Report 12, 1992

BINARY POWER GENERATION FROM WASTE HEAT: A FEASIBLE IMPROVEMENT TO OPERATING GEOTHERMAL POWER PLANTS

Myrna P. Sagun,
UNU Geothermal Training Programme,
Orkustofnun - National Energy Authority,
Grensasvegur 9,
108 Reykjavik,
ICELAND

Permanent address:
Philippine National Oil Company -
Energy Development Corporation,
Engineering and Construction Dept.,
Makati, Metro Manila,
PHILIPPINES

ABSTRACT

Geothermal steam power plants often utilize separated steam to run steam turbines and generate electric power. Except for the case of high enthalpy geothermal reservoirs, the steam flow from the separators is often only 20-25% of the total flow. Therefore, the waste water disposed from the separators represents a significant fraction of the total thermal energy of the resource. Likewise, in the case of non-condensing geothermal steam turbines, the exhaust steam is generally released to the atmosphere at temperatures of above 100°C. The poor efficiency of both cases results in rapid reservoir depletion and loss of useful heat. A binary power plant is considered suitable for the utilization of these resources in the expansion of an operating power plant's installed capacity to generate additional electric power.

The report deals with two cases; the existing binary power plant installation at the Svartsengi geothermal power plant in Iceland, where the turbine exhaust steam is used for production of power, and a proposed binary power plant in Bacman I geothermal power plant in the Philippines, where the heat source for the binary cycle is the waste water from the separators.

Different modes of operating the Svartsengi power plant are considered and estimates made of the potential binary power which can be generated from the surplus steam of the co-generation plant. The concept of the Rankine cycle was used in the power calculations. For Bacman I, the Svartsengi computational procedure was applied for determining the potential binary power generated. Some parameters were, however, modified, i.e. isopentane cycle conditions, cooling water temperatures and heat source temperatures to suit the conditions for Bacman I.

Thermal efficiency calculations of the two distinct power plants give results showing how the additional power from the binary plant can improve the utilization of the mined heat from the geothermal resource. The results show that the production of electric power from the "waste" heat, either in the form of turbine exhaust steam or waste water can improve the thermal efficiency of an operating geothermal power plant. For Svartsengi, the results show that the 10 MW_e of potential binary power which can be generated when the plant is operating at full capacity can increase the plant's thermal efficiency by 7%, whereas when the plant is operating during summertime, there is a 9 MW_e of potential binary power which can be generated from the surplus steam which will increase the thermal efficiency by 14.7%. For the Bacman I geothermal power plant, the potential binary power which can be generated from the wastewater is about 4.6 MW_e if the brine temperature is dropped from 190°C or 180°C to 165°C which corresponds to a 4.3% increase in thermal efficiency. The potential binary power calculations for other water outlet temperatures are also presented.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1. INTRODUCTION

Most of the geothermal resources utilized at present for the production of electricity have temperatures above 150°C. The utilization is mainly achieved by direct-steam or flashed-steam installations, which require high resource temperatures. The process of using geothermal energy to produce power leaves, however, a considerable amount of thermal energy that goes to waste. This thermal energy is sufficient to generate additional electricity by applying non-conventional methods of geothermal power generation.

Interest in the recovery of power from low grade heat sources during the past years has encouraged the development of a number of interesting thermodynamic systems designed to maximize the conversion of thermal energy to mechanical or electric power. Few of these are recent in origin and in some instances were overtaken by later developments in power plant design. One of these to which the present system will be related is the Organic Rankine Cycle (ORC) otherwise known as the Binary Cycle.

A geothermal binary cycle, from a thermodynamic point of view, is simply a heat-operated Rankine cycle in which the primary source of hot fluid is the geothermal fluid. The geothermal fluid is passed through a heat exchanger to heat up a secondary fluid which is vaporized before entering a turbine. This secondary fluid is usually a low-boiling-point organic liquid kept in a closed circuit. The binary cycle has the advantage of being more efficient than the steam cycle for the utilization of low to moderate enthalpy geothermal fluids.

In existing geothermal power plants using high temperature resources, the exhaust steam from the turbines (especially if the turbine is non-condensing), and/or the separated brine are potential low grade heat sources for binary power generation. The utilization of the waste heat from these plants improves their thermal efficiency.

This paper is a preliminary study of the applicability of the binary cycle power generation from waste heat as a means of improving the thermal efficiency of specific geothermal plants. The study puts emphasis on the Svartsengi power plant in Iceland and Bacman-I geothermal power plant in the Philippines. The geothermal power plant at Svartsengi utilizes the exhaust steam from the back-pressure turbines for binary power generation, while Bacman-I will be based on using the high-pressure separated brine from the separators.

2. SELECTED TOPICS

2.1 Svartsengi geothermal power plant (Iceland)

The geothermal power plant at Svartsengi utilizes a high-temperature geothermal reservoir of the same name in Iceland. The geothermal fluid is a brine with a temperature of 230°C to 240°C coming from six production wells. This geothermal fluid could not be used directly for district heating due to the high temperature and salinity of the water. Instead, the brine is first flashed in a high pressure separator at 5.5 bara pressure from which the separated high pressure steam is used to generate electric power in back pressure turbines. The steam exhaust is then used in the heat exchange process to heat fresh water for the district heating system, in what is commonly referred to as co-generation. The separated brine from the high pressure separator is then flashed for the second time in a low pressure flasher and the separated low pressure steam is used in the second stage of the four-stage heating process to heat up fresh cold water to produce potable hot water. After deaeration the hot water is pumped at 80°C to 120°C to the district heating systems for communities on the Reykjanes Peninsula and Keflavik airport.

The power plant at Svartsengi was constructed in two stages with somewhat different process designs but intended to produce similar results. Power plant I was fully commissioned in late 1978 and power plant II in 1981 (Björnsson and Albertsson, 1985). The diagram in Figure 1 shows one of four parallel units in power plant I. Figure 2 shows one of three parallel flow diagrams in power plant II. In the years that followed, a decision to put up a binary power plant was made. This plant is designed to utilize the surplus low pressure steam exhaust from the turbines which would otherwise be wasted.

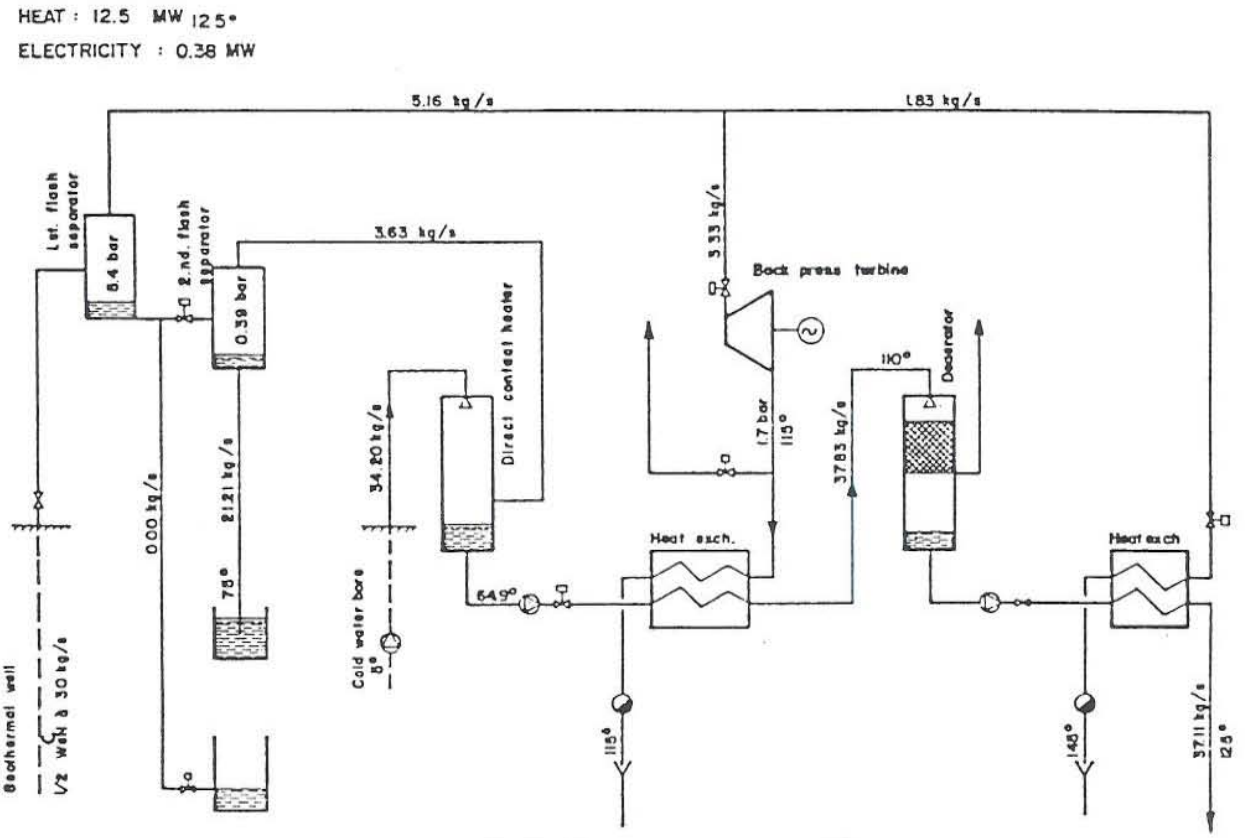

FIGURE 1: Svartsengi power plant I process flow diagram for one flow stream (12.5 MW_t) (Bjornsson and Albertsson, 1985)

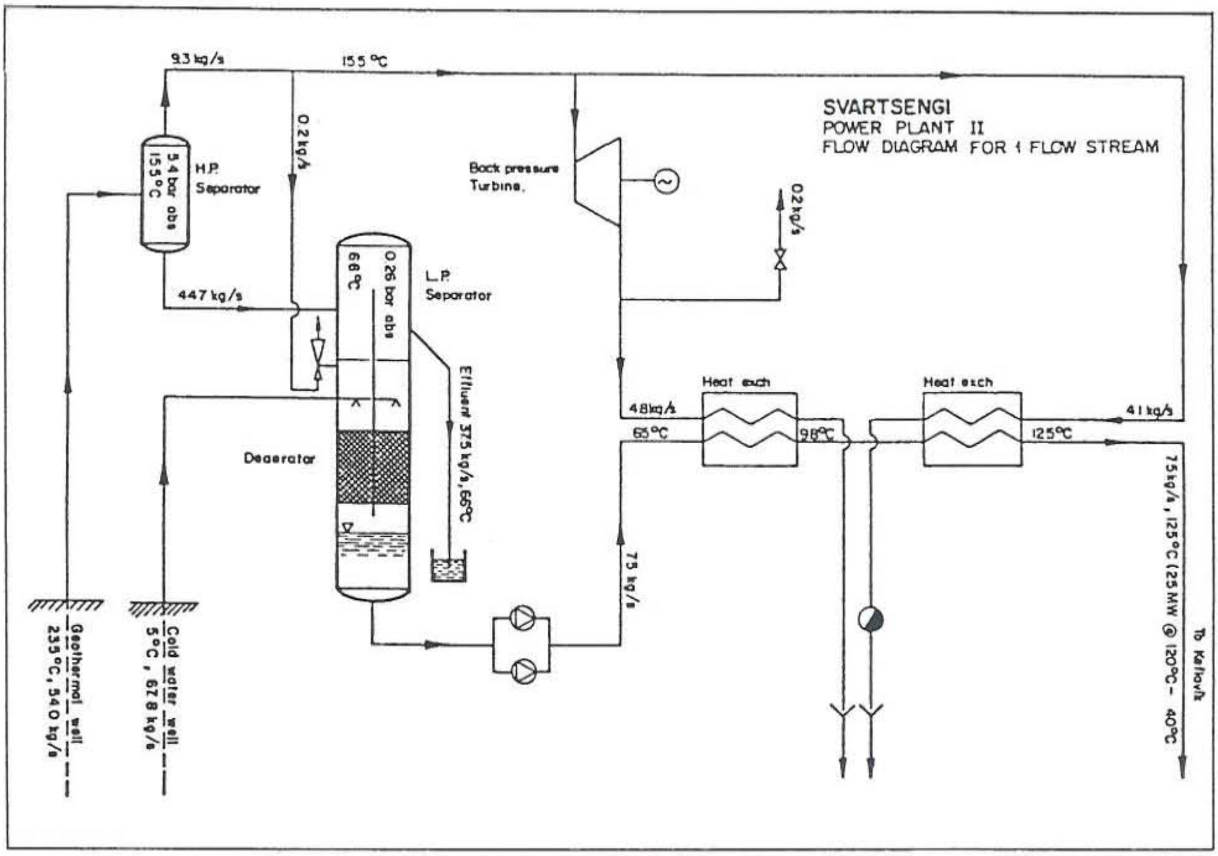

FIGURE 2: Svartsengi power plant II process flow diagram for one flow stream (25 MW_t) (Bjornsson and Albertsson, 1985)

Figure 3 shows a schematic diagram of one of three identical and parallel Ormat units installed in Svartsengi (Ormat Turbines Ltd., Israel). Currently, the installed capacity of the Svartsengi power plant is **125 MW_t** thermal power for district heating systems, **8 MW_e** electric power from conventional steam turbines and **4 MW_e** electric power from three Ormat binary turbines.

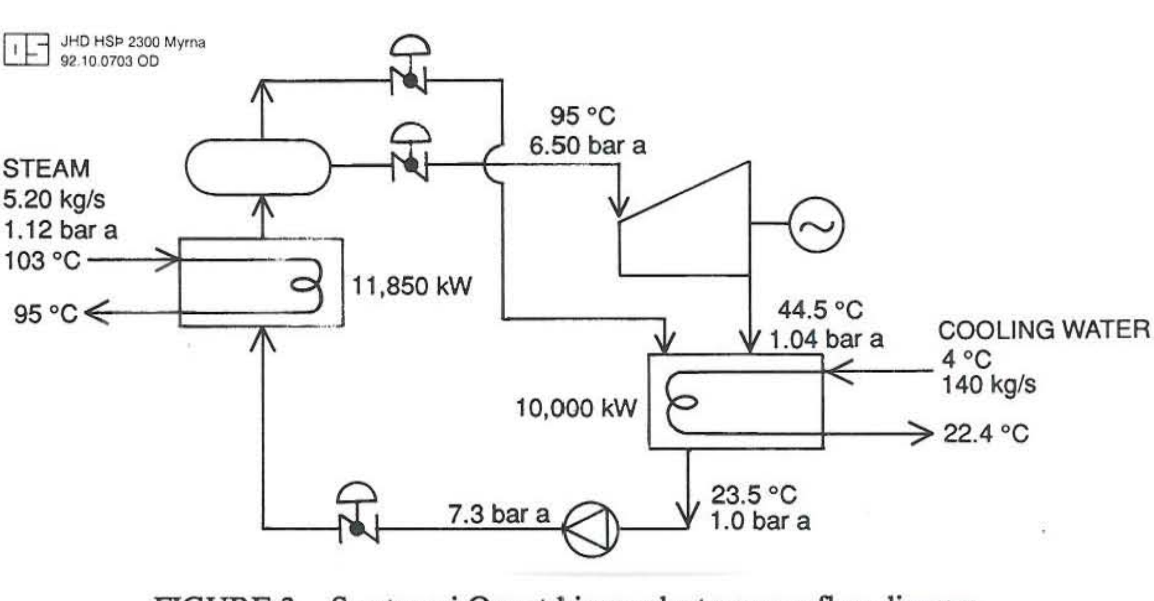

FIGURE 3: Svartsengi Ormat binary plant process flow diagram for one module (1.34 MW_e)

2.2 Bacman I geothermal power plant (Philippines)

Bacman I geothermal power plant is located on the boundary of the towns of Bacon, Sorsogon and Manito, Albay in the southern part of Luzon. The plant has an installed capacity of **110 MW_e** and will be supplied with geothermal fluid from 12 production wells distributed in three production pads namely Pad E, Lower pad H (LPH), and Upper pad H (UPH) with average enthalpies in the range of **1330-1670 kJ/kg**. Figure 4 shows the general arrangement plan of the Bacman I geothermal power plant.

The construction of the fluid collection and disposal system (FCDS) was started in the middle of 1990 and was completed and commissioned in the middle of 1992. The construction of the power house is scheduled to be completed in May 1993 and it is expected that the power plant will be operational by the end of 1993.

The plant is designed in such a way that the two-phase fluid coming from the wells will be piped to the two central separator stations, namely Pad E and Pad H stations. The high pressure steam for the 2 x 55 MW condensing turbines will be separated at these stations at a pressure of 10 bara in Pad H station and 12.5 bara in Pad E station. The separated steam from the two stations will go to a common header and will pass through a pressure reducing station to deliver the steam at a pressure of 7 bars which is the desired inlet pressure for the turbines. The separated brine on the other hand is piped to four reinjection wells distributed in two reinjection pads RA and RC/RD. The topography of the site is such that sufficient head is available from the separator station to the reinjection pad so that reinjection by gravity flow is feasible. It is this reject brine that is being considered for use as the heat source in the proposed binary plant. Application of this technology in the Philippines is the object of this evaluation. It will give an idea of the possible additional energy that can be derived from the reject water of the conventional geothermal power plant.

