CHAPTER 3

STEAM TURBINE POWER PLANT DESIGN

Section 1. TYPICAL PLANTS AND CYCLES

3-1. Introduction

a. Definition. The cycle of a steam power plant is the group of interconnected major equipment components selected for optimum thermodynamic characteristics, including pressure, temperatures and capacities, and integrated into a practical arrangement to serve the electrical (and sometimes by-product steam) requirements of a particular project. Selection of the optimum cycle depends upon plant size, cost of money, fuel costs, non-fuel operating costs, and maintenance costs.

b. Steam conditions. Typical cycles for the probable size and type of steam power plants at Army establishments will be supplied by superheated steam generated at pressures and temperatures between 600 psig (at 750 to 850°F) and 1450 psig (at 850 to 950° F). Reheat is never offered for turbine generators of less than 50 MW and, hence, is not applicable in this manual.

c. Steam turbine prime movers. The steam turbine prime mover, for rated capacity limits of 5000 kW to 30,000 kW, will be a multi-stage, multi-valve unit, either back pressure or condensing. Smaller turbines, especially under 1000 kW rated capacity, may be single stage units because of lower first cost and simplicity. Single stage turbines, either back pressure or condensing, are not equipped with extraction openings.

d. Back pressure turbines. Back pressure turbine units usually exhaust at pressures between 250 psig and 15 psig with one or two controlled or uncontrolled extractions. However, there is a significant price difference between controlled and uncontrolled extraction turbines, the former being more expensive. Controlled extraction is normally applied where the bleed steam is exported to process or district heat users.

e. Condensing turbines. Condensing units exhaust at pressures between 1 inch of mercury absolute (Hga) and 5 inches Hga, with up to two controlled, or up to five uncontrolled, extractions.

3-2. Plant function and purpose

a. Integration into general planning. General plant design parameters will be in accordance with overall criteria established in the feasibility study or planning criteria on which the technical and economic feasibility is based. The sizes and characteristics of the loads to be supplied by the power plant, including peak loads, load factors, allowances for future growth, the requirements for reliability, and the criteria for fuel, energy, and general economy, will be determined or verified by the designer and approved by appropriate authority in advance of the final design for the project.

b. Selection of cycle conditions. Choice of steam conditions, types and sizes of steam generators and turbine prime movers, and extraction pressures depend on the function or purpose for which the plant is intended. Generally, these basic criteria should have already been established in the technical and economic feasibility studies, but if all such criteria have not been so established, the designer will select the parameters to suit the intended use.

c. Coeneration plants. Back pressure and controlled extraction/condensing cycles are attractive and applicable to a cogeneration plant, which is defined as a power plant simultaneously supplying either electric power or mechanical energy and heat energy (para. 3-4).

d. Simple condensing cycles. Straight condensing cycles, or condensing units with uncontrolled extractions are applicable to plants or situations where security or isolation from public utility power supply is more important than lowest power cost. Because of their higher heat rates and operating costs per unit output, it is not likely that simple condensing cycles will be economically justified for a military power plant application as compared with that associated with public utility ‘purchased power costs. A schematic diagram of a simple condensing cycle is shown on Figure 3-1.

3-3. Steam power cycle economy

a. Introduction. Maximum overall efficiency and economy of a steam power cycle are the principal design criteria for plant selection and design. In general, better efficiency, or lower heat rate, is accompanied by higher costs for initial investment, operation and maintenance. However, more efficient cycles are more complex and may be less reliable per unit of capacity or investment cost than simpler and

