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MASTER

Waste Heat Rejection from Geothermal Power Stations

Roy C. Robertson

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National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
Price: Printed Copy $8.00; Microfiche $3.00

ORNL/TM-6533
Dist. Category UC-66d

Contract No. W-7405-eng-26

ENERGY DIVISION

WASTE HEAT REJECTION FROM GEOTHERMAL POWER STATIONS

Roy C. Robertson

Date Published: December 1978

Prepared as a section for the *"Sourcebook on the Generation of Electricity from Geothermal Energy,"* sponsored by the Department of Energy, Division of Geothermal Energy

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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.

Recent conceptual design studies made for 50-MW(e) geothermal power stations at Heber and Niland, California, are of particular interest. The former uses a flashed-steam system and the latter a binary cycle that uses isopentane. In last-quarter 1976 dollars, the total estimated capital costs were about $750/kW and production costs about 50 mills/kWhr. If wet/dry towers were used to conserve 50% of the water evaporation at Heber, production costs would be about 65 mills/kWhr.

1. GENERAL

1.1 Introduction

This study of waste heat rejection from geothermal power stations is concerned only with the heat rejected from the power cycle. The heat contained in reinjected or otherwise discharged geothermal fluids is not included with the waste heat considered here. The heat rejected from Rankine power cycles is primarily the heat of condensation of the working fluid, a quantity that may be defined in terms of the thermal efficiency of the cycle, η, as $Q_{in}(1 - \eta)$, where Q_{in} is the net heat input to the working fluid cycle.* In flashed-steam systems, Q_{in} may be considered as the difference between the enthalpy of the steam at the turbine throttle and the enthalpy of the condensate at the condenser hot well times the mass flow rate of the steam to the turbine. This study does not consider the heat contained in the underflow from the flashtanks in such systems as part of the heat rejected from the power cycle. By following this definition of the waste heat to be rejected, this study can discuss various methods of waste heat dissipation without regard for the particular arrangement to obtain heat from the geothermal source.

Fundamental to all heat-power systems working on the Rankine cycle is that a portion of the heat supplied to the cycle must be rejected. 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 1.1 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%.

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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.

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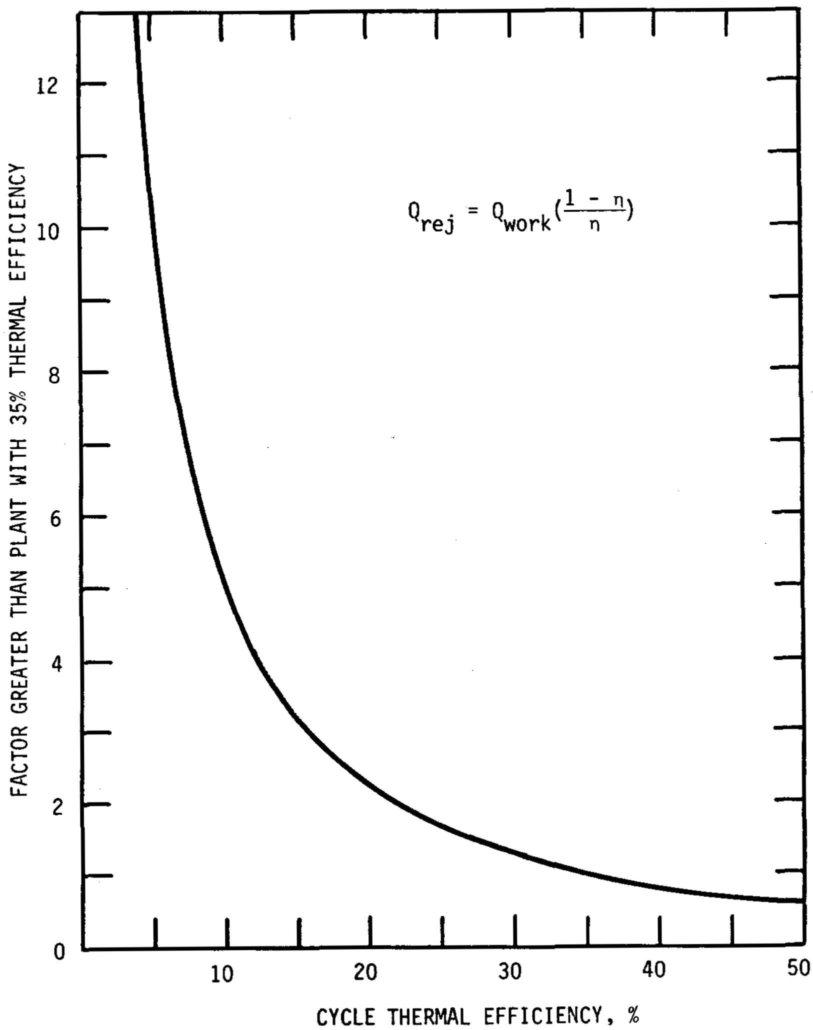

Fig. 1.1. Cycle heat rejection vs thermal efficiency compared to a cycle having an efficiency of 35%.

Essentially all of the waste heat discharged from a thermal power station is ultimately absorbed by the earth's atmosphere, regardless of the method used for heat dissipation. The methods may differ, however, in the lowest effective sink temperature for the cycle that can be realized, a factor having an important influence on the thermal efficiency of the cycle. The methods may also differ in costs, water consumption rates, and such environmental impacts as noise, drift deposition rates, fogging and icing potentials, and aesthetic appearances. With the exception of cooling coils that are operated dry, all of the heat dissipation methods involve the evaporation of water, either in a cooling tower, from a spray pond, or from increased evaporation rates at the surface of a lake or river due to raising the water temperature.

The large amount of heat wasted from thermal power cycles has been studied intensively to search for useful applications. Because the thermal energy exists at temperatures only a few degrees above ambient and because power stations are often well removed from places where the heat could be used for space heating, economic considerations of pumping and piping costs have thus far limited practical use of the rejected heat to only a few special situations. However, rising energy costs and fuel conservation measures will undoubtedly change this picture, and industries may someday locate near geothermal power stations to take advantage of the waste heat.

The waste heat rejection system for a geothermal power plant will cost proportionately more of the total station cost than the waste heat portions of conventional plants. The amount of heat to be rejected per kilowatt of output will be three to six times greater than that of a nuclear plant (Fig. 1.1). 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. Noncondensable gases may be higher than in conventional systems, affecting heat transfer area requirements and gas-handling costs. In view of these considerations, the waste heat rejection system for a geothermal power plant assumes an importance essentially equal to that of the steam supply or turbine-generator system and should receive as much attention in effecting the most economical arrangement.

Selection of a site for nuclear- and fossil-fueled power stations is influenced strongly by such considerations as the availability of a suitable heat sink, nearness to load centers, and acceptable environmental impacts, but geothermal power plants must be located where the energy is found. Thus, methods of waste heat rejection must make optimal use of what is available, and this availability is often limited. At the present time, the greatest U.S. potential for geothermal energy is found in the relatively hot, water-short areas of the Imperial Valley of Southern California. Regardless of the geographical area, however, water conservation is now an issue of prime importance in the siting of almost any power station. Economic 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 complex at the same location are simply not available. For example, a 500-MW(e) geothermal complex with an average thermal efficiency of 15% would require an annual average water makeup rate of about 0.9 m^3/sec (14,700 gpm) and an annual consumption of 29 x 10^6 m^3/year (23,700 acre-ft/year). This problem probably exists even if condensate from a flashed-steam geothermal cycle is used for the 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. Large stations at such sites would thus require use of significantly more expensive dry or wet/dry cooling towers.

1.2 Typical Waste Heat Rejection Systems and General Design Considerations

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. 1.2. Figure 1.3 exhibits binary cycles using wet mechanical-draft towers, and Fig. 1.4 shows the Heller-type cycle using a direct-contact condenser in conjunction with air-cooled coils. Figure 1.5 exemplifies the air-cooled coil with direct condensation of the exhaust steam.

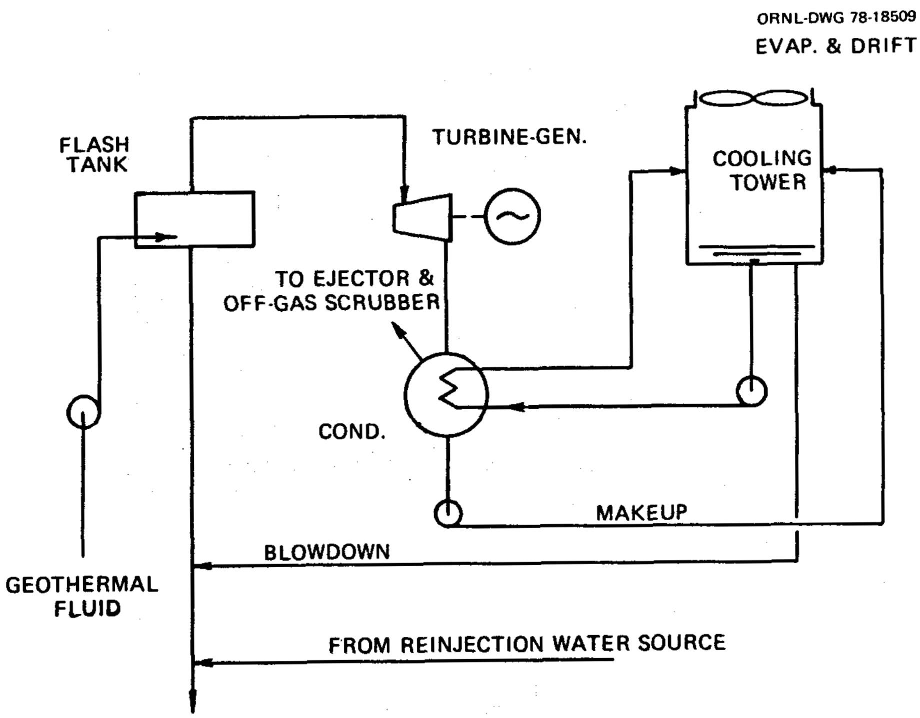

Fig. 1.2. Flashed-steam cycle using surface condenser and wet mechanical-draft cooling tower.

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

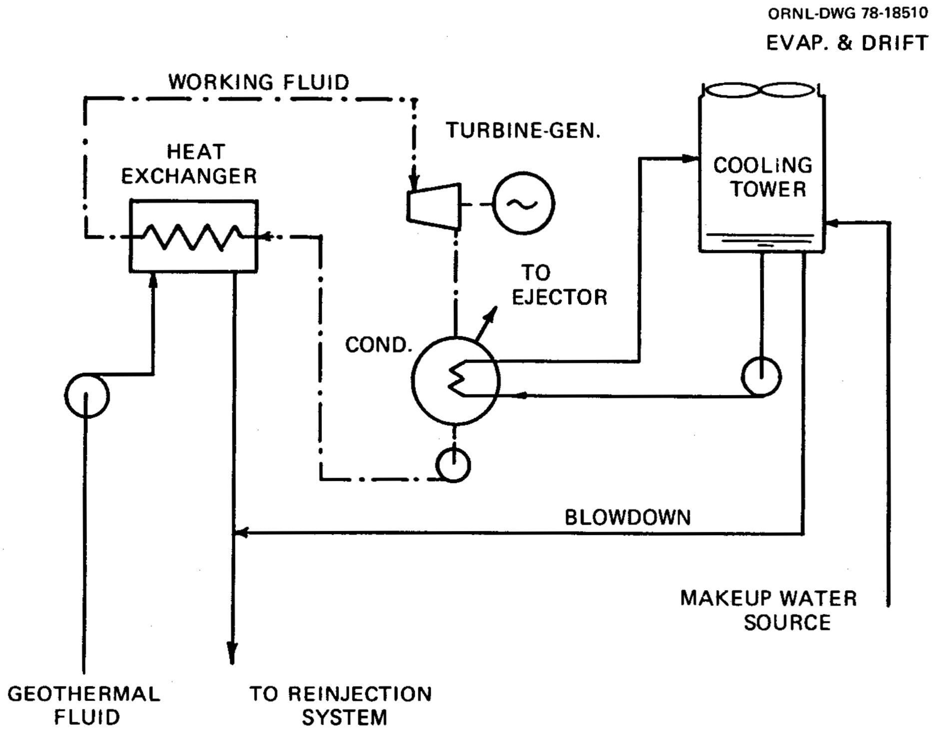

Fig. 1.3. Binary cycle using surface condenser and wet mechanical-draft cooling tower.

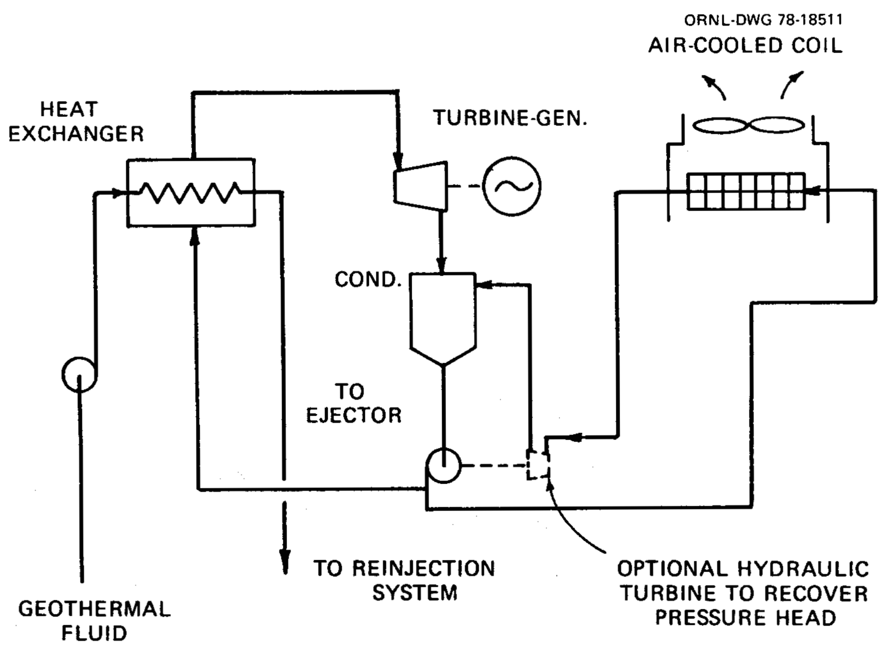

Fig. 1.4. Heller-type cycle using closed heat-exchanger, direct-contact condenser, and dry-type cooling tower.

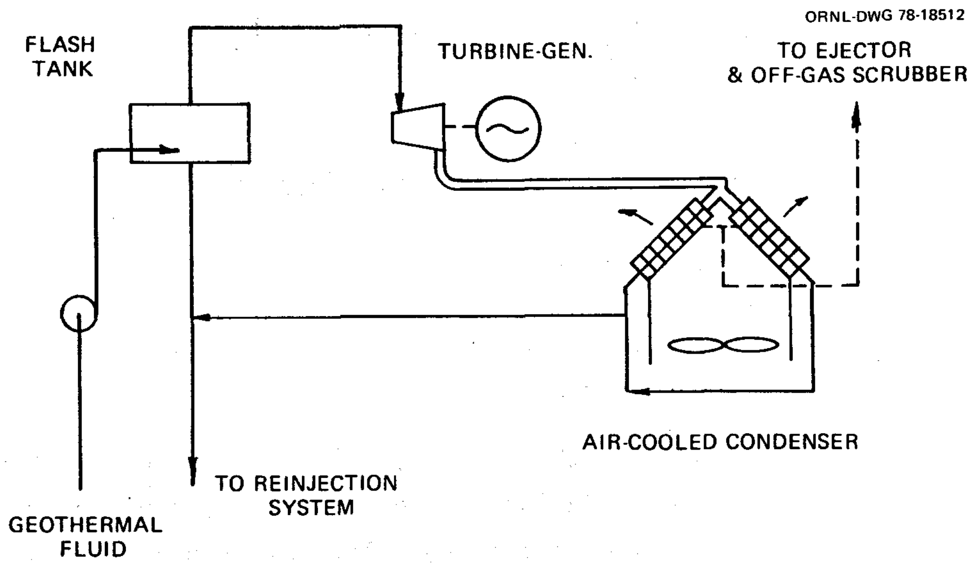

Fig. 1.5. Flashed-steam cycle with direct condensation of steam in dry cooling tower.

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. 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.

Although the waste heat rejection systems for geothermal power stations are relatively simple in concept, selection and design of a dissipation system will be influenced strongly by the relatively high capital and operating costs and by the effects on the station performance. The waste heat rejection system can have an important influence on obtaining licenses and construction permits because of water consumption and other environmental impacts. The designer of the waste heat system must strive toward

1. lowest capital costs,
2. lowest auxiliary power requirements,

3. lowest cooled water temperature (highest station efficiency),
4. ability to meet peak loads,
5. reliability and minimum downtime,
6. clean circulating water that will not corrode or foul heat transfer surfaces,
7. lowest evaporative water losses,
8. acceptable environmental impacts, such as control of noxious gases, thermal and chemical discharges, noise, etc., and
9. reasonable delivery and construction schedules.

Many of the above factors are overlapping, interrelated, and perhaps in opposition, and the selected design must be a compromise of these many aspects. At the present time, there are relatively rapid changes taking place in state and federal regulations, tax structures, costs of capital and labor, escalation rates, and conventional fuel costs. The latter influences the production cost of electricity, with which the geothermal stations compete, as well as affects the costs of reserve capacity to meet outages and peak loads.

As discussed in the previous section and illustrated in Fig. 1.1, the amount of heat to be rejected from a station is strongly dependent on the thermal efficiency of the power cycle. The amount of heat rejected per megawatt (electrical) of station capacity is indicated in Table 1.1. To this rejection must be added the miscellaneous waste heat sources, such as generator cooling, vent condensers, and air conditioning. Installations in arid regions will impose unusual circumstances that can add to costs.

Table 1.1. Heat rejected as a function of the thermal efficiency of the power cycle

Thermal efficiency (%)	Heat rejected per megawatt (electrical) of station capacity	
	MW(t)	(Btu/hr)
5	19.00	64.9×10^6
10	9.00	30.7×10^6
15	5.67	19.3×10^6
20	4.00	13.7×10^6

Hydrogen sulfide scrubbers may be required in off-gas systems. Because of the lack of experience and relatively wide choices that can be made in types of systems, materials, and equipment, there are no reliable guidelines for estimating the costs of waste heat rejection systems for geothermal applications. In fossil- and nuclear-fueled stations where the waste heat rejection systems may amount to approximately 15% of the total capital cost, some geothermal station cost estimates indicate that the waste heat systems may amount to 30% of the total capital cost.

1.3 Waste Heat Utilization

This study is concerned only with the waste heat rejected from the power cycles of geothermal power stations. Thus the broader subject of thermal uses for geothermal energy will not be discussed except to note that geothermal heat utilization has been studied at the Oregon Institute of Technology. The proceedings of the international conference held there in 1974 contain many references to successful applications of geothermal heat, such as those in Hungary, Iceland, and New Zealand.[1] The Idaho National Engineering Laboratory also has a program related to local and regional uses of geothermal heat. Although the technology is of interest, these activities are generally concerned with uses for geothermal heat and not with the utilization of relatively low-temperature heat that exists in turbine exhaust steam.

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 if, for example, standby oil- or gas-fired boilers and fuel storage must be provided to protect against freezing and loss of a valuable greenhouse crop. An additional factor is that whereas the utility may initially give the heat away and encourage its use if it lowers cooling tower operating costs, the time may come when it decides that the waste heat is a marketable product. All of these aspects affect the delivered cost of the heat. The delivered cost, in 1977 dollars, should be below about \$3.40/MW(t) (\$1/Btu x 10^6) in order to be economical.

In discussing why there has been relatively little use of central power station waste heat, Beall[2] stated that the stations need to seek a variety of waste heat users that could be clustered about the station so that all could benefit from a total thermal load large enough to allow a low-cost system. For example, if the utility could get one commitment for large greenhouses, another for fish cultures, one for raising chickens and for egg production, the heat delivery system costs could be substantially less than for a single waste heat project. The absence of organized effort in this respect and the lack of profit incentives for the utility (other than those that could accrue from good public relations or from rental of unused land) may be the reason why present-day applications are primarily of a demonstration nature.

In a project near Eugene, Oregon, warm water from a paper mill was used to provide frost protection, irrigation, subsoil heating, and crop cooling in the summer. The project, while not necessarily economical, was successful in protecting the field crops from frost and in increasing greenhouse yields by soil heating. Heat utilization in greenhouses is particulary attractive because of the high cash value of the crops that can be grown. For example, at a demonstration now under way at Becker, Minnesota, warm water from a 1400-MW(e) power plant will provide soil and air heating for a 0.2-ha (0.5-acre) greenhouse. The Tennessee Valley Authority (TVA) has several heat utilization projects under way, including a greenhouse demonstration similar to the Minnesota project, a catfish culture demonstration, and tests of effects of soil heating on crop yields. The TVA has received funds from the Environmental Protection Agency to demonstrate use of waste heat from a power station to stimulate the growth of algae from livestock wastes, with the algae being subsequently fed to amur (carp) fish. The fish will then be harvested for livestock food.[3] A bioconversion facility is under construction at Lamar, Colorado, to use waste heat from a local plant to warm digesters that will produce methane gas from the raw manure of 50,000 feedlot cattle. The methane will be burned in the power station boiler. Both algae grown in the effluent purification process and also a portion of the waste solids will be sold as a protein source for cattle feed. Other solids will be used as fertilizer.[4] There are numerous other demonstration projects.

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.

1.4 Water Availability and the Law

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.

Peterson and Sonnichsen studied the regional limitations of surface-water supplies with respect to the consumptive use requirements of wet cooling towers and spray ponds.[5] A primary objective of the study was to determine the regional needs for dry cooling towers. The conclusion drawn by the study is that, except for isolated cases where water for cooling is physically unavailable or in short supply, there are economic alternatives to dry or wet/dry cooling up to about 1990. It was predicted that between 1990 and 2000 there would be increased use of water. After the turn of the century, major to severe water problems will have developed in the lower Colorado and California regions, and moderate to major problems in the Great Basin, Upper Colorado, Rio Grands, Texas Gulf, Missouri, and Middle Atlantic regions.[6] Shifting societal pressures could, of course, alter these predictions.

Unfortunately, the regions considered now to be most promising for the development of geothermal energy are in areas with the most severe water shortages. The geothermal power industry should have need for large dry cooling towers well in advance of other types of electric generating stations and may have to develop much of the technology.

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. Moore[7] used a rather novel approach to illustrate the large amount of water consumed by evaporation in wet cooling towers for conventional (fossil and nuclear) power stations. He estimated that a typical 1000-MW(e) plant would need a runoff area in km^2 equal to $\frac{224}{r}$, where r is the runoff rate in centimeters per year (an area in square miles equal to $\frac{34}{r}$, where r is in inches per year). In the Northeastern United States, if 10% consumption of a runoff rate of 86 cm/year (34 in./year) is allowed, about 26 km^2 (10 sq miles) would be needed. If only about 1% of a runoff rate of 5 cm/year (2 in./year) in an arid region is permitted, then about 4400 km^2 (1700 sq miles) are required.

Expansion of electric power at today's reduced 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. In brief, Montana, Idaho, Wyoming, Nevada, Utah, Colorado, Arizona, and New Mexico as a holdover from the early gold mining days recognize the appropriation doctrine, which says that a pioneer who first takes and uses water can continue to do so and that between competing users priority will be given to the first user. The water need not be utilized in the same watershed as the one from which it is taken. The doctrine also permits market transfer of the rights as if they were property. In the more water-plentiful Eastern states of Michigan, Maine, New Hampshire, Vermont, New York, Connecticut, Massachusetts, Rhode Island, Pennsylvania, New Jersey, Delaware, Illinois, Indiana, Ohio, Missouri, Kentucky, Virginia, West Virginia, Arkansas, Tennessee, Alabama, Georgia, South Carolina, and Louisiana, the common doctrine is riparian law, which gives the owners of land adjoining the streams the superior right to use of the water. Minnesota, Wisconsin, Iowa, Maryland, North Carolina, and Florida use riparian law plus statutory regulations. Washington, Oregon, California, North Dakota, South Dakota, Nebraska, Kansas, Oklahoma, Texas, Mississippi, Alaska, and Hawaii use laws that are a combination of appropriation and riparian rights.[8] States sharing the watershed of a river may act jointly, as in the upper and lower Colorado River compacts.[9] Superimposed on the laws and regulations of the states are federal controls, such as the Wild and Scenic Rivers Act, the Wilderness Area Act, the National Park System, the National Forests, the National Wildlife Refuges and Ranges, and the various water pollution and environmental protection regulations. 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.

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2. CONDENSERS

2.1 General

All of the thermal power systems discussed in this study assume a Rankine cycle with a turbine as the prime mover. To obtain good Rankine cycle thermal efficiencies, the turbine exhaust pressure must be as low as can be economically attained with the available cooling water or air temperature. Table 2.1 lists the condensing pressures associated with the condensing temperatures of the five working fluids most commonly used in geothermal power cycles: water, ammonia, Freon-22, isobutane, and isopentane.* For the thermodynamic properties of the working fluids, the references given in Table 2.1 are recommended, but other sources are usually adequate. A convenient reference for these and other working fluids is Chap. 31 of the ASHRAE *Handbook of Fundamentals.*[1]

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. Tests show a decrease in steam consumption for a given power output of about 6 to 8% for each 25 mm (1 in.) of

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*The Starling[2] data for isobutane is at 5.6°C (10°F) intervals, and the isopentane data is, in part, at 11.1°C (20°F) intervals. Straight-line interpolation was therefore not used in Table 2.1 for values requiring interpolation of the Starling data. The equation $p = \exp(a + b/T + c \ln T)$, which gives good agreement for p-T relationships, was used to write three simultaneous equations based on representative points for the temperature in question. The solution gave values of a, b, and c that were used to calculate the intermediate temperature.

Table 2.1. Saturation pressure of common working fluids as a function of temperature

		Water[a]		Ammonia[b]		Freon-22[c]		Isobutane[d]		Isopentane[d]	
°C	°F	kPa	psia[e]	kPa	psia	kPa	psia	kPa	psia	kPa	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.

mercury reduction of exhaust pressure in the absolute pressure range of 50 to 127 mm (2 to 5 in.) of mercury. However, the specific volume of steam increases rapidly with a decrease in pressure, and the physical size of the exhaust system becomes proportionately larger and more expensive as the exhaust pressure is reduced. The properties of steam are also such that in the turbine the amount of moisture in the steam increases progressively during the expansion process in the turbine. 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.

Important to the thermal efficiency of a steam cycle is that the turbine operate at the design exhaust, or back, pressure. If the back pressure is higher than the design pressure, expansion in the turbine is incomplete and the heat rate increased. If the back-pressure is lower than the rated value, the steam velocity leaving the last turbine stage tends to exceed sonic values, and the turbine will "choke," also tending to increase the heat rate. Typically, the exhaust pressure of a steam turbine is allowed to vary by 25 to 76 mm (1 to 3 in.) of mercury. Maintaining the pressure within this narrow range requires careful operation to accommodate such variables as the plant electric load, cooling-water temperatures, fouling of heat transfer surfaces, etc.

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 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.

Steam condensers are not normally designed for high pressures on the shell side and are generally limited to about 138 kPa (20 psi) above atmospheric pressure. A sudden loss of coolant could cause the condenser pressure to mount rapidly, and protection must be provided against over-pressure. Blowout disks of thin metal are often used because these can be large enough to release quickly the necessary volume of steam.

Because steam condensers operate at high vacuum, the possibility exists that under some abnormal condition condensate could be drawn from the hot well back up into the unit, possibly allowing liquid to enter the off-gas pump or, in extreme cases, to damage the turbine blades. Vacuum breakers, or equivalent protection, have to be provided.

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. Because the heat transfer from a superheated vapor to a cooled surface is significantly lower than from a condensing vapor to the surface, in some cases it may be more economical to install desuperheaters between the turbine and the condenser.

2.2 Effect of Condensing Temperatures on Geothermal Station Performance

Geothermal power cycles generally have lower working fluid vaporizing temperatures than conventional nuclear- or fossil-fueled cycles. Examination of the Carnot efficiency term, $(T_1 - T_0)/T_1$, shows that the lower the heat source temperature, T_1, the more sensitive the efficiency becomes to the sink temperature, T_0. (Khalifa[3] has made a detailed study of the effect of the sink temperature on the performance of ideal cycles.) The sink temperature is a function of the temperature of the available cooling water, which will vary seasonally. Plants relying on dry cooling towers for heat dissipation, for example, would have unusually wide swings in the cooling-water temperature because of variations in the dry-bulb temperature between summer and winter. Figure 2.1 shows the effect on the performance of water and isopentane cycles designed for a 48.9°C (120°F) condensing temperature when operating at above and below the design point.

The values shown in Fig. 2.1 are for illustrative purposes and reflect only the change in available energy with exhaust pressure; they do not include the relatively small effects of the exhaust pressure on the internal efficiency of the turbine. Among other things, the internal efficiency is a function of the exhaust-end loading of the turbine — a turbine with light loading showing more improvement as exhaust pressures are lowered below the design point than one with heavy loading that cannot accommodate increased velocities. Applying steam-turbine-based condensing pressure adjustment factors to the internal efficiency of isopentane turbines would be particularly inappropriate. At the present time, definitive data is not readily available on the effects of exhaust pressure variations on the performance of isopentane units.