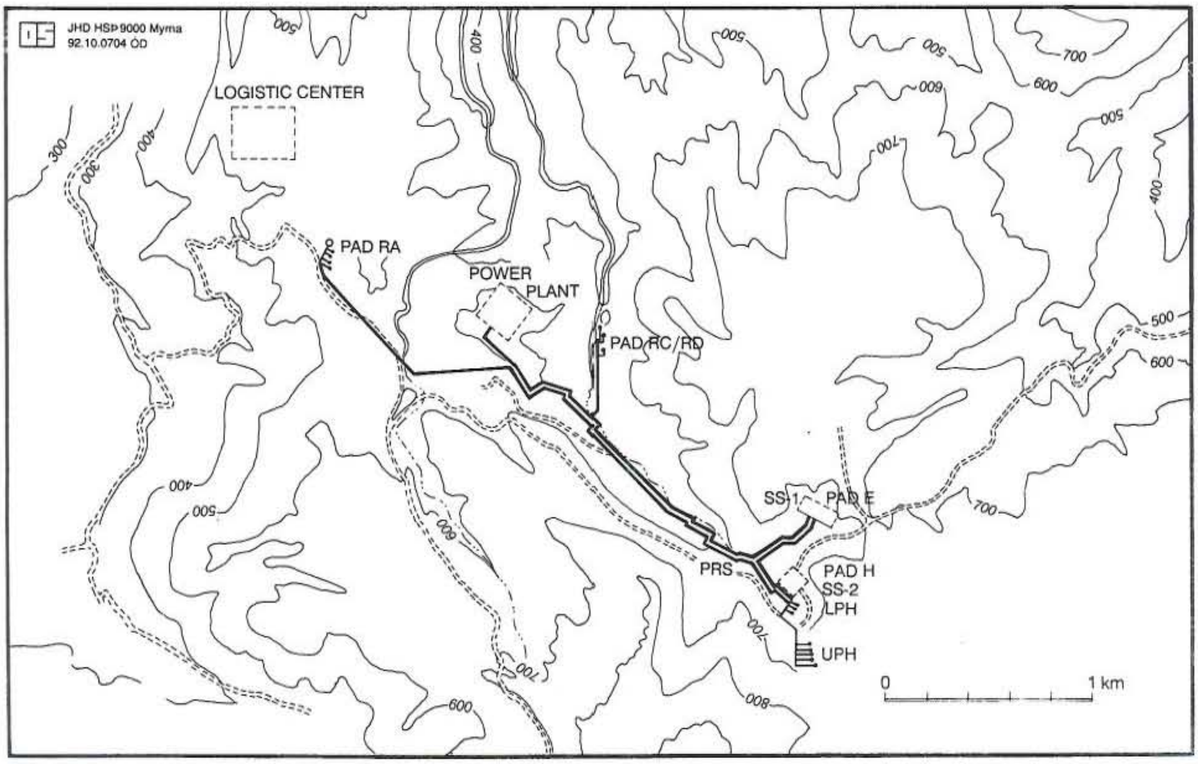

FIGURE 4: Bacman I power plant general arrangement plan

3. GENERAL DESCRIPTION OF BINARY POWER GENERATION

A binary power plant is designed to utilize low grade heat source or otherwise unused process heat to produce useful electric or shaft power. Like the steam turbine, the system is based on the Rankine power cycle but uses the advantages that an organic working fluid has at low temperatures. Such a recovery system that uses waste heat from a geothermal power plant is shown in Figure 5.

In its simplest form, the cycle starts with the heat recovery boiler or evaporator which provides the saturated or superheated vapor. The Rankine-cycle fluid, a hydrocarbon chosen according to heat source, is heated and vaporized in a heat exchanger by heat transfer from the hot geothermal fluid. Droplets of liquid carried over with the vapor must be separated to prevent them from impinging on the turbine blades. The pressurized vapor expands through the organic vapor turbine from which the shaft work produced is used to drive a generator to produce electric power.

The low pressure exhaust vapor from the turbine is passed to the shell side of the condenser where it is cooled and condensed to liquid by cooling water or condensed in large fin air heat exchangers. The resultant organic fluid is pumped back to the vaporizer by a feedpump thus completing the cycle.

The cooling water from the condenser either goes to a cooling tower and is recirculated to the condenser or used as a source of thermal energy in some heating process as in the case for Svartsengi.

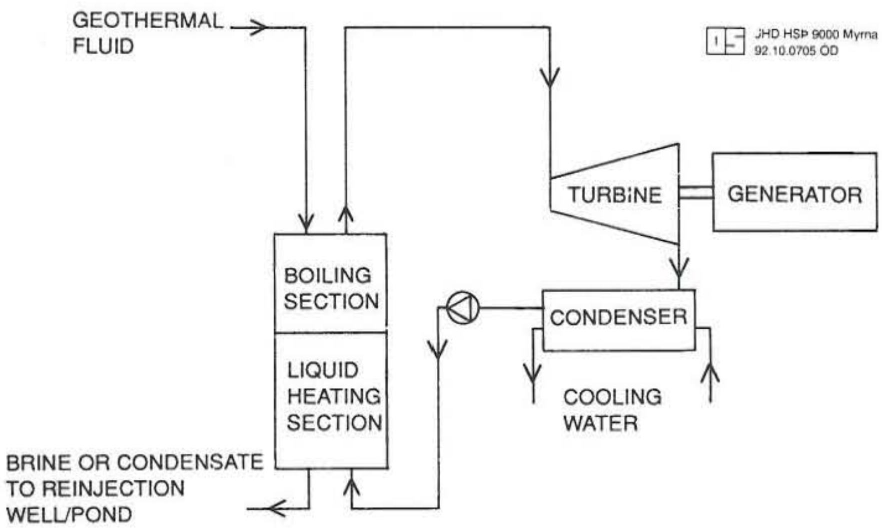

FIGURE 5: Schematic diagram of binary cycle with geothermal fluid as heat source

4. THEORETICAL PRINCIPLES USED IN THE CALCULATIONS

4.1 The ideal Rankine cycle

The basic Rankine cycle is presented schematically and on a *Ts* diagram in Figure 6. The ideal cycle of a simple Rankine power cycle then consists of

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

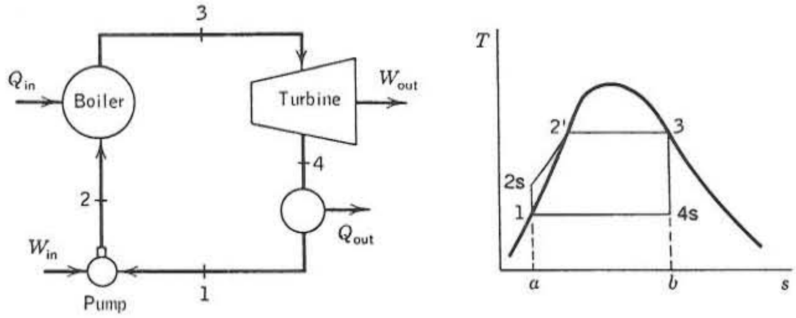

FIGURE 6: Schematic flow and *Ts* diagram of an ideal Rankine cycle (Wark, 1988)

The heat added in process 2-3 may come from a geothermal fluid as will be discussed in this paper or any other low-to-medium temperature source. If the changes in potential and kinetic energies are neglected, the heat transfer to the fluid in the boiler which in this case is a hydrocarbon, is presented on the *Ts* diagram by the area enclosed by states 2s-2'-3-*b*-*a*-2s. The area enclosed by states 1-4s-*b*-*a*-1 then represents the heat removed from the fluid in the condenser. The first law for an open cyclic process indicates that the net heat effect equals the net work effect. Hence, the net work is represented by the difference in the areas for the net input and heat rejection, i.e., area 1-2s-2'-3-4s-1.

The thermal efficiency for the cycle is defined by Equation 1 as

$$\eta_{th} = \frac{W_{net}}{Q_{input}} \tag{1}$$

This parameter is often used to measure the effectiveness of any cyclic heat-work converter. Expressions for the work and heat interactions in the ideal cycle are found by applying the steady flow energy equation to each separate piece of equipment. If kinetic- and potential-energy changes are neglected , the basic equation for each process reduces to

$$q + w = h_{out} - h_{inlet} \tag{2}$$

The isentropic pump work is given by

$$w_{input,pump} = h_2 - h_1 \tag{3}$$

The pump work is also frequently determined within the desired accuracy from the relation

$$w_{input,pump} = v_f(P_2 - P_1); \qquad s_1 = s_2 \tag{4}$$

where v_f is the saturated-liquid specific volume at state 1.

The heat input, the isentropic work output from the turbine, and the heat rejection in the condenser, all on a unit-mass basis, are

$$q_{input,boil.} = h_3 - h_2; \qquad P_3 = P_2 \tag{5}$$

$$w_{out,turb.} = h_3 - h_4; \qquad s_3 = s_4 \tag{6}$$

$$q_{out,cond.} = h_4 - h_1; \qquad P_4 = P_1 \tag{7}$$

The thermal efficiency then of an ideal Rankine cycle may be written as

$$\eta_{th} = \frac{w_T - w_P}{q_{input}} \tag{8}$$

4.2 The irreversible Rankine Cycle

The Rankine cycle discussed in the preceeding section is an idealized process of the conversion of heat to an equivalent amount of net work output. It is a concept which can be approximated closely at times by actual devices, but never matched. Therefore, the thermal efficiencies of the exchange of heat from a source initially in equilibrium and the production of net work output must be less than 100 percent. The effect of irreversibilities on vapor-power-cycle performance can be seen in Figure 7. The *Ts* diagram in this figure represents the actual cycle of the process. In all cases there are frictional losses in the piping which lead to pressure drops. Heat transfer losses occur throughout the equipment and these irreversibilities caused by the flow of the fluid are especially important to consider in the turbine and in the pump. Then, Equations 3, 4, and 6 in the ideal cycle must be corrected by a factor known as the *isentropic,* or *adiabatic efficiency,* η.

The *isentropic turbine efficiency* η_T is defined as the actual work output w_a to the work w_s that could be achieved while expanding isentropically to the same final pressure. This is expressed as

$$\eta_T = \frac{w_a}{w_s} \tag{9}$$

From Figure 7, if kinetic energy change across the turbine is negligible, the isentropic efficiency may be approximated by

$$\eta_T \approx \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \tag{10}$$

FIGURE 7: *Ts* diagram of a power cycle with irreversible turbine and pump performance (Wark, 1988)

Similarly for the pump, for the *same initial state,* and the *same final pressure*

$$\eta_P = \frac{w_s}{w_a} \tag{11}$$

or,

$$\eta_P \approx \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{12}$$

5. SVARTSENGI POWER PLANT EVALUATION

The approach taken in the evaluation of the Svartsengi power plant is first to determine the different possible modes of operating the power plant. By identifying these operating modes, estimates can be made of the surplus steam which is going to the chimney of the power plant in various operating schemes. From these, the potential binary power which can be generated will be calculated. The applicability of this method of using the surplus steam for power production will be assessed according to the amount of heat recovered from the vented steam and its contribution to the thermal efficiency of the power plant. The plant thermal efficiency will also be used as an indicator of the improved exploitation scheme of a geothermal field.

Furthermore, in the Svartsengi evaluation, the calculation for the energy and mass balance in thermal power production was performed using a spreadsheet (Lotus 123) program modified by the author from a model made by S. Thorhallsson. This program makes an energy and mass balance across each piece of process equipment in the power plant. By inputting the data on temperature and flow of heated water either on the inlet or outlet stream, the unknown massflow of the heating steam will be automatically calculated by the program. Printed results of these calculations are shown in the appendices.

5.1 Operation based on initial design: hot water production (125 MW_t)

5.1.1 Energy and mass balance

The Svartsengi power plant design capacity is based on **125 MW_t** thermal power and **8 MW_e** electric power. This thermal power production corresponds to the production of about **148 kg/s** of **120°C** hot water from power plant I (PP I) and **225 kg/s** of **120°C** hot water from power plant II (PP II). The hot water after passing through the radiators in each house is either rejected or recirculated at **40°C** (Björnsson and Albertsson, 1985).

An energy and mass balance calculated using the program developed for PP I and PP II is summarized in Appendix I. It shows that for the original design conditions of the plant i.e. without the Ormat binary turbines, the total production of **373 kg/s** of **120°C** hot water will require **20.64 kg/s** and **27.90 kg/s** of high pressure steam in PP I and II respectively, based on the required massflow of brine in the second flasher. The low pressure steam produced in the second flasher is used to heat the cold water in the direct contact condenser. Also, from the same calculation, **7.32 kg/s** of this high pressure steam is used to heat up the hot water to **120°C** in the final heater in power plant I. Similarly, the final heater in power plant II uses **12.23 kg/s** of high pressure steam. This results in a massflow of high pressure steam for electricity generation equivalent to **13.32 kg/s** in power plant I and **15.67 kg/s** in power plant II.

5.1.2 Calculation of power from steam turbines

Power plant I

To calculate the electric power generated from the available high pressure steam flow in power plant I, first a *Ts* diagram is drawn for the steam cycle and this is illustrated in Figure 8. Using Equation 6 and taking into account the irreversibility of the process as discussed

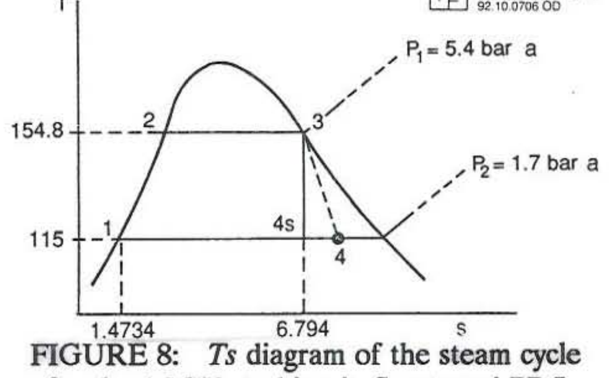

FIGURE 8: *Ts* diagram of the steam cycle for the 1 MW_e turbine in Svartsengi PP I

in Section 4.2, the turbine power output can be estimated by

$$W_{out,turb.} = m_s \eta_T (h_3 - h_{4s}) \tag{13}$$

From the thermodynamic properties of steam, the enthalpy values at the turbine inlet and outlet, denoted by states 3 and 4, respectively, in the *Ts* diagram shown in Figure 8 are found to be

$$h_3 = 2752.1 \ kJ/kg, \qquad H_{4s} = 2550 \ kJ/kg$$

The steam turbines in power plant I have a design steam rate of **8.88 kg/s-MW** (net) (Thorhallsson, S., pers. comm.). Assuming the *generator efficiency*, η_G as 0.95; then, the *isentropic efficiency* of the 1MW turbine can be found from the following equation:

$$\eta_T = \frac{W_G}{\eta_G m_s (h_3 - h_{4s})} \tag{14}$$

where
W_G = net out put of the generator.

Substituting in the equation, the *isentropic efficiency* of the **1 MW** turbine in PP I is

$$\eta_T = \frac{1000}{0.95 \times 8.88 \times 202.10} = 0.59$$

This is rather low efficiency for the turbine. It can be attributed to the small size of the units and that these units were not tailor-made for the specific steam conditions.

Referring to the massflow of steam from the energy and mass balance calculations, it has been found to be equal to:

$$m_s = 13.32 \ kg/s$$

The turbine work produced from this high-pressure steamflow is

$$W_T = 0.59 \times 13.32 \times 202.10 = 1{,}588 \ kW_e$$

The net power output will be equal to

$$W_G = \eta_G W_T \tag{15}$$

and for power plant I this is

$$W_G = 1{,}509 \ kW_e = 1.51 \ MW_e$$

This calculated power output for PP I coincides with the plant installed capacity of 2 x 1 MW turbines.

Power plant II

The *Ts* diagram for the cycle in power plant II is also shown in Figure 9. The *isentropic efficiency*, η_T, of the turbine in PP II is calculated similarly as for PP I but using the design steam rate for the **6 MW** turbine as **6.175 kg/s-MW(net)** (Thorhallsson, S., pers.comm.) and the steam enthalpy at the inlet and exhaust of the turbine as

$h_3 = 2752.1\ kJ/kg, \qquad H_{4s} = 2492\ kJ/kg$

This is found to be approximately equal to

$$\eta_T = 0.66$$

Referring to the energy and mass balance calculations, the available massflow of high pressure steam for the turbine in PP II is

$$m_s = 15.67\ kg/s$$

If a similar calculation is performed, the turbine work output from this steamflow in PP II is:

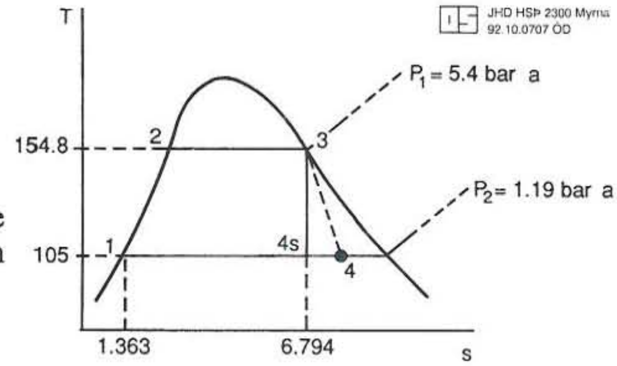

FIGURE 9: *Ts* diagram of the steam cycle for the 6 MW_e turbine in Svartsengi PP II

$$W_T = 2{,}690\ kW_e$$

Assuming the same efficiency for the generator, the net power output in MW_e is

$$W_G = 2{,}556\ kW_e = 2.56\ MW_e$$

This is approximately 40 percent of the 1 x 6 MW installed capacity in PP II.

5.1.3 Calculation of surplus steam and resultant "waste" heat

The Svartsengi power plant uses a non-condensing turbine and the exhaust steam which still has a fairly large amount of thermal energy is used in the co-generation plant for the production of the hot water for district heating. In the process flow diagrams of the Svartsengi power plant shown in Figures 1 and 2, part of the exhaust steam from the turbines is used in the pre-heaters to further heat up the hot water to the desired temperature before going to the final heater and the excess is vented to the atmosphere. For PP I, the hot water is heated to 110°C then it goes to a deaerator where the hot water is flashed and deaerated at 100°C before it eventually goes to the final heater. In PP II the hot water is heated to 98°C before eventually going to the final heater.