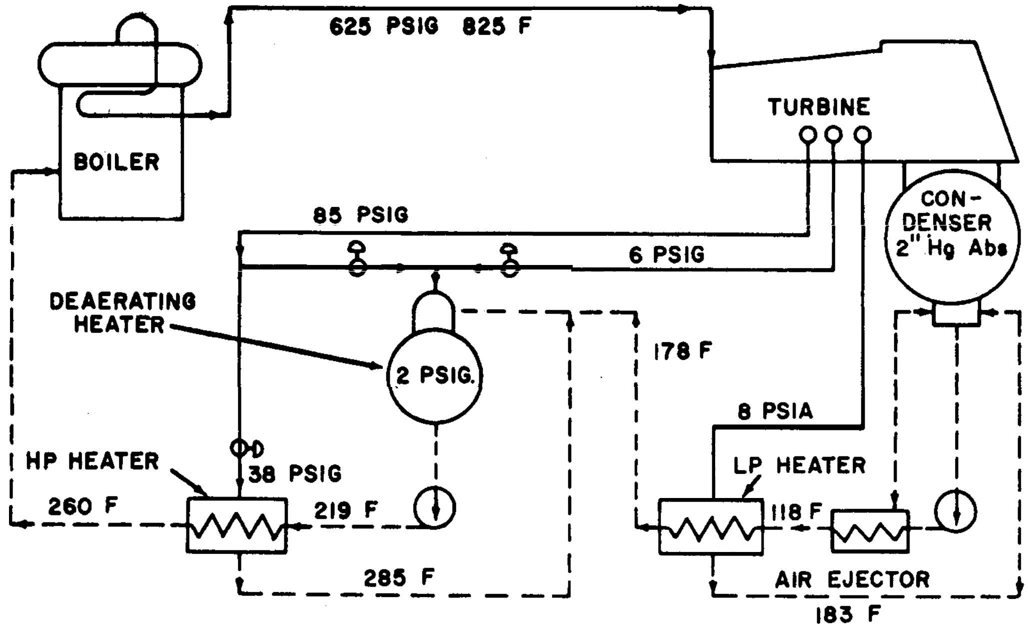

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Figure 3-1. Typical straight condensing cycle.

less efficient cycles. Efficiency characteristics can be listed as follows:

(1) Higher steam pressures and temperatures contribute to better, or lower, heat rates.

(2) For condensing cycles, lower back pressures increase efficiency except that for each particular turbine unit there is a crossover point where lowering back pressure further will commence to decrease efficiency because the incremental exhaust loss effect is greater than the incremental increase in available energy.

(3) The use of stage or regenerative feedwater cycles improves heat rates, with greater improvement corresponding to larger numbers of such heaters. In a regenerative cycle, there is also a thermodynamic crossover point where lowering of an extraction pressure causes less steam to flow through the extraction piping to the feedwater heaters, reducing the feedwater temperature. There is also a limit to the number of stages of extraction/feedwater heating which may be economically added to the cycle. This occurs when additional cycle efficiency no longer justifies the increased capital cost.

(4) Larger turbine generator units are generally more efficient that smaller units.

(5) Multi-stage and multi-valve turbines are more economical than single stage or single valve machines.

(6) Steam generators of more elaborate design, or with heat saving accessory equipment are more efficient.

b. Heat rate units and definitions. The economy or efficiency of a steam power plant cycle is expressed in terms of heat rate, which is total thermal input to the cycle divided by the electrical output of the units. Units are Btu/kWh.

(1) Conversion to cycle efficiency, as the ratio of output to input energy, may be made by dividing the heat content of one kWh, equivalent to 3412.14 Btu by the heat rate, as defined. Efficiencies are seldom used to express overall plant or cycle performance, although efficiencies of individual components, such as pumps or steam generators, are commonly used.

(2) Power cycle economy for particular plants or stations is sometimes expressed in terms of pounds of steam per kilowatt hour, but such a parameter is not readily comparable to other plants or cycles and omits steam generator efficiency.

(3) For mechanical drive turbines, heat rates are sometimes expressed in Btu per hp-hour, excluding losses for the driven machine. One horsepower hour is equivalent to 2544.43 Btu.

c. Heat rate applications. In relation to steam power plant cycles, several types or definitions of heat rates are used:

(1) The turbine heat rate for a regenerative turbine is defined as the heat consumption of the turbine in terms of “heat energy in steam” supplied by the steam generator, minus the “heat in the feedwater” as warmed by turbine extraction, divided by the electrical output at the generator terminals. This definition includes mechanical and electrical losses of the generator and turbine auxiliary systems, but excludes boiler inefficiencies and pumping losses and loads. The turbine heat rate is useful for

performing engineering and economic comparisons of various turbine designs. Table 3-1 provides theoretical turbine steam rates for typical steam throttle conditions. Actual steam rates are obtained by dividing the theoretical steam rate by the turbine efficiency. Typical turbine efficiencies are provided on Figure 3-2.

ASR =

where: ASR = actual steam rate (lb/kWh)

TSR = theoretical steam rate (l/kWh)

n_t = turbine efficiency

Turbine heat rate can be obtained by multiplying the actual steam rate by the enthalpy change across the turbine (throttle enthalpy - extraction or exhaust enthalpy).

C_t = $ASR(h_1 - h_2)$

where = turbine heat rate (Btu/kWh)

ASR = actual steam rate lb/kWh)

h_1 = throttle enthalpy

h_1 = extraction or exhaust enthalpy

$$\frac{TSR}{n_t}$$

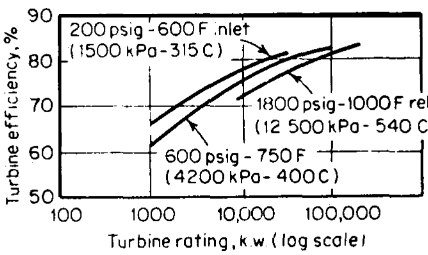

Figure 3-2. Turbine efficiencies vs. capacity.

