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An Introduction to Condensers and Auxiliary Equipment for Steam Power Plants

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J. Paul Guyer, P.E., R.A.

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1. STEAM CONDENSERS

1.1 CONDENSER TYPES

1.1.1 SPRAY TYPE. Spray condensers utilize mixing or direct contact of cooling water and steam. Cooling water is distributed inside the condenser in the form of a fine spray that contacts and condenses the steam. This type has application where dry cooling towers are used. Part of the condensate from the condenser is circulated through dry cooling towers and returned to and sprayed into the condenser. The balance of the condensate, which is equal to the steam condensed, is pumped separately and returned to the feedwater cycle.

1.1.2 SURFACE TYPE. Surface condensers are basically a shell and tube heat exchanger consisting of water boxes for directing the flow of cooling water to and from horizontal tubes. The tubes are sealed into fixed tube sheets at each end and are supported at intermediate points along the length of the tubes by tube support plates. Numerous tubes present a relatively large heat transfer and condensing surface to the steam. During operation at a very high vacuum, only a few pounds of steam are contained in the steam space and in contact with the large and relatively cold condensing surface at any one instant. As a result, the steam condenses in a fraction of a second and reduces in volume ratio of about 30,000:1.

1.1.2.1 PASS CONFIGURATION. Condensers may have up to four passes; one and two pass condensers are the most common. In a single pass condenser, the cooling water makes one passage from end to end, through the tubes. Single pass condensers have an inlet water box on one end and an outlet water box on the other end. Two pass condensers have the cooling water inlet and outlet on the same water box at one end of the condenser, with a return water box at the other end.

1.1.2.2 DIVIDED WATER BOX. Water boxes may be divided by a vertical partition and provided with two separate water box doors or covers. This arrangement requires two separate cooling water inlets or outlets or both to permit opening the water boxes on one side of the condenser for tube cleaning while the other side of the condenser remains in operation. Operation of the turbine with only half the condenser in service is limited to 50 percent to 65 percent load depending on quantity of cooling water flowing through the operating side of the condenser.

1.1.2.3 REHEATING HOTWELL. The hotwell of a condenser is that portion of the condenser bottom or appendage that receives and contains a certain amount of condensate resulting from steam condensation. Unless the condenser is provided with a reheating hotwell (also commonly called a deaerating hotwell), the condensate, while falling down through the tube bundle, will be subcooled to a temperature lower than the saturation pressure corresponding to the condenser steam side vacuum. For power generation, condenser subcooling is undesirable since it results in an increase in turbine heat rate that represents a loss of cycle efficiency. Condenser subcooling is also undesirable because the condensate may contain noncondensible gases that could result in corrosion of piping and equipment in the feedwater system. Use of a deaerating hotwell provides for reheating the condensate within the condenser to saturation temperature that effectively deaerates the condensate and eliminates subcooling. Condensers should be specified to provide condensate effluent at saturation temperature corresponding to condenser vacuum and with an oxygen content not to exceed 0.005cc per liter of water (equivalent to 7 parts per billion as specified in the Heat Exchange Institute (HEI), Standards for Steam Surface Condenser, 1970.

1.1.2.4 AIR COOLER SECTION. The condenser tubes and baffles are arranged in such a way as to cause the steam to flow from the condenser steam inlet toward the air cooler section. The steam carries with it the noncondensible gases such as air, carbon dioxide, and ammonia that leave the air cooler section through the air outlets and flow to air removal equipment. Any residual steam is condensed in the air cooler section.

1.2 CONDENSER SIZES. The proper size of condenser is dependent on the following factors:

a) Steam flow to condenser.

b) Condenser absolute pressure.

c) Cooling water inlet temperature.

d) Cooling water velocity through tubes.

e) Tube size (O.D. and gauge).

f) Tube material.

g) Effective tube length (active length between tube sheets).

h) Number of water passes.

i) Tube cleanliness factor.

1.2.1 CONDENSER HEAT LOAD. For approximation, use turbine exhaust steam flow in pound per hour times 950 Btu per pound for non-reheat turbines or 980 Btu per pound for reheat turbines.

1.2.2 CONDENSER VACUUM. Condenser vacuum is closely related to the temperature of cooling water to be used in the condenser. For ocean, lake, or river water, the maximum expected temperature is used for design purposes. For cooling towers, design is usually based on water temperature from the tower and an ambient wet bulb that is exceeded not more than 5 percent of the time. The condenser performance is then calculated to determine the condenser pressure with an ambient wet bulb temperature that is exceeded not more than 1 percent of the time. Under the latter condition and maximum turbine load, the condenser pressure should not exceed 4 inches Hg Abs. Using the peak ambient wet bulb of record and maximum turbine load, the calculated condenser pressure should not exceed 4-1/2 inches Hg Abs. The turbine exhaust pressure monitor is usually set to alarm at 5 inches Hg Abs, which is near the upper limit of exhaust pressure used as a basis for condensing turbine design.

1.2.3 COOLING WATER TEMPERATURES

1.2.3.1 INLET TEMPERATURE. Economical design of condensers usually results in a temperature difference between steam saturation temperature (ts) corresponding to condenser pressure and inlet cooling water temperature (t1) in the range of 20 degrees F (11 degrees C) to 30 degrees F (17 degrees C). Table 13 shows typical design conditions:

Cooling water temperature F(t_1)	Condenser pressure (in-Hg-abs)	degrees F ($t_s - t_1$)
50	1.0	29.0
55	1.0 – 1.25	24.0 – 30.9
60	1.0 – 1.5	19.0 31.7
65	1.5 – 1.75	26.7 – 31.7
70	1.5 – 2.0	21.7 – 31.1
75	2.0 – 2.25	26.1 – 30.1
80	2.0 – 2.5	21.1 – 28.7
85	2.5 – 3.0	23.7 – 31.1
90	3.0 – 3.5	25.1 – 30.6

Table 1

Typical design conditions for steam condensers

1.2.3.2 TERMINAL DIFFERENCE. The condenser terminal difference is the difference in temperature between the steam saturation temperature (t_s) corresponding to condenser pressure and the outlet cooling water temperature (t_2). Economical design of condenser will result with t2 in the range of 5 degrees F (2.8 degrees C) to 10 degrees F (5.6 degrees C) lower than t_s. The HEI Conditions limits the minimum terminal temperature difference that can be used for condenser design to 5 degrees F (2.8 degrees C).

1.2.3.3 TEMPERATURE RISE. The difference between inlet and outlet cooling water temperatures is called the temperature rise that will be typically between 10 degrees F (5.6 degrees C) and 25 degrees F (13.9 degrees C).

1.2.4 TUBE WATER VELOCITY. The maximum cooling water velocity through the tubes is limited by erosion of the inlet ends of the tubes and by the water side pressure drop (friction loss). Velocities in excess of 8 feet per second are seldom used. The normal tube water velocity ranges from 6 to 8 feet per second. Higher velocities provide higher heat transfer but will cause increased friction loss. Where conditions require the use of stainless steel tubes, the tube water velocity should be at least 7 feet per second to ensure that the tubes are continually scrubbed with oxygen for passivation of the stainless steel and maximum protection against corrosion. As a general rule, 7.5 feet per second water velocity is used with stainless steel tubes. When using admiralty tubes, water velocities should be limited to about 7 feet per second to prevent excessive erosion. Previous studies indicate that varying cooling water tube velocities from 6.8 to 7.6 feet per second has very little effect on the economics or performance of the entire cooling water system.

