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BERLÍN GEOTHERMAL PROJECT, PRELIMINARY POWER PLANT DESIGN

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ABSTRACT

The main objectives of this report are to determine the optimum pressure for the technical operation of a geothermal field and optimize the turbine selection. Calculations are made to guide the selection of the first condensing power plant in the Berlín geothermal field in El Salvador. Silica content in the Berlín wells is relatively high; saturation of amorphous silica is reached at about 10 bar-a. The most important result obtained showed that the optimum wellhead pressure is in the range 7-10 bar-a. For a single flow and single pressure turbine, a 560 mm last stage blade length is selected. The power output is 30 MWe with minimum exhaust loss of 15.63 kJ/kg, annular velocity of 177 m/s and a specific steam consumption of 1.85 kg/s per MWe. Management strategies to deal with silica scaling are presented.

1. INTRODUCTION

The Berlín geothermal field is located 100 km to the east of San Salvador, the capital city of El Salvador. Six deep wells (TR-1, 2, 3, 4, 5 and 9) that were drilled in the years 1978-1981 confirmed the existence of a reservoir of commercial interest for power generation (Figure 1). In 1992 the electric executive agency, Comisión Ejecutiva Hidroeléctrica del Río Lempa, CEL, commissioned two back pressure units of 5 MWe each. Using steam from wells TR-2 and TR-9, the maximum power generated is 8 MWe. All of the residual geothermal water is re-injected into wells TR-1, 8, 10 and 14.

The geothermal fluid exhibits a typical sodium chloride composition. The total dissolved solids (TDS) in the reservoir are 7,000 to 11,000 ppm and silica is up to 620 ppm, with a pH close to 6. Non-condensable gases amount to 0.25-0.50% by weight of the steam at a separation pressure of 8 bar-a. The silica concentration in the Berlín wells is relatively high. Calculations show that saturation conditions with respect to amorphous silica will occur at a separation pressure of 10 bar-a (Martínez, 1997). This pressure is assumed as a threshold, below which significant scaling of silica will occur in the separator, re-injection pipes and re-injection wells.

The production characteristics of wells TR-2, 3, 5 and 9 are summarized in Figure 2 and Table 1.

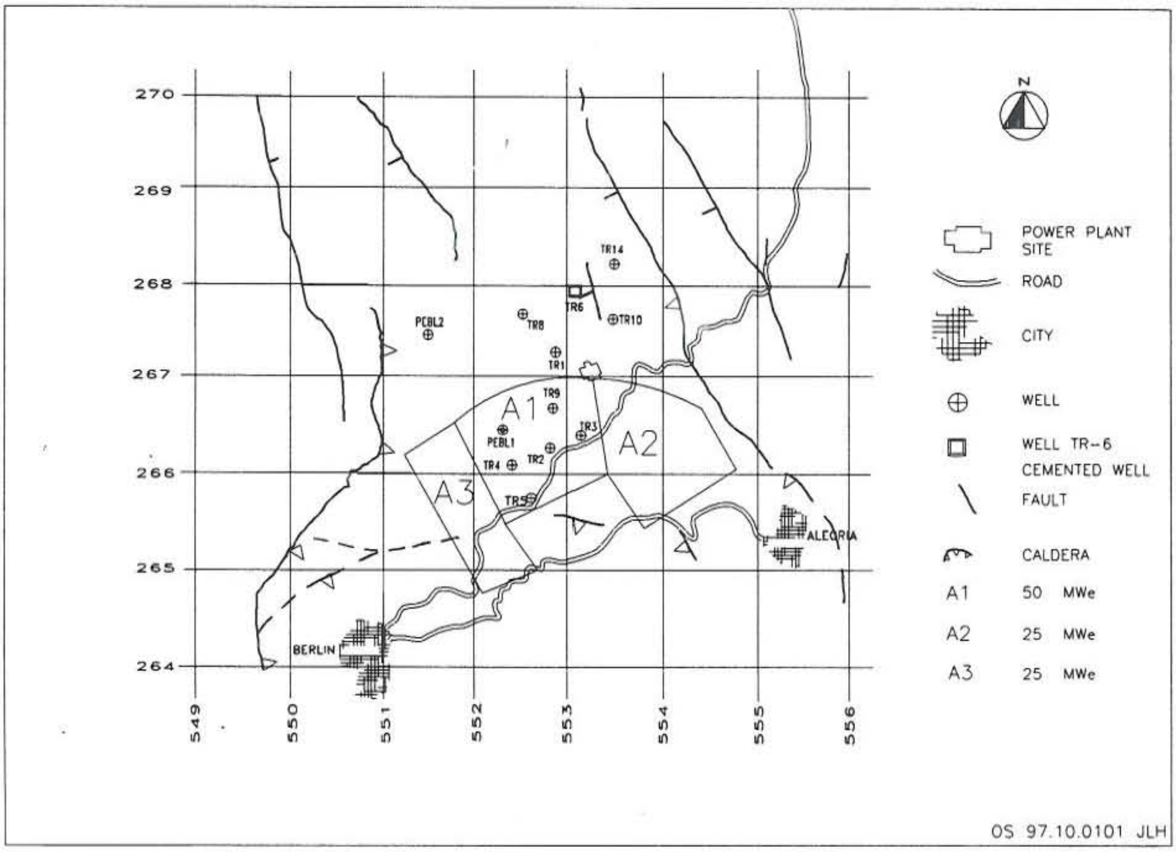


FIGURE 1: The Berlín geothermal field and areas of potential interest for power plants

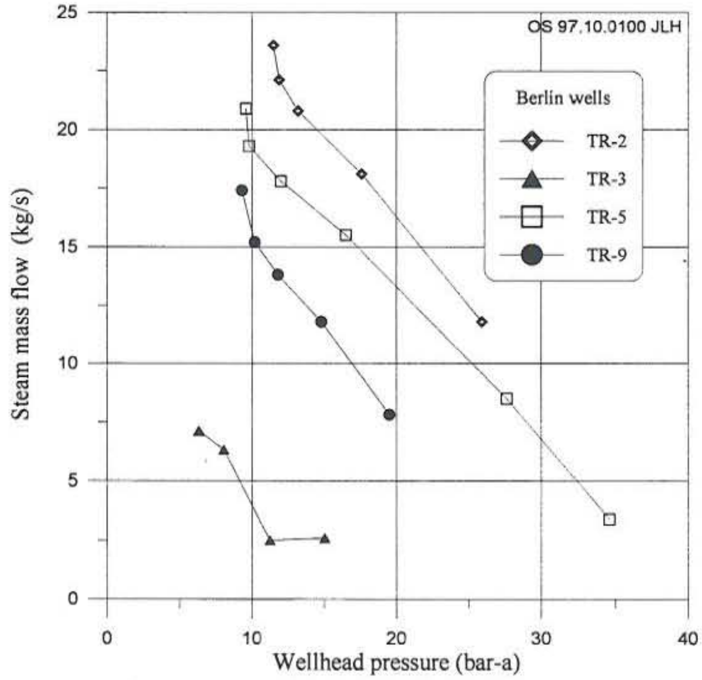


FIGURE 2: Output curves of wells TR-2, TR-3, TR-5 and TR-9

The conceptual model of the field (Parini et al., 1995) established the areas of potential interest for a commercially exploitable reservoir, based on of the results of the wells and geoscientific investigations (Figure 1). The proven area, identified as the zone directly tested by means of deep wells, covers a surface of 1.9 km^2. The probable area, identified as the zone associated with favourable structural conditions and with the presence of hydrothermal manifestation, covers 6.8 km^2. The possible area, identified as the zone included within the NNW-SSE graben, is 20 km^2.

Parini et al.'s resource assessment for the Berlín geothermal field indicates a minimum potential of 50 MWe, with a high probability for of a potential of at least 100 MWe. Large scale exploitation of the proven potential of 50 MWe is planned to start at the end of 1998.

TABLE 1: Production characteristics of Berlín wells

Well	Total flow (kg/s)	Enthalpy (kJ/kg)	Steam fraction (%)	Steam flow (kg/s)	Power (MWe)	Depth (m)	Maximum temperature (°C)
TR-2	78	1384	30	23.4	11.7	1903	293
TR-3	42	1216	22	9.2	4.6	2300	293
TR-5	58	1434	29	16.7	8.3	2086	301
TR-9	65	1293	18	11.7	6	2298	293

2. DESIGN FEATURE CONSIDERATIONS

2.1 Single flash power plant

The flow diagram for the Berlín development is outlined in Figure 3. To prevent scale deposition in re-injection wells, a single flash system and hot re-injection is adopted. In its simplest form, it consist of a production well(s), steam separator(s), a turbine-generator, a condenser, cooling tower and injection well(s).

The phase diagram is shown in Figure 4 in temperature-entropy coordinates with the corresponding state points. The double dome curve in Figure 4 is the state of saturation, left side water saturation and right side steam saturation. The maximum point on the saturation curve is the critical point.