Figure 2.1 shows that the reduction in work output due to a condensing temperature increase is about equal to the increase in work output when the condensing temperature is decreased by an equal amount. If the design point is selected judiciously, seasonal swings in the condensing conditions would not seriously affect the annual average power production rate. Short-term effects can be substantial, however, particularly if they result in the need to purchase power to make up for lost capacity.

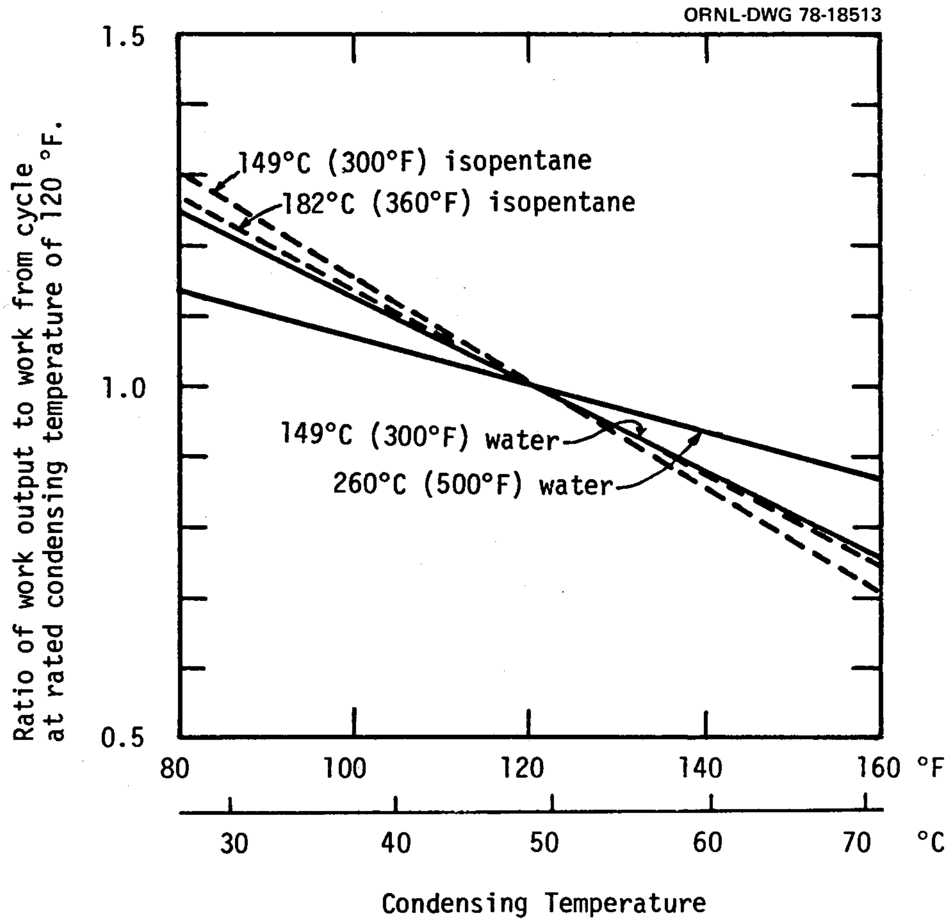

Fig. 2.1. Effect of condensation temperature on useful work output of water and isopentane power cycles. Does not include effects on turbine internal efficiency. See text.

2.3 Direct-Contact Condensers

2.3.1 General

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 and are discussed in Sect. 2.3.2. Direct-contact condensers are being considered for binary geothermal power cycles where the turbine exhaust would be condensed by direct contact with a different immiscible fluid. This type of condenser is called a "two-fluid" type and is discussed in Sect. 2.3.3.

2.3.2 Single-fluid direct-contact condensers

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.

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. 1.4, 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. 2.2. 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

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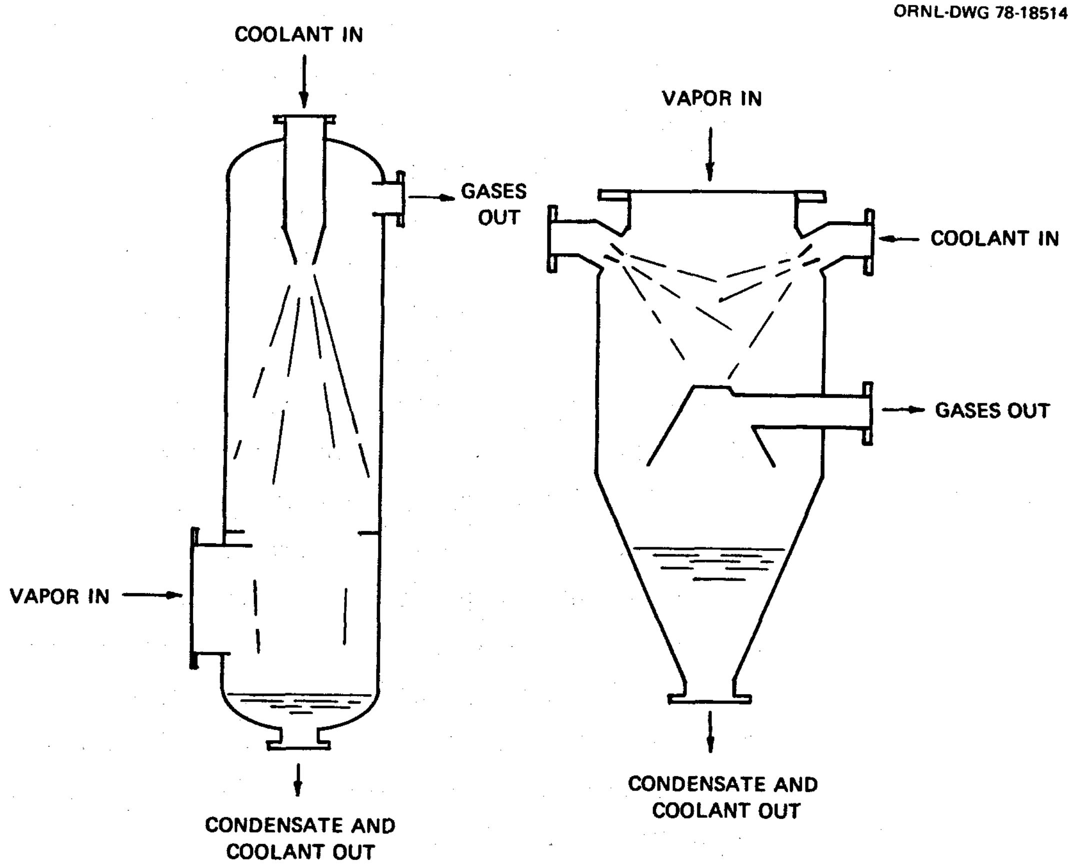

Fig. 2.2. Schematic of flow arrangement in direct-contact condensers: (a) counterflow and (b) parallel flow. Source: H. R. Jacobs and Heimer Fannar, *Direct Contact Condensers — A Literature Survey*, DGE/1523-3, UTEC 77-081, Mechanical Engineering Dept., University of Utah, Salt Lake City, (February 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. 2.2(a), or parallel to the spray, as in Fig. 2.2(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.[6] 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.

Satisfactory surface contact between the vapor and the cooling water can be achieved by the sprays alone, but a combination of sprays and cascades provides better deaeration. As in any condensing system, noncondensable gases must be removed continuously if the vacuum and performance are to be maintained. (Section 2.4.4 explains the effect of noncondensable gases on condenser performance.) Interior baffles must be provided to direct the gases to one or more takeoff points. Although vacuum pumps are now gaining in favor (Sect. 2.4.4), two-stage steam-actuated ejectors are commonly employed for scavenging.

Calculations of the cooling-water requirements for mixing condensers can be based on simple energy balances. The Heat Exchange Institute[7] recommends that for steam turbines the difference between the enthalpy of the entering steam and the enthalpy of the leaving mixture be taken as 2210 J/g (950 Btu/lb). Design of the mixing chamber has evolved over a number of years and is based largely on judgment and experience.

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[8] 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 (such as Ingersol Rand) rely heavily on previous experience and experimental testing programs. There is very little specific design information in the literature.

Direct-contact condensers are commonly used with air-cooled coils, as in the Heller system. Convincing arguments can also be made for use of surface condensers (Sect. 3.5.6.3).

2.3.3 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[9] 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.[10] 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.

2.4 Surface Condensers

2.4.1 General

Surface-type condensers are most commonly used in large, modern steam power stations. This usage stems primarily from the need to recover the high-quality condensate for return to the boiler. In binary geothermal cycles, the surface condenser must be used to separate the working fluid and the condenser coolant. The condensers are the shell-and-tube type, and almost without exception the steam or other working fluid is in the shell side and the coolant flows through the tubes. The term "surface condenser" is now reserved for tubular condensers of this design.[11]

A diagrammatic section through a horizontal, two-pass, shell-and-tube condenser is shown in Fig. 2.3 The coolant enters a water box on one end and flows through the lower half of the tube bundle to a water box on the other end; the flow then reverses and returns through the upper half of the tube bundle. Single-pass condensers have the water inlet and outlet on opposite ends. Although single-pass condensers tend to be lower in initial cost, they may not produce as good a vacuum and as high and sometimes do not fit well into the piping layout. In general, these condensers are used when circulating water is plentiful and where the fixed pumping head is not high. Single-pass condensers usually require less surface area but more circulating water than do multipass condensers. To keep the coolant velocity in the tubes sufficiently high for good heat transfer, it is necessary to use either relatively small diameter tubes or relatively few tubes of longer length. A velocity of about 1.8 m/sec

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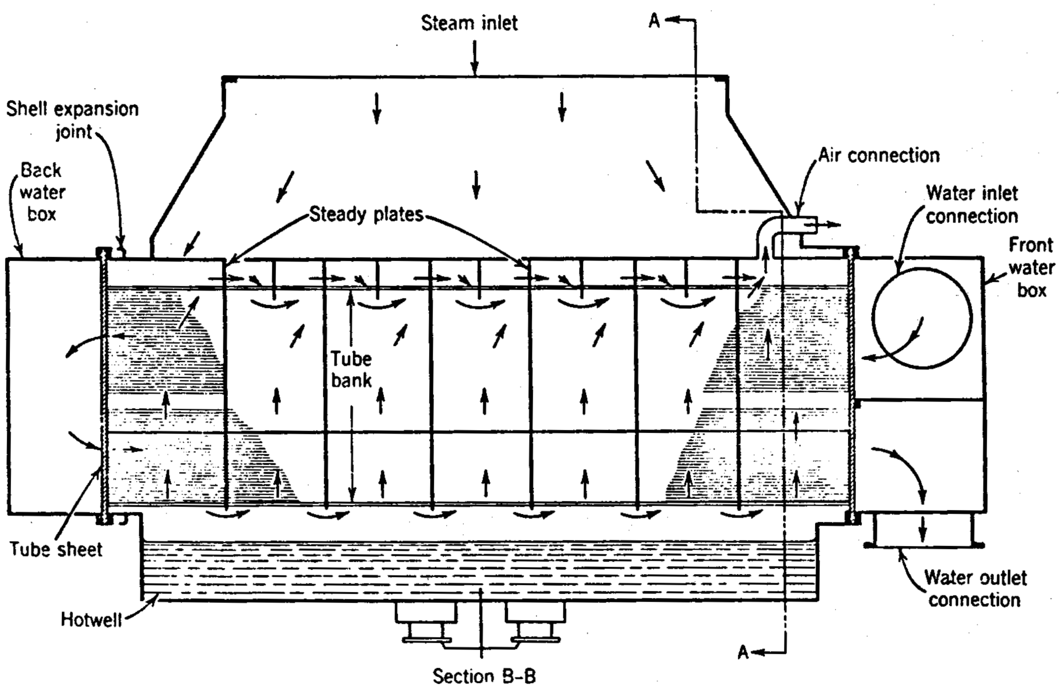

Fig. 2.3. Sections through a typical two-pass surface condenser for a large steam power plant. Reprinted by permission from Allis-Chalmers Corporation.

(6 ft/sec) or more is usually considered sufficient. In general, tubes of less than -1.6-cm (5/8-in.) OD clog too easily, and tubes of 2.5-cm (1-in.) OD and larger require excessive amounts of water to achieve desirable velocities. Long tube lengths cause more thermal expansion problems and require more room at the ends of the condenser to clean and replace tubes. Combinations of the above factors tend to make the two-pass arrangement favorable for most applications, including the steam power field. The colder water in the first pass is capable of condensing more steam, and the division of the duty is often about 60% in the first pass and 40% in the second.

Relatively large volumes of water are required for condensing service. A geothermal power station with a thermal efficiency of about 15% would need about 0.16 m^3/sec (2500 gpm) of cooling water per MW(e) of installed capacity. Thus, pipes supplying the water are of relatively large diameter and pose layout problems. A current trend is toward use of fiberglass-reinforced plastic pipes in some portions of the system because of their light weight in comparison to the steel and concrete pipes conventionally used.

One of the major difficulties in the design of surface condensers is accommodating the differential expansion of the tubes and the shell. Although various arrangements are used, the most common is an expansion joint that permits one tube sheet to move independently of the shell. Tubes are usually joined to the tube sheets by roller expansion, a method that is suitable for commonly used tubing materials and that will give satisfactory leak-tightness for most steam turbine condenser applications. Other methods do not prove to be as effective. Brazing, for example, requires large furnaces for vacuum brazing, and soldering may compromise corrosion resistance. Adhesives, such as the expoxies, involve very strict quality control if they are to be effective. Welding of the tubes on the face side gives the most leak-tight joints and is essentially the only alternative when using less formable materials, such as titanium. However, welding is more expensive and requires care to prevent cracking at the welds. Despite care in design and manufacture, surface condensers may develop leaks between the shell-side and the tube-side fluids. The designer must arrange for access to plug leaky tubes and to repair joints, as well as allow sufficient room (tube pull space) to replace tubes.

The concentration of noncondensable gases on the shell side increases progressively in the direction of the vapor flow. The takeoff to the noncondensable gas-removing equipment should therefore be located near the end of the vapor path where the greatest density of the gases occurs. The sweeping effect of the steam flowing through the condenser is an important factor in concentrating the noncondensable gases at the bottom of the condenser and in scavenging them from condensing surfaces. Stagnant zones where the gas could collect should be avoided. Provisions should be made for cooling gas that is not condensed to reduce the size of the gas-pruging equipment and to remove as much moisture (working fluid) as possible from the gas. These provisions can be accomplished by shrouding some of the tubes to draw the gases over the cooled surfaces.

A stream velocity of 30 to 60 m/sec (100 to 200 ft/sec) through the first row of tubes is good practice at vacuums of about 660 mm (26 in.) of mercury, and 60 to 120 m/sec (200 to 400 ft/sec) is used at vacuums of about 737 mm (29 in.) of mercury. The steam-side pressure drop varies from about 1.3 mm (0.05 in.) of mercury to 13 mm (0.5 in.) of mercury.

The cooling-water velocity through the tubes is usually in the range of 1.8 to 2.4 m/sec (6 to 8 fps). The tube diameters are commonly 20 to 25 mm (3/4 to 1 in.) in outside diameter, and tube lengths vary from about 5 to 10 m (15 to 30 ft). For freshwater service, the tubes are ordinarily of a copper alloy, such as admiralty metal (70% copper, 29% zinc). Saline water installations may use a nickel alloy, such as Monel, or titanium. A tube wall thickness of 18 British wire gage (BWG), 1.25 mm or 0.049 in., will withstand ordinary circulating system pressures and is commonly used. Tube sizes are shown in Table 2.2, and mechanical and physical properties are given in Table 2.3. At the present time, tubes with outside diameters in even fractions of an inch and with wall thickness specified in BWG are still being manufactured in the United States. Conversion factors for the SI system are given in footnotes to the tables.

Because the cleanliness of the condenser surfaces has a significant influence on the heat transfer and on the performance of the power cycle, the cooling-water treatment and the tube-cleaning system are very important adjuncts to the condensing system. These aspects are discussed in Sect. 2.4.5.

2.4.2 Heat transfer in surface condensers

The resistances to transfer of heat from the condensing vapor on the shell side to the fluid flowing inside the tubes of a surface condenser are expressed as

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{A_o}{A_i}\frac{1}{h_i} + R_{fo} + \frac{A_o}{A_i}R_{fi} + \frac{D_o \ln(D_o/D_i)}{2k} ,$$

Table 2.2. Heat exchanger and condenser tube data[a]

Tube OD, in.	BWG	Wall thick- ness, in.	ID, in.	Flow area per tube, in.²	Surface per lin ft, ft²	
					Outside	Inside
½	12	0.109	0.282	0.0625	0.1309	0.0748
	14	0.083	0.334	0.0876		0.0874
	16	0.065	0.370	0.1076		0.0969
	18	0.049	0.402	0.127		0.1052
	20	0.035	0.430	0.145		0.1125
¾	10	0.134	0.482	0.182	0.1963	0.1263
	11	0.120	0.510	0.204		0.1335
	12	0.109	0.532	0.223		0.1393
	13	0.095	0.560	0.247		0.1466
	14	0.083	0.584	0.268		0.1529
	15	0.072	0.606	0.289		0.1587
	16	0.065	0.620	0.302		0.1623
	17	0.058	0.634	0.314		0.1660
	18	0.049	0.652	0.334		0.1707
1	8	0.165	0.670	0.355	0.2618	0.1754
	9	0.148	0.704	0.389		0.1843
	10	0.134	0.732	0.421		0.1916
	11	0.120	0.760	0.455		0.1990
	12	0.109	0.782	0.479		0.2048
	13	0.095	0.810	0.515		0.2121
	14	0.083	0.834	0.546		0.2183
	15	0.072	0.856	0.576		0.2241
	16	0.065	0.870	0.594		0.2277
	17	0.058	0.884	0.613		0.2314
	18	0.049	0.902	0.639		0.2361
1¼	8	0.165	0.920	0.665	0.3271	0.2409
	9	0.148	0.954	0.714		0.2498
	10	0.134	0.982	0.757		0.2572
	11	0.120	1.01	0.800		0.2644
	12	0.109	1.03	0.836		0.2701
	13	0.095	1.06	0.884		0.2775
	14	0.083	1.08	0.923		0.2839
	15	0.072	1.11	0.960		0.2896
	16	0.065	1.12	0.985		0.2932
	17	0.058	1.13	1.01		0.2969
	18	0.049	1.15	1.04		0.3015
1½	8	0.165	1.17	1.075	0.3925	0.3063
	9	0.148	1.20	1.14		0.3152
	10	0.134	1.23	1.19		0.3225
	11	0.120	1.26	1.25		0.3299
	12	0.109	1.28	1.29		0.3356
	13	0.095	1.31	1.35		0.3430
	14	0.083	1.33	1.40		0.3492
	15	0.072	1.36	1.44		0.3555
	16	0.065	1.37	1.47		0.3587
	17	0.058	1.38	1.50		0.3623
	18	0.049	1.40	1.54		0.3670

[a]Conversion factors: mm = in. x 25.40; mm^2 = $in.^2$ x 645.160; m^2/m = ft^2/ft x 0.30480.

Source: Donald Q. Kern, *Process Heat Transfer*, McGraw-Hill, New York, 1950. Reprinted by permission.

Table 2.3. Mechanical and physical properties of condenser tube materials[a]

Material Category	Material	Typical Specification	PHYSICAL PROPERTIES (See Note a, b, and c below)			
			Density lbs/cu. in.	Thermal Conductivity B.T.U./hr, Ft², F, in.	Coefficient of Thermal Expansion in/in, F. See Note c	Modulus of Elasticity p.s.i.
COPPER BASE ALLOYS (a.1)	Admiralty	ASTM.B.111 Alloy 443	0.308	768 at 68 F	11.2×10^{-6}	16×10^6
	Admiralty	ASTM.B.111 Alloy 444	0.308	768 at 68 F	11.2×10^{-6}	16×10^6
	Admiralty	ASTM.B.111 Alloy 445	0.308	768 at 68 F	11.2×10^{-6}	16×10^6
	Alum. Brass	ASTM.B.111 Alloy 687	0.301	696 at 68 F	10.3×10^{-6}	16×10^6
	Alum. Bronze	ASTM.B.111 Alloy 608	0.295	552 at 68 F	10.0×10^{-6}	17.5×10^6
	Copper Nickel 70-30	ASTM.B.111 Alloy 715	0.323	204 at 68 F	9.0×10^{-6}	22×10^6
	Copper Nickel 90-10	ASTM.B.111 Alloy 706	0.323	312 at 68 F	9.5×10^{-6}	18×10^6
	Arsenical Copper	ASTM.B.111 Alloy 142	0.323	1344 at 68 F	9.8×10^{-6}	17×10^6
	Copper Iron 194	ASTM.B.543 Alloy 194	0.317	1800 at 68 F	9.0×10^{-6}	17.5×10^6
STAINLESS STEELS (a.2)	Stainless Steel	ASTM.A.249 Type 304	0.29	113 at 212 F	9.9×10^{-6}	28×10^6
	Stainless Steel	ASTM.A.249 Type 316	0.29	113 at 212 F	9.0×10^{-6}	28×10^6
	Stainless Steel	ASTM.A.269 Type 304	0.29	113 at 212 F	9.9×10^{-6}	28×10^6
	Stainless Steel	ASTM.A.269 Type 316	0.29	113 at 212 F	9.0×10^{-6}	28×10^6
TITANIUM (a.3)	Titanium	ASTM.B.338 Grade 1	0.163	Approximately 114 at 68 F	5.1×10^{-6}	14.9×10^6
	Titanium	ASTM.B.338 Grade 2	0.163	Approximately 114 at 68 F	5.1×10^{-6}	14.9×10^6
CARBON STEEL	Carbon Steel	ASTM.A.179	0.283 (a.4)	324 at 212 F (a.4)	6.44×10^{-6} (a.5)	27.7×10^6 (a.5)

[a]Conversion factors: lbf/in.^{2} x 6894.757 = Pa, or N/m^2; lbm/in.^{3} x 27,679.90 = kg/m^3; Btu·in./hr·ft^2·°F x 0.144279 = W/m·K; in./in.·°F x 1.4 = m/m·K.

Source: *Standards for Surface Condensers*, 6th ed., Heat Exchanger Institute, New York, 1970. Reprinted by permission.

where

U_o = overall heat transfer coefficient based on the outside surface area of the tube, $W/m^2 \cdot K$ ($Btu/hr \cdot ft^2 \cdot °F$);

h = film coefficient, $W/m^2 \cdot K$ ($Btu/hr \cdot ft^2 \cdot °F$);

A = area, m^2 (ft^2);

R = resistance, $m^2 \cdot K/W$ ($hr \cdot ft^2 \cdot °F/Btu$);

D = tube diameter, m (ft);

k = thermal conductivity of the tube wall, $W/m \cdot K$ ($Btu/hr \cdot ft^2 \cdot °F$);

Subscripts: o = outside, i = inside, f = fouling.*

Two of the most frequently used heat transfer correlations for laminar, filmwise condensation of a single vapor on the outside of a smooth, horizontal tube are[12]

$$h_o = 0.951 \left(\frac{k_f^3 \, \rho_f^2 \, g}{\mu_f} \cdot \frac{L}{W} \right)^{1/3} = 0.725 \left(\frac{k_f^3 \, \rho_f^2 \, g}{\mu_f} \, \frac{h_{fg}}{N D \Delta t} \right)^{1/4} ,$$

where

h_o = outside film coefficient, $W/m^2 \cdot K$ ($Btu/hr \cdot ft^2 \cdot °F$);

k_f = thermal conductivity of the condensing liquid, $W/m \cdot K$ ($Btu/hr \cdot ft \cdot °F$);

ρ_f = density of the condensing liquid, kg/m^3 (lbm/ft^3);

g = gravitational acceleration, m/sec^2 (ft/hr^2);

μ_f = absolute viscosity of the condensing liquid at film temperature, $Pa \cdot sec$ ($lbm/ft \cdot hr$);

L = length of tubes, m (ft);

w = flow rate of condensate from the lowest point on the condenser surface, kg/sec (lbm/hr);

h_{fg} = latent heat of vaporization of the condensing fluid, $W \cdot sec/kg$ (Btu/lbm);

N = number of tubes in the vertical tier;

D = tube outside diameter, m (ft);

Δt = temperature difference between the saturation temperature of the condensing fluid and the tube surface, K (°F).

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*Conversion factors: $W/m^2 \cdot K = (Btu/hr \cdot ft^2 \cdot °F) \times 5.67827$
$W/m \cdot K = (Btu \cdot ft/hr \cdot ft^2 \cdot °F) \times 1.73073$

Condensation in steam-power condensers is almost always laminar. The first correlation is convenient when the amount of fluid to be condensed is known; the second is useful when the difference between the condensing temperature and the tube surface temperature is available. Some selected values for the thermal conductivity, absolute viscosity, and specific heat and density of ammonia, Freon-22, isobutane, and water are listed in Table 2.4. Values of the outside condensing film coefficient, h_o, have been calculated using the properties from this table and the above equations at 15, 40, and 60°C for ammonia, Freon-22, isobutane, and water as functions of L/w and $h_{fg}/ND\Delta t$, as shown in Tables 2.5 and 2.6. The values of h_o at a condensing film temperature of 40°C (74°F) have been plotted as a function of L/w in Fig. 2.4. The effect of the presence of noncondensable gases on the outside film coefficient is discussed in Sect. 2.4.4.

For water, the resistance to heat transfer in the boundary layer inside liquid-cooled surface condenser tubes may be one of the most important of the resistances. In steam-power condenser, the condition inside the tube is almost always in the turbulent, forced-convection regime, and the following relationship is commonly used:

$$\frac{h_i d}{k} = 0.023\left(\frac{dV\rho}{\mu}\right)^{0.8}\left(\frac{\mu C_p}{k}\right)^{0.4} ,$$

where

h_i = inside film heat transfer coefficient, W/m²·K (Btu/hr·ft²·°F);
d = inside diameter of tube, m (ft);
k = thermal conductivity of fluid, W·m/m²·K (Btu/hr·ft·°F);
V = velocity in tube, m/sec (ft/hr);
ρ = density of fluid, kg/m³ (lbm/ft³);
μ = absolute viscosity of fluid, Pa·sec (lbm/ft·hr);
C_p = specific heat of fluid, W·sec/kg·K (Btu/lbm·°F).

All of the thermal properties of the fluid flowing inside the tube are evaluated at the bulk temperature. The inside film coefficient for 2.25-cm (7/8-in.) OD No. 18 BWG tubes for water in the temperature range

Table 2.4. 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·m/m²·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/m³		
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/m³) x 0.062428 = lbm/ft³; 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.

Table 2.5. Condensing film coefficient outside of horizontal tubes as a function of L/w at temperatures of 15, 40, and 60°C

	L/w^a	W/m²•K		
		15°C	40°C	60°C
Ammonia	2×10^0	1,738	1,610	1,520
	2×10^1	3,743	3,469	3,275
	2×10^2	8,065	7,474	7,055
	2×10^3	17,375	16,102	15,199
Freon-22	2×10^0	461	393	341
	2×10^1	992	848	734
	2×10^2	2,138	1,826	1,582
	2×10^3	4,606	3,935	3,409
Isobutane	2×10^0	357	334	318
	2×10^1	768	719	686
	2×10^2	1,655	1,548	1,478
	2×10^3	3,566	3,335	3,184
Water	2×10^0	1,501	1,808	2,239
	2×10^1	3,234	3,894	4,823
	2×10^2	6,967	8,390	10,390
	2×10^3	15,010	18,075	22,386

[a]Notes and conversion factors: L = length of tubes, m; w = flow rate of condensate from lowest point of condensing surfaces, kg/sec; ($W/m^2 \cdot K$) x 0.176110 = $Btu/hr \cdot ft^2 \cdot °F$; (m•sec/kg) x 413.3789 x 10^{-6} = ft•hr/lbm.

of 15°C (59°F) to 60°C (140°F) is plotted in Fig. 2.5 as a function of the velocity inside the tube. Adjustment factors for other tube diameters are also given in Fig. 2.5. The heat transfer coefficient decreases with temperature, which negates slightly the advantages of using colder water as the coolant.

The resistance to heat transfer in the tube wall is a function of the wall thickness and the conductivity of the metal. Table 2.3 shows the thermal conductivities of some of the commonly used condenser tubing materials. The conductivity increases with temperature in some metals (e.g., aluminum, brass, and the stainless steels) and decreases with temperature in materials such as copper, steel, iron, etc. However, over the temperature range of usual application in steam-power condensers, the changes are not great. In any event, the resistance to heat transfer due to the wall is such a small percentage of the total resistance that

Table 2.6. Condensing film coefficient outside of horizontal tubes as function of $h_{fg}/ND\Delta t$ at temperatures of 15, 40, and 60°C

	$h_{fg}/ND\Delta t^a$	W/m²·K		
		15°C	40°C	60°C
Ammonia	10^5	3,030	2,862	2,740
	10^6	5,388	5,089	4,873
	10^7	9,581	9,049	8,666
	10^8	17,037	16,092	15,410
Freon-22	10^5	1,119	995	893
	10^6	1,990	1,769	1,588
	10^7	3,539	3,145	2,824
	10^8	6,294	5,593	5,023
Isobutane	10^5	924	879	849
	10^6	1,643	1,562	1,509
	10^7	2,921	2,778	2,683
	10^8	5,195	4,941	4,772
Water	10^5	2,715	3,121	3,664
	10^6	4,828	5,550	6,515
	10^7	8,585	9,869	11,586
	10^8	15,266	17,549	20,603

[a]Notes and conversion factors: h_{fg} = latent heat vaporization of condensing fluid, J/kg, or W·sec/kg; N = number of tubes in vertical tier; D = outside diameter of tubes, m; Δt = temperature difference across film, °C; ($W/m^2 \cdot K$) x 0.176110 = $Btu/hr \cdot ft^2 \cdot °F$; (W·sec/kg·m·K) x 72.80023 x 10^{-6} = Btu/lbm·ft·°F.

the variation of the thermal conductivity with temperature need not be taken into account. A log mean thickness for the tube wall may be used instead of the wall thickness (as indicated in the equation), but here again the refinement will have only a small effect on the calculated overall heat transfer coefficient for the condenser.