1.2.5 TUBE OUTSIDE DIAMETER AND GAUGE. Condenser tubes are available in the following six outside diameters: 5/8-inch, 3/4-inch, 7/8-inch, 1-inch, 1-1/8 inch, and 1-1/4 inch. For power plants, 3/4-inch, 7/8-inch, and 1-inch OD tubes are the most prevalent sizes. As a general rule, 3/4-inch tubes are used in small condensers up to 15,000 square feet, 7/8-inch tubes are used in condensers between 15,000 and 50,000 square feet. Condensers larger than 50,000 square feet normally use at least 1-inch tubes. Condenser tubes are readily available in 14, 16, 18, 20, 22, and 24 gauge. For inhibited Admiralty or arsenical copper, 18 BWG tubes are normally used. For stainless steel tubes, 22 BWG tubes are normally used.

1.2.6 TUBE LENGTH. The length of tubes is important because of its direct relation to friction loss and steam distribution over the tube bundle. The selection of tube length depends on condenser surface required, space available for the installation, and cooling water pump power required. Normally, economical tube length for single pass condensers will fall in the ranges as shown in Table 2. Two pass condensers will normally have shorter tube lengths.

Tube length, ft	Condenser surface, sq. ft.
16 – 24	Less than 20,000
22 – 30	20,000 to 50,000
30 – 36	50,000 to 100,000
32 - 44	100,000 to 500.000

Table 2

Typical condenser tube length vs. surface

Turbine generator, Kw	Condenser surface, ft^2		Cooling water, gpm		Tube length, feet	Tube O.D., inches
	Single pass	Two pass	Single pass	Two pass		
5,000	3,836	5,147	6,607	4,433	20	¾
7,500	5,754	7,721	9,911	6,650	20	¾
10,000	7,096	9,522	12,223	8,201	20	¾
20,000	12,728	17,079	21,924	14,701	20	¾
30,000	18,486	24,637	32,301	21,525	24	7/8
40,000	24,071	33,290	36,051	24,929	28	7/8
50,000	30,704	43,211	42,921	30,202	30	7/8
60,000	34,705	46,889	57,205	38,645	30	1
80,000	38,706	52,295	63,800	43,100	30	1
100,000	48,180	65,096	79,418	53,650	30	1

Note: Based on use of Admiralty tubes, 85 degrees F (29.4 degrees C) cooling water inlet, 2-1/2 inch Hg Abs. condensing pressure, 85 percent cleanliness factor, 6.5 ft/sec tube water velocity, and 18 BWG tube wall thickness

Table 3

Typical condenser size and cooling water flow

1.2.7 NUMBER OF WATER PASSES. A single pass condenser is commonly used where the water is supplied from natural sources such as rivers or oceans. If the source of circulating water is at all limited, a two pass condenser will probably be the best selection since a single pass condenser requires more cooling water per square foot of condenser surface and per kilowatt of electrical generation. Usually, a two pass condenser is used with cooling towers or a cooling lake. Plant layout and orientation with respect to cooling water source may also dictate the use of a two pass condenser. If sufficient water is available, the most economical condenser is a single pass. A single pass condenser is normally smaller in physical size than the equivalent two pass unit. Typical condenser sizes and cooling water flows for a given turbine generator capacity are given in Table 3.

1.2.8 TUBE CLEANLINESS FACTOR. Design tube cleanliness can vary from 70 to 95 percent depending on tube water velocity, cooling water cleanliness, and cooling water scale-formation characteristics. As condenser tubes become dirty, the heat transfer coefficient is reduced and the condenser vacuum is decreased. When the cooling water is clean or is chlorinated, a factor of 0.85 is normally used. For bad water conditions, a lower value should be used. If the cooling water conditions are very good, a value of 0.90 or 0.95 could be used. For a cooling tower system with stainless steel condenser tubes, it is practical to use a value of 0.90. For a cooling tower system with Admiralty condenser tubes, a tube cleanliness factor of 0.85 should probably be used because of lower tube water velocities through the tubes. In general, with all types of cooling water (river, ocean, lake, cooling tower) a factor of 0.85 is commonly used for copper alloy tubes and 0.90 is commonly used for stainless steel tubes.

7.2.9 Surface. The condensing surface may be calculated by use of eq 1.

$$A = Q/U_m \qquad \text{(eq 5)}$$

where:

A = condensing surface (outside active tube area), ft^2.
Q = condenser heat load = W x Hr, Btu/h
W = exhaust steam from turbine, lb/h
H_r = heat rejected latent heat of exhaust steam, 950 Btu/lb for nonreheat unit, 980 Btu/lb

for reheat unit

U = heat transfer coefficient = $C \times V^{0.5} \times C_1 \times C_2 \times C_f$

V = velocity of cooling water through tubes, fps

$C \times V^{0.5}$ = heat transfer coefficient at 70 deg F, (see Figure 30)

C_1 = correction factor for inlet water temperatures other than 70 degrees F, (see Figure 1)

C_2 = correction factor for tube material and thickness other than No. 18 BWG Admiralty (see Figure 1)

C_f = correction factor for tube cleanliness, (see para. 1.2.8)

m = logarithmic mean temperature difference = $(t_2 - t_1)/\{\log_e [(t_s - t_1)/(t_s - t_2)]\}$

t_2 = cooling water outlet temperature, degrees F

t_1 = cooling water inlet temperature, degrees F

t_s = saturation temperature, degrees F of exhaust steam corresponding to condenser pressure

1.2.10 COOLING WATER FLOW. May be calculated by use of eq 2.

$$G = Q/500(t_2 - t_1) \qquad \text{(eq 2)}$$

where:

G = Condenser cooling water flow, gpm

1.3 CONDENSER MATERIALS. Recommended tube, tube sheet, and water box materials are shown in Table 4.

1.3.1 SHELL. The condenser shell is usually welded steel construction reinforced against collapsing forces resulting from high vacuum. Carbon steels ASTM A283 Grade C, Specification for Low and Intermediate Tensile Strength Carbon Steel Plates, Shapes, Shapes, and Bars, ASTM A285 Grade C, Specification for Pressure Vessel Plates, Carbon Steel, Low and Intermediate Tensile Strength, and ASTM A516 Grade 70, Specification for Pressure Vessel Plates, Carbon Steel, for Moderate and Lower Temperature Service, are commonly used without preference of one type over the others. NEI standards require 1/32-inch corrosion allowance and 1/16-inch corrosion allowance is usually specified.

1.3.2 TUBE SUPPORT PLATES. Tube support plates are located at periodic intervals along the length of the tubes to steady and prevent vibration of the tubes that could otherwise result from impingement of high velocity steam or possibly from cooling water flow. Carbon steel plate, either ASTM A283 Grade C or ASTM A285 Grade C material is usually used. Tube support plates should not be less than 3/4 inch thick. The spacing of the tube support plates shall be in accordance with HEI standards. The maximum spacing for 1-inch, 22 gauge Type 304 stainless steel tubes shall not be greater than 48 inches.

1.3.3 TUBES. Recommended condenser tube gauge, water velocity, and application are shown in Table 5. The relative resistance to various failure mechanisms of most widely used materials is shown in Table 6. A final choice of tube material should not be made without a thorough understanding of the effects and problems related to the following:

a) Tube metal corrosion.
b) Tube metal erosion.
c) Tube water velocity.
d) Cooling water scaling characteristics.
e) Foreign body contamination, particularly seashells.
f) Biofouling.
g) Chemical attack.
h) Galvanic corrosion and protection.
i) Dealloying such as dezincification.
j) Stress corrosion cracking.
k) Tube impingement and vibration.
l) Cooling water characteristics such as freshwater, seawater, brackish water, polluted water, concentrated cooling tower water, etc.

