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A WASTE-HEAT REJECTION SYSTEM FOR THE TOTAL-FLOW TURBINE CONCEPT
USING HOT GEOTHERMAL BRINES

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December 18, 1974

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Prepared for U. S. Atomic Energy Commission under contract no. W-7405-Eng-48

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A WASTE-HEAT REJECTION SYSTEM FOR THE TOTAL-FLOW TURBINE CONCEPT USING HOT GEOTHERMAL BRINES

ABSTRACT

A waste-heat rejection system, for use in a total fluid flow method proposed for producing electrical power from high-saline geothermal brines, is described. Because of the unique conditions imposed by the total flow concept, major design modification will be necessary in conventional heat rejection equipment if it is to be used. These changes are technically feasible without adding significantly to either total capital or operating costs. Design parameters, equipment choice, and costs are discussed for a waste-heat rejection system for a plant module size of 220 MW.

INTRODUCTION

A total fluid flow concept has been proposed as a method for producing electrical power from high-saline geothermal brines by Austin et al.¹ of Lawrence Livermore Laboratory. This method would increase geothermal resource utilization by converting the thermal energy in both the liquid and vapor phases into kinetic energy. (In conventional flashed steam systems, the liquid phase is exhausted at high temperatures.) In the total fluid flow method, the liquid-vapor mixture will be expanded through a converging-diverging nozzle. The resulting high-velocity output stream will drive a modified hydraulic impulse turbine operating at 11.7 kPa (1.7 psia) Hg and 322 K (120°F).

An integral part of any steam-turbine electrical generation plant is the waste heat rejection system. Application of the total fluid flow method will introduce major differences in the design of this system. We consider here some of the design parameters for this waste-heat rejection system. Equipment choice, size, and costs are discussed.

Austin et al.¹ developed their concept for the high-salinity brines of the Salton Sea geothermal area in California's Imperial Valley, although it is applicable to any water-dominated geothermal system. Salton Sea brines contain up to 30% dissolved solids at depths greater than 914 m (3000 ft) and at temperatures of approximately 573 K (572°F).²

HEAT REJECTION SYSTEMS

In conventional steam-turbine electrical generation plants, subatmospheric exhaust pressures are maintained by steam condensation. Latent heat is rejected downstream of the turbines into the environment (either into the lakes or rivers in once-through

systems or into the atmosphere by cooling towers). Geothermal power plants (e.g., The Geysers in Northern California) reject heat at a rate of 4.5 to 5 kW into the environment for every kilowatt of power generation.³ This compares with about 2 kW and 1.4 kW per kilowatt of power generation for nuclear and fossil fuel plants, respectively.^{4,5} The only other major difference between heat rejection systems in geothermal power plants and systems in nuclear or fossil fuel plants is that in the former there is a net accumulation of water rather than a deficit. This surplus water could be useful in arid areas.

Application of the total flow method using high saline brines will require major design modifications in the waste-heat rejection systems currently in use for geothermal systems. By virtue of the total flow utilization, vapor-liquid separation occurs at subatmospheric pressure after the brine mixture passes through the turbine. This pressure is 11.7 kPa (~1.7 psia) compared to 0.7 MPa (~100 psi) in flashed steam systems. Separation equipment downstream must, therefore, be designed to handle extremely high volumetric flow rates with little pressure drop available. Other design differences include higher operating salt concentrations in the condenser and cooling towers and higher noncondensable gas content.

In addressing these design problems, the following assumptions and simplifications were made:

- Thermodynamic properties of pure water were used for the brine, even though the salt content of the Salton Sea brines is about 25 wt%.
- The effects of high saline brine on choice and longevity of materials and equipment are not addressed, except that the brine is assumed to be extremely corrosive and to produce unacceptable scaling. Therefore, the design minimizes contact between the brine and equipment surfaces.
- Turbine operating back pressures greater than 11.7 kPa (1.7 psia) are not acceptable due to severe loss in operating efficiency.
- Equipment design is based on a dry bulb temperature (dbt) of 311 K (100°F) and a wet bulb temperature (wbt) of 302 K (83°F). (These are estimated 5% level temperatures based on data compiled by the Imperial Valley Conservation Research Center⁶ between 1967 and 1973.)
- A plant module size of 220 MW is assumed, supplied by 10 producing wells driving 10 turbine systems (22 MW/turbine).

