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WASTE HEAT REJECTION FROM GEOTHERMAL POWER STATIONS*

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ABSTRACT

Waste heat rejection systems for geothermal power stations have a significantly greater influence on plant operating performances and costs than do corresponding systems in fossil- and nuclear-fueled stations. With thermal efficiencies of only about 10%, geothermal power cycles can reject four times as much heat per kilowatt of output. Geothermal sites in the United States tend to be in water-short areas that could require use of more expensive wet/dry or dry-type cooling towers. With relatively low-temperature heat sources, the cycle economics are more sensitive to diurnal and seasonal variations in sink temperatures. Factors such as the necessity for hydrogen sulfide scrubbers in off-gas systems or the need to treat cooling tower blowdown before reinjection can add to the cost and complexity of geothermal waste heat rejection systems.

Working fluids most commonly considered for geothermal cycles are water, ammonia, Freon-22, isobutane, and isopentane. Both low-level and barometric-leg direct-contact condensers are used, and reinforced concrete has been proposed for condenser vessels. Multipass surface condensers also have wide application. Corrosion problems at some locations have led to increased interest in titanium tubing. Studies at ORNL indicate that fluted vertical tubes can enhance condensing film coefficients by factors of 4 to 7.

Once-through cooling of geothermal power plants is not likely, and cooling lakes and ponds will probably have limited application. Spray ponds and canals can be considered, but cooling towers will more than likely find the widest use. These will be mechanical-draft types because natural-draft towers do not function well in areas of high dry-bulb temperatures and low relative humidity. Most U.S. geothermal sites are in areas where maximum system electric loads occur in the summer months when tower cooling capacity is restricted and water supplies are more scarce. Although capital costs can be significantly higher, shortage of water will undoubtedly lead to increased use of wet/dry and dry-type cooling towers. Wet/dry towers are probably best arranged with the air flow in parallel through the wet and dry sections but with the water flow in series and entering the dry section first. Deluge cooling of dry-type towers to meet peak loads has sufficient merit to warrant more study and development. Phased cooling, whereby storage capacity is provided for warmed circulating water until nighttime conditions are more favorable for heat rejection, may have application at some locations.

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DISCLAIMER

INTRODUCTION

The waste heat rejection system for a geothermal power station is of much greater importance than are the corresponding systems in nuclear and fossil fueled power plants.

Fossil-fueled plants presently waste about 60% of their heat input, nuclear-fueled plants about 70%, and geothermal power plants, because of the low thermal efficiency inherent to relatively low-temperature heat sources, 85% or more. Figure 1 shows the ratio between the amount of heat rejected from a power cycle at various thermal efficiencies and the amount rejected from a power cycle having an efficiency of 35%. The amount of heat rejected from a geothermal station per kilowatt of output will thus be three three to six times greater than that of a nuclear plant. Because geothermal stations will tend to be smaller in size, equipment costs per kilowatt will be greater. The auxiliary power requirements, such as pumping, will tend to be proportionately higher. If dry or wet/dry cooling towers are required, the cost of the waste heat rejection system can be substantially higher. It is too early to tell how much the use of binary cycles will add to the cost of geothermal stations, but this cost may be significant. The presence of hydrogen sulfide in the off-gases can require more expensive materials to combat corrosion and may necessitate off-gas scrubbers in the waste heat rejection system. Noncondensable gases may be higher than in conventional systems, affecting heat transfer area requirements and gas-handling costs.

Because of the relatively large amounts of cooling water needed per kilowatt generated, geothermal power plants are particularly sensitive to the adequacy of the water supply. Unlike conventional stations, geothermal plants must be located where the energy is found, and at the present time, the greatest U.S. potential for geothermal energy is in the relatively hot, water-short areas of the Imperial Valley of Southern California. This problem can exist even if condensate from a flashed-steam geothermal cycle is used for cooling tower makeup because an equivalent amount of water may be needed from some other source to be reinjected into the ground to prevent subsidence. It should be noted that economic

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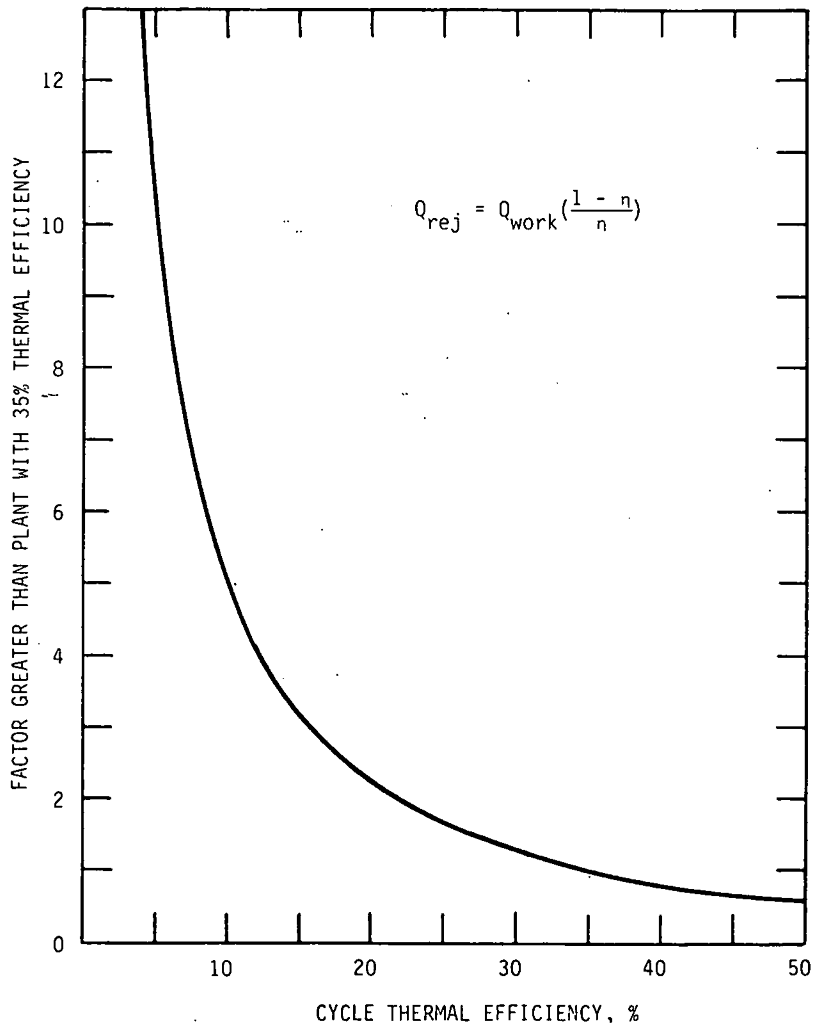

Fig. 1 . Cycle heat rejection vs thermal efficiency compared to a cycle having an efficiency of 35%.

operation of small-scale, demonstration geothermal power plants with sufficient water available for cooling tower makeup is misleading if amounts of makeup water needed for a large power station at the same location are not available.

The heat to be dissipated from a geothermal power cycle consists almost entirely of the heat of condensation of the turbine exhaust vapor. The condensers will be either direct-contact or surface types. The circulated coolant will in most cases be water, which will give up its heat in a spray pond or cooling tower. The towers will be either wet, dry, or a combination of the two, and, although natural-draft towers may be considered for larger stations, the flow of air will probably be induced by fans. Direct condensation of the steam in air-cooled coils may be feasible for some installations. Simple schematic flow diagrams of typical possible heat rejection system arrangements for a flashed-steam cycle with surface condenser and wet mechanical-draft cooling tower are shown in Fig. 2. Figure 3 exhibits binary cycles using wet mechanical-draft towers, and Fig. 4 shows the Heller-type cycle using a direct-contact condenser in conjunction with air-cooled coils. Figure 5 exemplifies the air-cooled coil with direct condensation of the exhaust steam.

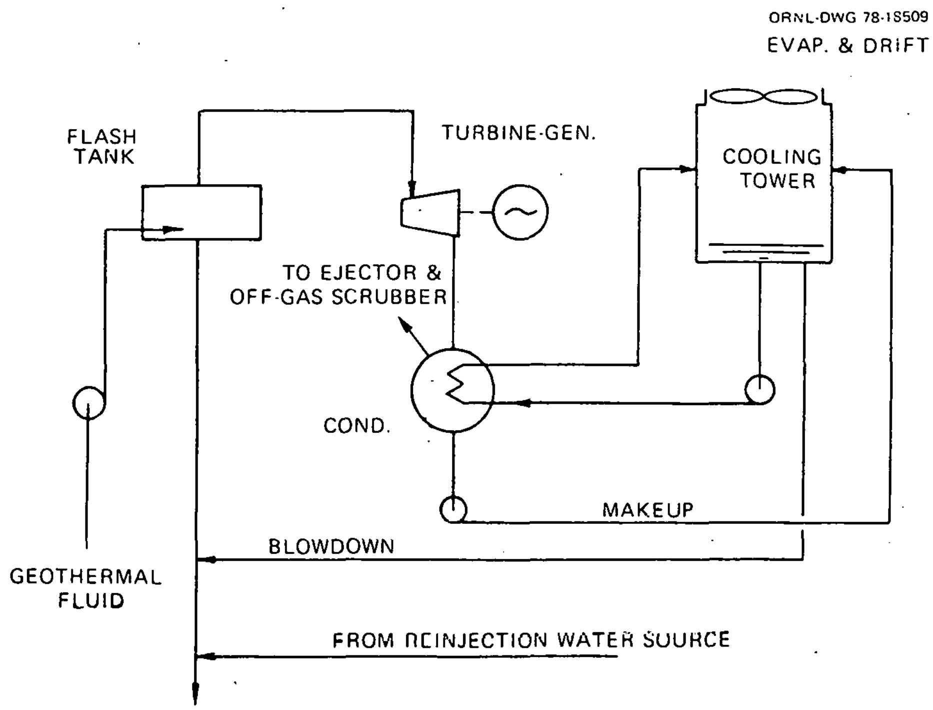

Fig. 2. Flashed-steam cycle using surface condenser and wet mechanical-draft cooling tower.

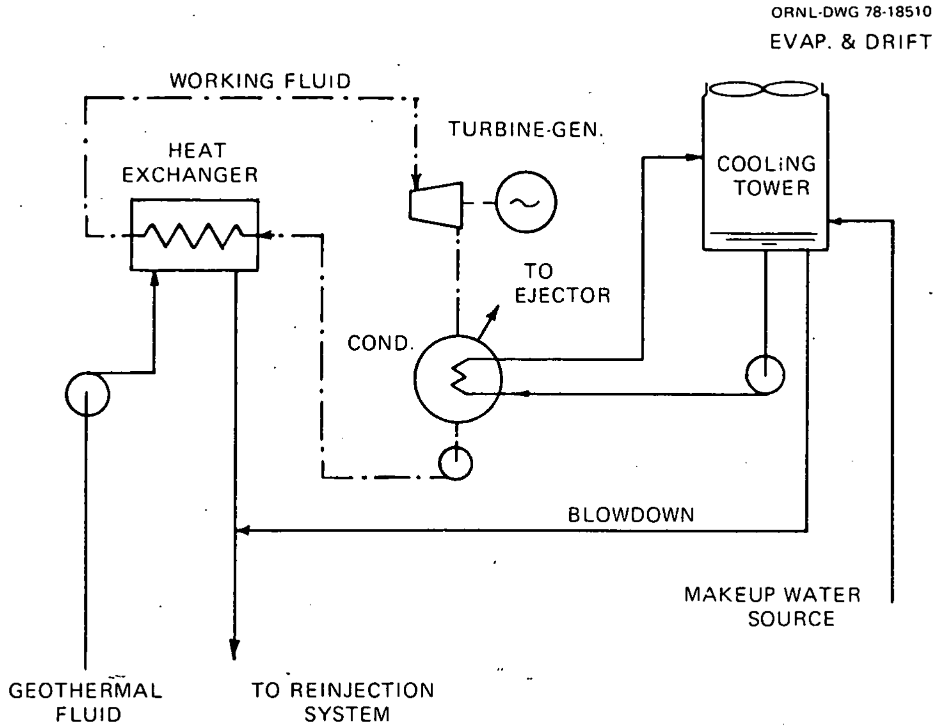

Fig. .3. Binary cycle using surface condenser and wet mechanical-draft cooling tower.

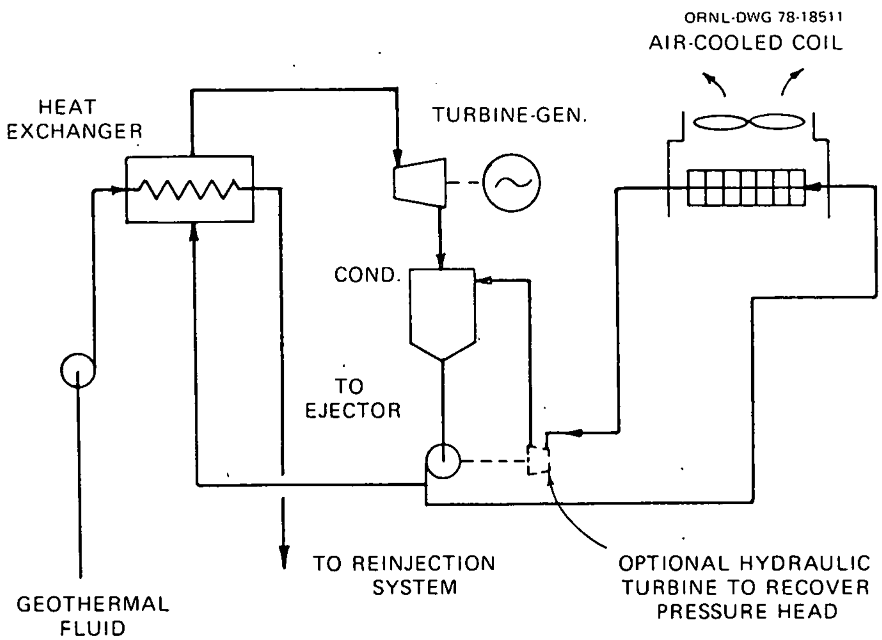

Fig. .4. Heller-type cycle using closed heat-exchanger, direct-contact condenser, and dry-type cooling tower.

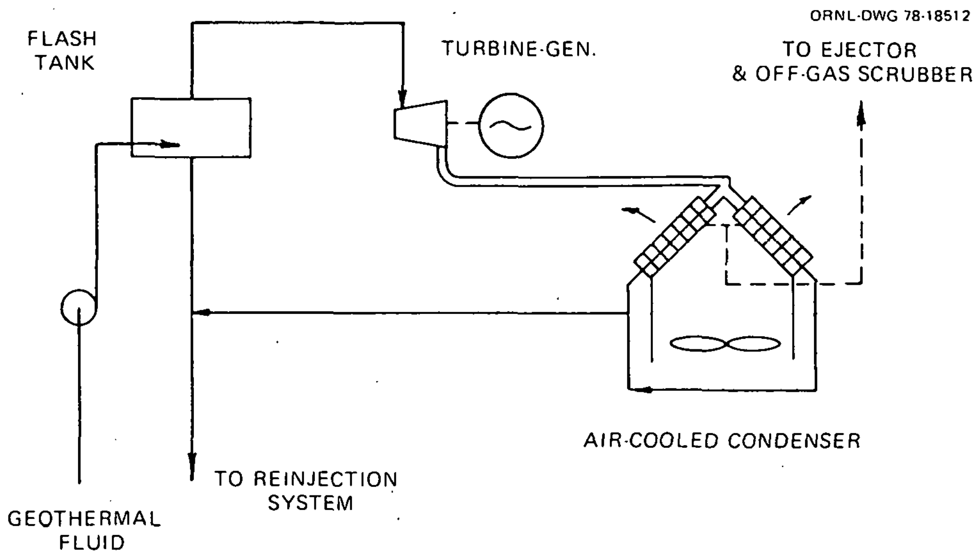

Fig. 1.5. Flashed-steam cycle with direct condensation of steam in dry cooling tower.

