INCREASED EXERGETIC EFFICIENCY BY USING A BACK PRESSURE TURBINE FOR HIGH WELLHEAD PRESSURES AT HELLISHEIDI GEOTHERMAL POWER PLANT

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ABSTRACT

A study of utilizing geothermal wells with high wellhead pressures by using a back-pressure turbine as a topping unit, operating between 17.5 bar-a inlet pressure and 10 bar-a exhaust pressure, was applied for the existing Hellisheidi geothermal power plant. The study was carried out to determine the electrical power output that could be generated and the loss of exergy throughout the process. Using engineering equation solver (EES) and manual calculations, an output of 7.75 MW of electricity was calculated for the topping unit with 98% energy efficiency and 81.76% exergetic efficiency for a back-pressure turbine with an isentropic efficiency of 85%. The total exergy available in the steam is 89 MW while the exergy available in the steam exiting the turbine is 79 MW. Due to irreversibilities, 1.8 MW is destroyed in the turbine. In comparison, 6 MW of exergy is destroyed if a throttle valve is used instead of a back-pressure turbine to reduce the steam pressure from 17.5 bar-a down to 10 bar-a in order to be compatible with the inlet pressure at Hellisheidi geothermal power plant.

1. INTRODUCTION

1.1 General background

Renewable energy resources are a priority in today's global energy outlook due to the drastically increasing prices of fossil fuels and their high gaseous emissions. Geothermal energy, the thermal energy stored in the earth's crust, is considered to be an attractive renewable energy option.

Geothermal energy has been used since ancient times in many countries in the world, but was mainly used for bathing, cooking and washing. More recently, geothermal energy has been utilised for electricity production, district heating, green house farming, fish farming, drying, snow melting, and many other uses (Cordova, 2013).

Commercial electricity power generation from geothermal power plants started in 1914 when a 250 kW unit at Larderello in Italy provided electricity to the nearby cities of Volterra and Pomarance (DiPippo, 2007). Since then, many geothermal power plants have been installed around the world.

In the author's opinion, there can be many reasons for producing electricity from geothermal resources. At many locations, it is technically, economically and environmentally feasible, due to high availability, a high capacity factor, the ability to be used as a base load, low operational and maintenance cost and low CO_2 emissions.

Geothermal power plants can be divided into two main groups according to the enthalpy of the wells: steam cycles which are used at higher well enthalpies, and binary cycles for lower enthalpies.

In the steam cycles the geothermal fluid boils partly due to flashing in the wells; then the steam is separated from the brine and finally expanded in a turbine, producing positive work. Usually, the brine is rejected to the environment (e.g. re-injected into the reservoir or released at the surface), or it can be flashed again at a lower pressure before final disposal. The first type of cycle in which the fluid boils and the brine is separated at a high pressure is called the Single Flash cycle (SF), and the second one in which the brine is flashed again at a lower pressure is called a Double Flash cycle (DF) (Valdimarsson, 2011).

A binary cycle uses a secondary working fluid in a closed power generation cycle. A heat exchanger is used to transfer heat from the geothermal fluid to the working fluid, and the cooled brine is then rejected to the environment. The Organic Rankine Cycle (ORC) and Kalina cycle, used to exploit low to medium enthalpy geothermal resources, are both binary cycles with different types of secondary working fluids (Valdimarsson, 2011).

Geothermal resources have been classified as low, intermediate or high enthalpy resources according to their reservoir temperatures. The temperature ranges used for these classifications are arbitrary and they are not generally agreed upon (Lee, 1996). Temperature is used as the classification parameter because it is the easiest to measure and understand. In addition, temperature or enthalpy alone can be ambiguous in defining a geothermal resource because two independent thermodynamic properties are required to define the thermodynamic state of fluid. Geothermal energy is already in the form of heat, and from the thermodynamic point of view, work is more useful than heat because not all the heat can be converted to work. Therefore, geothermal resources should be classified by their exergy, a measure of their ability to do work. The basic concept of exergy is that it is the maximum work (or power) output that could theoretically be obtained from a substance at specified thermodynamic conditions relative to its surroundings (DiPippo, 2007). The exergy analysis is based on the Second Law of thermodynamics. Exergy analysis is applied for all components of a power plant in order to identify the exergy efficiencies, sometimes called Second Law efficiencies or utilization efficiencies.

1.2 Area of study

This study, to analyse the exergy balance of pressure reduction by the installation of a back pressure turbine as a topping unit in an existing power plant setup, was applied to the Hellisheidi geothermal power plant. The geothermal power plant, (Figure 1), is situated in the Hengill area, an active volcanic ridge in southwest Iceland. The plant´s purpose is to meet an increasing demand for electricity and hot water for space heating in the industrial and domestic sectors. The maximum production capacity is 303 MW of electricity and currently 133 MW of thermal energy. Production of electricity began in 2006 when two 45 MW turbines were started, and then in 2007 an additional 30 MW low-pressure turbine was brought online. In 2008, two 45 MW turbines were added, using steam from boreholes in Skardsmýrarfjall among others. The thermal station producing hot water for space heating was introduced in 2010 and is currently producing 133 MW of hot water. In 2011, additional 2 x 45 MW units were installed in a separate power house (Kárason et al., 2012).

Geothermal activity in the Hengill area is connected with three volcanic systems. At least three volcanic eruptions have occurred in the Hengill area in the last 11,000 years, the most recent being 2,000 years ago. The Hengill area is part of the Hengill region, which covers 112 km^2 and is one of the most extensive geothermal areas in Iceland (Reykjavik Energy, 2014).

FIGURE 1: Hellisheidi geothermal power plant
(Power-Technology.com, 2014)

The idea is to install a back pressure turbine unit into the existing system instead of reducing pressure by using a throttling valve to utilize the geothermal energy of four new wells that operate at a high pressure and are located in Hverahlíd geothermal field, which is at a 5 km distance from Hellisheidi power plant. They are connected to the Hellisheidi plant to maintain the production of both electricity and hot water due to an annual 2.3% steam flow decline, due to pressure drawdown, from the primary reservoir utilized for the plant. Since the wells operate at a higher pressure than other wells connected to the plant, the pressure must be decreased by some process before the steam can be utilised in the existing turbines. It has become clear that the cost of connecting those four wells with high pressure to a back pressure turbine, and then to connect the exhausted steam to the existing power plant at Hellisheidi, is significantly less than the cost of drilling new wells to compensate for the pressure drop in the system. Another reason is to observe and study the behaviour of the Hverahlíd reservoir for a period of years in order to gain operational experience of the reservoir; thus, if the geothermal energy production from those wells is stable and steady, then the idea of implementing a new power plant in the Hverahlíd area would be an option.

1.3 Background review

The design inlet pressure for a condensing steam turbine is often below 8 bar-a in order to meet the estimated pressure drop until the geothermal reservoir reaches steady state. However, some of the geothermal well fields produce steam at much higher pressures than required for the condensing steam turbine, which could be considered unused energy. In this situation, the steam pressure must be reduced to the required inlet pressure of the turbine. Usually, the inlet pressure of the condensing steam turbines is restricted and cannot be increased; therefore, the idea of using a back-pressure turbine to extract this unused energy and to reduce the pressure becomes very attractive.