To calculate the electric power output from the generator, the following parameters must be known (Jónsson, 1997):

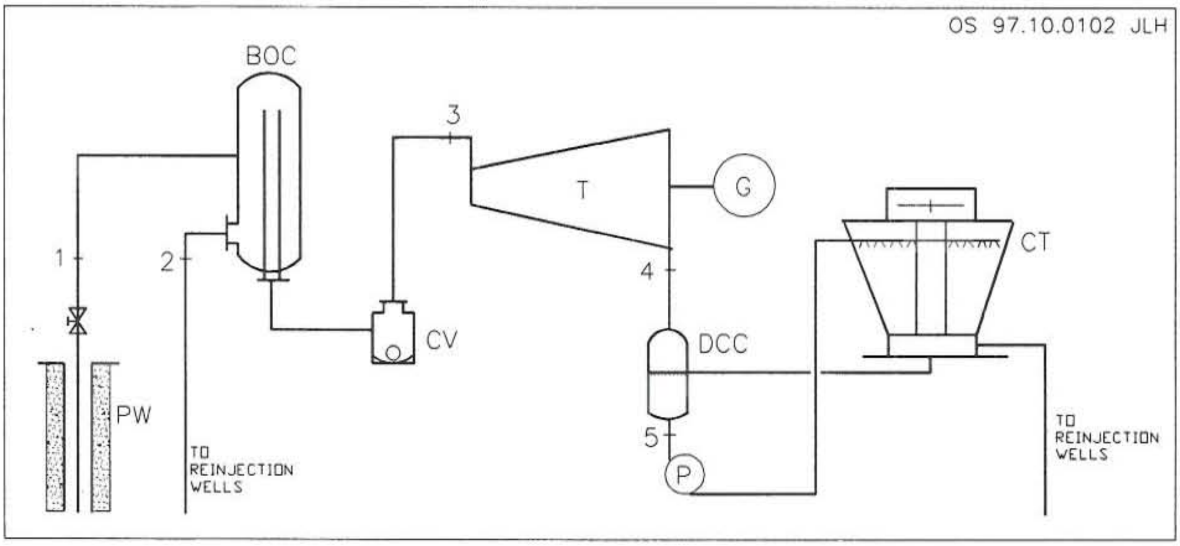


FIGURE 3: Single flash flow diagram for the Berlín field, PW = Production well; BOC = Bottom outlet centrifugal separator; CV = Check valve; T = Turbine; G = Generator; DCC = Direct contact condenser; P = Pump; CT = Cooling tower

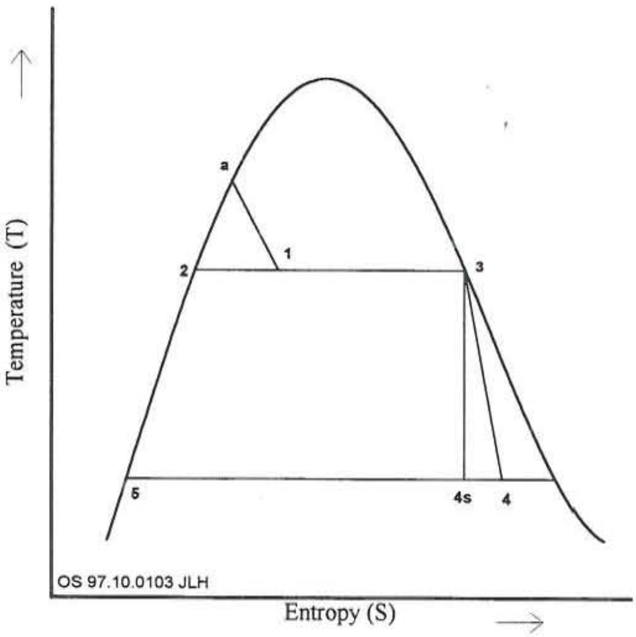


FIGURE 4: Temperature-entropy diagram for a single- flash power plant, see Figure 3 for location of state points

Enthalpy of geothermal fluid (h_a), or the temperature of a liquid-dominated reservoir (t_a);
Total mass flow from the well (m_t);
Pressure in the steam separator or turbine inlet (p_3);
Pressure in the condenser (p_4);
Isentropic efficiency of the turbine (η_i) and generator efficiency (η_g).

If the steam fraction at the separator inlet is x_1, the enthalpy of the mass flow entering the separator is

$$h_1 = x_1 h_{g3} + (1 - x_1) h_{f2} \qquad (1)$$

The total mass flow from the well is m_t; hence the steam mass flow to the turbine is

$$m_s = x_1 m_t \qquad (2)$$

The available isentropic mechanical work from the turbine is equal to the change in enthalpy (h_3-h_4) times the steam mass flow. To calculate the electric power output (*MW*), the isentropic and generator efficiency must be taken into account:

$$MW = m_s (h_3 - h_{4s}) \eta_i \eta_g \qquad (3)$$

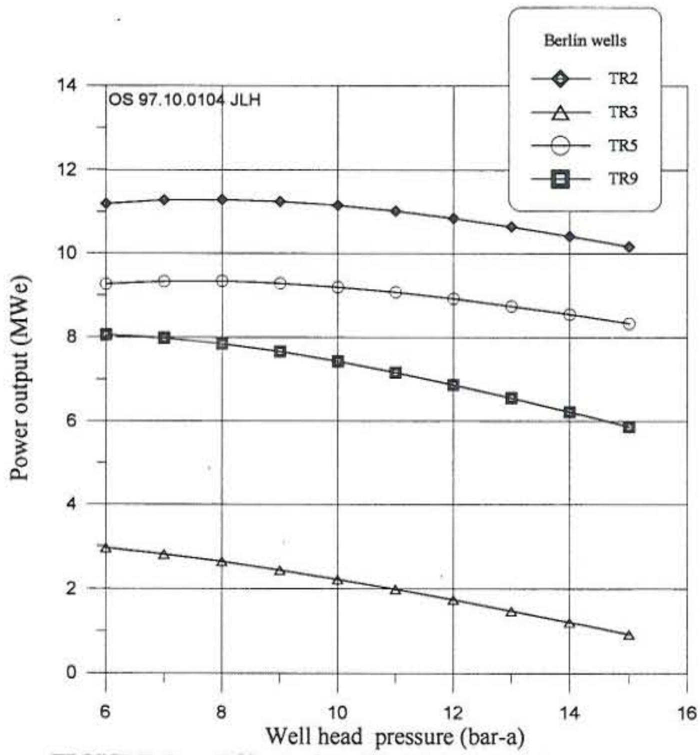


FIGURE 5: Effect of wellhead pressure on power output wells TR-2, TR-3, TR-5 and TR-9

From Equation 3 it can be seen that the electrical power generation for isentropic expansion through the turbine depends on the steam flowrate, and the enthalpy difference between the inlet and exit of the turbine. The amount of steam produced depends on the separation pressure. The lower the pressure, the greater the steam fraction and hence more power production. On the other hand, the lower the inlet pressure to the turbine is, the lower the inlet enthalpy (h_3) and hence lower power output. It follows that there must be an optimum condition where electrical power output from a given well shall be at a maximum. Figure 5 shows output vs. wellhead pressure and the maximum for wells TR-2, 3, 5 and 9 in the Berlín geothermal field.

2.2 Optimum turbine inlet pressure

The choice of turbine inlet steam pressure is governed principally by the pressure/flow characteristics of the wells. Individual wellhead pressures are related to the turbine inlet pressure by adding the pressure drop in the pipework, separator, valves etc. between the turbine and the well.

The power potential for wells TR-2, 3, 5 and 9 is plotted against varying wellhead pressure (Figure 5). For these calculations the turbine outlet pressure was kept constant at 0.1 bar, isentropic efficiency at 0.9 and generator efficiency at 1.0. The optimum well head pressure which allows the maximum generation of energy from wells TR-2 and 5 ranges from 7 to 10 bar-a. To avoid the prospect of silica deposition inside the re-injection and steam pipelines, a turbine inlet pressure of 10 bar-a is selected. Assuming a pressure drop of 2 bars between the well head and turbine means that the operation pressure for the wells will be 12 bar-a. The turbine power potential that will be extracted from each well at well head pressure 12 bar-a is 4% below the maximum.

In order to determine whether the wells can sustain production at the proposed wellhead pressure, output curves for the most probable case of field evolution must be generated. This has been done by coupling the results from the reservoir simulation to a wellbore simulator (Cozzini et al., 1995).

2.3 Steam turbine

A fundamental factor in the design of a geothermal power plant is the capacity of the turbine. Upon knowing the production capacity of the reservoir, a selection of the turbine size follows. Some factors that influence the selection are available steam, thermodynamic and chemical characteristics of the steam, type of turbine, effect of the natural decline in flowrate and pressure of the wells, increase or decrease of the non-condensable gases, and financial factors at present and in the future.

2.3.1 Energy conversion efficiency

It can be shown from the second law of thermodynamics that no heat engine can be more efficient than a reversible heat engine working between the same temperature limits. Carnot showed that the most efficient cycle possible is one in which all the heat supplied is supplied at one fixed temperature, and all the heat rejected at a fixed lower temperature. The cycle, therefore, consists of two isothermal processes joined by two adiabatic processes.

