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SIMULATION AND PERFORMANCE ANALYSIS OF THE NEW GEOTHERMAL CO-GENERATION POWER PLANT (OV-5) AT SVARTSENGI, SW-ICELAND

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ABSTRACT

The modular method is used to simulate and analyse the performance of the new co-generation power plant (OV-5) at Svartsengi that generates electricity via a single-flash process and supplies turbine extraction steam to heat exchangers where fresh water is heated up to 110°C for a district heating system. The utilization efficiency of the power plant is improved by 15%, using extraction steam instead of high-pressure steam, and 14-22% at different heat loads. The turbine model shows that at 24 MW_e electrical output and the rated heat load, the pressure of the first and second extraction drops below 2.5 bar-a and 1.3 bar-a, respectively. At this point, it is necessary to supply high-pressure steam to heat exchangers. The optimum main steam pressure for the turbine is 5-7 bar-a in order to maximize the power output and limit the exhaust wetness to 12-13%.

1. INTRODUCTION

Geothermal power plants, particularly those operating on the flash-steam principle, offer the opportunity to combine electricity generation with direct heat applications. The latter utilization can be accomplished using the thermal energy available in a waste brine and rejected heat in a condenser to heat fresh water which can then be distributed to a variety of end users. The technical feasibility and design of such co-generation power plants depends on a number of factors including the reservoir temperature of the geothermal fluid, the type of flash system used in the power plant (single or double-flash), the distance to end users and the types of applications. The climate, topography and cost of other energy alternatives will also influence the final decision on using geothermal co-generation power plants.

The Sabalan area is the most promising geothermal area in the northwest of Iran and has been selected for an exploration drilling and feasibility study of the first geothermal power plant in Iran. This area consists of three geothermal fields located in the northern, eastern, and southern parts of the Sabalan central volcano. Preliminary studies show that these fields have a common source, and temperature in excess of 200 °C is expected to be found at depth (Khosrawi, 1996). The cold climate, high elevation, and difficulties of fuel transportation are the main reasons for making a geothermal co-generation power plant feasible for this area.

This report presents the main characteristics of this kind of a geothermal power plant. For this purpose, the new co-generation power plant at Svartsengi, SW-Iceland (OV-5) is simulated based on the modular method and, its thermodynamic design parameters, such as utilization efficiency and properties of extraction steam, are analysed.

The two main geothermal co-generation power plants existing are at Svartsengi (Björnsson and Albertsson, 1985) and Nesjavellir (Gunnarsson et al., 1992) in Iceland and their thermodynamic cycles are described in the following sections. The thermal output of these power plants is, respectively, 125 MW_t and 200 MW_t and is used for district heating systems. There are also many co-generation power plants in Japan such as Otake, Mori, and Hatchobaru, but their thermal outputs are less than 4 MW_t and are mainly used for greenhouses, bathing and snow melting (Uchida, 1997).

1.1 The Svartsengi power plant

At Svartsengi the geothermal fluid cannot be used directly for heating because of its high mineral content. On cooling, it releases great quantities of hard deposits (silica) which lodge themselves in pipes and other equipment, making them inoperable after a short time. For this reason, the geothermal fluid is flashed twice in this power plant and the heat of the steam is utilized, while deposits from the steam are minimal. As shown in Figure 1, high-pressure geothermal steam (6.5 bar-a) is harnessed in steam turbines for the generation of electricity (8 MW_e) and for the final heating of the utility water (110°C). Low-pressure geothermal steam (103°C) is used for the direct warm-up of fresh water (preheating) and for the generation of electricity (8.4 MW_e) in Ormat binary generators. Thus fresh water is warmed up

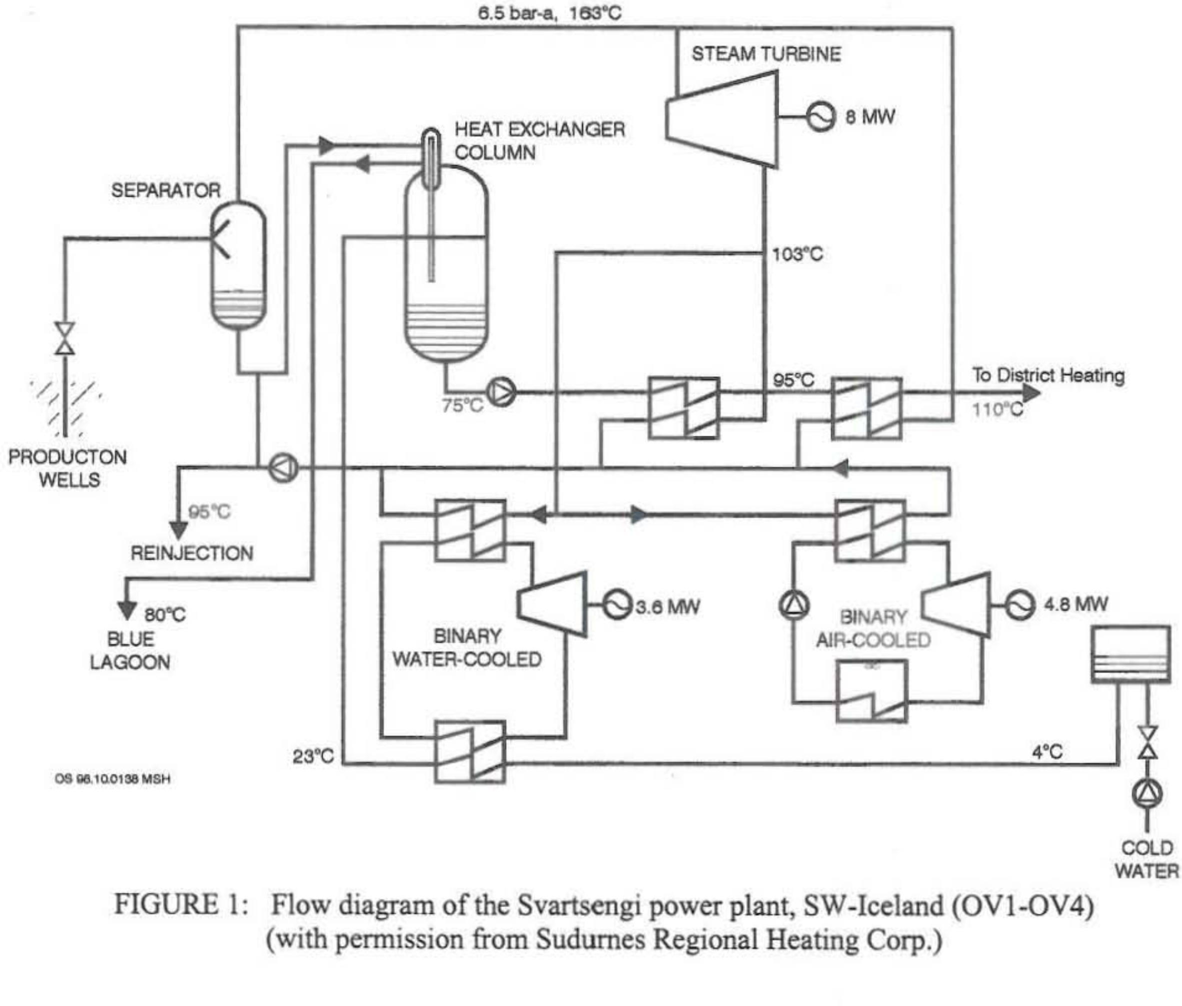

FIGURE 1: Flow diagram of the Svartsengi power plant, SW-Iceland (OV1-OV4) (with permission from Sudurnes Regional Heating Corp.)

from 4 to 25°C in the Ormat generators, de-aerated and heated with low-pressure steam to about 75°C in direct-contact heat exchangers. Then it is heated to about 95°C in plate heat exchangers with 103°C hot steam coming out of the exhaust from steam turbines, and finally the fresh water is warmed up with high-pressure steam to about 110°C in plate heat exchangers. Geothermal brine, at about 80°C, which is not used for heating finally forms the Blue Lagoon along with the condensate from the geothermal steam. The total thermal and electrical power of the plant are 125 MW_t and 16.4 MW_e, respectively. The new co-generation power plant at Svartsengi (OV-5) is designed to produce 30 MW_e electricity and about 75 MW_t of hot water. Its thermodynamic cycle is described in Chapter 2.