The technology of using a back-pressure turbine to extract energy from the excess steam pressure in condensing turbine geothermal power plants, which is sometimes called a topping unit, has been used in Leyte geothermal optimization project in the Philippines since 1997. It was done by ORMAT as an EPC turnkey contract. This project consists of four individual power plants, three of which have been using this technology of a topping unit, and the fourth using another technology called a bottoming unit. The total net power gained by using this technology is 49 MW which represents almost 10% of the total installed capacity of the four power plants of 502.5 MW. The net power gained is composed of 35.65 MW from topping units and 13.35 from the bottoming unit. The three power plants, which used topping units are: Tongonan power plant, Mahanagdong A and Mahanagdong B (Kaplan and Schochet, 2000).

Tongonan topping plant
The total installed capacity of the main Tongonan power plant was 112.5 MW. The inlet pressure of the plant is 6.83 bar-a, and the steam pressure from the well field is 11.14 bar-a. The ORMAT topping

units were added to generate maximum power while reducing the pressure from 11.14 bar-a to 6.83 bar-a. The topping unit consists of two 3.25 MW back-pressure turbines with high efficiency and reliability and simple construction. Each turbine is directly coupled to the opposite side of a common generator. The Tongonan topping plant consists of 3 topping units, generating 16.95 MW net power (Kaplan and Schochet, 2000).

Mahanagdong A topping plant
The total installed capacity of the main Mahanagdong A power plant is 120 MW. The inlet pressure of the plant is 6.83 bar-a while the resource pressure is 10.8 bar-a. The ORMAT topping units were added to generate maximum power while reducing the pressure from 10.8 bar-a to 6.8 bar-a. The Mahanagdong A topping tlant consists of 2 topping units generating 12.45 MW net output (Kaplan and Schochet, 2000).

Mahanagdong B topping plant
The total installed capacity of the main Mahanagdong B power plant is 60 MW. The inlet pressure of the plant is 6.83 bar-a and the resource pressure is 10.8 bar-a. The ORMAT topping unit was added to generate maximum power while reducing the pressure from 10.8 bar-a to 6.8 bar-a. The Mahanagdong B topping plant consists of one topping unit generating 6.25 MW net (Kaplan and Schochet, 2000).

The total net power gained by the three topping plants is 35.65 MW in cost effective power (Kaplan and Schochet, 2000).

This technology has also been used in Wairakei power project, New Zealand since 1956 when it was redesigned by replacing the machines of the heavy water distillation plant with new turbines. Wairakei geothermal power plant has a total capacity of 47 MW. The redesign resulted in implementation of a new plant, called B station, beside the old plant, which is called A station, and the addition of new turbines in A station itself.

Wairakei A station
This plant consists of three pressure levels: high pressure, intermediate pressure and low pressure. The high pressure is 13.5 bar-a which enters 2 x 11.5 MW turbines and 2 x 6.5 MW turbines, and exits at 4.45 bar-a as intermediate pressure. This intermediate pressure from the high pressure turbines exhaust is accompanied by steam which comes from the intermediate pressure wells and goes to the intermediate pressure manifold. The steam from the intermediate pressure manifold goes through two branches to the intermediate pressure turbines and to B station. The 2 x 11.2 MW intermediate pressure turbines receive the steam at 4.45 bar-a, and exhaust steam at 1.0345 bar-a. The steam at 1.0345 bar-a enters 4 x 11.2 low pressure turbines which are condensing turbines, designed to exhaust steam under vacuum (Thain and Carey, 2009).

Wairakei B station
The Wairakei B station consists of 4 x 30 MW condensing turbines. These turbines work at an intermediate pressure of 4.45 bar-a inlet pressure and exhaust the steam under vacuum. The turbines also accept the pass-in steam from low pressure at 1.1 bar-a (Thain and Carey, 2009).

The total installed capacity of Wairakei geothermal power plant increased from 47 MW to 193.2 MW, 103.2 MW from A station and 90 MW from B station (Thain and Carey, 2009).

2. TECHNICAL DATA

2.1 Plant description

Hellisheidi power plant is the largest geothermal power plant in Iceland. The maximum capacity is 303 MWe of electricity and the production of hot water is currently 133 MWth.

Electricity generation was started in 2006 when 2 x 45 MW high-pressure turbine units were installed. In 2007, a 33 MW low-pressure turbine was installed and brought online. Additional 2 X 45 MW high-pressure turbine units were installed in 2008. In 2011, 2 x 45 MW high-pressure turbine units were installed in a separate power house. The production of hot water for space heating started in 2010. The high-pressure turbines are working at 8 bars gage pressure and the low pressure turbine works at 2.1 bars pressure (Kárason et al., 2012).

Hellisheidi geothermal power plant is owned and operated by ON Power (Orka Náttúrunnar), ON, which is a new subsidiary company of Reykjavik Energy (Orkuveita Reykjavíkur). ON is a power company that produces electricity, mainly by harnessing geothermal energy, to more than half of the population of Iceland. The company is one of the world leaders in the utilisation of geothermal energy for production of hot water and electricity.

Orkuveita Reykjavíkur has already purchased 2 x 45 MW condensing turbines planned to be commissioned in the Hverahlíd area. The project has been postponed, but 4 production wells have already been drilled and are able to provide steam for 45 MW of electricity production in a condensing unit. ON is connecting these wells to Hellisheidi power plant to gain operational experience with the reservoir. The connection is assumed to be in operation at the beginning of 2016.

After testing and an analysis of the geothermal fluid of the wells in Hverahlíd, it has become clear that the optimal separation pressure is higher than the initial design, indicating 17.5 bar-a instead of 10 bar-a. However, to connect the steam to the Hellisheidi power plant, the pressure has to be dropped to 10 bar-a. Therefore, it is of interest to investigate the feasibility of installing a back pressure turbine at Hverahlíd. The turbines have already been purchased for Hverahlíd Power Plant with 7.5 bar-a turbine inlet pressure; the back pressure turbine is expected to be a permanently installed unit and to be in operation for at least 30-50 years.

2.2 Project description

This study aims to analyse the energy and exergy of a back pressure turbine which could be added in the Hverahlíd area and to compare the benefits of adding this turbine to the benefits of using a throttling valve. The two options for connecting high enthalpy wells are shown in Figure 2.

For scenario one, the steam from high pressure wells goes to the separator through a throttling valve which adjusts the steam pressure to 17.5 bar-a. The separated steam enters the back pressure turbine at 17.5 bar-a of saturated steam and the separated water goes through a water phase pipeline to Hellisheidi power plant for further utilization before it ends in the re-injection system. The exhaust steam from the back pressure turbine at 10 bar-a goes to Hellisheidi Power Plant, reaching there at 9 bar-a due to pressure drop along the pipeline. For scenario two, the steam from the wells goes through a throttling valve, reducing its pressure from 17.5 bar-a to 10 bar-a, after which the steam is led directly to Hellisheidi geothermal power plant.