Hence, we have the Carnot cycle efficiency:

$$\eta_{Carnot} = 1 - \frac{T_2}{T_1} \tag{4}$$

To achieve theoretical efficiency, the isothermal parts of the cycle have to be carried out infinitely slowly so that the working substance can come into thermal equilibrium with the heat reservoir. Under these conditions, the power output is clearly zero since it takes an infinite time to do a finite amount of work. To obtain a finite power output the cycle is speeded up; an expression for the efficiency has been developed under conditions of maximum power output, presented below (Curzon and Ahlborn, 1975).

$$\eta' = 1 - \left(\frac{T_2}{T_1}\right)^{\frac{1}{2}} \tag{5}$$

Considering the selected inlet and outlet conditions of the turbine as a source and sink temperatures, with T_2 = 46°C and T_1 = 180°C, the theoretical efficiency for maximum power output for the Berlín geothermal field is on the order of 16%.

2.3.2 Effect of last-stage blade length

In practice, the internal efficiency of a steam turbine does not include the loss at the turbine exhaust end. The exhaust end loss occurs between the last stage of a low-pressure turbine and the condenser inlet, which very much depends on steam velocity. The exhaust end loss generally includes actual leaving loss, gross hood loss, annulus-restriction loss, and turn-up loss. Under full load conditions, the exhaust loss is typically around 3% of the turbine's available energy. One way of reducing exhaust loss is to reduce the absolute steam velocity at the last stage. This can be achieved by increasing the last stage blade length or the number of steam flows (single, double, four flow).

In order to evaluate the optimum capacity of the turbine, a series of equations are presented below, where the potential output of the turbine is written as a function of the allowable centrifugal (tension) stress of the last stage blade.

The total blade exit area must be such as to satisfy the condition of continuous mass flow of steam. The steam mass flow can be written (Li and Priddy, 1985):

$$m_s = 2\pi r_m l \frac{V_{an}}{v(1-0.01y)} \tag{6}$$

The centrifugal (tension) stress in a blade will generally be highest at the root and depends upon the distribution of the mass and sectional area throughout the blade length. For a blade of uniform cross-section, the centrifugal stress is given by (Vincent, 1950):

$$S_c = \rho_m w^2 l r_m \; ; \quad (w = 2\pi N) \tag{7}$$

Using Equations 3, 6 and 7 gives the following result:

$$MW = \frac{S_c V_{an}}{2\pi v \rho_m N^2 (1-0.01y)} (h_3 - h_4)\, \eta_i \eta_g \tag{8}$$

Consider the following data for Berlín:

Inlet turbine pressure:	1.0 MPa-a.;
Condenser pressure:	0.01 MPa-a;.
Maximal allowable tension stress:	480 MPa;
Exhaust loss:	15.63 kJ/kg;
Angular velocity:	3600 rpm;
Density blade's material:	8000 kg/m³;
Steam moisture:	18%;
η_i:	0.9;
η_g:	1.0.

With Equation 8, the power that can be obtained in a turbine of single flow is 30 MWe. For a tension stress of 480 MPa, bucket material density of 8000 kg/m^3 and a relation of mean bucket diameter and blade length of 2.7, the last stage blade length is calculated as 560 mm, using Equation 7. An exhaust loss of 15.63 kJ/kg corresponds to an annulus velocity of 177 m/s, (Li and Priddy, 1985). Using Equation 6, with a steam density of 8.98 kg/m^3 at the outlet of the turbine, the specific steam consumption is 1.85 kg/s per MWe with a volumetric flowrate at the outlet of 512 m^3/s.

2.4 Cooling system

2.4.1 Climatic aspects in Berlín

The performance of cooling towers is directly influenced by atmospheric conditions, or more particularly the ambient wet bulb temperature. The meteorological parameters of the Berlín area have been collected from the Berlín station, 4 km southwest of the power plant site, at 1050 m a.s.l., and from the Santiago de María station, 6 km to the southeast, at 920 m a.s.l. The historical record is shown in Figure 6.

The site selected for the power plant in the Berlín geothermal field is at 660 m a.s.l. Considering the mean atmospheric thermal gradient in El Salvador, 0.6°C/100 m, the values measured at the meteorological stations will be increased by about 2°C.

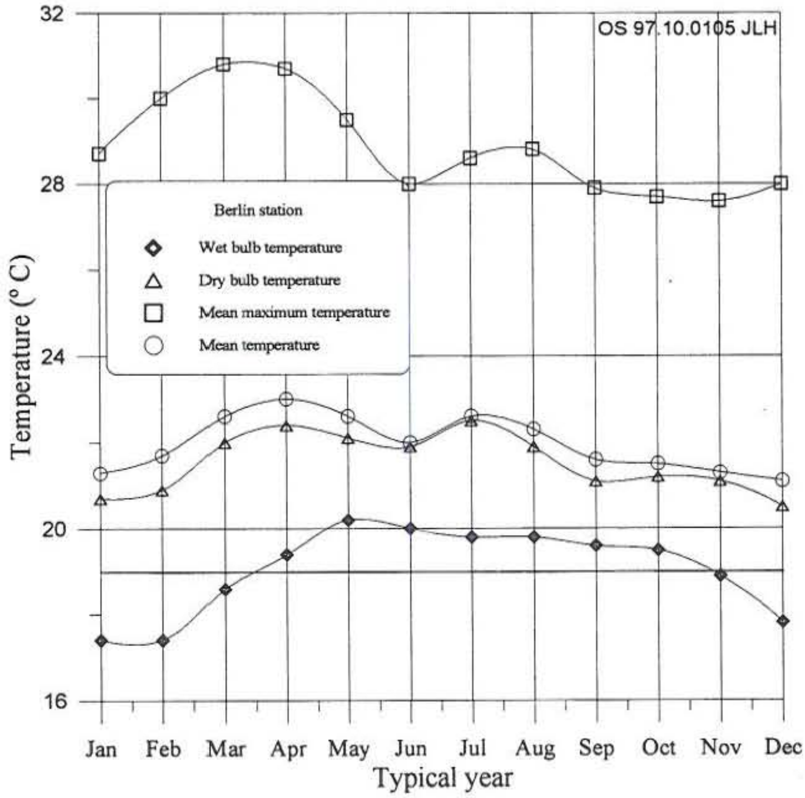


FIGURE 6: Meteorological data. Berlín and Santiago de María stations

The wet bulb temperature for the power plant site is within the range 19-22°C, and for the purpose of this report 21°C is taken as design wet bulb temperature. The main wind direction in the zone is predominantly from the south.

2.4.2 Evaporative cooling towers

Evaporative cooling towers are devices that cool water by bringing it into contact with air. Wet cooling towers dissipate heat rejected by the plant to the environment by (1) sensible heat transfer, owing to the difference in the temperature of water and air and (2) latent heat transfer, owing to evaporation of a portion of the re-circulation water itself.

Wet cooling towers are classified as either mechanical-draft or natural-draft cooling towers. Each of these types is further classified as counter-flow or cross-flow cooling towers (see Figure 7).

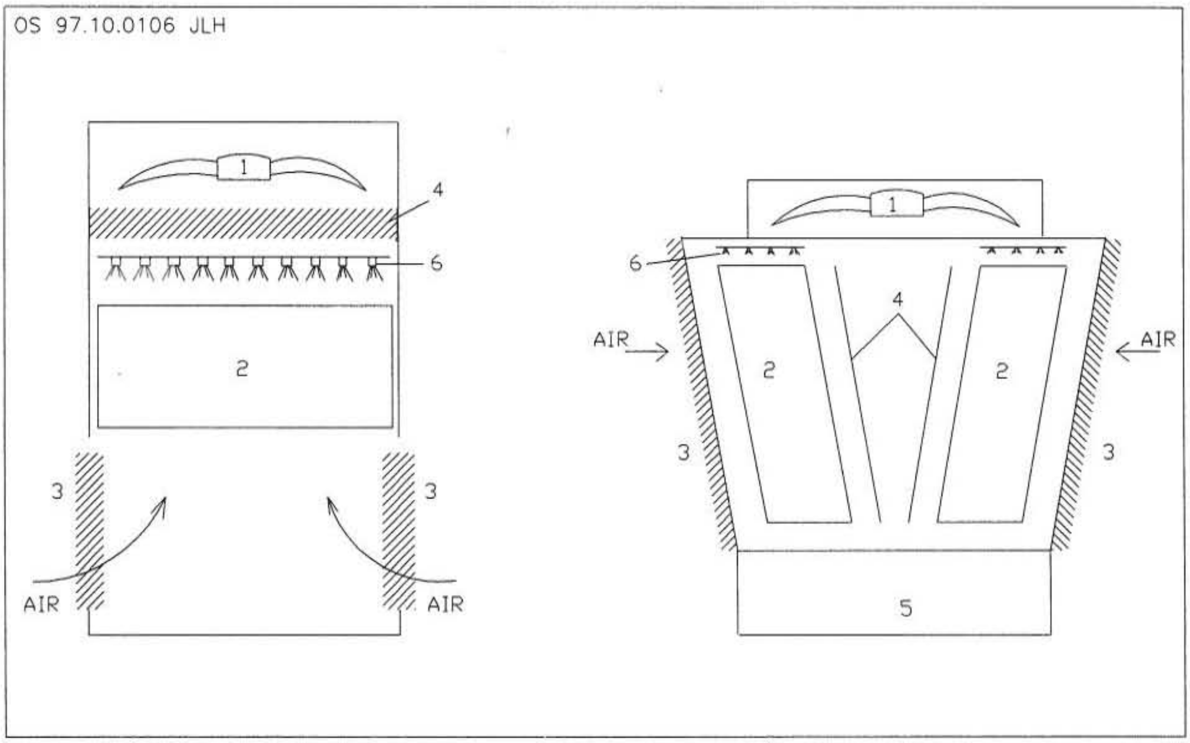


FIGURE 7: Mechanical draft cooling tower, 1 = Fan; 2 = Tower fill matrix; 3 = Inlet louvers; 4 = Drift eliminators; 5 = Water basin; 6 = Water spray system

A mechanical-draft cooling tower as treated here is usually composed of the following components:

1. Air moving equipment such as fans;
2. Tower fills (tower packing);
3. Air inlet louvers;
4. Drift eliminators;
5. Water storage basins;
6. Inlet water distributors.