Studying trends in waste heat rejection system designs by examining existing or planned geothermal power stations is not conclusive at this time. Examples of almost all kinds can be found. For instance, The Geysers field in California uses direct-contact condensers in conjunction with wet mechanical-draft cooling towers. Both barometric leg and low-level types of condensers are used, the latter being the most recently installed. The Cerro Prieto Station in Mexico uses direct-contact condensers with barometric legs and wet mechanical-draft cooling towers. A Bechtel study for 50-MW(e) stations at Heber and Niland, California, differ in that the former is a flashed-steam system using a direct-contact condenser and the latter is a binary (isopentane) cycle with a horizonal shell-and-tube condenser.[1] Both cycles employ wet mechanical-draft cooling towers. There are many examples in Europe of air-cooled coils used either for direct condensation of the turbine exhaust steam or for cooling the circulated condenser coolant. The only consistency noted thus far at the various stations is that the U.S. geothermal applications have been too small in size or located in a climate too dry to encourage use of natural-draft cooling towers.

This paper is concerned only with the waste heat rejected from the power cycles of geothermal power stations; the broader subject of thermal uses for geothermal energy will not be discussed.

There are many examples of multipurpose, or cogeneration, power stations that partially expand steam in a turbine generator and then use the relatively high-temperature exhaust steam for industrial purposes, space heating, etc. Because geothermal power plant steam turbines will generally be supplied with relatively low-temperature saturated steam and because other higher-temperature heat sources may be available from the geothermal fluid, it seems doubtful that multipurpose cycles of this kind would have significant use at geothermal installations. The following comments are directed to the use of waste heat at conventional turbine discharge temperatures.

There is continuing interest in the utilization of the waste heat from power stations. Because the amounts rejected are so large, the use of even a small percentage of it would represent important energy savings. There have been relatively few useful applications of the waste heat to date, however, primarily because of poor economics. It is usually uneconomical to convey the heat for several miles, either as very low-pressure, high-specific-volume steam or as warm water only a few degrees above ambient temperature, because of pumping, piping, and right-of-way costs. The utilization factor (i.e., the ratio of actual heat use to design capacity) is of particular importance in the cost analysis. The reliability of the waste heat supply is also an important consideration. In discussing why there has been relatively little use of central power station waste heat, Beall[2] stated that the absence of an organized effort in this respect by the utilities industry and the lack of profit incentives for a utility (other than those that accrue from good public relations or from rental of unused lands) may be the reason why present-day applications are primarily of a demonstration nature.

Some of the best geothermal sites in the United States are in the lower Imperial Valley of California where many food crops are grown. It may be that food drying, greenhouse heating, and canning operations would profit from use of the waste heat from geothermal power cycles rather than using prime heat from the geothermal wells. The temperature of the steam exhausting from the turbine of a station using dry cooling towers would be high enough for absorption refrigeration system, such as those operating on the lithium-bromide cycle. This refrigeration could freeze food, provide cold storage, and be used for air conditioning. The steam temperature would also be high enough for use in water-desalting plants, which may become profitable in the Imperial Valley because of the shortage of potable water.

In 1972 Congress enacted amendments to the Federal Water Pollution Control Act (FWPCA), which generally required the use of the best available technology to dissipate heat produced in the generation of electric power. Subsequent guidelines, proposed by the EPA in 1974, found that this goal could be met only by closed recirculating cooling systems. Although use of cooling towers for heat dissipation would reduce the sometimes severe environmental impacts associated with once-through cooling systems, it is recognized that wet cooling towers are not without their drawbacks.[3] Aside from increased costs, towers can cause discharges of vapor plumes, ground level fog, undesirable aerosol drift (especially when saline water is used for makeup) and can generate noise, be visually conspicuous, and last, but perhaps most importantly in water-short areas, consumptively evaporate significant quantities of water. A 50-MW(e) geothermal power station with an overall thermal efficiency of 10% and a concentration factor of 2 in the cooling towers would require about 0.4 m^3/sec (6000 gpm) of makeup water.
Expansion of electric power at today's growth rates indicates a future need for cooling tower makeup water that will further stress an intensely competitive situation with regard to water availability. Water quality as well as quantity is also now a serious issue. Wet cooling towers not only consume water, but if the untreated blowdown is returned to a diminished stream, the concentration of solids and impurities is increased downstream.

Increasing demands will be placed on the courts to decide water rights issues. Each state has its own laws and regulations concerning water use, and although there has been much progress in achieving uniformity, the disparities are still sufficient to make a detailed treatment of the subject beyond the scope of this discussion. The legal aspects are thus complex and the subject of much litigation. Despite the weight given to precedent by the courts, legal decisions are influenced by changing societal pressures, and in no area is this more evident perhaps than in water rights rulings. Decisions handed down a few years ago may no longer seem proper when viewed in the light of today's water demands. Courts will become increasingly involved in evaluation of the relative merits of water uses, such as irrigation vs cooling tower makeup. The stakes are very high, and the technical and economic aspects in such controversies may tend to get lost in the intensely political climates in which the issues will be settled.

CONDENSERS

To obtain good Rankine cycle efficiencies the turbine exhaust pressure must be as low as can economically be attained. Table 1 lists the condensing pressure associated with the condensing temperatures of five working fluids commonly used in geothermal power cycles: water, ammonia, Freon-22, isobutane, and isopentane.*

The exhaust pressure of a high-efficiency steam turbine must be well below atmospheric. The vacuum is achieved by condensing the exhaust steam, which also serves the important function in closed cycles of allowing recovery of the condensate. The degree of vacuum obtained depends on the turbine loading, the amounts of noncondensable gases present in the condenser because of inleakage and other sources, the cleanliness of the condenser tube surface, and most importantly the condensing temperature of the steam as influenced by the temperature of the cooling water (or other heat sink) available. The condensing temperature is usually in the range of 3 to 6°C (5 to 10°F) above the average temperature of the cooling water used as the heat sink or about 8°C (15°F) above the average dry-bulb temperature of the ambient air if dry cooling towers are used.

The amount of moisture present in the last stages of the turbine is often the limiting factor in the expansion process. The optimum design exhaust conditions will thus vary with the steam and cooling-water costs.

The amount of noncondensable gases present in the condenser is dependent on the tightness of the system against air inleakage, the amounts of gases entrained or dissolved in the steam supply to the turbine, and the amounts of gases released by chemical reactions in the water. Thermal power stations utilizing steam flashed from a geothermal fluid may have to contend with relatively large amounts of noncondensable gases being swept through the turbine and into the condenser. The noncondensable gases, even in amounts of less than 1% of the throttle

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*Where feasible, this study gives SI units followed in parentheses by the commonly used English units. Conversions were made by using the following accepted reference: American Society for Testing and Materials, *Standard for Metric Practice*, E 380-76, Philadelphia (1976). The method to show factors for converting units is illustrated by the following example: Btu/hr·ft·°F = W·m/m^2·K x 0.577789. This conversion states that, to obtain the units of Btu/hr·ft·°F, values expressed in W·m/m^2·K should be multiplied by 0.577789.

Table 1. Saturation pressure of common working fluids as a function of temperature

°C	°F	Water[a] kPa	Water[a] psia[e]	Ammonia[b] kPa	Ammonia[b] psia	Freon-22[c] kPa	Freon-22[c] psia	Isobutane[d] kPa	Isobutane[d] psia	Isopentane[d] kPa	Isopentane[d] psia
15	59	1.705	0.2473	730	105.8	789	114.5	257	37.3	64	9.2
20	68	2.339	0.3392	858	124.5	910	132.0	301	43.6	77	11.1
25	77	3.169	0.4596	1004	145.7	1044	151.4	348	50.5	92	13.3
30	86	4.246	0.6158	1168	169.4	1192	172.9	401	58.2	109	15.8
35	95	5.628	0.8163	1352	196.1	1355	196.5	461	66.9	129	18.7
40	104	7.384	1.0710	1557	225.8	1534	222.4	527	76.4	151	21.9
45	113	9.593	1.3913	1785	258.8	1729	250.8	599	86.8	177	25.6
50	122	12.349	1.7911	2036	295.3	1942	281.7	679	98.4	205	29.8
55	131	15.758	2.2855	2314	335.5	2175	315.4	767	111.2	237	34.4
60	140	19.940	2.8921	2618	379.7	2427	351.9	863	125.2	272	39.5
65	149	25.030	3.6303	2953	428.2	2607	378.1	967	140.2	312	45.2
70	158	31.190	4.5237	3317	481.1	2996	434.5	1080	156.6	355	51.5

[a] J. H. Keenan and F. G. Keyes, *Steam Tables — Metric Units*, Wiley New York, 1969.

[b] S. L. Milora and S. K. Combs, *Thermodynamic Representations of Ammonia and Isobutane*, ORNL/TM-5847 (May 1977).

[c] *Handbook of Fundamentals*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, New York, 1972.

[d] K. E. Starling, *Fluid Thermodynamic Properties for Light Petroleum Systems*, Gulf Publishing Company, Houston, 1973.

[e] kPa (kilopascal) = psia x 6.894757.

steam flow, can reduce markedly the performance of the condensing equipment unless adequate provisions to accommodate and remove these gases are provided. The hydrogen sulfide, which may make up a high percentage of the noncondensable gases, is toxic and corrosive and has an objectionable odor even in very small concentrations. Means will be required at most installations to collect and dispose of the condensables in an approved manner. Cooling-system equipment probably will require special corrosion protection. The system designer must keep in mind the time required to lower the condenser pressure to the design turbine back pressure for startup. Special high-capacity pumping equipment, called the "hogging" system, is needed if the time is kept to within reasonable limits (about 30 min to 1 hr). Many power stations have two-stage pumps or ejectors, and some arrange the first stage for hogging.

The above comments apply to condensing systems for steam turbines. In contrast, working fluids other than water result in cycles having pressures greater than atmospheric throughout, and air inleakage into the systems is not as great a problem. In fact, the condensing pressures are high enough in Freon-22 and ammonia systems to make it more economical at times to use several small condensers rather than a single large shell designed to withstand the pressure. Another distinct difference between the use of water and other working fluids is that although the thermodynamic properties of water result in a turbine exhaust in the saturated vapor region, the exhausts are in the superheated region when using isobutane, Freon-22, or ammonia.

Single-fluid direct-contact condensers

In a typical direct-contact condenser, the vapor is condensed by spraying the subcooled liquid of the same fluid into it. Condensation occurs on the falling, relatively cool, liquid droplets. The most common forms are the low-level and barometric-leg types used to condensate the turbine exhaust in steam power plants. Another form bubbles the vapor to be condensed through a pool or stream of the liquid. Both of these types, having only a single fluid present, are termed "single-fluid" condensers. (Direct-contact condensers are also being investigated for binary geothermal power cycles where the turbine exhaust is condensed by direct contact with a different immiscible fluid. This type of condenser is called a "two-fluid" type and is dicussed below).

Direct-contact condensers tend to be simpler in design than surface condensers, and they have appreciably lower initial costs, particularly in geothermal power applications where the direct-contact condensers do not require complex internals. There are not as many leakage problems with spray condensers as with the multiplicity of tube joints in surface condensers. Unlike the latter, the spray condensers require little maintenance or cleaning, and the heat transfer performance does not deteriorate with time. Spray condensers may occupy about one-third the space of a surface condenser for the same duty, and there will be a corresponding reduction in costs of turbine pedestals and other concrete work.

The disadvantages of direct-contact condensers are mainly associated with the fact that the condensate is mixed with the cooling water. The contaminated condensate would require deaeration and treatment before it could again be used as boiler feedwater. This factor of water quality prevented widespread use of direct-contact condensers in large steam power stations for several decades. It was not until the relatively recent interest in dry cooling towers and the Heller system that the direct-contact condensers have again come into limited use. In this system, as was indicated in Fig. 4, the condenser water is cooled in a closed loop in an air-cooled coil so that it can be maintained at a high quality and with low gas content. Geothermal power cycles have also given impetus to the use of direct-contact condensers because the condensate is not recovered in many instances. If deaeration of the condensate and minimum subcooling are not of perticular interest, condensers do not need a complicated internal arrangement of nozzles, baffles, and trays.

A further disadvantage of direct-contact condensers is that near saturation conditions at the hot-well pump inlet necessitate that the cooling-water pumps operate with low-net-positive suction heads if flashing is to be avoided. The pumps that circulate cooling water to the condensers may also operate at a higher head than pumps that supply water to a surface condenser. In the latter, the only pumping head is due to fluid friction, and the whole system operates above atmospheric pressure. In direct-contact condensing systems used in conjunction with air-cooled coils (as in the Heller arrangement), operation of the coil portion of the system above atmospheric pressure to minimize air inleakage is desirable. In this case, the water pumps must supply this head plus the pressure drops at the spray nozzles. [The latter is usually in the range of about 34.5 kPa (5 psi)]. In theory, a hydraulic turbine could be used between the coils and the condenser as a pressure letdown device to recover a portion of the pumping head, but this system may be marginal and each particular case needs to be studied. To prevent flooding of the condenser in the Heller system in the event of failure of the water-circulating pumps, slow-closing stop valves are needed in the supply line to the water spray nozzles. The air-cooled coils also need to be protected from any excessive pressure surges in the condenser.[4] Mixing, or direct-contact, condensers can be classified as either (1) barometric or (2) low-level types:

1. A barometric condenser is shown in Fig. 6. The turbine exhaust enters at the top of the mixing chamber where it meets the cooling water, which is either injected by spray nozzles or allowed to splash to form curtains through which the steam must pass. The condensed steam and cooling water collect in the bottom of the vessel and drain into a tail pipe [10.4 m (34 ft) or more in height], which acts as a

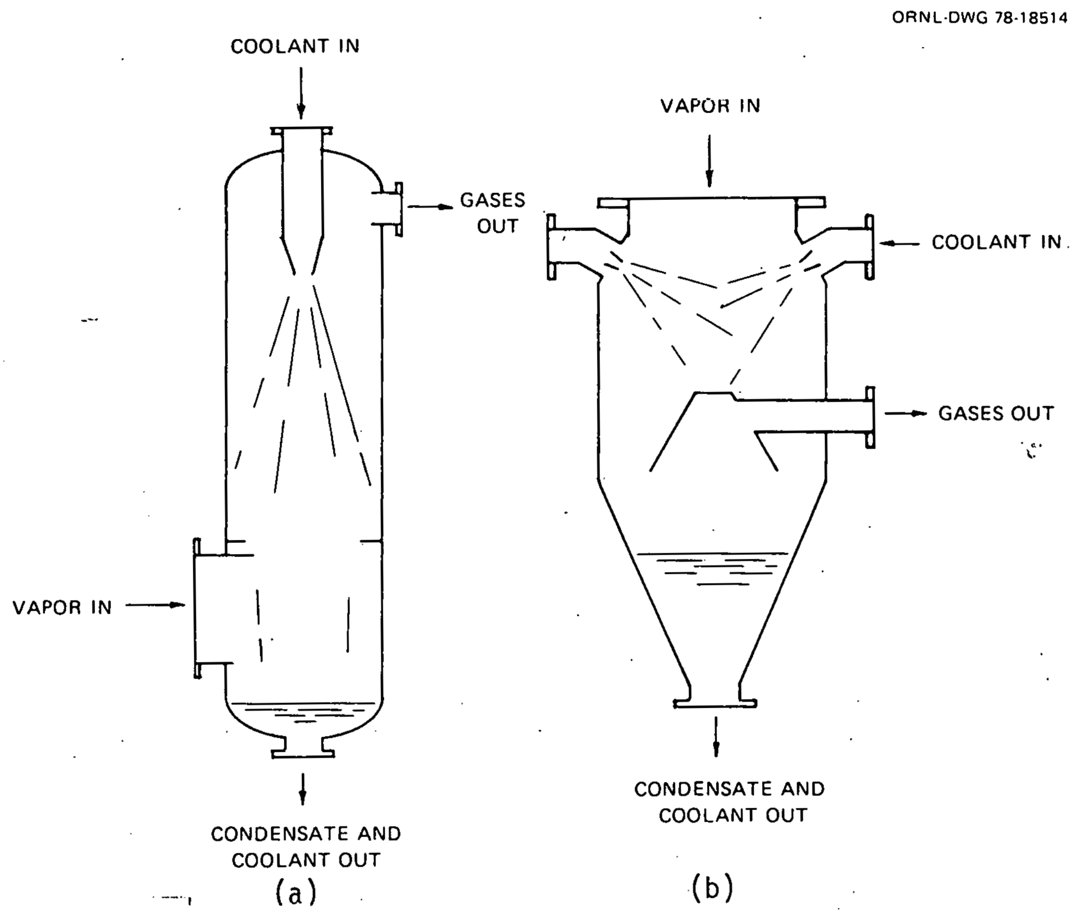

Fig. 6. Schematic of flow arrangement in direct-contact condensers: (a) counterflow and (b) parallel flow. Source: H. R. Jacobs and Heimer Fannar, *Direct Contact Condensers — A Literature Survey*, DGE/1523-3, UTEC 77-081, Mechanical Engineering Dept., University of Utah, Salt Lake City, (February 1977), Fig. 2.A (2-1). Reprinted by permission.

barometric leg or column to allow the condensate to flow out by gravity through a water seal, or air trap. This arrangement eliminates the need for the condensate pump, vacuum breaker, and pressure-relief devices but has the disadvantage of requiring 12 m (40 ft) or more of headroom. The mixture of noncondensable gases and water vapor collects at the top of the vessel after being in contact with the coolest water, and the gas is removed by either vacuum pump or steam ejector. Mixing condensers can differ in design according to whether the steam flow is countercurrent to the water spray, as in Fig. 6 (a), or parallel to the spray, as in Fig. 6 (b), but the performance characteristics are similar.