2.3 Back pressure turbine technical specifications

The technical specifications of the back pressure turbine, to be installed in the Hverahlíd area, should be compatible with the steam conditions from the four wells with high pressure in that area. The chemical composition of the steam is shown in Table 1, and the steam conditions for the back pressure turbine are shown in Table 2.
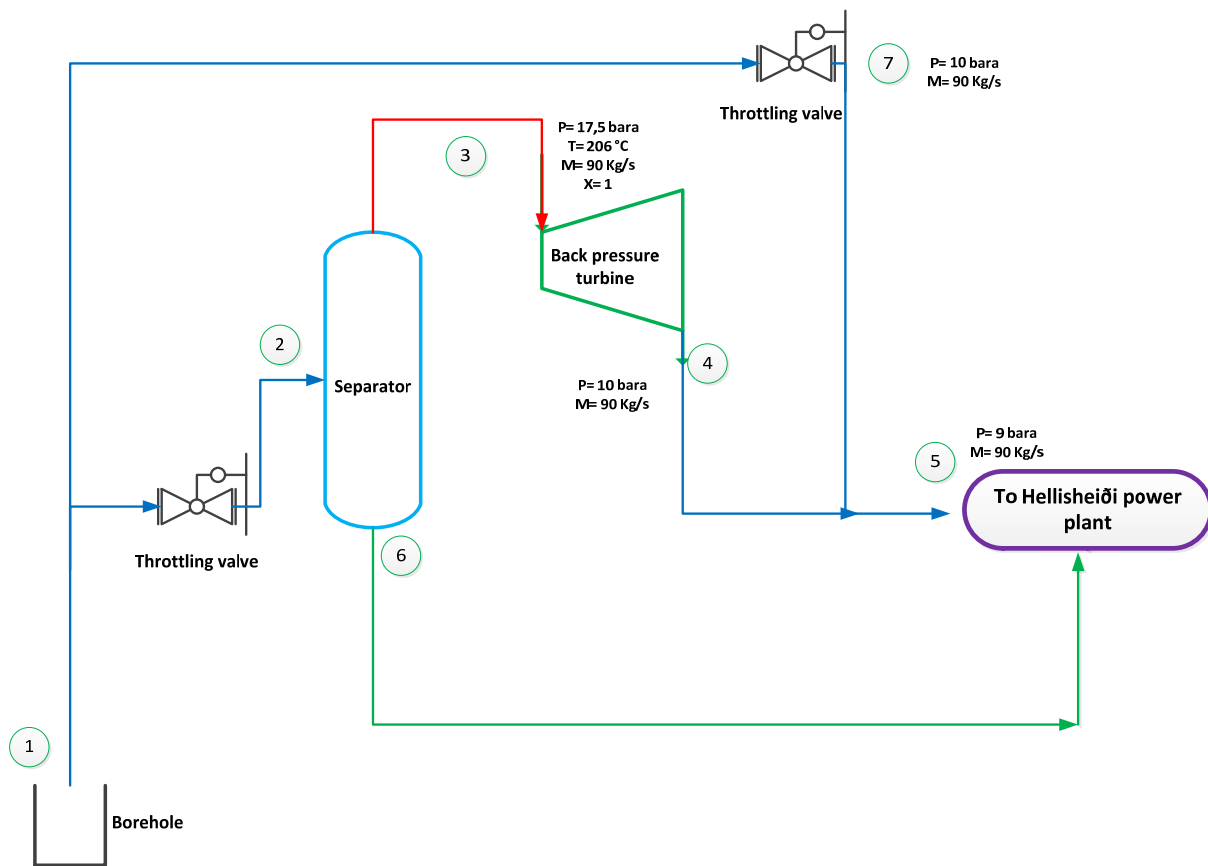
FIGURE 2: Flow diagram showing the two options of connecting the high enthalpy wells

TABLE 1: Chemical composition of the steam

Steam at 17.5 bar-a, 206°C	Range	Rated conditions
CO_2	3000-10.000 ppm	8.000 ppm
H_2S	400 – 2.000 ppm	1.500 ppm
H_2	4 – 200 ppm	100 ppm
CH_4	0 – 5 ppm	2 ppm
N_2	100 – 700 ppm	400 ppm

TABLE 2: The steam conditions

Conditions	Rated conditions	Unit
Rated steam consumption	90	kg/s
Main steam inlet pressure	17.5	bar-a
Main steam outlet pressure	10	bar-a
Main steam inlet temperature	206 (saturated steam)	°C
Frequency	50	Hz
Nominal voltage	11	kV
Generator power factor	0.80	-
Non condensable gas in steam	1.00	% by weight

Based on the above data, the specifications of the proposed back-pressure turbine and the generator are shown in Table 3. This study is applied for a specific turbine with an isentropic efficiency of 85%.

TABLE 3: Technical specifications for the back-pressure turbine-generator set

Parameter	Value	Unit
Throttle pressure	17.5	bar-a
Throttle temperature (Sat)	206	°C
Inlet enthalpy	2795	kJ/kg
Net throttle flow to turbine	90	kg/s
Included NCG flow	1.0	% by weight
Velocity through 2 x 400 mm DIN PN25 valves	44	m/s
Exhaust pressure	10.0	bar-a
Exhaust enthalpy	2704	kJ/kg
Exhaust temperature	180	°C
Exhaust moisture	3%	-
Turbine speed	5000	RPM
Generator speed	1500	RPM
Gross generator output	7.75	MWe
Theoretical steam rate (TSR)	33.470	Ton/MW-hr
Actual steam rate (ASR)	41.800	Ton/MW-hr
Auxiliary load	50	kW
Four-pole generator efficiency	98	%
Speed reduction gear box efficiency	98.5	%

3. THEORETICAL ANALYSIS

3.1 Exergy

Exergy is the theoretical maximum amount of work that can be acquired from a system at any state, pressure and temperature when operating with a reservoir at a constant pressure and temperature (Eastop and McConkey, 2009). The constant pressure and temperature are the surrounding environmental conditions state, called dead state. Dead state means that the working fluid at the exhaust exits in thermodynamic equilibrium with the surrounding conditions, thus the fluid has no more energy to deliver work and can be considered dead. Thermodynamic equilibrium requires that the system is in mechanical equilibrium (i.e. equal pressure), thermal equilibrium (i.e. equal temperature), and chemical equilibrium (i.e. equal reaction potential) with its surroundings. Chemical equilibrium does not take place in a practical sense in most geothermal power plants and can thus be disregarded; only mechanical and thermal equilibrium at dead state need be dealt with in calculating the exergy (DiPippo, 2007).

Exergy is based on the Second Law of thermodynamics which deals with the quality of the energy in addition to the quantity, while the first law of thermodynamics deals only with the quantity of energy and is only concerned with the amount of energy with no regard to its quality. Exergy can be divided into four different exergy components such as: physical exergy, kinetic exergy, potential exergy and chemical exergy, and can be expressed as follows (Bore, 2005):

$$\psi_{total} = \psi_{PH} + \psi_{KE} + \psi_{PE} + \psi_{CH} \qquad (1)$$

where ψ_{total} = Total exergy;
ψ_{PH} = Physical exergy;
ψ_{KE} = Kinetic exergy;
ψ_{PE} = Potential exergy; and
ψ_{CH} = Chemical exergy.

A system may receive or discharge fluids from or to the surroundings, and exchange heat and work with the surroundings. The aim is to gain the maximum power output from the operation of the system. In order to achieve this ideal outcome, there are two thermodynamic conditions that must be met:

1. All processes taking place within the system must be perfectly reversible, which means that no losses occur because of friction, turbulence, or any other source of irreversibility; and
2. The state of all fluids being discharged from the system must be in thermodynamic equilibrium with the surroundings, which means that the leaving fluids have no more potential to do work relative to the surroundings (DiPippo, 2007).