1.2 The Nesjavellir power plant

The power plant at Nesjavellir, SW-Iceland can be divided into three parts, i.e. a steam supply system from boreholes, heating of fresh cold water and production of electricity. Figure 2 shows the flow diagram of this power plant. Mixed steam and water are conveyed from boreholes through collection pipes to the separation station where the water is separated from the steam. Excess steam and unutilized water go into a steam exhaust outside the separation station. From there steam and water proceed by separate pipes to the power plant at a pressure of about 12 bars and a temperature of 188°C. The steam is conveyed to two 30 MW_e steam turbines for electricity production. In the condenser, the exhaust turbine steam is utilized to preheat cold fresh water from 4 to 62°C. In the first shell-and-tube heat exchanger the separated water (brine) is utilized to heat cold water. The heated water is then mixed with preheated water before final heating occurs in the second shell-and-tube heat exchanger where fresh water is heated up to 88°C. The cold water is saturated with dissolved oxygen that corrodes steel after being heated. To get rid of the oxygen, the water is sent to a deaerator where boiling under low pressure releases the dissolved oxygen and other gases from the water. During the process, the water temperature cools to 83°C. Finally, a very small quantity of steam containing acid gases is mixed with the water to eliminate the last traces of dissolved oxygen and lower the pH of the water in order to prevent precipitation in the distribution system. The heated water is then pumped up to a storage tank at 230 m higher level from which it flows by gravity the 27 km distance to the Reykjavík area. On the way, the water cools about 1-2°C. The total thermal and electrical power of the plant is 200 MW_t and 60 MW_e, respectively.

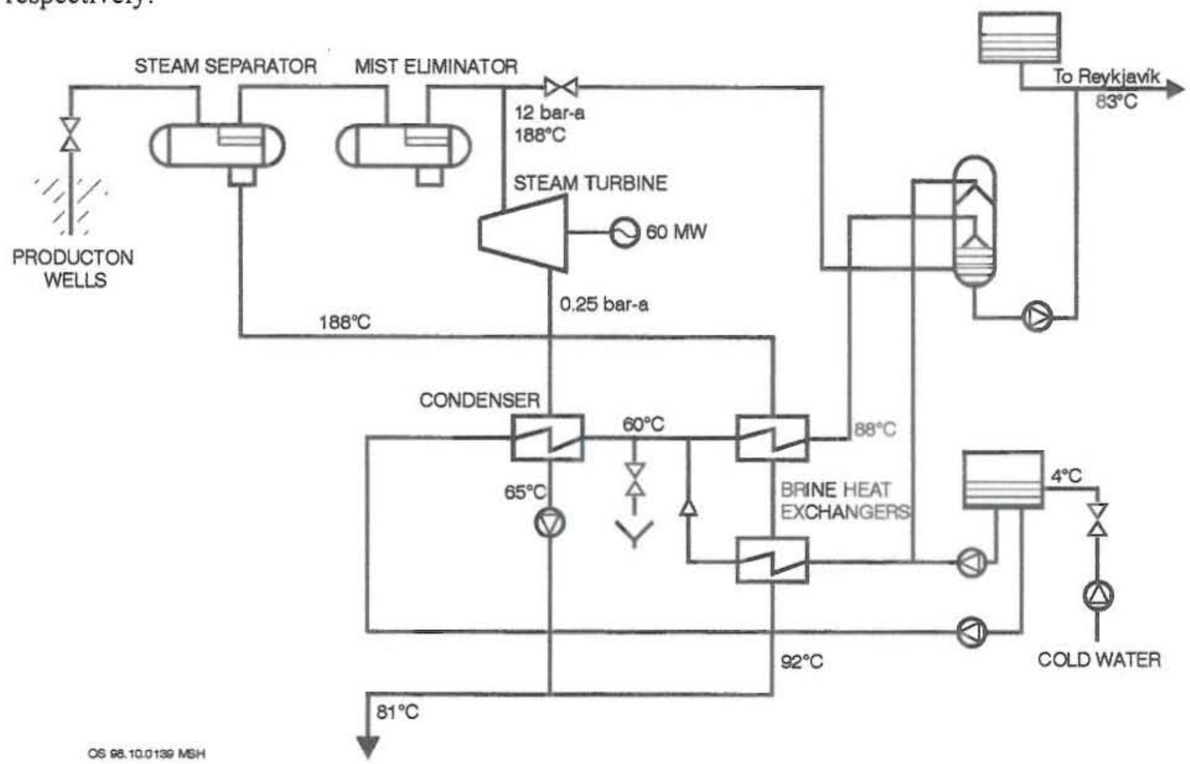

FIGURE 2: Flow diagram of the Nesjavellir power plant, SW-Iceland (modified after Reykjavík District Heating, 1998)

2. THE NEW CO-GENERATION POWER PLANT (OV-5) AT SVARTSENGI

The new power plant (OV-5) is designed for 30 MW_e electricity generation and 70 MW_t heating output. The district heating part is designed to heat up and de-aerate 240 kg/s of preheated fresh water derived from Ormat turbines from about 23°C to 90-95°C. Then pumps, final-heaters and coolers take care of pumping 70 kg/s of 85°C water to the town Grindavík and/or 240 kg/s of 110-115°C water to the town Njardvík (Figure 3). The maximum pumping in OV-5 to these places is 240 kg/s. Turbine extractions supply enough low-pressure steam for after-heating and final-heating of the district heating water.

In addition, it is possible to receive up to 150 kg/s of about 95°C hot district heating water from power plant 2 (OV-2), and heat it to 110-115°C together with the water that OV-5 produces itself. This is done because the steam in OV-5 is extraction steam (2.5 bar-a), whereas the steam in OV-2 is high-pressure steam (6.5 bar-a) that has been used as possible for electrical production. OV-2 takes care of pumping this water through the final heaters in OV-5. By this mean OV-5 can simulatinously produce 320 kg/s of 110-115 °C hot water to the Njardvík pipeline and 70 kg/s of 85 °C hot water to the Grindavík pipeline. Figure 3 shows a flow diagram of the OV-5 power plant.

The turbine is designed to operate at full-load and also supply extraction steam for the district heating system. If the power is reduced, it comes to the point where the extraction steam pressure becomes too low to be usable for district heating heat-exchangers, and instead of the extraction steam the high pressure steam, taken through the bypass valves, must be used for heating the district heating water.

The high-pressure steam pressure to the turbine will be controlled by the existing control valves in OV-2. In addition, the turbine will be equipped with a valve that reduces the turbine power if the steam pressure drops below 6.5 bar-a.