GENERAL PROCESS

The process of removing waste heat and, thus, maintaining turbine back pressure is as follows. The high-velocity vapor-liquid turbine exhaust mixture is separated in large, low-resistance cyclone separators. To achieve an acceptable pressure drop, the velocity of the turbine effluent is reduced by enlarging the flow area with a parallel network of conduits. Each conduit feeds a separator. The vapor streams from each

separator are collected and condensed by direct contact with cooling water. Non-condensable gases are compressed and ejected up a stack. Combined condensate and cooling water are pumped to cooling towers, where heat is rejected into the atmosphere. Additional water is required for blowdown due to the high salt content of the brines and the high dbt in the Imperial Valley. This differs from other geothermal plants (e.g., The Geysers) where there is sufficient overflow of water from the cooling towers to maintain a reasonable operating salinity in the towers.

Because in the total flow concept the separation of the vapor-liquid brine occurs after it passes through the turbines, the separators must accept and efficiently separate huge volumes of this low-density mixture without an appreciable pressure drop. Assuming a turbine operating back pressure of 11.7 kPa (1.7 psia) at 322 K (120°F) and that the condenser-cooling tower system has a cooling range of 11.1 K (20°F) with a 5.5 K (10°F) approach to wbt, then the condenser terminal temperature difference is 3.9 K (7°F). The corresponding condenser pressure is 9.6 kPa (1.39 psia). Therefore, the total pressure drop between the turbine exhaust and condenser cannot exceed 2.1 kPa (0.3 psia) without exerting additional constraints on the already massive, recirculating condenser-cooling water system.

DESIGN VARIABLES

Turbine Exhaust Stream

The turbine exhaust conditions are taken from Austin et al.¹ In this design (shown in Fig. 1), a turbine with an 1.81-m^2 (19.5-ft^2) exhaust area drives a 22-MW generator.

Separators

To effectively condense the steam, a vapor-liquid separation is required. Condensation of steam without separation from the liquid brine in a direct contact container would require huge quantities of circulating cooling water (assuming cooled brine could not be the circulating coolant for obvious reasons). An indirect contact condenser (assuming no scaling) would still require a fresh water supply to replenish cooling tower losses. These water losses are significant. Vapor-liquid separation would eliminate scaling problems in the condenser and beyond and would reduce the amount of outside water required.

Inertial-type separators are inexpensive and relatively simple in design and should work efficiently for this system. However, at the high exhaust velocity of the turbine, the pressure drop (ΔP) across the separator would be prohibitive. For inertial-type cyclone separators, the relationship between the pressure drop and inlet stream

Fig. 1. Turbine exhaust conditions.

parameters can be estimated from design equations for various configurations. For a cylindrical cyclone separator with rectangular inlet and conical tail, this relationship is⁷:

$$\Delta P = (10 - 16)\rho v^2,$$

where ρ is the density and v is the inlet velocity.

A pressure drop of 1.38 kPa (0.2 psi) across cyclone separators corresponds to an inlet velocity (turbine exhaust velocity) of about 30.5 m/s (100 ft/sec). This drop is acceptable but requires that the turbine exhaust velocity be reduced tenfold in a diffuser section. The exhaust area of the diffuser and also the inlet area of the separators would be 18.1 m² (195 ft²). To physically provide this area, a number of cyclones operating in parallel would be necessary, as depicted in Fig. 2. Operating conditions are given in the figure.

The amount of salt carryover will depend on the separation efficiency between the vapor and liquid brine, initially 25 wt% salt. A vapor purity of 99.9% is attainable without special provisions. This gives 16.7 g (0.0367 lb) of salt per second carryover or a salt concentration in the vapor stream of 396 ppm. Thus, 99.94% of the salt remains in the separated liquid fraction which can be further processed if mineral recovery is viable or reinjected back into the formation. Beyond the separators, the process is relatively salt free.

Pressure drop in the conduit leading to and from the cyclone manifolds is governed mainly by losses in the upstream conduit where the turbine exhaust velocity is high. At these turbine exhaust superficial velocities [0.29 km/s (965 ft/sec) vapor and 40 mm/s (0.13 ft/sec) liquid], annular mist flow is assumed to prevail.⁷ At these conditions, the pressure drop for two-phase flow is about $2 \times \Delta P$ for isothermal flow of the pure vapor only,⁸ or about 226 Pa/m (0.01 psi/ft). This assumes the turbine exhaust area of 1.81 m² (19.5 ft²) is contained in a 1.52-m (5-ft)-diam conduit. Thus, the plumbing following the turbine exhaust and separators should be limited to approximately 3.05 m (10 ft) or equivalent so as not to exceed 0.69 kPa (0.1-psi) pressure drop. This pressure drop combined with a 1.38 kPa (0.2 psi) drop in the separators would be compatible with condenser operation.