Since the steam as it enters the turbine undergoes an expansion process, a fraction of the steam is condensed as it passes through the blades, hence, the exhaust steam is not 100% dry. The resultant condensate is discharged to the drains and only the steam is used in the hot water production process. The dryness fraction of the steam at the turbine exhaust is given by

$$x_4 = \frac{h_4 - h_{f4}}{h_{g4} - h_{f4}} \tag{16}$$

where h_4 is the enthalpy of the exhaust steam, h_{g4} and h_{f4} denote the saturation enthalpy of the steam and water at the turbine exhaust pressure and x_4 is the dryness fraction.

Using this equation, the steam dryness is found to be

$$x_4 = 0.95$$

To get the massflow of steam available for the preheaters, the total exhaust steam is multiplied by the dryness fraction and this has been found to be equal to

$$m_{s(EXHAUST)} = 27.54\ kg/s$$

From the energy and mass balance calculations, the massflow of low pressure steam that goes to the pre-heaters in PP I and PP II was found to be **8.05 kg/s** and **13.88 kg/s** respectively (refer to Appendix I). Since the total flow of the exhaust steam from the non-condensing turbines is more than the requirement in the pre-heaters, the flow of surplus steam that goes to the vent will be equal to

$$m_{s(VENT)} = m_{s(EXHAUST)} - m_{s(PRE\text{-}HEATER)} = 5.61\ kg/s$$

The steamflow that is vented to the atmosphere contains a large amount of thermal energy that is consequently wasted. This thermal energy can be estimated if we calculate the average enthalpy of the steam to be

$$h_{ave} = 2583\ kJ/kg$$

Then the maximum thermal energy recoverable from the steam assuming an average ambient temperature of 5°C is given by,

$$Q_R = m_s(h_{ave} - h_{5°C}) = 14{,}374\ kJ/s = 14.37\ MW_t \qquad (17)$$

5.1.4 Binary power calculation

The installation of a 3 x 1.34 MW Ormat binary power plant in Svartsengi was made to recover part of this otherwise excess steam from the power plant previously vented to the atmosphere through two tall chimneys. To estimate the potential binary power which can be generated from the surplus steam in the Svartsengi power plant (for this case was previously calculated as 5.61 kg/s), the *thermal efficiency*, η_{th}, of the binary cycle must be known. This can be calculated from the given design values provided by the manufacturer as shown in the process flow diagram in Figure 3. The *pressure-enthalpy* diagram of the cycle is also shown in Figure 10. To calculate for the thermal efficiency as per design, Equation 8 will be applied. The design values for the turbine work and heat input are given as follows (Ormat Turbines Ltd., Israel):

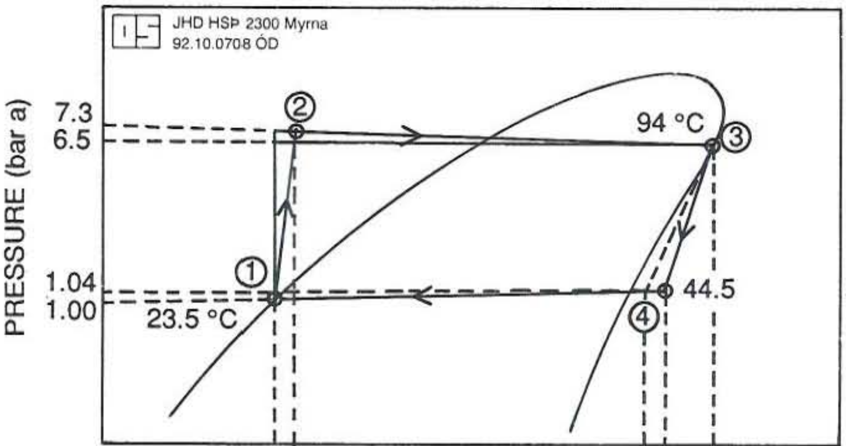

FIGURE 10: *Pressure-enthalpy* diagram of the isopentane cycle in Svartsengi Ormat binary plant

$$W_T = 1{,}340\ kW_e, \qquad Q_{input} = 11{,}850\ kW_t$$

then, Equation 8 as applied will become

$$\eta_{th} = \frac{1{,}340 - W_P}{11{,}850}$$

To complete the equation, the pump work shall be calculated using Equation 4. First, the mass of the binary fluid, which in this case is **isopentane**, is determined by an energy and mass balance across the evaporator using the design values of the entering and leaving streams as shown in Figure 3. The binary fluid flow is found to be equal to

$$m_B = 26.80\ kg/s$$

Since the efficiency of the pump is assumed as

$$\eta_P = 0.80$$

and the *specific volume*, v_f of isopentane at the pumping condition is

$$v_f = 1.63 \times 10^{-3}\ m^3/kg$$

solving for the pump work by Equation 4 gives

$$W_P = \frac{26.80 \times 1.63 \times 10^{-3} \times (7.3 - 1.0) \times 10^2}{0.80} = 34.40\ kW$$

The pump work as per design of the binary cycle is about **2.57%** of the turbine work. The enthalpy of isopentane at the pump discharge (corresponding to state 2 in the *pressure-enthalpy* diagram) can also be calculated using Equation 3 and this is found as

$$h_2 = 150.33\ kJ/kg$$

Substituting the calculated pump work into the main equation (Equation 8), the design cycle efficiency is thus

$$\eta_{th} = \frac{1340 - 34.40}{11{,}850} = 0.11$$

The binary power which can be generated from the 5.61.kg/s of waste steam in the power plant can now be calculated using the same equation (Equation 8) and substituting for the appropriate value of heat input. To simplify the equation, we can assume that the pump work is negligible as can be shown in the previous calculation at design conditions where pump work is only about **2.57%** of the work of the turbine.

Hence, the equation will be reduced to

$$\eta_{th} = \frac{W_T}{Q_{input}} \tag{18}$$

The missing parameter in the abovementioned equation is the heat input in the evaporator which is that coming from the surplus steam. If the steam is cooled down to 95°C as per design condition in the evaporator, the heat transferred to the isopentane in the evaporator considering the total steamflow going to the vent will be given by

$$Q_{input} = m_{s(VENT)}\,(h_{ave} - h_{95°C}) \tag{19}$$

Substituting the appropriate values in the abovementioned equation, the heat input is

$$Q_{input} = 5.61\,(2583 - 397.96)\ kJ/s = 12{,}258\ kW_t$$

This is about **85%** of the otherwise waste heat from the system.

Thus, the potential binary power which can be generated from the recovered thermal energy of the surplus steam is equal to

$$W_T = 0.11 \times 12{,}258 = 1{,}348\ kW_e$$

From these calculations, the *isentropic efficiency* of the Ormat turbine as designed can also be calculated using Equation 13 and with reference to Figure 10. For the Ormat turbines in Svartsengi, the *isentropic efficiency*, η_T, calculated from the data provided with the units is

$$\eta_T = \frac{1340}{26.8\,(593.23 - 526.98)} = 0.75$$

Since the net design output from each unit of the Ormat binary installation is given as

$$W_G = 1{,}200\ kW_e = 1.2\ MW_e$$

and the net output from the turbine after subtracting the pump work is **1305.6 kW**$_e$, the Ormat *generator efficiency* can be calculated by re-arranging Equation 15 as

$$\eta_G = \frac{W_G}{W_T}$$

Substituting in the equation, this is found to be equal to

$$\eta_G = \frac{1200}{1305.6} = 0.92$$

Thus, the net generator output will be

$$W_G = 0.92 \times 1{,}348\ kW_e = 1{,}240\ kW_e = 1.24\ MW_e$$

Therefore, considering the initial design conditions of the plant, whereby the operation is limited by the hot water production (125 MW$_t$), it is then possible to convert the waste heat into electric power and this will give a net binary power output of about **1.24 MW**$_e$.

5.2 Maximum production based on the available massflow from the wellhead

The calculation based on the initial design conditions in the previous sections is based on the assumption that the two-phase flow from the wellhead is exactly what is required during the operation of the plant. This means that no high pressure steam goes to the vent. However, the actual operation of the plant is such that the production wells are flowing at a steady rate controlled by a critical flow through an orifice and all of the two-phase fluid from the wells go to the power plant. For this case of plant operation, it is expected that the waste heat from the power plant as unused steam will be increased. Likewise, if this waste heat is converted to electric power, the binary power that can be generated will also increase.

The wells connected to the power plant with the corresponding massflows and enthalpy are shown in Table 1. From this table, it is shown that one of the wells (SG-10) which is connected to PP I is practically producing single phase steam. This is due to a steam cap that has formed in the reservoir during exploitation. Because of this, if it is considered to operate the plant in such a way that the available two-phase flow will be totally utilized, the total brine flow from the high pressure separators is not sufficient for the production of low pressure steam required to generate **50 MW$_t$** thermal power as per design of power plant I.

TABLE 1: Flowrate of Svartsengi geothermal wells, 1992

Well No.	Temp. (°C)	H kJ/kg	TP kg/s	Steam fraction	Water kg/s	Steam kg/s	Gas kg/s
SG-5	240	1038	0	0.174	0	0	0.000
SG-6	230	990	0	0.151	0	0	0.000
SG-7	240	1038	45	0.181	36.8	8.2	0.015
SG-8	240	1038	43	0.181	35.2	7.8	0.016
SG-9	240	1038	43	0.181	35.2	7.8	0.015
SG-10	235	2802	17	1.000	-	17.4	0.938
SG-11	240	1038	46	0.181	37.7	8.3	0.014
SG-12	230	990	33	0.151	28.0	5.0	0.008
Total			227		172.6	54.4	1.000

5.2.1 Energy and mass balance

The massflow of the brine from the high pressure separator can only produce **4.08 kg/s** of low pressure steam and this will limit the production of hot water in power plant I. The energy and mass balance across the direct contact condenser will show that the total hot water produced with this limitation is **54 kg/s**. This is equivalent to a thermal power of about **18.20 MW$_t$**. For this thermal power production in PP I, the high pressure steam flow required in the final heater is **2.79 kg/s**. Considering the total two-phase fluid from the wells in PP I, and the steam required in the final heater, the high pressure steam for electric power generation is

$$m_{s(PP\,I)} = 19.61\ kg/s$$

For PP II, the two-phase flow from the wells is sufficient to produce **75 MW$_t$** as per design of the plant. As previously calculated, the required steam in the final heater for the equivalent hot water production is **12.23 kg/s**. Considering also the available massflow from the wells and the requirement in the final heater, the high pressure steam which can be utilized for electric power generation in power plant II is

$$m_{s(PPII)} = 18.94 \ kg/s$$

5.2.2 Calculation of power from steam turbines

Power plant I

To estimate the electric power generation from the steam turbines based on the high pressure steam flow in PP I, the equations and steam rates used in Section 5.1.2 can be applied. The electric power which can be produced as a result of these calculations is **2.21 MW_e**. However, the maximum installed capacity in PP I is only 2.0 MW_e hence, **1.85 kg/s** of high pressure steam will be diverted to the vent.

Power plant II

The electric power which can be generated from the high pressure steam flow in PP II can be calculated similarly to that of PP I, but the corresponding steam rate for the turbine in PP II should be used. This will give an electric power output of **3.05 MW_e** when all of the steam is used in the turbine. Therefore, the total electric power generation from the steam turbines in PP I and PP II if all the two-phase fluid will be used in the co-generation plant is equal to **5.07 MW_e** and the total thermal power produced is **93.2 MW_t**.

5.2.3 Calculation of surplus steam and "waste" heat

With this kind of plant operation, assuming the same dryness of steam at the turbine exhaust as calculated in Section 5.1.3, the total massflow of low pressure steam coming from the turbine exhaust is

$$m_{s(EXHAUST)} = 36.72 \ kg/s$$

For the hot water production, the energy and mass balance across the pre-heaters also shows that for PP I **4.59 kg/s** of low pressure steam is needed and for PP II **13.88 kg/s** will be required. Subtracting these values from the total steamflow at the turbine exhaust, the corresponding flow of surplus steam in the vent is

$$m_{s(VENT)} = 18.25 \ kg/s$$

with an average enthalpy of **2600 kJ/kg**.

The recoverable heat assuming an average ambient temperature of 5°C is

$$Q_R = 47{,}060 \ kW_t$$

5.2.4 Binary power calculation

Applying the same calculation procedure as in Section 5.1.4, the heat input for the Ormat evaporator considering the total steamflow in the vent is

$$Q_{input} = 40{,}181 \ kW_t$$

The binary power which can be generated with this heat input in the evaporator will be

$$W_T = 0.11 \times 40{,}181 = 4{,}420\ kW_e = MW_e$$

and the net output from the generator is

$$W_G = 0.92 \times 4.42 = 4.07\ MW_e$$

This calculation also shows that if the plant will be operated such that the wells are discharging at present rates, and the balance of high pressure steam after subtracting the requirement in the final heater will all be used to generate power by the conventional steam turbines, the thermal energy wasted is increased by **30%** and the potential binary power which can be generated from this vented steam is likewise increased by the same magnitude.

5.3 Full capacity operation (8 MW_e + 125 MW_{th})

It has been mentioned in the preceeding sections that the total installed electric capacity of the Svartsengi power plant is 8MW_e. This is distributed as 2 x 1 MW in power plant I and 1 x 6 MW in power plant II. If it is desired to operate the plant at full capacity, i.e. 125 MW_t and 8MW_e, the total requirement for high pressure steam to the turbines using Equation 13 is **54.81 kg/s** and the high pressure steam requirement for hot water production as shown in Appendix I is **19.55 kg/s**. At the present well output shown in Table 1 the total high pressure steam which can be produced will not satisfy the full capacity requirement of the co-generation plant. For this type of plant operation, more production wells will be needed to be opened-up to the power plant to supply the additional high pressure steam requirement.

It is also indicated in the mass balance for PP I that the total brine which will be required in the low pressure flasher to produce the required low pressure steam for the direct contact condenser is **99.4 kg/s**. Hence, the additional two-phase flow requirement was found to be about **84.0 kg/s** for power plant I from which a surplus of high pressure steam of about **9.7 kg/s** will be produced. For PP II an additional two-phase flow requirement of **100 kg/s** is necessary to supply the additional high pressure steam for the power plant.

Moreover, for a full thermal power production, i.e. 125 MW_t, the total requirement of low pressure steam in the pre-heaters will still be **21.9 kg/s**. Assuming the same steam dryness of 0.95, the steam that will go to the vent will then be

$$m_{s(VENT)} = 40\ kg/s$$

with an average enthalpy of

$$h_{ave} = 2616\ kJ/kg$$

The equivalent thermal energy that is recoverable at an ambient temperature of 5°C is

$$Q_R = 103{,}801\ kW_t$$

However, the heat input to the evaporator at an outlet temperature for the steam of 95°C is

$$Q_{input} = 88{,}722\ kW_t$$

Translating this as turbine binary power output, this is equivalent to

$$W_T = 0.11 \times 88{,}722 = 9{,}759\ kW_e = 9.76MW_e$$

Since the Ormat generator efficiency as previously calculated is 0.92, the net power output will be

$$W_G = 0.92 \times 9{,}759 = 8{,}978\ kW_e = 9.0\ MW_e$$

5.4 Actual production of August 10, 1992

The calculations for the various modes of operating the power plant as discussed in the previous sections were based on the initial design values and maximum installed capacity of the Svartsengi geothermal power plant. However, it is always the case that the normal or actual operating parameters of any operating plant will deviate from the design values. Moreover, depending on the seasonal requirements, the operating load of the plant is at times below the total installed capacity. Such is the case in the Svartsengi geothermal power plant.

The hot water production varies with the season and the temperature requirement is also different from one area to another. In this case, the hot water temperature requirement is lower in Grindavik than in Keflavik. Further, during summertime the requirement of hot water for district heating is less than in wintertime. The demand for electric power, however, does not vary so much with the season.

Thus, it is during summertime when hot water demand hits the bottom and the waste heat from the power plant is maximum. This is so because less steam is used in the pre-heaters and most of the turbine exhaust steam is diverted to the chimneys in the power plant. To estimate the potential power that can be generated from the waste heat at the minimum thermal power production, a set of data was gathered during the actual operation of the plant on August 10,1992.

TABLE 2: Actual data logs (August 10, 1992)
of Svartsengi Ormat binary power plant

Unit no.	1	2	3
Vapor inlet temperature, °C	98	94	97
Vapor outlet temperature, °C	63	59	61
Cooling water outlet temperature, °C	20.8	21.2	21.9
Steam condensate outlet temp., °C	95	53	85
Vaporizer pressure, bar a	5.7	6.0	5.8
Condenser pressure, bar a	1.26	1.24	1.26
Feed pump pressure, bar a	6.4	6.4	6.4
Generator power output, kW	995	1022	993

Table 2 shows the actual data logs for the Ormat binary power plant. Table 3 shows the actual data logs for the steam turbines in PP I and PP II. The actual operating parameters of the thermal power production process is shown in Tables 4 and 5 for PP I and PP II respectively. The thermal power plant's operating data were taken by the author from the instrument indicators in the power plant's control room panel board, whereas the data in Table 3 for the steam turbines were taken from the operator's log book. For the Ormat binary plant, the data were taken from the data logging computer printout.