The medium pressure (first extraction) is variable depending on the district heating load, 2.7– 3 bar-a. If the turbine load is reduced, this pressure drops. In order to maintain minimum pressure, a bypass valve controls steam from the high pressure steam supply in order to prevent the medium pressure from dropping below 2.5 bar-a.

The low pressure (second extraction) is variable depending on the district heating load (1.4 bar-a at maximum and 1.9 bar-a at the minimum district heating load). A control valve between power plants OV-5 and OV-2 controls the extraction pressure based on a variable set-point that depends on the district

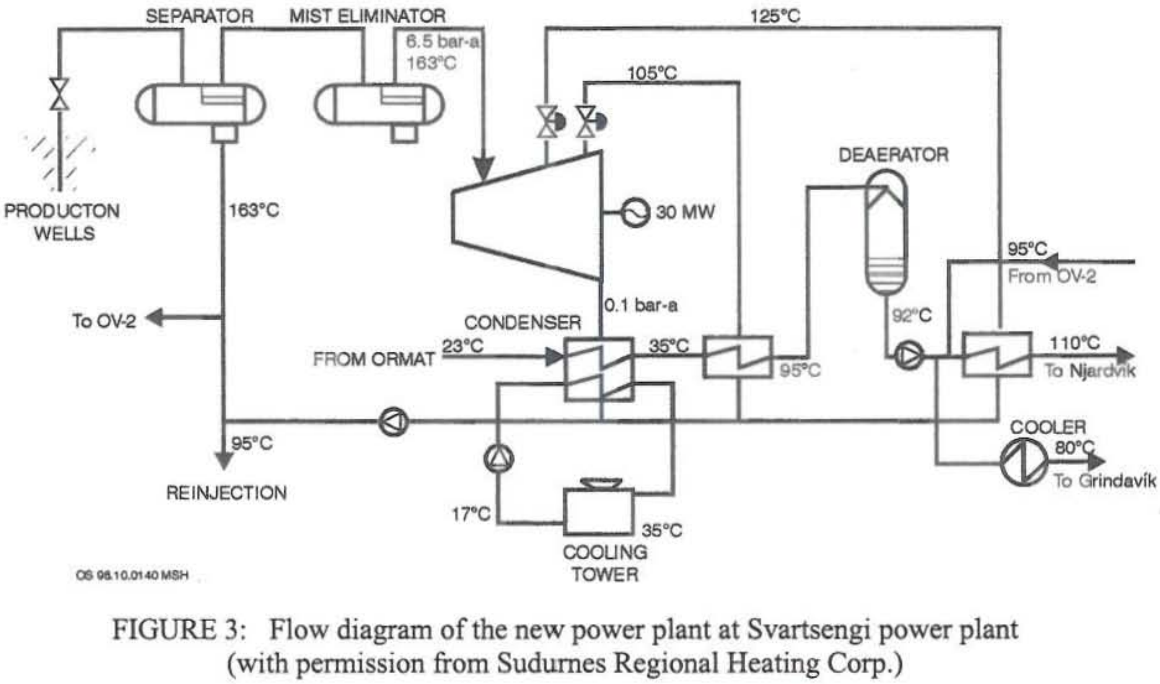

FIGURE 3: Flow diagram of the new power plant at Svartsengi power plant (with permission from Sudurnes Regional Heating Corp.)

heating load as measured by a flow meter. It is assumed that the turbine is run at maximum load (30 MW). If the turbine load is reduced the extraction pressure drops below 1.3 bar-a at some point. Then a by-pass control valve opens to maintain the pressure at 1.3 bar-a. At the same time, the check valve reduces the steam coming from the extraction. Chimney valves in OV-2 control the pressure at 1.3 bar-a at that side, so that the 6 MW turbine and the Ormat turbines will not be disturbed because of variability in low-pressure steam in OV-5.

The condenser pressure is controlled by the temperature of the cooling water from the cooling tower. The mixture of condensate water with brine is controlled by two valves that are operated by the same regulator (one opens whereas the other closes).

3. DESIGN PARAMETERS

Conservative design is the main need for co-generation geothermal power plants to operate at high availability and capacity factors. Geothermal fluids all tend to cause scaling and corrosion to some extent. Also, geothermal reservoirs degrade with time. Conservative design means using the minimum geothermal fluid consistent with plant economic factors and selecting equipment that will not be pushed to design limits and has reasonable margins for varying fluid conditions, scaling, corrosion and different loads. This includes the following items:

- High scaling factors for heat exchangers and the condenser
- Proper corrosion allowance for piping and vessels
- Larger turbines
- Wide-range control valves
- Redundant equipment for high maintenance services

Another important factor that should be considered in geothermal power plants is to keep the design as simple as possible. Piping systems, valves, expansion joints and connections can create maintenance problems.

3.1 Turbine

Because of the nature of geothermal resources, geothermal steam turbines differ in several aspects from conventional ones. The large mass flow rates, low density and enthalpy of geothermal steam require large inlet pipes, simplified control valves and relatively few turbine stages for generating moderate power. In addition, since geothermal steam usually contains high amounts of corrosive gas, special design features should be considered to extend service life that usually reduces turbine efficiency. In most other aspects, the geothermal steam turbines are fairly like conventional ones.

The configuration of the geothermal steam turbines depends primarily on the type of application and generating capacity. Generally, single-flow units are employed at 25-30 MW. Double-flow units may be used up to 60-70 MW, while four-flow units are required for higher ratings. For this reason, manufacturers usually prefer to build low capacity units that can be preassembled in the factory.

The internal efficiency of a turbine is defined as the expansion line efficiency for the entire group of stages. It includes losses due to moisture, leakage, windage and imperfections in the flow paths. It also includes performance gains due to reheating since losses in a given stage appear as heat that can be utilized by the following stage. The internal efficiency of a well-designed turbine can be estimated from the following relationship (Kestin et al., 1980) (variables are explained in Nomenclature):

$$\eta_{isen} = 0.87(1 - \frac{0.01 n \rho_i}{m_i}) f \tag{1}$$

where f is a correction factor, typically having a value between 0.92 and 0.94 depending on the main steam pressure.

Main steam pressure

Power generation potential of the production wells, wellhead pressure and rate of its decline, and wetness of the turbine exhaust are the main parameters that should be considered in the selection of the main steam pressure. Power output of a turbine is given by the following equation:

$$W_t = m_r \times \Delta h_{isen}\ \eta_{isen} \tag{2}$$

As can be seen from the above equation, the power output is proportional to the steam production of the well and the enthalpy drop Δh_s through the turbine. If separator pressure is set high, the total output and steam mass fraction of the well, x decreases based on its characteristic curve as shown in Figure 4, while enthalpy drop increases. Figure 5 shows the variation of power output versus main steam pressure for well SG-7 at the Svartsengi field.

The main steam pressure is also limited by the wetness of turbine exhaust which should not exceed 12-13%. Otherwise, severe blade erosion will result. As can be seen from Figure 5, for geothermal fluid that comes from the production wells (with enthalpy 1140 kJ/kg), main steam pressure of 5-6.5 bar-a is an acceptable range to limit exhaust wetness to the allowable value at the condenser pressure of 0.1 bar-a.

Another parameter that should be taken into account for selection of the main steam pressure is wellhead pressure and its decline rate. Figure 6 shows decay tendencies of the production wells at Svartsengi over 15 years operation with separator pressure about 5.5-6.5 bar-a. Obviously, throttling of the production wells prevents the fast fall in discharge of the early years and therefore the slower rate of decline. James (1995) recommended that discharge to be throttled to 2/3 as maximum horizontal values and relieved over the early years of production to sustain a constant discharge. According to this rule, the wellhead pressure for the Svartsengi production wells can be reduced up to 10-11 bar-a.