According to the definition of the system, geothermal power plants can be considered an open system operating in steady state once they operate at their design operating conditions. An open system is one in which fluids are exchanged with the surrounding environment through its boundaries during the operation and, in steady operation, indicates that the values of all thermodynamic properties at any point in the system remain constant without changing with time.

In order to get an efficient and effective power plant, it is important to consider the quality and quantity of the energy used to reach a given objective. As mentioned before, the first law of thermodynamics deals only with the quantity of energy and affirms that energy cannot be created or destroyed, meanwhile the second law of thermodynamics deals with the quality of energy, i.e. it is concerned with the quality of energy to cause change, declination of energy during a process, entropy generation and lost opportunities to do work.

In other words, exergy is the expression for loss of available energy due to the generation of entropy in irreversible systems or processes. The exergy loss in a system or component is calculated by multiplying the absolute temperature of the surroundings by the entropy generated. The entropy is the ratio of the heat immersed by matter to the absolute temperature of the surrounding environment.

In open systems operating in steady flow, we deal with the balance of mass, energy, entropy and exergy according to the first and second laws of thermodynamics. Those four balances should be applied (Hepbasli, 2006).

For mass balance:

$$\sum_{i=1}^{n} m_i = 0 \qquad (2)$$

where m_i = mass flow rate crossing each inlet or outlet;
i = an index that accounts for all inlets and outlets of the system; and
n = total number of inlets and outlets.

or:

$$m_i = m_{out} \qquad (3)$$

where m_i = mass flow rate at inlet; and
m_{out} = mass flow rate at outlet.

For energy balance:

$$\sum E_i = \sum E_{out} \qquad (4)$$

where E_i = Total energy input
E_{out} = Total energy output

or:

$$Q - W = -\sum_{i=1}^{n} m_i \left(h_i + 0.5\, v_i^2 + g z_i \right) \tag{5}$$

where   Q   = Heat energy (thermal power) added to the system from the surroundings;
         W   = Work output (mechanical power) developed by the system to the surroundings;
         h_i = Specific enthalpy of the fluid at each inlet or outlet;
         v_i = Velocity of the fluid at each inlet or outlet;
         g   = Gravitational acceleration corresponding to the elevation; and
         z_i = Elevation of each inlet or outlet.

For entropy balance:

$$S_i - S_{out} + S_{gen} = 0 \tag{6}$$

where S_i = Entropy at inlet;
         S_{out} = Entropy at outlet; and
         S_{gen} = Entropy generated by the system due to irreversibility.

For exergy balance:

$$\sum Ex_{in} - \sum Ex_{out} = \sum Ex_{dest} \tag{7}$$

where Ex_{in} = Exergy input;
         Ex_{out} = Exergy output; and
         Ex_{dest} = Exergy destruction due to irreversibility.

In geothermal power plants we focus only on physical and chemical exergies because they are low-quality forms of exergy associated with matter and cannot easily be converted to work, while the kinetic and potential exergies are high-quality exergy forms associated with matter and can be fully converted to useful work (Bore, 2005). Commonly, in power plants we concentrate on physical exergy which exists in the fluid stream, heat transfer and work transfer. Therefore, the general exergy balance can be written as follows:

$$\sum Ex_{heat} - \sum Ex_{work} + \sum Ex_{mass,in} - \sum Ex_{mass,out} = \sum Ex_{dest} \tag{8}$$

where Ex_{heat} = Exergy associated with heat transfer through the system boundaries;
         Ex_{work} = Exergy associated with work transfer through the system boundaries;
         $Ex_{mass,in}$ = Exergy associated with the fluid streams at the inlet; and
         $Ex_{mass,out}$ = Exergy associated with the fluid streams at the outlet.

Exergy associated with heat transfer through the system boundaries can be determined as follows:

$$Ex_{heat} = \sum \left(1 - \frac{T_0}{T_k} \right) * Q_k \tag{9}$$

where Q_k = Heat transfer rate through the boundary at temperature T_k at location k; and
         T_0 = dead state temperature.

Exergy associated with work transfer through the system boundaries can be written:

$$Ex_{work} = W \tag{10}$$

where   W   = Work.

Specific exergy associated with the fluid stream can be calculated by the following equation:

$$\psi = (h - h_0) - T_0(S - S_0) \tag{11}$$

where ψ = Specific exergy associated with the fluid stream;
h = Specific enthalpy;
S = Sspecific entropy;
h_0 = Enthalpy at dead state conditions; and
S_0 = Entropy at dead state conditions.

Therefore, the power output can be determined by:

$$W = m[(h_1 - h_2) - T_0(S_1 - S_2)] \tag{12}$$

And the maximum power output can be given by the equation:

$$W_{max} = m[(h_1 - h_0) - T_0(S_1 - S_0)] \tag{13}$$

where h_1, h_2 = Enthalpy at the inlet and outlet, respectively; and
S_1, S_2 = Entropy at the inlet and outlet, respectively.

3.2 Restricted dead state

Restricted dead state is the state in which a specific form of equilibrium is needed where only the mechanical and thermal equilibrium must be satisfied apart from the chemical equilibrium. This equilibrium is called thermo-mechanical equilibrium (Hepbasli, 2006).

3.3 Exergy balance (Exergy analysis)

An exergy analysis (or second law analysis) has been verified to be a powerful tool in the simulated thermodynamic analyses of energy systems. In other words, it has been commonly used in the design, simulation and performance evaluation of energy systems. The exergy analysis method is utilised to distinguish and to evaluate quantitatively the causes of the thermodynamic deficiency of the process under consideration. It can, therefore, indicate the possibilities of thermodynamic enhancement of the process under consideration, but only an economic analysis can decide the practicality of a possible enhancement.

Exergy analysis affords a mathematical method with which to evaluate the maximum work extractable from matter relative to a reference state (i.e. dead state) (Hepbasli, 2006). A simple diagram showing exergy balance is shown in Figure 3.

Exergy balance methods can indicate the quantity and quality of heat losses in the system or component and delineate the location of energy degradation (measure

FIGURE 3: A simple diagram showing exergy balance

and ascertain causes of energy degradation). Most of the thermodynamic limitations cannot be detected by an energy analysis. Some certain processes like throttling, heat transfers, expansion and friction contain no energy losses but they degrade the quality of energy and its ability to do work and, therefore, contain exergy losses (Bore, 2005). Exergy balance shows the exergy flow through a system or component, shown by the Sankey diagram in Figure 4.

3.4 Exergetic efficiency

In geothermal power plants it is better to calculate efficiency based on exergy, which is sometimes called the Second Law efficiency or utilization efficiency, rather than calculating the thermal efficiency (First Law efficiency). There are two different methods to determine the exergetic efficiency: "brute-force" and rhe "functional" method (Hepbasli, 2006). In the "brute-force" method, exergy efficiency for a system can be expressed as the ratio of the sum of all output exergy to the sum of all input exergy. In the "functional" method, exergy efficiency for a system can be expressed as the ratio of the exergy associated with the desired energy output to the exergy associated with the energy consumed to achieve the desired output. The brute-force method can be applied directly for each system regardless of the nature of the system, once all exergy flows have been determined.
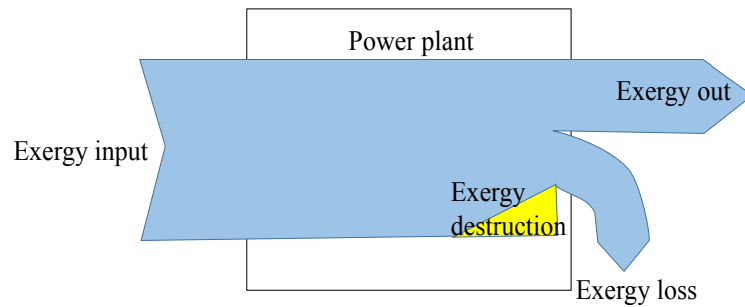
FIGURE 4: A Sankey diagram of exergy flow

The functional method cannot be applied until the purpose of the system is known and the working form of the efficiency equation can be formulated. Exergetic efficiency can be applied for a power plant or for each component separately.