2. A low-level direct-contact condenser may be essentially the same as a barometric condenser except that the condensate is removed by pumping rather than by gravity flow through the barometric leg. Elimination of the tail pipe usually makes it possible to install the condenser in the optimum position directly coupled to the turbine exhaust. The space requirements are no more than those for a surface condenser. The condensate removal can be effected by either centrifugal pumps, the kinetic energy of water jets, or a combination of the two.

A variation of the low-level direct-contact condenser is the multi-jet ejector condenser. Here, the cooling water flows at high velocity from converging jets into a Venturi section that aspirates the exhaust steam into the throat and condenses it. This arrangement has the advantage of removing the noncondensable gases along with the vapor and can achieve high vacuums. The cooling-water consumption is higher than the consumption for the mixing chamber type of condenser. However, for relatively small geothermal applications (such as well head installations) where the amount of noncondensable gases are relatively high and unpredictable, the ejector condenser may be considered.

The shells for direct-contact condensers have typically been made of carbon steel. However, the Bechtel Corporation study for a 50-MW(e) geothermal power plant at Heber, California, proposed that the low-level direct-contact condenser shells be fabricated of reinforced concrete designed for 517 kPa (75 psia) and full vacuum.[5] A condenser of similar design is being installed at Hatchobaru, Japan. The interior surface of the concrete would be impregnated with an epoxy mixture to seal against air inleakage through hairline cracks. Satisfactory operating experience with similarly sealed concrete vessels in desalting plants is cited by Bechtel.[5] The weight of the concrete shells is sufficient to anchor the condensers even where groundwater elevations are relatively high.

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 splashing action at saturation temperature provides good deaeration. The terminal temperature difference (i.e., the temperature difference between the leaving water and the condensing steam temperature) theoretically could be zero, but in actual practice may be as high as 6°C (10°F). For a given cooling-water inlet temperature, the direct-contact condenser will provide lower turbine back pressures than would a surface condenser.

The heat transfer processes in the condenser are complex and highly dependent upon the physical dimensions of the system. Development of mathematical models for spray condensers would depend on knowing stripping and diffusion coefficients. Much of this information is considered proprietary. The two parameters that probably have the greatest influence on the performance are the surface area of the condensing water in contact with the steam and the relative velocity between the steam and the condensing water. There is, then, an advantage to smaller water droplets and longer fall times. A terminal temperature difference of about 3°C (5°F) is common practice, although this is dependent on the amount of noncondensable gases present. The Electric Power Research Institute[6] has investigated the modeling of direct-contact condensers so that the dimensions and costs can be roughly estimated. The model makes assumptions such as the holes in the trays are 1.27 cm (0.5 in.) in diameter, the height of the water in the tray is six times the hole diameter, and the height of the condenser from the bottom of the first tray to the water outlet is twice the diameter of the tray. Such rules of thumb are sufficient for the modeling purposes intended but are, of course, not reliable design guides. In design of the units, major manufacturers rely heavily on previous experience and experimental testing programs. There is very little specific design information in the literature.

Two-fluid direct-contact condensers

In geothermal binary power cycle applications, this type of condenser would condense the turbine exhaust vapor mixture by direct contact with a cooling fluid that is immiscible with the working fluid. The vapor may be condensed by contact with sprayed droplets from the cooling fluid, by bringing the vapor into contact with a film of the coolant liquid, or by bubbling the vapor through a pool of the cooling fluid. The working fluids generally considered for this type of geothermal cycle are light hydrocarbons and halogenated hydrocarbons, and the obvious selection for the coolant is water because of its superior thermal properties, lower pumping energy requirements, and lower cost. A typical binary geothermal cycle, direct-contact condenser application would condense a mixture of about 90 to 95% working fluid (such as isobutane or isopentane) and about 5% steam by transferring heat (1) into water droplets through a water film or (2) from collapsing vapor bubbles into a water pool.

Unlike direct-contact boilers or heat exchangers, which are of primary interest in geothermal cycles because they reduce the scaling and fouling problems, direct-contact condensers would be justified mainly on the basis of lower capital costs, closer approach temperature, and more efficient separation of the two fluids. Direct-contact condensers bringing the working fluid vapor into contact with falling water droplets would not be unlike the single-fluid direct-contact types described in Sect. 2.3.2. Film-type direct-contact types would use a packed bed of rings or saddles. Bubble-type condensers have been of interest primarily as open feedwater heaters and vapor suppression systems in reactor containments and in condensers for seawater distillation.

Jacobs and Fannar[7] have reviewed the state of the art of direct-contact condensers and have published a comprehensive literature survey on the subject, covering both U.S. and British sources. Many of the above comments on direct-contact condensers were extracted from their work. Work is in progress at the University of Utah's Mechanical Engineering Department on direct-contact heat exchangers.[8] These tests are related to those now being conducted at East Mesa, California, by DSS Engineering (Ft. Lauderdale, Florida) on mixing-type heat exchangers. The DOE-sponsored tests at The Great Lakes Chemical Company in El Dorado, Arkansas, will also investigate direct-contact boilers and condensers.

Surface condensers

In binary geothermal power cycles a surface condenser is commonly used to separate the working fluid and the condenser coolant. The condensers are the shell-and-tube type, and almost without exception the condensing vapor is on the shell side and the coolant flows through the tubes. Surface condensers are widely used in steam power stations and the construction features and heat transfer relationships are well documented in the literature. Some selected properties of liquid ammonia, Freon-22, isobutane and water are given in Table 2.

Table 2. . Selected properties of liquid ammonia, Freon-22, isobutane, and water at temperatures of 15, 40, and 60°C

Fluid	Properties for given temperatures 15°C	40°C	60°C
	k^a = thermal conductivity, $W \cdot m/m^2 \cdot K^b$		
Ammonia	0.505	0.445	0.400
Freon-22	0.0928	0.0803	0.0704
Isobutane	0.111	0.101	0.0935
Water	0.592	0.631	0.654
	μ = absolute viscosity, Pa·sec or kg/m·sec[b]		
Ammonia	0.000160	0.000122	0.0000984
Freon-22	0.000213	0.000183	0.000162
Isobutane	0.000179	0.000142	0.000122
Water	0.00112	0.000632	0.000452
	c_p = specific heat, J/kg·K or W·sec/kg·K[b]		
Ammonia	4678	4870	5117
Freon-22	1217	1323	1495
Isobutane	2450	2647	2843
Water	4192	4179	4188
	ρ = density, kg/m^3		
Ammonia[c]	618	580	546
Freon-22[c]	1232	1131	1032
Isobutane[c]	563	532	504
Water[d]	999	992	983
	h_{fg} = latent heat vaporization, MJ/kg or MW·sec/kg		
Ammonia[c]	1.21	1.10	1.00
Freon-22[c]	0.193	0.167	0.141
Isobutane[c]	0.337	0.307	0.280
Water[d]	2.47	2.41	2.36

[a]Conversion factors: k: (W/m·K) x 0.577789 = Btu/hr·ft·°F; μ: (Pa·sec) or (kg/m·sec) x 2419.09 lbm/ft·hr; c_p: (J/kg·K) or (W·sec/kg·K) x 0.0002388 = Btu/lbm·°F; ρ: (kg/m^3) x 0.062428 = lbm/ft^3; h_{fg}: (J/kg) or (W·sec/kg) x 429.5911 = Btu/lbm.

[b]*Thermophysical Properties of Refrigerants*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, New York, 1976.

[c]*Handbook of Fundamentals*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, New York, 1972.

[d]J. H. Keenan et al., *Steam Tables, Thermodynamic Properties of Water, Including Vapor, Liquid and Solid Phases*, Wiley, New York, 1969.

Noncondensable gases

The percent of noncondensable gases in geothermal steam varies from well to well but averages less than 1% by weight. Typical ranges of concentrations of the various gases in geothermal steam at The Geysers are shown in Table 3. Carbon dioxide is by far the major constituent, amounting to 75 to 95% of the noncondensables. The flashed-steam system at Cerro Prieto, Mexico, has steam entering the turbines which contains impurities in the following average amounts: CO_2, 14,000 ppm; H_2S, 1500 ppm; NH_4, 110 ppm; chlorine, 0.8 ppm; sodium, 0.4 ppm; SiO_2, 0.2 ppm.[9] Although hydrogen sulfide (H_2S), a highly toxic gas, is present in much smaller amounts, it can be detected by smell at such low concentrations that a layman's impression is that it exists in large amounts at geothermal power installations. Its odor is a nuisance at concentrations as low as 0.07 ppm, and it causes eye irritation at 1 ppm. Prolonged exposure at concentrations of 100 ppm can be fatal, and about 1 hr of exposure to concentrations above 600 ppm is fatal. The potential hazard of H_2S is increased because it cannot be detected by smell at the higher concentrations. Mercury and radon, which may be present in trace amounts at some wells, are of particular concern because they are also toxic at very low concentrations.[9] Ammonia is a potential hazard, but it usually exists at too low a concentration to be of concern.[10]

Geothermal power systems, particularly those utilizing steam flashed from geothermal fluids, may have to contend with significant amounts of noncondensable gases being swept through the system and into the turbine condenser. The gases reduce the turbine efficiency by expanding with less enthalpy drop than steam,[5] but, perhaps more importantly, the gases can reduce the condensing coefficient on the shell side by factors of 2 or more. The effects of the gases on the condensing process can be explained as follows:

Table 3. Percent by weight of constituent gases in geothermal steam at The Geysers

Gas	Low	High	Design
Carbon dioxide	0.0884	1.90	0.79
Hydrogen sulfide	0.0005	0.160	0.05
Methane	0.0056	0.132	0.05
Ammonia	0.0056	0.106	0.07
Nitrogen	0.0016	0.0638	0.03
Hydrogen	0.0018	0.0190	0.01
Ethane	0.0003	0.0019	
Total	0.120	2.19	1.00

Source: J. P. Finney, *The Design and Operation of The Geysers Power Plant, Geothermal Energy,* ed. by Paul Kruger and Carel Otte, Stanford University Press, Stanford, Calif., 1973, Table 1, p. 148.

1. As the water vapor proceeds through the condenser shell, with only a relatively small pressure drop due to flow friction, condensation causes the proportion of the noncondensable gases to increase and the partial pressure of the water vapor to decrease. The condensing temperature of the water vapor is thereby reduced, as well as the effective temperature differences for heat transfer. Increasing amounts of noncondensable gases therefore reduce the temperature difference across the tube wall.

2. The water vapor, driven by the partial pressure difference, will move toward the cooler walls of the tubes and in doing so will carry the noncondensable gases with it to the tube walls. The gas film surrounding the tubes, unless adequately swept away by the stream velocities, acts as a barrier through which the water vapor must diffuse. The rate of condensation is thus controlled by the laws of vapor diffusion through a film of noncondensable gases[11] rather than the usual laws of heat transfer by conduction and convection. This effect can cause a significant reduction in the overall heat transfer coefficient.

#11

3. The water vapor pressure difference that must exist to cause diffusion of the vapor through the gas film causes a further reduction of the effective temperature difference for heat transfer.

4. The increase in the total bulk pressure in the condenser caused by the presence of the noncondensable gases represents a corresponding increase in the turbine exhaust pressure and a reduction in the enthalpy drop experienced by the steam in expanding through the turbine. Consequently the thermal efficiency of the power cycle is reduced.

5. The steam jet ejectors, or the vacuum pumps, necessary to remove the noncondensable gases from the condenser to prevent a buildup of the gases require an energy input that is large enough to have an important effect on the thermal efficiency of the cycle. Increases in the rate at which noncondensable gases enter the system thus demand correspondingly greater energy expenditures for the gas removal.

6. In conventional power cycles, the noncondensable gases can be vented from the system to the atmosphere without incurring significant environmental problems. Most geothermal power cycles, however, may have hydrogen sulfide or other objectionable noncondensable gases that will require capital expenditures for scrubbers.

Noncondensable gases are conventionally removed fron condensers by either steam-jet ejectors or by vacuum pumps. Single-stage units can operate at condenser pressures down to 100 to 200 mm (4 to 8 in.) mercury absolute, but two stages or more are needed for lower pressures. The gas-water vapor mixture entering an ejector or pump typically will contain about 30% gas and about 70% water vapor by weight. As the condensing temperature increases, the water vapor portion increases. An intercooler between the stages is advantageous in reducing the gas temperature and the power requirements, and substantial amounts of water can be removed in the intercooler.

Frequently the first stage of a two-stage ejector or vacuum pump unit has a high capacity and is used alone during startup as a hogger to reduce the condenser pressure to the operating range. Another common arrangement is to provide two units (one of which is for standby), and interconnections allow the first stages of both units to operate together for hogging. Even with such special provisions, it may require 30 min to an hour to pump the condensing system down sufficiently for startup.

Steam-jet ejectors have the advantage over vacuum pumps of having no moving parts; therefore, they require less maintenance, do not need lubrication, have lower initial costs, are not threatened by slugs of water in the suction, and tend to produce higher vacuums. Vacuum pumps have significantly lower operating costs, generally do not generate as much noise, and lend themselves better to computer-controlled systems. Some systems may use a combination of the two, with the ejectors used for the first stage and for hogging. With increased fuel costs in power stations, operating costs are becoming more important, and there is a trend in modern steam-power stations to use pumps rather than ejectors.

Mah[12] has calculated the steam consumption for one- and two-stage ejectors for various amounts of gas present. Figure 7 shows steam requirements at 690 kPa (100 psi) absolute operating steam. Adjustment factors for other steam supply conditions are given in Fig. 7. The ejector steam consumption at various operating geothermal power installations is shown by Mah[12] in Fig. 8. About 5% of the total steam flow is typical of the amount needed to operate the ejectors at most stations; however, at Cerro Prieto, Mexico, the consumption rate is 15%.

