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**AN INTRODUCTION TO ELECTRIC ENERGY
CONVERSION SYSTEMS FOR GEOTHERMAL
ENERGY RESOURCES**

By
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An Introduction to
Electric Energy Conversion Systems
For Geothermal Energy Resources

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1. General comments

The use of geothermal energy for electrical power production dates from about 1900, but only recently has this resource been exploited to any significant extent.

While documentation exists [ENEL] which shows that the natural steam fields in the Larderello region of Italy were recognized as early as the 3rd century and that the commercial potential of these mineral-laden waters led to wars between the Tuscan republics during the Middle Ages, it was not until 1904 that Prince Piero Ginori Conti first harnessed the power of the natural steam to produce electricity [Conti]. Conti's system used a reciprocating engine which received steam separated from the water. The engine was of the non-condensing type, exhausted to the atmosphere and produced about 15 kW of electricity from a DC generator to provide lighting for the boric acid factory at Larderello. This original engine was replaced by a turbo-alternator of 250 kW capacity in 1913, marking the beginning of the production of electricity from geothermal energy on a commercial scale.

In order to permit the recovery of boric acid from the endogenous steam and to eliminate the need for the consumption of large amounts of energy for cooling water circulation and gas extraction, the first energy conversion systems used the endogenous steam as a 'fuel' to boil pure water in a secondary power loop. Thus, the direct, corrosive steam did not come in contact with the turbine, but was used instead in vertical-tube heat exchangers. In modern terminology, such early systems might be called "endogenous steam/pure-water binary cycles". Eventually, turbine materials technology improved, troubles with heat exchanger performance arose leading to the development of effective separators, and the binary cycle was abandoned in favor of reaction

turbines of the back-pressure and condensing type fed directly with endogenous steam.

The worldwide use of geothermal energy for electrical power production is growing at an accelerating rate. For the period 1960-1973, the average annual growth rate was about 4%; from 1973-1978 it has been about 14%; and the anticipated rate for the period 1978-1982 is about 21%. At that time it is expected that a total of about 3600 MW of electrical power capacity will be installed in about ten countries. Table I shows the present and projected near-term installed capacity of geothermal-generated electric power for the world. It is clear that sizable additions are expected in Mexico, the Philippines and the United States.

2. Resource utilization and energy conversion systems

With the exception of the Wairakei plant in New Zealand, it has been only within the last five years that any resource other than a dry steam or vapor-dominated one has been exploited for electrical power production. The natural steam at The Geysers, California and at Larderello and Monte Amiata, Italy is a resource that is relatively easy to utilize and which does not strain the capabilities of present energy conversion technology. High-temperature liquid-dominated resources are far more challenging, but hold a great deal of potential since they are estimated to be about twenty times more abundant than dry steam resources [White, et al]. To date, approximately 58% of the installed capacity in the world is associated with dry steam or vapor-dominated geothermal fields, with the remaining 42% obtained from liquid-dominated resources.

The types of energy conversion systems in use may be classified as follows:

Table 1

Summary of World Utilization of
Geothermal Energy for Electric Power Production

Country	Installed Capacity (1978)	Planned Additions (1982)	Total Capacity (1982)
United States	502	1175.55	1677.55
Philippines	169.2 ⁽¹⁾	605	774.2
Italy	420.6	---	420.6
Japan	166	100-500	266-666
New Zealand	202.6	--- ⁽²⁾	202.6
Mexico	78.5	105	183.5
El Salvador	60	35	95
Iceland	33	30	63
Soviet Union	5	58	63
Turkey	0.5	14	14.5
	1637.4	2122.55- 2522.55	3759.95- 4159.95

(1) 4.2 MW on-line; 165 MW expected on-line during 1978.

(2) 150 MW expected by 1984.

Average annual growth rate $\approx$ 20%.

- Direct, dry steam
- Separated steam
- Single-flash steam
- Double-flash steam
- Multi-flash steam
- Brine/Freon binary cycle
- Brine/isobutane binary cycle.

In the subsections which follow, we discuss the thermodynamics of each of these with reference to simplified flow diagrams. Typical existing power plants are identified for each type of system. Full details on all operating geothermal power plants will be found in the reports devoted to each country which produces electricity from geothermal resources.

2.1 Direct, dry steam

Of all types of geothermal resources, the simplest to exploit for electrical power production is the direct, dry steam (DDS) resource. Power plants at The Geysers (California), Larderello and Monte Amiata (Italy) and Matsukawa, Onikobe and Kakkonda (Japan) operate with direct dry steam.