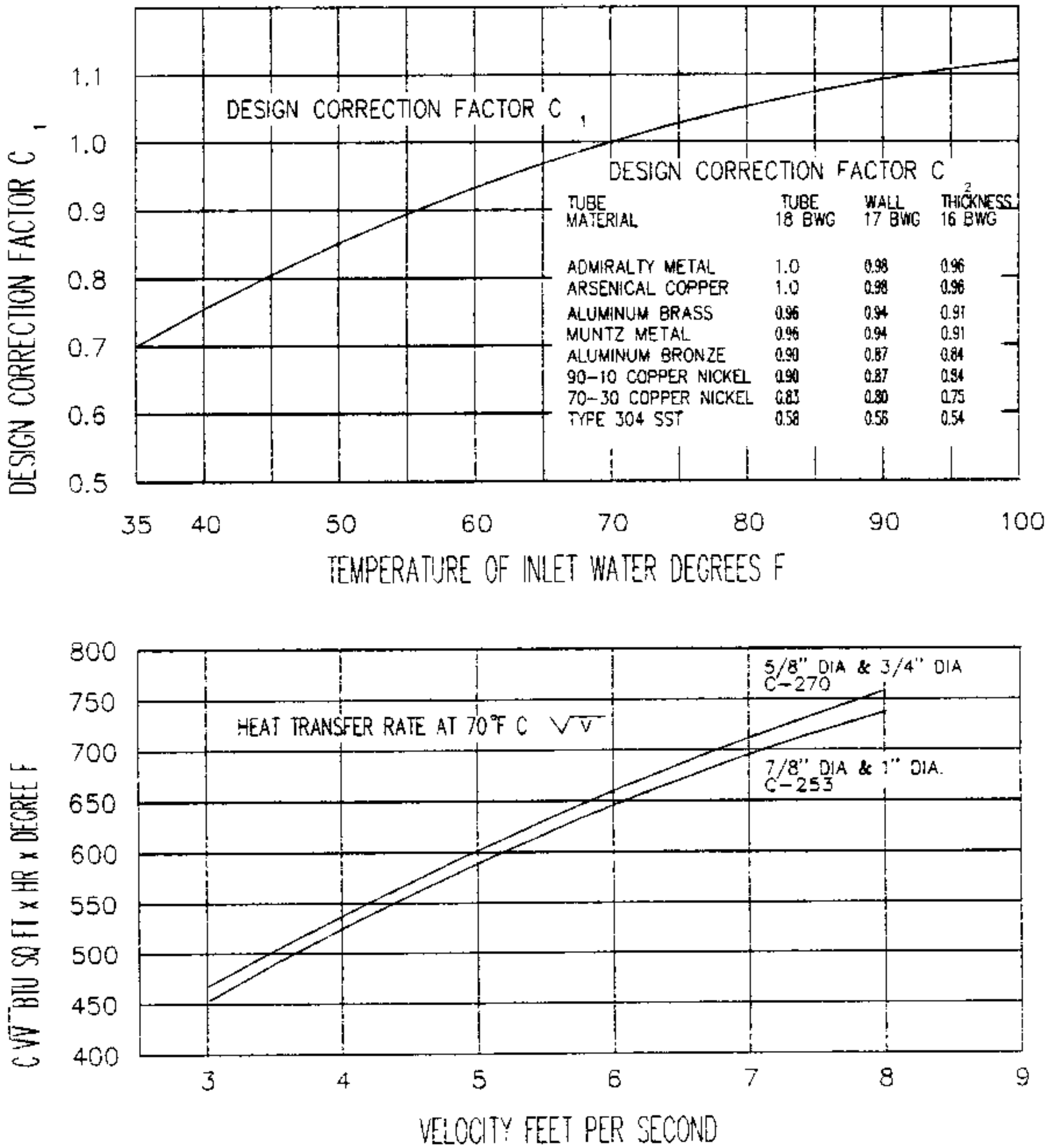

Figure 1

Heat transfer through condenser tubes

Water type	Specific conductance/chlorides[1]		Tube material	Tube sheet material	Interior water box material	Water box coating	Cathodic protection type
	Ms/cm	Mg/L					
Freshwater	2,000	250	304 ss, ASTM 249	Carbon steel ASTM A 283 Gr C	Carbon steel[2]	None	None
Freshwater	6,000	1,000	304 ss, ASTM 249	Carbon steel ASTM A 283 Gr C	Carbon steel[2]	None	Sacrificial anodes
Freshwater	9,000	1,500	304 ss,	Carbon steel ASTM A 283 Gr C	Carbon steel[2]	Yes	Sacrificial anotes
Brackish water and high TDS freshwater	9,000	1,500	90-10 Cu-Ni	Aluminum bronze (Alloy D) C61400	Carbon steel[2]	Yes	Impressed current
Clean seawater	30,000	15,000	90-10 Cu-Ni, C70600	Aluminum bronze (Alloy D) C61400	Carbon steel[2]	Yes	Impressed current
Polluted seawater	30,000	15,000	Titanium, ASTM B 338 Gr	Aluminum bronze (Alloy D) C61400	Carbon steel[2]	Yes	Impressed current

[1] Other chemical characteristics of the cooling water must be considered, such as ph and iron and manganese concentration. Full classification of the cooling water must be on a project by project basis.
[2] ASTM A 285 Gr C or ASTM A 283 Gr C.
[3] Polluted seawater includes water with sulfide related content. Sulfides and sulfide related compounds may be found in cooling waters other than seawater.

Table 4

Recommended tube, tube sheet and water box materials

Freshwater allows	Gauge BWG	Velocity FPS	Possible applications		
			Main body	Air removal section	Periphery
Once-through system					
Admiralty brass	18	8 max	X		
90-10 copper nickel	20	10 max	X	X	X
70-30 copper nickel	18, 20	15 max		X	X
304 stainless steel	22	5 min	X	X	X
Recirculating system					
90-10 copper nickel	20	10 max	X	X	X
70-30 copper nickel	18, 20	15 max		X	X
304 stainless steel[1]	22	5 min	X	X	X
Once-through system					
90-10 copper nickel	18, 20	8 max	X	X	X
85-15 copper nickel		20 max	X	X	X
70-30 copper nickel	18, 20	15 max	X	X	X
"Super" stainless steel	22	5 min	X	X	X
Titanium	22	5 min	X	X	X

1. Low chloride content waters only.

Table 5

Recommended tube gauge, water velocity and application

Failure mechanism	Admiralty	90-10 Cu-Ni	70-30 Cu-Ni	Stainless steel	Titanium
General corrosion	2	4	4	5	6
Erosion corrosion	2	4	5	6	6
Pitting (operating)	4	6	5	4	6
Pitting (stagnate)	2	5	4	1	6
High water velocity	3	4	5	6	6
Inlet end corrosion	2	3	4	6	6
Steam erosion	2	3	4	6	6
Stress corrosion	1	6	5	1	6
Chloride attack	3	6	5	1	6
Ammonia attack	2	4	5	6	6

NOTE: Numbers indicate relative resistance to the indicated cause of failure on a scale of 1 (lowest) to 6 (highest)

Table 6

Relative residtance of most widely used tube material to failure

1.3.3.1 FRESHWATER SERVICE. The most commonly used tube materials for freshwater service are Type 304 stainless steel (ss), 90-10 copper nickel and, to a lesser extent, Admiralty metal. Stainless steels, both type 304 and 316, provide excellent resistance to all forms of corrosion in fresh water. However, stainless steels are susceptible to biofouling, and scale buildup can also be a problem. Almost all failures of stainless steel tubes, because of corrosion, can be traced to the problem of tube fouling (including seawater applications). Type 304 stainless steel is a good selection for freshwater makeup cooling tower systems. Stainless steel provides a good resistance to sulfide attack, but the chloride levels must be kept low. For 304 stainless steel, chlorides less than 1500 mg/L should be acceptable. Copper alloys have also been used successfully in freshwater applications. Their main advantage over stainless steels is better resistance to biofouling. Admiralty and 90-10 copper-nickel have been used in both once-through and recirculating freshwater cooling systems. Admiralty provides good corrosion resistance when used in freshwater at satisfactory velocities (less than 8 fps), good biofouling resistance, good thermal conductivity and strength, and some resistance to sulfide attack. Admiralty is susceptible to stress corrosion cracking if ammonia is present. Admiralty should not be used in the air removal sections. Admiralty is also susceptible to dezincification. Because copper alloys are susceptible to ammonia based stress corrosion cracking, to blockage induced erosion/corrosion, and to deposit related attack, stainless steel (Type 304) is the best tube material for freshwater once-through or recirculating cooling water systems.