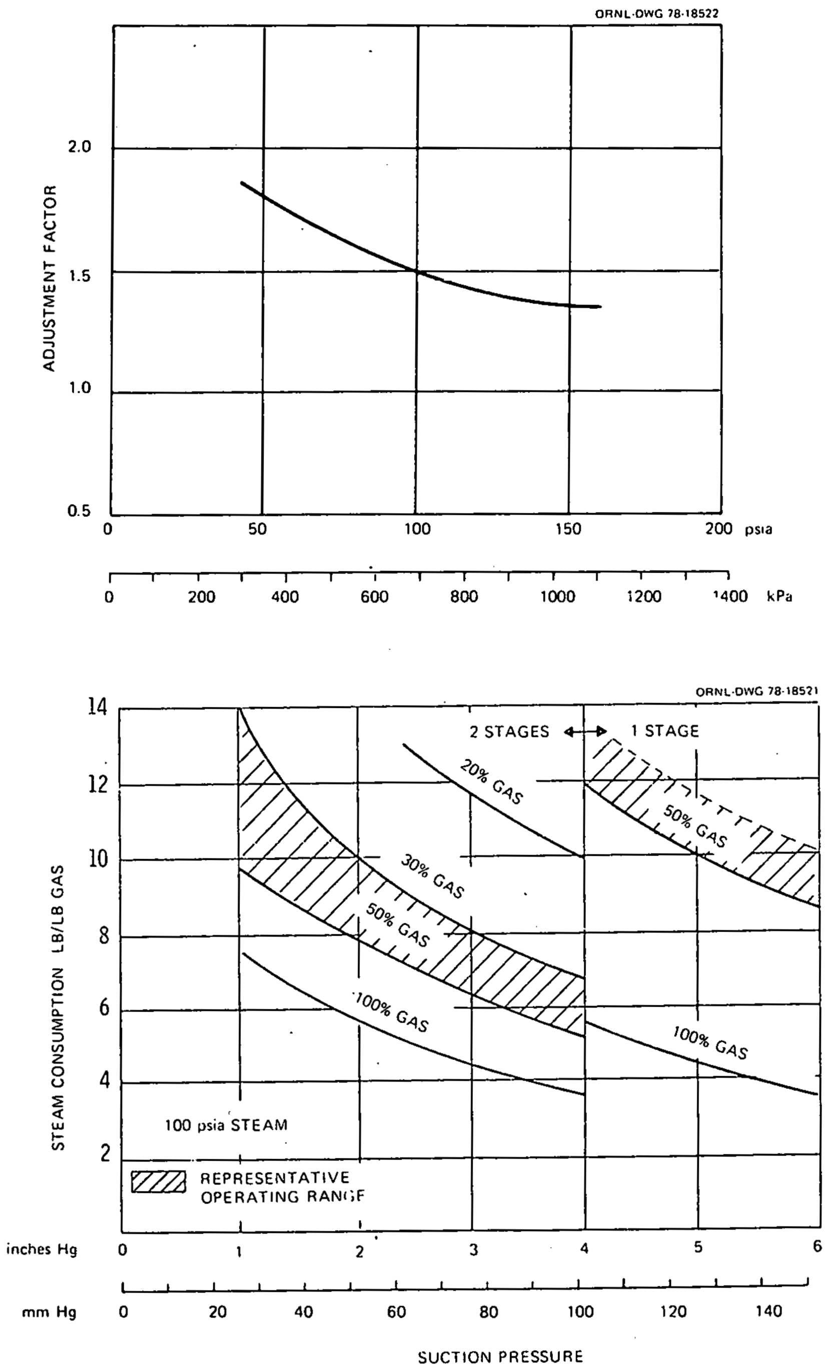

Fig. 7 . Steam requirement for ejectors as a function of the amount of noncondensable gases present and ejector suction pressure when using 690 kPa (100 psia) of saturated steam supply. Adjustment factors for other steam pressures are given in the top graph. Source: Clifford Mah, "Effect of Noncondensable Gases on the Performance of Rankine Cycles," presented at the Eighth CATMEC Meeting, San Diego, Calif., Feb. 21, 1978. Reprinted by permission of the Aerojet Energy Conversion Company (Aerojet Code PRA-SA 1 November 1978).

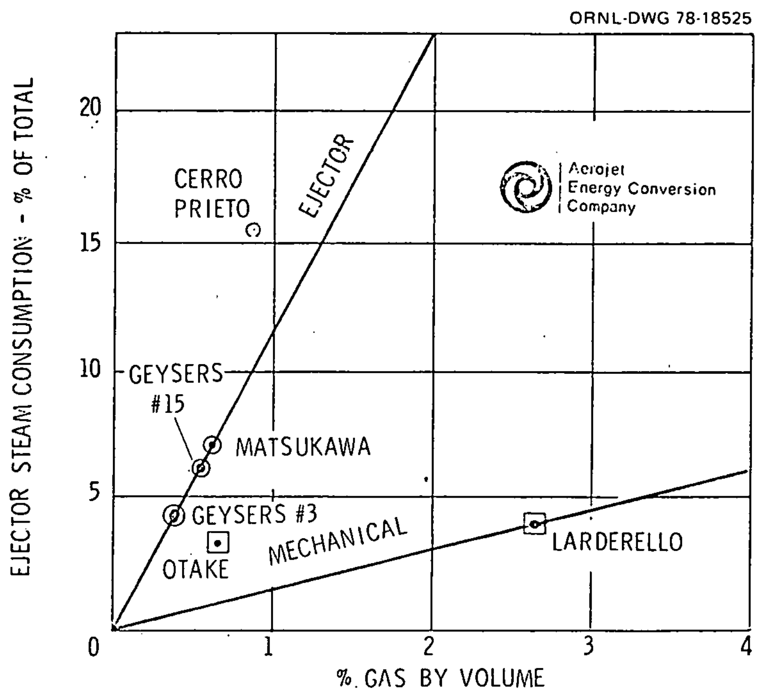

Fig. 8. Ejector steam consumption at some operating geothermal power stations. Source: Clifford Mah, "Effect of Noncondensable Gases on the Performance of Rankine Cycles," presented at the Eighth CATMEC Meeting, San Diego, Calif., Feb. 21, 1978. Reprinted by permission of the Aerojet Energy Conversion Company (Aerojet Code PRA-SA 1 November 1978).

2.4.5 Condenser tube fouling

In selecting the site for a power station, much emphasis is placed on achieving the lowest available average sink temperature because this temperature has a direct influence on the thermal efficiency of the Rankine cycle. Equal attention should be given, however, to the quality of the cooling water because reduction of surface condenser capacity through fouling of the tubes can also have a direct bearing on the cycle efficiency and the useful life of the equipment. If periodic plant shutdowns are required for cleaning of condenser tubes, fouling rates can have a significant impact on the costs of producing electric power.

Geothermal power stations will probably use wet cooling towers to the extent that makeup water is available. A wide variety of situations will exist at geothermal sites regarding the quality of the water. At geothermal stations using steam flashed from a geothermal fluid, the condensate can be used as makeup for the cooling towers. Although the quality of the makeup water will be relatively high in such systems, the quality of the water circulated through the condensers will be dependent on the cycles of concentration in the towers and on the amounts of dust, fumes, insects, and other atmospheric pollutants washed from the air that passes through the cooling towers. Makeup water taken from groundwater or surface water supplies may be so high in dissolved or suspended solids that clarifiers and other treatment would be necessary. Generally, the large quantities of water needed for cooling tower makeup make it uneconomical to provide extensive treatment, but where the only alternative is the use of dry cooling towers, relatively elaborate water treatment facilities may be more economical. For example, the proposed Sundesert Nuclear Power Station near Blythe, California, would have cleaned up high salinity water taken from the Palo Verde Outfall Drain for use as cooling tower makeup.

The use of relatively dirty water will cause deposits such as sediments, scale, algae, and slime on the condenser tube surfaces and markedly reduce the condenser heat transfer coefficient. The amount of deposits present will obviously depend upon the deposition rate and the time elapsed since the last cleaning. Buildup of deposits can be rapid when using water of poor quality; instances causing a 15 to 20% reduction in condenser capacity after 10 hr of operation have been cited.[13] Factors influencing the rate of fouling and the types of deposits are the kind of metal used for the tubes, the character of the tube surface, the water velocity through the tubes, and cooling-water properties such as temperature, dissolved solids, pH, and bacterial content. In the absence of definitive information on the average amount of deposits, a factor of 0.80 to 0.85 is commonly applied to the overall heat transfer coefficient to take care of fouling and other uncertainties. A common design fault is not to allow enough excess area. On the other hand, care should be taken not to be too generous with this excess capacity because it is expensive. Furthermore, when commencing operation with clean tubes in an oversized condenser, the plant operator may cut back on the coolant flow rate, thereby reducing the water velocity through the tubes and increasing the wall temperatures. Both of these processes tend to hasten the buildup of fouling deposits.

Condenser tube failure

A comprehensive study of 30 power stations disclosed that the major impact of surface condenser tube leakage is the value of the electric power generation lost when the unit is taken off-line to repair the leaks.[14] In the main condensing section in freshwater-cooled condensers, all the commonly used materials have a high probability of lasting the lifetime of the plant. In the air-removal section, however, Admiralty metal tubes gave less satisfactory service. In the saltwater-cooled condensers, it was found that, of all the materials surveyed, only titanium had a high probability of lasting the lifetime of the plant without the need for retubing. Aluminum-brass, aluminum bronze, and 90-10 copper-nickel have less than a 50% probability of lasting 40 years with only 10% of the tubes plugged. The study was less definitive in evaluating materials used in wet cooling tower systems. Stainless steel tubes appear to give good service in both once-through and cooling tower systems. The Bechtel publication, *Steam Plant Surface Condenser Leakage Study,*[14] contains an extensive bibliography on condenser tube materials, failure modes, and cleaning methods.

Approximately one-half of the failures of tubing are directly attributable to vibration damage.[14] Tube vibration is usually caused by high shell-side cross-flow steam velocities in the upper portions of the tube bundle or at poorly baffled drain lines. Improper support plate spacing or steam flow distribution or unusual condenser operating conditions are typical sources of vibration problems. Chenoweth has prepared a comprehensive report on the state of the art on the prediction of flow-induced vibrations in shell-and-tube heat exchangers.[15]

Titanium for condenser tube service

There will be an incentive to use titanium for condenser tubes at some geothermal plants because of relatively severe operating conditions. Although the superior resistance of titanium to corrosion and erosion has been known since the 1950s, its use has been limited because of its comparative high cost. However, the decreasing price of titanium in the 1970s, the increasing costs of other tubing materials, and the increased worth of reducing plant downtime for condenser maintenance have all brought titanium into a more competitive position.

Titanium is equal in strength and toughness to stainless steels and has better resistance to corrosion in severe applications. Table 2.9 shows that it has good abrasion resistance, is not subject to stress corrosion cracking or crevice corrosion, and will not corrode in the presence of sulfides, chlorides, mercury, and other man-made pollutants in the cooling water. On the steam side, titanium tubing is not attacked by noncondensable gases such as carbon dioxide, ammonia, and oxygen. It is also resistant to the erosive action of the high-velocity steam entering from the turbine exhaust.

Because titanium tends to retain a smooth surface, it resists biological fouling to a better degree than the commonly used tubing materials. Cleanliness factors for titanium, like stainless steel, are in the 0.9 to 1.0 range as compared to a value of 0.85 commonly used for other materials.

The thermal conductivity of titanium is about 16 to 20 W/m·K (109 to 138 Btu/in.·hr·ft^2·°F), as shown in Table 2.3, which is about the same as the thermal conductivity of the 304, 316, and 347 stainless steels but less than that of 410 or 501 stainless. The conductivity is less than that of the copper-nickel alloys by a factor of about 2, less than that of Admiralty metal by a factor of about 7, and less than that of aluminum by a factor of about 10. In heat transfer through condenser tubes, however, the resistance of the wall is usually relatively small in comparison to the film coefficients and fouling resistances. Further, the higher strength of titanium and the essentially zero allowance needed for corrosion and erosion allow use of No. 22 BWG tubing rather than the No. 18 BWG usually used for other materials. Because the resistance to cavitation and erosion is good, it may be economical to design for higher velocities in the tubes, perhaps 2.4 to 3 m/sec (8 to 10 fps), which can improve the inside film coefficient by 25 to 30% over those obtained with more conventional lower velocities. Figure 9 shows the relative heat transfer performance of titanium tubes compared to other materials if credits are taken for fouling factors, water velocities, and wall thickness. On this basis, titanium compares favorably with the coefficients of some of the other commonly used tubing materials, although still falling short of aluminum or Admiralty metal.

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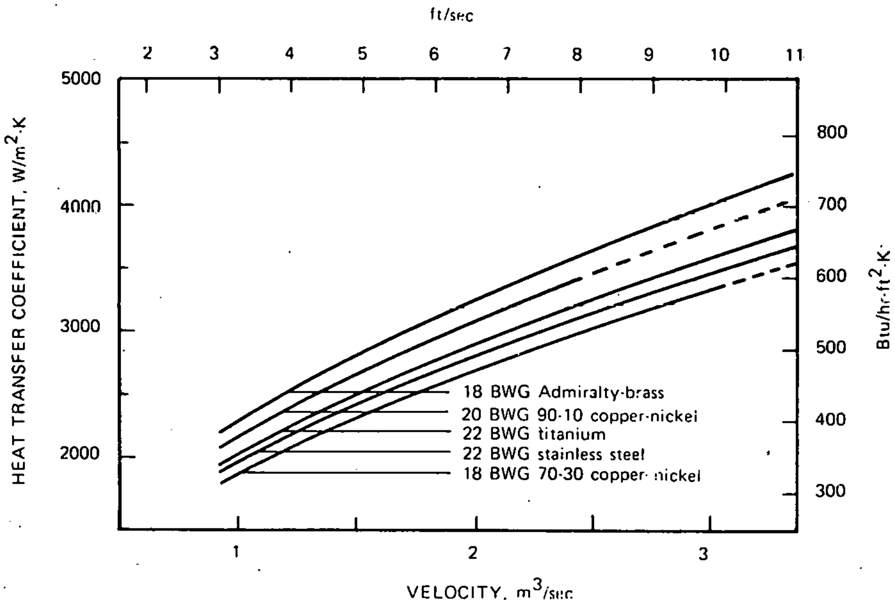

Fig. 9 |. Overall heat transfer coefficient as a function of water velocity in 2.5-cm (1-in.) OD tubes. The values shown are the result of applying appropriate fouling factors and adjustments for wall thermal conductivities to the heat transfer coefficients. Source: *Titanium Tubing for Surface Condenser Heat Exchanger Service,* Timet Bulletin SC-4, Titanium Metals Corporation. Reprinted by permission.

2.4.8 Enhanced-surface condenser tubes

The Oak Ridge National Laboratory has investigated the performance of vertical condenser tubes with fluted outside surfaces. The studies, initiated in 1964 to improve the performance of water-desalting plants, were encouraging and thought to have application in obtaining power from geothermal sources and ocean temperature gradients. The mechanism by which fluted surfaces improve the condensation heat transfer performance of a vertical tube is illustrated in Fig. 10 . The surface tension forces in the condensate film act to draw the liquid from the crests into the troughs, thereby reducing the resistance to heat transfer in the crest areas. Although the resistance is increased somewhat in the trough areas, the net effect is an improvement in the heat transfer performance over that attained with conventional smooth tubes.[16].

Experimental studies have been made of the performance of vertical tubes with enhanced surfaces on which various fluorocarbons, ammonia, and isobutane were condensed.[17] The tubes were 2.21 cm (7/8 in.) to 2.54 cm (1 in.) OD and had the cross-sectional profiles shown in Fig. 11 . Most of the tubes tested were of aluminum, but other materials were studied also (Fig. 11). The heat fluxes for ammonia varied from about 5000 to 50,000 W/m^2 (1600 to 16,000 Btu/hr·ft^2). When condensing fluorocarbons or isobutane, the heat fluxes varied from 5000 to 30,000 W/m^2 (1600 to 10,000 Btu/hr·ft^2). The condensing coefficient was found to improve by factors or 4 to 7 times over that of a smooth tube, depending on the heat flux and the geometry of the flutes.[17] The condensing film coefficients obtained when condensing ammonia are shown in Fig. 12 , and the coefficients for condensing isobutane are shown in Fig. 13 . The most efficient flute geometry depends on the surface tension of the condensed liquid, on the flute efficiency for condensate drainage, and possibly in some situations, on the conductivity of the tube wall. When condensing ammonia (Fig. 12), the tube with 60 square ridges (Tube E) gave somewhat lower condensing coefficients than the tube with 48 corrugations (Tube F). When condensing isobutane, however, Tube E gave a higher

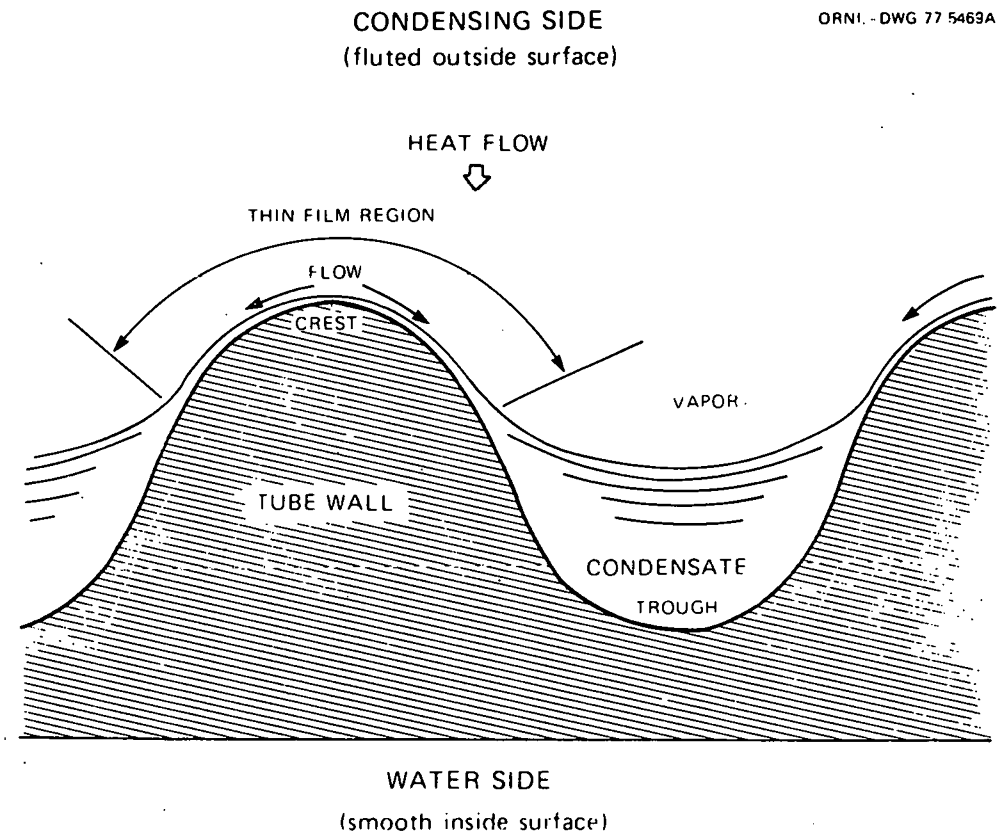

Fig. 10. Fluted tube principle of operation (condensation mode). Surface tension forces act to push condensate from crests into troughs. Source: S. K. Combs, *An Experimental Study of Heat Transfer Enhancement for Ammonia Condensing on Vertical Fluted Tubes*, ORNL-5356 (April 1978).