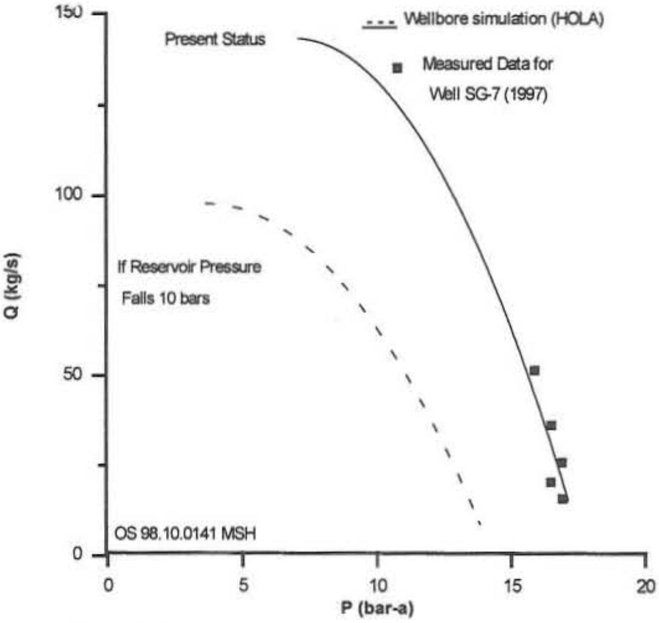

FIGURE 4: Pressure-flow characteristics of production wells

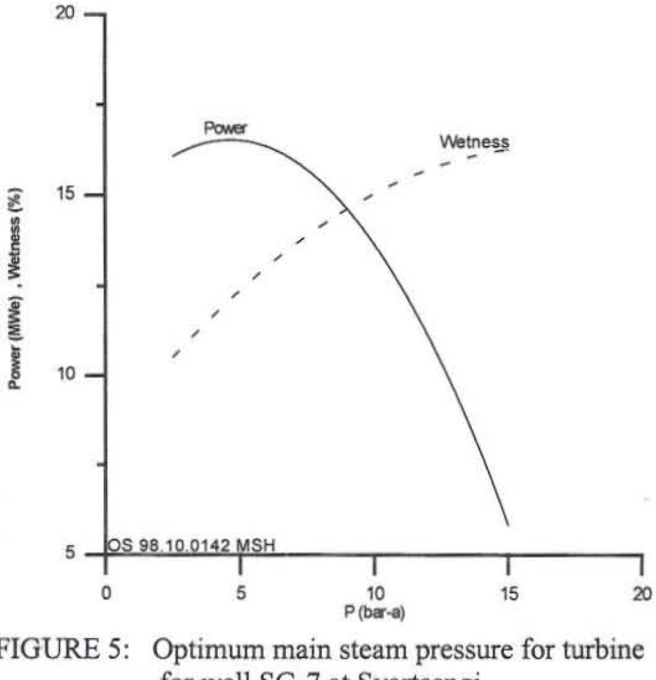

FIGURE 5: Optimum main steam pressure for turbine for well SG-7 at Svartsengi

Special design features

The enthalpy of geothermal steam is much smaller than for thermal power plants. Therefore, it is very important to minimize the leaving loss of the turbine which varies from10 to 40 kJ/kg according to the turbine capacity and the last-stage blade length, in order to improve the steam consumption. This value for the OV-5 turbine is about 16.5 kJ/kg which is determined based on the last stage efficiency and annulus velocity at turbine exhaust.

Integral shroud design (ISB) increases the flexibility of a long blade to minimize the leaving loss. With integral shroud design, adjacent blades come in contact at the shroud due to twist-back force generated by centrifugal force resulting from revolution of the rotor. The mechanical damping generated by this contact reduces vibration stress to less than 20% of the vibration stress of the grouped blades (Yoshida and Saito, 1997).

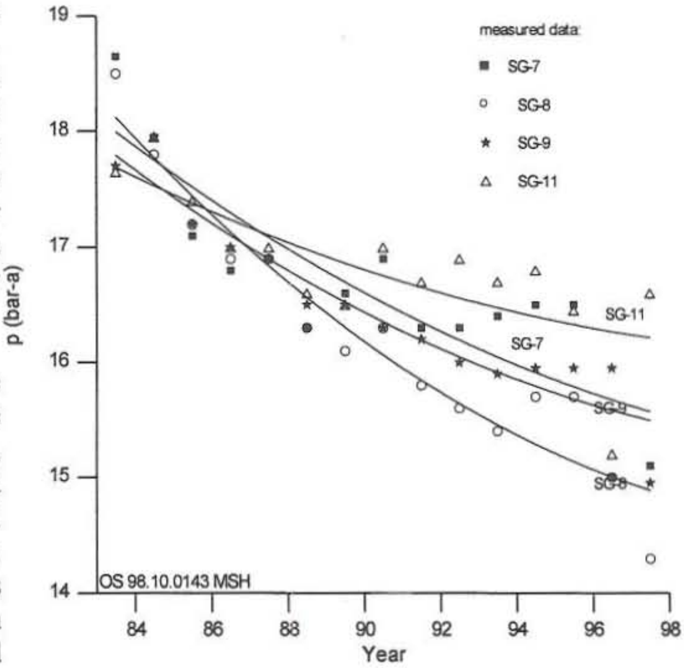

FIGURE 6: Wellhead pressure decline of production wells at Svartsengi

The higher exit wetness of geothermal turbines, compared with conventional ones, requires the following design considerations for the last-stage blades:

- Drain catcher and removal groove for exiting water droplets to improve stage efficiency.
- Stellite strip protection on blade tips of the last stage (1/3 of blade length) that are fixed to the blade silver brazing.
- Impinging damage is considerably lower with a reaction turbine compared to an impulse turbine because of its lower steam velocity.

Because of the low enthalpy, the number of stages for geothermal turbines is less than for conventional ones. The OV-5 turbine has 10 reaction stages.

3.2 Condenser

In co-generation geothermal power plants, there is the possibility of absorbing rejected heat in the condenser for a district heating system. Total rejected heat in a condenser is given by the following equation:

$$\frac{Q_c}{W_t} = \frac{1}{\eta_u} - 1 \tag{3}$$

It means that a geothermal power plant with 10% efficiency rejects 9 kWh of heat for every 1 kWh of generated power. Hence, the utilization factor of geothermal fluid can be improved considerably in co-generation plants. For this purpose, a shell-and-tube condenser is the proper type.

Non-condensible gases in the main steam, which is several hundred times as high in concentration as in a thermal power plant, have great effects on the optimum condenser pressure. Non-condensible gases in the condenser are extracted by a gas extractor and discharged and diffused in the atmosphere. As the condenser pressure becomes low, the steam required to drive the jet-steam ejector or the auxiliary power for the vacuum pump increases, so the net output decreases no matter which kind of gas extractor is used. Therefore, the optimum condenser pressure for geothermal plants, when gas concentration in the main steam is high, tends to be high.

Generally, steam ejectors are economic when the weight fraction of non-condensible gases is less than 1% while the liquid -ring vacuum pumps are suitable for 1-2% gas fractions (Lazzeri et al., 1995). For higher amounts of non-condensible gases, a hybrid system and compressors are used. For gas contents exceeding about 10%, it is necessary to use a back-pressure turbine instead of a condensing turbine.