TABLE 3: Actual data logs (August 10, 1992)
of steam turbine operation in Svartsengi power plants I and II

Unit no.	1	2
POWER PLANT I		
Turbine inlet pressure, bar a	6.1	6.1
Turbine exhaust pressure, bar a	1.7	1.7
Steam inlet temperature, °C	160	160
Steam outlet temperature, °C	115	115
Generator output, kW	980	940
POWER PLANT II		
Turbine inlet pressure, bar a	5.4	
Turbine exhaust pressure, bar a	1.19	
Steam inlet temperature, °C	157	
Steam outlet temperature, °C	105	
Generator output, kW	4300	

TABLE 4: Actual operating parameters (August 10, 1992) for Svartsengi power plant I
hot water production, low-pressure separators for units 2 and 4 are not in use

Stream location	Unit 1	Unit 2	Unit 3	Unit 4
Temperature of steam from H.P. separator, °C	152	157	164	163
Temperature of steam from L.P. separator, °C	82.0	-	70.9	-
Temperature of hot water from D.C. condenser, °C	50.5	23.0	60.7	22.7
Temperature of hot water from pre-heater, °C	104.9	107.3	107.1	108.1
Temperature of hot water from final heater, °C	51.0	111.3	106.7	101.6
Supply temperature of hot water to Grindavik	83.7	-	97.8	-
Hotwater flowrate, kg/s	27.0	15.3	14.5	15.6
Turbine exhaust pressure, bar g.	0.68	-	0.96	-

TABLE 5: Actual operating parameters (August 10, 1992)
for Svartsengi power plant II hot water production

Stream location	Unit 6	Unit 7	Unit 8
Temperature of steam from H.P. separator, °C	-	159	157
Temperature of steam from L.P. separator, °C	74.6	88.9	77.1
Temperature of hot water from D.C. condenser, °C	80.2	80.3	76.9
Temperature of hot water from pre-heater, °C	89.9	99.3	95.7
Temperature of hot water from final heater, °C	85.4	102.8	102.3
Hotwater flowrate, kg/s	35.0	35.0	35.3

A process flow diagram of power plant I and II for the day's operation is shown in Figures 11 and 12 respectively. Each stream in the figure is numbered to correspond to the numbers in the columns of the table for the energy and mass balance.

The steam and water flows from the two-phase production of the day were calculated based on the recorded separation pressure and this is shown in Table 6.

TABLE 6. Svartsengi borehole production of August 10, 1992

Power plant I	Sep. pressure: 6.10 bar a				
	Enthalpy (kJ/kg)	Two-phase (kg/s)	x	Water (kg/s)	Steam (kg/s)
SG - 5	1038	0	0.175	0	0
SG - 6	990	0	0.152	0	0
SG - 10	2802	17	1.000	0	17
SG - 12	990	33	0.152	27.98	5.02
Power plant II	**Sep. pressure: 6.0 bar a**				
	Enthalpy (kJ/kg)	Two-phase (kg/s)	x	Water (kg/s)	Steam (kg/s)
SG - 7	1038	45	0.176	37.07	7.93
SG - 8	1038	43	0.176	35.43	7.57
SG - 9	1038	43	0.176	35.43	7.57
SG - 11	1038	46	0.176	37.90	8.10

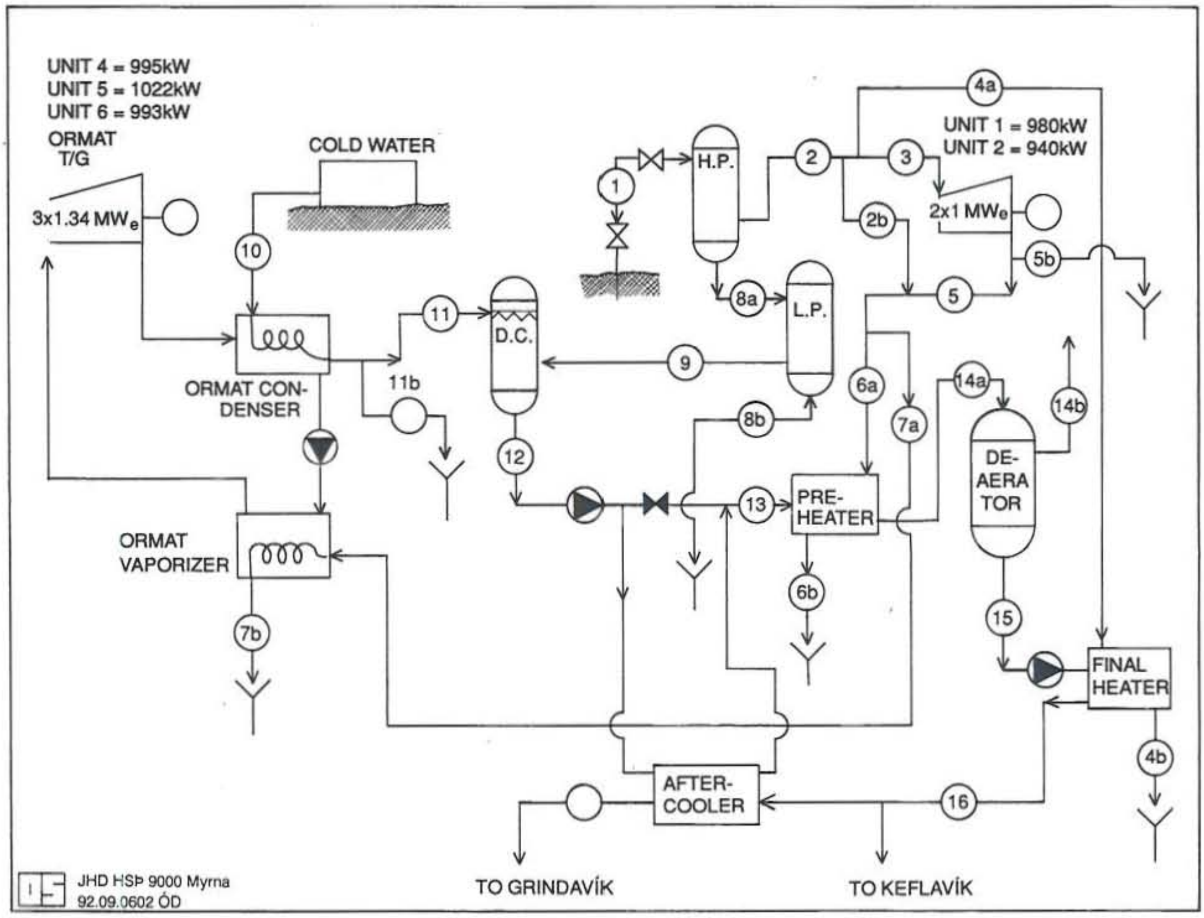

FIGURE 11: Svartsengi PP I process flow diagram during actual operation (Aug. 10, 1992)

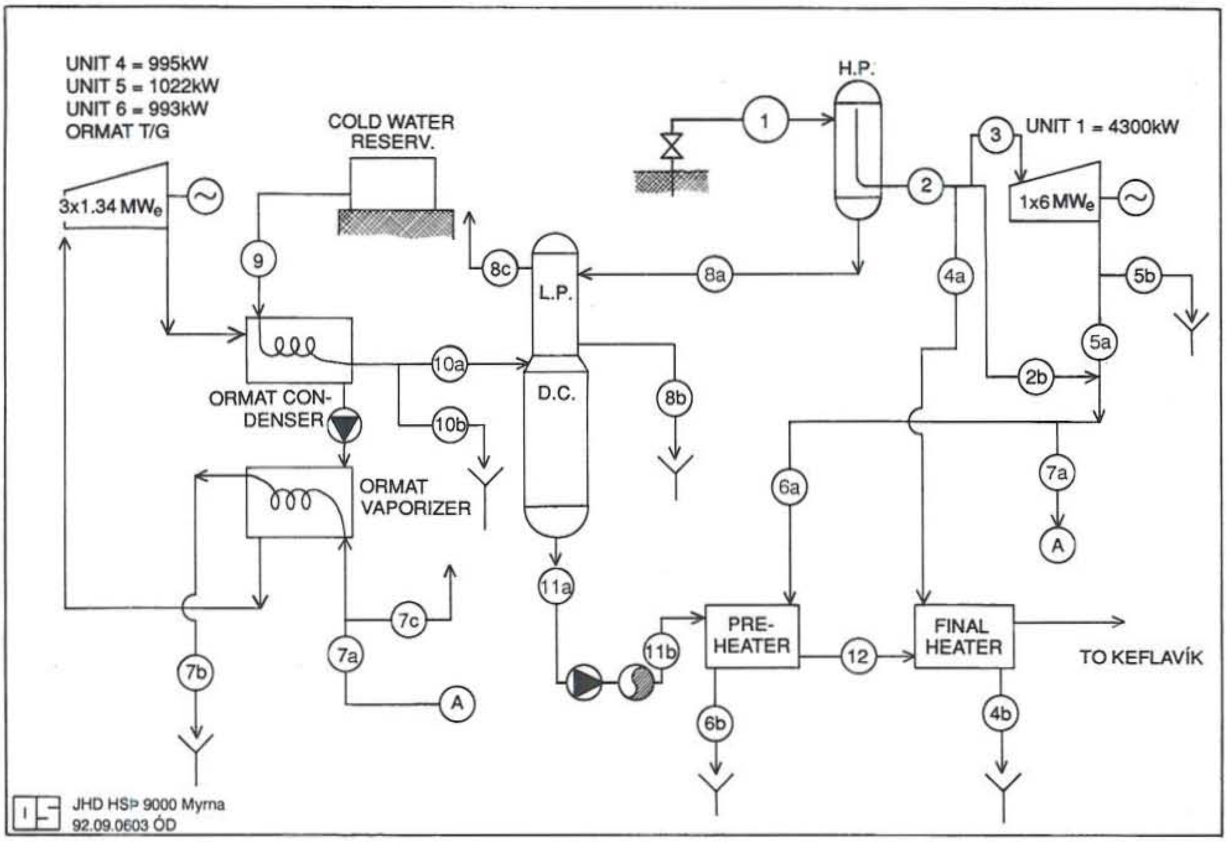

FIGURE 12: Svartsengi PP II process flow diagram during actual operation (Aug. 10, 1992)

The hot water from Units 1 and 3 are delivered to Grindavik

5.4.1 Energy and mass balance

The results of the energy and mass balance calculation for the day's operation is summarized in Tables 7 and 8 and the details in each stream for each unit of the power plant are shown in Appendix II. The thermal energy, ***Q (kJ/s)***, shown in this table for the steam and water at each stream is calculated as maximum i.e. with reference temperature of 0°C.

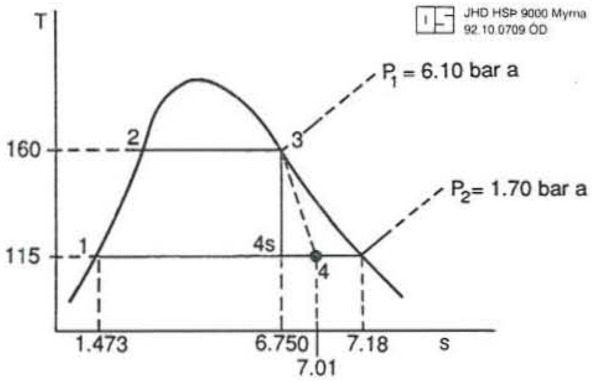

FIGURE 13: *Ts* diagram of actual steam cycle for the 1 MW_e turbine

The total flow of high pressure steam used in the turbines was calculated from the recorded net power output and using the equations of the Rankine cycle. The *Ts* diagram for the actual steam cycle in power plant I and II is shown in Figures 13 and 14, respectively. By assuming the same values for the turbine and generator efficiencies as found for the design conditions, the total flow of high pressure steam used in the turbines is **46.64 kg/s**.

TABLE 7: Summary of energy and mass balance (August 10, 1992) for Svartsengi power plant I

Stream	1	2a	3	4a	5	6a	7a	8a	9	10	11	12	13	14a	15
m, kg/s	50.00	22.02	17.00	0.55	16.15	10.68	9.94	27.98	2.44	413.05	72.40	74.84	74.84	74.84	74.00
t, °C	232	160	160	160	115	124	124	160		5	21.3				100
Q^{**}, kJ/s	80,304	60,733	31,875	1,517	42,614	28,507	28,801	18,902	6,405	8,665	6,472	12,768	9,553	33,324	31,046
		2b		4b	5b	6b	7b	8b			11b			14b	
m, kg/s		4.47		0.55	0.85	10.68	9.94	25.54			223.05			0.84	
t, °C		160			115		95				21.3			100	
Q^{**}, kJ/s		12,329		257	411	4,071	3,956	8,500			19,940			2,248	
	16		To Grindav.	From Grindav.		To Keflav.	From Keflav.								
m, kg/s	74.00		43.45	43.45		30.54	30.54								
t, °C			88.65	40.00		106.41	40.00								
Q^{**}, kJ/s	26,400		16,012	7,281		13,572	5,118								

Q: reference temperature is 0°C

TABLE 8: Summary of energy and mass balance (August 10, 1992), Svartsengi power plant II

Stream	1	2a	3	4a	5a	6a	7a	8a	9	10a	11a	12	To Keflav.	From Keflav.
m, kg/s	177.0	31.17	29.64	0.58	28.16	2.58	26.53	145.83	413.05	105	116	116	116	116
t, °C	240	159	157	159	105	107	107	159	5	21.3	79.14		97	40
Q^{**}, kJ/s	183,726	85,936	61,515	1,599	73,696	6,752	70,371	97,881	8,665	9,387	38,362	45,677	46,654	19,438
		2b		4b	5b	6b	7b	8b		10b	11b			
m, kg/s		0.95		0.58	1.482	2.58	3.70	124.65		223.05	116			
t, °C		159			105		95			21.3	79.14			
Q^{**}, kJ/s		2,619		365	652	1,251	7,501	41,892		19,940	38,362			
							7c	8c						
m, kg/s							23.00	10.20						
t, °C							107							
Q^{**}, kJ/s							60,636	10,192						

Q: reference temperature is 0°C

To determine the massflow of high pressure steam used in the hot water production, the actual data gathered on the hot water conditions at certain points in the process, i.e. temperature and hotwater flowrate, was used as input data in the spreadsheet program for the energy and mass balance across the heat exchangers. The total massflow of high pressure steam used in the process is determined to be equal to **1.13 kg/s**.

Considering the total production from the wells during the day's operation, Table 6 shows that a total massflow of **53.2 kg/s** of

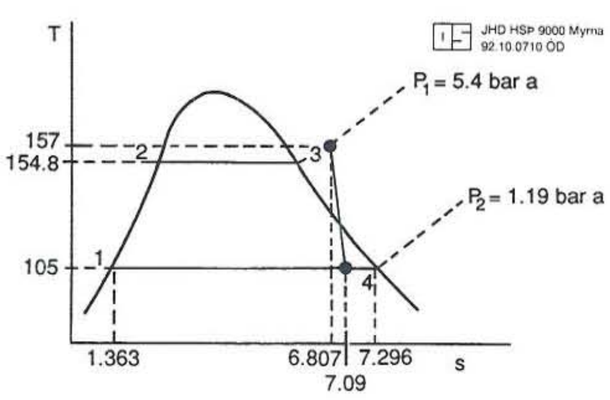

FIGURE 14: *Ts* diagram of actual steam cycle for the 6 MW_e turbine

high pressure steam was delivered to the power plant. This indicates that a surplus of high pressure steam production was diverted to the vents. By mass balance, the flowrate of high pressure steam that went to the vent during the day's operation was found to be **5.42 kg/s**.

5.4.2 Calculation of surplus steam and "waste" heat

In the previous sections, the calculations of binary power was always referred to as potential power generation. This is so because these calculations were based on possible future plant operation. However, during the actual plant operation of August 10, binary power was generated from the turbine exhaust steam of this day. The net power output from the Ormat units is shown in Table 2. For this day, a total net power output of **3.01 MW$_e$** electric power from the binary plant was generated which is about the total installed capacity. The surplus steam for this day's operation will then be just the balance of the unused steamflow after deducting the steam used in the Ormat evaporators and the steamflow to the pre-heaters.

To calculate the steamflow in the Ormat evaporator, the flowrate of isopentane was first determined from the actual conditions of the fluid at the turbine inlet and outlet (shown in the *pressure-enthalpy* diagram for each unit in Figure 15) and from the turbine output assuming the same generator efficiency used in the design calculations. The results of the calculations from the actual data is summarized in Table 9. It is shown here that the total steam used in the Ormat evaporators is **13.64 kg/s**.

