooling water, respectively. Assuming an isentropic process in the pumps, the power of the pumps, respectively, is given by:

$$P_1 = \dot{m}_2(h_5 - h_4) = \dot{m}_2(h_{5s} - h_4)/\eta_t \qquad (19)$$

$$P_2 = \dot{m}_{iso}(h_{10} - h_9) = \dot{m}_{iso}(h_{10s} - h_9)/\eta_t \qquad (20)$$

$$P_3 = \dot{m}_{cw}(h_{15} - h_{14}) = \dot{m}_2(h_{15s} - h_{14})/\eta_t \qquad (21)$$

where h_{5s} = Ideal enthalpy of the condensate at the exit of pump 1 (kJ/kg);
h_5 = Real enthalpy of the condensate at the exit of pump 1 (kJ/kg);
h_{10s} = Ideal enthalpy of the working fluid at the exit of pump 2 (kJ/kg);
h_{10} = Real enthalpy of the working fluid at the exit of pump 2 (kJ/kg);
h_{14} = Enthalpy of the cooling water at the inlet of pump 3 (kJ/kg);
h_{15s} = Ideal enthalpy of the cooling water at the exit of pump 3 (kJ/kg);
h_{15} = Real enthalpy of the cooling water at the exit of pump 3 (kJ/kg);
η_t = Efficiency of the pumps (assumed the same for all three).

5. OPTIMISATION OF THE NET POWER OUTPUT

As said in the introduction of this report, all data necessary for this study are not available and some assumptions have been made in order to calculate and optimise the output power from both steam and binary turbines.

5.1 Restrictions of the geothermal reservoir

During the thermo-dynamical modelling of the power plant, the following assumptions regarding the behaviour of the geothermal system were made:

- The reservoir temperature was assumed to be 210°C according to the geothermometer results; the corresponding enthalpy calculated with EES (F-Chart Software, 2002) is about 900 kJ/kg;
- The flow in the well was assumed to be saturated water up to the wellhead;
- The reinjection temperature was assumed to be 110°C;
- The calculations were done using a mass flow of 1 kg/s.

5.2 Power plant equipment

The power plant equipment includes turbines, compressors, pumps and heat exchangers. As turbines, compressors and pumps cannot operate at 100% efficiency, their respective efficiencies should be taken into account when designing the power plant. The assumptions that have been made for isentropic efficiency are given in Table 3.

TABLE 3: Isentropic efficiency of different power plant equipment

Equipment	Isentropic efficiency
Turbines	85%
Compressor	85%
Pumps	75%

The overall heat transfer coefficients used to calculate the total heat exchanger area for each heat exchanger in the modelled power plant are given in Table 4 (Páll Valdimarsson, pers. comm.).

TABLE 4: Overall heat transfer coefficient for various heat exchangers

Fluid	Overall heat transfer coeff., U (W/m^2 °C)
Steam-isopentane (vapour)	2000
Steam-isopentane (liquid)	1500
Condensate-isopentane	800
Brine-isopentane	1200
Water-isopentane	1200
Isopentane-isopentane	1200

- The calculation of heat exchanged in each heat exchanger was done using the LMTD method;
- The cooling water temperature enters the condenser at 15°C and the pinch was assumed to be 5°C for all heat exchange except for the regenerator, where it is 8°C.
- The cooling tower fans were estimated to consume 0.2 kW/kg.
- The condenser pressures in the ORC cycle is 1 bar and the pressure of the exit steam from the back-pressure turbine is 1.5 bar.

5.3 Optimum power output

The optimisation of the maximum net power output for the integrated single-flash ORC cycles has been done using the EES software (F-Chart Software, 2002). The thermo-dynamical variable analysed is the separation pressure, as the exit pressure of the atmospheric exhaust turbine is fixed.   Note that this pressure imposes the ORC working pressure in such a way that the difference in temperature between the working fluid and the brine at the inlet of the evaporator shouldn't be less than the assumed pinch.
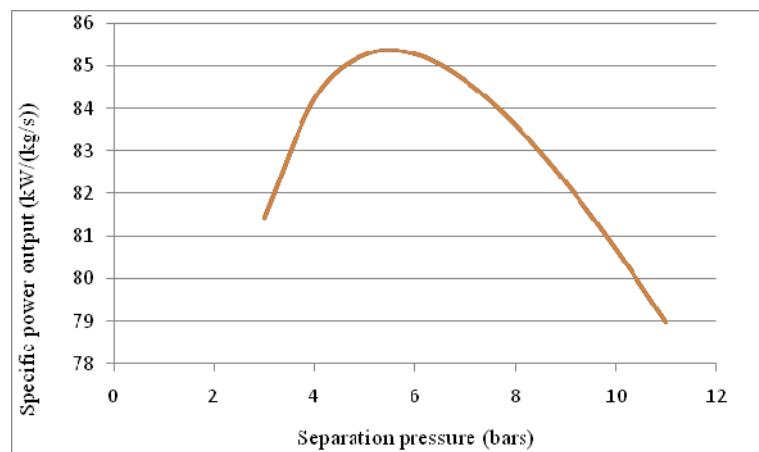