The weight fraction of non-condensible gases of the production wells in the Svartsengi field in recent years is 0.15-3.5%, measured at separator pressure about 5.5 bar-a (Bjarnason, 1996). The highest gas concentration belongs to well SG-10 that is located at the steam cap of the field. A hybrid system of steam-jet ejector and liquid-ring vacuum pump has been selected for power plant OV-5 to sustain 0.1 bar-a vacuum in the condenser. This system contains two 50% units and a two-stage steam ejector as a back-up.

3.3 Heat exchangers

Plate and shell-and-tube heat exchangers are the most usual types used as an after-heater and final-heater in a co-generation geothermal power plant. Type selection of heat exchangers depends on the following parameters:

- Flow and fouling rates of geothermal fluid
- Temperature and pressure at both sides
- Capital and maintenance cost

All of the heat exchangers of power plant OV-5 are shell-and-tube type. The amount of heat transfer can be determined by the following equations:

$$q = U_o A_o F \Delta T_m = m\, C_p (T_{co} - T_{ci}) \tag{4}$$

$$\frac{1}{U_o} = \frac{1}{K_{hi}}\left(\frac{A_o}{A_i}\right) + r_i\left(\frac{A_o}{A_i}\right) + \frac{D_o}{2K_w} Ln\left(\frac{D_o}{D_i}\right) + r_o + \frac{1}{K_{ho}} \tag{5}$$

$$\Delta T_m = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{Ln\dfrac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}} \tag{6}$$

As can be seen from the above equations, the fouling factors (r_i and r_o) and the terminal temperature difference has a significant impact on the amount of heat transfer. The large difference between clean and fouled conditions can produce many operational problems. Since no reliable method is available to predict heat exchanger fouling, testing is required to determine the most economical cleaning frequency of a heat exchanger. The design fouling factors of after-heaters and final-heaters in OV-5 are 0.00012 and 0.00035 m^2 K/W, respectively.

The terminal temperature difference is also an important parameter in optimizing heat exchanger design. It defined as

$$TTD = T_{hi} - T_{co} \tag{7}$$

If this value increases, the logarithmic mean temperature also increases, which reduces the required heat transfer surface area. However, as terminal temperature difference increases, the efficiency of the thermodynamic cycle decreases and, therefore, it should be selected properly. This value is, respectively, 10 and 15°C for after-heaters and final-heaters of power plant OV-5.

4. MODELLING

4.1 Simulation environment

The Modular Modelling System (MMS) is a simulation code for modelling power plant dynamic performance (Framatome Technologies, 1998). It is intended for use during both plant design and operation. The MMS code permits simulation, during the design stage, of a wide range of transients which may occur with a proposed plant configuration, for example the consequence of a heater drain pump trip. It permits investigation of a possible cause of a transient that has actually occurred and its consequence had it been permitted to continue without interruption to completion. It can be used for performance analysis of a cogeneration power plant at different conditions, such as different conditions of production wells or different thermal and electrical production of the power plant. It also permits testing of alternative control system configurations during the design stage, leading to an optimized design.

A model of a geothermal power plant of an arbitrary configuration may be assembled from pre-programmed generic MMS component models, uniformly called *a module* in MMS. Each significant physical component of the plant is represented by a module. A model is produced by invoking as many of these generic modules as necessary. The interconnection and sufficient data to describe the physical component should be defined. Because the MMS is so structured that each module may be added, replaced, or removed without changing the other modules, it is easy to explore the effect of design alternatives. The modularity concept includes system analysis and an integration package used by MMS.

The modules in the MMS code are developed from the first principles (conservation of mass and energy) as a lumped parameter model of components which are recognizable as equivalent to components used in a power plant such as heaters, turbine, and condenser. For example, a single module representing a reaction turbine stage is provided; this module can be employed several times with different parameter inputs to represent the multi-stage reaction turbine model. The lumped parameter approach also permits a single component to be represented by different modules at different levels of complexity to meet different modelling needs.

The procedure for developing a geothermal power plant model from conception to loading the model can be summarized as follow:

- **Draw model schematic**. The model schematic is a simplified process schematic showing all plant components to be modelled.
- **Draw interconnection schematic**. The interconnection schematic is a reconstruction of the model schematic using MMS modules.
- **Prepare model parameters**. This step consists of calculation or definition of all constants and interconnections for each module in the interconnection schematic.
- **Model control system**. The control system used in the model is defined.
- **Fill out worksheets**. User worksheets for each module in interconnection schematic are completed with data from the parameterization and control system definition.
- **Code model**. The model is coded directly from the worksheets.
- **Generate executable program**. The program is translated and compiled using the appropriate command for the computer.

4.2 Governing equations for the turbine

The turbine model simulates ten reaction stages, and a multiple turbine control valve. The turbine is formulated for variable speed operation but the shaft speed is calculated by the generator model. The performance calculations are applicable for speed variations.

The TURBR module models a reaction stage of a multi-stage turbine, assuming 50% reaction. TURBR optionally models exhaust and windage losses. Heat loss from metal to the ambient is modelled and is used to calculate the turbine casing temperature. In addition, another energy balance is used to calculate the temperature of the rotor/blade, which is heated by entering steam.

The TURBR module has up to three energy balances. One energy balance is for the steam, the second is for the turbine casing metal, and the third energy balance calculates the rotor/blade ring temperature, which approaches the entering steam temperature. In contrast, the turbine casing metal temperature approaches the leaving steam temperature. The energy balance for the steam accounts for the shaft power produced by the fluid and the heat transferred to the metal. The casing metal energy balance includes heat transferred from the steam and heat lost to the ambient. The rotor/blade metal energy balance is calculated using the TMETAL subroutine. TMETAL models conservation of energy for the metal and includes the heat transferred from the steam and heat lost to the ambient.

The TURBR module assumes no energy storage in the steam. The exit specific enthalpy, before accounting for heat transfer to the metal, is calculated using the efficiency as follows:

$$h_{el} = h_{se} - (h_{se} - h_{isen})\eta_t \tag{8}$$

For a 50% reaction turbine stage, the stage efficiency can be calculated as follows:

$$\eta_t = 2K_e \frac{V_b}{V_i}\left[(\cos\alpha)\frac{\sqrt{2}}{2} - \frac{V_b}{V_i} + (\cos\alpha)\sqrt{1 + (\frac{V_b}{V_i})^2 - \sqrt{2}(\cos\alpha)\frac{V_b}{V_i}}\right] \tag{9}$$

For a reaction stage, the kinetic energy entering the moving blades is proportional to the static specific enthalpy drop across the stage, and the steam velocity V_I may be calculated using

$$V_I = \sqrt{2(h_{se} - h_{sl})} \tag{10}$$

To simplify the calculations, TURBR uses h_{isen} to approximate h_{sl}. The velocity ratio may be expressed using a proportionality constant as follows:

$$V_I = \sqrt{2(h_{se} - h_{isen})} \tag{11}$$

The velocity ratio may be expressed as follows:

$$\frac{V_b}{V_I} = \frac{K_{v1}N}{\sqrt{h_{se} - h_{isen}}} \tag{12}$$

The velocity parameter K_{v1} may be backed out from the turbine performance data at the design operating point, which may be assumed to be the point of maximum efficiency. By differentiating the equation given for efficiency with respect to the velocity ratio and setting it equal to zero, we can solve the velocity ratio where the efficiency is at a maximum. In this way, we find that the maximum efficiency is at the following velocity ratio:

$$\left[\frac{V_b}{V_i}\right]_{\max} = \frac{\sqrt{\frac{1}{\cos^2\alpha} - 1 + \frac{\cos^2\alpha}{2}} - \frac{\sqrt{2}}{2}\cos\alpha}{\frac{1}{\cos^2\alpha} - 1} \tag{13}$$