The schematic in fig. 1a is a highly simplified flow diagram for such a system. The sketch shows a condensing turbine with a mechanically induced draft cooling tower. All Italian geopower plants use natural draft towers and some of their plants use noncondensing, exhausting-to-atmosphere turbines.

The energy conversion processes are shown in the temperature-entropy (T-s) diagram in fig. 1b. The expansion process from 1 to 2 takes place irreversibly (i.e. non-isentropically). The turbine power output is expressed as the produce of the mass flow rate of steam $\dot{m}$ and the enthalpy

(a) SCHEMATIC DIAGRAM: HYDROTHERMAL STEAM ENERGY CONVERSION PROCESS

(b) CYCLE DIAGRAM: HYDROTHERMAL STEAM ENERGY CONVERSION PROCESS

Figure 1. SCHEMATIC AND STATE DIAGRAMS FOR THE DIRECT DRY STEAM (DDS) ENERGY CONVERSION PROCESS.

drop through the turbine $\Delta h = h_1 - h_2$; i.e.

$$\dot{W} = \dot{m} (h_1 - h_2) \quad (1)$$

The turbine isentropic efficiency η_t is often used to characterize the expansion process. This is defined as the ratio of the actual power (or work) output to the isentropic power (or work) output; i.e.

$$\eta_t = \frac{\dot{W}}{\dot{W}_s} = \frac{h_1 - h_2}{h_1 - h_{2s}} \quad (2)$$

Thus, the actual turbine output may be written as

$$\dot{W} = \dot{m} \eta_t (h_1 - h_{2s}) \quad (3)$$

where $h_{2s} = h(P_2, s_1)$.

Since the entire expansion process takes place within the two-phase, liquid-vapor region, the turbine efficiency will be lower than the value for dry expansion in the superheated vapor region. If we denote the former value by η_{tw} and the latter by η_{td} (for "wet" and "dry" expansion, respectively), then the relation between these is given by the modified Baumann rule; i.e.

$$\eta_{tw} = \eta_{td} [1 - \alpha (1 - x_{2s})] \quad (4)$$

where α is an adjustable parameter whose value is close to unity ($\alpha = 1.2$ yields reasonable values for η_{tw}) and x_{2s} is the quality the steam would have if it expanded isentropically through the turbine.

For analytic work, η_{td} may be taken as 0.85 (85% isentropic efficiency for dry expansion).

Finally, the specific output of the turbine may be satisfactorily represented by the equation

$$w = \dot{W}/\dot{m} = \eta_{td} [1 - \alpha (1 - x_{2s})] (h_1 - h_{2s}) , \quad (5)$$

or

$$w = (1.02 x_{2s} - 0.17) (h_1 - h_{2s}) , \quad (6)$$

As can be seen from the T-s diagram of fig. 1b, the working fluid does not undergo a cycle in the usual sense of the word. It is admitted to the turbine at 1, condensed at 3 and either evaporated from the stack of the cooling tower or reinjected into the reservoir as liquid from the cooling tower cold well overflow. Thus, the usual definition of the cycle efficiency should not be used to assess the overall performance of the plant. In that definition, the efficiency was given as

$$\eta = \dot{W}_{net} / \dot{Q} \quad (7)$$

where $\dot{W}_{net}$ is the net plant output (turbine output minus pumping work) and $\dot{Q}$ is the rate at which heat is supplied to the working fluid. Since $\dot{Q}$ is produced geothermally and not by the consumption of fuel, its determination becomes ambiguous.

The appropriate measure of plant performance in the thermodynamic sense is the utilization efficiency η_u which compares the plant output to the maximum theoretically obtainable output; i.e.

$$\eta_u = \dot{W}_{net} / \dot{E} = \dot{W}_{net} / \dot{e} , \quad (8)$$

where $\dot{E}$ is called the exergy and is defined as the difference between the availability of the entering working fluid and that of the working fluid at the ambient or sink condition. It can be shown that the exergy per unit mass

is given by

$$e = (h_1 - h_o) - T_o (s_1 - s_o) , \quad (9)$$

where the subscript o is used to denote the sink condition at pressure P_o and temperature T_o .