Fig. 11. Characteristics of fluted tubes. <u>Source</u>: J. W. Michel et al., *Energy Div. Annu. Prog. Rep. Period Ending Sept. 1977*, ORNL-5364, Table 5.2, p. 182.

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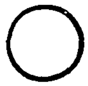
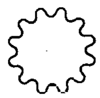
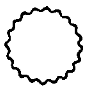
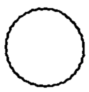
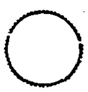
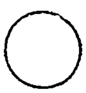
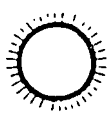
Cross section	Tube designation	Material	External perimeter (cm)	External surface area (m^2)	Number of external flutes	Number of internal flutes
	A	Aluminum	8.00	0.0973		
	B	Al-brass	11.94	0.1296	12	12
	C	Al-brass	8.90	0.0967	20	20
	D	CuNi (90 10)	8.90	0.0967	36	36
	E	Aluminum	12.71	0.1490	60	0
	F	Aluminum	8.26	0.0964	48	0
	G	Aluminum	9.75	0.1143	24	0
	H	Aluminum	14.00	0.1522	42	34
	J[a]	Aluminum	26.61 8.00[b]	0.3110 0.0973[b]	36	0

[a] A duplicate of tube A, with 36 stainless steel blades loosely attached.
[b] Numerator includes blades; denominator is base tube only.

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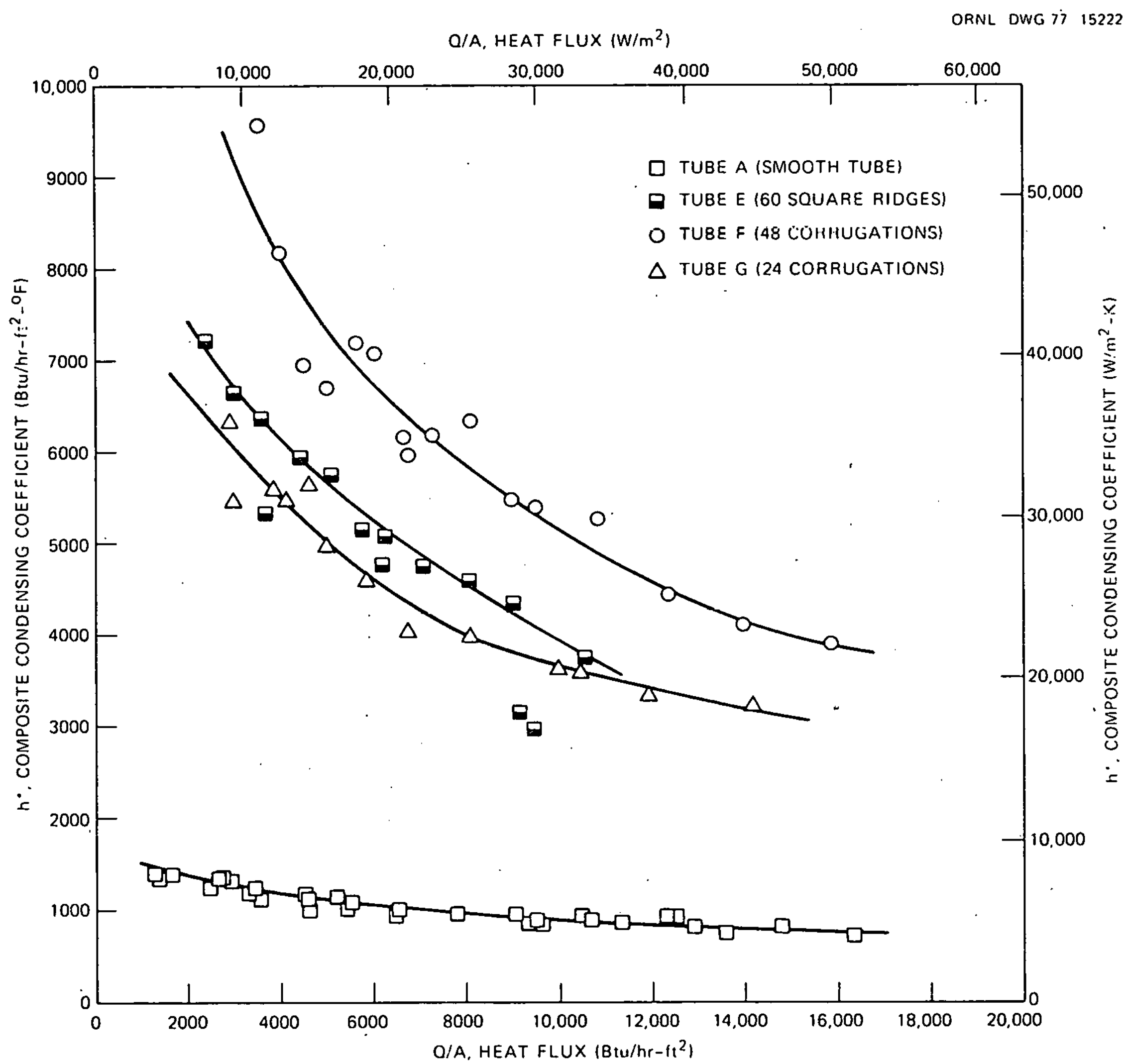


Fig. 12. Composite condensing heat transfer coefficients for ammonia using enhanced-surface tubes. <u>Source</u>: S. K. Combs, *An Experimental Study of Heat Transfer Enhancement for Ammonia Condensing on Vertical Fluted Tubes*, ORNL-5356 (April 1978).

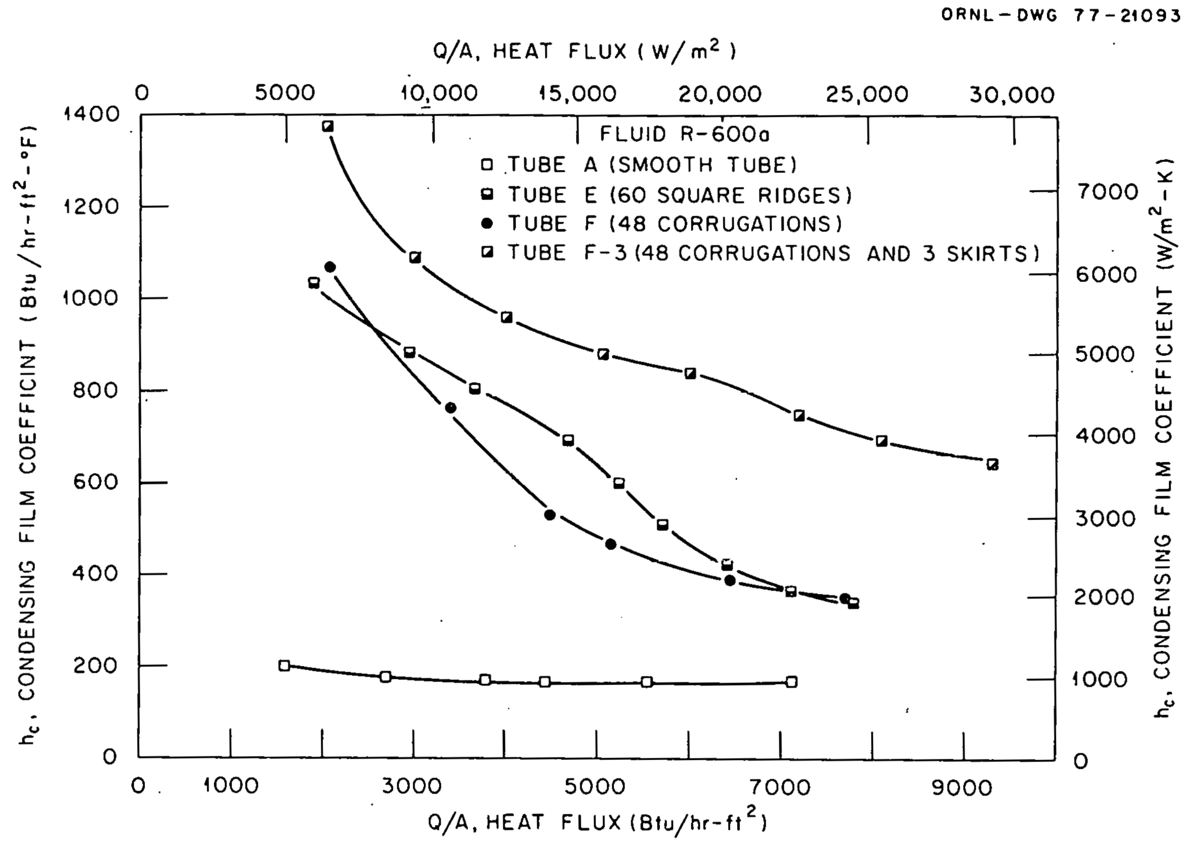


Fig. 13 . Condensation heat transfer coefficient for isobutane using enhanced-surface tubes. Source: J. W. Michel et al., *Energy Div. Annu. Prog. Rep. Period Ending Sept. 1977*, ORNL-5364, Fig. 5.4, p. 185.

composite condensing coefficient (Fig. 13). The condensing coefficients shown are based on the actual area of the tube surfaces. Even though type F tubes gave higher coefficients for condensing ammonia, the fact that type E tubes have more surface area per unit length (Fig. 11) would make this tube outperform the type F tubes with ammonia on a unit-length basis.[17] In measuring the film coefficients when using isobutane, the resistance of the tube walls is negligible in comparison to that of the liquid film. In the case of ammonia, however, the relatively high coefficients of the ammonia add importance to the tube wall conductivity, and as a result the heat transfer performance in those studies is reported in Fig. 2.17 on the basis of a composite coefficient* that includes both the liquid film and the tube wall resistances.[17]

Condensate flowing down the vertical tubes increases the liquid film thickness in the lower sections to cause "flooding" and a marked reduction in the heat transfer performance. This effect is less pronounced when condensing ammonia rather than isobutane, possibly because the higher latent heat of vaporization of the ammonia results in less condensate. The flooding effect when condensing isobutane can be significantly improved by equipping the tubes with drain-off skirts located about every 30 cm (12 in.) along the tube length. The skirts divert the condensate away from the tube wall and result in the improved performance indicated in Fig. 2.18. When this effect is combined with the improvements caused by the enhanced surfaces, the result is strikingly better performance for the system over that obtained with nonskirted smooth vertical tubes.[17]

Fouling of enhanced tube surfaces has a more pronounced effect on the heat transfer coefficient than fouling of smooth tubes. Figure 14 shows the deterioration of the overall heat transfer performance on the enhanced Tube F (with drain-off skirts) and a smooth tube, type A, as the fouling resistance is increased. With relatively clean water and occasional cleaning, the fouling resistance will normally fall in the 0.00009-0.0002 $m^2 \cdot K/W$ (0.0005-0.001 $hr \cdot ft^2 \cdot {}^\circ F/Btu$) range. The effect on enhanced surfaces would be significant if the fouling resistance were about five times higher, as may be encountered with relatively dirty cooling water or infrequent cleaning. The mechanical cleaning systems will thus probably be particularly important for enhanced tube systems. To date, the comparative effects of noncondensable gases on the performance of enhanced- and smooth-surfaced tubes have not been studied extensively.

The enhanced-surface tubing can be formed to U-tubes if filled with a low-temperature alloy during the bending process. An experimental four-pass 40-tube, vertical condenser of U-tube design has been fabricated

*No completely satisfactory method has been developed to compute the wall resistance for a fluted tube.

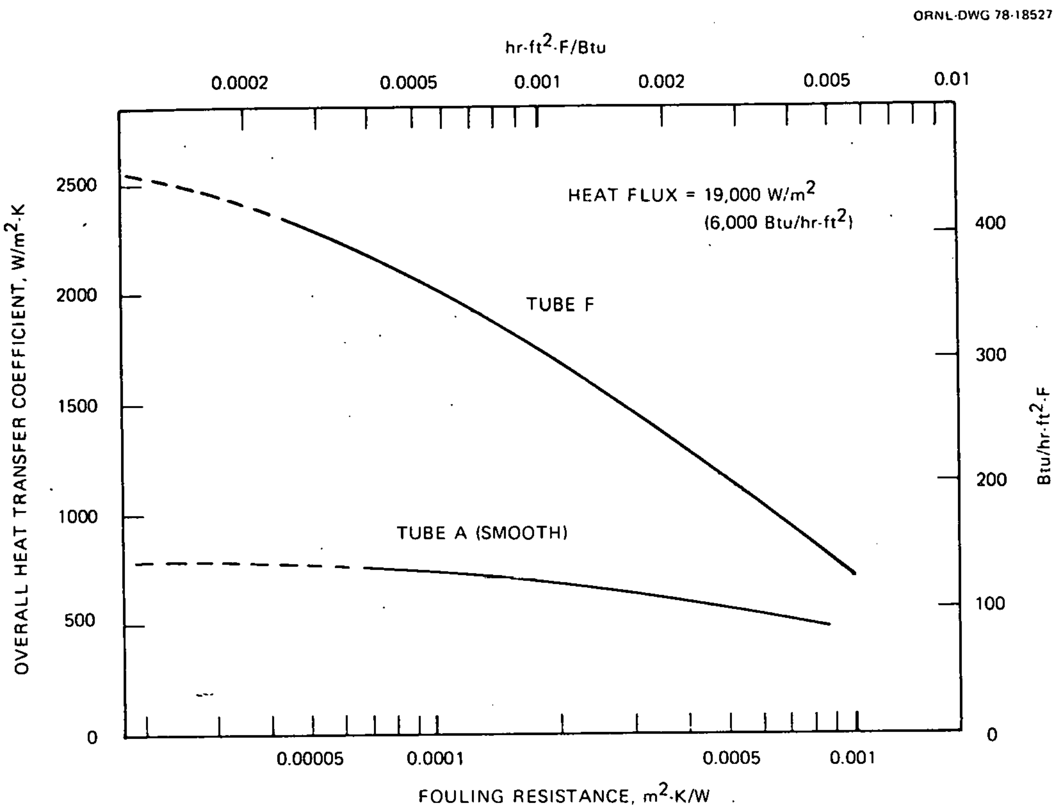


Fig. 14 Effect of fouling resistance on overall condensing heat transfer coefficient of enhanced-surface tubes.

and will be tested at East Mesa in the California Imperial Valley. The unit will condense isobutane containing a small percentage of water vapor and noncondensable gases. The neoprene drain-off skirts are sealed to the tubes with an adhesive that is compatible with isobutane and other materials in the system. The skirts will also serve as cross-baffles to direct the flow of vapor across the tube bundle and to act as vibration dampers.[17]

Manufacturing facilities are currently available for production of enhanced-surface tubing in quantity, generally by extrusion or by drawing. If manufactured in large quantities, the incremental fabrication cost for forming the special surface becomes small. Enhanced-surface tubes contain about 20% more material than plain tubes of the same inside diameter and wall thickness, and large-order prices would probably be in about this same proportion.