A detailed discussion of cooling tower theory is not included here. In brief, in a counter-flow wet cooling tower, there are L kilograms of water and G kilograms of dry air per second over a unit area flowing through the tower, under steady-state and steady-flow conditions. Assuming that the evaporation rate in the control volume is negligible, then

$$L C_{pw} dt_w = G dH_a \tag{9}$$

Applying the principles of heat transfer and mass transfer to the interface between the water and air will lead to Merkel's equation (Li and Priddy, 1985):

$$Ka\frac{V}{L} = \int_{t_e}^{t_i} \frac{C_{pw}\, dt_w}{H_w - H_a} \tag{10}$$

The theoretical derivation of Equation 8 is specific only to counter-flow fill with water and air at moderate temperatures. It has been successfully applied to cross flow-fill by segmenting the volume into a two-dimensional array of elemental volumes of a unit depth. The tower characteristics for a cross-flow wet cooling tower has an equivalent form of KaY/L.

Ka in Equation 10, can be expressed as a function of L and G. A Kelly and Swenson counter-flow tower fill (Deck D), at a height of 6 m, has (Li and Priddy, 1985)

$$Ka = 0.07\, L^{0.46}\, G^{0.54} \tag{11}$$

Consider the conditions for Berlín where wet bulb temperature is 21°C; approach is 8°C and the cooling range 12°C. Using Equations 9, 10 and 11, for a counter-flow cooling tower and a turbine of 30 MWe, the result is 2.0 m^3/s water flow to the cooling tower and 589 m^2 area of the cooling tower. The Tchebycheff four-ordinate method (Wilbur, 1985) was used to solve Equation 10.

2.4.3 Definition of terms used with cooling towers and typical values (Kestin et al., 1980; Wilbur, 1985)

Wet-bulb temperature is the lowest temperature at which evaporation can occur for the specific conditions of the atmosphere. Selection of a design wet-bulb temperature that will not be exceeded more than 2 or 3% of the time is customary.

Dry-bulb temperature is the sensible temperature of the atmosphere as measured by a mercury-in-glass thermometer or other devices.

Approach is the temperature difference between the liquid that was cooled and the wet bulb temperature. The wet tower approach limit can be as low as 4°C to the ambient-design wet-bulb temperature in a mechanical draft cooling tower, but typical values used are 6-11°C (10-20°F).

Range refers to the temperature range between the initial and final cooled temperatures, namely, the hot and cold temperatures. It is usually between 7 and 17°C (12-30°F).

Water load, L, is usually given as a total quantity per unit of time per area of the tower fill. It is usually 7-12 $m^3/h\text{-}m^2$ (3-5 gpm/ft^2) for a counter-flow wet tower and 30-40 $m^3/h\text{-}m^2$ (12-16 gpm/ft^2) for a typical cross-flow tower.

Air load, G, the gas or air cooling the liquid, is expressed like the water load. Generally, it is in the range 7500-12500 kg/h-m^2 (1500 to 2500 lb/h-ft^2).

Re-circulation is the percentage of the exhaust of the cooling tower drawn back into its inlet. It is a function of the size of the tower and the magnitude and direction of the wind speed and causes an increase in the air temperature at the inlet. Though it can in some cases be as high as 10%, it is usually 2-4% and the result is a 1°C (1-2°F) average inlet rise in the wet-bulb temperature.

Drift loss is the quantity of liquid entrained in the exhaust of the tower. It can be expressed as a total quantity or percentage of flow. The total drift is typically below 0.005% of the quantity of water cooled.

L/G is the ratio of liquid cooled to air (gas) flow. It ranges from 1.3 to 1.8.

Condenser terminal difference is the difference in temperature between the water leaving the condenser and the condensing steam temperature.

2.4.4 Wet cooling tower performance

The interaction of the fill thermal capability with particular performance requirements is analogous to

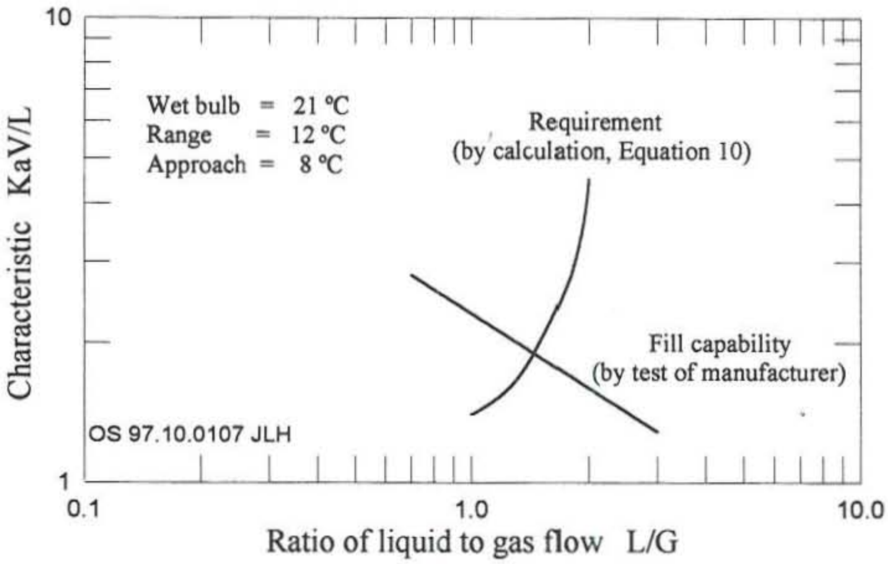


FIGURE 8: Wet cooling tower characteristic curve

that of a pump curve and system head curve. The tower operates at the intersection of the two curves as depicted by Figure 8. For a given fixed condition of wet bulb temperature, approach and range, the requirement curve in Figure 8 is constructed using Equation 10 and a set of L/G values.

2.5 Condenser

Two methods of condensing a vapour are; (1) mixing the vapour with a liquid so that the vapour can reject its latent energy to the liquid, with a consequent increase in the temperature of the liquid, and (2) transferring the latent energy of the vapour through a surface to another fluid that is at a lower temperature.

The exhaust pressure of a high-efficiency steam turbine must be well below atmospheric pressure. Vacuum is achieved by condensing the exhaust steam. The degree of vacuum obtained depends on the turbine loading, the amounts of non-condensable gases present in the condenser, due to gas in the steam and in-leakage, and most importantly the condensing temperature of the steam as influenced by the temperature of the cooling water leaving the condenser. In a typical single-fluid direct contact condenser, cooling water is sprayed into the turbine exhaust steam, and condensation occurs on the water droplets. The terminal temperature difference theoretically could be zero, but in actual practice may be as high as 6°C (Kestin et al., 1980). The mixing or direct-contact condenser is of the low-level jet type.

Heat balance calculation for a direct-contact condenser assumes that the condenser is perfectly insulated, the process is one of steady flow, and that the energy loss from the vent to the atmosphere or some other region is negligible.

$$W_1 = S \frac{(h_s - h_{w2})}{(h_{w2} - h_{w1})} \tag{12}$$

$$W_2 = S \frac{(h_s - h_{w1})}{(h_{w2} - h_{w1})} \tag{13}$$

The Heat Exchange Institute recommends that for steam turbines, the difference between the enthalpy of the entering steam and the enthalpy of the leaving mixture be taken as 2210 kJ/kg (Kestin et al., 1980).

For the Berlín area, with a vacuum pressure of 0.1 bar-a and steam mass flow of 55.5 kg/s entering the condenser, and using Equations 12 and 13, the water flow entering the condenser is 2008 kg/s and the water flow leaving the condenser is 2064 kg/s. The condenser heat rate is 4.6×10^8 kJ/kg.

2.6 Turbine exhaust pressure

The basic factors to be considered in the optimisation of the turbine exhaust (outlet) pressure are the steam consumption of the turbine, the steam consumption of the air ejectors, the power consumption of the circulating water pumps, the power consumption of the cooling tower fans, and the capital cost of the plant and wells. These factors depend on the choice of design ambient wet-bulb temperature and condenser temperature terminal difference.

The ambient wet-bulb temperature is the factor which governs the limit of evaporative cooling in a cooling tower and hence the lowest water temperature, which in turn influences the maximum attainable condenser vacuum. When selecting a high vacuum pressure the turbine output is increased. As geothermal steam contains large amounts of non-condensible gases the power required to drive the gas extraction system has to be factored in, resulting in large construction costs for the gas extraction and cooling system.