FIGURE 17: Net power output versus separation pressure

The results for the optimised specific net power output for the whole cycle are shown in Figure 17. It is clear that the total net power output from this plant is obtained at a separation pressure of about 5.6 bar when the working fluid in ORC circle is 8 bar and when the exit pressure of the back-pressure turbine is 1.5 bar.

5.4 Results

After obtaining the optimum values of different thermo-dynamical variables, calculations were made using EES (F-Chart Software, 2002), and the results are shown in Table 5.

TABLE 5: Results of thermo-dynamic optimisation

Specific net power output (kW/(kg/s))	Output power from ORC cycle (kW/(kg/s))	Output power from single-flash cycle (kW/(kg/s))	Exergy efficiency (%)	Thermal efficiency (%)
85.4	70.5	23	47.5	10.4

The first law and second law of utilisation efficiencies, respectively, are calculated as follows:

$$\eta_{th} = \frac{W}{Q_{in}}$$ (22)

$$\eta_u = \frac{W_{net}}{E}$$ 23)

5.5 Comparison with a single-flash ORC bottoming power plant

A single-flash plant with an ORC bottoming power plant (Figure 18), is similar to the integrated power plant; the difference is that in an ORC bottoming cycle, the steam is condensed and the ORC unit uses the brine only. EES modelling results for an integrated single-flash ORC power plant and a single-flash ORC bottoming power plant are shown in Table 6.

FIGURE 18: Schematic diagram of a single-flash power plant with an ORC bottoming unit;
this power plant is included in this study for purposes of comparison.

TABLE 6: Comparison of an integrated single-flash ORC (1) and
single-flash ORC bottoming (2) power plant

	Specific net power output (kW/(kg/s))	Output power from ORC cycle (kW/(kg/s))	Output power from single-flash cycle (kW/(kg/s))	Exergy efficiency (%)	Thermal efficiency (%)
1	85.4	70.5	23	47.5	10.4
2	83.7	28	69.4	46.5	9.7

6. CONCLUSIONS

A back-pressure turbine is recommended for a pilot geothermal power plant in Rwanda because it is the simplest, easiest and fastest in installation. It has also the lowest capital cost of all geothermal power plants which makes it feasible for pilot projects. As shown in Table 5, under the same conditions of flow rate, well head pressure and reinjection temperature, the back-pressure turbine produces about a third of the power from the condensing turbine, but this concern can be eliminated by integrating an ORC unit after running the plant for some time and being sure that the geothermal field is economically exploitable. The integrated single-flash ORC cycles will then produce more than 85 kW/(kg/s) while the condensing single-flash power plant with ORC bottoming cycles can produce only 83 kW/(kg/s).

Using the EES software, the net power output together with condenser and heat exchangers have been calculated and the results show that it is possible to extract more than 85 kW from 1 kg/s of geothermal fluid having enthalpy of 900 kJ/kg. This type of power plant is also associated with very low emissions and is mostly advised where environmental concerns are significant. However, this study was limited to technical designs. A cost analysis between the two types of plants is essential for determining which one is economically more feasible.

NOMENCLATURE IN FIGURES

C = Condenser;
CT = Cooling tower;
E = Evaporator;
G = Generator;
$G_{1,2}$ = Generator 1 and 2, respectively;
NCG-Unit = Non condensable gas unit;
P = Pre-heater;
$P_{1,2,3}$ = Pumps 1,2,3, respectively;
R = Recuperator;
S = Separator;
T = Turbine;
$T_{1,2}$ = Turbines 1 and 2, respectively;
WH = Well head.

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