COOLING TOWERS

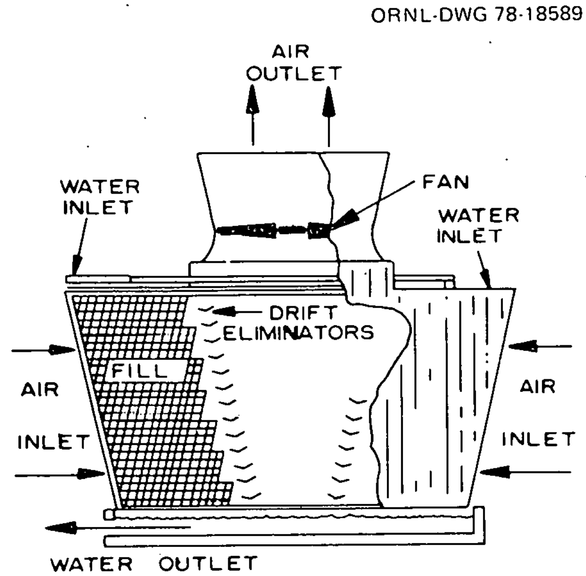
Because of the decline in use of once-through cooling systems, most large power stations installed recently in the U.S. rely on evaporative cooling for waste heat rejection. Although spray ponds and canals have some use, the greater proportion of of evaporative cooling is accomplished in cooling towers. All evaporative, or wet, cooling towers operate on the same basic principle of bringing the water to be cooled into intimate contact with a moving air stream where about 75% of the water-cooling process takes place by evaporation and the remainder by conduction to raise the dry-bulb temperature of the air. The air stream usually leaves thetower very close to the aturated condition.

Selection of the optimum design for a cooling tower is very site specific and is influenced by a large number of factors, such as land area requirements, quality of makeup water, environmental considerations, costs of labor and materials, worth of electricity, capitalization costs, etc. The large number of variables, including the input of historical meteorological data for the site, as well as the iterative nature of the solutions, has led to considerable reliance on computer analyses. The computer models, together with necessary performance information, are considered proprietary by most manufacturers.

Wet mechanical-draft cooling towers

Mechanical-draft cooling towers employ motor-driven fans to provide a positive air flow through the tower fill. The units may be designed for the fans to provide either forced or induced draft, but the latter is more commonly used in larger, present-day towers. The water to be cooled is pumped to the top of the tower, distributed through headers, then falls to the basin at the bottom over a series of splash boards, or slats. The direction of the air flow relative to the falling water droplets may be either countercurrent or cross flow. Typical induced-draft, crossflow and counterflow, wet mechanical-draft cooling tower modules are shown in Fig. 15. The principal advantages of counterflow towers are that the process is more efficient and that they can be adapted better to restricted spaces. The more widely used crossflow type has the advantages of lower air-side pressure drops and more uniform distribution of both airstreams and water streams. Each module, or cell, is a separate unit with its own fan, and the louvered openings are on only two sides — permitting the cells to be arranged side by side in long rows in a so-called rectangular layout. The sloped trapezoidal sides of the tower conform to the path of the falling water profile as it is pulled toward the center by the airstream that is flowing horizontally. This shape eliminates unused fill space and reduces the basin size and cost. The geometry also facilitates ice melting because, with the fans turned off, the warm water will spill down over the louvers.

Internal supports may be redwood, treated fir, concrete, or cast iron. The fill hangers in which the slats are mounted are often glass-reinforced polyester. The splash boards, or slats, may be polyvinyl chloride (PVC), asbestos cement board (ACB), or plastic. The tower basin is reinforced concrete. Concrete may also be used to a greater extent in the future for the support structures. Although they have an initial cost of about 1.5 times that of wood towers, the improved fire resistance of the concrete towers both eliminates the need for sprinkler systems and lowers the insurance rates. The growing trend toward use of concrete and plastics in cooling towers will produce longer life and less maintenance. This aspect is of particular importance in geothermal power stations where the presence of hydrogen sulfide can cause relatively severe corrosion of metal cooling tower fittings. For this reason, stainless steel will also probably be used increasingly for the hardware in cooling towers for geothermal plants.



MECHANICAL DRAFT
CROSS-FLOW TOWER

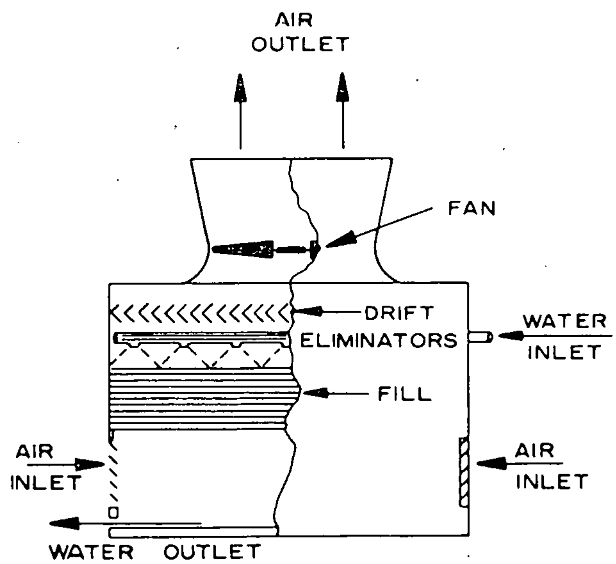


Fig. 15 Mechanical-draft cooling towers. Source: J. D. Holmberg and O. L. Kinney, *Drift Technology for Cooling Towers,* The Marley Company, Mission, Kans., 1973. Reprinted by permission.

Large present-day mechanical-draft towers may have fans with high-tensile-strength fiberglass blades up to 8.5 m (28 ft) in diameter that operate at top speeds of 60 m/sec (12,000 fpm) and are driven by 150-kW (200-hp) motors. A gear reducer and an extension shaft are provided so that the drive motor operates outside the moist airstream. Even larger towers, with 300-kW (400-hp) motors driving fans with blades 12 m (40 ft) in diameter, may possibly be developed in the near future. Each fan discharges into a premolded fiberglass vent stack designed to recover up to about 75% of the velocity head. The air typically exhausts from the stack at about 10 m/sec (30 fps).

The reinforced concrete basin at the bottom of the tower collects the cooled water and acts as a sump for the circulating-water pumps. The basin is sized to hold several hours of inventory in the event that the makeup supply is lost. A drain is provided for removal of silt and is combined with an overflow pipe. Screens are provided at the pump suction.

The drift eliminators, indicated in Fig. 3.7, consist of baffles arranged to change the direction of air flow and catch most of the water droplets entrained in the airstream by impingement on the baffles. More complex baffling provides lower drift rates but adds to the initial cost and the air-side friction losses. Drift rates for mechanical-draft cooling towers are typically in the range of 0.05 to 0.2%, but because of present-day concerns for the environmental impacts of drift deposition, drift rates as low as 0.001% have been specified.

Mechanical-draft cooling towers are typically designed for wet-bulb temperatures of 19 to 28°C (66 to 82°F), ranges of 7 to 17°C (12 to 30°F), approaches of 6 to 11°C (10 to 20°F), evaporation losses of 1.5 to 2.5%, drift losses of 0.02 to 0.2%, and blowdown rates of 0.5 to 3% of the water circulation rate.[18] The wet-bulb temperature is the most important parameter in designing a wet cooling tower for a given site. Because the maximum wet-bulb temperature that has historically occurred at a given location will probably exceed greatly the average value, designing for the maximum would result in an oversized tower. Selection of a design wet-bulb temperature that will not be exceeded more than 2 or 3% of the time is customary. Designing for the minimum wet-bulb temperature can also be considered, as will be discussed subsequently.

Multiple-cell towers are versatile in that they can be designed for a broad range of capacities by varying the number of cells used. A broad variety of cooling system requirements can be met by varying the fan speed and other operating parameters. Drift-rate specifications can be met by arrangement of the drift eliminators and fill-packing materials and configurations; fill height and packing depths can be optimized for specific design requirements. The modular arrangement of mechanical-draft towers has another distinct advantage in that, if the installed capacity is later demonstrated to be inadequate, it is relatively simple to add more cooling capacity.

The air–water vapor mixture leaving mechanical-draft cooling towers may at times be cooler than the air entering; that is, the plume from the tower may have negative buoyancy. The orientation of the tower with respect to the prevailing wind direction can have an important influence on recirculation, areas of drift deposition, icing, and ground-level fogging. In general, recirculation will be at a minimum if the row of cells is at right angles to the wind-direction and located well away from structures, trees, or terrain features that could restrict or deflect air movements into and away from the units. Recirculation will tend to be at maximum when the wind is blowing along the line of towers. On the other hand, larger plumes tend to be lofted to higher altidudes than smaller ones because of the entrainment effect, and a wind blowing along the line of cells tends to combine the effluent into a larger plume. This effect may help with problems of ground-level fogging during certain months of the year but may or may not help drift deposition problems in that it may be preferable for the major amount of the drift to be deposited within the plant boundaries rather than on public or private lands.

Tower-induced icing and snowfall effects on plant structures, nearby highways, etc., must also be taken into consideration when siting and orienting the cooling towers. Because the ground effects of the relatively short mechanical-draft towers are greater than those of the much taller natural-draft towers, the mechanical-draft types may have to be located at a greater distance from the turbine building and perhaps spread over a greater area, resulting in a proportionate increase in land and piping costs.

The construction time for large wet mechanical-draft cooling towers may be about six months. Field erection costs can vary widely depending on the tower size and geographical location but may be roughly estimated at about one-fourth of the total tower cost. Fire protection systems, if required, can add about 15% to the capital cost. In 1974, some adjusted costs of large mechanical-draft towers were reported as varying from about $1 to $2.23 per kilowatt of fossil-fired steam station capacity.[19] If adjusted to a geothermal station having only 10% thermal efficiency and to 1978 costs with an average inflation rate of 7% per annum, the costs could range between about $10 to $20 per kilowatt of installed capacity.

Circular mechanical-draft cooling towers

A relatively recent configuration for wet cooling towers is the cross-flow, circular mechanical-draft type developed by Marley. An overall view and a half-section of this type of tower are shown in Fig. 16. The tower fill is arranged in a large annular area, as in a natural-draft tower, with an array of induced-draft fans clustered in the center to replace the chimney of the natural-draft type. The advantages of the arrangement over a rectangular mechanical-draft layout are that recirculation effects are reduced and that the plume from the circular tower will be lofted to greater heights to reduce ground-level fogging and drift problems. The circular layout also will probably reduce the ground area needed, circulating-water piping costs, and fan power requirements. The advantages of the circular tower over the natural-draft type include a lower profile that makes it less visible, a lower capital cost, and probably a greater ability to resist seismic disturbances. It will also operate in meteorological conditions that would not be tolerable for a natural-draft tower.

As with natural-draft towers, the circular mechanical-draft type will be most economical in larger sizes. The first full-scale installation

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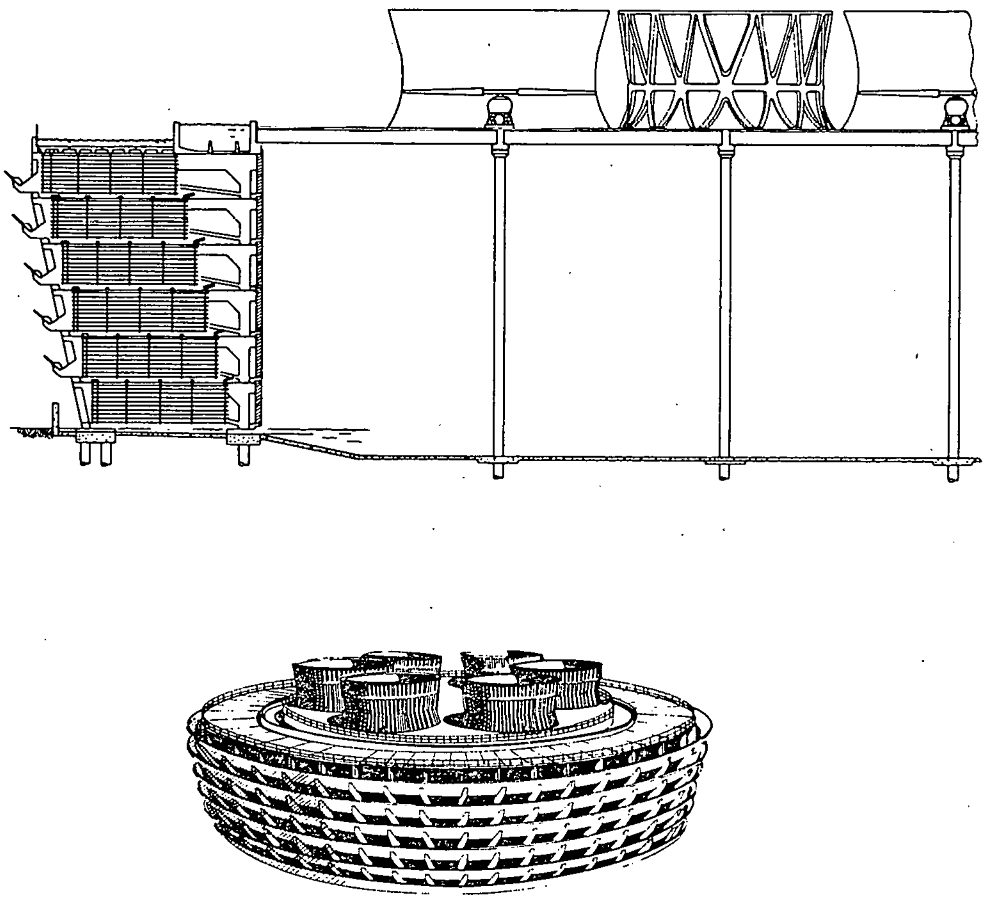

Fig. 16 Circular mechanical-draft cooling towers. Source: *Round Towers,* Publication RT-75, The Marley Company, Mission, Kans. Reprinted by permission.

of this tower type was in 1973-74 at the 500-MW(e) Jack Watson Station of the Mississippi Power Company at Gulfport. The plume behavior of the operating tower has conformed well with the predictions obtained from model tests.[20] This circular type of tower has been specified for several power station projects now planned or under construction. The costs of the towers will probably not be significantly greater than that of mechanical-draft towers in a rectangular layout, particularly when land and piping costs are considered.

The circular mechanical-draft cooling tower will probably find application for large geothermal power stations. It will function in arid regions where the natural-draft type will not, may help with drift problems at certain sites, and will offer economies in land costs and fan power consumption.

Wet natural-draft cooling towers

Natural-draft cooling towers may have limited use in geothermal power plant applications because they are best suited for very large installations and they do not function well in climates having high wet-bulb temperatures and low relative humidities, such as occur in the Imperial Valley region of southern California. The seismic activity and ground subsidence tendencies often associated with geothermal sites may also be a deterrent to their use. Nevertheless, because a 100-MW(e) geothermal power plant may reject as much heat as a 600-MW(e) fossil-type station and potential geothermal resources are not necessarily confined to arid, seismically active regions, natural-draft cooling towers cannot be completely eliminated from consideration.

As in mechanical-draft towers, there are two basic types, the crossflow and the counterflow. In the former, the fill or packing, is located in a ring outside the base of the tower and the inside serves primarily as a chimney. In the counterflow type, the fill is inside the base and elevated so that the air entering around the periphery moves upward through the falling water droplets. The counterflow design provides more efficient heat transfer because the coolest water contacts the coolest air. In the crossflow arrangement, the air and water are distributed more uniformly, and there is less air-pressure drop across the fill. Sketches of both types of towers are shown in Fig. 17.