1.3.3.2 BRACKISH WATER SERVICE. Brackish water is defined as any water with chlorides in the range of 1500 mg/L to 12000 mg/L and associated high concentrations of total dissolved solids. Brackish water also refers to the recirculating systems with freshwater makeup where the cycles of concentration produce high chlorides and total dissolved solids. In spite of overall excellent corrosion resistance, stainless steels have not been used extensively in brackish or seawater. Type 316 stainless has been used successfully in a few instances and where special care was taken to keep the tubes free of fouling. Because of stringent preventive maintenance requirements and procedures, Type 316 stainless steel is not considered the best tube material for use in brackish water applications. For condenser cooling water with high chloride concentration, increased attention is being given to newly developed austenitic and ferritic stainless steels. It is generally accepted that for austenitic stainless steel to resist corrosion, the molybdenum content should be 6 percent with a chromium content of 19 to 20 percent. For ferritic steels, the molybdenum content should be at least 3 percent and the chromium content should probably be 25 percent or more. Copper alloys, including aluminum brass, aluminum bronze, and copper-nickel have been used extensively in brackish water applications. Because of overall failure rate experience, 90-10 copper-nickel is the recommended tube material for use with brackish water in the main body of tubes with 70-30 copper-nickel in the air removal sections. When inlet end infringement and erosion attack due to water flow is a potential problem, 85-15 copper-nickel should be considered. However, if the brackish cooling water is also characterized by high sulfide concentrations, consideration for use of the "super" stainless steels is recommended.

1.3.3.3 SEAWATER SERVICE. Seawater materials are considered wherever the chlorides in the cooling water are greater than 15,000 ppm. Seawater also includes cooling tower systems where brackish water is concentrated and high chlorides and total dissolved solids result. As long as the seawater is relatively clean and free of pollution, the recommendations for brackish water materials are applicable. Titanium tubes are being used with increasing frequency for seawater application. Titanium is essentially resistant to all oxidizing media by virtue of the stable, protective oxide film. The major problems with titanium tubes include its high fouling rate in low water velocity systems, its susceptibility to hydrogen embrittlement, and its low modulus of elasticity. Where scale formation or microbiological slimes can possibly occur, an on-line mechanical tube cleaning system is required to maintain a high tube cleanliness factor. Careful attention must also be given to support plate spacing to avoid vibration when using thin walled titanium tubes and extra support plates are needed. Because of the expense and potential problems with titanium tubes, 90-10 copper-nickel and 70-30 copper-nickel tubes are considered better selections for clean seawater applications. When the seawater is also characterized by a high sulfide concentration, the new austenitic and ferritic stainless steel condenser tube alloys should be considered since 90-10 copper-nickel and 70-30 copper-nickel are highly susceptible to sulfide pitting attack.

1.3.3.4 POLLUTED WATER SERVICE. Polluted water materials should be used whenever sulfides, polysulfides, or elemental sulfur are present in the cooling water. Sulfides produce and accelerate corrosion of copper alloys. Therefore, copper based alloy tubes are not considered feasible polluted water materials. Stainless steel is also not acceptable since the polluted water is usually brackish or seawater. This leaves titanium and the new austenitic and ferritic stainless steels. The most acceptable of these tube materials is titanium based mainly on its greater experience. However, the "super" stainless steels, which were created predominantly for use with polluted cooling water, are less expensive than titanium and are not expected to experience any of the problems with cathodic protection systems that are possible with titanium tubes. The majority of installations using these new "super" stainless materials are located in coastal areas with polluted cooling water. To date, the results have been favorable for the "super" stainless steels used in this application.

1.3.4 TUBESHEETS. In order to prevent galvanic action between tubes and tubesheets, the obvious selection of tubesheet material for new units is the use of same material as the tubes. However, this may be prohibitively expensive. The next best choice is to use materials that are as close as possible to one another in the galvanic series or ensure satisfactory performance by using coatings or cathodic protection.

1.3.4.1 FRESHWATER SERVICE. Tubesheet material compatible with stainless steel tubes are carbon steel and stainless steel. Carbon steel has been used successfully for tubesheet material. Since it is less expensive than stainless steel, it is the obvious tubesheet material selection for use with stainless steel tubes. Muntz metal is the most widely used tubesheet material with copper-nickel tubes. Muntz metal is also suitable for use with Admiralty tubes.

1.3.4.2 BRACKISH AND SEAWATER SERVICE. Because of their relatively high yield strengths, aluminum bronze and silicon bronze provide good tube-to-tubesheet joint integrity and good pull-out strength. The materials are compatible with all copper alloy tubes. Even if titanium tubes are used, aluminum bronze is the most common choice for tubesheet material involving welded tubes. (Cathodic protection is required, however.) Silicon bronze tubesheets are not widely used. Aluminum bronze is the preferred tubesheet material for copper-nickel installations since silicon bronze is not easily weldable and since Muntz metal does not provide as much strength. Also, as a tubesheet material, aluminum bronze is significantly less susceptible to galvanic corrosion than Muntz metal. The cost difference between aluminum bronze and silicon bronze is slight. The actual material cost of silicon bronze is slightly lower but the added thickness or supports required for high pressure designs negates any material savings. Muntz metal is not the best material since it does not have the strength required to ensure adequate tube-to-tubesheet integrity. However, as with freshwater, Muntz metal tubesheets are often used with copper-nickel tubes. New condenser tubesheet materials are under consideration as a result of the ever increasing use of the new austenitic and ferritic stainless steel condenser tube materials. Stainless steels such as Type 316L and other proprietary alloys are similar to Type 304 stainless steel but with the addition of molybdenum that offers increased resistance to general corrosion, pitting, and crevice corrosion attack. Galvanic corrosion between the new "super" stainless steel condenser tubes and these tubesheet materials is minimized because their similar compositions places them relatively close on the galvanic series chart. The 316L and similar tubesheet materials are also slightly cathodic to the "super" stainless steel tube alloys. This is desirable since whatever corrosion takes place, if any, will occur on the thicker tubesheet instead of the thinner walled tubes.

1.3.4.3 POLLUTED WATER SERVICE. Aluminum bronze is the preferred tubesheet material for titanium tubes. For extremely polluted water, a titanium tubesheet (or titanium cladded tubesheet) should be considered. This arrangement would prevent any potential galvanic corrosion of the aluminum bronze as well as eliminate any problem with corrosion due to the sulfides. A properly designed cathodic protection system should protect the aluminum bronze tubesheet. Recommended tubesheet materials for use with tubes made of the new austenitic and ferritic stainless steels are the same as described under brackish water.

1.3.5 WATER BOXES

1.3.5.1 FRESHWATER SERVICE. Use carbon steel ASTM A285 Grade C or ASTM A283 Grade C with copper alloy or stainless steel tubes.

1.3.5.2 BRACKISH AND SEAWATER SERVICE. Water box materials include carbon steel, stainless steel, and 90-10 copper-nickel. In brackish water, there is no advantage to using stainless steel over carbon steel since stainless steel is more expensive and is also susceptible to corrosion. A feasible alternative is 90-10 copper-nickel but it is significantly more expensive than carbon steel. Carbon steel is an acceptable choice assuming that the interior of the water box is properly coated and that some form of cathodic protection for the water box is provided.