3.4.1 Exergetic efficiency for turbine

With regard to the two different methods for calculating the exergetic efficiency, both methods can be applied for calculating the exergetic efficiency of the turbine. Assume that we have a turbine with steam entering through the inlet and leaving from the exhaust, as shown in Figure 5.
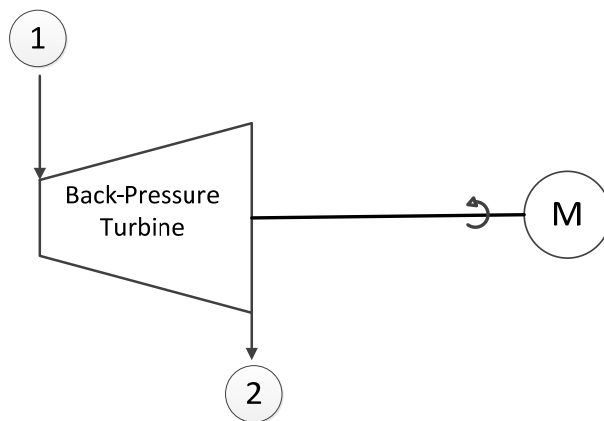

FIGURE 5: Simple turbine with inlet and outlet

The brute-force efficiency can be expressed as shown in Equation 14 and the functional efficiency can be expressed as in Equation 15.

$$\eta_{B.F}^{II} = \frac{W + Ex_2}{Ex_1} \tag{14}$$

$$\eta_{FUN}^{II} = \frac{W}{Ex_1 - Ex_2} \tag{15}$$

where $\eta_{B.F}^{II}$ = Brute-force efficiency;
       η_{FUN}^{II} = Functional efficiency;
       W = Work output; and
       Ex_1 and Ex_2 = Exergy at inlet and outlet, respectively.

4. RESULTS

A model was built using the software Engineering Equation Solver (EES)to calculate the energy and exergy flow of the back pressure turbine to be installed at Hverahlíd area for scenario one, shown in T-s diagram in Figure 6, and also to calculate the energy and exergy flow through the throttling valve for scenario two, shown in T-s diagram in Figure 7.
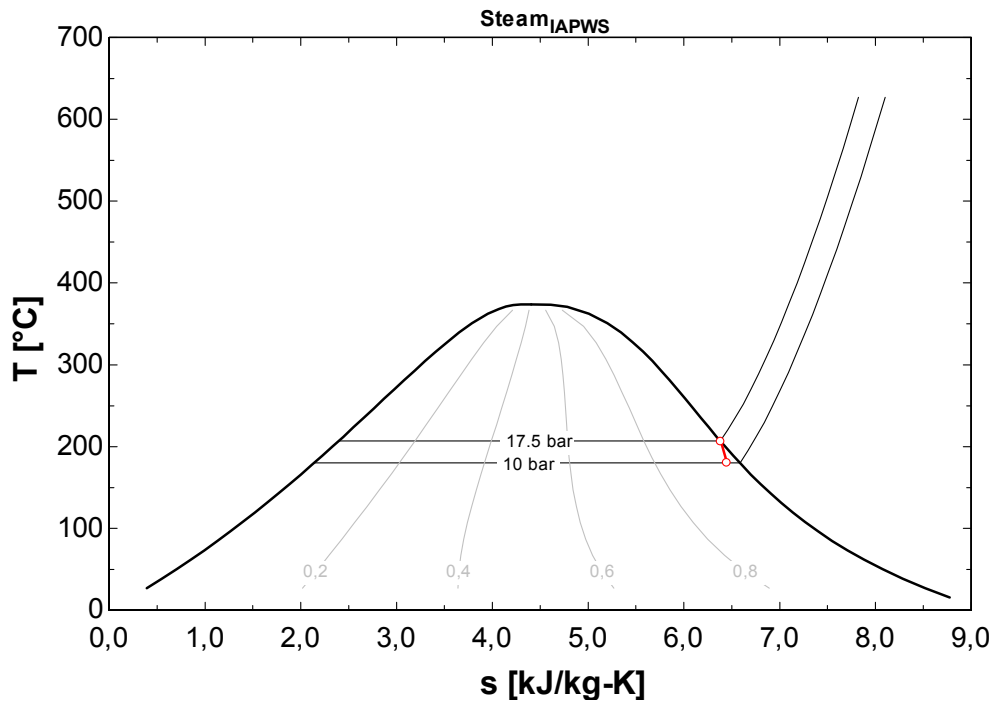

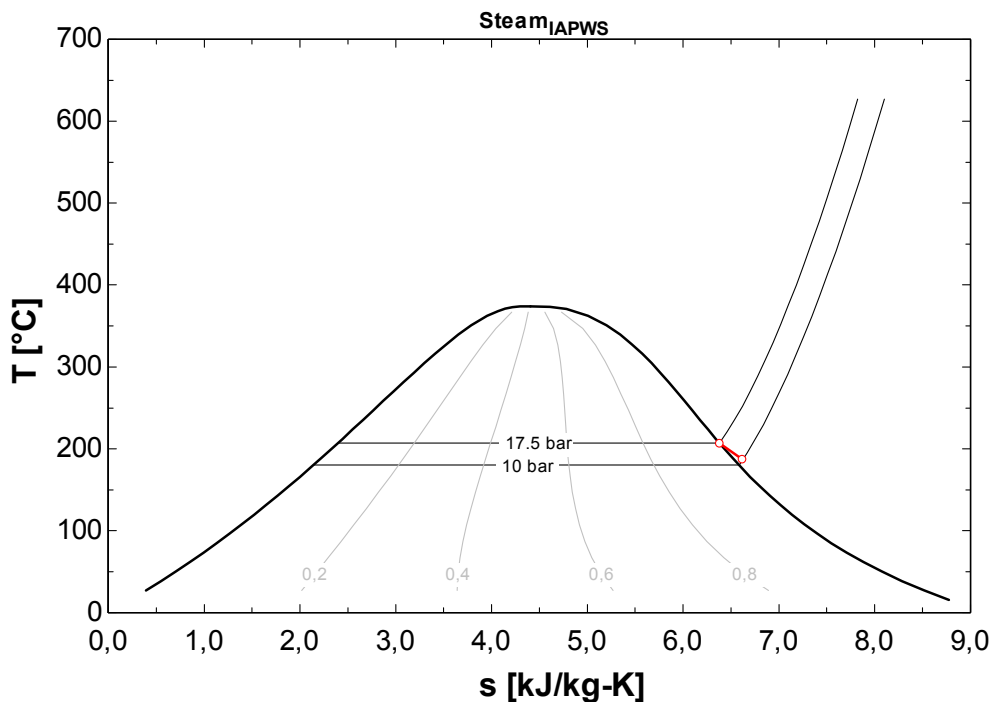FIGURE 6: T-S diagram showing the process of expansion in the back-pressure turbine

FIGURE 7: T-S diagram showing the processes of throttling due to the throttling valve

Many assumptions were made to simplify these calculations:

1. The flow of geothermal fluid is steady which means the mass flow in is equal to the mass flow out and there are no losses.
2. The turbine is thermally isolated which means there is no heat exchange between the turbine and the surrounding environment.
3. There is no heat exchange between the valve and the surroundings and no work is performed or produced during the process. Thus, the pressure drop across the valve is assumed to be isenthalpic.
4. Geothermal fluid has the same properties as pure water.
5. Kinetic exergy, potential exergy and chemical exergy are neglected.