Although there may seem to be an incentive to use smaller diameter tubing to obtain the maximum heat transfer surface per unit volume in the condenser, tubes smaller than 15-mm (5/8-in.) OD become plugged too easily and are excessively difficult to clean. In larger sizes of condensers, tube diameters of 22 mm (7/8 in.) and 25 mm (1 in.) are used so that the tube length can be increased without causing excessive frictional pressure losses.

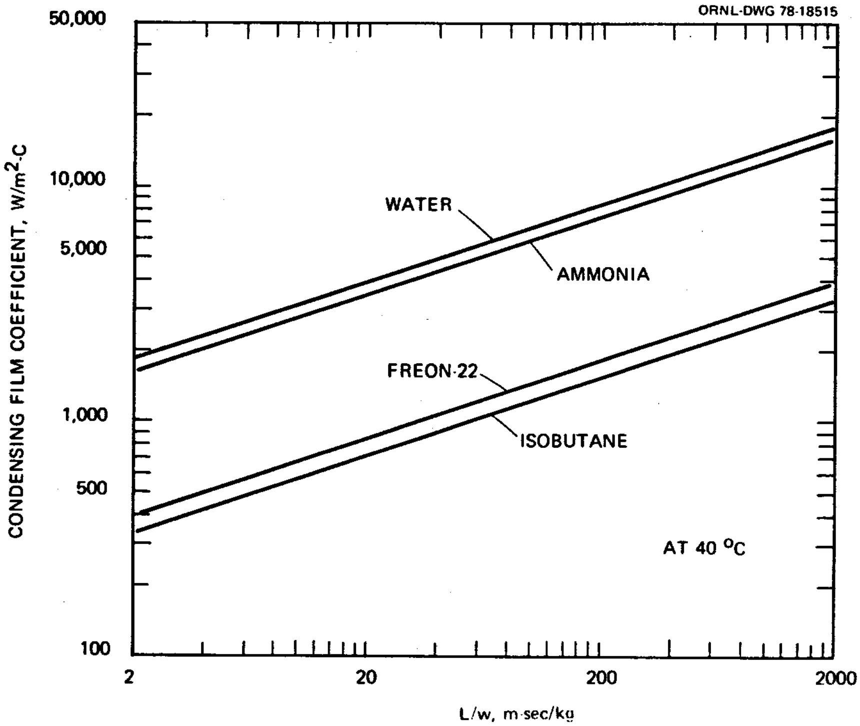

Fig. 2.4. Condensing film heat transfer coefficient at 40°C (74°F) as a function of L/W. Refer to Table 2.5 where Btu/hr•ft^2•°F = W/m^2•K x 0.176110.

The resistance of the foulants on the tube surface is typically taken as about 0.00018 m^2•K/W (0.001 hr•ft^2•°F/Btu) for tubes that use relatively clean water and that need only occasional cleaning. Fouling resistances of about one-fifth of this value may be attained using very clean water and frequent regular cleaning of the surfaces. For relatively dirty surfaces, which might be encountered when using seawater as the coolant, the resistance should probably be taken as five times the above value. The performance of enhanced-surface tubing is particularly vulnerable to fouling (Sect. 2.4.8).

In power plant applications, where water is condensed outside of smooth tubes and coolant water flows through the tubes, the resistances may be distributed as follows: (1) 20% due to the outside film coefficient, (2) 4% due to the tube wall resistance, (3) 55 to 60% due to the inside film coefficient, and (4) about 15 to 20% due to the fouling on both sides of the tube. One source[13] states that the distribution is

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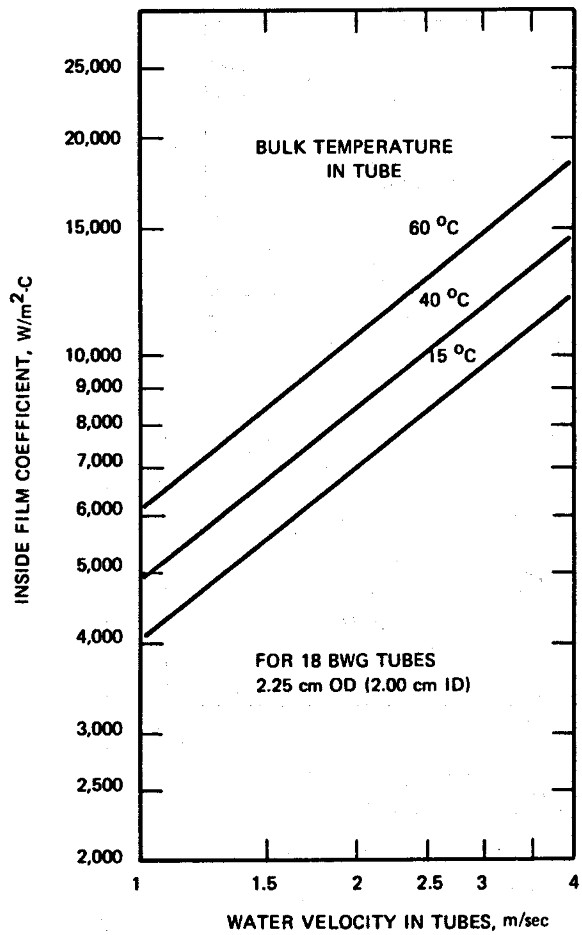
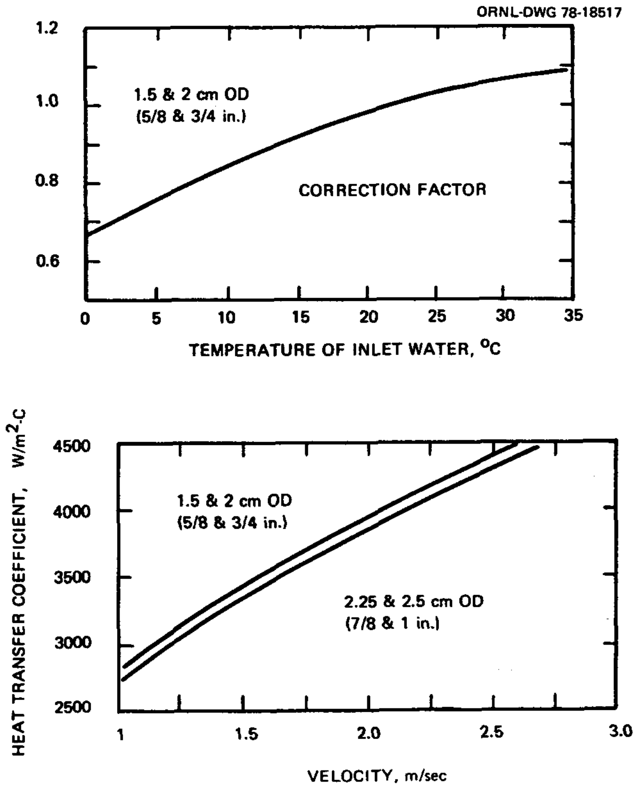


FOR OTHER 18 BWG TUBE DIAMETERS:

cm	in.	Multiply by:
1.75	5/8	1.06
2.00	3/4	1.03
2.25	7/8	1.00
2.50	1	0.98

Fig. 2.5. Inside liquid water film heat transfer coefficient as function of water velocity in No. 18 BWG tubes (W/m^2·K x 0.176110 = Btu/hr·ft^2·°F).

usually 18% due to the outside film, 8% due to fouling outside the tube, 2% due to the tube wall, 33% due to fouling inside the tube, and 39% due to the inside film coefficient. The difference presumably lies in the degree of fouling considered normal. The major resistances are the inside film coefficient and the resistance of the dirt and slime deposits if the tubes are not clean. The overall heat transfer coefficient, U_o, with the area based on the outside diameter of the tubes and water on both sides of the tubes, will probably fall in the 2300- to 4000-W/m^2·K (400- to 700-Btu/hr·ft^2·°F) range (Fig. 2.6). Correction factors for other tube materials are given in Table 2.7 and in Fig. 2.14. For other than typical installations and where fluids other than water are used,



To obtain	Multiply	By
Btu/hr-ft^2-F	W/m^2-K	0.176110
ft/sec	m/sec	0.3048

Fig. 2.6. Overall heat transfer coefficient for condensing on No. 18 BWG Admiralty metal tubes, for a cooling-water temperature of 21°C (70°F), based on external surface area of tube. A fouling factor of 0.85 is commonly applied to the coefficients. Source: *Standards for Surface Condensers*, 6th ed., Heat Exchanger Institute, New York, 1970. Reprinted by permission.

one must examine the individual resistances. The film coefficients are relatively well studied and the literature extensive, but data on fouling resistances is not as easily available because of the variations between specific applications. Also, such studies do not lend themselves as well to a systematic approach and to university-related research projects. There is often the dilemma of the design engineer who can calculate the film coefficients and the wall resistance in detail because information is readily available but can do little to account accurately for noncondensable gases and fouling, which can be a major resistance to the heat transfer.

Table 2.7. Correction factors for any tube gage or material other than No. 18 BWG Admiralty metal with which to multiply the heat transfer coefficients obtained from Fig. 2.6

Tube Materials	Tube Wall Gauge — BWG						
	24	22	20	18	16	14	12
Admiralty Metal	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Arsenical Copper	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Aluminum	1.06	1.04	1.02	1.00	0.96	0.92	0.87
Aluminum Brass	1.03	1.02	1.00	0.97	0.94	0.90	0.84
Aluminum Bronze	1.03	1.02	1.00	0.97	0.94	0.90	0.84
Muntz Metal	1.03	1.02	1.00	0.97	0.94	0.90	0.84
90-10 Cu-Ni	0.99	0.97	0.94	0.90	0.85	0.80	0.74
70-30 Cu-Ni	0.93	0.90	0.87	0.82	0.77	0.71	0.64
Cold-Rolled Low Carbon Steel	1.00	0.98	0.95	0.91	0.86	0.80	0.74
Stainless Steels							
Type 410/430	0.88	0.85	0.82	0.76	0.70	0.65	0.59
Type 304/316	0.83	0.79	0.75	0.69	0.63	0.56	0.49
Type 329	0.78	0.76	0.74	0.69	0.65	0.60	0.54
Titanium (Tentative)	0.85	0.81	0.77	0.71	—	—	—

Source: *Standards for Surface Condensers*, 6th ed., Heat Exchanger Institute, New York, 1970. Reprinted by permission.

For a detailed treatment of heat transfer, the reader should consult the literature — such as the text of McAdams.[12] The American Society of Heating, Refrigerating, and Air-Conditioning Engineers' (ASHRAE) *Handbook of Fundamentals*[1] contains a good summary of heat transfer relationships (Chap. 2), and compilations of thermophysical properties are given in Chap. 14. The ASHRAE work also serves as a reference for geothermal applications because, in addition to water, it covers such fluids as isobutane, ammonia, and Freons.

2.4.3 Multipressure condensers

Turbines with compounded low-pressure ends may use a separate condenser shell for each of the turbine exhausts. The cooling water may be arranged to flow through the condensers either in parallel or in series. With parallel flow, the condensing steam temperatures will be essentially the same. With series flow, the temperature in each condensing zone will be different, and the average temperature will be less than for the parallel-connected arrangement. The former is termed a "single-pressure" system and the latter a "multipressure" system. The lower average backpressure for the multipressure arrangement improves the cycle heat rate.

In closed cycles, an improvement to the heat rate can also be made by passing the condensate from the lower-pressure condenser into the higher-pressure unit for direct contact with the condensing steam, thereby raising the average temperature of the condensate above the temperature that could be attained in a single-pressure system. In general, multipressure systems are most advantageous in systems with low terminal-temperature differences and high temperature-rises (ranges) for the cooling water.

Although multipressure arrangements improve the cycle efficiency, the capital costs of the system are greater than those of a single-pressure system. The turbines must be designed for the specific exhaust pressures, and lower back-pressures will increase the capital cost. The water-circulating pumps will require more power because of the increased pumping head needed for the series flow arrangement, possibly causing the required piping layouts to be more expensive. Palmer and Miller[14] point out that, rather than operating at reduced back-pressure, it may be more economical to reduce the surface area and the water flow rates by amounts that will result in the same heat rate as that for a single-pressure system. The savings in capital and operating costs in this arrangement frequently exceed the savings resulting from reduced heat rates.

An evaluation must be made of each particular application to weigh the heat rate advantages of the multipressure arrangement against the relative simplicity and lower initial cost of the single-pressure system. The potential heat rate gain for multipressure systems is difficult to estimate because of the complex relationships existing among the low-pressure turbine, the condenser, and the first low-pressure heater.[15] Because the cost differentials require detailed study, it is difficult to present any generalized rules concerning which of the arrangements will be the best choice. A first approximation of the amount the condensing temperature will be lowered can be obtained from the Palmer and Miller[14] equation shown below. This expression assumes that the total duty, total heat transfer area, and total temperature rise of the cooling water remain the same in all cases and are divided equally among the condensing zones. The overall heat transfer coefficient and the log-mean Δt are assumed to be the same in each unit as they are in a single condenser. The relationships among the temperature range (TR), the terminal temperature difference (TTD), and the initial temperature difference (ITD) are shown in Fig. 2.7 for a two-zone surface condenser. The difference between the single-pressure condensing temperature and the average multipressure condensing temperature, ΔT_s, is then given by Palmer and Miller[14] as

$$\Delta T_s = \mathrm{ITD} - \frac{\mathrm{TR}(n-1)}{2n} - \frac{\mathrm{TR}}{n}\left[\frac{\left(\frac{1}{1-R}\right)^{1/n}}{\left(\frac{1}{1-R}\right)^{1/n} - 1}\right],$$

where

ΔT_s = difference between the single-pressure condensing steam temperature and the average multipressure condensing temperature, °F,

n = number of condensing zones,

TR = total temperature range of cooling water for single-pressure condenser, °F,

ITD = initial temperature difference between entering cooling water and condensing steam for single-pressure condenser, °F (ITD = TR + TTD),

TTD = terminal temperature difference,

R = TR/ITD.

If the terminal temperature difference is zero (ITD = TR and R = 1), the above expression reduces to

$$\Delta T_s = \mathrm{TR}\left(\frac{n-1}{2n}\right) .$$

Values of ΔT_s are plotted against TR in Fig. 2.8 for TTDs of 0, 1.5, 2.5, and 4°C. For typical conditions with a temperature range of 8 to 11°C (15 to 20°F) and a terminal temperature difference of about 2.8°C (5°F), the value of ΔT_s is 0.7 to 1.1°C (1.2 to 2.0°F). If the single-pressure condenser were condensing at 32°C and 36 mm of mercury absolute (90°F

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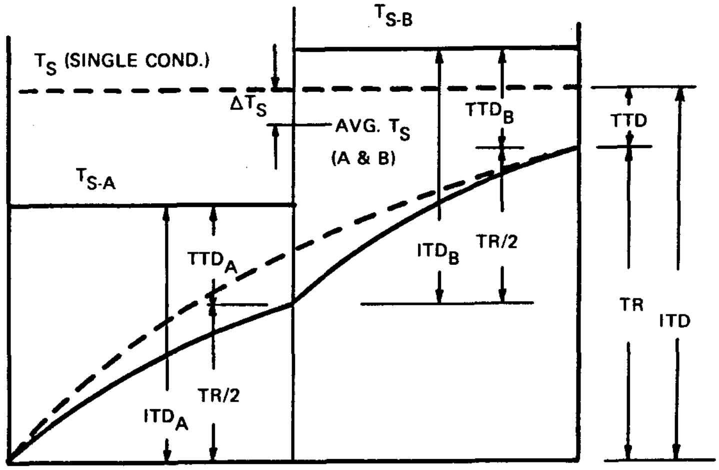

TEMPERATURE RELATIONSHIPS IN TWO-ZONE CONDENSER

T_S = CONDENSING STEAM TEMPERATURE
ITD = INITIAL TEMPERATURE DIFFERENCE
TTD = TERMINAL TEMPERATURE DIFFERENCE
TR = TEMPERATURE RANGE

Fig. 2.7. Temperature relationships in multipressure condensing system with two condensers in series in the cooling-water circuit.

and 1.42 in. of mercury absolute), the resulting decrease in the heat rate due to the reduced turbine back-pressure would be in the range of 0.9 to 3.5 W/kW (3 to 12 Btu/kWhr). There would also be a reduction in the heat rate of 0.9 to 1.5 W/kW (3 to 5 Btu/kWhr) because of increased condensate temperature. With direct-contact condensers, the value of ΔT_s would be higher, about 2.8°C (5°F), and the corresponding gains in heat rate would be greater. However, it is not clear whether geothermal power stations in the near-term state of the art will have more than one condenser per turbine and thus be suitable for multipressure arrangements. Further, the added complexity on the basis of relatively small improvements to the cycle heat rate may not be justified.

When using thermodynamic fluids other than water, the condensing pressures at ordinary condensing temperatures may be high enough to require the use of multiple condenser shells, which could be connected in series in the cooling-water circuit. The gain in efficiency resulting

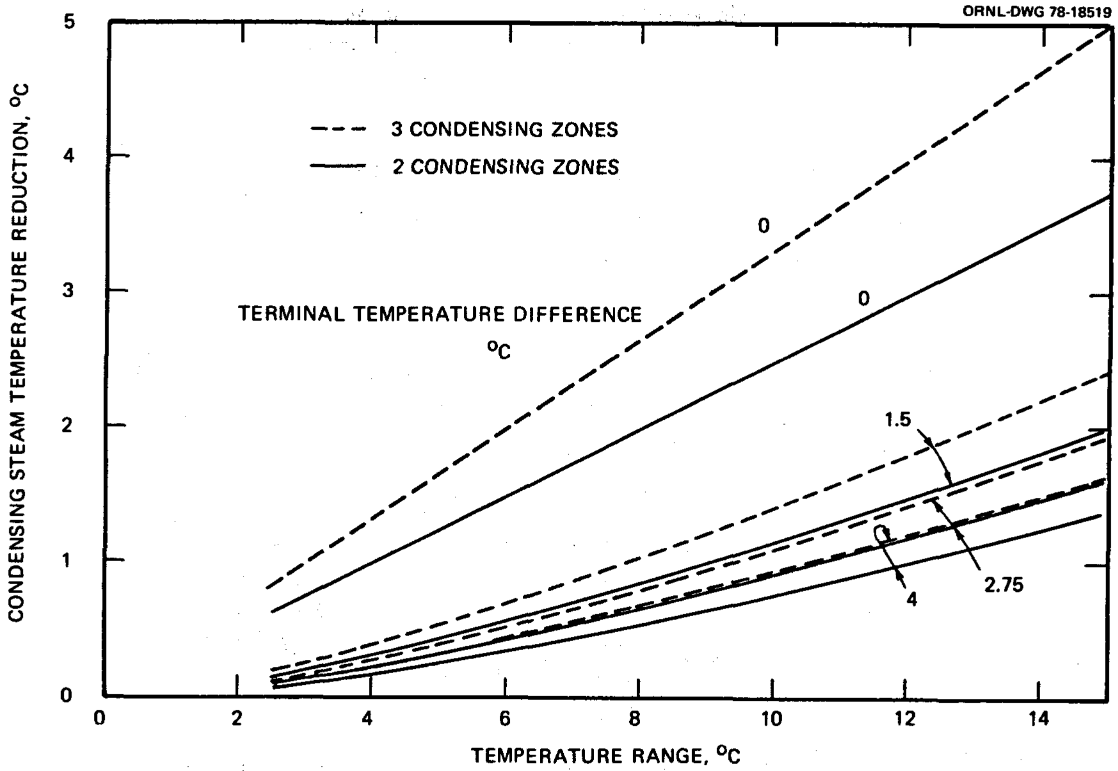

Fig. 2.8. Difference in condensing temperatures between single-pressure condenser and multipressure arrangements.

from the reduced back-pressure for these fluids has not been studied fully. Moreover, sufficient cost information has not been accumulated to serve as a basis for judgment at this time.

2.4.4 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 2.8. 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.[16] 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

Table 2.8. 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.

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.[16] Ammonia is a potential hazard, but it usually exists at too low a concentration to be of concern.[17]

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 markedly the performance of the condensing equipment. Othmer[18] investigated the effect of noncondensable gases in 1929, and his work was substantiated by Henderson and Marchello[19] in 1969 when they used steam-air and toluene-nitrogen mixtures condensing on a horizontal pipe. Othmer (as indicated in Fig. 2.9) showed that 1% of gas by volume in the steam to a surface condenser with a temperature difference of about 11°C (20°F) can cause the condensing coefficient to be about 6250 $W/m^2 \cdot K$ (1100 $Btu/hr \cdot ft^2 \cdot °F$) compared to about 11,400 $W/m^2 \cdot K$ (2000 $Btu/hr \cdot ft^2 \cdot °F$) when no gas is present. Figure 2.9 illustrates that if the gas concentration is 2% by volume the condensing coefficient is only about 4260 $W/m^2 \cdot K$ (750 $Btu/hr \cdot ft^2 \cdot °F$).

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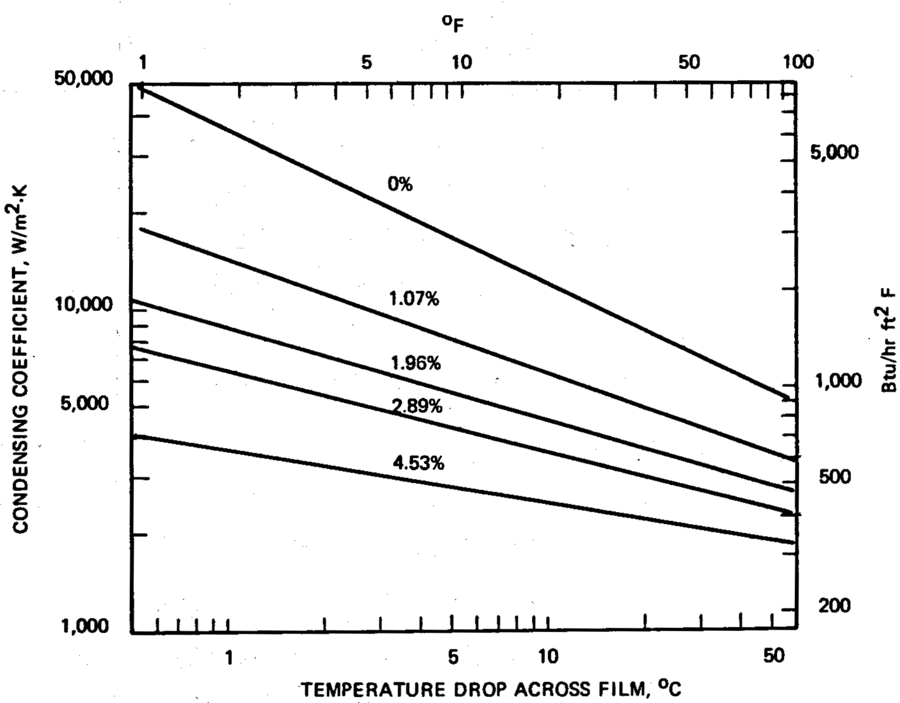


Fig. 2.9. Influence of air on the condensing film heat transfer coefficient of 110°C (230°F) steam. Source: D. F. Othmer, *Ind. Eng. Chem.* 21, 576 (1929).

Keeping in mind that the total pressure in a condenser is the sum of the partial pressures of the water vapor, air, hydrogen sulfide, carbon dioxide, or any other gases that may be present (Dalton's Law), the major undesirable effects of noncondensable gases on the condensing process in surface condensers can be explained as follows:

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[12] 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.

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.

These aspects interact to make the analysis of the condensing process with noncondensable gases present relatively complex, and the heat transfer relationships under such conditions have not been explored fully. The condenser designer must be careful not to apply heat transfer correlations or data that were taken on single tubes, where gas blanketing may not have been significant, to tube bundles. (This is also true because the rain of condensate on lower tubes in the bundles can cause flooding and significant impairment of the heat transfer coefficient.) Vendors of "off-the-shelf" equipment provide extra surface to compensate for the presence of noncondensable gases, for the cleanliness of surfaces, for the operation at off-design conditions, etc. The amount allowed is a matter of judgment and experience and is based on performance data for specific applications. Vendors use calculating procedures that are usually considered proprietary.

Eissenberg[20] tested a tube bundle with condensation taking place in the presence of varying amounts of noncondensable gas (nitrogen). He then correlated the combined effects of the amount of gas, the temperature difference, the steam temperature, and the steam velocity, using a modified Colburn-Hougen mass transfer factor. A critical steam mass velocity was found, below which noncondensables tended to accumulate on the surfaces. This minimum rate is a function of the condenser design and the condensing rate. Eissenberg suggests that rather than provide additional condensing surface to compensate for the presence of relatively large and unknown amounts of noncondensable gases, extra steam ejector or vacuum pump capacity would perhaps be as effective and require less capital expenditure.[21]

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.

The vacuum pumps are most commonly the rotary type, including the liquid-ring, rotary-screw, or the sliding-vane type. The liquid-ring units, such as those manufactured by Nash, have an advantage in that slugs of water in the suction can be accommodated, making an inlet separator unnecessary. The rotary-screw type, such as manufactured by Ingersol-Rand, does require a separator, and the bearings must be lubricated. Seals and oil slingers are designed to prevent loss of oil into the pumped fluid. The sliding-vane pump (exemplified by the Allis-Chalmers units) requires lubricants to be introduced into the pumped fluid at the suction and removed by wire mesh demisters, separators, and settling tanks at the exit. Typical oil consumption rates for small units are around 4 liters (1 gal) per day.[22]

The power requirements for noncondensable gas removal, whether it be steam for ejectors or electricity to drive vacuum pumps, increase almost linearly with the amount of gases to be handled. This effect, and the general method of calculating condenser noncondensable gas pressures, is illustrated in the equation below.

Dalton's Law states that the total pressure is the sum of the partial pressures of the constituent gases. The partial pressure of the water vapor at a given condensing temperature may be found in tables of the thermodynamic properties of water[23] or calculated from equations of state. The partial pressure of a constitutent gas may be found from the perfect gas relationship

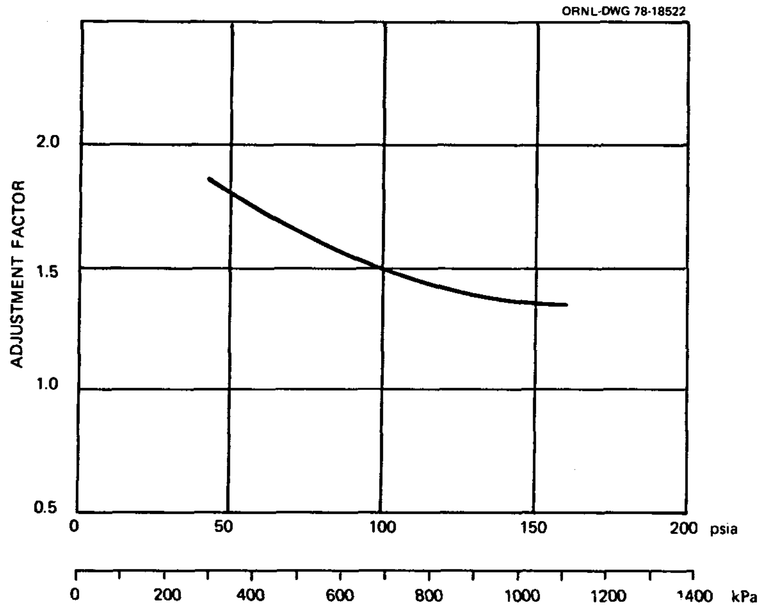
$$p = \frac{8314.91\, wt}{Mv_g} ,$$

where

p = partial pressure of the gas, N/m^2 or Pa (N/m^2 x 0.00014505 = psi),
w = weight of the gas per unit weight of the steam in the condenser,
t = condensing temperature, K,
M = molecular weight of the gas: air = 28.967, H_2S = 3408, and CO_2 = 44.005,
v_g = specific volume of steam at condensing temperature, m^3/kg (m^3/kg x 16.0184 = ft^3/lb).

A temperature drop across the gas film surrounding the tube can be assumed (perhaps 3°C) and the partial pressures at the tube wall calculated. The pressure fraction of each constituent gas is also the mole fraction, and if one knows the molecular weights of the gases, the weight fraction of each can be calculated. The weight of the gas-water vapor mixture that must be removed from the condenser per unit mass of steam entering can then be determined and the pumping effort estiamted.