For the pre-heaters, the energy and mass balance for the actual operation of the power plant shows that the combined massflow of steam that went to the pre-heaters in PP I and PP II was **13.26 kg/s**. Hence, the surplus steam from the power plant will be given by the following equation:

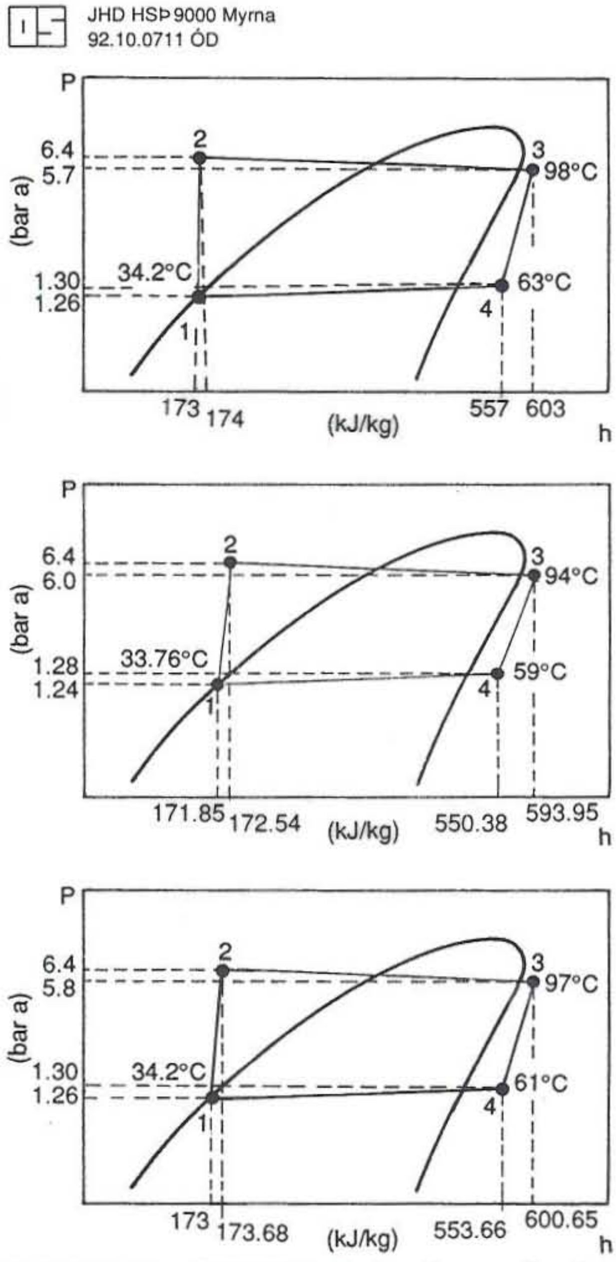

FIGURE 15: *Pressure-enthalpy* diagram for the Ormat units during actual operation (Aug. 10, 1992)

$$m_{s(VENT)} = [m_{s(H.P.)} + m_{s(EXHAUST)}] - [m_{s(ORMAT)} + m_{s(PRE\text{-}HEATERS)}]$$

TABLE 9. Summary of calculations from actual data (August 10, 1992) for Svartsengi Ormat binary power plant

Unit no.	1	2	3
Net output, kW	995	1022	993
Turbine work, kW	1081.52	1110.87	1079.35
Pump work	16.38	17.52	15.70
Turbine Efficiency, η_T	0.810	0.767	0.840
Thermal Efficiency, η_{th}	0.105	0.103	0.110
Isopentane flow, kg/s	24.00	25.50	22.94
Steam flow, kg/s	4.60	4.46	4.30

Considering the steam dryness at the turbine exhaust which was calculated to be **0.97** and **0.96** for the turbines in power plants I and II respectively, the vented steam is equal to

$$m_{s(VENT)} = 23 \ kg/s$$

with an average enthalpy of

$$h_{ave} = 2633 \ kJ/kg$$

Based on an outlet temperature of 95°C in the Ormat evaporator, the thermal power from the surplus steam that will be available for binary power generation is

$$Q_{input} = 51{,}406 \ kW_t = 51.41 \ MW_t$$

The preceeding calculations illustrate that even with the installed Ormat binary power plant operating at full capacity, there is still a considerable massflow of surplus steam that goes to the vent during summertime operation. Furthermore, the potential binary power which can be generated from the waste heat of the power plant justifies the current additional Ormat units being installed in the power plant.

5.4.3 Calculation for additional binary power

The steam vented to the atmosphere during the actual operation of August 10, 1992 is still a potential source of power. It can be deduced from the preceeding section, that with an ambient temperature of 5°C only 32% of the waste heat from the power plant has been recovered by the existing binary power plant. If the remaining surplus steam is also used to generate binary power, the calculated heat input in Section 5.4.2 for the additional binary units will result in a turbine work output of

$$W_T = \eta_{th} \times 51{,}406 \ kW_e$$

Using the design value for the thermal efficiency as

$$\eta_{th} = 0.11$$

then the turbine output is

$$W_T = 5{,}655\ kW_e$$

Using also the design value for the generator efficiency as

$$\eta_G = 0.92$$

then the net power output will be

$$W_G = 5{,}203\ kW_e = 5.20\ MW_e$$

The over-all recovery then of the otherwise waste heat from the power plant expressed as percent of the maximum if the ambient tempearture is 5°C will be equal to

$$\%R = \frac{81{,}809}{95{,}260} \qquad giving \qquad R = 86\%$$

The total potential recovery of waste heat as calculated from the actual data agrees with the calculated value in the previous sections based on the operation of the power plant as designed. Further, it also shows that the potential maximum net power generation if the newly installed binary units will be operational is about **8.0 MW$_e$**

The calculations from the actual performance of the Ormat turbines is very close to the design conditions with regard to the turbine efficiency and thermal efficiency. However, the *pressure-enthalpy* diagram for each unit of the actual cycle shows the actual conditions of the isopentane to be superheated as it enters the turbine. This is probably done due to a higher operating pressure in the condenser compared to the design which can lower the cycle efficiency and the turbine efficiency. The high pressure in the condenser may be due to air leakage during evacuation of the condenser prior to operation.

5.4.4 Hot water production

The mass of the cooling water requirement for the Ormat condensers was also calculated using the sensible heat equation for the cold water and by applying the steady flow energy equation to each piece of equipment in the binary cycle.

If the pump work is neglected, the heat removed from the isopentane in the condenser is

$$q_{cond} = q_{evap} - w_T \tag{20}$$

and the sensible heat in the cooling water is expressed as

$$Q_w = m_w C_p (t_2 - t_1) \tag{21}$$

From these, the total water used in the Ormat cycle was found to be **413 kg/s**. However, the actual data gathered shows that only a total of **190 kg/s** of hot water was delivered. Hence, a balance of **223 kg/s** of cooling water at about 22°C went into the drain during this actual production. This is equivalent to **15,258 kW** of thermal energy that went to the drain.

5.5 Thermal efficiency of the Svartsengi power plant

The various calculations in the previous sections were basically focused on the recovery of the thermal energy that is wasted during the operation of the power plant. Calculations for thermal efficiencies were also made, however, these pertain to the heat-work cycle of the binary system. The thermal efficiency in a broader sense can also be applied to the operation cycle of the whole power plant.

For this case, the thermal efficiency can be simply defined as

$$\eta_{th(PP)} = \frac{energy\ used}{energy\ supplied}$$

This means that, for a geothermal power plant, the over-all thermal efficiency will be expressed as

$$\eta_{th(PP)} = \frac{\Sigma(MW_e + MW_{th})}{m_{TP}(h_{ave} - h_{fC})} \tag{23}$$

where the numerator is the sum of the thermal and electric power produced and the denominator is the maximum heat that can be recovered from the two-phase fluid from the boreholes at an average ambient temperature.

A summary of the power output calculation as discussed in the preceeding sections and the calculated plant thermal efficiency at various mode of plant operation is shown in Table 10.

TABLE 10: General summary of calculations for the Svartsengi power plant

Operating conditions	Design limited by hot water product.	Consider. available massflow from well	Installed capacity operat. of plant	Actual operation August 10, 1992	Addit. Ormat units installed
Massflow of hot water, kg/s	373.44	279.00	373.44	34 31 116	
Hotwater temperature, °C	120.0	120.0	120.0	89 106 97	
Thermal power, MW_t*	125.0	93.2	125.0	54.42	
Electric power, MW_e	4.035	5.050	8.000	6.530	
Ormat power, MW_e	1.348	4.420	9.759	3.340	5.660
Massflow of surplus steam, kg/s	6.71	18.25	40.00	13.64	23.00
Tot. heat flow, boreholes, kJ/s**	313,264	259,902	439,436	259,902	259,902
Thermal eff.of plant, $\eta_{th(PP)}$ (%)					
a. without binary power	41.20	37.80	30.27	23.45	
b. with binary power	41.62	39.50	32.50	24.74	26.91

* Assuming the hot water is returned or rejected by the users at 40°C

** Assuming an outside temperature of 5°C and the average enthalpy of additional wells where required to be 1,000 kJ/kg

Design Conditions:
The thermal efficiency of the power plant if operated as originally designed, i.e. the maximum thermal power output is **125 MW_t** and the electric power generation which for this case is **4.035 MW_e** dependent on the hot water production, is calculated to be about **41.2%**. For this plant efficiency, no binary power is generated and an ideal mass balance whereby no high pressure steam is diverted to the vent is considered. However, if the waste heat from the plant which is basically the exhaust steam from the turbine, is converted to electric power by the binary cycle,

then it is expected that the power output will be increased by **1.348 MW_e** and the thermal efficiency of the plant is thus increased by **1%**.

Maximum production considering the available massflow from wells:
When the power plant is be operated in such a way that it is desired to produce maximum power from the current available two-phase fluid from the wells, the thermal efficiency of the power plant is only **37.8%**. This corresponds to a thermal power production of only **93.2 MW_t** and an electric generation of **5.05 MW_e**. This is because, at present capacity of the field, the total requirement of flashed geothermal brine in power plant I to operate at maximum designed thermal capacity is more than what is presently available. This is due to the fact that presently, one of the wells in the Svartsengi field has a very high enthalpy due to a formation of a steam cap in the reservoir. This mode of operation results in an excess of high pressure steam in power plant I that goes to the vent. If, however, the plant will use all the exhaust steam to generate electric binary power, this will result in an additional production of **4.42 MW_e** electric power which translates to an increase in thermal efficiency of the plant by about **4.5%**.

Full capacity operation of the power plant:
The thermal efficiency of the power plant when it is operated at the full installed capacity, i.e. **125 MW_t** thermal power and **8.0 MW_e** electric power, is calculated to be about **30.3%**. At this condition, more production wells will be required to be hooked-up to the plant. This will also result in an increased massflow of high pressure steam to the turbines which will increase the low pressure exhaust steamflow to the vents. Moreover, to satisfy the requirement of high pressure brine for thermal power production, more high pressure steam will be produced in power plant I resulting in an additional high pressure steam that will go to the vent. If all of the waste steam is used for binary power generation, an additional production of **9.76 MW_e** electric power will increase the plant's thermal efficiency by about **7%**.

Actual operation (August 10, 1992):
In the actual operation of the Svartsengi power plant whereby **54.42 MW_t** of thermal power and **6.53 MW_e** of electric power from the steam turbines were generated, the calculations show that the thermal efficiency of the power plant is only **23.5%** if the binary power production is not counted. However, for this day, a total of **3.34 MW_e** of binary turbine power was produced using a fraction of the waste steam from the plant. It is also shown in the calculation that a total of **5.42 kg/s** of high pressure steam was diverted to the vent during this day's operation. Hence, taking into account the binary power production for this day, the thermal effciency of the power plant has been increased by about **5.5%** thus making it **24.7%** with still a total of **60 MW_t** thermal power wasted in the vents.

Additional binary power production from actual operation:
If the waste heat from the actual operation of the plant is converted to electric power, then this will generate a potential binary power of **5.66 MW_e** which will further increase the thermal efficiency of the power plant to **26.9%**. The thermal efficiency of the plant in this case is about **8.8%** higher than the actual operation.

6. THE BACMAN I GEOTHERMAL POWER PLANT

6.1 Plant description

The Bacman I geothermal power plant is the most recently constructed geothermal power plant in the Philippines. It is the third geothermal power plant built by the Philippine National Oil Company's Energy Development Corporation (PNOC-EDC) since 1983. The plant has an installed capacity of **110 MW$_e$** and will utilize two-phase fluid from about twelve production wells. The power facility which is going to supply a baseload market is expected to be operational by the first half of 1993. Its capacity will add to the existing 660 MW$_e$ geothermal power installation in Luzon and raise the nationwide total to 1041 MW$_e$ (Tolentino and Alcaraz, 1986) by early 1993.

The two-phase geothermal fluid is separated in two central separator stations located apart from each other in Pad E and Pad H as shown in Figure 4. The separators at Pad E station separates the two-phase fluid at a pressure of 12.5 bara whereas, in Pad H station, steam is separated at a pressure of 10.0 bara. The separated steam is delivered to the power plant at a design rate of 238 kg/s at a single pressure, 7.0 bara, at a saturation temperature of 165°C. A schematic pressure and flow diagram of the Bacman I steam supply lines is shown in Figure 16. At the power generation facility, steam enters the two 55 MW$_e$ condensing turbines and is exhausted to the condenser.

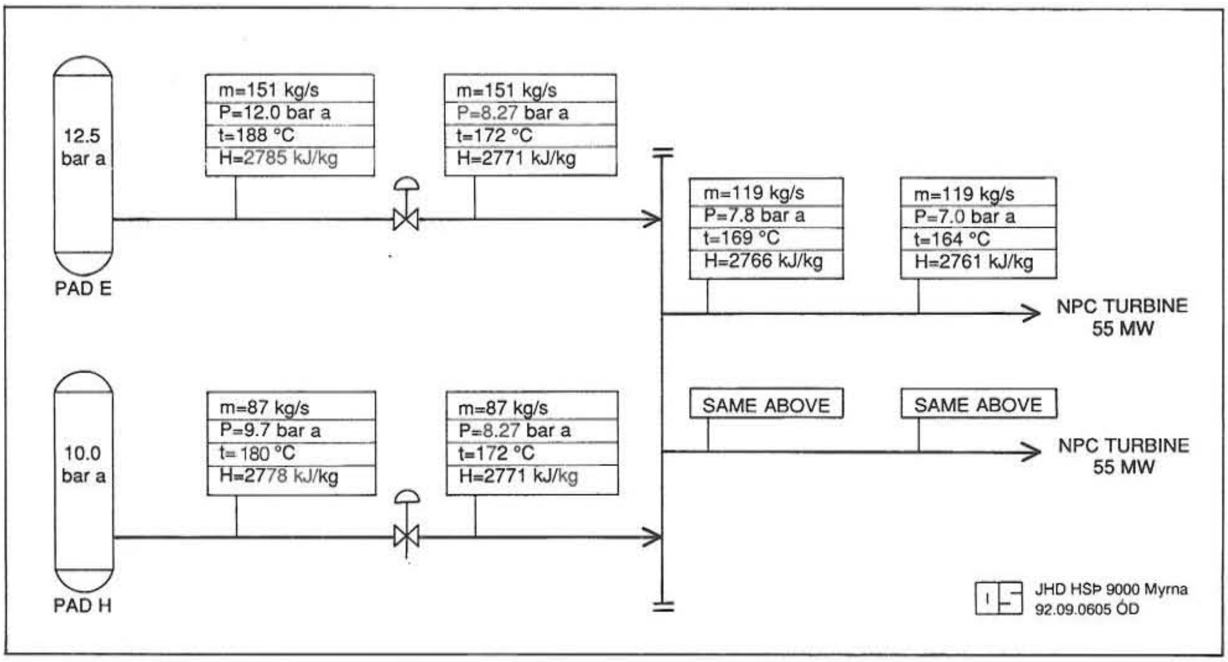

FIGURE 16: Bacman I pressure and flow schematic diagram of steam lines

The waste water from the separator station is piped to two reinjection pads where injection by gravity is used. There are four wells currently hooked-up to the system intended for reinjection purposes. In each reinjection pad, a storage pond is provided to collect the waste water during startup or shutdown of the plant or in some emergency cases during the operation of the plant. The total flow of waste water from the two separator stations is estimated to be 531 kg/s. The temperature of the reinjection brine is 190°C from Pad E station and 180°C from Pad H station. A schematic diagram of the Bacman I reinjection line is shown in Figure 17 and the proposed tapping point for the binary power generation facility is also shown in the figure.

6.2 Evaluation of binary power potential for Bacman I

The approach taken in this study is merely to estimate the binary power which can be generated when the waste water from the separators is used. The water from the separators has a temperature which is high enough to generate power by binary cycle. Based on the available data of the field and the design of the binary power plant at Svartsengi geothermal plant, initial evaluation is done on the potential binary power which can be derived from the waste water from the Bacman I power plant.

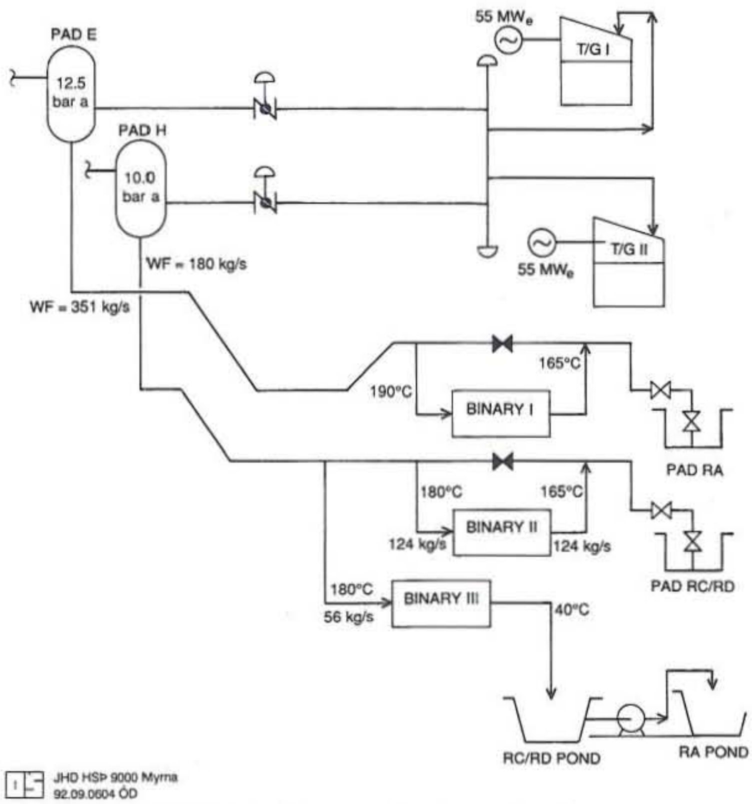

FIGURE 17: Bacman I schematic diagram of waste water line

6.2.1 Assumptions

In making the preliminary cycle calculations for Bacman I, the design parameters of the Svartsengi binary power plant are used. However, due to the difference in climatic conditions at the locations of the Svartsengi and Bacman I power plants, the temperature parameters of the cycle has to be scaled up to be applicable to the Philippine conditions. Relative to this the following assumptions are used:

a. The inlet temperature of the cooling water to the condenser is 27°C (based on the average water temperature in the Philippines) and the outlet temperature from the condenser is 45°C based on the temperature gradients in the condenser at Svartsengi power plant.
b. The temperature difference in the Svartsengi design is also used in scaling up the isopentane temperatures at the condenser outlet and turbine exhaust, hence, for Bacman I this is equal to 46.5°C and 67.5°C, respectively, as will be discussed in the succeeding section.
c. The binary fluid used for this case is isopentane and it enters the turbine at saturated condition and leaves at superheated condition.
d. The pump, turbine and generator efficiencies used in the calculations are the same as that calculated from the manufacturer's data provided with the Ormat units in Svartsengi which are found to be equal to $\eta_P = 0.80$, $\eta_T = 0.75$ and $\eta_G = 0.92$.
e. The outlet temperature of the brine from the evaporator shall be based on acceptable silica saturation index (SSI).
f. The feedpump discharge pressure is 0.80 bar higher than the pressure at the evaporator based on the Svartsengi design.