Condenser

Since the separated vapor phase is relatively salt free, it can be mixed with the circulating cooling water in a direct-type condenser. Either barometric or low-level condensers can be used. Cooling water directly contacts the separated vapor of the turbine exhaust through use of distribution nozzles and cascading trays. Because heat and mass transfer rates are high in direct contact equipment, internal transfer surfaces are minimized, thus reducing both capital and maintenance costs. The condensate and cooling water are pumped directly to the cooling towers.

Fig. 2. Vapor-liquid separation scheme.

The noncondensable gases, assumed to be mainly CO_2 , are removed during the condensation stage of the process. Helgeson² reports that the fugacity of CO_2 in Salton Sea brine can be as high as 2.53 MPa (25 atm) (the estimate for the No. 1 State well). Using 2.53 MPa (25 atm) and data for CO_2 solubility in 2M NaCl at 573 K (300°C) from Ellis and Golding,⁹ one obtains 0.0032 mole fraction or 7800 ppm CO_2 in Salton Sea brine. This concentration is conservatively high for two reasons. First, it is based on the No. 1 State well which is an order of magnitude higher in CO_2 fugacity than other wells (for example, the No. 2 IID²). Secondly, CO_2 concentration is calculated from 2M NaCl data; the greater salinity of the brine would tend to suppress CO_2 solubility. Using the conservative figures, the partial pressure of CO_2 in the condenser, assuming all dissolved CO_2 is evolved in the process, is

$$P_{\text{CO}_2} = \frac{(0.0032)(9.6)}{0.37},$$

where

0.0032 = moles CO_2 /mole brine,⁹

0.37 = steam quality,

9.6 kPa (1.39 psia) = condenser operating pressure.

This results in a CO_2 pressure in the condenser of 83 Pa (0.012 psia), which is 2 wt% CO_2 in the vapor fraction stream of the separators.

For a two-stage steam ejector [9.6 → 34 → 103 kPa (1.4 → 5 → 15 psia)] with interstage condensation, approximately 3 kg steam at 0.7 MPa (100 psi) is required for each kilogram of CO_2 compressed. If the wellhead product can be separately flashed to give 0.29 kg of steam at 0.7 MPa/kg brine, then 6.2% (or potentially 1.36 MW) of well production would be needed to operate the steam ejector. Although steam ejectors are relatively maintenance free, they are inefficient.

The steam requirements calculated for the two-stage steam ejector above assume pure CO_2 in the suction section of the ejector. In actual operation, water vapor in the condenser would be unavoidably intermixed with the ejected CO_2 . The amount of motive steam required per kilogram of CO_2 must in practice be higher. Assuming that the condenser operates ideally at 9.6 kPa (1.39 psia) total pressure (partial pressure CO_2 and H_2O) and that equilibrium can be established at the cold water inlet temperature of 307 K (93°F) [where the vapor pressure of water is 5.3 kPa (0.767 psia)], then the $\text{H}_2\text{O}/\text{CO}_2$ mass ratio in the ejected mixture before combination with the motive steam is 0.5. With water intermixed with CO_2 in this ratio, the motive steam requirements increase to 5 kg steam/kg CO_2 , which would be about 10% of well production (2 MW). Equilibrium at the cold water inlet temperature is possible in principle only. Operating conditions will be at least a few degrees above this, in which case even greater ejection capacity would be required due to the higher partial pressure of water. Lower wbt will offset this 95% of the time, which will allow a wider cooling range and hence lower cold water operating temperatures. Clearly, the most economical operating point must be established taking into account weather patterns, power usage, and the precise costs of steam, cooling water, and equipment.

It is probably more efficient to use a pump compressor for removing the non-condensable gases in the condenser. A two-stage compressor with interstage cooling would require about 0.3 MW power to remove the $\text{CO}_2\text{-H}_2\text{O}$ ejector mixture described above. Mechanical compression of the noncondensable gases would use considerably less power, but maintenance costs would be higher. Overall, a pump compressor ejection system would probably be superior, unless motive steam were available at lower cost.