Mah[24] has calculated the steam consumption for one- and two-stage ejectors for various amounts of gas present. Figure 2.10 shows steam



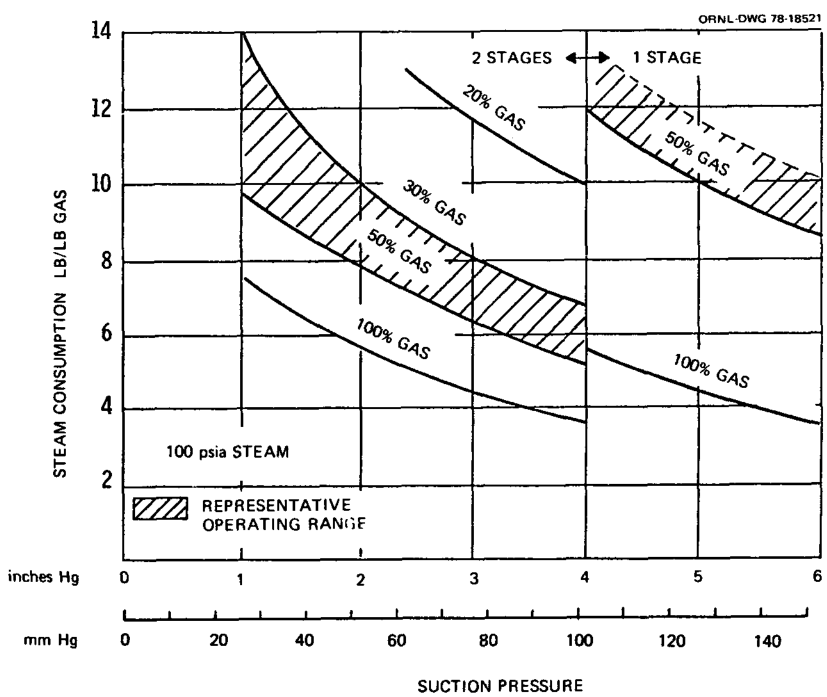


Fig. 2.10. Steam requirement for ejectors as a function of the amount of noncondensable gases present and ejector suction pressure when using 690 kPa (100 psia) of saturated steam supply. Adjustment factors for other steam pressures are given in the top graph. Source: Clifford Mah, "Effect of Noncondensable 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).

requirements at 690 kPa (100 psi) absolute operating steam. Adjustment factors for other steam supply conditions are given in Fig. 2.10. The amount of steam required for the ejectors as a percentage of the condenser flow can then be estimated as a percentage of the condenser flow (Fig. 2.11). If vacuum pumps are used, the work of compression is calculated assuming two-stage compression with optimum intercooler pressure (determined for the mixture by iteration) and assuming a compression efficiency of 83%. The estimates are given in Fig. 2.12 as a function of the amount of noncondensable gases to be handled. The power requirements for both ejectors and pumps are almost linear with the amount of gas to be handled, and the power expenditure becomes significant at amounts of over about 1%. (For simplicity, both Figs. 2.11 and 2.12 were calculated assuming that the noncondensable gases consisted of equal weights of air and carbon dioxide. The graphs should not be used for design purposes without verifying the appropriateness of these and the several other assumptions.)

The presence of noncondensable gases in a direct-contact, or mixing, condenser does not have as serious an effect on the heat transfer performance as in surface condensers, but the gases must be removed from the condenser if the low turbine back-pressure is to be maintained. The amount of gases to be removed from the mixing condenser, however, may be considerably greater than for an equivalent surface condenser because of the dissolved or entrained gases in the cooling water sprayed into the unit. As the cooling water enters the mixing condenser, it is both heated and reduced in pressure. Both of these processes reduce gas solubility. Analysis of the performance of direct-contact condensers is impeded by the need to know stripping and diffusion coefficients, which have not been studied extensively to date.

The ejector steam consumption at various operating geothermal power installations is shown by Mah[24] in Fig. 2.13. 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%.

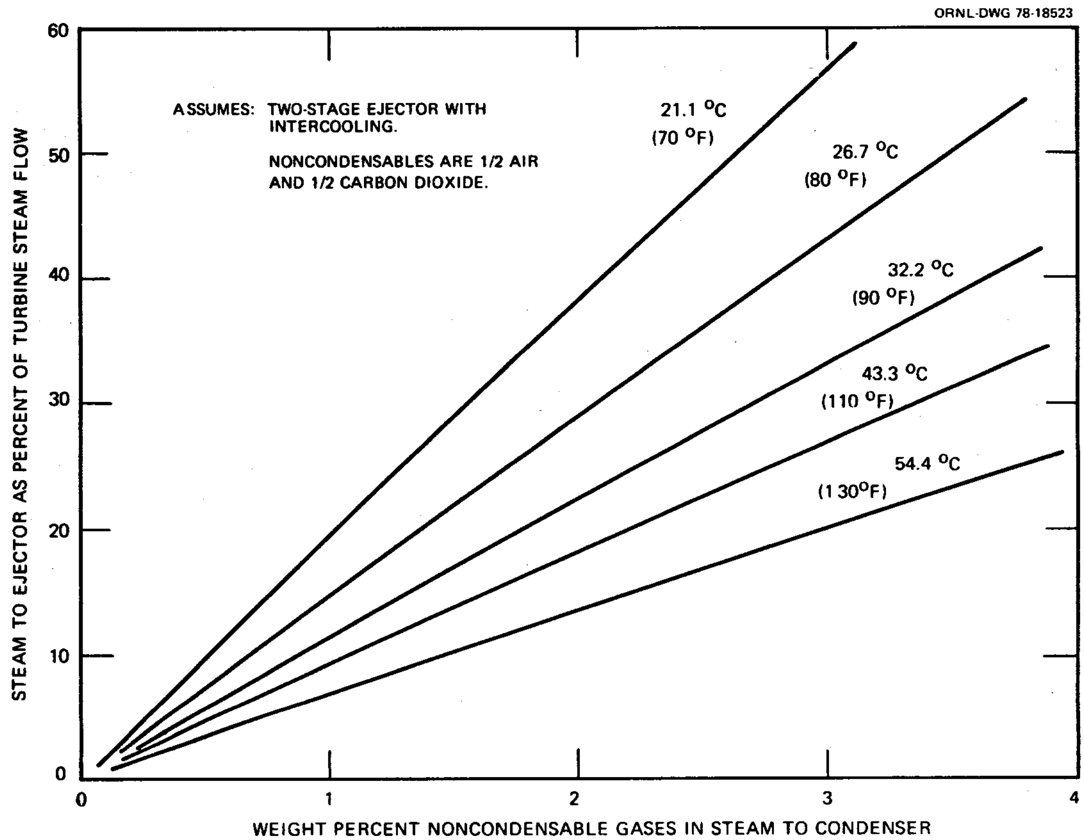


Fig. 2.11. Two-stage ejector steam consumption to remove noncondensable gas from surface condenser. Note assumptions.

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

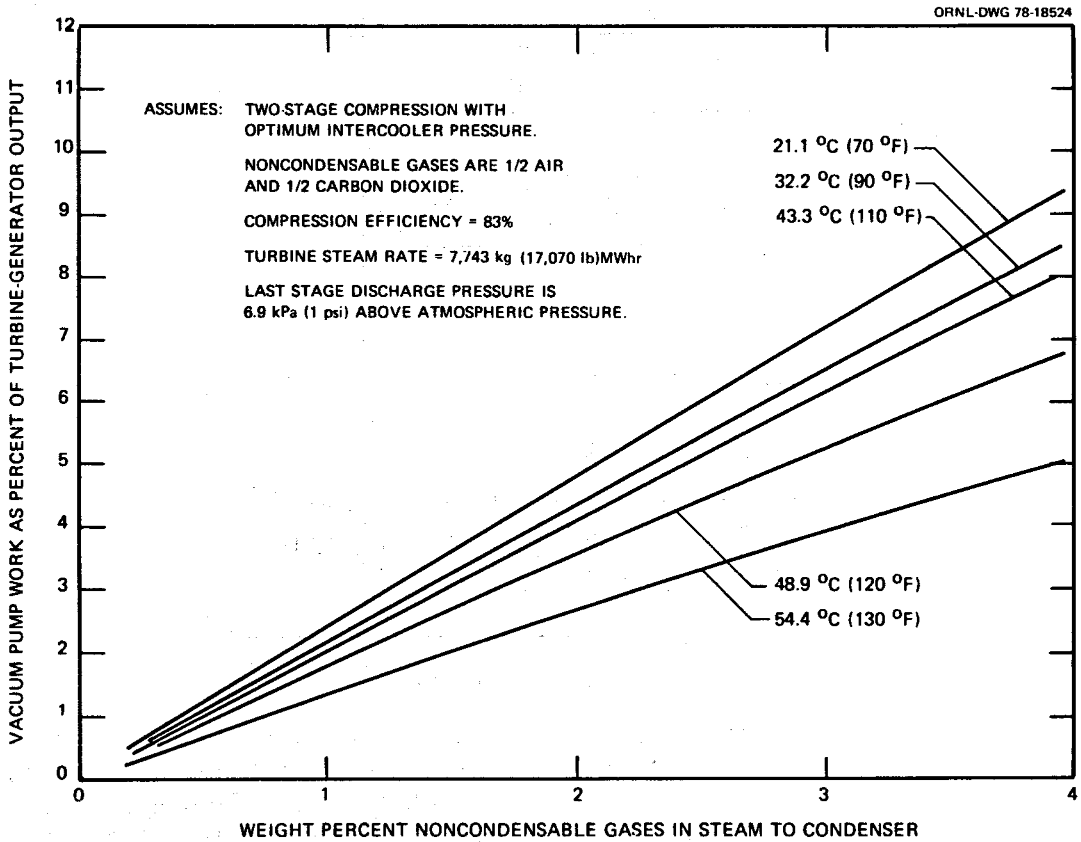


Fig. 2.12. Estimated two-stage vacuum pump work to remove noncondensable gases from surface condenser. Do not use for estimating purposes without checking the several assumptions shown in the figure.

geothermal stations using steam flashed from a geothermal fluid, the condensate can be used as makeup for the cooling towers. The quantity of condensate available should be sufficient because the heat given up by condensation of a unit mass of steam can be dissipated in the tower by evaporation of a unit mass of the condensate. 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

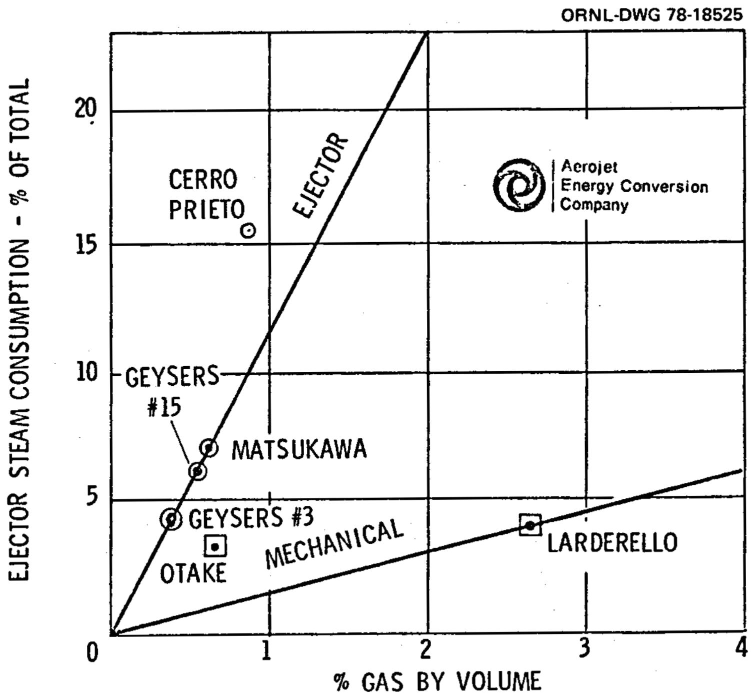


Fig. 2.13. Ejector steam consumption at some operating geothermal power stations. Source: Clifford Mah, "Effect of Noncondensable Gases on the Performance of Rankine Cycles," presented at the Eighth CATMEC Meeting, San Diego, Calif., Feb. 21, 1978. Reprinted by permission of the Aerojet Energy Conversion Company (Aerojet Code PRA-SA 1 November 1978).

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. In more typical circumstances, the quality of the makeup water is high enough so that the only necessary treatment is periodic additions of small amounts of chlorine to retard the buildup of biological growths in the circulating systems. These amounts are usually added as hypochlorites, but where relatively high chlorine addition rates are required and saline water is available, studies have indicated that it may be more economical to generate chlorine by electrolysis.

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.[25] 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.

Shutdown of power stations for manual cleaning of condenser tubes can be costly because of revenue loss from sale of power; thus, economic studies are required to equate the cost of condenser capacity loss against shutdown costs. Presentation of guidelines with regard to typical cleaning schedules is impossible because of the great difference between fouling tendencies at specific plants. 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.

Several methods have been developed, or are proposed, for cleaning condenser tubes without taking the unit out of service. These methods include Amertap, MAN, water jets, slurries of abrasives, hydraulically propelled brushes, and possible pretreatments. One of the more commonly used methods is the Amertap system.[13] Amertop involves using the coolant pressure to force sponge rubber balls through the tubes on the average of about one every 5 min. The balls wipe the surface and materially aid in maintaining the condenser efficiency. They are recovered from the effluent water and recirculated. The length of time the balls can be used before replacement varies with the specific application. The MAN system consists of a stiff brush retained in each condenser tube by suitable end-cages. The circulating-water piping is arranged to allow periodic reversal of flow. Each flow reversal drives the brush through its respective tube.[26] Although the economics of mechanical tube-cleaning systems generally favor larger power stations, the systems have been used in smaller applications. However, no specific rules can be given concerning their usefulness in geothermal power plants. Consideration of the costs of mechanical cleaning systems must include a comparison between the installation and operating costs and the cycles of concentration in the cooling tower.

Besides causing deterioration of the heat transfer rate, deposits can also accelerate corrosion of the tubes. Some of the more commonly used tubing materials are copper, cupro-nickels, aluminum, stainless steel, and titanium. When stainless steel is used as tubing material, two of the more troublesome deposits are calcium carbonate containing chlorides and hydrous oxides of maganese (in cases where seawater is circulated). Tubing materials containing no copper are particularly susceptible to the deposition of slimes. The tubing material itself and the cooling water may be incompatible, particularly if saline water is used. The stress corrosion cracking of stainless steels, the pitting of aluminum, and the incompatibility of ammonia and the copper-bearing alloys are particular areas of trouble. Despite the fact that the literature contains a great deal of information, much study is needed on the corrosion of condenser tubes in geothermal applications. Use of titanium tubes eliminates many of the concerns about corrosion but at some cost penalty, as discussed in Sect. 2.4.7.

The condensers are often supplied in the purchase of steam turbine generators. In such cases, the vendor assumes responsibility for meeting the performance specifications for the condenser. However, the burden may be on the purchaser to specify correctly the cooling-water quality that will exist and to maintain the quality during operation.

If condensers of special design are fabricated, the manufacturer guarantees only to meet the fabrication specifications, leaving the responsibility for performance with the customer or his consultants. As in any instance where knowledge and experience are important, such information tends to have market value and is not freely given. Much of the investigative work on condenser fouling, condenser design, and condenser costs, etc., is considered proprietary; there is relatively little definitive information available in the open literature.

2.4.6 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.[27] Very small leaks can be detected by analysis of the water chemistry of the turbine cycle, and nearly all of the surveyed stations have automatic analyzers for this purpose. Hydrotesting of the steam side was reported as the most sensitive method of locating a leak, but plastic film or foam applied to the tube sheet of an evacuated condenser is also used.

The tube exposure environment was defined in terms of three circulating-water categories: freshwater once-through, salt water, and wet cooling towers with freshwater makeup. Three condenser service sections were also defined: impingement section, main condensing section, and air removal section. 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,*[27] 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.[27] 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. The major excitation forces causing tube vibration result from the cross-flow steam velocity. The resulting drag force and vortex shedding cause tubes to vibrate at the natural frequency, f_n, given by the following equation:[27]

$$f_n = C(gEI/wL^4)^{1/2} ,$$

where

g = gravitation constant,

E = modulus of elasticity,

I = moment of interia of the tube cross section,

w = weight per unit length of the tube filled with water,

L = tube span between supports,

C = end support constant.[28]

The deflection, Y, of the vibrating tube is given proportionally as[27]

$$Y \propto kW_D L^4/EI ,$$

where

W_D = (drag) force per unit length causing the deflection,

k = end support constant.[28]

For a given steam flow condition, W_D, applitude reduction may be accomplished by (1) decreasing the tube span, L; (2) increasing the modulus of elasticity, E; or (3) increasing the moment of inertial, I. The natural frequency will also increase with these changes. 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.[29]

2.4.7 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

Table 2.9. Comparative corrosion resistance of tube materials

	Titanium	Copper-nickel	Stainless steel
Ammoniated condensate	Excellent	Fair	Excellent
Velocity effects	Excellent	Fair	Excellent
Hydrogen sulfide	Excellent	Poor	Fair
Chlorides	Excellent	Fair	Poor
Deposit attack	Immune	Susceptible	Very susceptible
Stress corrosion cracking	Immune	Good	Poor
Fouling resistance	Excellent	Poor	Excellent
Metal ion pollution	None	Considerable	None
Metal ion carry-over	None	Considerable	None

Source: *Titanium Tubing for Surface Condenser Heat Exchanger Service,* Timet Bulletin SC-4, Titanium Metals Corporation. Reprinted by permission.

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 2.14 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.

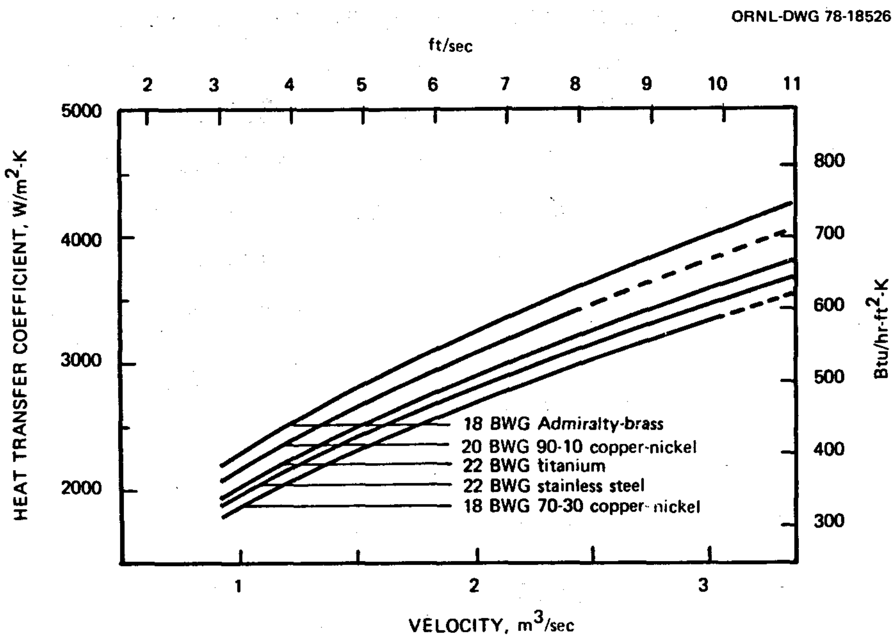


Fig. 2.14. Overall heat transfer coefficient as a function of water velocity in 2.5-cm (1-in.) OD tubes. The values shown are the result of applying appropriate fouling factors and adjustments for wall thermal conductivities to the heat transfer coefficients. Source: *Titanium Tubing for Surface Condenser Heat Exchanger Service,* Timet Bulletin SC-4, Titanium Metals Corporation. Reprinted by permission.

Number 22 BWG titanium and stainless steel and No. 18 BWG copper-nickel alloys will all have about the same resistance to vibration. This factor affects the spacing of tube supports and is of particular importance when considering replacement of existing tubes with titanium because the spacing of the tube supports cannot be easily altered.

One of the most common and economical methods of joining titanium tubes to tube sheets is by roller expansion. Mechanical bonds are subject to leakage, however, and where the consequences of leakage are severe, such as in the mixing of highly saline water with ammonia, etc., the joints may not be acceptably tight. Joints such as O-rings and X-rings have been used, but they also may not meet exacting leakage specifications. Welding, usually on the face side, produces the most leak-proof joints. There are many examples of successful welding of titanium tubes to titanium sheets using inert atmosphere techniques. Explosion bonding has been used successfully to join titanium to titanium, or titanium to steel tube sheets. Titanium may also be successfully brazed to titanium or steel using either a vacuum or high-purity inert atmosphere. Silver braze alloys are the most useful.[30]

The 1977 cost of uninstalled titanium tubing was in the order of $54/m^2$ ($5/ft^2$). This cost was subject to specific market conditions, the quantity purchased, and the purchase specifications, such as the degree of testing and inspection required. An appreciation of the comparative cost of titanium may be seen in Table 2.10, which lists 1975 costs of some common tube materials.

Table 2.10. Comparative costs of condenser tubing in 1975

Material	$/ft^2
Admiralty metal	2.56
Type 304 stainless steel	2.37
Titanium	3.97
Allegheny-Ludlum AL6X	4.09

Source: J. K. Rice, "Evaluating Cooling Systems with Zero Aqueous Discharge," pp. 81-86 in *The 1976 Generation Handbook*, McGraw-Hill, New York, 1976.

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. 2.15. 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.[31]

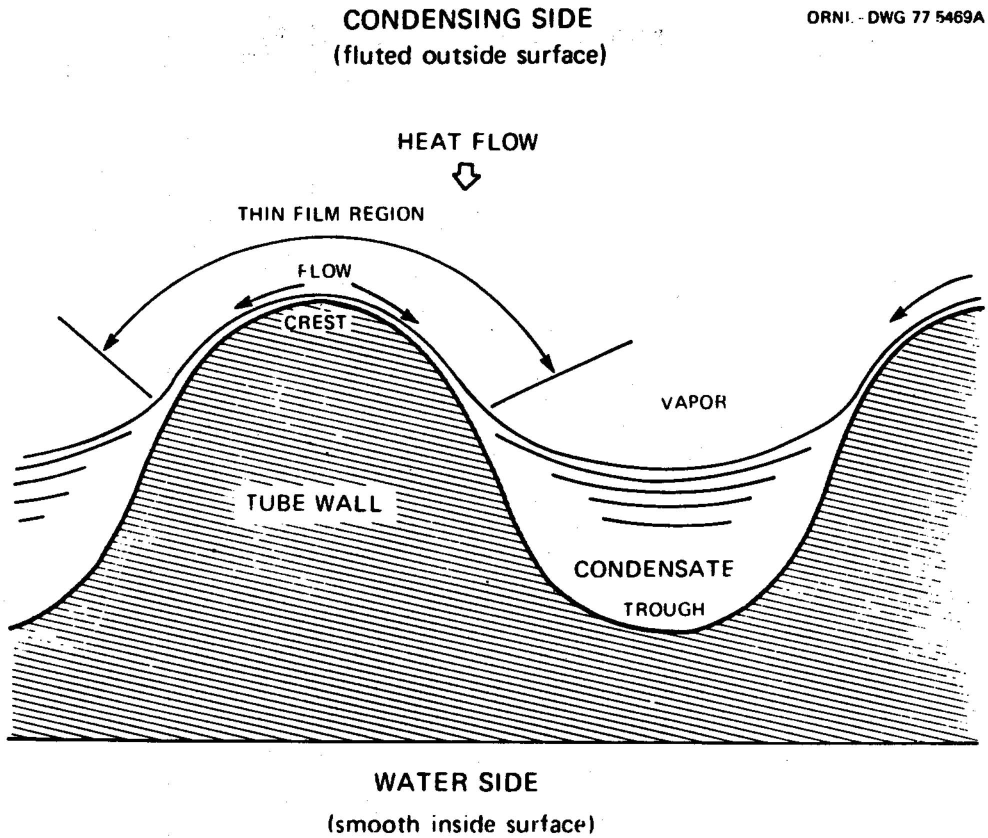


Fig. 2.15. Fluted tube principle of operation (condensation mode). Surface tension forces act to push condensate from crests into troughs. Source: S. K. Combs, *An Experimental Study of Heat Transfer Enhancement for Ammonia Condensing on Vertical Fluted Tubes*, ORNL-5356 (April 1978).

Experimental studies have been made of the performance of vertical tubes with enhanced surfaces on which various fluorocarbons, ammonia, and isobutane were condensed.[32] 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. 2.16. Most of the tubes tested were of aluminum, but other materials were studied also (Fig. 2.16). 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.[32] The condensing film coefficients obtained when condensing ammonia are shown in Fig. 2.17, and the coefficients for condensing isobutane are shown in Fig. 2.18. The

Fig. 2.16. Characteristics of fluted tubes. Source: 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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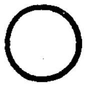
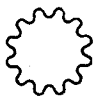
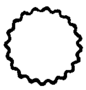
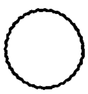
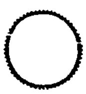
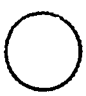
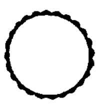
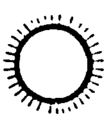
Cross section	Tube designation	Material	External perimeter (cm)	External surface area (m^2)	Number of external flutes	Number of internal flutes
	A	Aluminum	8.00	0.0973		
	B	Al-brass	11.94	0.1296	12	12
	C	Al-brass	8.90	0.0967	20	20
	D	CuNi (90 10)	8.90	0.0967	36	36
	E	Aluminum	12.71	0.1490	60	0
	F	Aluminum	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.

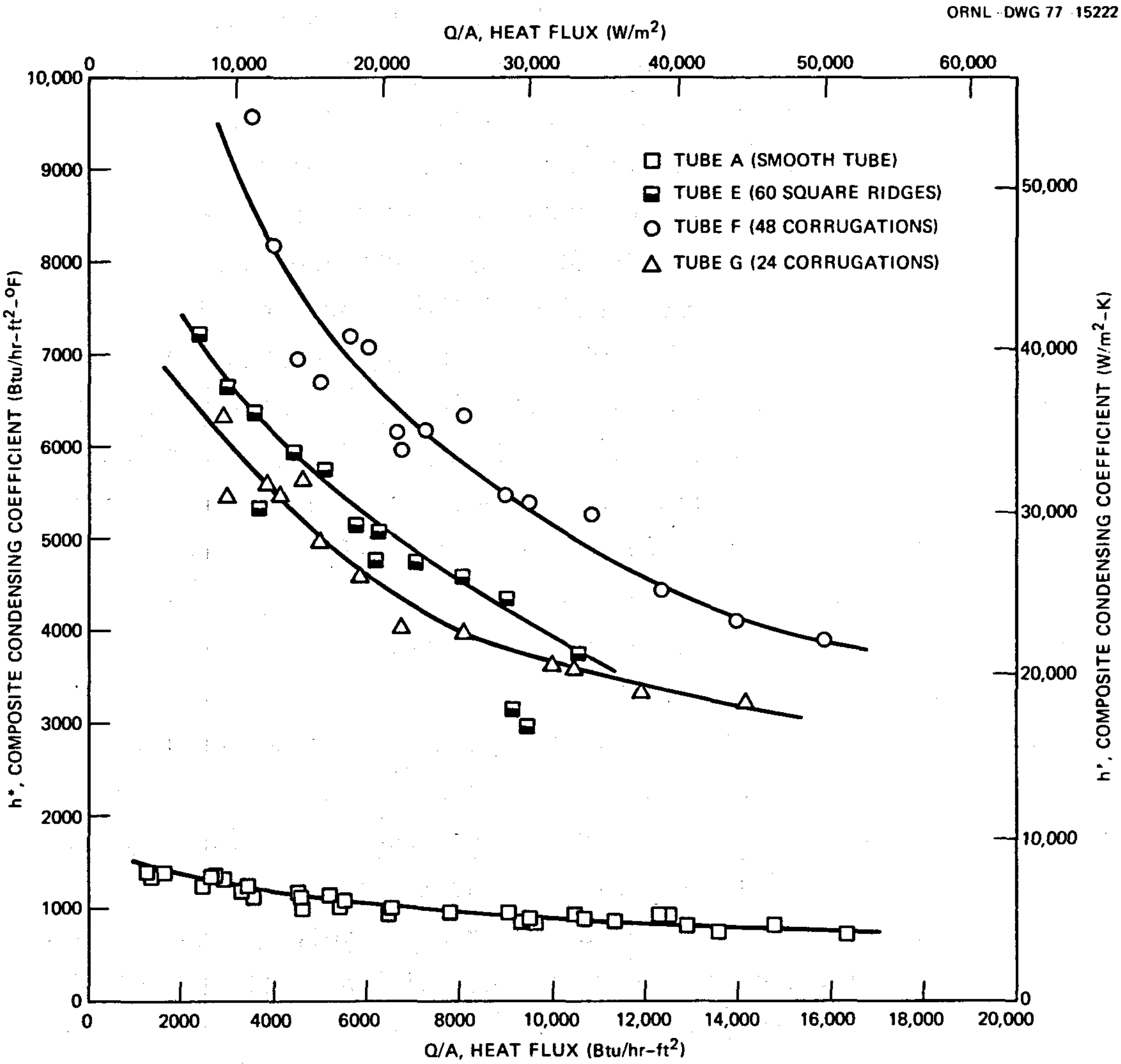


Fig. 2.17. Composite condensing heat transfer coefficients for ammonia using enhanced-surface tubes. Source: S. K. Combs, *An Experimental Study of Heat Transfer Enhancement for Ammonia Condensing on Vertical Fluted Tubes*, ORNL-5356 (April 1978).