Another factor that controls the selection of exhaust (outlet) pressure of the turbine is the exit steam wetness which should not exceed 12.5%. Otherwise, severe blade erosion will result.

The general objective is to determine the optimum combination of waste heat rejection equipment and steam condenser size and rating for a given turbine-generator unit. In a case where the waste heat rejection system is specified, for example, a mechanical-draft cooling tower, the scope will then include the application of cooling towers of various water temperature approaches and cooling ranges. The equipment must, thus, be matched to get the most economical combination of condenser and cooling tower.

Following the procedure described above, it can be shown that improved efficiency of the turbine results from a reduction in exhaust (outlet) pressure beyond 0.1 bar-a. It, however, costs more than the value that can be attributed to improved steam consumption. A design value of 0.1 bar-a has, therefore, been selected as the turbine exhaust pressure for the purpose of this report.

2.7 Gas removal system

Geothermal steam contains non-condensible gases in large amount compared with that of conventional thermal power plants. For that reason, geothermal power plants require large capacity non-condensible gases removal systems, which play a very important role in geothermal power generation. They also contribute a large portion of the total plant cost and total auxiliary power consumption.

Most geothermal power plants currently use a train of steam-jet ejectors to extract the non-condensible gases. Steam-jet ejectors have a relatively low capital cost and are highly reliability as they have no moving parts. They are, however, low efficiency devices and require a high flowrate of motive steam.

To reduce the power required by the non-condensable gases extraction system, some geothermal power plants utilize a higher performance hybrid system. These systems use steam-jet ejectors for high vacuum compression and liquid-ring vacuum pumps for low vacuum compression. Since the efficiency of a liquid-ring vacuum pump is superior to a steam-jet ejector, this hybrid approach reduces the power required by the extraction system. However, liquid-ring pumps have a higher capital cost.

Very high performance non-condensable gases extraction can be achieved by using centrifugal compressors which have higher efficiencies than ejectors or liquid-ring pumps. These compressors have, however, seen limited use in non-condensable gases extraction systems due to maintenance, high capital cost and reliability concerns about the high speed rotating assemblies (Forsha and Lankford, 1994).

In general, the choice of a particular extraction system depends on the amount of non-condensable gases. Economical considerations would suggest the use of steam ejectors for fractions below on percent, and of the compressor above two percent; the liquid ring pump has merits in between these values with strong overlapping (Lazzeri et al., 1995). The amount of non-condensable gases in the Berlín geothermal field is only 0.4% by weight of steam, thus, ejectors have been selected for their extraction.

2.8 Auxiliary power consumption

The power consumption of the electrical fan (cooling tower) and cooling water pumps can be calculated using the following basic equations (Hicks, 1987).

Electrical fans:

$$BHP = \frac{(ACFM)\ (SP + VP)}{6356\,(EFF)} \tag{14}$$

Cooling water pumps:

$$BHP = \frac{(gpm)\,H_t\,s}{3960\,e} \tag{15}$$

Consider the following information for Berlín:

Saturated air temperature at cooling tower exit:	34.7°C
L/G:	1.0
Total pressure drop in cooling tower:	15 mm of water
Water leaving the condenser:	2064 kg/s
Total head on the pump:	30 m
Specific gravity:	0.99
Fan mechanical efficiency:	0.8
Pump efficiency:	0.8

With Equations 14 and 15, the auxiliary power consumption for the fan and pump are 345 and 751 kW.

3. SPECIAL DESIGN CONSIDERATIONS

3.1 Steam purity

Inadequate steam purity from liquid-dominated or vapour-dominated geothermal resources can be detrimental to the long term economical and reliable operation of geothermal power plants. Contaminants in the motive steam of geothermal power plants cause scale buildup in the inlet nozzles which, in time, reduces power output.

There are two basic types of contaminants in geothermal steam, liquid entrainment, and volatile chemical species. Liquid entrainment can generally be resolved adequately using mechanical separators. The volatile species consist of slightly volatile substances such as silica, arsenic and boron, as well as highly volatile substances such as carbon dioxide, hydrogen sulfide and ammonia.

Bechtel National, Inc. conducted a study to establish the steam purity criteria for geothermal power plants and methods to achieve these requirements commercially. The steam purity criteria for silica and total dissolved solids were developed and are shown in Table 2 (Van der Mast et al., 1986).

TABLE 2: Steam purity criteria

	SiO_2 (ppm)	**TDS (ppm)**
Desired (conventional boiler criteria)	0.02	5
Allowable (slow scale buildup but no need for maintenance for at least 2 years; less than 10% power loss in 2 years)	0.1	15
Marginal (20% power loss in 1 year)	1	50

3.2 Control of steam impurities

3.2.1 Bottom outlet cyclone separator

The bottom outlet cyclone separator (BOC) is most often used in the geothermal industry for liquid-dominated resources as a primary separator. It is easy to operate and a highly efficient device that yields a separation efficiency of up to 99.95 percent.

The bottom outlet cyclone separator uses centrifugal action to assist in reducing moisture and dirt in the steam. To improve the separation efficiency, a wire mesh type demister incorporated in the same vessel as the bottom outlet cyclone separator has been installed in Svartsengi, Iceland. Figure 9 shows the location of the wire mesh inside it. An alternative location of the demister is in a second separator, typically located by the power house.

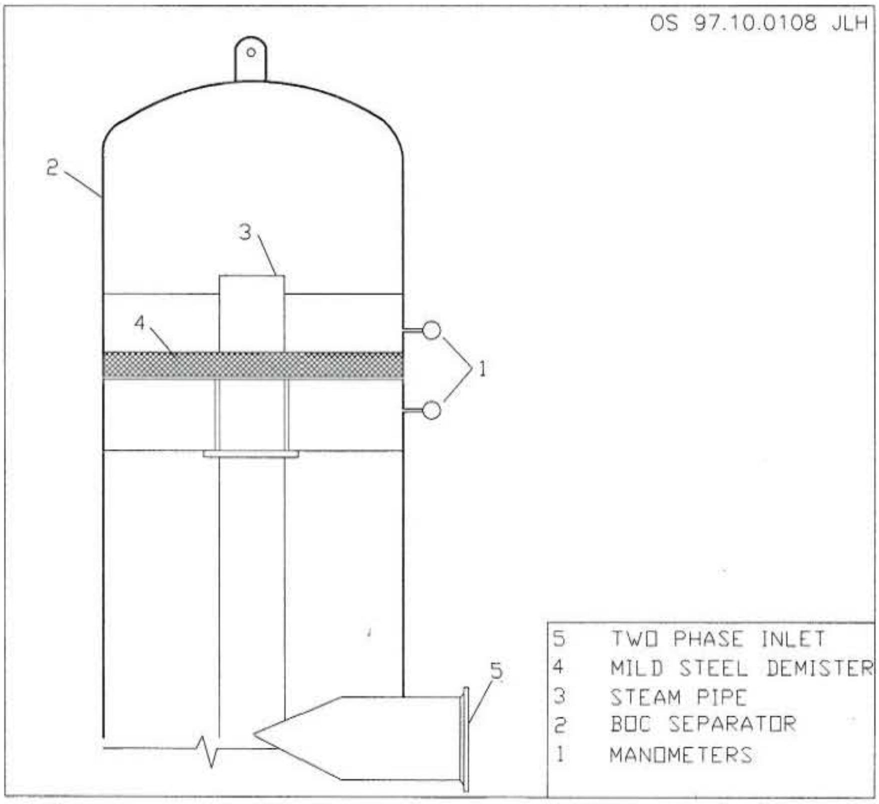


FIGURE 9: Bottom outlet cyclone separator with internal demister

3.2.2 Use of long steam pipes to obtain a scrubbing effect

Work carried out at Wairakei, New Zealand, (Reimann, 1993) has confirmed that steam lines can help in removing minerals from steam. Considering that a thin scale inside the pipes may effectively act as an anti-corrosive coating, attention should be payed to the effect of drain pots for long steam-pipes to avoid corrosion problems (Freeston, 1981).

3.2.3 Steam washing

Steam washing is a basic steam scrubbing technique of injecting steam condensate into the steam flow up-stream of a final separator. This will collect unwanted substances entrained and dissolved in the steam into the wash water. This is followed by separation of the liquid fraction from the flow. TDS, silica, boron, and arsenic can all be removed readily in this manner. Scrubbers can also be used to remove ammonia. However other non-condensable gases such as carbon dioxide and hydrogen sulfide cannot be removed readily by scrubbing. In order to optimize condensate water injection rates used for steam scrubbing, an on-site analytical test procedure must be developed, based on the turbine scale composition (Van der Mast et al., 1986).

3.3 On-load turbine washing

Turbine washing is a procedure used to clean steam turbine blading of solid deposition by intermittently injecting heated condensate into the steam. The cleaning process is carried out with the turbine operating at partial load. This technique has been successfully used in geothermal steam turbines for a number of years. It is generally not recommended for frequent use, due to the inherent danger of water droplet erosion and thermal shock effects. Turbine wet-steam washing must, for the above reasons, always be carried out with great care and under carefully controlled conditions.