1.3.5.3 POLLUTED WATER. Coated carbon steel water boxes with cathodic protection is the recommended choice for use with titanium or the new austenitic and ferritic stainless steel tubes.

1.3.6 EXHAUST NECK. The connection piece extending from the turbine exhaust flange to the main body of the condenser and often referred to as the condenser neck is made of the same material as the condenser shell.

1.3.7 EXPANSION JOINTS

1.3.7.1 EXHAUST NECK. For bottom supported condensers, an expansion joint made of copper, stainless steel, or rubber is located between the turbine exhaust flange and the main body of the condenser, either as a part of the exhaust neck of the condenser or separate component. Corrosion of copper joints has caused the use of this material to be essentially discontinued. The use of stainless steel is satisfactory but expensive. The majority of all condensers are now furnished with a rubber (dogbone type) expansion joint. The rubber dogbone type is preferred because it can more easily be replaced as compared to a stainless steel joint.

1.3.7.2 SHELL. Depending upon the type of tube to tubesheet joining, there can be and usually is a difference in expansion between the shell and tubes during operation. Suitable means must be incorporated in the design of the condenser to provide for this differential expansion. Both flexing steel plate and U-bend type have been used; however, the majority of condensers are furnished with a steel U-bend type that is usually located adjacent to one of the tube sheets.

1.4 CONDENSER SUPPORT

1.4.1 BOTTOM SUPPORT. Bottom support is the simplest method and consists of mounting the condenser rigidly on its foundation. The condenser dome, turbine exhaust extension piece, or condenser neck as it is commonly called is attached to the turbine exhaust flange by bolting or welding and contains an expansion joint of stainless steel, copper, or rubber.

1.4.2 SPRING SUPPORT. The condenser is bolted directly to the turbine exhaust flange and supported at the bottom feet by springs to allow for expansion. This avoids the use of an expansion joint in the condenser neck. However, all piping connected to the condenser for auxiliaries must be provided with expansion joints to permit free movement of the condenser. This method is seldom used.

1.4.3 RIGID SUPPORT. The condenser is bolted to and supported from the turbine exhaust. The center of gravity of the condenser must be centered on the turbine exhaust. As with the spring support method, all auxiliary piping must be provided with expansion joints. The use of this method is restricted to small turbine generator units.

1.5 CONDENSER AIR REMOVAL

1.5.1 CONTINUOUS AIR REMOVAL. Continuous air removal is accomplished by use of either a steam jet air ejector or mechanical air exhausters (vacuum pumps). Recommended capacities of air removal (venting) equipment for single shell condensers should not be less than shown in Table 7.

1.5.2 HOGGING AIR REMOVAL. For evacuating steam space, when starting up to a condenser pressure of about 10-inch Hg Abs., a steam operated hogging ejector or mechanical air exhausters (the same equipment as used for continuous air removal) must be used. Hogger capacities are shown in Table 8.

Turbine exhaust steam flow, lb/hr	Steam/air mixture SCFM
Up to 25,000	3.0
25,001 to 50,000	4.0
50,001 to 100,000	5.0
100,001 to 250,000	7.5
250,001 to 500,000	10.0
500,001 to 1,000,000	12.5

Table 7

Venting equipment capacities for single shell condenser

Turbine exhaust steam flow, lb/hr	Dry air, SCFM (at 1.0” Hg absolute suction pressure)
Up to 100,000	50
100,001 to 250,000	100
250,001 to 500,000	200
500,001 to 1,000,000	350

Table 8

Hogger capacities

2. AUXILIARY EQUIPMENT

2.1 CONDENSATE STORAGE AND TRANSFER. About 0.5 percent of the steam flow to the turbine is lost from the cycle. These losses occur at points such as the deaerator continuous noncondensibles and steam vent, pump glands, valve packing leaks, continuous boiler blowdown, and continuous water and steam samples. Demineralized water is also required for filling the boiler/turbine generator unit system initially prior to startup and during times of boiler or cycle maintenance and chemical cleaning. The condensate storage and transfer equipment is illustrated in Figure 2.

2.1.1 CONDENSATE STORAGE TANK. For normal operation, the excess or deficiency of cycle water caused by load changes is usually handled by providing a condensate storage tank which can accept and hold excess condensate or provide condensate makeup for cycle water deficiency. A tank sized for twice the cycle water swell volume will usually provide sufficient capacity for normal condensate makeup and dump requirements. Condenser vacuum is normally used as the motive force to draw condensate from the storage tank to the condenser through makeup control valves. Condensate dump from the cycle to the storage tank usually is made from the condensate pump discharge through dump control valves. For cogeneration plants, the function of condensate return from heating and other processes is usually combined with the function of condensate storage using a single tank.

2.1.2 DEIONIZED OR DEMINERALIZED WATER STORAGE TANK. Water required for filling the cycle or boiler either initially, for maintenance or for chemical cleaning, is usually stored in separate tanks which contain deionized or demineralized water. The amount of storage required is about 1,000 gallons per MW of installed electric generating capacity which is usually divided into not less than two tanks. Provide two pumps for transfer of water as needed from these tanks to the condensate storage/return tank. If an evaporator is used for cycle water makeup, a similar amount of 1,000 gallons per MW storage capacity is necessary.

2.1.3 CONDENSATE RECEIVERS AND PUMPS SIZING. For sizing of condensate receivers and associated pumps, see the technical literature.

2.1.4 CONDENSATE PUMPS. Condenser condensate pumps are used for pumping condensate from the turbine condenser to the deaerator through the low pressure feedwater heaters, the steam jet air ejector, and the turbine gland steam condenser (if any). Two condenser condensate pumps, each capable of handling full load operation, shall be provided of either the horizontal split case or vertical can type. The vertical can type pumps are often used because the construction and installation provides for net positive suction head (NPSH) requirements without the use of a pit for pump location.

2.1.5 CONDENSATE TRANSFER PUMPS. Condensate transfer pumps are used to pump condensate from the main condensate storage/return tank to the deaerator. Two condensate transfer pumps, each capable of handling full load operation, shall be provided of either the horizontal split case or vertical can type. The standby condensate transfer pump can be used for boiler fill, emergency condensate makeup to the deaerator, and initial fill of the condenser.

2.1.6 CONDENSATE CLEANING. Oil and other undesirable matter should be removed from condensate returned from the process and fuel oil tubular heat exchangers. Oil will cause foaming and priming in the boilers as well as scale.

2.1.6.1 WASTAGE. Condensate containing oil should be wasted.

2.1.6.2 FILTRATION. Where the amount of oil-contaminated condensate is so great that it would be uneconomical to waste it, provide cellulose, diatomite, leaf filters, or other acceptable methods to clean the condensate.