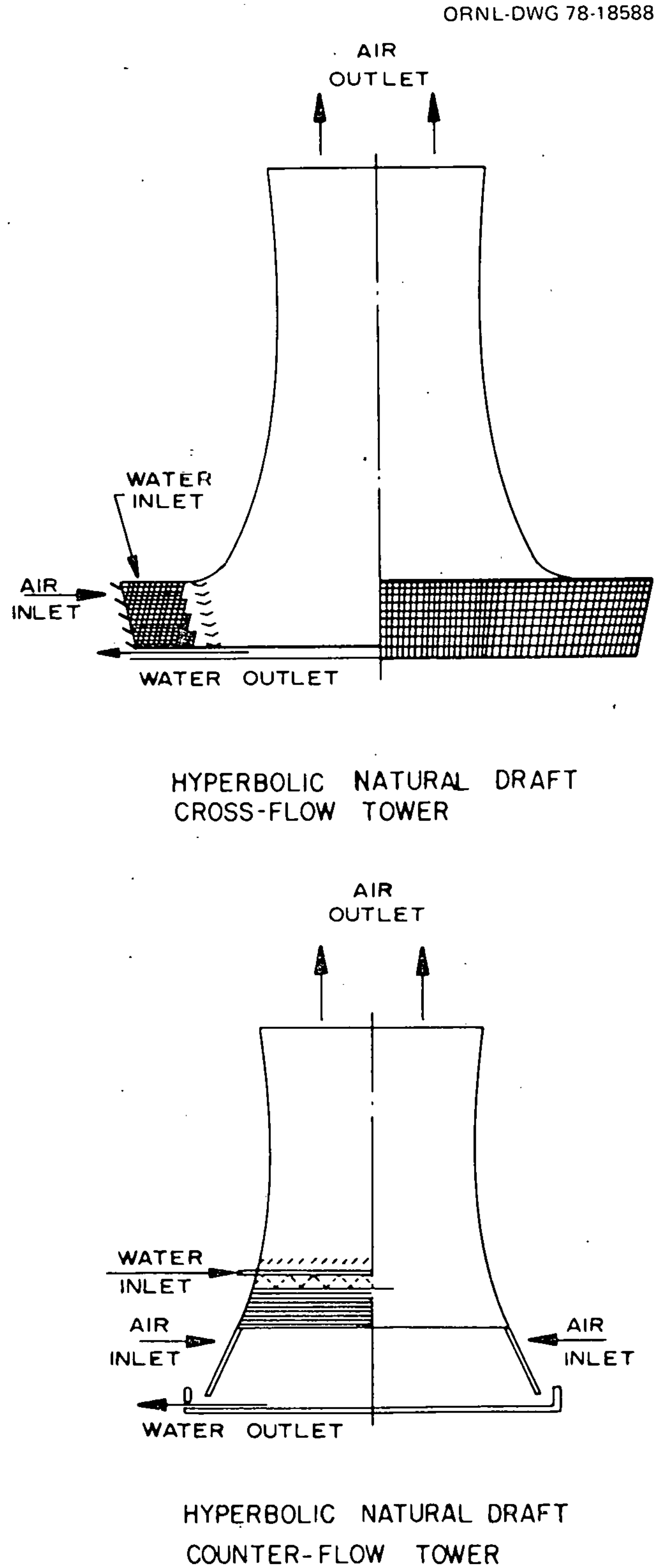

Fig. 17 | Wet natural-draft cooling towers. Source: J. B. Dickey and R. E. Cates, *Managing Waste Heat with the Water Cooling Tower*, 2d ed., The Marley Company, Mission, Kans., 1970. Reprinted by permission.

The hyperbolic shape of a natural-draft cooling tower matches the natural air flow through the unit, but, perhaps more importantly, it has strength characteristics that permit economizing on materials of construction. The shell of a natural-draft cooling tower may be fabricated of reinforced concrete, of structural steel with aluminum skin, or, as at Schmehausen, Germany, of a suspended cable net with an aluminum skin.[21] Reinforced concrete has been used almost exclusively in the United States. Concrete shells are poured by leapfrogging forms and construction platforms up the structure and usually require about two years for completion. The shells must be designed for the dead weight of the tower plus loads resulting from buckling, vibrations, wind, seismic forces, and thermal stresses. Induced stresses due to poor subsoil or nonuniform settling must also be considered and may take on more importance in geothermal areas subject to subsidence. The shells are surprisingly thin and, in some towers, are only about 15 cm (6 in.) in thickness at the waist. Models have been developed for analysis of the membrane stress to minimize the shell thickness required.[22] The design wind load is usually 161 km/hr (100 mph).

The shells of natural-draft towers are more subject to seismic damage than mechanical-draft towers, particularly those of wooden construction. Micro-earthquakes, that is, earthquakes with magnitudes of less than 4 on the Richter scale, have been observed near many geothermal areas around the world, including The Geysers and the Imperial Valley. However, earthquakes having magnitudes greater than 4.5 and the potential to cause significant surface damage have rarely been observed in geothermal areas. One possible explanation is that the frequent micro-earthquakes in geothermal areas may tend to relieve regional tectonic stress, thus reducing the possibility of a major earthquake. Earthquakes have been associated with the injection of fluids into oil fields, and it is hypothesized that similar events could occur as a result of geothermal development. Much remains to be learned in this area, and detailed seismic monitoring is being conducted at The Geysers and in the Imperial Valley.[23]

A variation of the natural-draft tower is the fan-assisted type. The tower fill arrangement is much the same as for a counterflow type, but fans are arranged around the periphery to force air into the tower and thereby reduce the stack height required. The capacity of the tower becomes less dependent upon the meteorological conditions, and the tower height may be less obtrusive. The reduced tower height makes the unit perform in a manner similar to the circular mechanical-draft type discussed previously.

Concentration relationships in wet cooling towers

The rate that makeup water must be added to a wet cooling tower is the sum of the rates at which water is lost from the tower system by evaporation, blowdown, drift, and leakage. The blowdown rate determines the level to which the concentration of dissolved solids in the water are allowed to accumulate. The concentration factor, or cycles of concentration, is defined as the ratio of the total dissolved solids (TDS) in the circulating (or blowdown) water to the TDS in the makeup water. Where the environmental impacts of the blowdown may be a serious problem and ample makeup water is available, the concentration factor may be as low as 2 to 4. The proposed Sundesert Nuclear Station near Blyth, California, was an example of the other extreme, where even low-quality makeup water would have been scarce. The blowdown would have been so laden with salt that it was to have been disposed of by evaporation. A concentration factor of about 20 was proposed, and side-stream clarifiers were to have been used in the circulating system. The Sundesert conditions may be typical of those for large geothermal power plants located in the Imperial Valley of southern California.

The relationships between the mass flow rate per unit time of makeup, blowdown, evaporation, and drift rates can be formulated in terms of the concentration factor (on a weight basis).

Let:

M = makeup rate,

B = blowdown rate,

E = evaporation rate,

D = drift rate,

C = concentration of solids in stream, lb/lb,

CF = concentration factor, or cycles of concentration.

Then, if leakage is ignored,

$$M = B + E + D \ ,$$

$$CF = C_B/C_M \ ,$$

$$C_M M = C_B B + C_B D \ ,$$

$$C_M = C_B(B + D)/M \ ,$$

$$CF = C_B/[D_B(B + D)/M] \ .$$

It then follows that

$$CF = M/(B + D) \ ,$$

$$B = [E/(CF - 1)] - D \ ,$$

$$M = CF[E/(CF - 1)] \ ,$$

$$E = (CF - 1)(B + D) \ .$$

3.5.9 Drift from wet cooling towers

Drift is the portion of the spray pond or cooling tower circulating water that becomes entrained in the moving air stream and is carried out of the system in droplet form. The amount of drift is commonly expressed as a percentage of the circulating-water flow rate on a weight basis. However, the amount of drift from a tower is not a strong function of the water loading but is primarily dependent on air flow rates and velocities. Drift may also be expressed as the weight of the droplets per unit weight of air, in parts per million (ppm):

$$\text{ppm} = \frac{\%\ \text{drift}}{100} \times 10^6 \times \frac{L}{G}\ ,$$

where L/G is the weight ratio of the water and air flow rates. Cooling tower drift rates vary over a wide range, from about 0.001 to 0.2%. At a typical L/G ratio of 1.5, this amount is equivalent to 15 to 3000 ppm of drift in the air stream.

The percentage of drift is small, and the water lost from the tower due to drift is insignificant in comparison to that lost by evaporation. Drift does not contribute significantly to the visibility of cooling tower plumes. The primary concerns with regard to drift are the environmental impacts of the salts that are dissolved in water droplets being deposited in the vicinity of the cooling tower, and the possibility for icing of nearby roads and equipment during the winter months. Salt deposition due to drift, for example, in the power station electrical switchyard can affect insulator efficiency and accelerate corrosion. A frequent complaint is salt spotting of windows and finishes of automobiles parked near the cooling towers. More importantly, some forms of vegetation may be damaged by the chlorides and other salts dissolved in the circulating water. Plants differ markedly in the amount of salt they can tolerate at various times in the growing season. This aspect is of considerable concern and is often one of the environmental impacts that is an issue in the granting of construction permits for power plants using spray ponds or cooling towers for waste heat rejection.

The magnitude of the drift problem can be appreciated by considering that if a cooling tower for a 50-MW(e) geothermal power station had a thermal efficiency of 10%, an 11°C (20°F) water temperature range, cooling tower makeup water having 500 ppm TDS, a concentration factor of 5 in the towers, a tower drift rate of 0.005%, and a plant capacity factor of 0.8, then a total of about 3.0×10^4 kg (6.7×10^4 lb) of salt and other dissolved solids would leave the tower each year and be deposited within a few miles of the tower. If the wind tends to be from the same quarter all year, the annual deposition rate would be concentrated in a smaller area than if the wind direction were more variable. Maximum deposition rates tend to be within about 1.6 km (1 mile) of mechanical-draft towers and can amount to 10 or more $g/m^2 \cdot year$ (100 lb/acre·year). As a general rule, greater amounts cause concern for the biota, depending on the kinds of vegetation involved, the season of the year, and the amount of rainfall available to wash the plants and to leach the salt from the soil.

During the past few years, improvements in techniques for measuring drift confirmed that actual drift rates from towers were substantially below the values of about 0.2% that manufacturers had been citing up to that time. Guaranteed drift rates of 0.05% are now commonplace, and values as low as 0.001% may now be specified. Improvements in measurement techniques have also contributed to the drift eliminator designs in that the performance of various arrangements can be more meaningfully tested.

Dry cooling towers

In dry cooling towers the heat to be rejected from the power cycle is transferred through the walls of an air-cooled heat exchanger to raise the dry-bulb temperature of the air stream. Power cycles using dry cooling towers are illustrated in Figs. 4 and 5.

Unfavorable economics in the past have resulted in relatively little interest in dry cooling towers in the United States. Rather than use dry towers where conventional sources of water are not available, the electric utilities have found that is generally more economic to find water by some means (such as buying agricultural land for the water rights, piping sewage treatment plant effluent, using salt water, or pumping groundwater). The growing scarcity of even these water sources, however, and the unacceptable environmental impacts that may be attendant to their use are now drawing more attention to use of dry cooling methods. The Department of Energy (DOE) has sponsored several studies of both dry and wet/dry towers. These subjects cover the state of the art, economics, and projected future needs in light of available water supplies in the United States.

Dry cooling towers have been used in Europe for 15 years or more, as listed by Miliaras.[24] In the United States, the Wyodak plant — a joint project at Gillette, Wyoming, of the Black Hills Power and Light Company and the Pacific Power and Light Company — is the most notable example at the present time. This 330-MW(e) station is the largest in the world using dry cooling towers. A power station using a dry air-cooled condenser supplied by CE-Lummus, has also been completed recently at Valdez, Alaska.[25] Dry cooling towers are of two types, direct and indirect.

In the direct arrangement, the turbine exhaust is ducted to an extended-surface, air-cooled heat exchanger. In conventional steam-power cycles, the condensate is collected and returned to the boiler via the feedwater heating system. When the vapor is condensed at pressures less than atmospheric, provisions must be made for purging of noncondensable gases. If the condensing vapor is steam, then the exhaust ducting must be relatively large [e.g., about 2 m (6 ft) in diameter for a 40-MW(e) turbine] to keep exhaust pressure-drop losses acceptably low but, at the same time, sufficiently flexible to accommodate the pipe stresses. Mechanical-draft is used on most dry-type towers constructed to date, but studies have indicated that, although the initial costs are more than twice that of mechanical-draft types,[26] natural-draft towers may also be feasible.

The air-cooled coils are usually made up of parallel-connected banks (commonly arranged in a "W" form) with the fans located underneath. The vapor is distributed to the heat exchangers through headers, which may be arranged to deliver the steam either to the top or to the bottom of the extended-surface coils. With top delivery, in what is called the uniflow arrangement, the condensate flows down the inside walls of the tubes in the same direction as the vapor flow, and the pressure drop caused by the flow will result in a saturation temperature for the condensate several degrees below the entering vapor temperature. With the vapor delivered to the bottom of the exchangers, in a counterflow arrangement, the vapor flows upward against the down-flowing condensate. Although the condensate leaves at the entering steam temperature, the counterflow arrangement is not as efficient in terms of heat transfer and requires more area. A combination of the uniflow and counterflow arrangements can also be used.

In the indirect system, the exhaust vapor from the turbine is condensed by a coolant circulated through the condenser and to the dry cooling tower where the heat is transferred in extended-surface, air-cooled heat exchangers. The condensers can be either the surface or direct-contact type. An indirect system using a direct-contact condenser, called the Heller system, is illustrated in Fig. 4. Although a surface condenser could be used with the indirect system, all units up to the present time have used direct-contact condensers in a Heller-type system. This usage permits a condensing steam temperature lower by the amount of the terminal temperature difference that would have been required for the surface condenser, resulting in lower plant heat rates and capital costs. However, a 1973 study made by Beck and Associates[27] for 1000-MW(e) fossil and nuclear power plants, in which both mechanical-draft and natural-draft dry cooling towers were considered, indicated that the cost of producing electricity using surface condensers was competitive with the direct-contact condensers and that the surface condensers offered significant operating advantages. Although the study was based on plant sizes much larger than those presently considered for geothermal power installations, on meteorology typical of the eastern United States, and on fuel costs that are no longer applicable, the factors involved in the assessment are of interest.

To maintain air in-leakage into the working fluid (steam) within practical limits, the water that is inside the dry cooling coils should operate above atmospheric pressure. When using surface-type condensers, this requirement can be easily accomplished. When using direct-contact condensers, however, the condenser effluent must be pumped up to pressure for passage through the air-cooled surfaces and then let down in pressure before it is sprayed into the direct-contact condenser (Fig. 1.4). The pressure reduction can be either through a throttling valve or through a hydraulic turbine direct-coupled to the condensate pump where about 80 to 90% of the pressure head can be recovered. However, this aspect adds to the cost and complexity of the direct-contact condenser system. A further operating advantage of the surface condenser is that, because the coolant and the working fluid do not mix, it can be used in cycles where the working fluid is not water or where a fluid other than water is used as the coolant. For example, a glycol solution can be used as the coolant to protect the air-cooled coil from freezing. The Beck[27] study considered a 30% (by volume) inhibited ethylene glycol solution and an increase in electrical production costs of 0.07 to 0.17 mills/kWhr over the costs when using plain water. Ammonia has also been considered as a heat transport fluid (to be discussed subsequently).

One of the distinguishing features of all dry-type cooling towers is that the temperature at which the turbine exhaust steam is condensed is dependent on the ambient dry-bulb temperature rather than primarily on the wet-bulb temperature, as in evaporative-type cooling towers. Typically, the condensing pressure for a dry-type cooling tower system will be in the order of 203 mm (8 in.) of mercury absolute as compared to 64 to 102 mm (2.5 to 4 in.) of mercury absolute for a wet cooling tower. The high back-pressure will necessitate a different design for the turbine if the size of the tower and the capital costs are to be kept to a minimum. Although U.S. manufacturers state that they could supply high back-pressure turbines , they report that there has been little interest to date in them.[26]

Also resulting from the dependency of dry cooling towers on the dry-bulb temperature of the ambient air is that one must either (1) design for high dry-bulb temperatures and operate at efficiencies significantly less than the plant would be capable of if designed for lower dry-bulb or (2) suffer a loss in capacity when the ambient dry-bulb temperatures exceed the design value. This effect is particularly disadvantageous in regions where air-conditioning loads cause peaks on the system to occur at times of maximum dry-bulb temperature. Two ways to offset this situation are to use a combination of wet and dry cooling towers (Sect. 3.5.7) or to wet the outside surface of the dry cooling tower during periods of high dry-bulb temperature and peak loads. Both ways require the consumptive use of water during the hot season when water supplies tend to be restricted.