The condensate washing action functions principally in two ways

1. It dissolves soluble solids contained in the deposited scale and so weakens the scale's matrix..
2. The weakened scale matrix, consisting largely of silica and its compounds, is then mechanically washed away by droplet impingement action and liquid flow.

It is generally recommended that steam entering the inlet nozzles of the turbine during washing has a dryness ranging between 90% and 95% (by weight) and that all casing and labyrinth seal drains be kept open. To minimize thermal shock and improve the homogeneity of the wet steam entering the turbine, it is advised that the temperature of the condensate injected into the steam flow be at least 100°C. The wet steam washing should be started gently, and the steam wetness controlled. The liquid injection quantity should be monitored and can be gradually increased until the values obtained indicate that the inlet steam is within the above wetness range.

It is recommended that the progress of the washing be initially gauged by chemical analysis of the condensate from the turbine casing drains. When concentrations measured in the condensate have reached normal values, the washing can stop. A recovery of any lost generating capacity also indicates adequate cleaning. It is further recommended that the cleaning be carried out at an approximately constant load, somewhere in the mid-load range. This improves the ability to accurately control the wetness of the inlet steam and reduces possible erosion effects while keeping steam velocities through the passages at a reasonable level for efficient scale removal.

The process is described by the schematic in Figure 10.

3.4 Inspection and cleaning ports in the gathering system

Scaling of steam/water transmission lines and especially injection pipelines is common in many geothermal operations. In previous works by Stock (1990) and Brown et al. (1995), methods were discussed for monitoring scale build-up in injection lines by pressure drops, and Klein (1995) recently discussed methods for predicting and suppressing scaling in re-injection lines and wells. As a part of the whole design of the gathering system, and in order to have scale management at a low level of over-saturation, ports must be considered for inspections and cleaning. The port design and location considered for the gathering system in the Berlín geothermal field are presented in Figure 11.

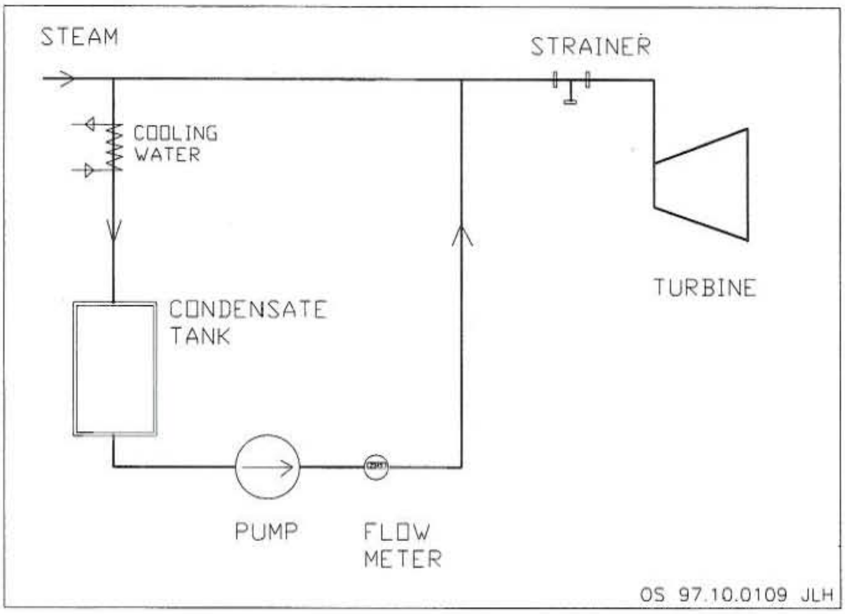


FIGURE 10: On-load turbine washing system

4. PROPOSED DESIGN FOR THE BERLÍN GEOTHERMAL FIELD

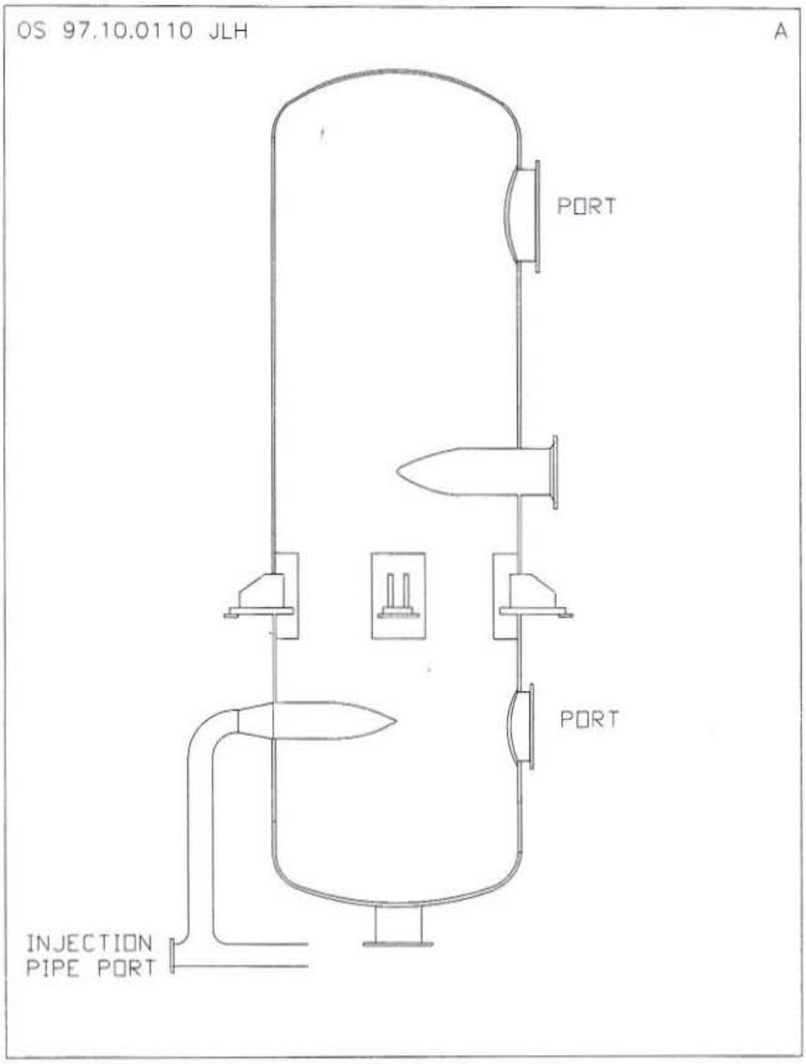
4.1 Gathering system

The layout of the Berlín geothermal steam gathering system and re-injection system is shown in Figure 12. The design is based on a centralized separator station with long two phase flow pipelines, separator of the BOC type with spiral inlet, steam supply pipelines, mist eliminators before the steam enters the turbine, and re-injection pipelines for the waste water.

Production wells. A set of 10 production wells are to be connected to the separation station by means of two-phase pipelines. Four wells are to be connected from the pad of well TR-5, two from the pad of well TR-4 and two from PEBL-1. The total mass production for each well has been assumed to be 57.5 kg/s, with a steam mass fraction of 28.7%.

Steam separators. A set of four bottom outlet cyclone separators with a mist eliminator at the top will be used to separate the steam from the geothermal fluid. The separators will be sized to minimize the carry-over of geothermal water with the geothermal steam. The operation pressure in the separators will be 11 bar-a. Over-pressure protection will be provided by safety valves with bursting discs. Water dump valves will be installed to discharge geothermal water to a holding pond in the event of a high water level in the water vessel.

Steam pipeline. Two pipelines, 711 mm in diameter and 600 m long each, will connect the separation station to the power plant (see Table 3).



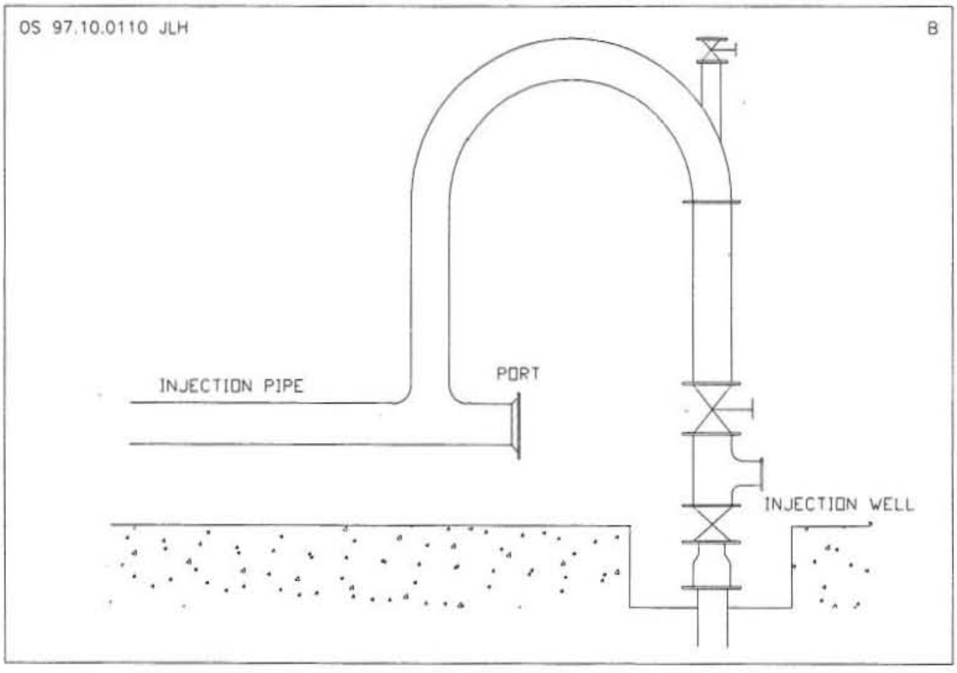


FIGURE 11: The Berlín geothermal steam gathering system; a) Ports design; b) Location