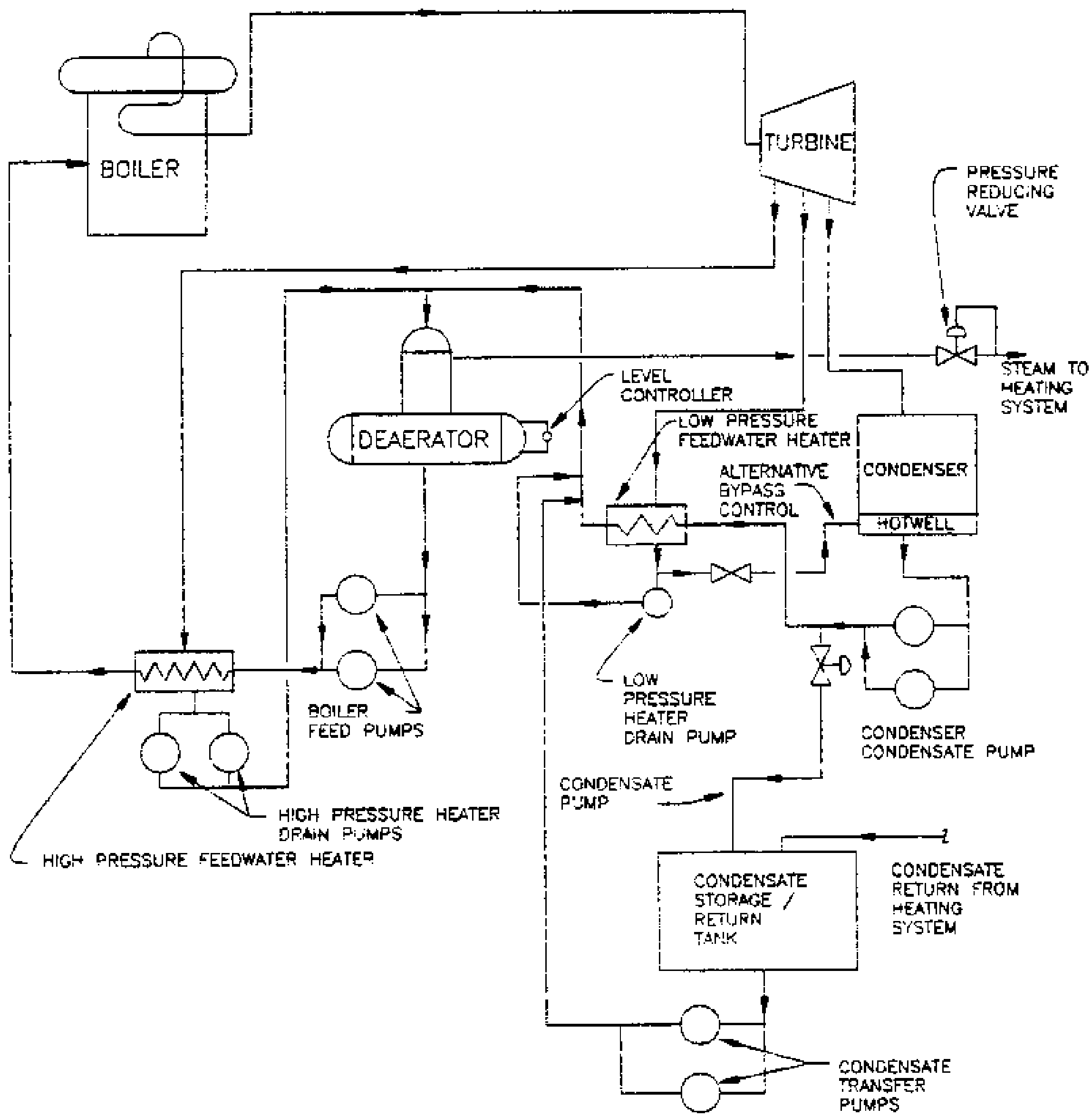

Figure 8

Typical steam plant flow diagram

2.2 FEEDWATER HEATERS. Low pressure feedwater heaters are used in the condensate system between the condensate pump discharge and boiler feed pumps, and utilize low pressure turbine extraction or auxiliary turbine exhaust steam for heating the condensate. High pressure feedwater heaters are used in the feedwater system between the boiler feed pump discharge and the boiler, and utilize high pressure turbine extraction steam for heating the feedwater. The condensate or feedwater temperature increase for each feedwater heater will be in the range of 50 to 100 degrees F (28 to 56 degrees C) with the actual value determined by turbine manufacturer's stage location of steam extraction nozzles. Depending on turbine size, some turbines offer alternate number of extraction nozzles with usually a choice of using the highest pressure extraction nozzle. The selection, in this case, of the total number of feedwater heaters to use should be based on economic evaluation. The feedwater heater equipment is illustrated in Figure 2.

2.2.1 LOW PRESSURE HEATER(S). Use one or more low pressure feedwater heaters to raise the temperature of condensate from condensate pump discharge temperature to the deaerator inlet temperature. The heater drains are cascaded from the higher pressure heater to the next lower pressure heater with the lowest pressure heater draining to the condenser.

2.2.2 HIGH PRESSURE HEATER(S). Use one or more high pressure feedwater heaters to raise the temperature of feedwater from deaerator outlet temperature to the required boiler economizer inlet temperature. The heater drains are cascaded from heater to heater, back to the deaerator in a fashion similar to the heater drain system for the low pressure heaters.

2.3 HEATER DRAIN PUMPS

2.3.1 LOW PRESSURE HEATER DRAIN PUMP. Low pressure heater drain pumps may be used for pumping drains from the lowest pressure heater to a point in the condensate piping downstream from the heater in lieu of returning the drains to the condenser. Pumping of the heater drains in this fashion provides recovery of heat which would otherwise be lost to the condenser. The use of low pressure heater drain pumps can be decided by economic evaluation. Use only one pump and provide alternate bypass control of drains to the condenser for use when the drain pump is out of service.

2.3.2 HIGH PRESSURE HEATER DRAIN PUMPS. High pressure heater drain pumps are required, when high pressure heater drains are cascaded to the deaerator, in order to overcome the elevation difference between the lowest high pressure heater and deaerator. Use two full capacity pumps with one of the two pumps for standby use.

2.4 DEAERATORS. Provide at least one deaerator for the generating plant. The deaerator usually is arranged in the cycle to float in pressure with changes in extraction pressure (which changes with turbine load). Deaerator(s) for power plants usually heat the condensate through a range of 50 to 75 degrees (28 to 42 degrees C).

2.4.1 DEAERATOR FUNCTION. The primary function of the deaerator is to remove dissolved oxygen from the condensate in excess of 0.005 cc of oxygen per liter of condensate at all loads. In addition, the deaerator will normally perform the following functions:

a) Heat the condensate in the last stage of condensate system prior to the boiler feedwater system.
b) Receive the boiler feed pump recirculation.
c) Provide the boiler feed pumps with the required net positive suction head.
d) Receive water from the condensate system and provide surge capacity in the storage
tank.
e) Provide hot water for air preheating and combustion gas reheating (if any) and/or other auxiliary heat requirements.
f) Receive drains from high pressure heaters.
g) Receive high pressure trap drains.
The deaerator functions are illustrated in Figure 3.

2.4.2 DEAERATOR DESIGN PRESSURE

a) Turbine manufacturers indicate that the extraction pressure quoted on the heat balances may vary as much as plus or minus 10 percent. Considering operation with extraction heaters out of service, the manufacturers recommend that the deaerator be designed for a possible 15 percent increase in the pressure from that shown on the manufacturer's heat balance.

b) Safety valve manufacturers recommend that a suitable margin be provided between the maximum operating pressure in a vessel and the set pressure of the lowest set relief valve. This prevents any undesirable operation of the relief device. They suggest that this margin be approximately 10 percent above the maximum operating pressure or 25 psi, whichever is greater.

c) The deaerator design pressure shall be specified with a design pressure equal to maximum extraction pressure x 1.15 plus allowance for safety valve. The allowance for safety valve shall equal maximum extraction pressure x 1.15 x 0.1 or 25 psi, whichever is greater. The design pressure should be rounded up to the nearest even 10 psi. The maximum allowable working pressure shall be assumed to be equal to the design pressure.

d) If the deaerator design pressure is 75 psi or greater, the deaerator shall also be designed for full vacuum (thereby eliminating the need for a vacuum breaker.)