6.2.2 Choice of isopentane cycle parameters

As mentioned in the preceeding section, the isopentane cycle temperatures for Bacman I are determined based on the temperature of the cooling water in the condenser. Assuming a cooling water temperature of 27°C and using the same temperature difference in Svartsengi , it is found that the turbine exhaust temperature is **67.5°C** and the condenser outlet temperature of isopentane will be **46.5°C**. Since the isopentane is superheated at the turbine exit then at this condition, if a 5.5°C temperature difference is maintained between the cooling water outlet temperature and the saturation temperature of the isopentane, it is found that the saturation temperature of isopentane is **50.5°C** and the corresponding condenser pressure is determined to be **2.10 bara**. Moreover, for the conditions at Svartsengi, there is a 0.04 bar pressure loss from the turbine exhaust to the condenser. Hence, the turbine exhaust pressure for Bacman I is **2.14 bara**.

To get the temperature and pressure of the isopentane at the turbine inlet, it is assumed that the isopentane condition at the turbine exhaust temperature and pressure values, which were previously determined based on Svartsengi, is the isentropic state. Therefore, it can be said that

$$t_{4s} = 67.5°C, \qquad P_{4s} = 2.14\ bara, \qquad h_{4s} = 561.26\ kJ/kg, \qquad X_{4s} = 1.087.$$

For an isentropic expansion, the entropy, s, at the turbine inlet can be determined from the abovementioned values and this is equal to

$$s_{4s} = s_3 = 2.453\ kJ/kg\text{-}K$$

Since it is assumed that the isopentane is saturated at the turbine inlet, then the properties of the binary fluid can be found from the calculated entropy and these are

$$h_3 = 615.82\ kJ/kg, \qquad t_3 = 111.69°C, \qquad P_3 = 9.4\ bara.$$

Finally, the actual condition of the isopentane at the turbine exhaust can now be calculated using the isentropic efficiency found for Svartsengi, i.e. η_T=0.75.

Hence, the properties of isopentane at the actual condition are

$$h_4 = 574.90\ kJ/kg,$$

$$t_4 = 75.27°C$$

From the values calculated above, the *pressure-enthalpy* diagram of the isopentane cycle can be illustrated as shown in Figure 18.

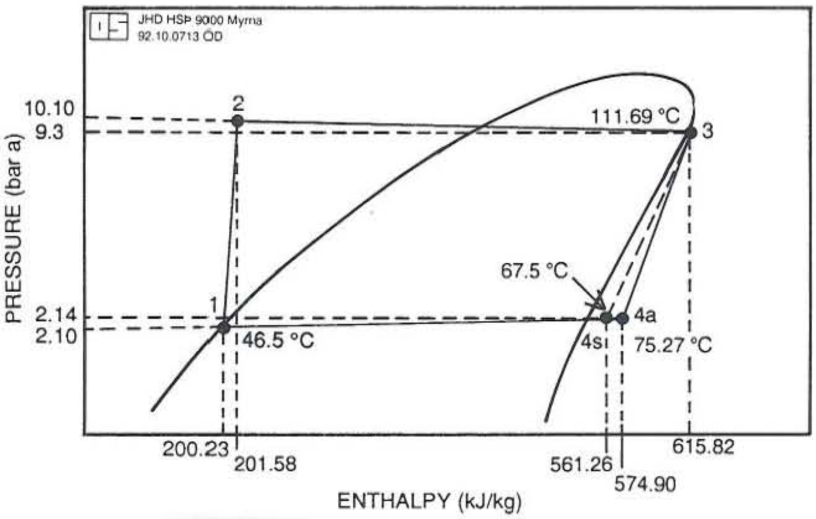

FIGURE 18: *Pressure-enthalpy* diagram of the isopentane cycle for the Bacman I binary plant

6.2.3 Choice of outlet temperature of the geothermal brine

Using the assumptions stated in the preceeding section, the first step of the power output calculations consisted of deciding upon an acceptable brine outlet temperature in the evaporator. The acceptability of the outlet brine temperature in this study will be related to the calculated SSI, meaning whether it is oversaturated with respect to amorphous silica using the equation:

$$SSI = \frac{\textit{silica concentration of the brine}}{\textit{equilibrium amorphous silica solubility}}$$

Silica deposition has been encountered in virtually all high enthalpy, liquid-dominated, geothermal fields (Thomas and Gudmundsson, 1989). This is because, high temperature geothermal water is invariably saturated with respect to quartz. With decrease in temperature and phase separation, the residual water becomes supersaturated with respect to amorphous silica (Solis et al., 1991). Hence, in deciding the brine outlet temperature, the equilibrium amorphous silica solubility, c_e, was calculated from the equation of Fournier and Rowe (1977) at various temperatures of the brine based on the separator outlet condition:

$$\log c_e = \frac{-731}{T} + 4.52 \tag{24}$$

TABLE 11: Silica saturation at different brine temperatures, Bacman I geothermal power plant

Location	Pad E					
Temperature, °C	190*	175**	165**	160**	155**	150**
Silica conc. in brine, ppm ***	890	890	890	890	890	890
Silica solubility, ppm	873	773	710	679	649	619
Silica saturation index	1.019	1.151	1.253	1.311	1.371	1.438
Location	Pad H					
Temperature, °C	180*	175**	165**	160**	155**	150**
Silica conc. in brine, ppm ***	813	813	813	813	813	813
Silica solubility, ppm	806	773	710	679	649	619
Silica saturation index	1.009	1.051	1.145	1.198	1.253	1.312

* Separated brine temperature ** Proposed brine outlet temperature
*** Average from well

The results of these calculations are summarized in Table 11. The estimated degree of silica supersaturation of the brine from the heat exchanger is in the range of 1.05-1.44 at outlet temperatures of 175-150°C. From actual field experience in Tongonan I and Palinpinon I, operation of these plants with an SSI of 1.2-1.3 has an average scaling rate of 5-10 mm/year. The corresponding schedule for shutdown to clean the reinjection line is every 2 years (Jabonillo, 1992). Hence, for the proposed binary power plant in Bacman I, an outlet brine temperature of 165°C from the heat exchanger will be considered acceptable. This is also the same temperature at which waste water from Tongonan and Palinpinon power stations is reinjected.

Due to the limited capacity of the hot reinjection wells in Bacman I, a portion of the separated brine will be for cold reinjection. Experiments in reinjecting the effluent at relatively low temperatures have successfully been undertaken at the Bacman geothermal field (Solis et al., 1991). The results from these experiments may allow some portion of the total waste brine to be cooled down to 40°C in the heat exchanger for binary power generation. However, the binary power calculations for this cycle will not be dealt with in this report.

6.2.4 Binary power calculations

In calculating the binary power which can be generated from the waste water in Bacman I power plant the total massflow of the brine that will be considered is that which can only be cooled down to 165°C and this is equal to **475 kg/s**. Also, the assumptions stated earlier in Section 5.2 is applied with reference to the *pressure - enthalpy* diagram of the cycle as shown in Figure 18. The same equations for the Rankine cycle as discussed in the previous sections are applied to calculate the electrical turbine output, heat flows in the cycle, and thermal efficiency of the cycle. Figure 19 shows the process flow diagram for the Bacman I binary cycle. Since the waste water in Bacman I has a high temperature, it has a large potential for power generation. Hence, a calculation for brine outlet temperatures other than 165°C was made separately for Pad E and Pad H as well and the results of this are shown in Tables 12 and 13.

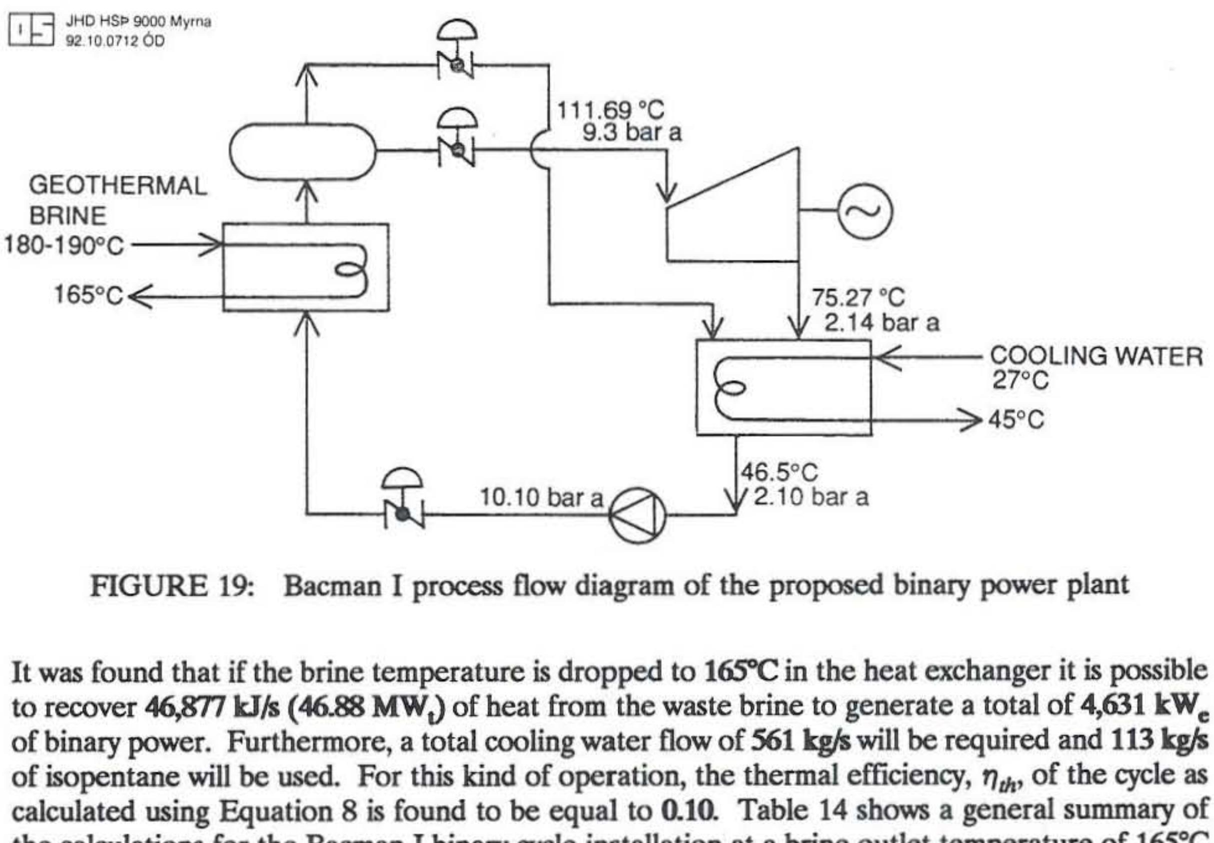

FIGURE 19: Bacman I process flow diagram of the proposed binary power plant

It was found that if the brine temperature is dropped to **165°C** in the heat exchanger it is possible to recover **46,877 kJ/s (46.88 MW$_t$)** of heat from the waste brine to generate a total of **4,631 kW$_e$** of binary power. Furthermore, a total cooling water flow of **561 kg/s** will be required and **113 kg/s** of isopentane will be used. For this kind of operation, the thermal efficiency, η_{th}, of the cycle as calculated using Equation 8 is found to be equal to **0.10**. Table 14 shows a general summary of the calculations for the Bacman I binary cycle installation at a brine outlet temperature of 165°C and 40°C.

6.2.5 Thermal efficiency of the power plant

Using the equations applied in Section 5.5 for Svartsengi, the thermal efficiency of the power plant, $\eta_{th(PP)}$, is calculated to be **10.8%** without the production of binary power. This is based on a total two-phase flow of **502 kg/s** in Pad E and **267 kg/s** in Pad H with average enthalpy of **1457.90 kJ/kg** and **1424.60 kJ/kg**, respectively. With binary power generation this can be increased by about **4.3%** which will make the thermal efficiency of the power plant equal to **11.3%**.

TABLE 12: Potential binary power at the different brine outlet temperatures of 175, 165, 160 and 155°C for Bacman I Pad E station

Turbine output, kW	2304.02	3823.73	4579.26	5332.01
Turbine efficiency, η_T	0.75	0.75	0.75	0.75
Pump work, kW	95.66	158.76	190.13	221.39
Pump efficiency, η_P	0.80	0.80	0.80	0.80
Generator output, kW	2119.70	3517.83	4212.92	4905.44
Generator efficiency, η_G	0.92	0.92	0.92	0.92
Thermal efficiency, η_{th}	0.10	0.10	0.10	0.10
ENTHALPY, kJ/kg				
Brine: inlet	807.62	807.62	807.62	807.62
Brine: outlet	741.17	697.34	675.55	653.84
Isopentane: Turbine inlet	615.82	615.82	615.82	615.82
Isopentane: Turbine exhaust	574.90	574.90	574.90	574.90
Isopentane: Vaporizer inlet	201.58	201.58	201.58	201.58
FLOW, kg/s				
Brine	351.00	351.00	351.00	351.00
Cooling water	279.37	463.64	555.25	646.53
Isopentane	56.31	93.44	111.91	130.30
TEMPERATURE, °C				
Brine: inlet	190	190	190	190
Brine: outlet	175	165	160	155
Cooling water: inlet	27	27	27	27
Cooling water: outlet	45	45	45	45
Isopentane to turbine	111.69	111.69	111.69	111.69
Isopentane from turbine	75.27	75.27	75.27	75.27
Isopentane from condenser	46.5	46.5	46.5	46.5
PRESSURE, bar a				
Brine saturation: inlet	12.50	12.50	12.50	12.50
Brine saturation: outlet	8.92	7.01	6.18	5.43
Vaporizer	9.30	9.30	9.30	9.30
Turbine exhaust	2.14	2.14	2.14	2.14
Condenser	2.10	2.10	2.10	2.10
Feed pump discharge	10.10	10.10	10.10	10.10
SPECIFIC VOLUME, m^3/kg				
Isopentane	0.0017	0.0017	0.0017	0.0017
HEAT FLOW, kW				
Heating stream	23323.95	38708.28	46356.57	53976.78
Cooling water	20924.27	34725.79	41587.18	48423.39

6.3 Potential problems

The utilization of wastewater from the separators for binary power generation is not without problems especially involving the extraction of heat from the brine which subsequently lowers the temperature, increasing the silica supersaturation index. For the operation of a binary power plant in Bacman I, the following major problem areas have been identified:

1. Scaling in heat exchangers, pipelines and reinjection wells;
2. Cooling water supply;
3. Possible detrimental effects to production zone due to injection of relatively cold fluid.

TABLE 13: Potential binary power at different brine outlet temperatures for Bacman I Pad H station

Turbine output, kW	806.97	1073.88	1339.81
Turbine efficiency, η_T	0.75	0.75	0.75
Pump work, kW	33.51	44.59	49.79
Pump efficiency, η_P	0.80	0.80	0.80
Generator output, kW	742.41	987.97	1232.63
Generator efficiency, η_G	0.92	0.92	0.92
Thermal efficiency, η_{th}	0.10	0.10	0.10
ENTHALPY, kJ/kg			
Brine: inlet	763.22	763.22	763.22
Brine: outlet	697.34	675.55	653.84
Isopentane: Turbine inlet	615.82	615.82	615.82
Isopentane: Turbine exhaust	574.90	574.90	574.90
Isopentane: Vaporizer inlet	201.58	201.58	201.58
FLOW, kg/s			
Brine	124.00	124.00	124.00
Cooling water	97.85	130.21	162.46
Isopentane	19.72	26.24	32.74
TEMPERATURE, °C			
Brine: inlet	180	180	180
Brine: outlet	165	160	155
Cooling water: inlet	27	27	27
Cooling water: outlet	45	45	45
Isopentane to turbine	111.69	111.69	111.69
Isopentane from turbine	75.27	75.27	75.27
Isopentane from condenser	46.50	46.50	46.50
PRESSURE, bar a			
Brine saturation: inlet	10.02	10.02	10.02
Brine saturation: outlet	7.01	6.18	5.43
Vaporizer	9.30	9.30	9.30
Turbine exhaust	2.14	2.14	2.14
Condenser	2.10	2.10	2.10
Feed pump discharge	10.10	10.10	10.10
SPECIFIC VOLUME, m^3/kg			
Isopentane	0.0017	0.0017	0.0017
HEAT FLOW, kW			
Heating stream	8169.12	10871.08	13563.12
Cooling water	7328.64	9752.61	12173.52

6.4 Potential solutions

Scaling

Siliceous scale is typically inert to most chemicals (Thomas and Gudmundsson, 1989) and once deposited, it is very resistant to mechanical removal. Hence, most treatment methods focus on prevention of silica deposition or on controlling the morphology of the silica deposited. The present efforts being done at PNOC-EDC in this respect include restricting the injected water to

TABLE 14: General summary of calculations for Bacman I power plant

Turbine output (BINARY), kW	806.97	3,823.73
Turbine output (STEAM), kW	40,255	69,759
THERMAL EFFICIENCY, $\eta_{th(PP)}$(%)		
Without binary	10.83	
With binary	11.61	
FLOW, kg/s		
Brine	124	351
Cooling water	97.85	463.64
Isopentane	19.72	93.44
Two-phase from boreholes	184	502
TEMPERATURE, °C		
Brine: inlet	180	190
Brine: outlet	165	165
HEAT FLOW, kW		
Heating stream (Evaporator)	8,169.12	38,708.28
Cooling water (Condenser)	7,328.64	34,725.79
From wells	346,817	668,784

temperatures at which silica supersaturation is minimized. Present field tests to address this problem also include acidification of the brine phase to inhibit silica deposition and field testing (in cooperation with Ciba-Geigy) of other chemical inhibitors. This process of inhibiting silica deposition is important before the brine enters the heat exchanger.