Using data at the design operating conditions and assuming that the maximum efficiency is at those conditions, we can use the above equations to back out K_{v1} as follows:

$$K_{v1} = \frac{\left(\frac{V_b}{V_i}\right)_{\max} \sqrt{h_{se,design} - h_{isen,design}}}{N} \tag{14}$$

The exhaust loss at the last stage of a multi-stage turbine has the effect of reducing the energy extracted from the steam and increasing the leaving specific enthalpy. The exhaust loss is assumed to be proportional to the square of the steam leaving velocity. Therefore, the exit specific enthalpy after accounting for the exhaust loss is

$$h_{ue} = h_{el} + K_{el}\left[\frac{m_{sl}}{\rho_{sl}}\right]^2 \tag{15}$$

Shaft power produced by the turbine stage includes the effect of windage loss. The windage torque is assumed to be proportional to the square of the speed and to the density of the steam. Therefore, the windage power loss is proportional to the cube of the speed

$$W_t = m_{se}(h_{se} - h_{ue}) - K_{wl}\rho_{sl}N^3 \tag{16}$$

The effect of heat transfer to the metal is calculated by using the following energy balance:

$$h_{sl} = h_{ue} + \frac{q_s}{m} \tag{17}$$

The rate of heat transfer to steam from metal has up to two terms. They correspond to the heat transfer to the casing metal and the optional heat transfer to the rotor/blade

$$q_s = q_{rs} + q_{cs} \tag{18}$$

The casing metal temperature T_c can be calculated by the following equations:

$$\frac{dT_c}{dt} = \frac{q_{cs} - q_{ca}}{K_{mc}} \tag{19}$$

$$q_{cs} = K_{um}(T_{ca} - T_{se}) \tag{20}$$

$$q_{ca} = K_{ua}(T_{ca} - T_{amb}) \tag{21}$$

The flow through TURBR is calculated using the following equation:

$$m_{se} = m_{sl} = K_w\sqrt{\rho_{se}p_{se}\left[1-\left(\frac{p_{sl}}{p_{se}}\right)^2\right]} \tag{22}$$

The rotor temperature can be calculated based on a metal energy balance (TMETAL subroutine):

$$\frac{dT_r}{dt} = \frac{q_{rs}}{K_{cm}} = \frac{K_h K_w}{K_{cm}}(T_r - T_{se}) \tag{23}$$

The steady-state rotor metal temperature is closer to the steam temperature for larger values of the weighting factor K_{wt}

$$T_{final} = K_{wt} T_{se} + (1 - K_{wt}) T_{ca} \tag{24}$$

The response time can be slowed by increasing the metal mass or its heat capacity and by decreasing the heat transfer parameter:

$$\tau = \frac{K_{cm}}{K_{ht}} \tag{25}$$

The TURBV module models multiple control valves of the governing stage of a multi-stage steam turbine. In order to reduce throttling losses, the inlet steam flows through two or more valves. Each controls the flow to a separate group of first stage nozzles. This module calculates the total flow admitted to the first stage, accounting for the effect of choking. In addition, the TURBV module models the valve program, determining the number of nozzles receiving flow and passes that information to the first stage of the turbine to account for the effect of multiple admission valves on its pressure-flow calculation. The pressure-flow relationship used by the first stage is a function of the number of valves open and partially open in TURBV. The valve position, which varies from zero to one, determines the number of valves that are open or partially open according to the following logic:

$$K_v = \min\left[\max\left(\frac{Y}{K_{vp}}, 1\right), 10\right] \tag{26}$$

The flow through TURBV is calculated using the following equation:

$$m_{sl} = \max(Y, K_{vp} Y_{\min}) . K_{wv} (1 - 0.0725 \Delta p_v) \sqrt{\rho_{se} p_{se} \Delta p_v} \tag{27}$$

4.3 Model parameters

Preparation of model parameters is the most important step in the model generation process. The main objectives are to find accurate parameters for each component in the model and to define the model initial conditions accurately enough to bring the model to an initial steady state. The following procedure is used for preparing required parameters of the turbine model (Figure 7):

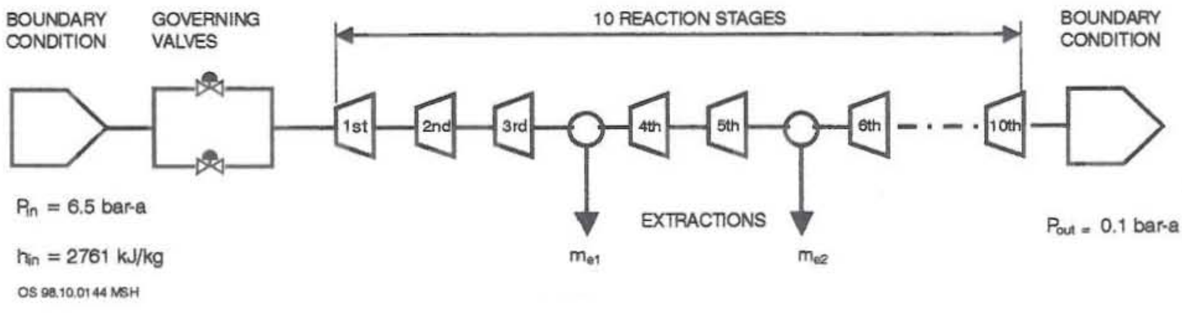

FIGURE 7: Schematic of the turbine model

Collect required physical data

- Number of control valves
- Number of extractions
- Annular exhaust area

Specify required operating point data

- Inlet and outlet mass flow
- Inlet pressure and enthalpy
- Mass flow of extractions
- Isentropic turbine efficiency
- Condenser pressure
- Steam wetness of each stage

Calculate required turbine parameters

Exhaust enthalpy of a turbine can be calculated by the following equations:

$$h_{out} = ELEP + K_{el}\left(\frac{m_{sl}}{\rho_{sl}}\right)^2 \tag{28}$$

$$ELEP = h_{in} - \Delta h_{isen} \cdot \eta_{isen} \tag{29}$$

Inlet enthalpy for each stage is determined based on the assumption that the works of each stage are equal

$$h_{j+1} = h_j - \frac{m_{tot}(h_{in} - h_{out})}{m_{tot}\sum_{j=1}^{10}\frac{m_j}{m_{tot}}} \tag{30}$$

The efficiency of each stage is estimated by the Baumann rule for a wet turbine performance which states that the actual efficiency equals the dry expansion efficiency (assumed to be 0.85), multiplied by the average dryness fraction in the stage.

Check internal variables

Some variables such as enthalpy and quality at extraction can be checked to be sure that they are in acceptable ranges.

5. PERFORMANCE ANALYSIS

The utilization efficiency of a geothermal co-generation power plant can be defined as that proportion of the heat input which is utilized as work and heat. The heat input is the difference between the enthalpy of the geothermal fluid (h_r) coming from the well and the enthalpy of the saturated water at reference temperature (h_o). This difference represents the maximum heat which can be extracted from the fluid.

$$\eta_u = \frac{W_t + Q_h}{m_r(h_r - h_o)} \tag{31}$$

The discharge enthalpy of five deep production wells in the Svartsengi geothermal field has been measured 1010-1140 kJ/kg (Björnsson et al., 1998). In the following calculations, 1075 kJ/kg is considered as the average enthalpy of the geothermal fluid and 5°C is used as the reference temperature which is the mean ambient temperature in Iceland.