(2) Plant heat rates include inefficiencies and losses external to the turbine generator, principally the inefficiencies of the steam generator and piping systems; cycle auxiliary losses inherent in power required for pumps and fans; and related energy uses such as for soot blowing, air compression, and similar services.

(3) Both turbine and plant heat rates, as above, are usually based on calculations of cycle performance at specified steady state loads and well defined, optimum operating conditions. Such heat rates are seldom achieved in practice except under controlled or test conditions.

(4) Plant operating heat rates are long term average actual heat rates and include other such losses and energy uses as non-cycle auxiliaries, plant lighting, air conditioning and heating, general water supply, startup and shutdown losses, fuel deterioration losses, and related items. The gradual and inevitable deterioration of equipment, and failure to operate at optimum conditions, are reflected in plant operating heat rate data.

d. Plant economy calculations. Calculations, estimates, and predictions of steam plant performance will allow for all normal and expected losses and loads and should, therefore, reflect predictions of monthly or annual net operating heat rates and costs. Electric and district heating distribution losses are not usually charged to the power plant but should be recognized and allowed for in capacity and cost analyses. The designer is required to develop and optimize a cycle heat balance during the conceptual or preliminary design phase of the project. The heat balance depicts, on a simplified flow diagram of the cycle, all significant fluid mass flow rates, fluid pressures and temperatures, fluid enthalpies, electric power output, and calculated cycle heat rates based on these factors. A heat balance is usually developed for various increments of plant load (i.e., 25%, 50%, 75%, 100% and VWO (valves wide open)). Computer programs have been developed which can quickly optimize a particular cycle heat rate using iterative heat balance calculations. Use of such a program should be considered.

e. Cogeneration performance. There is no generally accepted method of defining the energy efficiency or heat rates of cogeneration cycles. Various methods are used, and any rational method is valid. The difference in value (per Btu) between prime energy (i.e., electric power) and secondary or low level energy (heating steam) should be recognized. Refer to discussion of cogeneration cycles below.

3-4. Cogeneration cycles

a. Definition. In steam power plant practice, cogeneration normally describes an arrangement whereby high pressure steam is passed through a turbine prime mover to produce electrical power, and thence from the turbine exhaust (or extraction) opening to a lower pressure steam (or heat) distribution system for general heating, refrigeration, or process use.

b. Common medium. Steam power cycles are particularly applicable to cogeneration situations because the actual cycle medium, steam, is also a convenient medium for area distribution of heat.

(1) The choice of the steam distribution pressure will be a balance between the costs of distribution which are slightly lower at high pressure, and the gain in electrical power output by selection of a lower turbine exhaust or extraction pressure.