Cooling Tower

The steam condensate and cooling water leave the condenser at 318 K (113°F) and 9.6 kPa (1.39 psia). Turbine operating conditions and the local weather conditions constrain the operating characteristics of the cooling tower. A 5.5 K (10°F) approach to wet bulb temperature is standard design. A closer approach requires longer liquid-vapor contact which adds to capital and operating costs. With a 5% seasonal level wbt of 302 K (83°F), a 11.1 K (20°F) cooling range is available. Steam mass flow is 41.9 kg/s (92.5 lb/sec), of which a minimum of 0.5 kg is ejected for each kilogram of CO_2 . Since CO_2 is about 2 wt%, the amount of ejected steam from the condenser is at least 0.34 kg/s (0.75 lb/sec). Therefore, assuming 41.7 kg (91.75 lb) of steam/s at 318 K (113°F) is condensed, the cooling water circulation rate, m , is:

$$m = \frac{41.7 h_1}{C_p \Delta T} = 2.15 \text{ Mg/s (4720 lb/sec)},$$

where

h_1 = latent heat per kg,

C_p = 4.2 kJ/kg·K (1 Btu/lb·°F),

T = 11.1 K (20°F) range.

The cooling tower heat load is:

$$41.7 [h_{\text{steam at 318 K}} - h_{\text{H}_2\text{O at 307 K}}] = 102 \text{ MJ/s (96,250 Btu/sec)},$$

where 41.7 is the steam mass flow rate. Assuming air enters the tower at 311 K (100°F) dry bulb and 50% relative humidity and exits at 316.7 K (110°F) and 90% relative humidity, approximately 0.91 kg of H_2O is evaporated per kilogram of condensate produced. This ratio is relatively large due to the high ambient temperatures in the Imperial Valley. The corresponding minimum air circulation rate through the tower is:

$$\frac{(41.7)(0.91)}{0.032} = 1.18 \text{ Mg (2600 lb) of air/s} = 1.1 \times 10^3 \text{ m}^3/\text{s} (2.34 \times 10^6 \text{ cfm}),$$

where

- 41.7 = steam mass flow rate, kg/s,
- 0.91 = kg H₂O evaporated/kg condensate,
- 0.032 = humidity difference between exit and inlet air, kg H₂O/kg air.

The flow streams for the heat rejection equipment are summarized in the heat and mass flow diagram, shown in Fig. 3.

Due to entrainment in the separators, 16.7 g (0.037 lb) of salt/s is carried over into the condenser and eventually into the cooling tower, which operates at some practical saline level. The higher the salinity, the smaller the volume of blowdown. Assuming 1000 ppm salt is tolerable, then 13 kg/s (28.5 lb/sec) of fresh make up water is required. Of this, 12.1 kg/s (26.7 lb/sec) is for blowdown and the remainder is for the slight deficit from combined drift and evaporation.

EQUIPMENT SIZE AND COST - 22-MW UNITS

Separators

For turbine flow rates of 113.6 kg/s (250 lb/sec), the exhaust area of the conduit needs to be enlarged to reduce the stream velocity to acceptable limits for vapor-liquid separation in inertial-type separators. A bank of 10 cyclone separators with combined flow area of 18.1 m² (195 ft²) and intake velocity of 29.4 m/s (96.5 ft/sec) would provide the necessary capacity. The approximate dimensions of a single cyclone separator would be 4 m (13 ft) diam by 16 m (52 ft) long with an inlet port of 0.9 m (3 ft) by 2 m (6.5 ft). Vessel wall thickness would be 25 mm (~1 in.), capable of withstanding 0.14 MPa (20 psi) external pressure. Each cyclone would weigh 28 Mg (31 tons). Estimated cost for a bank of 10 cyclones is \$200,000. The separators would be preceded by a large distribution manifold. Vapor exhaust streams would be combined and channeled to the condenser. This extensive piping network is estimated to cost \$50,000.

Condenser

A direct contact, single-shell barometric (or low-level) condenser for 41.7 kg (91.75 lb) of steam would contain distribution nozzles, internal baffling, and ejector pump-compressor for noncondensable gases - mainly CO₂. The condenser would be on the order of 4.6 m (15 ft) diam by 12.2 m (40 ft) high. For low-level condensers, lack of structural supports for elevating the condenser would result in some cost savings. However, additional controls would be required to assure that water does not enter the turbine.

Fig. 3. Heat and mass flow diagram for the heat rejection equipment. Symbols are: V, vapor; G, noncondensable gas; h, enthalpy; T, temperature; P, pressure; L, liquid; CW, cold water; and RH, relative humidity.