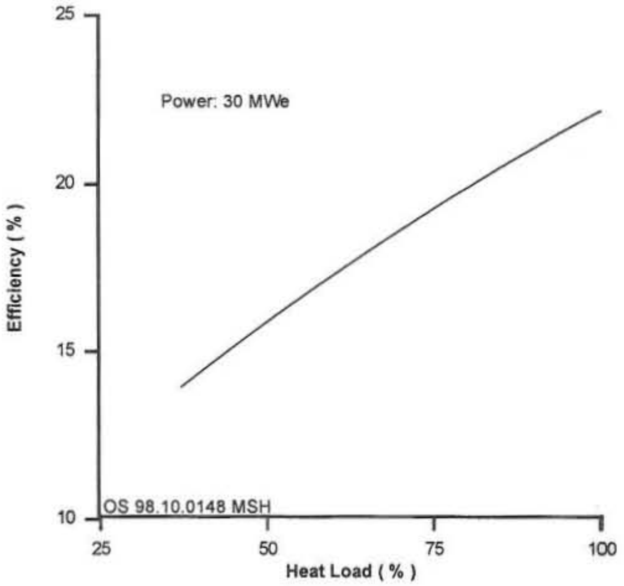

FIGURE 8: Utilization efficiency of Svartsengi power plant OV-5

Figure 8 shows the utilization efficiency of the OV-5 power plant in Svartsengi at different heat loads and 30 MWe power output. As can be seen, efficiency increases with heat load. This is not only due to the increased thermal output, because the well flow rate (m_r), also increases. But the main reason is that the power output is fixed and, therefore, the turbine exhaust flow and rejected heat at the condenser increases. The energy balance calculations show that the utilization efficiency of the OV-5 power plant is improved by 15% using extraction steam instead of high pressure steam for district heating.

The energy balance for the Svartsengi power plant is shown in Figure 9. The main losses are the rejected heat through the brine and cooling water. If the mineral content of the brine permits, it can be used for heating the fresh water, like the Nesjavellir power plant (Figure 10). But the recovery amount of the rejected heat at the condenser is determined by the district heating load and power generation of the plant.

As mentioned before, the thermal output of power plant OV-5 is controlled by turbine extractions. Figure 11 shows the temperature distribution through the turbine stages that are determined by the turbine model. In order to heat fresh water up to 95°C and 110°C by after-heaters and final-heaters, it is necessary to use turbine extractions at 100-110°C and 115-125°C, respectively. As can be seen in