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. 2.17), 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 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.[32]

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.[32]

Fouling of enhanced tube surfaces has a more pronounced effect on the heat transfer coefficient than fouling of smooth tubes. Figure 2.19 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 $hr \cdot ft^2 \cdot °F/Btu$) 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

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

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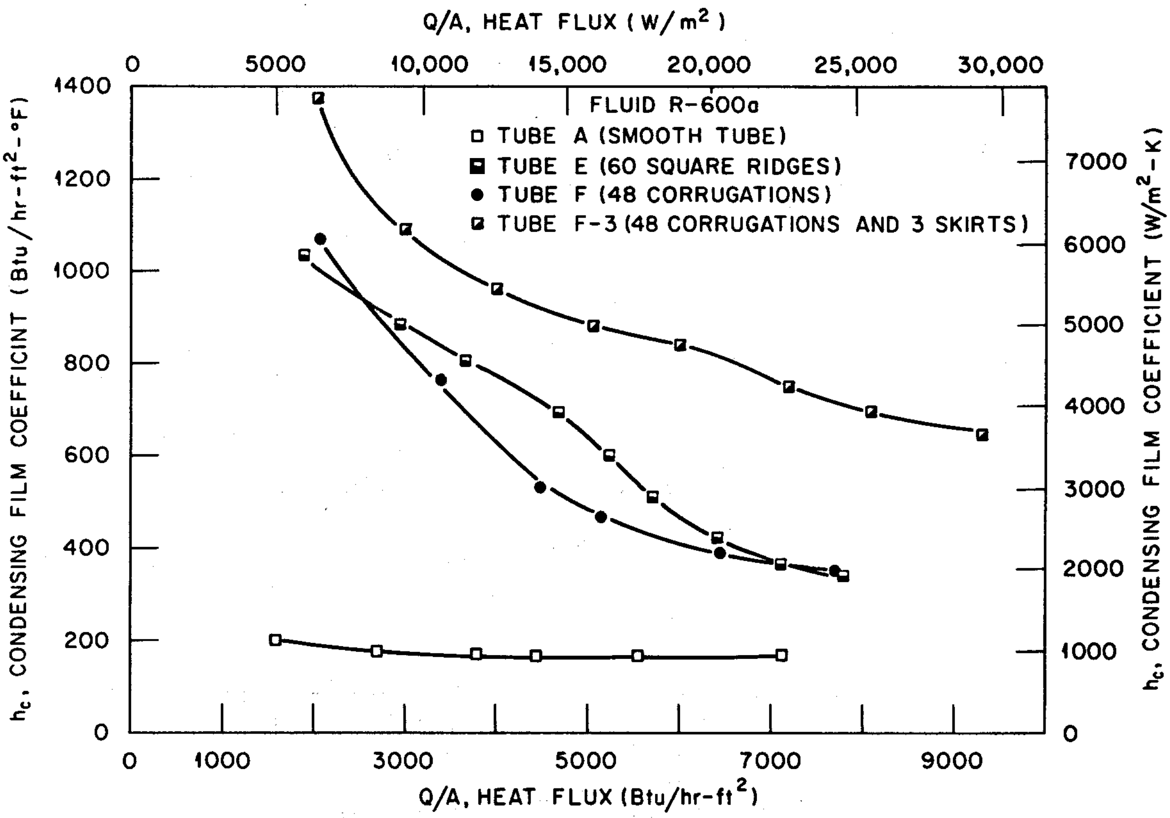

Fig. 2.18. Condensation heat transfer coefficient for isobutane using enhanced-surface tubes. Source: J. W. Michel et al., *Energy Div. Annu. Prog. Rep. Period Ending Sept. 1977*, ORNL-5364, Fig. 5.4, p. 185.

composite condensing coefficient (Fig. 2.18). 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. 2.16) would make this tube outperform the type F tubes with ammonia on a unit-length basis.[31] 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

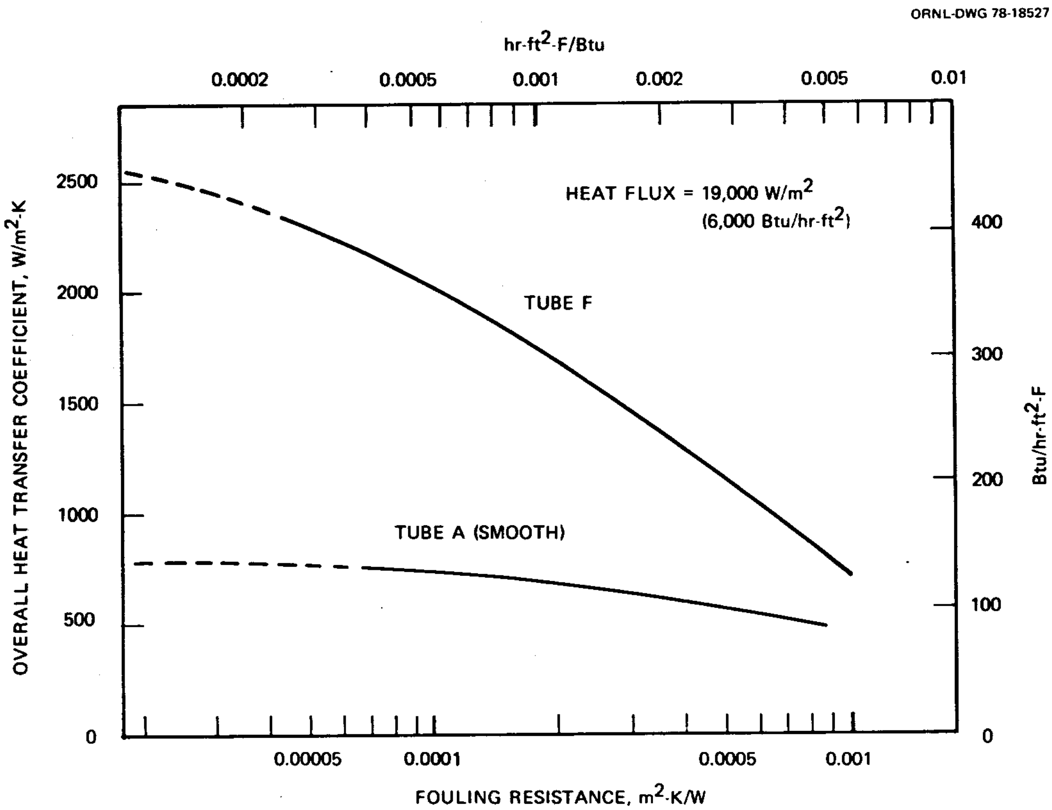

Fig. 2.19. Effect of fouling resistance on overall condensing heat transfer coefficient of enhanced-surface tubes.

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.[32]

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.

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3. HEAT DISSIPATION SYSTEMS AND EQUIPMENT

3.1 General

Electric power stations are normally classified in terms of their net electrical capacity in megawatts. This classification provides a mental image of the general size and cost of the plant, with the steam supply system and the turbine-generator unit dominating. In picturing a geothermal power station, however, it is necessary to include the waste heat rejection system as one of the dominent features. A 50-MW(e) geothermal power plant, for example, would probably require a wet cooling tower of the size usually asscoiated with a 250- or 300-MW(e) conventionally fueled station. If the geothermal plant were located in a water-short area, the cooling towers would probably have to be the wet/dry or dry type, adding further to the cost and size of the equipment. The waste heat rejection system for a geothermal plant is therefore a principal feature of the station, and selection of the equipment is one of the major design considerations.

In view of the growing scarcity of water, the Department of Energy, the Electric Power Research Insitute, and others have recognized the probable future need for wet/dry and dry cooling towers for conventionally fueled as well as geothermal power plants, and have sponsored a relatively large number of studies on the subject. Although this review cannot be exhaustive, selected references will be presented to establish the general state of the art and the probable trends as they relate to near-term geothermal power.

3.2 Once-Through Cooling Systems

Once-through cooling systems take water from an abundant source, such as an ocean, large lake, or river; pump it through the turbine condenser, where the water temperature is raised 8 to 11°C (15 to 20°F); and return it to the source. This method is usually the most economical means of heat rejection from a power station. Capital costs of a once-through system for a geothermal power plant would probably fall in the range of

$15 to $30 per kilowatt of installed capacity. Up until the 1960s it was the most favored method of heat rejection, and water availability was a prime requirement in selecting a site for a new power station. However, once-through cooling systems usually have more environmental impact than other methods of heat dissipation because of the relatively large amounts of water that must be drawn into the intakes. A geothermal power station with a thermal efficiency of about 15% would require about 0.17 m^3/sec (6 cfs) of cooling water per megawatt electrical of installed capacity.

The primary environmental impacts are the entrainment of fish eggs and plankton, which have a high mortality rate in the passage through the condenser, and the killing of fish due to impingement on the intake screens.[1] Attempts to prevent fish from entering the intake area, for example, by means of air-bubble curtains or underwater noise generators, have achieved only limited success. Equipping the traveling screens with fish lifts, or baskets, has given better results, but the impacts are still being evaluated and are of concern. To keep impingement losses within reason, it is necessary to design water velocities approaching the screens low enough for fish to escape laterally and to provide a passageway for the fish to return to the water body. An approach velocity greater than 0.3 m/sec (1 fps) is usually considered unacceptable. (The approach velocity is not to be confused with the face velocity at the screen. The latter takes into account the obstructions to flow due to the wire mesh, screen support frames, and structures. The free area for flow is usually taken as about 65% of the total face area.)

The discharge of the heated water back into the source can create objectionable zones of elevated temperatures that can disturb migration patterns of fish, influence spawning, or be lethal to some aquatic species.[2] Heating of significant volumes of water to temperatures greater than 32°C (90°F), or to temperatures of 1.7 to 2.8°C (3° to 5°F) in excess of ambient, usually violates state water quality criteria and raises serious questions of environmental acceptability. A design objective is to mix the discharged water as rapidly as possible with the receiving water by use of jets, or nozzles. The desirable distance for the nozzles to be submerged beneath the surface, and the spacing between them, is site specific.[3] In many instances, the only acceptable arrangement is a diffuser pipe that distributes the water over a relatively large area.

Fouling of the condenser heat transfer surfaces is a primary consideration in once-through cooling systems, particularly if the water is drawn from an ocean or estuary. Few locations exist today where water is available in sufficient quantity and is also of high enough quality to be used without chemical treatment and/or routine mechanical cleaning of the condenser tubes.

Based on present knowledge of the most feasible locations for geothermal power stations, it is reasonable to conclude that water sources are not likely to be available for once-through cooling systems. In any event, once-through systems are now so lacking in favor with the Environmental Protection Agency and other regulatory bodies that construction permits are most difficult to obtain.

3.3 Cooling Lakes and Ponds

The costs and environmental impacts of using relatively large amounts of land for a cooling lake, the lack of suitable sites, and the evaporation and seepage losses from such a water body have all generally discouraged construction of lakes solely for the purposes of waste heat dissipation. If, however, it is desirable to provide a storage lake for other purposes, such as for recreation, to ensure an adequate water supply during dry seasons, to implement stream flows, or to control runoff, then use of the same lake for heat dissipation may be an attractive possibility. One important advantage of lakes as a heat sink is that they have a thermal inertia which enables a power station to better meet daytime peak loads. Lakes created to serve power stations have historically assumed recreational and wildlife importance, and aspects such as fish impingement on the intake screens, waterfowl habitat, minimum drawdown to protect swimming and boating, and aesthetic aspects have become serious environmental issues.

A water body will naturally evaporate at a rate of 2.5 to 5 mm (0.1 to 0.2 in.) per day, depending upon the meteorological conditions.

If the surface temperature of the water is raised by using it as a heat sink for a power cycle, the surface evaporation rate increases. This induced evaporation will not be as great as that required in a cooling tower to dissipate the same amount of heat because the water body has the advantage of also losing heat by radiation. The total of the natural plus induced evaporation rates, however, can be greater than the water consumption of a cooling tower having the same duty. Water seepage from an unlined or leaky impoundment may also be a significant water loss. Solely from the standpoint of water conservation, construction of a lake specifically for the purposes of heat dissipation is likely to be a poor choice when compared to spray canals or wet cooling towers.

A rough notion of the lake area required to dissipate heat can be obtained by assuming that 4000 m^2 (1 acre) can dissipate about 1.5 MW (5×10^6 Btu/hr) of heat. It must be cautioned, however, that this value can vary widely with the water temperature, the wind velocity, and other meteorological conditions. Construction costs can also vary markedly, depending on the site location, character of the terrain, and whether the basin must be lined to reduce seepage losses. In 1976, cooling lakes for nuclear power plants were estimated to cost in the range of $14 to $22 per kilowatt of installed capacity; a geothermal power station rejecting two to four times as much heat per kilowatt could thus have 1978 costs of over $100 per kilowatt.

The rate of heat transfer from a water surface has been studied by several investigators. Some of the more representative results have been obtained using the approach of Edinger and Geyer.[4] The method involves the concept of an equilibrium temperature, t_E, and a surface heat exchange coefficient, K. The heat flux from the surface is expressed as

$$\phi = K(t_s - t_E) ,$$

where t_s is the actual water surface temperature and t_E is defined as the water surface temperature that, for a given set of meteorological conditions, makes the back-radiation, evaporation, and conduction losses equal to the heat inputs by solar radiation. A water body with t_s greater than t_E will tend to decrease in temperature, and t_s will approach t_E.

If t_s is less than t_E, the opposite will be true. The surface heat exchange coefficient, K, is defined to give the incremental change of net heat transfer induced by an incremental change in the water surface temperature. In 1973 Ryan and Harleman[5] devised a method for approximating the values of t_E and K. The procedure summarized below is taken from that source. (For a more detailed treatment, see refs. 4 and 5.)

Input data:

t_D = dew-point temperature, °F,
t_a = ambient air temperature measured 6 ft above surface, °F,
RH = relative humidity, as a decimal,
W = wind speed above surface, mph,
ϕ_d = direct incoming solar radiation, Btu/day·ft^2 (see Fig. 3.1),
C = Brundt coefficient (i.e., a function of the air temperature and the ratio of the measured solar radiation to the clear-sky solar radiation). A typical value is 0.6.

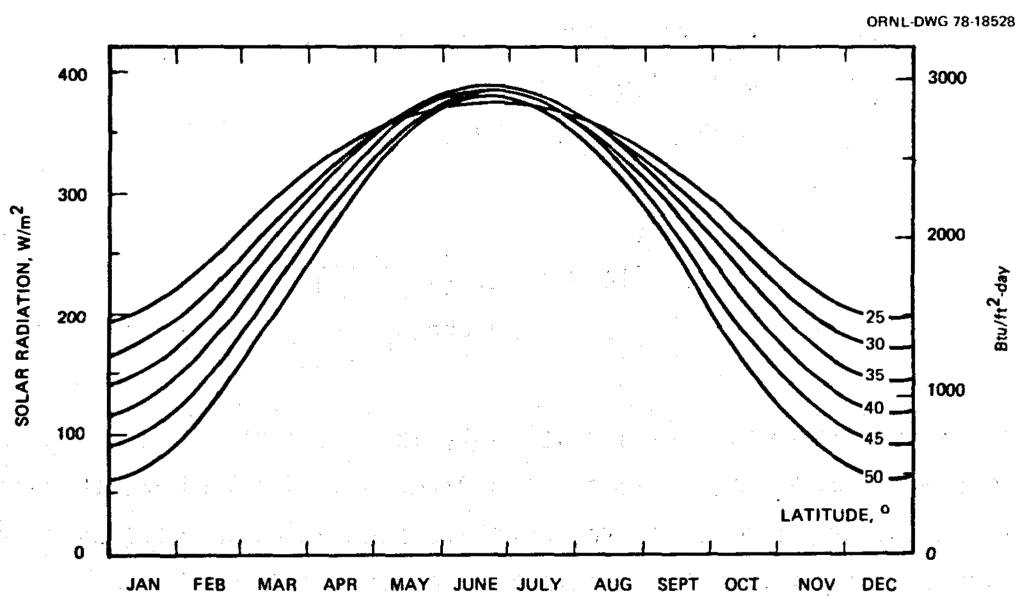

Fig. 3.1. Clear sky radiation. Source: J. R. Edinger and J. C. Geyer, *Cooling Water Studies for Edison Electric Institute, Project No. RP-49 — Heat Exchange in the Environment*, The Johns Hopkins University, Baltimore, June 1, 1965.

Calculate:

$$T_a = t_a + 460 \text{ , °R ,}$$
$$e_a = (25.4)(\mathrm{RH})\exp[17.62 - (9500.8/T_a)] \text{ , mm Hg ,}$$
$$\phi_a = 4.18 \times 10^{-8}(T_a)^4(C + 0.031\sqrt{e_a}) \text{ ,}$$
$$\phi_r = \phi_a + \phi_d \text{ .}$$

Assume t_E and calculate:

$$T^* = (t_E + t_D)/2 + 460 \text{ , °R ,}$$
$$\beta = [25.4(9500.8)\exp 17.62 - 9500.8/T^*]/(T^*)^2 \text{ ,}$$
$$t_E = \frac{\phi_r + 17W(\beta t_D + 0.255t_a) - 1600}{23 + 17W(\beta + 0.255)} \text{ .}$$

Assume a new value for t_E, repeat previous step, and continue iteration until a satifactory value is found for t_E.

Calculate:

$$T_s = t_s + 460 \text{ , °R ,}$$
$$e_s = (25.4 \exp(17.62 - 9500.8/T_s) \text{ , mm Hg ,}$$
$$T_{s-v} = T_s/[1 - (0.378/760)e_s] \text{ , °R ,}$$
$$T_{a-v} = T_a/[1 - (0.378/760)e_a] \text{ , °R ,}$$
$$\Delta\Theta_v = T_{s-v} - T_{a-v} \text{ ,}$$
$$t_s^* = 0.55t_s + 0.45t_E \text{ ,}$$
$$K = 23 + (\beta + 0.255)[14\,W + 22.4(\Delta\Theta_v)^{1/3}] + 7.5(\Delta\Theta_v)^{-2/3}[e_s - e_a + 0.255(t_s^* - t_a)] \text{ , Btu/day·ft}^2\text{·°F .}$$

The literature contains a relatively large amount of information on cooling pond performance. Gibbons and Pike[6] have listed more than sixty studies on cooling ponds. Jirka, Abraham, and Harleman[7] have assessed the techniques used to predict temperature distributions in ponds. Field temperature data for heated surface discharges into Lake Michigan have been analyzed by Kyser, Paddock, and Policastro.[8] Neill and Gibbons[9] discuss in some detail the procedures for sizing a cooling pond for a thermal power plant using steam turbines with a surface condenser. They show that the optimum size depends upon many parameters, including the plant efficiency, local climate, turbine exhaust temperature, condenser area, the water circulation rate, etc. The work is based on performance data taken under summer conditions in the southeastern United States. Mixing and thermal stratification in the pond are included as parameters. Some of the results are summarized in Fig. 3.2. In a sample calculation assuming midsummer conditions for a fossil plant having a thermal efficiency of 41%, Neill and Gibbons estimate about 10,000 m^2 (2.5 acres) of pond area would be required per megawatt (electrical) of installed capacity. Assuming the same conditions, but a geothermal plant with an efficiency of 15%, about four times that area would be required per megawatt (electrical); if the efficiency is 10%, about 6.4 times that area, or 65,000 m^2 (16 acres), would be needed per megawatt (electrical) of capacity. These adjustment factors must also be used with Fig. 3.2 if it is applied to geothermal stations of like efficiencies.

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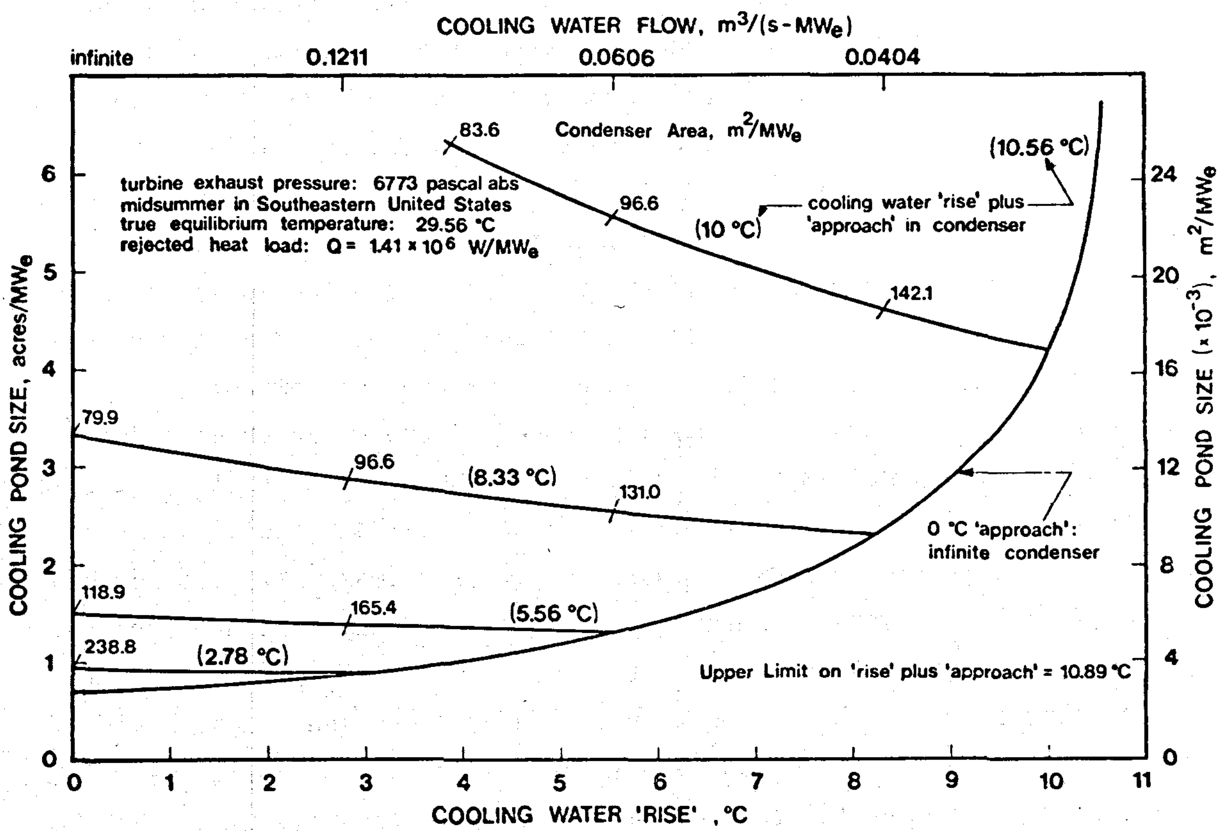

Fig. 3.2. Cooling pond size as function of temperature range and condensing temperature. Source: J. S. Neill and J. H. Gibbons, *Sizing a Cooling Pond for a Power Plant*, 75-WA/HT-63, American Society of Mechanical Engineers (August 1975). Reprinted by permission.

3.4 Spray Ponds and Canals

In contrast to cooling lakes or ponds, where the water is cooled primarily by evaporation from the surface, spray ponds and canals spray the water to be cooled into the air. By increasing the evaporation rate, the land area requirements are only about 5% of those needed for a cooling lake. The cooling process, which is about 80% by evaporation and 20% by convection, is essentially the same as that taking place in a cooling tower except that it occurs in the open atmosphere and the air flow is not induced, or channeled, as in a tower.

Spray ponds have been widely used in the past by industry and by smaller power plants. Some of the advantages are simplicity, low maintenance, ease of repair, and relatively low operating power requirements. Spray ponds are less visible than cooling towers and are judged by many to be more aesthetically acceptable. The noise level created by a spray pond of equivalent duty would probably be about equal to that of a natural-draft cooling tower but is probably less than that of a mechanical-draft cooling tower. The fogging effects of a spray pond tend to be more localized, and drift losses also tend to be confined to the immediate area, although more recent data indicate that the drift deposition may be more widespread than was first thought. Some important disadvantages of spray ponds and canals include the following: the land area requirements are substantially greater than those for cooling towers; the performance, being subject to the wind speed, is more variable; there may be freeze-up problems in the wintertime; the approach to the wet-bulb temperature is limited; and the design factors relating the fluid mechanics and thermodynamics are more complex and less well studied than those of cooling towers. Whether spray ponds or canals are preferred over cooling towers is thus very site specific as to land costs and other factors, and each case must be evaluated on an individual basis.

Two different types of spray systems are in use: (1) the traditional fixed-pipe system and (2) the modular, or unit-powered, system. The fixed-pipe arrangement consists of an array of nozzles fed by a central pump, and all the spray nozzles operate in parallel. The water to be cooled is sprayed into the air only once. The flow of sprayed water collected in the catch basin is not particularly directed. The modular system is comprised of floating units, each consisting of a pump and spray nozzles, moored in a canal so that the water flowing through the canal will pass in series through the modular units. A given particle of water is thus sprayed into the air many times in a single pass through the canal system. The temperature of the particle will approach the wet-bulb temperature of the air that is ambient to the last module in the series.

There has been considerable study to develop nozzle designs with low pressure drops and efficient spray patterns. Spary nozzles can be classified as pressure nozzles, spinning-disk types, and pneumatic, or gas-atomizing, nozzles. Pressure nozzles are most commonly used in spray ponds. Of this type, the so-called hollow-cone, ramp-bottom nozzle is widely used. It has a large free passageway, no interior vanes, and the right-angled inlet is offset to impart a whirling action that creates a hollow cone of spray water, as indicated in Fig. 3.3. Also, as shown in Fig. 3.3., air is induced downward into the central part of the cone by the relatively high velocity of water in the jet. The nozzles are usually arranged in groups of four. A typical ramp-bottom nozzle delivers 0.0033 m^3/sec (53 gpm) with a pressure drop of 48 kPa (7 psi) and 0.0040 m^3/sec (64 gpm) with a pressure drop of 69 kPa (10 psi). A rule of thumb is that the water will rise about 0.044 m for every kPa of pressure drop; thus the height of the spray cone for a nozzle with a ΔP of 48 kPa (7 psi) installed 1.52 m (5 ft) above the surface would be about 3.7 m (12 ft), and the total diameter of the spray pattern would be about 9.8 m (32 ft).[10] The fixed-pipe system produces relatively small drop sizes, in the order of less than 1 mm in diameter. The smaller drops enhance evaporation but increase the drift rate.

The modular system is now widely favored over the fixed-pipe arrangement for power plant applications. Much of the following discussion about modular units has been taken from Ryan.[11] The flow rate of 0.003 to 0.004 m^3/sec (40 to 70 gpm) per nozzle mentioned above for the fixed-pipe systems contrasts sharply with the 0.16 m^3/sec (2500 gpm) typically delivered from each nozzle of a modular system. Drop sizes are much larger than in the fixed-pipe system, being in the order of 5 to 10 mm in diameter, and thus drift losses also tend to be less. Each module

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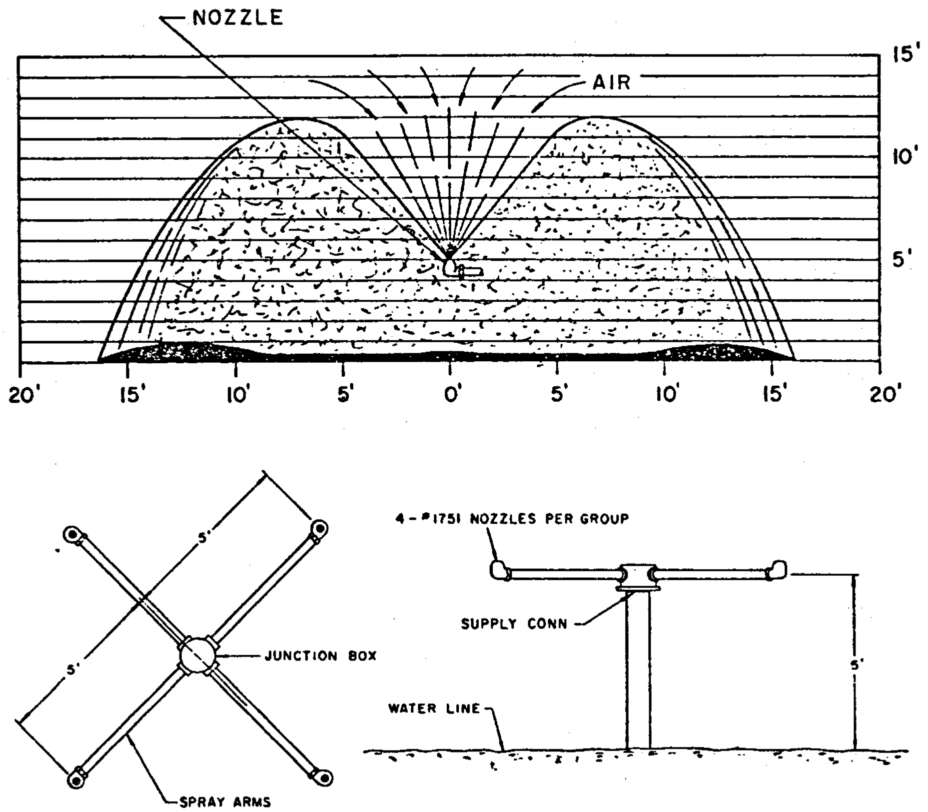

Fig. 3.3. Spray pattern and distribution. Source: E. Hebden and A. M. Shah, "Effects of Nozzle Performance on Spray Ponds," p. 1449 in *Proceedings of the American Power Conference*, vol. 38, Illinois Institute of Technology, Chicago, 1976. Reprinted by permission.

customarily has four nozzles with the spray pattern for each being about 12.2 m (40 ft) in diameter. The total power requirements for a module with four nozzles is about 56 kW (75 hp). The modules are moored in symetrical rows along the width and length of the canal. The performance is better when the wind direction is perpendicular to the canal, because air movement along the length tends to cause interference between the sprays and raise the local wet-bulb temperature. The spray pattern characteristics of unit sprays have been analyzed and reported in the literature.[12]

One approach to evaluating the performance of modular spray systems is to use overall performance information to construct simplified design curves. A set of these curves, developed by the Tennessee Valley Authority

(TVA),[13] is illustrated in Fig. 3.4. The use of the curves is explained in the figure caption.