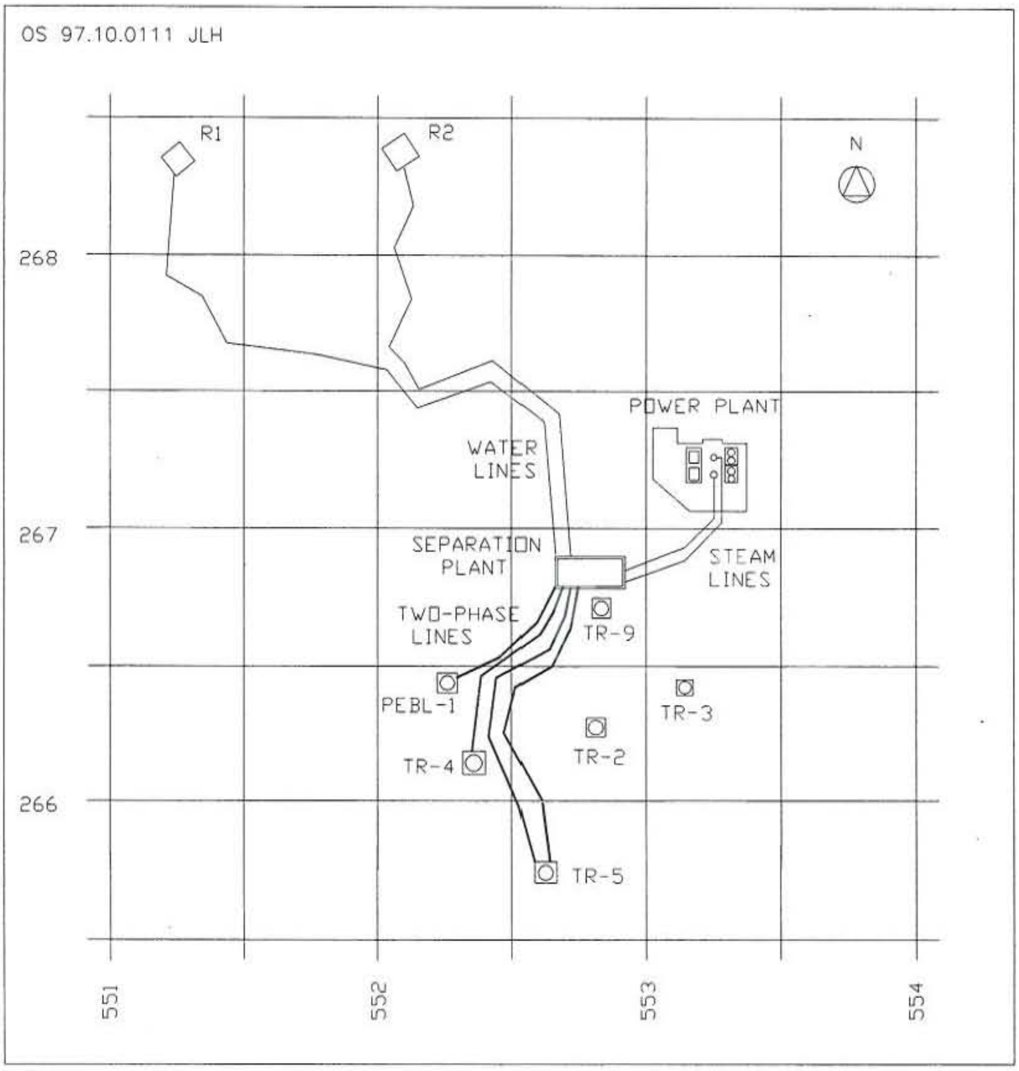


FIGURE 12: The steam gathering system for Berlín geothermal field

TABLE 3: Summary of pipeline sizing

Line identity	Mass flow (kg/s)	Pipe length (m)	Elev. shift (m)	Pipe diam. (m)	P_{in} (bar-a)	P_{out} (bar-a)
TPTR5a	115	2000	203.6	0.609	14.5	11
TPTR5b	115	2000	203.6	0.609	14.5	11
TPTR4	115	1500	118.1	0.609	14	11.5
TPEBL1	115	600	50.8	0.609	13	11.9
WR1	170	3200	250	0.406	11	34
WR2	170	3200	250	0.406	11	34
SP1	55	2000	10	0.609	10.5	10
SP2	55	600	10	0.609	10.5	10

TPTR5a = Two-phase pipeline from well TR-5 (TR-5 and TR-5a)
TPTR5b = Two-phase pipeline from well TR-5 (TR-5b and TR-5c)
TPTR4 = Two-phase pipeline from well TR-4
TPEBL1 = Two-phase pipeline from well EBL1
WR1 = Injection line 1 WR2 = Injection line 2
SP1 = Steam line 1 SP2 = Steam line 2

Water pipeline. Gravity re-injection is proposed into wells drilled to a depth of 2000 m. These wells are located to the north of the production zone and are approximately 2000-3000 m from the separation station (refer to Figure 12).

The system has been designed for gravity re-injection, in keeping with the principle of design simplicity. Two pipelines, 406 mm in diameter, were considered to connect the separation station to the re-injection wells. Considering the elevation shift between the separation station and the re-injection wells (250 m), and a case where the pipe is completely full of water, the well head pressure at the re-injection well will be in the order of 35 bar-a. No control is intended for the gravity re-injection system as this will be essentially a self-modulating system.

4.2 Power plant

Modular (portable) units present several advantages (Saito, 1993; Saito et al., 1995) suitable for the development stage in a proven area of 50 MWe such as at the Berlín geothermal field. According to the calculations presented above for a single flow, single pressure turbine the maximum power output is in the order of 30 MWe. Units with a capacity at 25-30 MWe will be preferred.

To summarize, the preliminary design parameters for the power plant in Berlín are as follows:

Turbine

Turbine gross output:	Two portable units, single flow, of 30 MWe each;
Inlet pressure range:	8-10 bar-a;
Condenser pressure:	0.1 bar-a;
Steam consumption:	1.85 kg/s per MWe;
Last stage blade length:	560 mm.

Cooling tower

Cooling tower type:	Counter-flow;
L/G ratio considered:	1;
Water flow:	2.0 m^3/s;
Area:	589 m^2 for fill package, 6 m long, and $Ka = 175$;
Power consumption	
Fans:	345 kW;
Pumps:	751 kW.

Condenser

Type:	Direct contact low level jet
Rate:	4.6×10^8 kJ/hr

The site layout shown in Figure 13 for the power plant was evaluated in the areas of constructibility, geology, hydrology and visual and noise impact. The distribution of the equipment (turbine-generator house, warehouse, workshop, cooling tower) is such as will allow adequate free area for future development. The cooling towers have been located for optimum exposure to the prevailing wind direction, while avoiding drift crossing the power plant and switch yard.

To check the selection of the turbine for the Berlín geothermal field against world-wide practice, a survey of existing geothermal power plants in the world was made. The database used was a revision of the Annual Report on Geothermal energy development in Japan, 1997, Japan Geothermal Energy Association (JGEA, 1997). The main conclusion is that units of 30 and 55 MWe output are most common with inlet turbine pressures ranging from 4.5 to 10.5 bar-a, as can be seen in Figure 14.