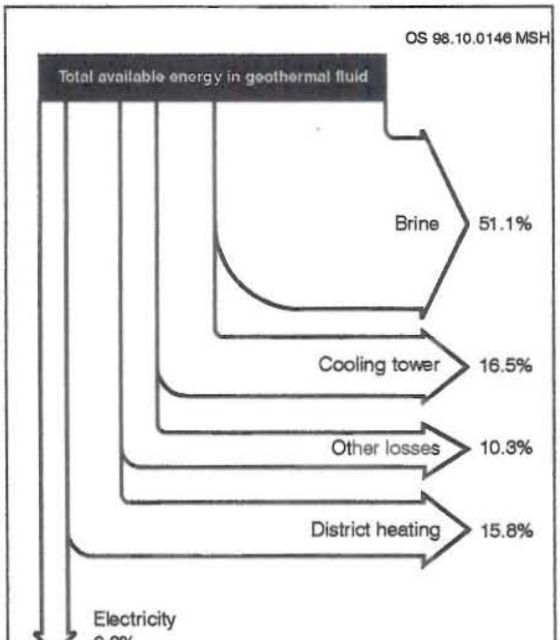

FIGURE 9: Energy balance for the Svartsengi power plant OV-5

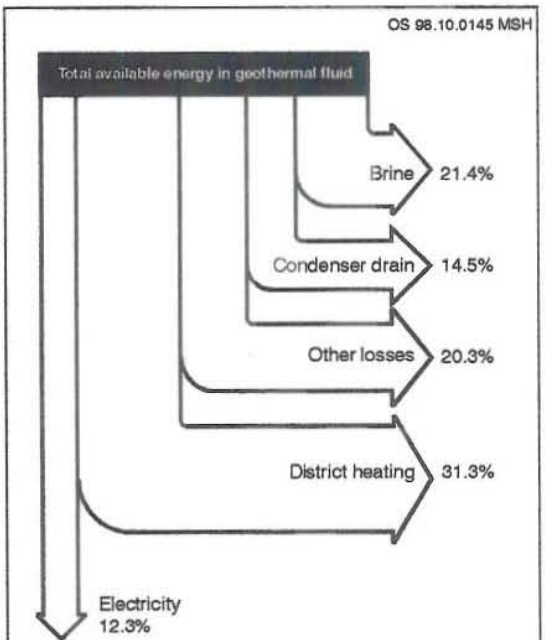

FIGURE 10: Energy balance for the Nesjavellir power plant

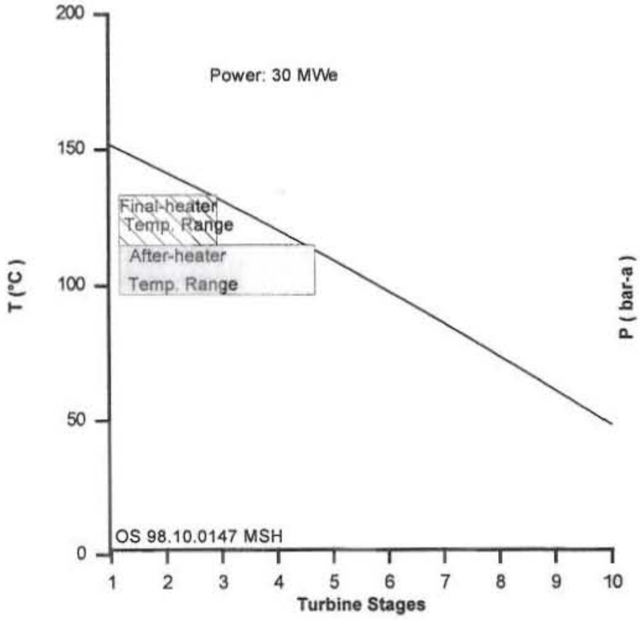

FIGURE 11: Temperature distribution through turbine stages

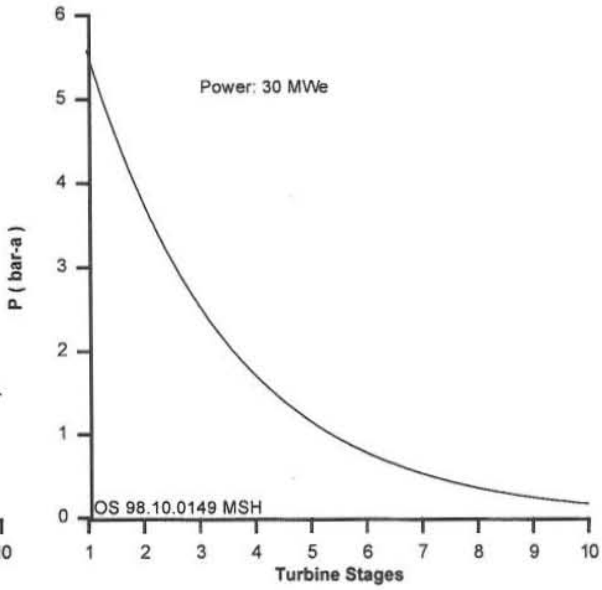

FIGURE 12: Pressure distribution through turbine stages

Figure 11, turbine extractions between stages 3 and 4 for final-heaters and 5 and 6 for after-heaters provide proper steam for this purpose. The corresponding pressures for these turbine extractions are respectively, 2.75 bar-a and 1.43 bar-a for the full-load power generation (Figure 12). The turbine model shows that at 24 MW_e rated heat load, the pressure of the first and second extraction drops below 2.5 bar-a and 1.3 bar-a, respectively (Figure 13), and the temperature of the extraction steams approach the minimum allowable values. At this point, it is necessary to supply enough steam from the main high-pressure steam. The electrical power of the turbine can be reduced to 21and 24 MWe at the medium and maximum heat load, respectively. Table 1 shows the comparison of the turbine model results and the heat balance diagrams of OV-5 at different heat loads. Figures 13 and 14 and Table 1 show that the static and dynamic behaviour of the power plant at the different electrical outputs and heat loads can be analysed by simulation.

TABLE 1: Comparison of the turbine model results and Fuji heat balance diagrams (Fuji,1998)

Thermodynamic properties	Maximum heat load		Rated heat load		Medium heat load		Minimum heat load	
	Model	**Fuji**	**Model**	**Fuji**	**Model**	**Fuji**	**Model**	**Fuji**
m_{in} (kg/s)	85.92	85.29	80.38	80.34	76.46	76.43	72.83	72.99
m_{out} (kg/s)	44.92	44.29	49.38	49.34	55.46	55.43	57.81	57.99
$p_{ext.1}$ (bar-a)	2.78	2.75	2.91	2.91	2.92	2.88	3	n.a.
$p_{ext.2}$ (bar-a)	1.47	1.43	1.62	1.62	1.83	1.85	1.92	1.95
$h_{ext.1}$ (k J/kg)	2640	2631	2650	2646	2655	2651	2663	n.a.
$h_{ext.2}$ (k J/kg)	2555	2545	2571	2566	2589	2591	2598	2601

n.a. = not available

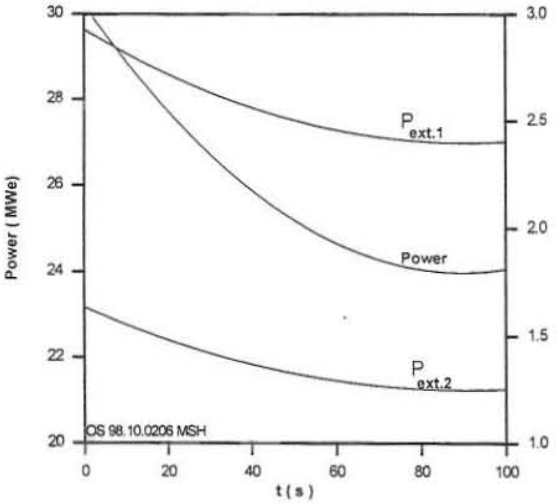

FIGURE 13: Pressure variations of extractions at the different electrical loads

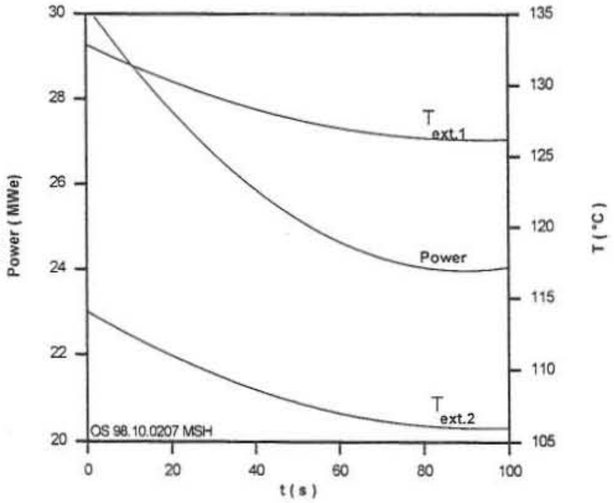

FIGURE 14: Temperature variations of extractions at the different electrical loads

6. CONCLUSIONS

1. The total performance of the new geothermal co-generation power plant at Svartsengi is improved by using turbine extractions instead of high pressure steam, for heating fresh water to 110°C in heat exchangers. Energy balance calculations show that the utilization efficiency of the power plant OV-5 is improved by 15% with this type of operation and 14-22% at different heat loads.
2. The turbine model shows that at 21-24 MW_e electrical outputs and different heat loads, the pressure of the first and second extractions drops below 2.5 bar-a and 1.3 bar-a, respectively. At this point, it is necessary to supply high-pressure steam to heat exchangers.
3. For dry saturated steam entering the turbine, the optimum pressure is about 5-7 bar-a, to maximize the power output and limit exhaust wetness to 12-13%.
4. The static and dynamic behaviour of the power plant at different electrical and heat loads can be analysed by simulation.
5. The utilization efficiency of geothermal co-generation power plants is usually 15-45% depending on the enthalpy and scaling potential of the geothermal fluid and also the electrical and heat loads of district heating system.
6. A technical and economic assessment is necessary to determine the optimum electrical and thermal output of a geothermal co-generation power plant according to the properties of the geothermal fluid and local energy and economic parameters.

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NOMENCLATURE

A = Area of heat transfer (m^2);
C_p = Specific heat (J/kg);
D = Diameter (m);
$ELEP$ = Expansion line end point (J/kg);
f = Efficiency correction factor;
F = Correction factor for heat exchanger;
h = Enthalpy (J/kg);
h_{ue} = Used energy endpoint (J/kg);
K_{cm} = Metal heat capacity (J/°C);
K_e = Efficiency parameter;
K_{el} = Exhaust loss parameter (m^{-4});
K_h = Heat transfer parameter (W/°C);
K_{mc} = Thermal capacity of casing metal (J/kg);
K_u = Parameter for heat transfer from/to casing metal (W/°C);
K_v = Number of governing valves;
K_{v1} = Velocity parameter (m/rad);
K_{vp} = Reciprocal of the number of governing valves;
K_w = Thermal conductivity (W/m°C);
K_{wt} = Weighing factor;
K_{wv} = Flow parameter (m^2);
m = Mass flow rate (kg/s);
n = Number of parallel flow in the turbine;
N = Turbine speed (rad/s):
p_{se} = Pressure of steam (Pa);
q = Rate of heat transfer (W);
Q_c = Rejected heat at condenser (W);
Q_h = Thermal output (W);
r = Fouling factor;
T = Temperature (°C);
TTD = Terminal temperature difference (°C);
U_o = Overall heat transfer coefficient (W/°C m^2);
V_b = Blade velocity (m/s):
V_i = Steam velocity (m/s);
W_t = Turbine power (W);
Y = Valve position;

α = Angle of approach of steam to moving blade (rad);
ΔT_m = Log mean temperature difference (°C);
Δp_v = Pressure ratio;
η_{isen} = Isentropic efficiency;
η_t = Turbine efficiency;
η_u = Utilization efficiency;
ρ = Density (kg/m^3);
τ = Time constant.

Subscripts
c = Cold;
C = Casing;
e = Entering;
h = Hot;
i = Inside;
l = Leaving;
o = Outside;
r = Reservoir;
s = Steam.

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