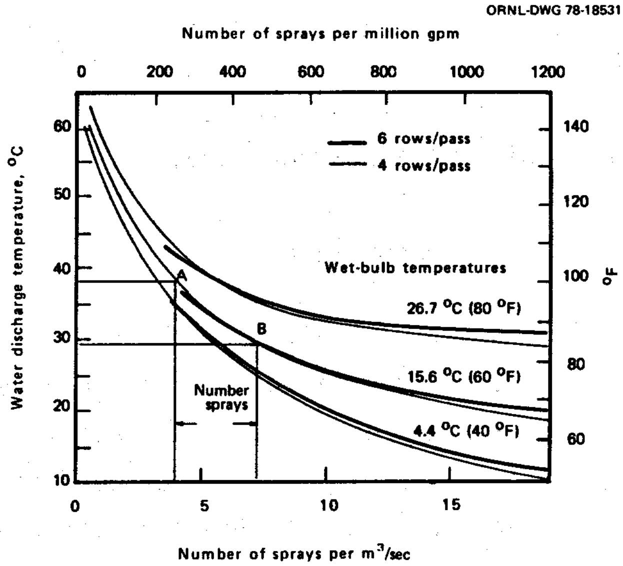

Fig. 3.4. Simplified spray pond design curves developed by TVA for 8 km/hr (5 mph) wind. Use of curves is explained by the following example: For hot water temperature of 38°C (100°F) and wet-bulb temperature of 15.6°C (60°F) [Point A], a flow rate of 31.5 m^3/sec (0.5 x 10^6 gpm), and four rows per pass, the number of sprays per cubic meter for each second is 3.9. For an allowable discharge temperature of 29.4°C (85°F) [Point B], the number of discharge sprays is 7.2 per m^3/s. The number of sprays required is (7.2 - 3.9)31.5 = 104 units. Source: C. Bowman, Tennessee Valley Authority, Knoxville, Tenn., private communication with P. J. Ryan, Bechtel, San Francisco. Reprinted by permission.

A more rigorous approach is to estimate the thermal efficiency of a single module as a function of the water temperature, wet-bulb temperature, and wind speed and then estimate the behavior of the system of modules, including the interference effects of the sprays. A widely used method of estimating the module performance is the so-called NTU Model. The module efficiency at a given wind speed is represented by the number of transfer units (NTU), a dimensionless quantity defined as follows:[11]

$$\overline{\mathrm{NTU}} = \frac{c_w \Delta T}{[h(T_H) + h(T_C)]/2 - h(T_{wba})} ,$$

where

c_w = specific heat of water,

h = total heat (sigma function) of air-vapor mixture of evaluated T,

T_H = nozzle (hot) water temperature,

T_C = sprayed (cold) water temperature,

$\Delta T = T_H - T_C$,

T_{wba} = local wet-bulb temperature.

Hoffman[14] obtained empirical values of the average NTU as a function of wind speed (Fig. 3.5). These values check reasonably well with field data, according to Porter and Chen,[15] who back-calculated from field data for NTU values (Fig. 3.5). The differences between the best-fit data and the summer data is apparent, as is also the difference between these data and the Hoffman values.

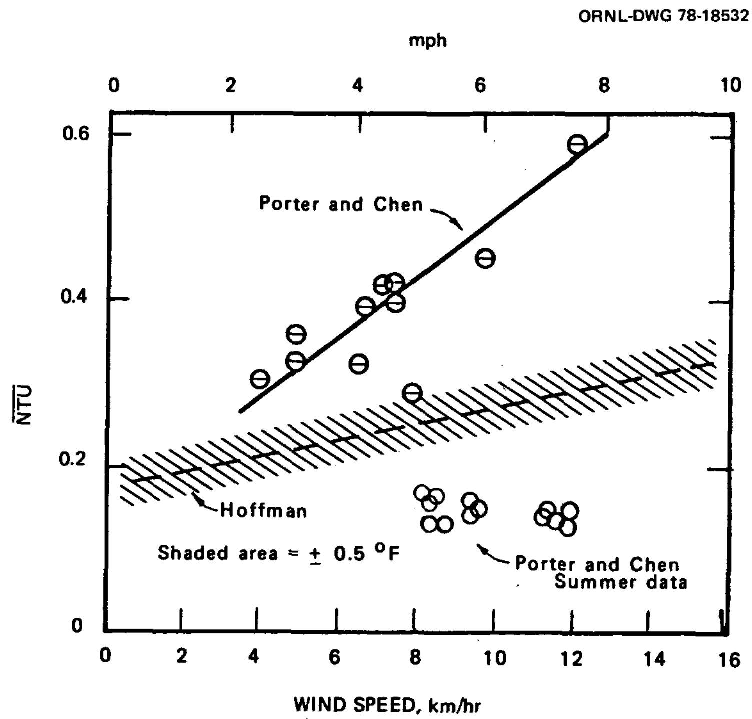

Fig. 3.5. Number of transfer units for spray ponds vs wind speed. Source: D. P. Hoffman, "Spray Cooling for Power Plants," p. 702 in *Proceedings of the American Power Conference*, vol. 35, Illinois Institute of Technology, Chicago, 1973. R. W. Porter and K. H. Chen, *J. Heat Transfer*, 96, 286-91 (August 1974).

Another approach to evaluating module performance is the cellular model, which examines the heat and mass transfer relationships from a single water droplet. Various modifications and improvements have been made of the original Elgawhary model,[16] and good agreement has been obtained with field data using a typical drop diameter of 5 mm. Because a comprehensive discussion of the cellular model cannot be included here, the reader is referred to Porter and Chaturvedi.[17]

The performance of a system of modules is evaluated by taking each row as a separate channel. A fraction of the flow in the channel is assumed to be pumped through a module, cooled, and mixed with the remaining flow in the channel. The mixed flow then proceeds to the next module. At each module the spray water temperature, the wind speed, and the local wet-bulb temperature are estimated. The temperature of the spray water is obtained from the individual module performance analysis and from an energy balance, as outlined above. The wind speed is obtained from local meteorological data or assumed to have a typical value of 8 km/hr (5 mph). The local wet-bulb temperature is an important parameter in estimating the performance of the system. A common assumption is that the wet-bulb temperature increases by 1.1 to 1.7°C (2 to 3°F) at the first row and 0.3 to 0.6°C (0.5 to 1°F) at each subsequent row thereafter. The local wet-bulb temperature, T_{wba}, may also be defined in terms of an interference factor:

$$T_{wba} = T_{wb} + f(T - T_{wb}) ,$$

where T is the water temperature, T_{wb} is the upwind wet-bulb temperature, and f is the interference factor. An average interference factor may be approximated by

$$\bar{f} = \frac{1}{n} [0.18 + 0.26(n - 2)] ,$$

where n is the number of rows of modules.[11] More rigorous methods of evaluating interference factors are given in the literature[17] but are beyond the scope of this section.

Present practice is for the plant designer to lay out the overall arrangement of the spray pond or canal but to leave the selection of the number of modules required to the supplier, who thus assumes responsibility for the performance of the system. Because the performance characteristics of spray modules are usually considered proprietary by the vendors, little application data is published. Although some predictive models have been developed, as briefly mentioned above, all of the models involve the use of empirical, or even arbitrary, factors, and considerable caution is advised in using them to design large systems. Examples are given by Ryan[11] in which the number of modules estimated for a given set of conditions using different models and assumptions differed by a factor of 2.5. There are several instances where the spray canal performances at actual generating stations have been underestimated and modifications have been required.

Until relatively recently it was assumed that the large-diameter drops associated with the modular units would quickly fall to the ground if blown outside the confines of the catch basin and that significant drift deposition would not occur further than about 100 m (100 yd) from the basin. However, Feder[18] has published drift data that extends the possible range of important drift deposition. Environmental Systems Corporation reported drift measurements made at a large spray canal installation at wind speeds varying from 8 to 24 km/hr (5 to 15 mph), as shown in Fig. 3.6. The salinity of the sprayed water was 22,000 ppm. The drift deposition rate as far as 3 km (10,000 ft) downwind was found to be twice that of the background (upwind) values, and some vegetation damage was noted as far away as 2 km (6500 ft) from the basin.[19]

3.5 Cooling Towers

3.5.1 Introduction

Because of the decline in use of once-through cooling systems, most large power stations installed recently in the United States rely on evaporative cooling for waste heat rejection. This method consists of evaporation of a small portion (1 to 3%) of the condenser circulating water to cool the remainder of the stream by 8 to 16°C (15 to 30°F).

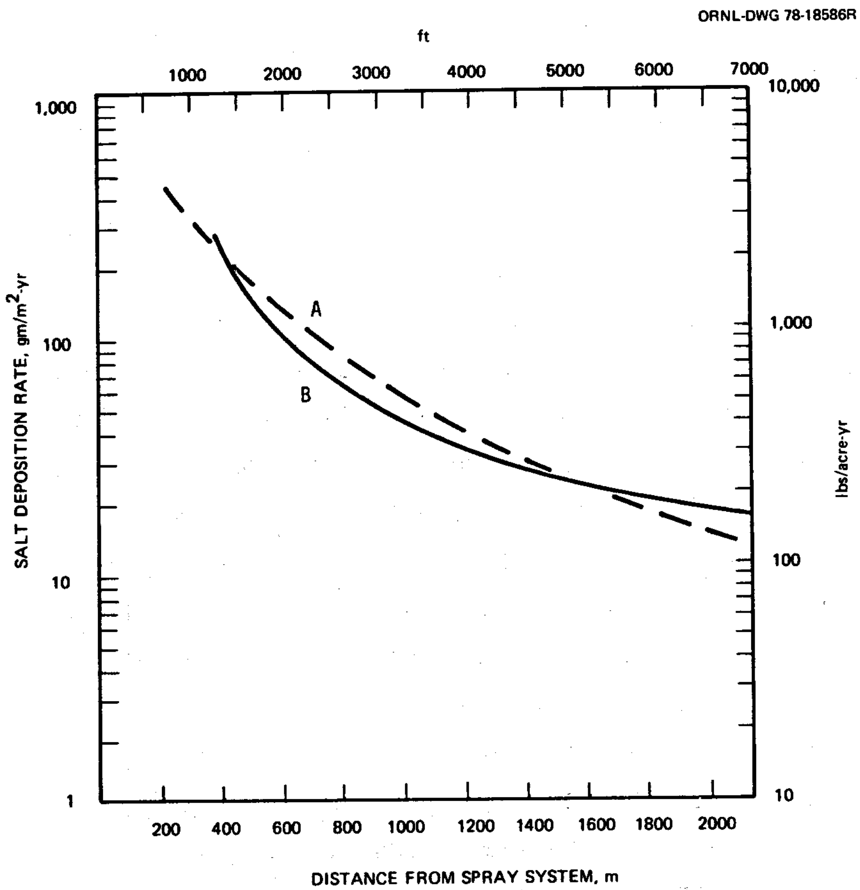

Fig. 3.6. Some measured salt deposition rates from spray canals as function of distance from source. Curve A is based on data from ref. 18 with TDS in canal water of 5000-7000 ppm. Curve B is based on data from ref. 19 with TDS in canal water of about 22,000 ppm.

Although spray ponds and canals have some use, the greater proportion 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 airstream 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 airstream usually leaves the tower very close to saturated condition. Cooling tower designs vary in the manner in which the water surface is presented to the air (droplets vs film), the arrangement of the water and air flows (crossflow vs counterflow), and the manner in which the air flow is created (mechanical draft vs natural draft). Each arrangement, as will be discussed subsequently, has particular advantages and disadvantages.

The performance of natural-draft cooling towers is more sensitive than the mechanical-draft type to local meteorological conditions. Natural-draft towers were first developed extensively in England and portions of Europe. Here the low wet-bulb temperatures and high relative humidities, coupled with maximum station loads that tend to occur in the winter when ambient air temperatures are lowest, provide favorable operating conditions. By contrast, natural-draft towers are ill-suited to the southwestern United States where wet-bulb temperatures tend to be high, relative humidities are low, and the maximum loads occur in the summer due to air-conditioning demands. Because the geothermal power sites currently thought to have the most promise in the United States occur in southern California, natural-draft cooling towers may have little current application for geothermal energy power stations.

A distinctly different form of heat rejection is dry cooling. The water to be cooled, or the steam (or other working fluid) to be condensed, is confined inside extended surface tubes over which the air passes. Because there is no evaporation, the method relies entirely on conduction to raise the dry-bulb temperature of the airstream. This method of cooling has been used for many years by industry and for air-conditioning applications in the arid regions of the southwestern United States. In small sizes, the arrangement is usually referred to as air-cooled coils, but in larger sizes (such as for power stations) it is commonly referred to as dry cooling towers. Dry towers have not been used to any great extent in the United States to date because of their relatively high cost and downgrading effect on plant thermal efficiency, but the growing scarcity and cost of water will undoubtedly change this situation. As has been stated, today one would probably look at dry cooling last — tomorrow one may have to look at it first.[20]

A detailed discussion of cooling tower theory is not included here. In brief, in a wet cooling tower, heat is removed from the circulating water by the transfer of sensible heat due to the temperature difference between it and the air and by the transfer of latent heat equivalent to the mass transfer that results from evaporation of a portion of the water into the air stream. In 1925, Merkel[21] ingeniously combined these two processes into a single equation, which is based on the enthalpy difference as the driving force (the specific heat of the water is assumed to be unity).

$$L\,dt = Ka\,dV(h' - h) = G\,dh\ ,$$

where

L = mass flow rate of the circulating water,
t = temperature of the circulating water,
K = overall unit conductance, based on enthalpy as the driving force,
a = area of water interface,
V = active tower volume per unit area,
h' = enthalpy of moist air at the bulk water temperature,
h = enthalpy of the moist air,
G = mass flow rate of the air.

Among the several simplifying assumptions usually made in applying this equation is that, in making the mass balance, the evaporation loss is ignored. Development of the equation has been summarized in many texts and references; Baker and Shryock[22] made a comprehensive review of the subject in 1961. Because the equation does not lend itself to direct mathematical solution, it is usually solved by some means of mechanical integration that considers the relative motion of the airstreams and water streams in counterflow and crossflow cooling towers. Advent of the computer made it possible to develop comprehensive sets of curves to aid in analyzing tower performance[23] and to mathematically model cooling tower processes. These models, however, depend on coefficients that need verification by field testing; many are considered proprietary. Simplified approximations of the cooling tower process are often sufficiently accurate, and many such methods have been described in the literature. One method, for example, follows Merkel's total heat theory and, embodying Lichnestain's empirical relationships, results in an equation subject to iterative solution.[24] The ASHRAE *Guide and Data Book* is one of the better references.[25]

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. The majority of mechanical-draft cooling towers in the United States have been furnished by Marley, Ecodyne, Zurn Industries, Research-Cottrell, and Westinghouse. Natural-draft towers have been provided primarily by Marley and Research-Cottrell.

3.5.2 Definitions of terms used with cooling towers

The following terms are often employed in discussing cooling towers:

Approach: The difference between the temperature of the cooled water leaving the tower and the ambient wet-bulb temperature.

Blowdown: The water discharged from the cooling tower basin to control concentration of salts and other impurities in the circulating water (Sect. 3.5.8).

Circulation rate: The rate at which the water taken from the cooling tower basin circulates through the condenser system.

Cold-water temperature: The temperature of the cooled water leaving the tower.

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

Cooling range, or range: The difference between the temperature of the water entering and leaving the tower.

Cycles of concentration: A comparision of dissolved solids in makeup water with solids in the circulating water. This value usually varies from about 2 to 20 (Sect. 3.5.8).

Drift, or drift rate: The water lost from the tower as droplets entrained in the effluent airstream. Typical values are 0.001 to 0.2% of the circulating-water rate.

Dual-service condenser: A surface condenser with divided water boxes that allows use of two separate cooling-water circulating systems.

Evaporation rate: The water lost from the tower by evaporation of a portion of the circulating water into the air stream.

Fill, or packing: The internals of a cooling tower, which, either by causing splashing of the falling water or by presenting films (wetted surfaces), provide maximum water-air interface.

Heat load, or duty: The heat removed from the circulating water in the cooling tower. It is usually expressed in megawatts (Btu/hr).

Hot water temperature: The temperature of water entering the tower.

Hyperbolic cooling tower: The shape of many natural-draft cooling tower shells.

L/G ratio: The ratio of the weight of water circulated in a cooling tower to the weight of air passing through the tower.

Makeup: The water added to the cooling tower basin to replace that water lost by evaporation, drift, blowdown, and leakage. The relationship between these water losses and cycles of concentration is discussed in Sect. 3.5.8.

Packaged cooling tower: These towers are small enough to be factory-assembled and require little or no field erection.

Parallel path: Wet/dry cooling towers are sometimes referred to as parallel path towers because the air flow is usually in parallel through the two sections.

Recirculation: The percentage of the exhausted air that reenters the tower.

Total dissolved solids (TDS): Total dissolved solids, usually expressed as ppm, contained in the circulating water.

Water loading: The circulating-water flow rate expressed in cubic meters per square meter per minute (gpm/ft^2) of effective horizontal wetted area of the tower.

Wet/dry cooling towers: Towers provided with both wet and dry sections. The airstreams usually move in parallel paths through the wet and dry portions, then combine before being exhausted by a single fan. The water usually flows in series, first through the dry coil and then through the wet section. This type of tower can control visible plumes and reduce annual water evaporation rates.

3.5.3 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. 3.7. 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 up to 120 m (400 ft) in length in a so-called rectangular layout. (A recent development in wet mechanical-draft cooling towers is the circular configuration discussed in Sect. 3.5.4.) 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

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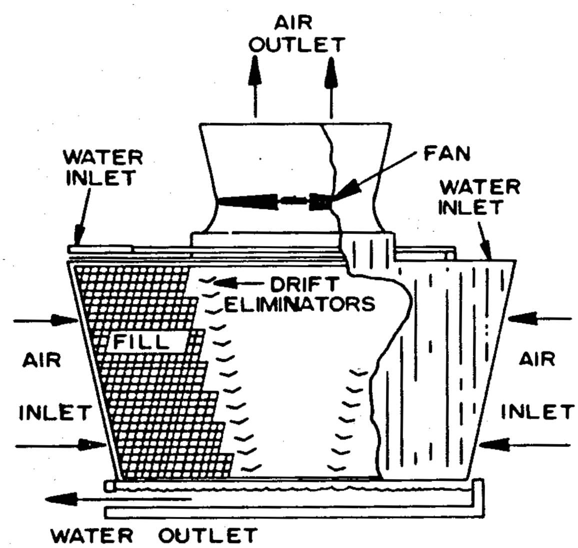

MECHANICAL DRAFT
CROSS-FLOW TOWER

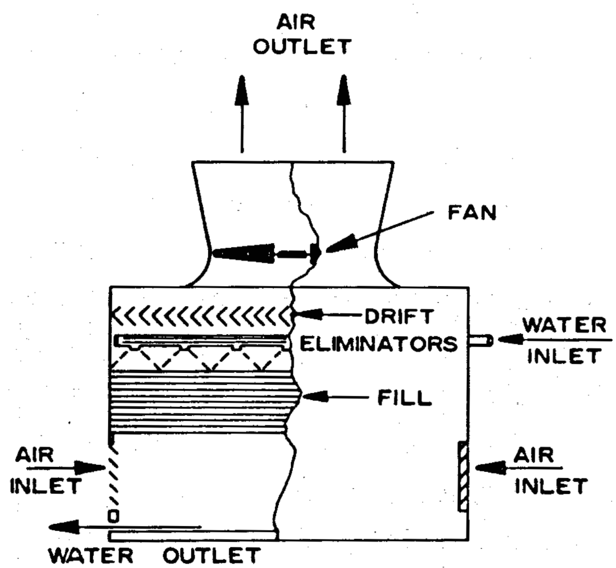

Fig. 3.7. Mechanical-draft cooling towers. <u>Source</u>: J. D. Holmberg and O. L. Kinney, *Drift Technology for Cooling Towers,* The Marley Company, Mission, Kans., 1973. Reprinted by permission.

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.

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. Section 3.5.9 provides a further discussion of drift and methods of measurement.

Wet mechanical-draft cooling towers may be classified as large, intermediate, and small (or package) units. The larger cells, or modules, may be about 15 m (50 ft) high, 21 m (70 ft) wide, and 12 m (40 ft) deep. The top of the fan stack can be as high as 30 m (100 ft) above grade level. These large cells can cool up to 1 m^3/sec (15,000 gpm) of circulating water and have fans up to 8.5 m (28 ft) in diameter driven by 150-kW (200-hp) motors. Intermediate sizes handle flow rates of 0.02 to 0.4 m^3/sec (300 to 6000 gpm) and may have fan diameters of 3 to 5 m (10 to 16 ft) driven by motors of 19- to 45-kW (25- to 60-hp) capacity. Packaged units may vary in capacity from about 1 to 100 liters/sec (15 to 1500 gpm) of circulating water, have fan motors from less than 1 to 19 kW (1.5 to 25 hp), and have a wide variety of designs and materials, including forced-draft counterflow arrangements. These relatively small factory-assembled packaged units have their greatest application in air conditioning.[25]

The circulating-water flow rates required for geothermal power stations will depend on the plant thermal efficiency and the plant electrical generating capacity. If a typical cooling-water temperature range of 11°C (20°F) is assumed, a plant with a thermal efficiency of 5% would require a circulating rate of about 0.4 liters/kW•sec (6.5 gpm/kW). With an efficiency of 10%, the rate would be about 0.2 liters/kW•sec (3.1 gpm/kW), and with 15% efficiency, the rates would be about 0.1 liter/kW•sec (1.9 gpm/kW). The relationships between makeup water addition rates, blowdown, evaporation loss, and drift and the concentration of dissolved solids in the circulation water are discussed in Sect. 3.5.8.

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.[20] 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 (Sect. 5).

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. Section 3.5.9 includes further discussion of drift. 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.[24] 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.

3.5.4 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. 3.8. 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

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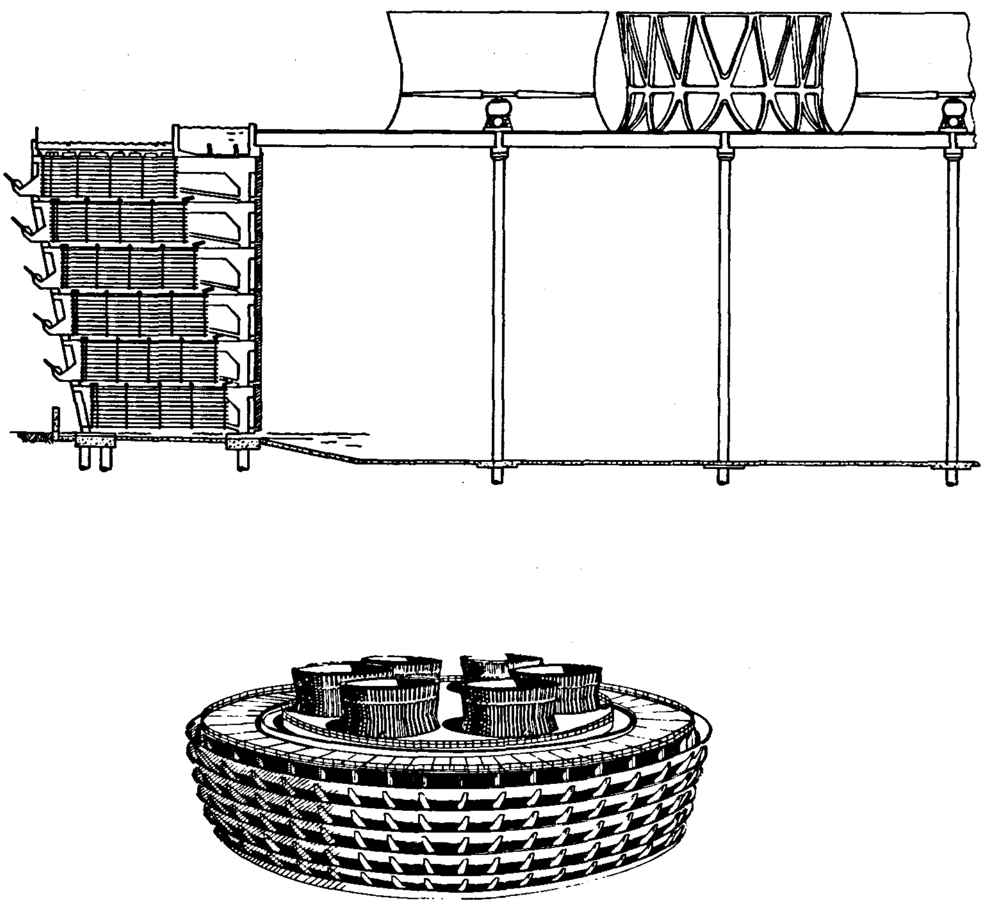

Fig. 3.8. Circular mechanical-draft cooling towers. Source: *Round Towers,* Publication RT-75, The Marley Company, Mission, Kans. Reprinted by permission.

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 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.[26] 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.

3.5.5 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.

The selection of natural-draft rather than mechanical-draft towers for many installations in the United States attests to their advantages in circumstances that permit their use. The favorable conditions may be summarized as

1. ambient conditions of low average wet-bulb temperature and high relative humidities;
2. a combination of low wet-bulb temperature and high inlet and exit water temperatures, that is, a broad cooling range and a large value for the approach temperature;

3. relatively large wintertime loads;
4. long amortization period;
5. relatively large station size;
6. need to restrict ground-level fogging and drift to a minimum; and
7. sites where visual impact, aircraft interference, etc., are not a problem.

Although the primary design parameter for mechanical-draft towers is the wet-bulb temperature, both the wet-bulb temperature and the relative humidity are important to the design of a natural-draft tower.[27]

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. 3.9. Both arrangements are in use, and selection depends upon the operating requirements of a particular site.

The fill may consist of either splash- or film-type packing. In the splash type, usually used in crossflow towers, the water to be cooled falls over wave-shaped slats in such a manner that the droplets are constantly reforming and presenting fresh interfaces for exchange of heat and mass.[28] In the film type, the fill consists of multiple vertical surfaces where the water flows down in thin continuous films. Although the film type occupies less volume and generally requires less shell height, it is more subject to clogging. The splash packing is easier to repair and replace.