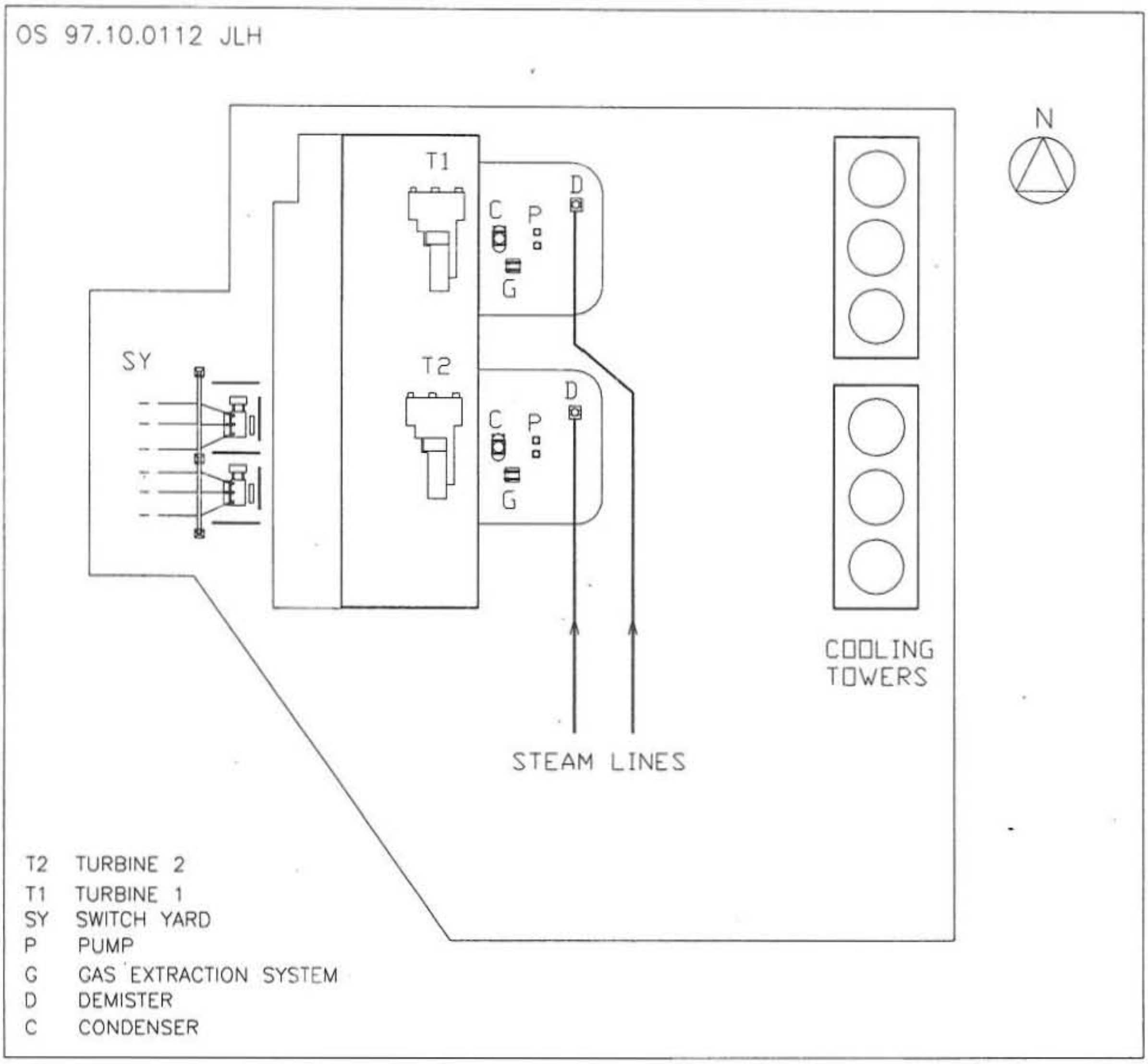


FIGURE 13: Layout of the Berlín power plant site

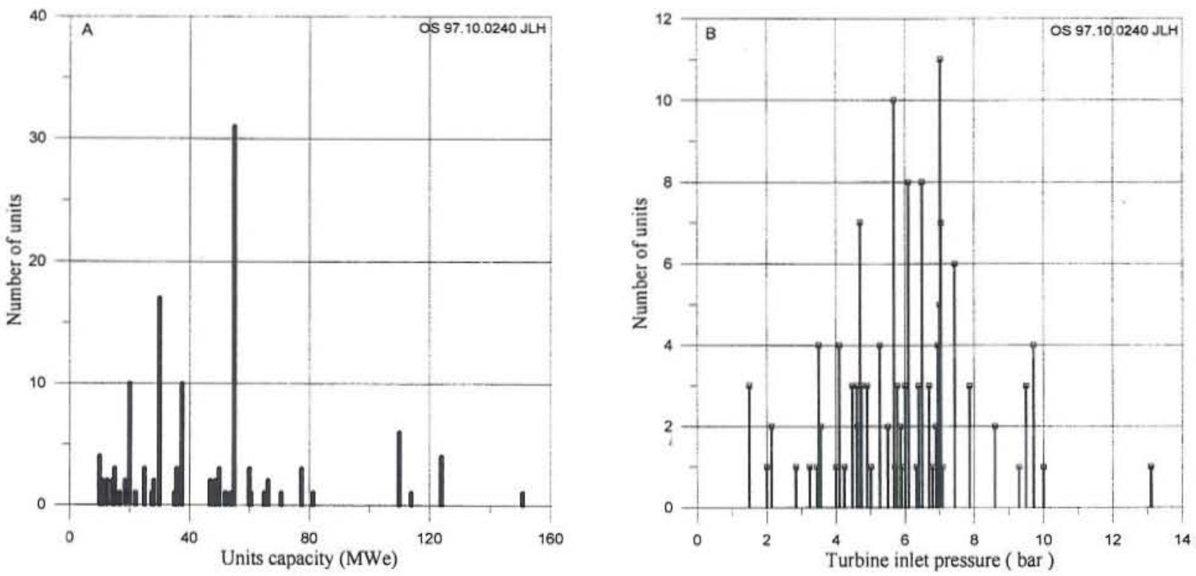


FIGURE 14: Survey of existing geothermal power plants; a) Capacity of units; b) Turbine inlet pressure

5. CONCLUSIONS

The optimum inlet turbine pressure for the Berlín geothermal field was found to be in the range of 7 to 10 bar-a. To avoid silica scaling in the gathering system a single flash power plant is selected. An inlet steam turbine pressure of 10 bar-a is selected and the exhaust pressure 0.1 bar-a. The turbine would have a last blade length of 560 mm, with optimum volumetric flow and velocity at the turbine exhaust being 512 m^3/s and 177 m/s. Portable units with a capacity of 30 MWe are selected for the first condensing development of the Berlín geothermal field.

Bechtel steam purity criteria for geothermal power plants is adopted. The methods to achieve these requirements are presented in the report; steam separator (horizontal or bottom outlet), steam washing, long steam pipelines and the use of wire mesh or mist eliminators.

On-load turbine washing should be used as a last resort to recover lost output due to scaling, to be carried out with great care under carefully controlled conditions. The gathering system has been designed to consider the possibility of operating at low levels of silica over-saturation. The centralized separator station will reduce the residence time of the water in the injection system, as well as provide a reasonable steam pipe length to allow it to function as a separator. Inspection and cleaning ports have been considered in the separator and re-injection lines. The ports in the re-injection lines will allow cleaning by means of a pig or hydro-blasting.

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NOMENCLATURE

a = Contact surface area per unit volume of tower packing (m^2/m^3);
$ACFM$ = Actual air volume flowrate (cfm, cubic feet per minute);
BHP = Pump or fan brake horsepower;
C_{pw} = Specific heat for water (kJ/kg-°C);
e = Pump efficiency (dimensionless);
EFF = Fan mechanical efficiency (dimensionless);
gpm = Cooling water flow (gallon per minute);
H_a = Enthalpy of moist air (kJ/kg);
h_a = Enthalpy of geothermal fluid (kJ/kg);
h_{f2} = Enthalpy of saturated liquid at state point 2 (kJ/kg);
h_{g3} = Enthalpy of saturated steam at state point 3 (kJ/kg);
h_s = Steam enthalpy (kJ/kg);
H_t = Total head on the pump (feet);
H_w = Enthalpy of saturated air at water temperature (kJ/kg);
h_{w1} = Enthalpy of water entering the condenser (kJ/kg);
h_{w2} = Enthalpy of water leaving the condenser (kJ/kg);

h_3-h_4 = Enthalpy drop in the turbine (kJ/kg);
h_1 = Enthalpy of geothermal fluid at state point 1 (kJ/kg);
h_3 = Enthalpy of the steam at state point 3 (kJ/kg);
h_{4s} = Enthalpy of the steam at state point 4s (kJ/kg);
K = Mass transfer coefficient at the interface (kg/s-m^2);
l = Bucket length (inches);
m_s = Steam flowrate (kg/s);
m_t = Total mass flow from the well (kg/s);
MW = Power output of the turbine (MWe);
N = Angular speed (rpm);
P = Pressure (Pa);
r_m = Bucket mean radius (inches);
S = Steam flow (kg/s);
s = Specific gravity of the cooling water, dimensionless;
S_c = Maximal allowance tension stress (MPa);
SP = Static pressure drop in a cooling tower (inches of water);
t_a = Liquid-dominated reservoir temperature (°C);
t_e = Water temperature at the tower exit (°C);
t_i = Water temperature at the tower inlet (°C);
t_w = Water temperature at any point in the cooling tower (°C);
T_1 = Heat source temperature (K);
T_2 = Heat sink temperature (K);
V = Volume of the tower fill (m^3);
v = Saturated dry specific volume (m^3/kg);
V_{an} = Annulus velocity (m/s);
VP = Effective velocity pressure (inches of water);
w = Angular speed (rad/s);
W_1 = Water entering the condenser (kg/s);
W_2 = Water leaving the condenser (kg/s);
x_1 = Steam fraction at state point 1 (%);
Y = Fill height in the cooling tower (m);
y = Percentage of moisture at the expansion line end point;

η_{Carnot} = Carnot cycle efficiency;
η_g = Generator efficiency;
η_i = Isentropic (internal) efficiency;
η' = Cycle efficiency for maximum power output;
ρ_m = Density of bucket material (kg/m^3).

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