Power plants such as those at The Geysers have utilization efficiencies in excess of 60%

2.2 Separated steam

It is common in the case of unpumped geothermal wells for the wellhead product to consist of a two-phase mixture of liquid and vapor. The quality of the mixture (i.e., the mass fraction of the vapor phase) depends on the reservoir properties and the wellhead pressure. It is not difficult to separate the phases, either at each wellhead, at centrally located stations or at the power house. Plants which use a single stage of steam separation are called separated steam (SS) plants. Examples of such plants are found at: Cerro Prieto (Mexico), Ahuachapán (El Salvador), Otake and Onuma (Japan) and Pauzhetka (Soviet Union).

In all likelihood, the fluid condition in the reservoir is that of a compressed liquid at elevated temperature. As the fluid comes to the surface under a reducing pressure, it flashes into steam and obtains a wellhead quality ranging from about 20 to 65% for individual wells. From fig. 2, it can be seen that the plant equipment is essentially the same as for the DDS system; the differences include the addition of the separator and a ball float check valve to prevent massive water ingestion by the turbine in the event of a liquid backup in the separator.

The thermodynamic analysis for the SS system is similar to that for the DDS. The mass flow rate through the turbine is equal to the mass flow rate

(a) SCHEMATIC DIAGRAM

(b) CYCLE DIAGRAM

Figure 2. SCHEMATIC AND STATE DIAGRAMS FOR THE SEPARATED STEAM (SS) ENERGY CONVERSION PROCESS.

of geofluid passing from the well times the quality of the geofluid at a;
i.e.,

$$\dot{m}_t = \dot{m}_a x_a \quad (10)$$

Also the exergy of the geofluid must be calculated from the availability of the geofluid at a and include the contribution of the liquid portion even though it is discarded as waste for this plant. Consequently, the utilization efficiency for SS systems is lower than that for either the DDS, or the double- or multi-flash systems to be described in **following sections**. The utilization efficiency of the Cerro Prieto plant, for example, is about 45%.

2.3 Single-flash steam

With reference to fig. 3, the special case when the geofluid emerges at the wellhead as a liquid under pressure (essentially a saturated liquid) leads to what is called a single-flash steam system (SFS). The flashing process, instead of occurring in the wellbore as in the case of the SS system, occurs in surface equipment designed to reduce the fluid pressure to some optimum value. Otherwise, the SFS system is quite similar to the SS system. The exergy in this case should be based on the wellhead fluid conditions at state w.

In most geopower plants which are fed from a number of unpumped wells, only a few may produce saturated (or compressed) liquid; the SFS, as defined in fig. 3, is not in practical use.

2.4 Double-flash steam

The so-called double-flash (or dual-flash) steam system is somewhat of a misnomer in that it more accurately should be called a separated-steam/single-flash system since plants of this type generally operate with a

(a) SCHEMATIC DIAGRAM

(b) CYCLE DIAGRAM

Figure 3. SCHEMATIC AND STATE DIAGRAMS FOR THE SINGLE-FLASH (SF) ENERGY CONVERSION PROCESS.

two-phase geofluid at the wellhead. The term double-flash arises from the fact that two flashes occur, one below the surface (typically in the well-bore) and one above the surface in a specially-designed flash tank. In any event the point of interest is that a two-stage process of steam production and utilization is used, the second stage capturing a portion of the energy which otherwise would be wasted.

Plants of this type are located, for example, at Hatchobaru (Japan) and Krafla (Iceland).

The schematic and T-s diagram for this system may be found in fig. 4. The essential difference between this and a single-stage system is that a dual-pressure (or pass-in) turbine is typically employed. Two individual turbines arranged in a tandem-compound fashion may be used in certain instances. The high-pressure steam from the separator flows through the high-pressure portion of the turbine and produces the following work output per unit mass of fluid entering at 1:

$$w_{HP} = x_1 (h_2 - h_{2f}) , \quad (11)$$

where x_1 is the quality of the wellhead fluid at the separator,,

$$x_1 = (h_1 - h_3)/(h_2 - h_3) , \quad (12)$$

and h_{2f} is the fluid enthalpy at the exhaust section of the HP turbine.