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

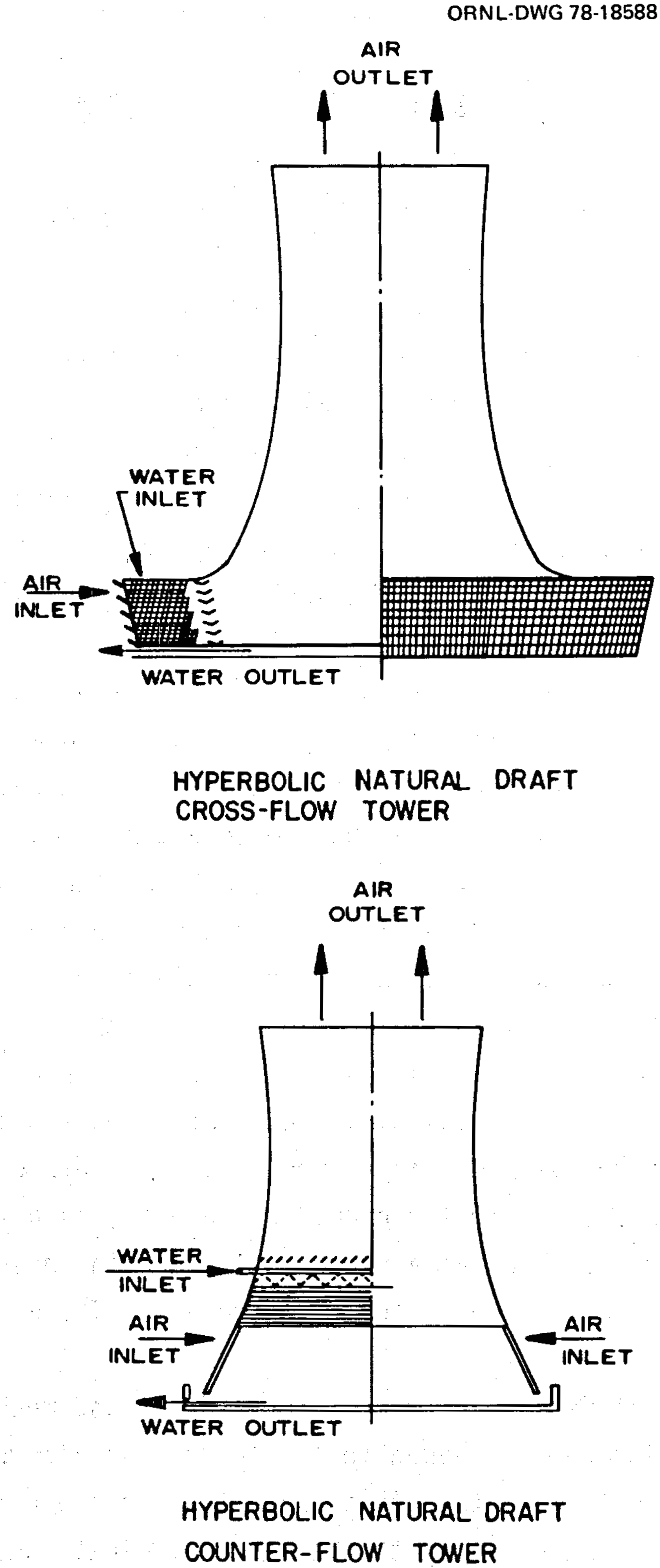

Fig. 3.9. Wet natural-draft cooling towers. <u>Source</u>: J. B. Dickey and R. E. Cates, *Managing Waste Heat with the Water Cooling Tower,* 2d ed., The Marley Company, Mission, Kans., 1970. Reprinted by permission.

of reinforced concrete, of structural steel with aluminum skin, or, as at Schmehausen, Germany, of a suspended cable net with an aluminum skin.[29] 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.[30] 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.[31]

An obvious difference between natural-draft and mechanical-draft cooling towers is that, although in the latter the air movement is somewhat under the control of the plant operators, in the natural-draft type the air flow rates vary with the meteorological and other tower operating conditions. Also, the water pumping costs will probably be less for a mechanical-draft tower. This is, however, more than offset by the auxiliary power savings made by natural-draft towers in not needing fans; the power to operate the fans of a mechanical-draft tower can amount to 0.5 to 1% of the total plant operating costs. Tower maintenance costs will also be appreciably higher for the mechanical-draft type, primarily because of the fans, gear reducers, and motor drives. Because mechanical-draft towers usually require four to five times more land area than natural-draft towers of the same duty, piping and grading costs will be correspondingly greater. Environmental impacts, such as ground-level fogging and icing are significantly less for the natural-draft tower, and there are fewer problems with recirculation. The capital cost of natural-draft towers, however, may be three to four times those of mechanical-draft units for smaller stations. As the size increases, however, there may be a crossover point at which the economics will favor the natural-draft type.

The size and appearance of natural-draft towers may be an important factor in their selection. Even though the economics might not be overwhelmingly in their favor, it is suspected that they are specified in some cases simply because of the distinctive hyperbolic shape. The pros and cons of the appearance of the towers was very well summed up by one writer, "The stark simplicity of the structure, the pleasing symmetry, and graceful upsweep of ribbed concrete — these excite the eye, stir the mind — yet they are offensive to some, who consider them disruptive elements imposed by man on a fast-dwindling rural landscape."[20]

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 in Sect. 3.5.4.

3.5.6 Dry cooling towers

3.5.6.1 General. 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 airstream. Power cycles using dry cooling towers are illustrated in Figs. 1.4 and 1.5.

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[32] in Table 3.1. 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.[33] Dry cooling towers are of two types, direct and indirect.

3.5.6.2 Direct. 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

Table 3.1. Generating stations with dry air-cooled coils rejecting heat to atmosphere

Year	Location	Condensing capacity (kg/sec)[a]	Plant capacity (MW)
	Direct		
1939	Ruhr, Germany	1.5	
1956	Dudelange Steel Works, Luxembourg	13.9	13
1956–57	Municipal Power Station, Rome, Italy	2 x 18.9[b] 23.9	2 x 29[c] 36
1962	Black Hills Power & Light, South Dakota	3	3
1960–61–65	Volkswagen Works, Wolfsburg, Germany	3 x 30.5 36	3 x 40 48
1968	Municipal Incinerator, Beremen, Germany	27.7	
1969	Black Hills Power & Light, South Dakota	19.3	20
1969	Mine-Mouth Station, Utrillas, Spain	87.2/96.8[d]	146/160[e]
1977	Wyodak Station, Gillette, Wyoming		330
1977	(Nuclear) Germany		300
	Indirect		
1965	(Prototype) Hungary	1.4	1.2
1961	Danube Steel Works, Hungary	13.2	13/16
1961	Rugeley, England		120
1964	Quetta Power Station, Pakistan		2 x 7.5
1967	Ibbenbüren, Germany	83.2	
1969	Gyongyos, Hungary		2 x 100
1971	Grootvlet, South Africa		200
1970–71–73	Razdan, U.S.S.R		3 x 220
1976	(Nuclear) Soviet Armenia		2 x 400

[a]kg/sec x 7936.64 = lbm/hr.

[b]Read as 2 units of 18.9 kg/sec condensing capacity each.

[c]Read as 2 units of 29 MW condensing capacity each.

[d]Read as 2 units of 87.2 and 96.8 kg/sec condensing capacity each.

[e]Read as 2 units of 146 and 160 MW condensing capacity each.

Source: E. S. Miliaras, *Power Plants with Air-Cooled Condensing Systems*, MIT Press, Cambridge, 1974. Reprinted by permission.

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,[34] 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.

3.5.6.3 Indirect. 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. 1.4. The condensers may be connected in multipressure arrangements (Sect. 2.4.3).

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[35] 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[35] 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 dry cooling tower and the capital costs are to be kept to a minimum. High back-pressure turbines have been produced in Europe in sizes up to about 200 MW(e) by eliminating the last row of blades of a conventional turbine; this design allows operation at back-pressures up to about 380 mm (15 in.) of mercury absolute. To offset the loss of turbine power, the high-pressure end was also modified to take a greater steam flow rate. In the United States, General Electric markets a high back-pressure turbine. The GE unit provided for the Wyodak plant has last-stage blades only about one-half as long as those in a low back-pressure turbine. Allis-Chalmers has completed designs for a high back-pressure unit and, if the need develops, could start supplying them by 1983. Both General Electric and Allis-Chalmers report that to date, however, there has been relatively little interest in dry cooling requiring high back-pressure units.[34]

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. This latter arrangement, called "deluge cooling" (Sect. 3.5.7.4) has a tendency to foul the outside surfaces with deposits left from the evaporation of the water and to create corrosion problems. In any event, both ways require the consumptive use of water during the hot season when water supplies tend to be restricted.

The performance of dry cooling towers is more sensitive than that of wet towers to wind and thermal inversion effects. Dry towers can be severely affected by high winds, probably because of the altered air flow across heat exchanger surfaces. Even at moderate wind speeds, wind interactions at the tower exit can be important. The effect of rain impinging directly on the heat exchanger surfaces could be expected to improve the performance, but in the case of natural-draft towers, rain falling through the rising air column will cool it and reduce the draft. Moore and Torrence[36] have studied wind effects on dry natural-draft towers at Rugeley, England; Grootvlei, South Africa; and Ibbenburen, West Germany. The performances at Rugeley and Ibbenburen are said to be reduced during heavy rain and fog. On the other hand, forced-draft dry towers at Wolfsburg, Germany, and Pignataro Majori, Italy, are said to have improved performances during rain. Moore[37] has developed performance equations for evaluating the aerodynamic losses in dry cooling towers and discusses various bundle configurations.

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.[38] Another approach is a foam-metal material proposed by ERG, Inc. Plastic tubes or plastic sheets have also been proposed by Roma at Italimpianti and Battelle-Geneva. (The air-side heat transfer coefficient dominates the process, and the conductivity of the wall is not a major concern.) Although the cost analysis is presently inconclusive, these designs may have a potential for being economically attractive. Some other, more revolutionary, designs have been proposed that may or may not have promise. One of these consists of metal disks that rotate through the water to be cooled and the moving airstream to transport the heat from the water to the air. An oil layer on the water would inhibit evaporation, but fouling of the surfaces and emulsification of the oil film are problems yet to be resolved.[33]

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.[33] 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.[38] 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.

A mechanical-draft dry cooling tower will probably cost five times as much as a wet mechanical-draft tower if the turbine back-pressure is about 305 mm (12 in.) of mercury absolute and about ten times as much if the pressure is 127 mm (5 in.) of mercury absolute.[24]

3.5.7 Wet/dry cooling towers

3.5.7.1 General. 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. 3.10. (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.

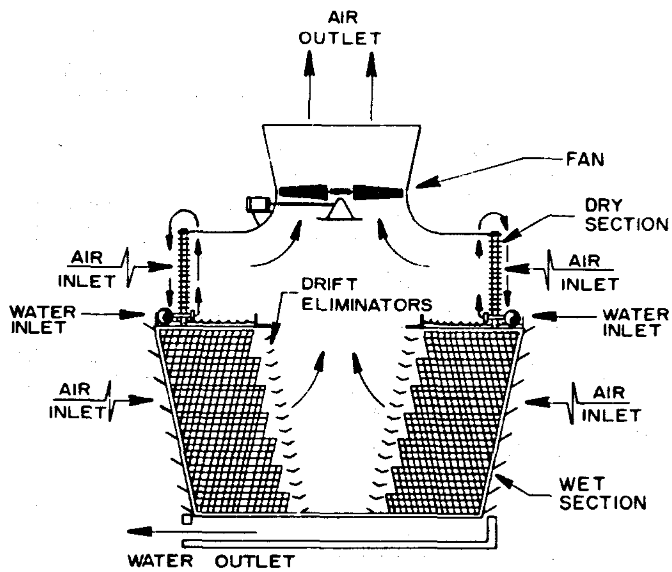

Fig. 3.10. Wet/dry mechanical-draft cooling tower. Source: J. D. Holmberg and O. L. Kinney, *Drift Technology for Cooling Towers*, The Marley Company, Mission, Kans., 1973. Reprinted by permission.

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.

3.5.7.2. Visible plume control. Although the visibility of the plume emitted from the cooling towers is likely to be of little concern in the climate of most geothermal plants, the psychrometric relationships are of interest. The changes that the properties of the air undergo in passing through a single-structure wet/dry cooling tower (Fig. 3.10) are illustrated in Fig. 3.11. The conditions depicted are most likely to

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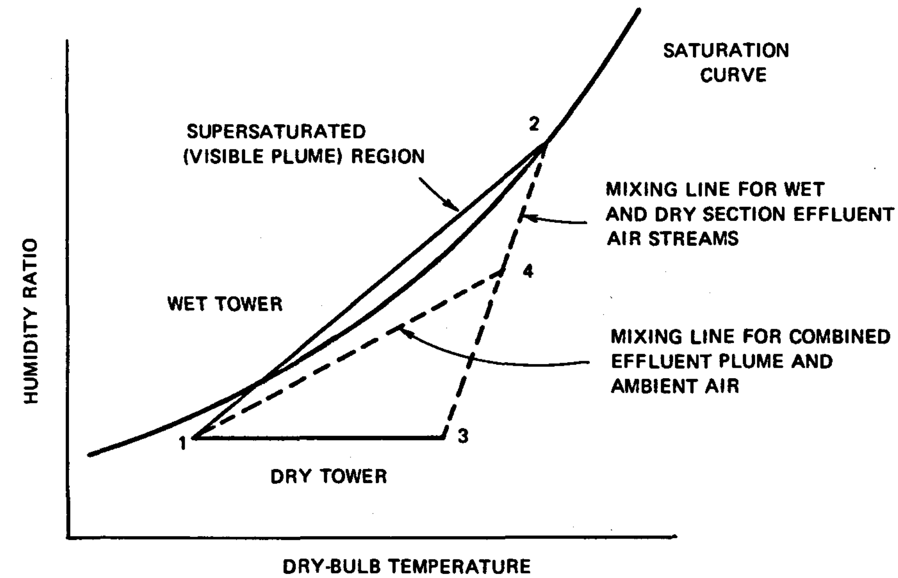

Fig. 3.11. Psychrometric relationships for visible plume abatement during wintertime operation of a single-structure wet/dry cooling tower.

occur during wintertime operation at low dry-bulb temperatures and high relative humidities. If a wet cooling tower alone were used in these circumstances, with ambient air entering at Point 1 in Fig. 3.11 and the air–water vapor mixture leaving the tower essentially staturated at Point 2, the effluent plume would mix with the ambient air along Line 1-2 and pass through the supersaturated region to form a visible vapor cloud, or plume. If a wet/dry tower were used, the evaporative process would perform as just stated, but the dry section would heat the air along Line 1-3, and the effluent from the wet and dry sections would mix along Line 2-3, with the resultant condition at Point 4. When this mixture leaves the tower and mixes with the ambient air along Line 1-4, the state point lies outside the supersaturated region and the vapor is not visible. The proportion of the duty carried by the wet and dry sections can be varied in the design, and the dampers can be adjusted during operation to provide visible plume control. Wet/dry towers have demonstrated the ability to virtually eliminate plume emissions. Landon and Houx[39] are among those studying the plume abatement aspects of wet/dry cooling towers.

3.5.7.3. Flow paths for air and water. 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[40] studied this aspect, and although his work was preliminary and concerned primarily with comparative performances and evaporation rates, the principles involved are of significance. The simplifying assumption was made that the heat rejection capacity of a cooling tower can be expressed as a function of the ITD as follows:

$$Q = K(t_{\mathrm{ITD}})^n ,$$

where K is a constant, and the exponent, n, is a function of the type of tower and the wet-bulb temperature. Loscutoff[40] used $n = 1$ for mechanical-draft dry towers, $n = 1.3$ for natural-draft dry towers, $n = 1.4$ for mechanical-draft wet towers, and $n = 1.8$ for natural-draft wet towers [at a wet-bulb temperature of 18.3°C (65°F)]. The ITD is taken as the difference between the entering water temperature and the ambient dry-bulb temperature in the case of the dry cooling towers and as the difference between the entering water temperature and the ambient wet-bulb temperature for the wet cooling towers. By assuming K to be constant for a given tower type, the performance of the various arrangements as they affect the ITD was compared. Loscutoff[40] 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.

3.5.7.4 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,[41] 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.

Kreid, Johnson, and Faletti[41] have made a comprehensive study of a method for approximating the heat transfer relationships from a wetted-finned heat exchanger, using the heat transfer performance of dry surfaces as a base. The analysis shows that the equations describing the combined latent and sensible heat transfer from a wet surface may be transformed to an approximate equation that involves the product of the heat transfer coefficient and a driving potential. For a wet surface, this potential is the moist-air enthalpy difference between the surface and the coolant stream.

A heat exchanger with a low ITD has more to gain from deluge cooling than one with a higher ITD. There is substantial enhancement of the heat transfer even if the entering air is saturated, because the surface saturation temperature will be higher than the air temperature. The rate of water consumption will be greater than if the same amount of cooling were achieved in a conventional wet cooling tower, but the evaporation rate will not be as great as for the adiabatic cooling method described below.

Wiles et al., of Pacific Northwest Laboratory[42] 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 developed by the HÖTERV Institute, Budapest, Hungary, and licensed in the United States by Babcock and Wilcox. Although this type of surface is more expensive than spiral-wrapped, finned-tube surfaces, it was judged that the flat surfaces were better adapted to deluge wetting. Of the various parameters assumed in the study, perhaps one of the more important is the assumption that loss in electrical generating capacity would be made up by gas turbine power, costing 24 mills/kWhr. The primary thrust of the study, however, was to evaluate the water consumption aspects. A 1000-MW(e) power plant under the assumed operating conditions would require about 11 x 10^6 m^3/year (9000 acre-ft/year) of makeup water if evaporative cooling towers were used. If no water were available, it is estimated that an all-dry cooling system would add about 2.4 mills/kWhr to the production cost; if about 3 x 10^6 m^3/year (2500 acre-ft/year) of water were available annually for consumptive use in deluge cooling, the incremental cost would be about 1.5 mills/kWhr (Fig. 3.12). 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.

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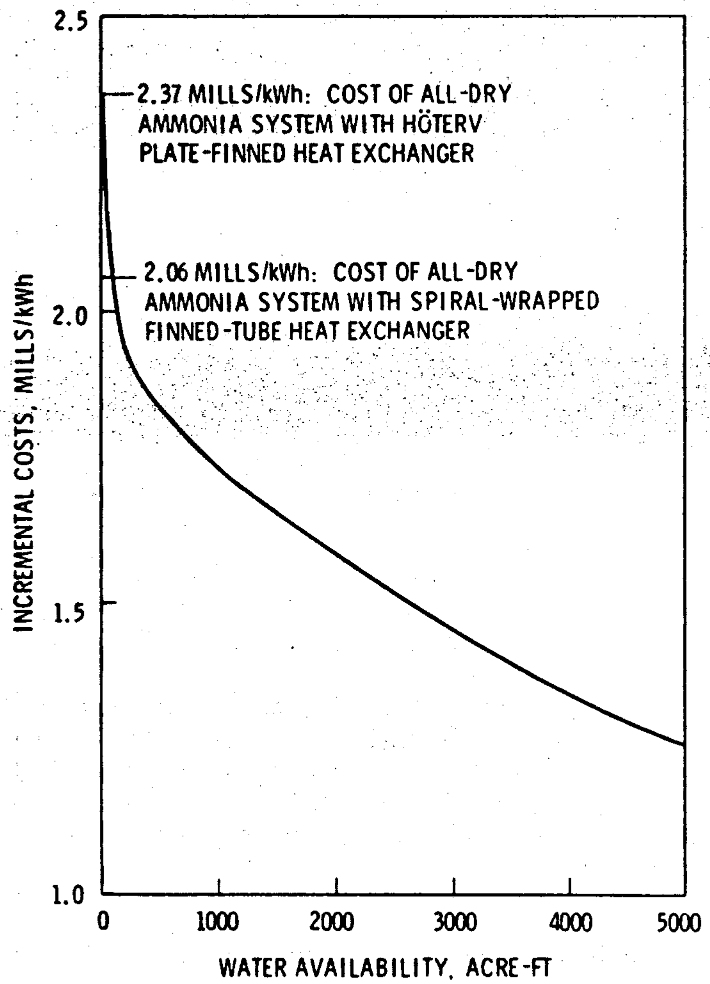

Fig. 3.12. Incremental cost of electricity attributable to a deluged dry cooling coil system as a function of the annual water availability to a 1000-MW(e) fossil-fueled power plant. (m^3/year = acre-ft/year x 1233.5). Source: L. E. Wiles et al., *A Description and Cost Analysis of a Deluge Dry/Wet System*, PNL-2498, Pacific Northwest Laboratory (June 1978). Reprinted by permission.

3.5.7.5. Evaporative, or adiabatic, cooling. One method of increasing the capacity of a dry cooling tower during peak loads or periods of high dry-bulb temperature would be to increase the temperature difference for heat transfer by cooling the air before it passes over the coils. This method could be achieved by evaporating water into the air in an adiabatic process that is commonly used in dry climates where the wet-bulb temperature is substantially below the dry-bulb temperature. (The heat exchanger coils, however, would not operate with wet surfaces as in the deluge system mentioned above.) The adiabatic

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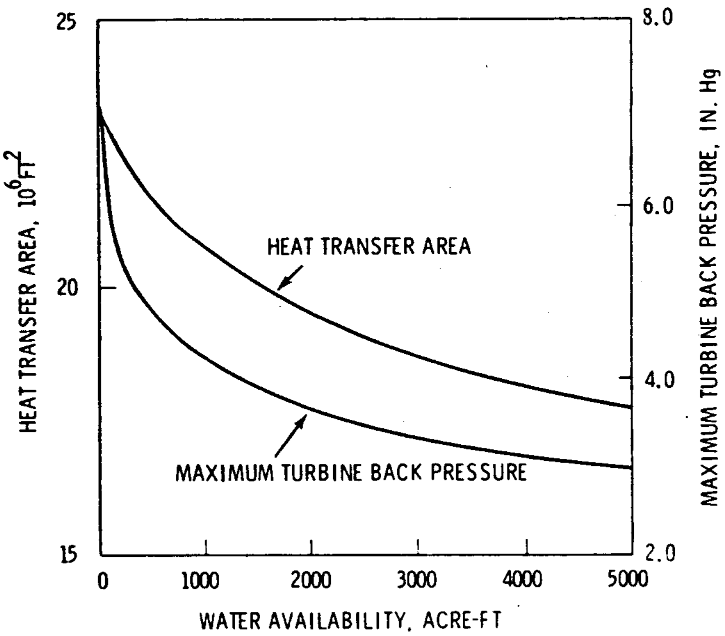

Fig. 3.13. Optimized heat transfer area and maximum turbine back pressure as a function of the annual water availability to a 1000-MW(e) fossil-fueled power plant. (m^3/year = acre-ft/year x 1233.5). Source: L. E. Wiles et al., *A Description and Cost Analysis of a Deluge Dry/Wet System*, PNL-2498, Pacific Northwest Laboratory (June 1978). Reprinted by permission.

cooling process is enhanced by decreasing the size of the water droplets sprayed into the air, but there is a tradeoff with the amount of power required for atomization. The quantity of water evaporated in the adiabatic cooling of air is significantly greater (perhaps about 50%) than the amount of water that would be evaporated to handle the incremental peak load in a conventional wet cooling tower. As mentioned in Sect. 3.5.7.4, the amount of water evaporated is also greater than that required for deluge cooling, but, if operated carefully, there will not be the same tendency for corrosion and scaling as in the intermittently wetted surfaces of the deluge system. Adiabatic cooling for an all-dry cooling system, handling the entire system heat rejection duty, may enable the plant to meet the peak loads at high ambient temperature conditions without the high cost of a wet/dry system.

3.5.7.6. Separate or single structures. The wet and dry sections of a wet/dry cooling tower can either be housed in a single structure

(Fig. 3.10) where there is a common fan or be kept separate as more or less conventional wet and dry cooling towers. Smith and Larinoff[43] have summarized some of the considerations involved in choosing between single and separate structures for wet/dry cooling towers:

1. Air recirculated into the dry section from the wet will contain moisture and chemicals used for water treatment that will accelerate corrosion of the extended-surface heat exchangers. A single-structure arrangement is more susceptible to recirculation problems because, with separate structures, the towers can be located to minimize the effect.
2. The single-structure configuration relies on louvers and damper doors to regulate the flow of air between the wet and dry sections. The dampers are relatively large and must operate in moist and corrosive conditions. Separate structures do not require dampers but lack the "fine tuning" aspects of the damper system.
3. Separate structures with separate fans provide more latitude in the design and in selecting operating modes.
4. Freeze protection is more difficult with single structures.
5. Plume abatement is more effective with a single structure because the effluent of the wet and dry sections can be mixed in the optimum proportions.
6. Separate structures will generally have a higher cost, although this difference tends to decrease in the larger sizes.

Smith and Larinoff[43] concluded that the various possible configurations did not have a significant influence on condensing steam temperatures nor on the annual water evaporation rates.

3.5.8 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

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

The makeup water rates required for power stations with various thermal efficiencies and concentration factors are shown in Table 3.2.

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

Table 3.2. Makeup water requirements vs cycle efficiency and concentration factor

Plant electrical capacity [MW(e)]	Cycle thermal efficiency (%)	Concentration factor	Makeup water rate	
			m^3/sec	gpm
1	5	2	0.0156	247
	10	2	0.0074	117
	15	2	0.0046	74
	20	2	0.0033	52
	5	4	0.0104	165
	10	4	0.0049	78
	15	4	0.0031	49
	20	4	0.0022	35
	5	6	0.0093	148
	10	6	0.0044	70
	15	6	0.0028	44
	20	6	0.0020	31
5	5	2	0.0779	1234
	10	2	0.0369	585
	15	2	0.0232	368
	20	2	0.0164	260
	5	4	0.0519	823
	10	4	0.0246	390
	15	4	0.0155	245
	20	4	0.0109	173
	5	6	0.0467	740
	10	6	0.0221	351
	15	6	0.0139	221
	20	6	0.0098	156
50	5	2	0.7785	123340
	10	2	0.3688	5845
	15	2	0.2322	3680
	20	2	0.1639	2598
	5	4	0.5190	8227
	10	4	0.2458	3897
	15	4	0.1546	2454
	20	4	0.1093	1732
	5	6	0.4671	7404
	10	6	0.2213	3507
	15	6	0.1393	2208
	20	6	0.0983	1559

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.

It can be assumed with very little error that the concentration of salts in the drift droplets is the same as that in the tower circulating water. Even though the droplets travel through the tower in an air stream saturated with moisture, the transient time is small (less than 1 min even in tall towers), and there is little opportunity for the droplets to grow in size.

The size distribution of the water droplets in drift varies with the tower design and operating conditions but typically may be about as follows for a mechanical-draft cooling tower:[44]

Average diameter (microns, μ)	Weight fraction
50	0.35
100	0.44
150	0.14
200	0.06
280	0.006
450	0.004

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·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.

The power plant designer can influence the average drift deposition rate in a particular area by consideration of the (a) prevailing direction of the wind, (b) quality of the makeup water and cycles of concentration, (c) tower height and exit velocity, (d) drift droplet size distribution, and (e) drift eliminator performance and other tower design features.

Until a few years ago economics determined such things as tower location and design; however, today the environmental impacts of drift are a major influence. Prior to making design decisions, the drift deposition rates of various cooling tower layouts, designs, and operating parameters are now calculated using either onsite or nearby meteorological data taken over a period of several years. Where salt-sensitive crops are grown, it may be that a seasonal sensitivity can be correlated with seasonal variations in the prevailing wind direction. Mechanical-draft towers cause more severe drift deposition problems than do natural-draft towers because the latter lofts the drift higher and disperses it over a wider area. Mechanical-draft towers that take advantage of the greater buoyancy of multiple plumes, such as the circular mechanical-draft type, also reduce the ground-level drift deposition rates.

A water droplet carried in an air stream has horizontal and vertical components of velocity. The magnitude of the vertical component is the resultant of the upward velocity due to being lifted by the air stream and the gravity-induced fall of the droplet. The flight path of an airborne drop is also affected by drag and momentum, the former being a function of the diameter squared and the latter a function of the diameter cubed. It is thus evident that the smaller droplets are more likely to remain airborne, and, after exiting from the tower, will have longer trajectories. It is also clear why a mechanical-draft cooling tower with greater air velocities at the exit will sweep larger drops from the tower than will natural-draft structures.

Drift eliminators for cooling towers use baffles to change the direction of the air flow so that entrained water droplets will impinge on and be trapped by the baffle surface. Care is taken to drain the eliminators so that water will not be reentrained. In addition to the direction change, the eliminator baffles may be followed downstream by a honeycomb-type "polisher." The honeycomb surfaces strip additional droplets from the air stream and also give more opportunity for gravity settling of the droplets. The best location for drift eliminators in a cooling tower is away from the water splash in the fill and where the air flow through the plane of the eliminators will be uniform. The general location of the eliminators is between the fill and the fan plenum, as indicated in Figs. 3.7 and 3.10. The eliminator baffles are commonly designed in either a herringbone or sinusoidal wave pattern and may be fabricated of asbestos-cement board.[28] Yao and Schrock[45] are among those studying improved designs for drift eliminator baffles. As the efficiency of the drift eliminators is increased, the capital costs and the fan power requirements also tend to increase. The compromise of these factors is often weighted in favor of lower drift rates because of the current sensitivity to environmental impacts.

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.

In measuring drift, values for both the amount leaving the tower and the droplet size distribution are needed for input into the predictive models used to calculate drift deposition. It is difficult to measure the drift rate in a cooling tower because the quantity of drift is very small compared to the other forms of water present.[45] One method used to estimate the amount of drift is to make a chloride balance on the tower. A more recent method is termed "isokinetic sampling." Several arrangements of this sampling method have been tried, but probably the most effective is to draw samples from points in a grid pattern in the effluent stream and trap the dry solids after evaporation of all the moisture in the sample. These are then washed from the sampling tube and the solution analyzed for the amount of a tracer, such as magnesium, existing in known amounts in the circulating water. Size distributions of the droplets can be investigated using sensitized paper and counting the number of water spots of various sizes after exposure to the air stream. A method now under study and development is an electro-optical system based on the light-scattering of laser beams. This method, termed "Particle Instrumentation by Laser Light Scattering" (PILLS),[46] has some limitations due to background fog but has the advantage of also indicating droplet size distributions.

Webb, Schrecker, and Guild[47] have described application of the sensitive paper, isokinetic sampling, and PILLS methods to the Potomac Electric Power Company's (PEPCO) natural-draft tower at Chalk Point, Maryland, in an attempt to validate various salt drift transport and deposition models. This work was jointly sponsored by the State of Maryland, the Electric Power Research Institute (EPRI), the Energy Research and Development Administration (ERDA), and PEPCO. Concern over the salt effects on the local tobacco crops was a motivating factor.