The cost of the condenser and auxiliaries (excluding liquid circulation pumps) is estimated at \$250,000. This estimate is based on 1970 cost data for fossil fuel plants⁴ and a 3.6 geothermal/fossil fuel heat rejection factor. A 7% per year adjustment was made for inflation from 1970 to 1974.

Cooling Tower

For the cooling tower operating conditions shown in Fig. 3, approximately $1.1 \times 10^3 \text{ m}^2$ (11,500 ft²) of tower area is required. In practice, this capacity would be divided into three cells, each with separate induction fans, but with common water distribution system, pumps, water well, structural members, etc. Using cost data for fossil fuel plants⁴ and again a 3.6 geothermal fossil-fuel heat rejection factor, the estimated cooling tower cost for a 22-MW power generating plant is \$225,000 excluding pump costs. Pump and conduit costs for the condenser and cooling tower are estimated to be \$210,000. Two $1.07\text{-m}^3/\text{s}$ (17,000-gpm) liquid pumps would be required for lifting water to the cooling towers. Barometric condensers could require two additional pumps to take water from the cooling towers up to the condenser. Each $1.07\text{-m}^3/\text{s}$ (17,000-gpm) pump draws 330 kW. Each induction fan draws 85 kW.

Spray Pond

Use of spray ponds for cooling requires substantial land usage. To meet heat load requirements for a 22-MW plant, a pond, 150-m (500 ft) wide by 30-m (100 ft) deep, would be required, assuming 8 km/h (5 mph) winds. An extensive network of approximately 80 distribution nozzles producing water mist 4.6 m (15 ft) above the pond would be required. Variable wind velocity and direction would reduce the efficiency of the spray pond. In design of power plants which must operate steadily and continuously, dependence on environmental factors such as wind conditions would probably be unacceptable.

ECONOMICS

Capital and Operating Costs

Table 1 summarizes estimates of major capital expenditures for heat rejection equipment. Capital costs for all equipment are taken from Austin et al.,¹ except for the cooling system which was increased from his estimate of \$3.84 million to \$10 million. Annual operating costs for the entire plants are increased slightly to reflect this capital cost increase of \$6.16 million. The cost of an external fresh water supply,

Table 1. Major equipment capital investment required for the heat rejection system of a 220-MW geothermal power plant utilizing the total flow concept (all figures in 1974 dollars).

Equipment	Cost
Turbine exhaust distribution manifold and piping	500,000
Separators	2,000,000
Condensers and ejector pumps	2,500,000
Cooling towers and basin	2,300,000
Circulation pumps and piping	2,100,000
Total	\$9,400,000 or \$10 million per 220-MW power plant

approximately 200 m³/d (50,000 gpd) during peak summer days, is not expected to be significant. Substituting these figures into the estimate made by Austin et al., total capital costs are estimated at \$45.1 million and annual operating costs are estimated at \$3.62 million.

Cash Flow and Profitability

The internal rate of return vs power sales price is shown in Fig. 4. An 80% load factor, 12.5% royalty, no depletion allowance, 48% federal income tax, and straight line depreciation for 20 yr was assumed. The 80% load factor is conservative enough to absorb approximately 5% of the design power generation needed for operating pumps, fans, and plant auxiliaries. The rate of return is approximately two percentage points less than the estimate of Austin et al., whose heat rejection capital equipment costs were 62% lower than the estimate given here. The effect of power sales price on rate of return is approximately the same in both estimates.

CONCLUSIONS

It is clear that the rate of return is strongly affected by power sales price. Higher turbine back pressures due to flow restrictions downstream will severely reduce turbine efficiency, which will be reflected in much lower power sales. On the other hand, adequate turbine back pressures can be maintained by a relatively small capital expenditure (almost insignificant) for additional downstream plumbing and equipment.

Fig. 4. Internal rate of return for a 220-MW power plant. The solid curve shows the effects of the higher capital costs estimate for the heat rejection system. The dashed curve, taken from Austin et al.,¹ is shown for comparison.

Rejecting the waste heat using conventional equipment is technically feasible. The estimated capital costs for the waste-heat rejection system are approximately 22% of the total plant capital costs.¹

An external source of fresh water for cooling tower blowdown is necessary due to the high salinity of the brines and the high ambient temperatures in the Imperial Valley. In the nonpeak summer season, this demand is reduced considerably.

The problems of corrosion and scaling are common to the entire system — from the well head through the nozzles and turbines, and to the cooling system. The solutions to these problems will undoubtedly affect system design, particularly materials selection and process stream handling. However, the basic heat rejection concept and type of equipment used will not differ significantly.

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