2.4.3 DEAERATOR STORAGE VOLUME. The deaerator storage volume, elevation, and boiler feed pump net positive suction head (NPSH) are related as outlined in Rodney S. Thurston's paper "Design of Suction Piping: Piping and Deaerator Storage Capacity to Protect Feed Pumps," Journal of Engineering for Power, Volume 83, January 1961, ASME pp 69-73. The boiler feed pump NPSH is calculated using the following equation:

$$NPSH = (P_a + P_s - P_v) \times (144/D) + h_s - f \qquad \text{(eq 3)}$$

where:

P_a = Atmospheric pressure, psia
P_s = Steam pressure in deaerator, psig
P_v = Vapor pressure of boiler feedwater at boiler feed pump suction, psia
D = Density of boiler feedwater, lb/cu. ft.
h_s = Static head between deaerator water level and centerline of boiler feed pump, ft
f = Friction loss in piping from deaerator to boiler feed pump suction, ft

a) Deaerator storage volume should be not less than the volume of feedwater equivalent to 10 minutes of feedwater flow at full turbine load.

b) The deaerator is usually located at the same elevation as the boiler main upper drum.

2.4.4 DEAERATOR RATING

2.4.4.1 RATED CAPACITY. The rated capacity of a deaerator is the quantity of deaerated water in pounds per hour delivered to the boiler feed pumps by the deaerating unit and includes all of the steam used for heating in the deaerator.

2.4.4.2 OXYGEN REMOVAL. Deaerators should be specified to provide condensate effluent, at all loads, at saturation temperature corresponding to deaerator pressure and with an oxygen content not to exceed 0.005cc of oxygen per liter of condensate.

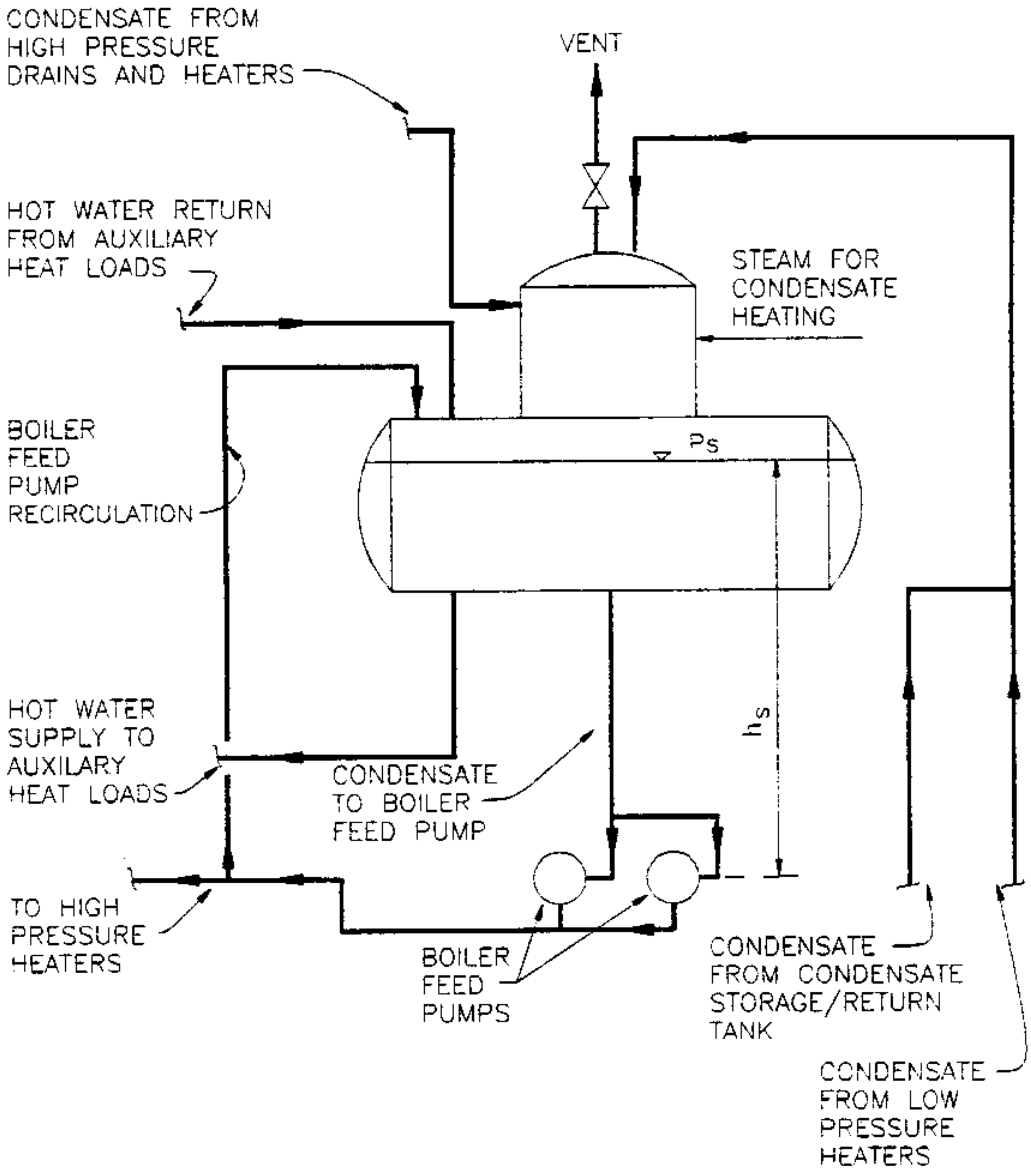

Figure 3

Deaerator functions

2.5 BOILER FEED PUMPS. For design and other data relative to boiler feed pumps and feedwater pumping systems, see the technical literature.

2.6 PRESSURE REDUCING AND DESUPERHEATING STATIONS. A pressure reducing and desuperheating station is shown in Figure 4.

2.6.1 PRESSURE REDUCING STATIONS. Typical use of pressure reducing control valves are as follows:

a) Boiler drum steam supply to auxiliary steam system supplying building heating equipment, fuel oil heaters, and deaerator standby steam supply.

b) Main steam supplemental and standby supply to export steam.

c) High pressure extraction bypass to deaerator.

d) Main steam supply to steam jet air ejector, if used.

For load variations of 3:1 and larger, use two parallel pressure reducing stations with a common valved bypass; use one for one-third of total load and the other for two-thirds load instead of single pressure reducing valve, or station.

2.6.2 DESUPERHEATING STATIONS. Desuperheating stations usually consist of a control valve station which is used to regulate the flow of desuperheating water (from boiler feed pumps or condensate pumps discharge, depending upon the reduced pressure of export steam) to the desuperheater. Water used for tempering must be of demineralized water or good quality condensate to avoid mineral deposits on the desuperheater. Desuperheaters may be of the steam or mechanically atomized type. Desuperheaters may also be used on the boiler steam headers for main steam temperature control, depending upon the design of the boiler.