Another approach which can be used to prevent massive scaling in the wellbore when reinjecting a relatively cold fluid has been successfully tested in Bacman (Solis et al., 1991) where flashed brine was first collected in sump ponds prior to reinjection. The process involves maximizing the amorphous silica supersaturation of the separated water, cooling the fluid as rapidly as possible and maximizing the fluid residence time in the ponds to precipitate and remove the silica prior to reinjection.

Cooling water supply

The source of cooling water for the condensers may pose some problems to the installation of the binary plant in Bacman I due to the high flow requirement. This will require the installation of a cooling tower for the plant so that the cooling water can be recirculated and only the make-up water will be required.

An alternative though, is to use aircooled condensers, however, additional power consumption will be required for the blowers.

Reinjection of cold fluid

With respect to this problem, PNOC-EDC has maintained a very cautious attitude and approach. The reinjection policy to reinject as far away as possible from the production boreholes and as deep into the outflow as may be allowed by the hydrology of the geothermal system is necessary to attain better management of the reinjected brine. This will require careful assessment of identified reinjection zones.

7. DISCUSSION OF RESULTS

The results of the analyses for Svartsengi have been summarized in Table 10 and for Bacman I these are shown in Table 14. By comparing the figures in these separate tables it is shown that generally the thermal efficiency of the Svartsengi power plant is higher than that of Bacman I. This agrees with the theory that the thermal efficiency of a co-generation plant, which in this case is the Svartsengi power plant, is higher than the thermal efficiency of an electrical generating plant which in this case is the Bacman I geothermal power plant.

Moreover, the results of the calculations from actual data show that the additional **3.34 MW_e** electric power generated from the binary power plant in Svartsengi generally accounts for the marginal increase of **6%** in the plant's thermal efficiency during the plant's operation in summer when a low demand for hot water is expected and thus a much higher steamflow is vented to the atmosphere. It is expected that the marginal increase in the plant's thermal efficiency will be **14.8%** when all the surplus steam from the power plant during the summertime operation will be used, thus increasing the binary power output to **9.0 MW_e**. This can be realized when the Ormat units being installed will become operational. This calculation will also justify the current upgrading being done at Svartsengi of the Ormat binary power plant capacity.

The production of **10 MW_e** binary power when the plant operates at its maximum installed capacity will likewise increase the thermal efficiency by **7%**. However, with the current massflows from the wells, it is not possible to operate the plant at full capacity, hence from the calculations it is shown that the binary power which can be generated from the waste heat of the plant is **4.42 MW_e** which can result in a marginal increase in the thermal efficiency of **4.5%**.

On the other hand, the **4.3%** marginal increase in the thermal efficiency for the Bacman I geothermal power plant, as a result of the installation of a binary power plant is a relatively lower figure compared to that of Svartsengi due to the limitations in the outlet temperature of the brine. However, depending on the success of the silica inhibition trials, this can be further increased to about **7.5%** by greater allowable temperature drop in the brine.

The thermal efficiency of a power plant though, does not tell the whole story for this case. It is the utilization of an otherwise waste heat into a more valuable megawatt power. The main point that is stressed in the result of this evaluation is that for Svartsengi the potential power which can be generated from its waste heat by binary cycle is about **10 MW_e** during full capacity operation, **4.4 MW_e** considering the current two-phase flow available and about **9.0 MW_e** when the plant is operating during warmer seasons as exemplified by the calculations based on actual data. For the case of Bacman I, the total potential power which can be generated from its waste heat by binary cycle is at least about **4.6 MW_e** if the brine is cooled down to 165°C. These improvements in the generating capacity of a power plant are several times more valuable than the thermal power generation.

8. CONCLUSIONS

It can be deduced from the results of this study that the installation of a binary power plant in an operating geothermal power plant can improve the use of the mined heat from a geothermal resource and thus can lead to a more efficient resource management. For instance, at a steam rate of 2.2 kg/s-MW for a steam turbine and an average well capacity of 5 MW_e, the maximum binary power which can be produced from the waste heat in Svartsengi will be equivalent to about 2 production wells and in Bacman I this will be equivalent to 1 production well. This installation, however, is more applicable in a power plant such as Svartsengi where the surplus steam used is a relatively clean source of heat compared to the geothermal brine of Bacman I, hence, a maximum heat recovery is possible.

For the case of the Bacman I geothermal power plant, PNOC-EDC is forging ahead with the study and evaluation of developing a binary power plant as an added module to the existing power plant. While the calculations present an attractive side of waste heat recovery, it is realized that the proposed system is more complex than the Svartsengi case which utilizes a cleaner geothermal fluid in the heat exchangers. The potential problems in the Bacman I case as discussed should be clearly looked into, and in addition to this, an economic study is needed to give an answer of go or no go. Furthermore, a design of a binary power cycle which is appropriate for the Bacman conditions must be made to arrive at a closer estimate of the power potentials.

ACKNOWLEDGEMENTS

Through this page I would like to express my indebtedness to the following people and entities who have contributed much in one way or another to the completion of this report:

- to the United Nations University under the directorship of Dr. Ingvar Birgir Fridleifsson and the Government of Iceland for affording me the opportunity to join the geothermal training course;

- to the National Energy Authority of Iceland for providing the facilities and supplies during the course;

- to my office, the Philippine National Oil Co.-Energy Development Corporation for believing in me;

- to the lecturers in the course for the invaluable knowledge imparted to us;

- to my advisor, Sverrir Thorhallson for seeing me through;

- to Valdimar K. Jonsson and Arni Ragnarsson for providing the much needed lectures and informations;

- to Ludvik S. Georgsson for being always on-hand to give assistance till the very end of the course;

- to Runolfur, Claus and Egill of VGK Engineering Consultants for sharing their expertise;

- to Bragi and Thor of Hitaveita Sudurnesja for accomodating me and patiently answering my querries;

- to the ladies in the drafting section for patiently drawing the figures;

- and to the very dear folks at home who provided from a distance the much needed motivation and inspiration that helped a great deal to pull myself through.

NOMENCLATURE

c_e	Equilibrium amorphous silica solubility	ppm
C_P	Specific heat capacity at constant pressure	kJ/kg-°C
h	Specific enthalpy	kJ/kg
m	Massflow	kg/s
P	Absolute pressure	bara
q	Specific heat flow	kJ/s-kg
Q	Total heat flow	kJ/s
Q_{input}	Total heat duty in the evaporator	kJ/s
R	Heat recovery	%
s	Entropy	kJ/kg-K
t	Temperature	°C
T	Absolute temperature	K
v_f	Specific volume at liquid state	m^3/kg
w_a	Specific actual work	kW/kg
w_s	Specific isentropic work	kW/kg
W	Total work	kW
x	Steam dryness fraction	

Subscripts

1	Pump suction
2*a*	Actual pump discharge condition
2*s*	Isentropic pump dischanrge condition
3	Turbine inlet
4*a*	Actual turbine outlet condition
4*s*	Isentropic turbine outlet condition
ave	Average value
B	Binary fluid
cond	Condenser
evap	Evaporator
*f*4	Liquid state at turbine outlet
*g*4	Vapor state at turbine outlet
G	Generator
H.P.	High pressure
P	Pump
PP	Power plant
R	Recoverable
s	Steam
t	Reference temperature
T	Turbine
th	Thermal
TP	Two-phase
w	Water

Greek letters and scientific extension

η	Efficienccy
Σ	Summation
log	Logarithm

REFERENCES

Bjornsson, G., and Albertsson, A., 1985: The power plant at Svartsengi, development and experience. Geothermal Resources Council, 1985 international volume, 427-442.

Fournier, R.O., and Rowe, J.J., 1977: The solubility of amorphous silica in water at high temperatures and high pressures. Am. Mineral, 62, 1052-1056.

Jabonillo, R.G., 1992: Palinpinon I proposed binary power plant. Memorandum to Taguines, M.L., (unpublished).

Solis, R.P., See, F.S., and Ruaya, J.R., 1991: Low temperature wastewater injection experiments at Bacman geothermal field. International Workshop on Geothermal Energy Training, Proceedings, Philippines.

Thomas, D.M., and Gudmundsson, J.S., 1989: Advances in the study of solids deposition in geothermal systems. Geothermics, 18 - 1/2, 5-13.

Tolentino, B.S., and Alcaraz, A.P., 1986: Strategies relating to the exploration and development of a geothermal field: A case for Bacon-Manito geothermal project, Albay/Sorsogon provinces, Philippines. In Tolentino, B.S.: Lectures on geothermal energy in the Philippines. UNU G.T.P., Iceland, report 12, 91-102.

Wark, K., 1988: Thermodynamics. McGraw-Hill, U.S.A., 787 pp

APPENDIX I:

Energy and mass balance calculation for Svartsengi power plants I and II, using the modified program

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
SUMMATION

STREAM LOCATION	I COLD WATER RESERVOIR	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	136.8						
initial temperature, °C	5.00						
waterflow to D.C. condenser, kg/s		136.8					
inlet temp. of cold fluid, °C		5.00					
outlet temp. of hot water, °C		64.80					
L.P. steamflow to D.C. condenser**, kg/s		14.53					
massflow from D.C. condenser, kg/s		151.33					
massflow to preheater, kg/s			151.33				
inlet temp. to pre-heater, °C			81.87				
outlet temp. from pre-heater, °C			110.00				
massflow of M.P. steam, kg/s			8.05				
inlet temp. to deaerator, °C				110.00			
outlet temp. from deaerator, °C				100.00			
massflow of hot water, kg/s				151.33			
steam fraction				0.02			
steam flashed, kg/s				2.55			
massflow from deaerator, kg/s				148.78			
massflow to final heater, kg/s					148.50		
inlet temp. to final heater, °C					100.00		
outlet temp. from final heater, °C					125.00		
massflow of H.P. steam, kg/s					7.32		
massflow required, kg/s						74.26	74.24
initial temperature, °C						124.98	100.00
supply temperature, °C						90.74	125.00
return temperature, °C						40.00	40.00
heat capacity at inlet, kJ/s	2874.17	2874.17	51318.56	69171.32	62230.18	38669.12	31111.01
heat capacity at inlet, MW	2.87	2.87	51.32	69.17	62.23	38.67	31.11
heat capacity at outlet, kJ/s		40533.05	69812.97	62349.32	77960.73	27991.36	38975.26
heat capacity at outlet, MW		40.53	69.81	62.35	77.96	27.99	38.98
d MW		37.66	18.49	-6.82	15.73	-10.68	7.86

**
brineflow required = 99.36 kg/s
high pressure steam (H.P.) = 20.64 kg/s
H.P. steam for turbine = 13.32 kg/s

SVARTSENGI POWER PLANT II
HEAT AND MASS BALANCE
SUMMATION

STREAM LOCATION	I COLD WATER RESERVOIR	II D.C. CONDSR./ DEAERATOR	III PRE- HEATER	IV FINAL HEATER	VI SUPPLY TO KEFLAVIK
cold water flow, kg/s	203.40				
initial temperature, °C	5.00				
massflow to D.C. condenser, kg/s		203.40			
L.P. steam to D.C. condenser**, kg/s		21.90			
inlet temp. of cold fluid, °C		5.00			
outlet temp. of hot water, °C		65.00			
massflow from D.C. condenser, kg/s		225.30			
massflow to preheater, kg/s			225.30		
inlet temp. to pre-heater, °C			65.00		
outlet temp. from pre-heater, °C			98.00		
massflow of M.P. steam, kg/s			13.88		
inlet temp. to final heater, °C				98.00	
outlet temp. from final heater, °C				125.00	
massflow of H.P. steam, kg/s				12.23	
massflow from final heater, kg/s				225.30	
massflow required, kg/s					225.3
supply temperature, °C					120
return temperature, °C					40
heat capacity at inlet, kJ/s	4273.43	4273.43	61288.27	92514.81	113488.12
heat capacity at inlet, MW	4.27	4.27	61.29	92.51	113.49
heat capacity at outlet, kJ/s		61288.27	92514.81	118280.08	37726.49
heat capacity at outlet, MW		61.29	92.51	118.28	37.73
d MW		57.01	31.23	25.77	-75.76

**
brineflow required = 134.1 kg/s
H.P. steamflow = 27.9 kg/s
H.P. steam for turbine = 15.67 kg/s

APPENDIX II:

Energy and mass balance calculation for August 10, 1992, in each stream for each unit in Svartsengi power plants I and II, using the modified program

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
SUMMATION

STREAM LOCATION	I ORMAT CONDENSER	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	72.4						
initial temperature, 'C	5.00						
final temperature, 'C	21.30						
steamflow to vaporizer(MP), kg/s	14.26						
binary fluid flow, kg/s	71.05						
ORMAT T/G output, MWe	3010.00						
waterflow to D.C. condenser, kg/s		72.4					
inlet temp. of cold fluid, 'C		21.30					
outlet temp. of hot water, 'C		42.55					
steamflow to D.C. condenser, kg/s		2.44					
massflow from D.C. condenser, kg/s		74.84					
massflow to preheater, kg/s			74.84				
inlet temp. to pre-heater, 'C			30.71				
outlet temp. from pre-heater, 'C			106.85				
massflow of M.P. steam, kg/s			10.68				
inlet temp. to deaerator, 'C				106.52			
outlet temp. from deaerator, 'C				100.00			
massflow of hot water, kg/s				74.84			
steam fraction				0.01			
steam flashed, kg/s				0.84			
massflow from deaerator, kg/s				74.00			
massflow to final heater, kg/s					73.99		
inlet temp. to final heater, 'C					100.00		
outlet temp. from final heater, 'C					85.40		
massflow of H.P. steam, kg/s					0.55		
massflow required, kg/s						43.45	30.54
initial temperature, 'C						70.61	100.00
supply temperature, 'C						88.65	106.41
return temperature, 'C						40.00	40.00
heat capacity at inlet, kJ/s	1521.12	6374.10	9404.31	33258.60	31005.77	12729.21	12798.09
heat capacity at inlet, MW	1.52	6.37	9.40	33.26	31.01	12.73	12.80
heat capacity at outlet, kJ/s	6374.10	13157.50	33258.60	31010.05	26334.11	16012.21	13571.98
heat capacity at outlet, MW	6.37	13.16	33.26	31.01	26.33	16.01	13.57
d MW	4.85	6.78	23.85	-2.25	-4.67	3.28	0.77

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
R-1 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	72.40						
initial temperature, 'C	5.00						
final temperature, 'C	21.30						
steamflow to vaporizer(MP), kg/s	14.26						
initial temperature, 'C	115.00						
final temperature, 'C	95.00						
binary fluid flow, kg/s	17.76						
generator output, kW	3010.00						
condenser outlet temperature, 'C	V						
vaporizer inlet temperature, 'C	V						
vaporizer outlet temperature, 'C	V						
T/G inlet fluid enthalpy, kJ/kg	1385.53						
T/G inlet pressure, bar a.	7.8						
T/G exhaust fluid enthalpy, kJ/kg	1385.53						
binary sat. vapor enthalpy, kJ/kg	1385.53						
binary sat. liquid enthalpy, kJ/kg	1106.10						
waterflow to D.C. condenser, kg/s		27					
inlet temp. of cold fluid, 'C		21.30					
outlet temp. of hot water, 'C		50.50					
steamflow to D.C. condenser, kg/s		1.35					
inlet temp. of L.P. steam, 'C		82.00					
massflow from D.C. condenser, kg/s		28.35					
massflow to preheater, kg/s			28.35				
inlet temp. to pre-heater, 'C			18.04				
outlet temp. from pre-heater, 'C			104.90				
massflow of M.P. steam, kg/s			4.71				
inlet temp. of M.P. steam, 'C			136.00				
outlet temp. of M.P. steam, 'C			128.00				
inlet temp. to deaerator, 'C				104.90			
outlet temp. from deaerator, 'C				100.00			
massflow of hot water, kg/s				28.35			
steam fraction				0.01			
steam flashed, kg/s				0.21			
massflow from deaerator, kg/s				28.14			
massflow to final heater, kg/s					28.14		
inlet temp. to final heater, 'C					100.00		
outlet temp. from final heater, 'C					51.00		
massflow of H.P. steam, kg/s					-2.74		
inlet temp. of H.P. steam, 'C					160.00		
outlet temp. of H.P. steam, 'C					150.00		
massflow required, kg/s						28.14	0.00
initial temperature, 'C						51.00	0.00
supply temperature, 'C						83.70	0.00
return temperature, 'C						40.00	40.00
heat capacity at inlet, kJ/s	1521.12	2377.08	2140.60	12360.22	11792.43	6006.25	0.00
heat capacity at inlet, MW	1.52	2.38	2.14	12.36	11.79	6.01	0.00
heat capacity at outlet, kJ/s	6374.10	5933.02	12360.22	11792.43	6006.25	9779.00	0.00
heat capacity at outlet, MW	6.37	5.93	12.36	11.79	6.01	9.78	0.00
d MW	4.85	3.56	10.22	-0.57	-5.79	3.77	0.00