The flasher produces additional steam from the liquid fraction of the separation process by an additional pressure drop. This steam is mixed with the steam from the high-pressure turbine at the pass-in section of the turbine, and the augmented steam produces work output in the low-pressure turbine per unit mass at 1:

$$w_{LP} = [x_1 + (1-x_1) x_a] (h_i - h_6) , \quad (13)$$

(a) SCHEMATIC DIAGRAM

(b) CYCLE DIAGRAM

Figure 4. SCHEMATIC AND STATE DIAGRAMS FOR THE SEPARATED-STEAM/SINGLE FLASH (SSSF) ENERGY CONVERSION PROCESS.

where

$$x_a = (h_3 - h_5)/(h_4 - h_5) \quad , \quad (14)$$

and h_i is the enthalpy of the mixed steam at the pass-in section of the low-pressure turbine:

$$h_i = [x_1 h_2 + (1-x_1)x_a h_4]/[x_1 + (1-x_1)x_a] \quad . \quad (15)$$

The special case where the geofluid at the wellhead is saturated (or compressed) liquid is shown in fig. 5. This schematic illustrates the tandem-compound turbine arrangement wherein two separate cylinders are used for the high- and low-pressure sections both exhausting to a common condenser. For this case, the term double-flash is accurate and appropriate.

2.5 Multi-flash steam

A multi-flash plant uses more than two levels of steam pressure at turbine entry points. There is only one example of this type in existence, namely the power plant at Wairakei, New Zealand. As is explained in more detail elsewhere (DiPippo, 1978(C)), this particular arrangement resulted from design considerations related not solely to the geothermal power station, but from requirements for an auxiliary plant that was to have used a portion of the steam from the power plant for industrial purposes.

2.6 Brine/fluorocarbon or brine/hydrocarbon binary cycle

These systems are usually called simply binary cycles because a secondary working fluid (a fluorocarbon or hydrocarbon) is used in a main Rankine cycle with the geofluid merely providing the required thermal energy to vaporize and superheat the secondary working fluid.

Figure 5. SCHEMATIC AND STATE DIAGRAMS FOR THE DOUBLE-FLASH (DF) ENERGY CONVERSION PROCESS.

A schematic and a cycle diagram in pressure-enthalpy coordinates are shown in fig. 6, for the case of isobutane as the secondary fluid. The case shown is for a supercritical heat exchange process from 6 to 1.

There are only two plants in the world operating on this principle and both are small. The Soviet Union has one at Paratun'ka on the Kamchatka peninsula and one exists at Otake, Japan. The Japanese pilot plant is rated at 1 MW while the Soviet unit is about half that size. Operating experience is meager. Several plants of this type are planned for sites in the Imperial Valley, CA. (See DiPippo, 1978(A) and 1978(B).)

A number of advantages are claimed for a binary cycle when compared to a flash steam system [Anderson]. These include:

- more suited to low-temperature hydrothermal resources
- smaller turbine size for given output
- less expensive turbine for given output
- high-pressure operation throughout, eliminating vacuum operating problems of air in-leakage, etc.
- non-corrosive working fluid in turbine
- higher isentropic turbine efficiencies
- lower turbine rotor inertia giving lower maximum coupling torque caused by generator overloads
- reduced stress and vibration problems with turbine blades because of lower tip speeds
- condensing temperatures can be lower giving better cycle efficiency
- less cavitation damage in boiler feed pumps
- completely dry expansion eliminating erosion problems.

Some of the disadvantages must also be cited and include:

Figure 6. SCHEMATIC AND STATE DIAGRAMS FOR THE BINARY ENERGY CONVERSION PROCESS.

suitable secondary working fluids are expensive
 no leaks can be permitted
 heat exchangers are major elements and are costly
 huge brine flow rates are needed for a reasonable size plant,
 leading to disposal problems
 flammability is a problem with hydrocarbon working fluids.

The thermodynamics of binary systems is not difficult. If $\dot{m}_b$ is the mass flow rate of brine and $\dot{m}$ is that of the secondary fluid, then with reference to fig. 6, the turbine output is

$$\dot{W}_t = \dot{m} (h_1 - h_2) , \quad (16)$$

the secondary fluid feed pump power requirement is

$$\dot{W}_p = \dot{m} (h_6 - h_4) , \quad (17)$$

giving a net plant output of

$$\dot{W}_{net} = \dot{m} (h_1 - h_2 - h_6 + h_4) . \quad (18)$$

If the brine requires pumping for any reason, then that power must be deducted from $\dot{W}_{net}$ in eq. (18) to obtain the true plant net power.

The brine flow is determined from an analysis of the primary heat exchanger:

$$\dot{m}_b (h_a - h_b) = \dot{m} (h_1 - h_6) . \quad (19)$$

The waste heat from the plant is given by

$$\dot{Q}_o = \dot{m} (h_2 - h_4) . \quad (20)$$

The cooling tower must be designed to accommodate this heat load.

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