4.1 Energy analysis

Energy can be defined as motion or the ability to cause motion. Commonly, energy is conserved in a process according to the First Law of Thermodynamics (Bore, 2005). In an energy analysis we calculate the energy input, the energy output, the isentropic work that can be done, the actual work and the energy efficiency.

4.1.1 Energy input

Energy input is the heat energy added into a turbine and is equal to the product of mass flow rate of the steam into a turbine and its enthalpy at entry.

$$E_i = m_i * h_i \tag{16}$$

where E_i = Energy input;
         m_i = Mass flow rate at the entry; and
         h_i = Enthalpy at the entry.

$$E_i = 90\frac{kg}{s} * 2795 \frac{kJ}{kg} = 251{,}550 \frac{kJ}{s}$$

The energy input is thus 251,550 kW.

4.1.2 Energy output

Energy output is equal to the heat exhausted which is equal to the mass flow rate of the steam that exits the turbine multiplied by the enthalpy at the exit.

$$E_{out} = m_{out} * h_{out} \tag{17}$$

where E_{out} = Energy output;
         m_{out} = Mass flow rate at the exit; and
         h_{out} = Enthalpy at the exit.

$$E_{out} = 90\frac{kg}{s} * 2704 \frac{kJ}{kg} = 243{,}360 \frac{kJ}{s}$$

The energy output is 243,360 kW.

4.1.3 Work done

Work done is the ideal or theoretical work that can be obtained in isentropic expansion in the turbine and is equal to the energy in the steam at the entrance into the turbine minus that at the exit.

$$W.D = E_i - E_{out} \tag{18}$$

where $W.D$ = Work done.

$$W.D = 251550 \frac{kJ}{s} - 243360 \frac{kJ}{s} = 8190 \, kW$$

Work done is 8190 kW.

4.1.4 Actual power developed by turbine shaft

The Actual Power developed by the turbine shaft is the mechanical power produced by the turbine and is equal to the net power output from the generator divided by the gearbox efficiency and the generator efficiency.

$$P = \frac{Generator \; Power}{\eta_{Gearbox} \; * \; \eta_{Generator}} \tag{19}$$

where P = Actual power developed by turbine shaft;
$\eta_{Gearbox}$ = Gear box efficiency; and
$\eta_{Generator}$ = Generator efficiency.

$$P = \frac{[7750 \, kW]}{0.98 * 0.985} = 8028.59 \, kW$$

The actual power developed by turbine shaft is 8028.59 kW.

4.1.5 Energy efficiency (1ˢᵗ Law efficiency) of the turbine

Energy efficiency is the ratio of the energy output as mechanical power to the total energy in the system:

$$\eta_U = \frac{(Actual \; power \; developed \; by \; turbine \; shaft)}{(E_i - E_{out})} \tag{20}$$

where η_U = Energy efficiency.

$$\eta_U = \frac{8028.59 \, kW}{\left(251,550 \frac{kJ}{s} - 243,360 \frac{kJ}{s}\right)} = 98.03\%$$

The energy balance for the back pressure turbine is shown in the Sankey diagram in Figure 8.

All of the above calculations are for scenario one. For scenario two, it is known from the basics of thermodynamics that the enthalpy remains constant through throttling which means the energy before throttling is equal to that after throttling. Therefore, there is no energy loss due to throttling, although exergy is destroyed during the process.
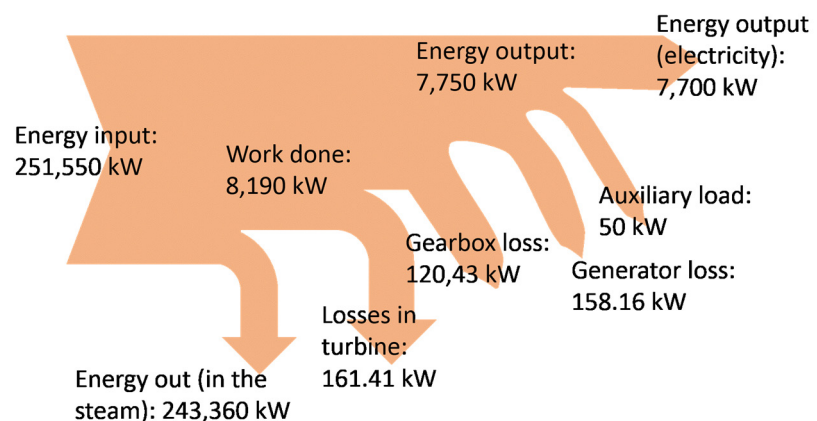

FIGURE 8: A Sankey diagram showing energy flow in the turbine

4.2 Exergy analysis

Exergy can be defined as work or the ability to cause work. Commonly, exergy is conserved in a reversible process, but is always consumed in an irreversible process according to the Second Law of Thermodynamics (Bore, 2005).

Exergy is always evaluated with respect to a reference environment (dead state). The reference environment is in stable equilibrium, acts as an infinite system and is a sink or source for heat and materials. It experiences only internal reversible processes, in which its intensive properties (i.e. temperature T_0 and pressure P_0) remain constant. In this analysis, the surrounding temperature and pressure are taken as $T_0 = 10°C$ (283 K) and $P_0 = 1$ bar, based on weather and climate conditions at Hverahlíd, Iceland. In exergy analysis we calculate the exergy input, the exergy out, exergy destruction and exergetic efficiency.

4.2.1 Exergy input

The total exergy associated with the fluid stream at the inlet of a turbine can be written as:

$$\psi_{in} = m_{in} * [\, h_{in} - h_0 - T_0\,(S_{in} - S_0)] \tag{21}$$

where ψ_{in} = Exergy input;
m_{in} = Mass flow rate input; and
h_{in} and S_{in} = enthalpy and entropy at the inlet, respectively.

$$\psi_{in} = 90\frac{kg}{s} * \left[2795\frac{kJ}{kg} - 42.12\frac{kJ}{kg} - 283\,K\,*\left(6.388\frac{kJ}{kg}.\,K - 0.1511\frac{kJ}{kg}.\,K\right)\right]$$
$$= 88905.36\frac{kJ}{s}$$

The exergy input is 88,905.36 kW.