The prospect of future electric power generation depending heavily on dry cooling towers has led to studies of methods of improving the performance and reducing the cost of the extended-surface heat exchangers. These studies have included methods for forming extended metal surfaces, such as the Curtis-Wright configuration with which it is claimed that manufacturing costs are substantially less and that a lowered pressure drop of the fluid through the exchanger can lead to significant savings in pumping costs.[28]

As mentioned above, a surface condenser used in conjunction with a dry cooling tower offers the possibility of a fluid other than water being used as the heat transport medium. A study by Franklin Institute, Battelle-Northwest, and Union Carbide under grants from DOE and the Electric Power Research Institute (EPRI) determined that ammonia was the most economical transport fluid.[25] The system would condense the turbine exhaust steam by use of ammonia, which would be evaporated in the process. The ammonia vapor would then be condensed in a dry cooling tower and recycled. The arrangement with the lowest cost utilized water deluge on the tower during the hottest ambient conditions to effect water savings of 80% over the amount of makeup water required for an all-evaporative cooling system. The estimated cost of the system is 15 to 30% lower than for a conventional wet/dry tower arrangement.[28] The major advantages of the ammonia system are that (1) less surface area is required, reducing the tower size and cost; (2) smaller transfer lines are needed between the tower and the turbine condenser; (3) freezing problems in the air-cooled exchanger are eliminated; and (4) no pumping power is required to move the vapor from the condenser-boiler to the tower and very little power is needed to return the liquid ammonia for recycle.

Wet/dry cooling towers

Wet/dry cooling towers combine evaporative and dry cooling of the circulating water. A typical wet/dry tower with the air flowing in parallel through the wet and dry sections is shown in Fig. 18. (Wet/dry cooling towers are sometimes reffered to as parallel-path towers.) The dry portion handles essentially all the cooling load when dry-bulb temperatures are low, and the wet portion serves as supplemental capacity when needed, such as in the summer months when dry-bulb temperatures are high. Wet/dry towers have a significantly higher cost than a wet or dry tower alone but offer the following advantages:

1. The visibility of the effluent plume can be controlled and essentially eliminated, if desired.

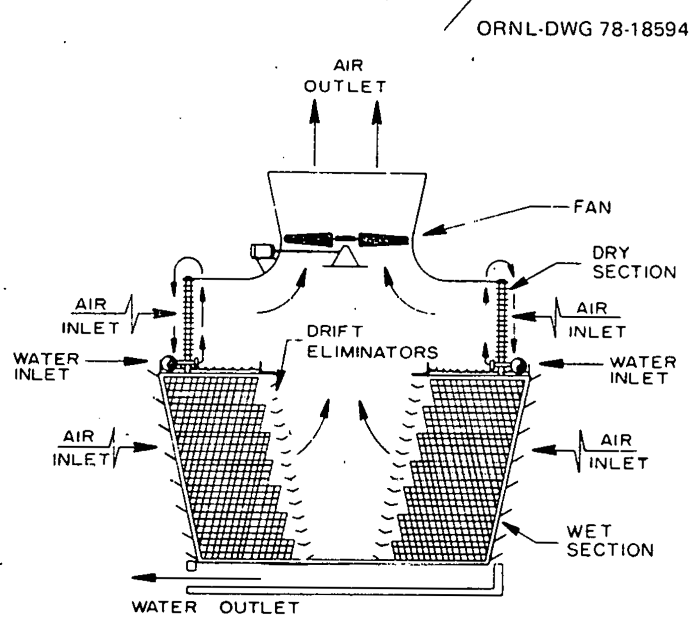

Fig. 18. Wet/dry mechanical-draft cooling tower. Source: J. D. Holmberg and O. L. Kinney, *Drift Technology for Cooling Towers*, The Marley Company, Mission, Kans., 1973. Reprinted by permission.

2. The water temperature leaving the cooling tower can be maintained at a sufficiently low value year-round to permit use of a low back-pressure turbine.
3. The evaporative loss of cooling water can be reduced substantially below that required if wet cooling alone were used.
4. The system is better able to meet peak loads than if dry cooling towers alone were used.

The relative importance of these factors is site specific, but at most geothermal power installations, the water consumption rate and the ability to meet summertime peak loads will probably be the major considerations.

Wet/dry towers may be arranged with the circulating water flowing in parallel or in series; in the series arrangements, either the wet or the dry section can be upstream of the other. The performance of the various arrangements can be compared in terms of cooling capacity, water evaporation rates, capital and operating costs, and environmental impacts. Although all these figures of merit are important, it seems certain that water conservation will have priority at most geothermal power installations in the Southwest. To obtain the lowest evaporation rates, the dry cooling section should carry as much of the heat rejection load as possible. A greater mass flow rate of air is needed through the dry section than through the wet section; a series flow path for the air results in a significant mismatch in this regard. Further, a greater initial temperature difference (ITD) is obtained when air at ambient conditions enters both the wet and dry sections, thereby increasing the capacity and minimizing the evaporation rate. A parallel path for the air flow is thus a common practice for wet/dry cooling towers.

Conclusions cannot be as readily drawn with regard to the water path. Loscutoff[29] studied this aspect, and although his work was preliminary and concerned primarily with comparative performances and evaporation rates, the principles involved are of significance. Loscutoff[29] concluded that, from the standpoint of capacity, series flow in the water path is better than parallel flow. Also, with series flow the capacity is slightly better if the dry section comes first when ambient dry-bulb temperatures are low and slightly better if the wet section comes first when ambient wet-bulb temperatures are high. The differences were very small, though, and do not appear to be conclusive. The evaporation rates, however, are significantly less with the series arrangement and with the circulating water flowing first through the dry section.

With the series flow path, if the full flow of the circulating water is first through the dry section and then the wet section, the temperature range for the water in the wet section will be smaller than ordinarily achieved in wet cooling towers. By reducing the size of the wet tower and increasing the range by bypassing a portion of the water flow around it, there would generally be a cost savings.

An advantage of the parallel-path arrangement is that, by use of separate or dual-service condensers, the coolant flow paths through the dry and wet sections can be kept entirely separate. Some considerations in this regard are as follows:

1. The relatively dirty, oxygen-laden, and chemically treated water from the wet cooling towers will present more corrosion problems to the materials commonly used for extended-surface heat exchangers than would the water that could be circulated in a closed loop through the dry coils. An antifreeze solution or other coolant fluid could be circulated through the dry section.
2. If separate or dual-service condensers are used, the condenser tubes cooled by the fluid circulated in the closed dry section loop will be significantly less subject to fouling. (Nitrogen-blanketing is sometimes used to protect extended-surface heat exchangers from corrosion when the system is drained for shutdown).
3. The additional circulating pumps and piping required for separate systems is offset to some extent by the by-pass and flow-regulating valves needed in a series-connected system.
4. In a parallel-connected system, the greater flow through the dry section when the wet section is removed from service will improve the water-side heat transfer coefficient, lower the cooling range, and increase the temperature difference in the condenser. One result of this is that the wet tower can be removed from service at a higher ambient dry-bulb temperature and thus conserve makeup water in the wet section.

Deluge cooling for dry towers.

A variation of the wet/dry tower, which has promise as a method of increasing the capacity of dry cooling towers during peak loads or periods of high ambient dry-bulb temperatures, involves the wetting of a portion of the dry surface. This method, called deluge cooling, is a compromise between wet cooling and dry cooling methods and minimizes water consumption while reducing the performance penalties associated with all-dry systems.

In deluge cooling, an excess of water is used to wet the surface, and the runoff is collected and recirculated. The transfer of heat is greatly enhanced. Factors of 2.5 are cited as commonly attainable,[30] but much higher factors are mentioned for certain conditions.[40] The wetted coils act much as a wet cooling tower, and although the heat transfer augmentation is greatest for higher air flow rates, it may be advantageous to reduce the rate of air flow to resemble more closely the *L/G* ratios commonly used in wet towers. There are added problems from corrosion and deposition of solids on the intermittently wetted surfaces, however, and these have yet to be resolved satisfactorily.

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Wiles et al., of Pacific Northwest Laboratory[31] have made a detailed cost analysis of a deluge wet/dry cooling system. In the system studied, ammonia was selected to transport the heat from the turbine exhaust to the ambient air. The system uses a horizontal-tube, vertical, plate-finned heat exchanger / The cost savings using deluge cooling result primarily from specifying a smaller heat exchanger and from lower turbine exhaust pressures (Fig. 3.13).

The potential advantages of deluge cooling in achieving the water conservation of dry towers without the high cost of wet/dry towers and in meeting peak load conditions appears sufficient to warrant continued research and development in this area.

PHASED COOLING

In power stations with evaporative cooling ponds or towers having a significant diurnal difference in cooling capability, significant reductions in water consumption may be possible by storing a portion of the warmed circulating water for nighttime heat dissipation. This arrangement has been termed "phased cooling."

MacFarlane, Goodling, and Maples[32] studied a phased-cooling concept in which two storage ponds were proposed, one to store cooled water to supply the condenser during hot daytime temperatures and the other to store the effluent warmed water. During the night, when conditions are more favorable for cooling, the warmed water is cooled and returned to the first pond. It was concluded that phased cooling might cut evaporation losses about in half, depending on the storage capacity provided in the ponds. The analysis was based on meteorological conditions in the southeastern United States, and allowance was made for about 1.3 m/year (52 in./year) of rainfall into the ponds and into the evaporation basin used for heat dissipation.

With phased cooling, essentially the same total amount of heat must be dissipated, but by postponing the heat release, the duty of the cooling equipment is increased accordingly. Costs, however, are not necessarily increased in direct proportion because of the size factor and because of the advantageous nighttime cooling conditions. Depending on the availability of land and on other construction costs of the water storage basins, phased cooling might reduce water consumption with less expense than would be required for either wet/dry or dry cooling towers. Phased cooling would have another important economic advantage in that the stored cooled water would increase the station's ability to meet peak daytime loads.

VARIABLE, OR "FLOATING," POWER COOLING CONCEPT

A power plant is customarily designed for the highest constant power output that can be assured year-round, as determined by the capacity of the heat rejection system at reasonable worst-case ambient conditions. However, in some special cases it may be more economical to design for a power output that will vary as the capacity of the waste heat rejection system is affected by ambient conditions. The cooling system would operate at full capacity all the time, and the turbine exhaust pressure would vary to allow the turbine to generate the maxiumum amount of power possible under the particular circumstances. In effect, the system would be designed for lowest wintertime ambient temperatures rather than highest summertime temperatures. Although something of a misnomer, this concept of variable power output has been termed "floating" power.

If the equipment is sized for the maximum attainable power output, it will operate at only partial capacity much of the time. The penalty in capital cost charges will, however, tend to be offset by the greater amount of power produced. Advantages of the variable power output concept will be greater at plant locations where there are wide swings in the available cooling-water temperature as a result of diurnal and seasonal changes. Use of dry cooling would enhance this effect because the water temperature would be dependent on the ambient dry-bulb rather than wet-bulb temperature. The advantages also tend to be greater for plants with higher average condensing temperatures, which, again, is a characteristic of systems using dry cooling. The concept also favors power cycles that have only moderate thermal efficiencies, such as those dependent on relatively low-temperature heat sources. Geothermal power stations using dry cooling are very likely candidates for the variable power output concept.

Pines, Green, Pope, and Doyle[33] made a study of variable power output for a 50-MW(e) binary geothermal power station concept for Heber, California. Both evaporative and dry cooling systems were considered. The study made use of the computer model GEOTHM to optimize the system design parameters in each case for minimum electrical energy production costs. The power production costs of base-loaded (constant power output) plants were compared to variable power output plants. The equipment efficiencies and costs used as input parameters were obtained by applying scaling factors to information taken from a study made for the Electric Power Research Institute (EPRI) by Holt-Procon[34] of a 50-MW(e) binary-cycle station located at Heber. Isobutane was used as the working fluid, and the dry cooling concept assumed direct condensation of the isobutane in an air-cooled coil. It was assumed that the additional power generated by the variable power output concept had the same monetary worth as that generated by the base-loaded plant. Seasonal shifts in the dry cooling cycle performance, based on the meteorology at Heber, were taken into account.

The results of the study are summarized in Fig. 19 and 20. In comparison to the 50 MW(e) of net power produced by the base-loaded plant

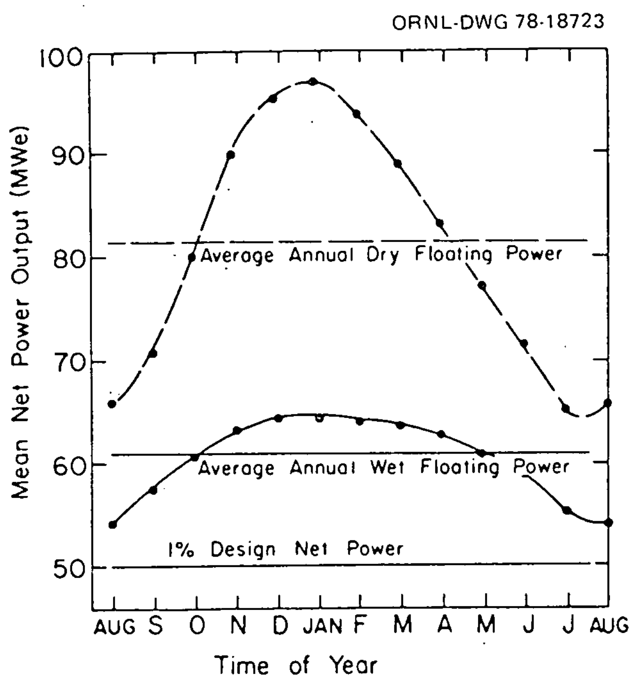

Fig. 19 Seasonal variation of net output of "floating" evaporative and dry-cooled binary geothermal power plants at Heber, California, which would generate 50 MW(e) (net) if designed on 1% basis. Source: H. S. Pines et al., *Floating Dry Cooling, A Competitive Alternative to Evaporative Cooling in a Binary Cycle Geothermal Power Plant*, LBL-7087, Lawrence Berkeley Laboratory (July 1978), paper presented at the American Society of Mechanical Engineers' 1978 Winter Annual Meeting at San Francisco, December 10-15, 1978. Reprinted by permission.

(based on design conditions that would exist 99% of the time), the average annual net power output of the variable power concept using wet cooling towers is about 61 MW(e). When using dry cooling, the annual average net output is about 81 MW(e), as shown in Fig. 5.1. The average power output is greater than 50 MW(e) because the monthly mean ambient temperatures are always lower than the design value. The electricity production (busbar) costs, compared in Fig. 5.2, show about a 10% lower cost for the variable power-output concept. The study concluded that geothermal power stations operating on the binary cycle and using dry cooling can be as economical as base-loaded plants using wet cooling towers.

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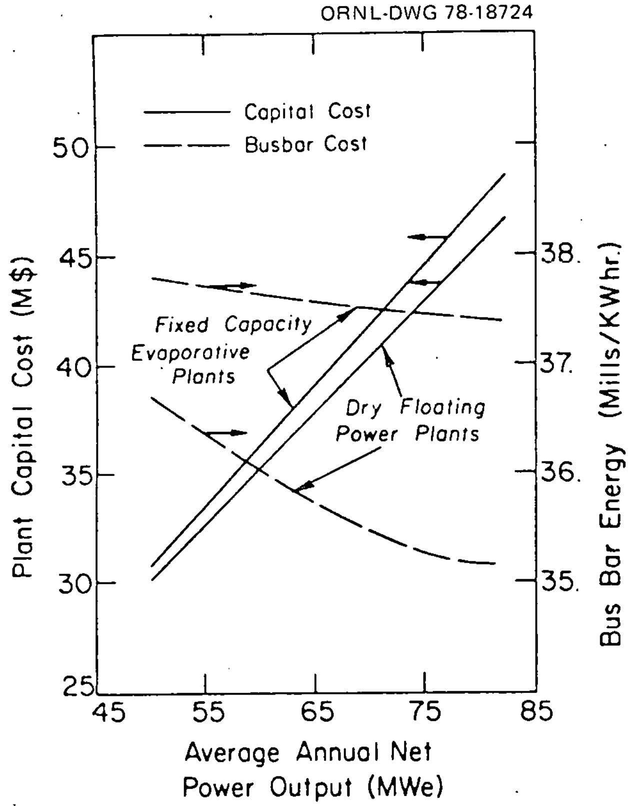

Fig. 20 Plant capital costs and busbar energy costs of base-loaded evaporative-cooled and "floating" dry-cooled plants as a function of the annual average power output. Source: V. W. Roberts, *Geothermal Energy Conversion and Economic Case Studies*, EPRI-301, Holt/Procon (November 1976). Reprinted by permission.

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