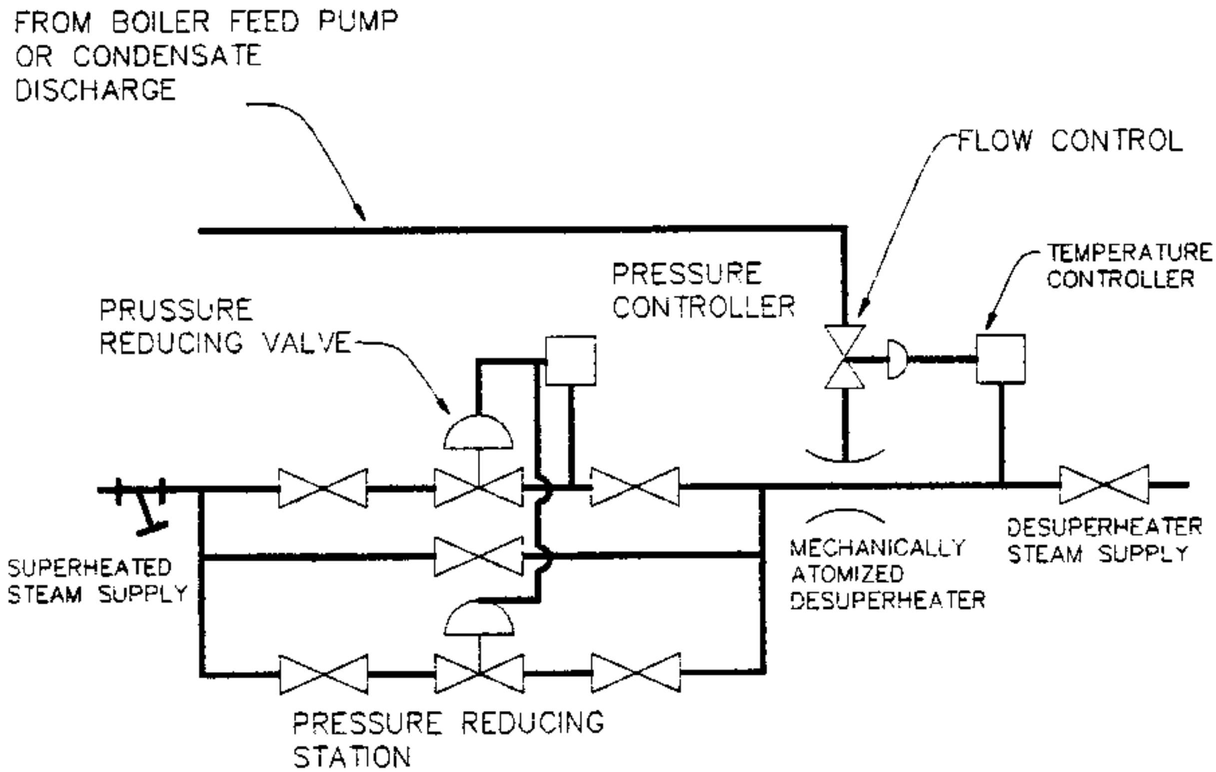

Figure 4

Typical pressure reducing and desuperheating stations

2.7 COMPRESSED AIR SYSTEM

2.7.1 APPLICATIONS. The major uses of compressed air for power plants are for plant service which includes boiler fuel oil atomizing, soot blowing, and instrument air supply. The use of compressed air for fuel oil atomizing should be economically evaluated versus steam or mechanical atomization. The use of compressed air versus steam blowing for soot blowers should also be economically evaluated.

2.7.2 EQUIPMENT DESCRIPTION, DESIGN, AND ARRANGEMENT. For description of types, design requirements, and arrangement of air compressors, aftercoolers, receivers, and air dryers, see the technical literature.

2.8 AUXILIARY COOLING WATER SYSTEM. A closed circulating cooling water system shall be provided for cooling the bearings of auxiliary equipment such as pumps and fans, for air compressor jackets and aftercoolers, turbine oil coolers, generator air or hydrogen coolers, and sample cooling coils. The system shall consist of two shell and tube heat exchangers, two water circulating pumps, one head tank, and necessary valve and piping. A typical auxiliary cooling water system is illustrated in Figure 5.

The auxiliary cooling water head tank is used as an expansion tank, to provide head on the system and to provide a still volume to permit release of air from the system. The normal operating water level of the head tank should be approximately 70 percent of the tank capacity. To provide for expansion from cold to operating temperatures, a volume equal to approximately one percent of the volume of the system should be provided between the normal operating level and the high water alarm. The tank should be provided with an overflow piped to drain. The tank should be located above the highest piece of equipment being cooled by the auxiliary cooling water system. This will assure a positive pressure throughout the system both during normal operation and in the shutdown mode. The temperature of auxiliary cooling water from the heat exchangers should be maintained constant by use of an automatic temperature control system which regulates a control valve to bypass auxiliary cooling water around the cooling water heat exchangers. The auxiliary cooling water in the system is treated initially upon filling with chemical additives to prevent corrosion throughout the system. Chemical concentration of water contained in the system is maintained during plant operation by periodic injection of chemicals. This is accomplished by means of a pot feeder located on the discharge of the auxiliary cooling water pumps.

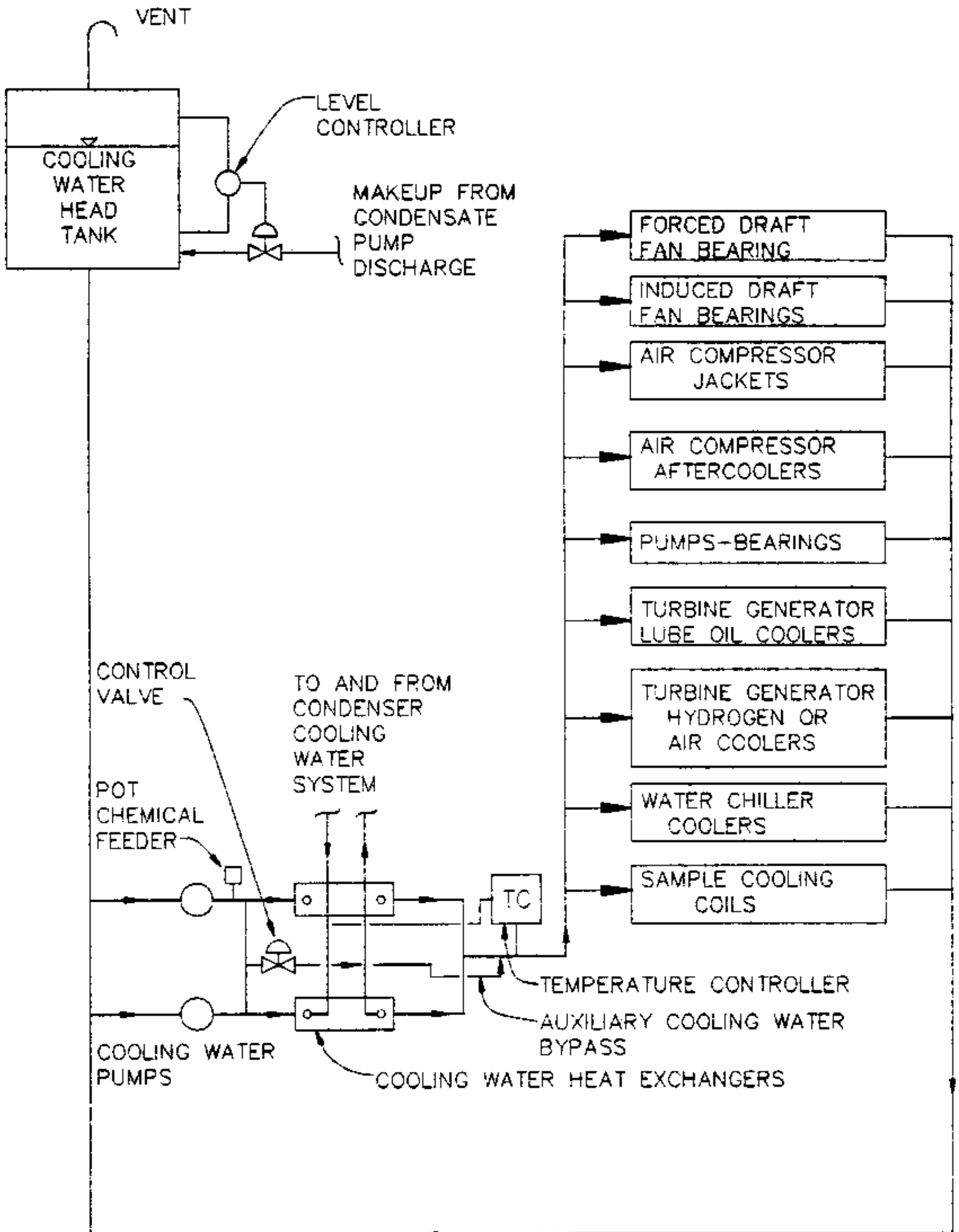

Figure 5

Typical auxiliary cooling water station