Laskowski,[48] Roffman and Grimble,[49] and Israel and Overcamp[50] are among those having developed mathematical models for predicting drift deposition. The Oak Ridge National Laboratory (ORNL) developed a fog and drift model for assessing environmental reports for nuclear power stations.[51,52] The necessity for preparing these reports has led to use of consultants by the electric power industry to perform the various required studies and analyses. Some of these consultants have also developed models for predicting drift transport and deposition but consider them to be proprietary. In 1976 ORNL offered the developers of drift deposition models the opportunity to compare the results if each used his model to analyze the same set of conditions. Nine parties responded, and the results have been described by Chen.[53] In general, there was satisfactory correspondence. Overcamp[54] has also made a sensitivity study and analysis of cooling tower drift deposition models. A test was made by Jallouk, Kidd, and Shapiro[55] in 1974 at cooling towers located in Oak Ridge which use chromium and zinc in the circulating water for corrosion control. The uptake of these elements was measured in the surrounding vegetation to obtain measurement of drift transport and deposition. Taylor, Gray, and Parr[56] found good correspondence with the tests and the predictions made using the ORFAD model.

To date, there are no reliable mathematical models for predicting the amount of ice buildup at various points from a cooling tower. The icing potential at a given installation may be surveyed, however, by noting from the predictive models the amount of wintertime drift deposition in critical areas, such as a nearby highway, and by equating this to the number of hours the temperature falls below freezing. Significant icing normally occurs only close to the base of mechanical-draft towers and is usually not a problem at a distance of a few hundred meters (yards). Few icing problems have been associated with natural-draft cooling towers.

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4. 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[1] 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.

A 50-MW(e) geothermal power station with an overall thermal efficiency of 10% and an 11°C (20°F) rise in the circulating-water temperature might require a cooled-water storage pond of about 5 ha (11 acres) if the water depth averaged 6 m (20 ft). This amount of water would be for 8 hr of storage. The optimum storage time would have to be determined by analysis but could well be less than 8 hr. A warm-water storage pond of about equal size would also be required. The construction cost of two 5-ha (11 acre) ponds would vary widely but might be in the order of 0.5 million. Although site-specific economic studies will be needed to evaluate the potential of the phased-cooling concept, the preliminary indication is that the arrangement has promise for some locations.

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5. 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[1] made a study of variable power output for a 50-MW(e) binary geothermal power station concept for Heber, California. (The 50-MW(e) conceptual designs for Heber discussed in Sect. 6 are for a flashed-steam rather than a binary system). 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[2] 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. 5.1 and 5.2. In comparison to the 50 MW(e) of net power produced by the base-loaded plant

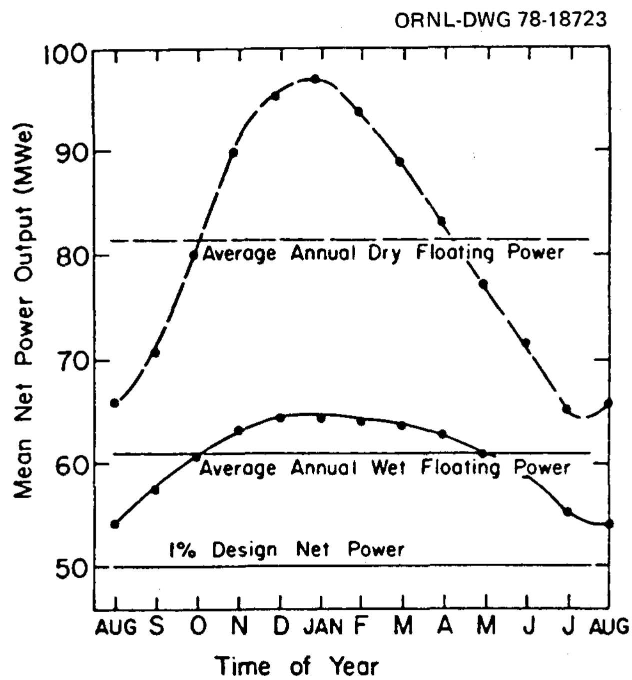

Fig. 5.1. Seasonal variation of net output of "floating" evaporative and dry-cooled binary geothermal power plants at Heber, California, which would generate 50 MW(e) (net) if designed on 1% basis. Source: H. S. Pines et al., *Floating Dry Cooling, A Competitive Alternative to Evaporative Cooling in a Binary Cycle Geothermal Power Plant*, LBL-7087, Lawrence Berkeley Laboratory (July 1978), paper to be 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.

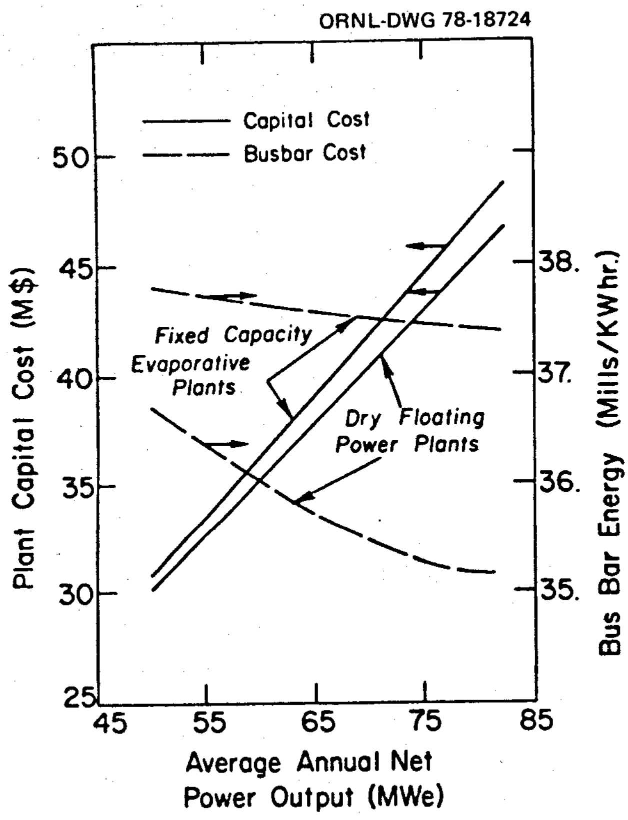

Fig. 5.2. Plant capital costs and busbar energy costs of base-loaded evaporative-cooled and "floating" dry-cooled plants as a function of the annual average power output. Source: V. W. Roberts, *Geothermal Energy Conversion and Economic Case Studies*, EPRI-301, Holt/Procon (November 1976). Reprinted by permission.

REFERENCES FOR SECTION 5

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2. V. W. Roberts, *Geothermal Energy Conversion and Economic Case Studies,* EPRI-301, Holt/Procon (November 1976).

6. HEAT REJECTION EQUIPMENT IN 50-MW(e)-STATION CONCEPTUAL DESIGNS AT HEBER AND NILAND, CALIFORNIA

6.1 General

Study of existing geothermal power plants has limited usefulness because the economic and environmental aspects of power plant design have changed markedly in the past few years. Perhaps of more interest are the recent conceptual designs made by Bechtel[1] of two different types of power cycles for the proposed geothermal power stations at Heber and Niland in the Imperial Valley of southern California. Also worth noting is a United Engineers[2] study of a wet/dry cooling tower for the proposed Heber station. The studies were based on a plant of 50 MW(e) net capacity; stations of greater capacity would be comprised of a multiplicity of the 50-MW(e) units.

The geothermal fluids at Heber and Niland are quite different. At Heber the design temperature of the fluid at the plant boundary is 170°C (338°F), and at Niland it is 228°C (443°F). At Heber the quantity of total dissolved solids (TDS) in the fluid is about 14,000 ppm, but at Niland it is 250,000 ppm. (The noncondensable gas content is about the same at 0.3 to 0.5% by weight). A two-stage flashed-steam system was selected for Heber, and a multistage binary system using isopentane as the working fluid was proposed for Niland. The salient features of the heat rejection systems of the two stations are compared in Table 6.1. The auxiliary power requirements are listed in Table 6.2, and the estimated costs of the heat rejection equipment and total plant, in last-quarter 1976 dollars, are given in Table 6.3.

6.2 Heber

The Heber concept includes two different design approaches to accommodate the expected droop in the brine temperature during the lifetime of the plant. The first considers a constant brine flow rate with drooping power output and the second, increased brine flow rate and constant power output. Although power stations are traditionally designed for a constant

Table 6.1. Salient features of the heat rejection systems and other aspects of the conceptual designs of the Heber and Niland geothermal power stations[a]

	Heber[b]	Niland[c]
Net electrical capacity, MW(e)	50	50
Geothermal fluid at plant boundary		
Temperature, °C (°F)	170 (338)	228 (443)
TDS, ppm	14,000	250,000
Noncondensable gases, wt %	0.3	0.5
Brine flow rate, lb/hr	12×10^6	3.7×10^6
Type of power cycle	Two-stage flashed steam	Multistage flashed/binary
Turbine		
Working fluid	Steam	Isopentane
Throttle pressure, psia	41.8/18.0	550
Exhaust pressure, psia	1.96	28.8
Turbine internal efficiency, %	77/79	85
Condenser type	Direct-contact	Shell-and-tube
Heat sink	Wet mechanical-draft towers	Wet mechanical-draft towers
Heat rejected in towers, Btu/hr	1.58×10^9	8.56×10^8
Cooling water flow rate, gpm	104,000	74,400
Water temperatures, in/out (°F)	116/87	110/87
Makeup water required, gpm	3380	3060
Makeup water source	Condensate	Alamo River
Reinjection water source	New River	Tower blowdown
River water consumption, acre-ft/year[d]	3972	3397
Special equipment	H_2S scrubber	H_2S scrubber Blowdown deaerator Isopentane storage
Cycle thermal efficiency, %	9.8	16.5
Specific net energy, Whr/lb brine	4.2	13.6
Estimated total cost, $1000[e]	36,000	39,200

[a]

To obtain Column I	Multiply Column II	by Column III
kg/sec	lbm/hr	125.998×10^{-6}
kPa	psia	6.89476
MW(t)	Btu/hr	293.071×10^{-9}
m^3/sec	gpm	6.309×10^{-5}
m^3/sec	acre-ft/year	3.9113×10^{-5}
Whr/kg	Whr/lbm	2.2046

[b]Assumes constant power output.

[c]Assumes regeneration.

[d]Based on 0.85 plant factor.

[e]In last-quarter 1976 dollars.

Source: *Advanced Design and Economic Considerations for Commerical Geothermal Power Plants at Heber and Niland, California, Final Report,* SAN-1124-2, Bechtel Corporation (October 1977).

Table 6.2. Auxiliary power requirements for the 50-MW(e) (net) Heber and Niland geothermal power stations

	Power requirements (kW)	
	Heber[a]	Niland[b]
Brine reinjection pumps	3870	1190
Circulating water pumps	2930	2130
Ejector condenser pumps	40	
Condenser vacuum holding pumps	85	
Low-pressure feed pumps		400
Makeup water pumps		90
Blowdown reinjection pumps		160
Scrubber water supply pumps	15	10
Cooling tower fans	1190	840
Other services, estimated	300	230
Total auxiliary power	8430	5050

[a]Heber concept with constant power output.

[b]Niland concept with regeneration.

Source: *Advanced Design and Economic Considerations for Commercial Geothermal Power Plants at Heber and Niland, California, Final Report,* SAN-1124-2, Bechtel Corporation (October 1977).

electrical output over the life of the station, there is no technical reason for not designing for decreasing power capacity if this produces the better economics (see Sect. 5). The study points out that use of the two different design approaches could produce significantly different design concepts for the station. The discussion here will be based primarily on the constant power output approach.

A highly simplified flow diagram of the two-stage flashed-steam system proposed for Heber is shown in Fig. 6.1. Geothermal fluid at downhole conditions of 173°C (344°F) and 855 kPa (124 psia) enters two high-pressure flash vessels at 170°C (338°F) and 793 kPa (115 psia), at a rate of 1512 kg/sec (12.0 x 10^6 lb/hr). A total of 114.4 kg/sec (908 x 10^3 lb/hr) of 310-kPa (45-psia) saturated steam is produced and supplied to the

Table 6.3. Total installed costs of heat rejection equipment and total plant costs at Heber and Niland, California

	Equipment and plant costs[a] ($$10^3$)				
	Heber			Niland	
	Base	Constant output	Constant flow rate	Without regeneration	With regeneration
Regenerator					1,010
Condensers	840	850	880	3,440	2,990
Cooling towers	1,290	1,290	1,415	1,220	1,010
Total plant	35,000	36,000	37,500	39,500	39,200
Total $/kW	700	720	750	790	784

[a]In last-quarter 1976 dollars.

throttle of two 3600-rpm, dual-admission steam turbines that are direct-connected to a common electrical generator having a gross output of 58.4 MW(e). Turbine internal efficiencies are 77% for the high-pressure unit and 79% for the low-pressure unit. The unflashed portion of the geothermal fluid flows to the two low-pressure flash vessels where a total of 75.6 kg/sec (600 x 10^3 lb/hr) of 124-kPa (18-psia) steam is produced and supplied to the low-pressure inlet of the turbine. The remaining 1336 kg/sec (10.6 x 10^6 lb/hr) of unflashed geothermal fluid, along with about 8.8 kg/sec (70 x 10^3 lb/hr) of cooling tower blowdown, is pumped at about 106°C (223°F) and 2172 kPa (315 psia) into the reinjection wells. System output is controlled by varying the rate of brine flow to the flash tanks.

At design conditions, cooling water enters the condenser at 30.6°C (87°F), and, after mixing with the steam condensate, is pumped to the mechanical-draft cooling towers at 46.7°C (116°F) at a rate of 6564 kg/sec (52.1 x 10^6 lb/hr; or 104,117 gpm). The condenser, a low-level, direct-contact, or spray, type located directly beneath the turbine, is constructed of reinforced concrete and is 9.5 m (31 ft) wide x 10.4 m (34 ft) high x 15.9 m (52 ft) long. The inside surface is sealed with an epoxy resin compound. The condenser duty is estimated to be 410.3 MW(t) (1.40 x 10^9 Btu/hr). The condenser was designed on the basis that the temperature

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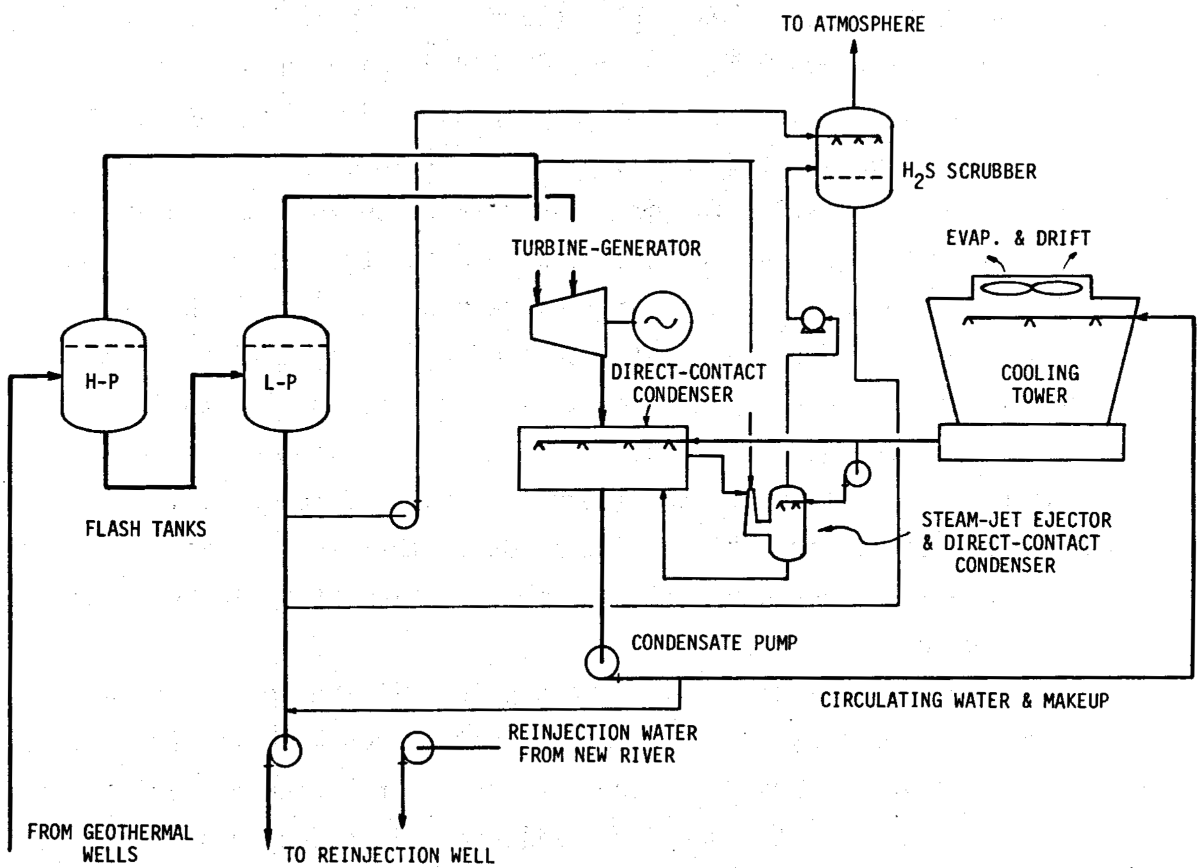

Fig. 6.1. Simplified flow diagram of two-stage flashed-steam power cycle for 50-MW(e) geothermal power station at Niland, California. Source: *Advanced Design and Economic Considerations for Commercial Geothermal Power Plants at Heber and Niland, California, Final Report*, SAN-1124-2 Bechtel Corporation (October 1977).

of the condensing steam would exceed that of the mixture leaving the condenser by about 5.6°C (10°F). Noncondensable gases are removed in two stages; the first is a steam-jet ejector and the second a mechanical vacuum pump. A steam-jet ejector could be used, though, for the second stage during the early life of the plant before the geothermal fluid temperature decreases. The noncondensable gases pass through a hydrogen sulfide scrubber before release to the atmosphere. It is tentatively proposed that the scrubbing medium be concentrated brine taken from the second-stage flash tanks, with small amounts of alkali added if required.

The cooling towers are the evaporative, mechanical, induced-draft type arranged in ten cells, each 11 m (36 ft) wide x 18.3 m (60 ft) high. The design wet-bulb temperature is 26.1°C (79°F), and the average evaporation and drift rate is given as 182.7 kg/sec (1.45 x 10^6 lb/hr, or 2898 gpm). Based on a blowdown rate of 8.8 kg/sec (70 x 10^3 lb/hr, or 140 gpm), the concentration factor is about 21. Because the amount of water available from condensing the steam exceeds the makeup rate required for the towers, the small excess will be reinjected along with the unflashed geothermal fluid. A small amount of hydrogen sulfide may be dissolved in the condensate and carried over to the cooling tower, where there will be a tendency for it to come out of solution and be released to the atmosphere. Although the amount is estimated to be small and within acceptable limits, this aspect needs further investigation, and it is possible that hydrogen sulfide removal equipment would be needed for the condensate.

The amount of geothermal fluid reinjected will be less than that taken from the ground by an amount approximately equal to the evaporation rate in the cooling towers. If necessary to control ground-level subsidence, water taken from the New River will be reinjected in separate wells. It may be necessary to chemically treat this water prior to reinjection to make it compatible with the downhole geothermal fluid.

6.3 Niland

A simplified flow diagram of the Niland binary isopentane multistage flash system is shown in Fig. 6.2. Two cases were studied by Bechtel,[1] one with regenerators in the turbine exhaust and the other a base case without regeneration. The regeneration case, shown in Fig. 6.2, is the system discussed below.

As previously stated, the geothermal fluid temperature at the Niland site is higher than at Heber, being about 288°C (550°F) downhole and about 228°C (443°F) at the plant boundary. The composition of the geothermal fluid, or brine, is close to one-fourth solids, or about 250,000 ppm, by weight. Almost all the brine supplied to the plant will be reinjected. As indicated in Fig. 6.2, the brine enters the separator–heat exchanger as a two-phase mixture of steam and brine. The flashed steam passes through a mist eliminator and condenses on the exterior of the heat exchange

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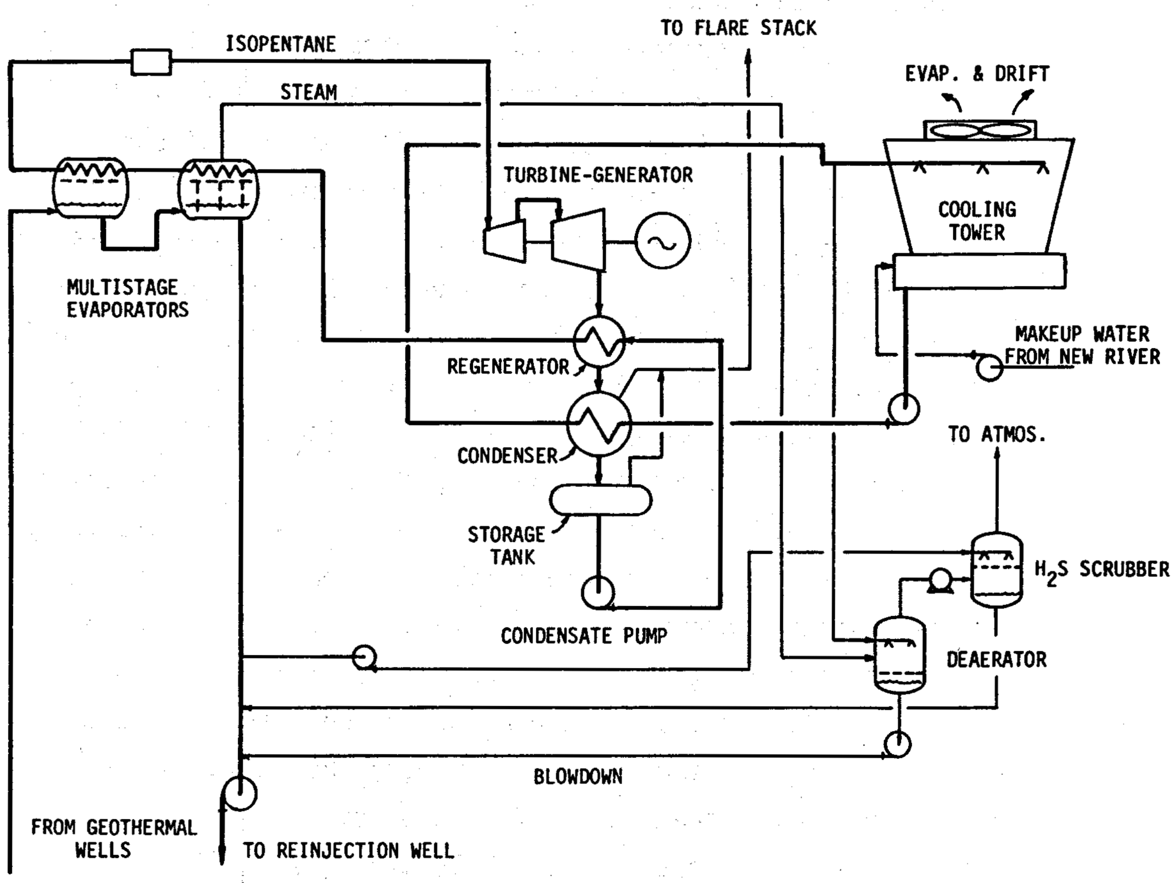

Fig. 6.2. Simplified flow diagram of multisage evaporation binary power cycle for 50-MW(e) geothermal power station at Niland, California. Source: *Advanced Design and Economic Considerations for Commercial Geothermal Power Plants at Heber and Niland, California, Final Report,* SAN-1124-2, Bechtel Corporation (October 1977).

tubes, giving up heat to the isopentane flowing inside the tubes. The condensate falling off the tubes mixes with residual brine and flows to the next lower stage where the process is repeated.

The isopentane circulates at a rate of about 706 kg/sec (5.6 x 10^6 lb/hr). It is pressurized to a supercritical pressure of about 3985 kPa (578 psia) and a temperature of 88.3°C (191°F) before it enters the tubes of the regenerator and multistage flash unit. The superheated isopentane vapor, at 3792 kPa (550 psia) and 199°C (390°F), enters the high-pressure turbine and expands to about 827 kPa (120 psia) and 135°C (275°F) before further expansion in the low-pressure turbines to 226 kPa (32.8 psia) and 106°C (223°F). The turbines are direct-connected to a 3600-rpm electric generator having a gross output of 55.6 MW(e). The turbine exhaust vapor, still in the superheated state, flows to the shell side of the two shell-and-tube regeneration heat exchangers where the temperature is reduced to 54.4°C (130°F) in heating the "boiler" feed from 47.2°C (117°F) to 88.3°C (191°F). The exhaust vapor then enters the shell side of the surface-type condenser. Condensation is at about 48.9°C (120°F) near the entrance but because of a 20.7-kPa (3-psi) pressure drop through the condenser is completed at about 45.6°C (114°F). The two condensers are the horizontal shell-and-tube type, each consisting of a 1.9-cm (0.75-in.) thick shell, 3.7 m (12 ft) in diameter and 18.3 m (60 ft) long, equipped with copper-nickel tubes. The condensation pressure is in the 172-193 kPa (25-28 psia) range, and because it is above a atmospheric, the noncondensable gases can be vented without use of ejectors or vacuum pumps. The gases are vented through a flare stack to burn off combustibles before release to the atmosphere. The isopentane condensate flows by gravity to a storage tank. Cooling water is circulated through the condensers at a rate of 4.7 m^3/sec (74,400 gpm). The water enters at 30.6°C (87°F) and leaves at 43.3°C (110°F), resulting in a cooling duty of 251 MW(t) (857 x 10^6 Btu/hr). The water is circulated to a mechanical-draft evaporative cooling tower that consists of seven cells and has an overall length of 77 m (252 ft) and width of 16 m (53 ft). The design wet-bulb temperature is 26.1°C (79°F) and the approach is 4.4°C (8°F). The evaporation rate is given as about 0.13 m^3/sec (2078 gpm). The concentration factor is about 3, and the blowdown is deaerated before reinjection along with the geothermal fluid. The gases vented from the deaerator are removed by a steam-jet ejector and pass through a hydrogen sulfide scrubber before release to the atmosphere. About 0.19 m^3/sec (3058 gpm) of makeup water for the cooling tower will be obtained from the Alamo River, where two 100%-capacity, vertical, centrifugal pumps, each developing 40 m (132 ft) of head at 3600 rpm, will be installed in an intake structure.

In brief, the effect of adding regeneration to the cycle is to improve the cycle efficiency from about 13.9% to 16.5%, which, for the same 50-MW(e) output, reduces the duty of the heat rejection system from about 290 MW(t) (9.9 x 10^8 Btu/hr) to 252 MW(t) (8.6 x 10^8 Btu/hr) and allows use of seven cooling tower cells rather than eight. The isopentane flow rate through the condenser is increased by use of regeneration from about 338 kg/sec (2.68×10^6 lb/hr) to 355 kg/sec (2.82×10^6 lb/hr). The geothermal fluid flow rate would remain the same, but the reinjection temperature would be raised by use of regeneration from 68°C (154°F) to 109°C (229°F). Although use of regeneration reduces the total plant cost by less than 1% (Table 6.3), the significant aspect is that cooling-water consumption is reduced by 19%.

6.4 Heber with Wet/Dry Cooling Tower

Because of the scarcity of water in the Imperial Valley of southern California, United Engineers[2] made a study of a 50-MW(e) Heber concept using a wet/dry tower rather than a conventional mechanical-draft evaporative type. The study postulated saving various amounts of cooling tower makeup water. In the assumptions used in the study, listed in Table 6.4, notice that a loss-of-capacity penalty charge of $1450/kW was applied. (The estimated cost of the waste heat rejection system using conventional mechanical-draft evaporative cooling towers was estimated at about 25% of the total station cost).

Table 6.4. Parameters used in study of wet/dry cooling towers for 50-MW(e) Heber geothermal plant

Year of pricing	1985
Average plant capacity factor	70%
Annual fixed charge rate	17%
Plant life	30 years
Capacity penalty charge rate	$1450/kW
Fuel cost	$0.952/million Btu
Operation and maintenance cost	9.36 mills/kWhr
Water cost	30¢/10^3 gal

Source: United Engineers, *Engineering and Economic Evaluation of Wet and Wet/Dry Cooling Systems for a Geothermal Power Plant*, Contract EY-76-C-02-2477, unpublished data, September 1977.

Reducing the water consumption to 40% of what it would be using conventional wet towers would make (1) the capital costs of the waste heat rejection system about 1.5 times greater and (2) the total cost, after capacity and operating penalties are applied, about 3 times greater. The total cost to produce electricity, shown in Table 6.5, is about 1.4 times greater than for a plant using conventional wet towers. If the water consumption were to be reduced to 5% of that required for conventional towers, the effects would be striking. The cost of the waste heat rejection system, after penalties are applied, is about 6 times greater, and the total cost to produce electricity is about 1.6 times greater. The study concluded that relatively small geothermal power plants tied into large grids might operate more economically using conventional rather than wet/dry towers and would not be base-loaded except in the winter months. In the summer, during extremely adverse conditions for heat rejection, the capacity would be allowed to drop off or the plant would be shut down completely.

Table 6.5. Effect of varying the amount of water conserved on cost to produce electricity at Heber

Evaporation (percent of wet mechanical-draft tower)	Power cost (mills/kWhr)
100	48.8
50	65.0
40	66.7
30	68.8
20	72.0
10	75.0
5	78.0

Source: United Engineers, *Engineering and Economic Evaluation of Wet and Wet/Dry Cooling Systems for a Geothermal Power Plant*, Contract EY-76-C-02-2477, unpublished data, September 1977.

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