4.2.2 Exergy out

The total exergy associated with the fluid stream at the exit of turbine can be written as:

$$\psi_{out} = m_{out} * [(h_{out} - h_0 - T_0(S_{out} - S_0)] \tag{22}$$

where ψ_{out} = Exergy out;
m_{out} = Mass flow rate at the outlet; and
h_{out} and S_{out} = Enthalpy and entropy at the outlet, respectively.

$$\psi_{out} = 90\frac{kg}{s} * \left[2704\frac{kJ}{kg} - 42.12\frac{kJ}{kg}\right.$$
$$\left. - 283\,K*\left(6.452\frac{kJ}{kg}.\,K - 0.1511\frac{kJ}{kg}.\,K\right)\right] = 79085.28\frac{kJ}{s}$$

The exergy out is 79,085.28 kW.

4.2.3 Exergy destruction in turbine

The total exergy destroyed in the turbine is due to irreversibility in the process or deficiencies of the turbine:

$$\psi_{des} = \psi_{in} - \psi_{out} - \psi_{power} \tag{23}$$

$$\psi_{des} = 889,015.36 \ \frac{kJ}{s} - 79,085.28 \ \frac{kJ}{s} - 80,28.59 \ kW = 1791.49 \ \frac{kJ}{s}$$

Exergy destruction in the turbine due to irreversibility is 1791.49 kW.

4.2.4 Exergy efficiency

For brute-force exergy efficiency, it is the ratio of the total exergies associated with the exergy out to the total exergies associated with the exergy input.
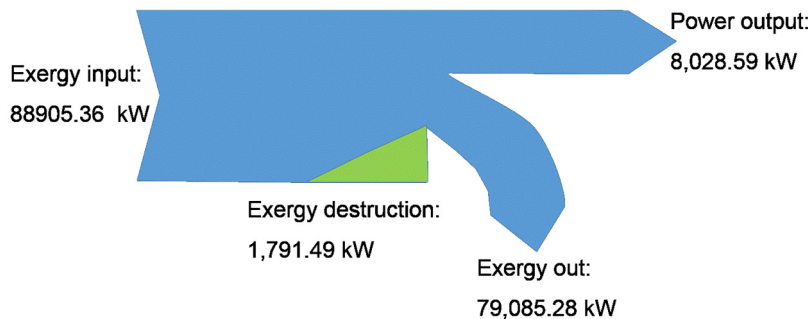
$$\eta_{B.F}^{II} = \frac{\psi_{power} + \psi_{out}}{\psi_{in}} \tag{24}$$

$$\eta_{B.F}^{II} = \frac{8028.59 + 79,085.28}{88,905.36} = 97.98\%$$

For functional exergy efficiency, it is the ratio of the exergy converted to useful work to the total exergy in the system:

$$\eta_{FUN}^{II} = \frac{\psi_{power}}{(\psi_{in} - \psi_{out})} \tag{25}$$

$$\eta_{FUN}^{II} = \frac{8028.59 \ kW}{88,905.36 \ \frac{kJ}{s} - 79,085.28 \ \frac{kJ}{s}} = 81.76\%$$

FIGURE 9: A Sankey diagram showing the exergy flow through the turbine

The exergy balance for the back pressure turbine is shown in the Sankey diagram in Figure 9.

The general layout of the back pressure turbine is shown in Figure 10.

For the second scenario, in which we connect these four wells with high wellhead pressure directly to Hellisheidi geothermal power plant through a throttling valve, there will be some losses of exergy which can be determined as follows:

$$Exergy \ loss \ (Destruction) = Ex_{Upstream} - Ex_{Downstream} \tag{26}$$

The explanation is that the loss of exergy is equal to the exergy of the fluid before the throttling valve (Up-stream) less the exergy of the fluid after the throttling valve (Down-stream). As we know the enthalpy remains constant due to using throttling valve:

$$Ex_{Upstream} = 90 \ \frac{kg}{s} * \left[2795 \ \frac{kJ}{kg} - 42.12 \ \frac{kJ}{kg} - 283 \ K * \left(6.388 \ \frac{kJ}{kg}.K - 0.1511 \ \frac{kJ}{kg}.K \right) \right]$$
$$= 88,905.36 \ \frac{kJ}{s}$$

The entropy at down-stream is the entropy at a pressure of 10 bar-a and enthalpy of 2795 kJ/kg.

P[1]= 17.5 bar-a
T[1]= 206 deg C
M= 90 kg/s
H[1]= 2795 kJ/kg

①

η= 85%

Back-Pressure Turbine

Actual power developed by turbine shaft: 8.028 MW

Gear Box

Generator

M

η= 98%

Gross generated power: 7.75 MW

η= 98.5%

Work done: 8.190 MW

②

P[2]= 10 bar-a
T[2]= 180 deg C
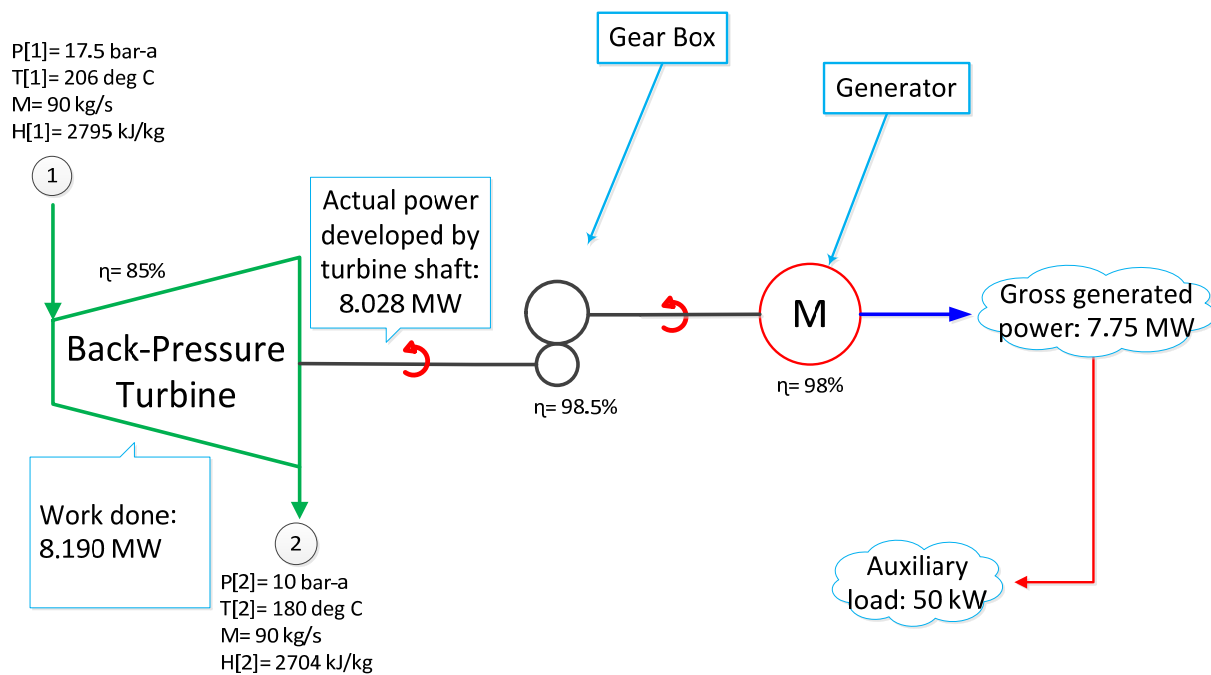M= 90 kg/s
H[2]= 2704 kJ/kg

Auxiliary load: 50 kW

FIGURE 10: General layout of the back-pressure turbine

$$Ex_{Downstream} = 90 \ \frac{kg}{s} * \left[2795 \ \frac{kJ}{kg} - 42.12 \ \frac{kJ}{kg} - 283 \ K * \left(6.625 \ \frac{kJ}{kg}. \ K - 0.1511 \ \frac{kJ}{kg}. \ K \right) \right]$$

$$= 82,868.97 \ \frac{kJ}{s}$$

$$Exergy\,loss(Destruction) = 88,905.36 - 82,868.97 = 6036.39 \ kJ/s$$

The exergy destruction (loss) due to the throttling valve is: 6036.39 kW.