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
R-2 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	72.40						
initial temperature, °C	5.00						
final temperature, °C	21.30						
steamflow to vaporizer(MP), kg/s	14.26						
initial temperature, °C	115.00						
final temperature, °C	95.00						
binary fluid flow, kg/s	17.76						
generator output, kW	3010.00						
condenser outlet temperature, °C	V						
vaporizer inlet temperature, °C	V						
vaporizer outlet temperature, °C	V						
T/G inlet fluid enthalpy, kJ/kg	1385.53						
T/G inlet pressure, bar a.	7.8						
T/G fluid exhaust enthalpy, kJ/kg	1385.53						
binary sat. vapor enthalpy, kJ/kg	1385.53						
binary sat. liquid enthalpy, kJ/kg	1106.10						
waterflow to D.C. condenser, kg/s		15.3					
inlet temp. of cold fluid, °C		21.30					
outlet temp. of hot water, °C		23.00					
steamflow to D.C. condenser, kg/s		0.04					
inlet temp. of L.P. steam, °C		28.80					
massflow from D.C. condenser, kg/s		15.34					
massflow to preheater, kg/s			15.34				
inlet temp. to pre-heater, °C			23.00				
outlet temp. from pre-heater, °C			107.30				
massflow of M.P. steam, kg/s			2.42				
inlet temp. of M.P. steam, °C			136.00				
outlet temp. of M.P. steam, °C			117.00				
inlet temp. to deaerator, °C				107.30			
outlet temp. from deaerator, °C				100.00			
massflow of hot water, kg/s				15.34			
steam fraction				0.01			
steam flashed, kg/s				0.20			
massflow from deaerator, kg/s				15.14			
massflow to final heater, kg/s					15.14		
inlet temp. to final heater, °C					100.00		
outlet temp. from final heater, °C					111.30		
massflow of H.P. steam, kg/s					0.32		
inlet temp. of H.P. steam, °C					160.00		
outlet temp. of H.P. steam, °C					120.00		
massflow required, kg/s						0.00	15.14
intial temperature, °C						0.00	100.00
supply temperature, °C						0.00	111.30
return temperature, °C						40.00	40.00
heat capacity at inlet, kJ/s	1521.12	1347.01	1479.34	6883.90	6345.93	0.00	6344.57
heat capacity at inlet, MW	1.52	1.35	1.48	6.88	6.35	0.00	6.34
heat capacity at outlet, kJ/s	6374.10	1479.34	6883.90	6345.93	7049.94	0.54	7048.43
heat capacity at outlet, MW	6.37	1.48	6.88	6.35	7.05	0.00	7.05
d MW	4.85	0.13	5.40	-0.54	0.70	0.00	0.70

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
R-3 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	72.40						
initial temperature, °C	5.00						
final temperature, °C	21.30						
steamflow to vaporizer(MP), kg/s	14.26						
initial temperature, °C	115.00						
final temperature, °C	95.00						
binary fluid flow, kg/s	17.76						
generator output, kW	3010.00						
condenser outlet temperature, °C	V						
vaporizer inlet temperature, °C	V						
vaporizer outlet temperature, °C	V						
T/G inlet fluid enthalpy, kJ/kg	1385.53						
T/G inlet pressure, bar a.	7.8						
T/G fluid exhaust enthalpy, kJ/kg	1385.53						
binary sat. vapor enthalpy, kJ/kg	1385.53						
binary sat. liquid enthalpy, kJ/kg	1106.10						
waterflow to D.C. condenser, kg/s		14.5					
inlet temp. of cold fluid, °C		21.30					
outlet temp. of hot water, °C		60.70					
steamflow to D.C. condenser, kg/s		1.01					
inlet temp. of L.P. steam, °C		70.90					
massflow from D.C. condenser, kg/s		15.51					
massflow to preheater, kg/s			15.51				
inlet temp. to pre-heater, °C			69.57				
outlet temp. from pre-heater, °C			107.10				
massflow of M.P. steam, kg/s			1.08				
inlet temp. of M.P. steam, °C			136.00				
outlet temp. of M.P. steam, °C			112.00				
inlet temp. to deaerator, °C				107.10			
outlet temp. from deaerator, °C				100.00			
massflow of hot water, kg/s				15.51			
steam fraction				0.01			
steam flashed, kg/s				0.20			
massflow from deaerator, kg/s				15.30			
massflow to final heater, kg/s					15.30		
inlet temp. to final heater, °C					100.00		
outlet temp. from final heater, °C					106.70		
massflow of H.P. steam, kg/s					0.18		
inlet temp. of H.P. steam, °C					160.00		
outlet temp. of H.P. steam, °C					99.00		
massflow required, kg/s						15.30	0.00
initial temperature, °C						106.70	100.00
supply temperature, °C						97.80	106.70
return temperature, °C						40.00	40.00
heat capacity at inlet, kJ/s	1521.12	1276.58	4477.69	6956.25	6412.62	6800.38	0.00
heat capacity at inlet, MW	1.52	1.28	4.48	6.96	6.41	6.80	0.00
heat capacity at outlet, kJ/s	6374.10	3893.29	6956.25	6865.11	6800.38	6219.20	0.00
heat capacity at outlet, MW	6.37	3.89	6.96	6.87	6.80	6.22	0.00
d MW	4.85	2.62	2.48	-0.09	0.39	-0.58	0.00

SVARTSENGI POWER PLANT I
HEAT AND MASS BALANCE
R-4 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II DIRECT CONTACT CONDENSER	III PRE-HEATER	IV DEAERATOR	V FINAL HEATER	VI SUPPLY TO GRINDAVIK	VII SUPPLY TO KEFLAVIK
cold water flow, kg/s	72.40						
initial temperature, 'C	5.00						
final temperature, 'C	21.30						
steamflow to vaporizer(MP), kg/s	14.26						
initial temperature, 'C	115.00						
final temperature, 'C	95.00						
binary fluid flow, kg/s	17.76						
generator output, kW	3010.00						
condenser outlet temperature, 'C	V						
vaporizer inlet temperature, 'C	V						
vaporizer outlet temperature, 'C	V						
T/G inlet fluid enthalpy, kJ/kg	1385.53						
T/G inlet pressure, bar a.	7.8						
T/G fluid exhaust enthalpy, kJ/kg	1385.53						
binary sat. vapor enthalpy, kJ/kg	1385.53						
binary sat. liquid enthalpy, kJ/kg	1106.10						
waterflow to D.C. condenser, kg/s		15.6					
inlet temp. of cold fluid, 'C		21.30					
outlet temp. of hot water, 'C		22.70					
steamflow to D.C. condenser, kg/s		0.04					
inlet temp. of L.P. steam, 'C		28.00					
massflow from D.C. condenser, kg/s		15.64					
massflow to preheater, kg/s			15.64				
inlet temp. to pre-heater, 'C			22.72				
outlet temp. from pre-heater, 'C			108.10				
massflow of M.P. steam, kg/s			2.48				
inlet temp. of M.P. steam, 'C			136.00				
outlet temp. of M.P. steam, 'C			113.00				
inlet temp. to deaerator, 'C				108.10			
outlet temp. from deaerator, 'C				100.00			
massflow of hot water, kg/s				15.64			
steam fraction				0.01			
steam flashed, kg/s				0.23			
massflow from deaerator, kg/s				15.40			
massflow to final heater, kg/s					15.40		
inlet temp. to final heater, 'C					100.00		
outlet temp. from final heater, 'C					101.60		
massflow of H.P. steam, kg/s					0.04		
inlet temp. of H.P. steam, 'C					160.00		
outlet temp. of H.P. steam, 'C					101.00		
massflow required, kg/s						0.00	15.40
initial temperature, 'C						0.00	100.00
supply temperature, 'C						0.00	101.60
return temperature, 'C						40.00	40.00
heat capacity at inlet, kJ/s	1521.12	1373.42	1442.21	7081.27	6454.78	0.00	6453.52
heat capacity at inlet, MW	1.52	1.37	1.44	7.08	6.45	0.00	6.45
heat capacity at outlet, kJ/s	6374.10	1442.21	7081.27	6454.78	6519.78	0.50	6518.51
heat capacity at outlet, MW	6.37	1.44	7.08	6.45	6.52	0.00	6.52
d MW	4.85	0.07	5.64	-0.63	0.07	0.00	0.06

SVARTSENGI POWER PLANT II
HEAT AND MASS BALANCE
SUMMATION

STREAM LOCATION	I ORMAT CONDENSER	II D.C. CONDSR./ DEAERATOR	III PRE- HEATER	IV FINAL HEATER	VI SUPPLY TO GRINDAVIK
cold water flow, kg/s	105.00				
initial temperature, °C	5.00				
final temperature, °C	21.30				
steamflow to vaporizer(MP), kg/s	14.45				
binary fluid flow, kg/s	25.76				
Ormat generator output, kW	3010.00				
condenser inlet temperature, °C	V				
condenser outlet temperature, °C	V				
vaporizer inlet temperature, °C	V				
vaporizer outlet temperature, °C	V				
T/G inlet fluid enthalpy, kJ/kg	1385.53				
T/G inlet pressure, bar a.	7.80				
T/G fluid exhaust enthalpy, kJ/kg	V				
binary fluid sat. temp., °C	V				
binary gaseous enthalpy, kJ/kg	1385.53				
binary liquid enthalpy, kJ/kg	1106.10				
massflow to D.C. condenser, kg/s		105.00			
steamflow to D.C. condenser, kg/s		11.00			
inlet temp. of cold fluid, °C		21.30			
outlet temp. of hot water, °C		79.14			
massflow from D.C. condenser, kg/s		116.00			
massflow to preheater, kg/s			116.00		
inlet temp. to pre-heater, °C			79.14		
outlet temp. from pre-heater, °C			94.97		
massflow of M.P. steam, kg/s			2.58		
inlet temp. to final heater, °C				94.97	
outlet temp. from final heater, °C				96.83	
massflow of H.P. steam, kg/s				0.58	
massflow from final heater, kg/s				116.00	
heat capacity at inlet, kJ/s	2206.05	9244.20	38362.18	45676.88	
heat capacity at inlet, MW	2.21	9.24	38.36	45.68	
heat capacity at outlet, kJ/s	9244.20	38362.18	45676.88	46653.57	
heat capacity at outlet, MW	9.24	38.36	45.68	46.65	
d MW	7.04	29.12	7.31	0.98	

SVARTSENGI POWER PLANT II
HEAT AND MASS BALANCE
R-6 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II D.C. CONDSR./ DEAERATOR	III PRE- HEATER	IV FINAL HEATER	VI SUPPLY TO KEFLAVIK
cold water flow, kg/s	105.00				
initial temperature, °C	5.00				
final temperature, °C	21.30				
steamflow to vaporizer(MP), kg/s	14.45				
initial temperature, °C	105.00				
final temperature, °C	95.00				
binary fluid flow, kg/s	25.76				
Ormat generator output, kW	3010.00				
condenser inlet temperature, °C	V				
vaporizer inlet temperature, °C	V				
vaporizer outlet temperature, °C	V				
T/G inlet fluid enthalpy, kJ/kg	1385.53				
T/G inlet pressure, bar a.	7.8				
T/G fluid exhaust enthalpy, kJ/kg	V				
binary fluid sat. temp., °C	V				
binary gaseous enthalpy, kJ/kg	1385.53				
binary liquid enthalpy, kJ/kg	1106.10				
massflow to the D.C. condenser, kg/s		35.00			
steamflow to D.C. condenser, kg/s		3.71			
inlet temp. of cold fluid, °C		21.30			
outlet temp. of hot water, °C		80.20			
inlet temp. of L.P. steam, °C		74.60			
massflow from D.C. condenser, kg/s		38.71			
massflow to preheater, kg/s			29.00		
inlet temp. to pre-heater, °C			79.14		
outlet temp. from pre-heater, °C			89.90		
massflow of M.P. steam, kg/s			0.59		
inlet temp. of M.P. steam, °C			120.00		
outlet temp. of M.P. steam, °C			116.00		
inlet temp. to final heater, °C				89.90	
outlet temp. from final heater, °C				85.40	
massflow of H.P. steam, kg/s				-0.25	
inlet temp. of H.P. steam, °C				159.00	
outlet temp. of H.P. steam, °C				129.00	
massflow from final heater, kg/s				29.00	
heat capacity at inlet, kJ/s	2206.05	3081.40	9590.54	10808.79	
heat capacity at inlet, MW	2.21	3.08	9.59		
heat capacity at outlet, kJ/s	9244.20	12964.56	10808.79	10321.32	
heat capacity at outlet, MW	9.24	12.96	10.81	10.32	
d MW	7.04	9.88	1.22	10.32	

SVARTSENGI POWER PLANT II
HEAT AND MASS BALANCE
R-7 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II D.C. CONDSR./ DEAERATOR	III PRE-HEATER	IV FINAL HEATER	VI SUPPLY TO KEFLAVIK
cold water flow, kg/s	105.00				
initial temperature, °C	5.00				
final temperature, °C	21.30				
steamflow to vaporizer(MP), kg/s	14.45				
initial temperature, °C	105.00				
final temperature, °C	95.00				
binary fluid flow, kg/s	25.76				
Ormat generator output, kW	3010.00				
condenser outlet temperature, °C	V				
vaporizer inlet temperature, °C	V				
vaporizer outlet temperature, °C	V				
T/G inlet fluid enthalpy, kJ/kg	1385.53				
T/G inlet pressure, bar a.	7.8				
T/G fluid exhaust enthalpy, kJ/kg	V				
binary fluid sat. temp., °C	V				
binary gaseous enthalpy, kJ/kg	1385.53				
binary liquid enthalpy, kJ/kg	1106.10				
massflow to D.C. condenser, kg/s		35.00			
steamflow to D.C. condenser, kg/s		3.77			
inlet temp. of cold fluid, °C		21.30			
outlet temp. of hot water, °C		80.30			
inlet temp. of L.P. steam, °C		88.90			
massflow from D.C. condenser, kg/s		38.77			
massflow to preheater, kg/s			29.00		
inlet temp. to pre-heater, °C			79.14		
outlet temp. from pre-heater, °C			99.30		
massflow of M.P. steam, kg/s			1.11		
inlet temp. of M.P. steam, °C			120.00		
outlet temp. of M.P. steam, °C			117.00		
inlet temp. to final heater, °C				99.30	
outlet temp. from final heater, °C				102.80	
massflow of H.P. steam, kg/s				0.19	
inlet temp. of H.P. steam, °C				159.00	
outlet temp. of H.P. steam, °C				129.00	
massflow from final heater, kg/s				29.00	
heat capacity at inlet, kJ/s	2206.05	3081.40	9590.54	12030.23	
heat capacity at inlet, MW	2.21	3.08	9.59	12.03	
heat capacity at outlet, kJ/s	9244.20	12986.03	12030.23	12397.07	
heat capacity at outlet, MW	9.24	12.99	12.03	12.40	
d MW	7.04	9.90	2.44	0.37	

SVARTSENGI POWER PLANT II
HEAT AND MASS BALANCE
R-8 AUGUST 10, 1992

STREAM LOCATION	I ORMAT CONDENSER	II D.C. CONDSR./ DEAERATOR	III PRE-HEATER	IV FINAL HEATER	VI SUPPLY TO GRINDAVIK
cold water flow, kg/s	105.00				
initial temperature, °C	5.00				
final temperature, °C	21.30				
steamflow to vaporizer(MP), kg/s	14.45				
initial temperature, °C	105.00				
final temperature, °C	95.00				
binary fluid flow, kg/s	25.76				
Ormat generator output, kW	3010.00				
condenser outlet temperature, °C	V				
vaporizer inlet temperature, °C	V				
vaporizer outlet temperature, °C	V				
T/G inlet fluid enthalpy, kJ/kg	1385.53				
T/G inlet pressure, bar a.	7.8				
T/G fluid exhaust enthalpy, kJ/kg	V				
binary fluid sat. temp., °C	V				
binary gaseous enthalpy, kJ/kg	1385.53				
binary liquid enthalpy, kJ/kg	1106.10				
massflow to D.C. condenser, kg/s		35.00			
steamflow to D.C. condenser, kg/s		3.51			
inlet temp. of cold fluid, °C		21.30			
outlet temp. of hot water, °C		76.90			
inlet temp. of L.P. steam, °C		77.10			
massflow from D.C. condenser, kg/s		38.51			
massflow to preheater, kg/s			29.00		
inlet temp. to pre-heater, °C			79.14		
outlet temp. from pre-heater, °C			95.70		
massflow of M.P. steam, kg/s			0.88		
inlet temp. of M.P. steam, °C			105.00		
outlet temp. of M.P. steam, °C			96.00		
inlet temp. to final heater, °C				95.70	
outlet temp. from final heater, °C				102.30	
massflow of H.P. steam, kg/s				0.39	
inlet temp. of H.P. steam, °C				159.00	
outlet temp. of H.P. steam, °C				159.00	
massflow from final heater, kg/s				29.00	
heat capacity at inlet, kJ/s	2206.05	3081.40	9590.54	11541.31	
heat capacity at inlet, MW	2.21	3.08	9.59	11.54	
heat capacity at outlet, kJ/s	9244.20	12252.14	11541.31	12397.07	
heat capacity at outlet, MW	9.24	12.25	11.54	12.40	
d MW	7.04	9.17	1.95	0.86	