Exergy destruction in the throttling valve can be expressed in the Sankey diagram in Figure 11.

Exergy at up-stream: 88,905.36 kW

Exergy at down-stream: 82,868.97 kW

Exergy destruction: 6,036.39 kW
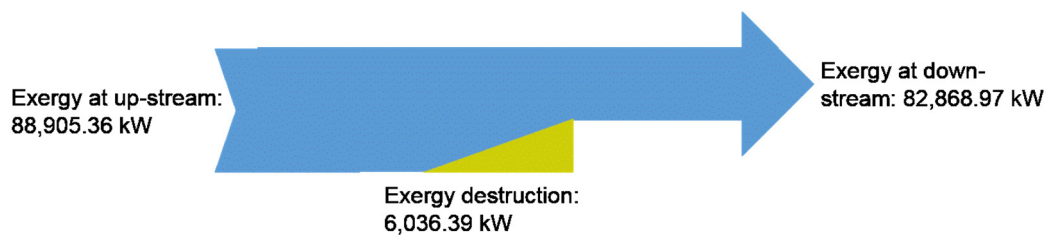
FIGURE 11: A Sankey diagram showing the exergy flow through a throttling valve

If a simple comparison is made between the two options (scenarios) of using a back-pressure turbine and reducing the pressure by using a throttling valve, we can clearly see that the steam leaving the back pressure turbine towards Hellisheidi geothermal power plant has an exergy of 79,085 kW while the steam from a throttling valve has an exergy of 82,869 kW. This results in higher potential for performing work in the steam if a throttle valve is chosen compared to the back pressure turbine. However, the difference in exergy of the exiting steam between the two scenarios is 3,783.69 kW, which is much less than the electricity produced by the back pressure turbine (7,750 kW). The back pressure turbine makes use of the exergy in the steam, and produces electricity, while the exergy drop within the steam led through a throttle valve is due to exergy destruction. Thus, a more reasonable way of reducing pressures of high-pressure geothermal wells, from a thermodynamic point of view, is by using a back pressure turbine instead of a throttle valve.

5. CONCLUSIONS AND RECOMMENDATIONS

From the study, the results can be summarized as:

1. The total exergy available in the steam is 89 MW.
2. A back-pressure turbine has an exergetic efficiency of 81.76% while its mechanical efficiency is 98%.
3. The exergetic efficiency gives a more accurate indication than the energy efficiency.
4. There is a possibility of producing electrical energy of about 7 to 8 MW by adding a backpressure turbine as a topping unit at Hverahlíd area with a drop of about 10 MW of exergy compared to an exergy drop of 6 MW when reducing the pressure from 17.5 bar-a down to 10 bar-a by using a throttling valve.
5. Some of the exergy is destroyed in the turbine (1.8 MW) and the throttling valve (6 MW) due to the irreversibility of the processes. Exergy destruction occurs in all components of a power plant due to irreversibilities.

A suggested recommendation is to conduct a study on the economic feasibility of installing a back pressure turbine, taking into account all the costs for installation and running and then comparing the costs with the revenue of selling the electricity generated from the back pressure turbine.

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APPENDIX I: EES calculation model

{Environment conditions}
P[0] = 1
T[0]= 10
h[0]=Enthalpy (Steam_IAPWS;P=P[0];T=T[0])
s[0]=Entropy (Steam_IAPWS;P=P[0];T=T[0])

"Steam inlet conditions"
m_dot[1]=90[kg/s] {Given in design documents}
P[1]=17,5 [bar] {Given in design documents}
x[1]=1 {Saturated steam at inlet}
T[1]=206

{Calculated enthalpy and entropy according to inlet info}
h[1]=Enthalpy (Steam_IAPWS;P=P[1];x=x[1])
s[1]=Entropy (Steam_IAPWS;P=P[1];x=x[1])

"Steam outlet conditions"
P[2]=10[bar] {given in design documents}
T[2]=180 {given in design documents}
m_dot[2]= m_dot[1]
x[2] = 0,97 {given in design documents 3% moisture content}

"Power production calculations"
W_out = 7750[kW] {Given in design documents}
eta_gearbox=0,985 {Value given in design documents}
eta_generator= 0,98   {Value given in design documents}

{Calculating turbine energy efficiency according to the power output given}
7750 = eta_turbine*eta_generator*eta_gearbox*m_dot[1]*(h[1]-h[2])

{Ideal entropy at turbine outlet assuming an isentropic process}
s_i[2] = s[1]

{Ideal enthalpy at turbine outlet}
h_i[2]=Enthalpy (Steam_IAPWS;P=P[2];s=s_i[2])

{Isentropic turbine power output}
W_is= m_dot[1]*(h[1]- h_i[2])

{Actual entropy at turbine outlet}
S[2] = Entropy (Steam_IAPWS;P=P[2];x=x[2])

{Assuming the isentropic efficiency of the turbine is 85%}
eta_is=0,85
{Calculating the enthalpy at the outlet according to the isentropic efficiency}
eta_is= (h[1]-h[2])/(h[1]-h_i[2])

{Actual power output developed by turbine shaft}
W_act= m_dot[1]*(h[1]- h[2])

{Energy analysis}

{Energy input= m_dot * h_ inlet}
E[1] = m_dot [1]* h[1]

{Energy output= m_dot * h_out}
E[2] = m_dot [2]* h[2]

{Work done = E_i - E_out}
W.D = E[1] - E[2]

{Energy efficiency (1st Law efficiency) of Turbine: eta_turbine= (Actual power developed by turbine shaft)] / (E_i- E_out)}
energyefficiency= W_act/(E[1] - E[2])

{Exergy analysis}

{Exergy input}
Ex[1]= m_dot[1] * (h[1]- h[0]- (T[0]+273)*(S[1]-S[0]))

{Exergy out}
Ex[2]= m_dot[2] * (h[2]- h[0]- (T[0]+273)*(S[2]-S[0]))

{Exergy distruction}
Ex[3]= Ex[1]- Ex[2]- W_act

{Exergetic efficiency}
Exergetic Efficiency = W_act / (Ex[1]- Ex[2])

{For throttling valve}

{Exergy before throttling valve}
ExVlve[1]= m_dot[1] * (h[1]- h[0]- (T[0]+273)*(S[1]-S[0]))

{Exergy after throttling valve}
h_th[2]= h[1] {Throttling}
ExVlve[2]= m_dot[2] * (h_th[2]- h[0]- (T[0]+273)*(S_th[2]-S[0]))

{Entropy after throttling}
S_th[2]=   Entropy (Steam_IAPWS;P=P[2];h=h_th[2])

{Exergy destruction due to throttling valve}
ExVlve_dest= ExVlve[1] - ExVlve[2]