(2) Often the early selection of a relatively low

Table 3-1. Theoretical Steam Rates for Typical Steam

Exhaust pressure	Initial pressure, lb/in² gage															
	150	250	400	600	600	850	850	9(	900	1,200	.250	1,250	'1,450	1,450	,800	2,400
	Initial temp °F															
	365.9	500	650	750	825	825	900	825	900	825	900	950	825	950	1000	1000
	Initial superheat, °I															
	0	94.0	201.9	261.2	336.2	297.8	372.8	291.1	366.	256.3	326.	376.	232.0	357.0	377.9	3 7.0
	nirial ha Btu/lb															
	,195.5	1,261.8	,334.9	,379.6	1,421.4	1,410.6	,453.5	1,408.4	,45 .6	1,394.7	1,438.4	46	,382.7	1,461.2	1,480.1	1,460.4
in Hg abs																
2.0	10.52	9.070	7.831	7.083	6.761	6.580	6.282	6.555	6.256	6.451	6.133	5.944	6.408	5.900	5.668	5.633
2.5	10.88	9.343	8.037	7.251	6.916	6.723	6.415	6.696	6.388	6.584	6.256	6.061	6.536	6.014	5.773	5.733
3.0	11.20	9.582	8.217	7.396	7.052	6.847	6.530	6.819	6.502	6.699	6.362	6.162	6.648	6.112	5.862	5.819
4.0	11.76	9.996	8.524	7.644	7.282	7.058	6.726	7.026	6.694	6.894	6.541	6.332	6.835	6.277	6.013	5.963
lb/in² gage																
5	21.69	16.57	13.01	11.05	10.42	9.838	9.288	9.755	9.209	9.397	8.820	8.491	9.218	8.351	7.874	7.713
10	23.97	17.90	13.83	11.64	10.95	10.30	9.705	10.202	9.617	9.797	9.180	8.830	9.593	8.673	8.158	7.975
20	28.63	20.44	15.33	12.68	11.90	11.10	10.43	10.982	10.327	10.490.	9.801	9.415	10.240	9.227	8.642	8.421
30	33.69	22.95	16.73	13.63	12.75	11.80	11.08	11.67	10.952	11.095	10.341	9.922	10.801	9.704	9.057	8.799
40	39.39	25.52	18.08	14.51	13.54	12.46	11.66	12.304	11.52	11.646	10.831	10.380	11.309	10.134	9.427	9.136
50	46.00	28.21	19.42	15.36	14.30	13.07	12.22	12.90	12.06	12.16	11.284	10.804	11.779	10.531	9.767	9.442
60	53.90	31.07	20.76	16.18	15.05	13.66	12.74	13.47	12.57	12.64	11.71	11.20	12.22	10.90	10.08	9.727
75	69.4	35.77	22.81	17.40	16.16	14.50	13.51	14.28	13.30	13.34	12.32	11.77	12.85	11.43	10.53	10.12
80	75.9	37.47	23.51	17.80	16.54	14.78	13.77	14.55	13.55	13.56	12.52	11.95	13.05	11.60	10.67	10.25
100		45.21	26.46	19.43	18.05	15.86	14.77	15.59	14.50	14.42	13.27	12.65	13.83	12.24	11.21	10.73
125		57.88	30.59	21.56	20.03	17.22	16.04	16.87	15.70	15.46	14.17	13.51	14.76	13.01	11.84	11.28
150		76.5	35.40	23.83	22.14	18.61	17.33	18.18	16.91	16.47	15.06	14.35	15.65	13.75	12.44	11.80
160		86.8	37.57	24.79	23.03	19.17	17.85	18.71	17.41	16.88	15.41	14.69	16.00	14.05	12.68	12.00
175			41.16	26.29	24.43	20.04	18.66	19.52	18.16	17.48	15.94	15.20	16.52	14.49	13.03	12.29
200			48.24	29.00	26.95	21.53	20.05	20.91	19.45	18.48	16.84	16.05	17.39	15.23	13.62	12.77
250			69.1	35.40	32.89	24.78	23.08	23.90	22.24	20.57	18.68	17.81	19.11	16.73	14.78	13.69
300				43.72	40.62	28.50	26.53	27.27	25.37	22.79	20.62	19.66	20.89	18.28	15.95	14.59
400				72.2	67.0	38.05	35.43	35.71	33.22	27.82	24.99	23.82	24.74	21.64	18.39	16.41
425				84.2	78.3	41.08	38.26	38.33	35.65	29.24	26.21	24.98	25.78	22.55	19.03	16.87
600						78.5	73.1	68.11	63.4	42.10	37.03	35.30	34.50	30.16	24.06	20.29

THE MATERIAL IN THIS TABLE IS ADAPTED FROM THEORETICAL STEAM RATE TABLES W TH THE PERM SSION OF THE PUBLISHER, THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

steam distribution pressure is easily accommodated in the design of distribution and utilization systems, whereas the hasty selection of a relatively high steam distribution pressure may not be recognized as a distinct economic penalty on the steam power plant cycle.

(3) Hot water heat distribution may also be applicable as a district heating medium with the hot water being cooled in the utilization equipment and returned to the power plant for reheating in a heat exchange with exhaust (or extraction) steam.

c. Relative economy. When the exhaust (or extraction) steam from a cogeneration plant can be utilized for heating, refrigeration, or process purposes in reasonable phase with the required electric power load, there is a marked economy of fuel energy because the major condensing loss of the conventional steam power plant (Rankine) cycle is avoided. If a good balance can be attained, up to 75 percent of the total fuel energy can be utilized as compared with about 40 percent for the best and largest Rankine cycle plants and about 25 to 30 percent for small Rankine cycle systems.

d. Cycle types. The two major steam power cogeneration cycles, which may be combined in the same plant or establishment, are:

(1) *Back pressure cycle.* In this type of plant, the entire flow to the turbine is exhausted (or extracted) for heating steam use. This cycle is the more effective for heat economy and for relatively lower cost of turbine equipment, because the prime mover is smaller and simpler and requires no condenser and circulating water system. Back pressure turbine generators are limited in electrical output by the amount of exhaust steam required by the heat load and are often governed by the exhaust steam load. They, therefore, usually operate in electrical parallel with other generators.

(2) *Extraction-condensing cycles.* Where the electrical demand does not correspond to the heat demand, or where the electrical load must be carried at times of very low (or zero) heat demand, then condensing-controlled extraction steam turbine prime movers as shown in Figure 3-3 may be applicable. Such a turbine is arranged to carry a specified electrical capacity either by a simple condensing cycle or a combination of extraction and condensing. While very flexible, the extraction machine is relatively complicated, requires complete condensing and heat rejection equipment, and must always pass a critical minimum flow of steam to its condenser to cool the low pressure buckets.

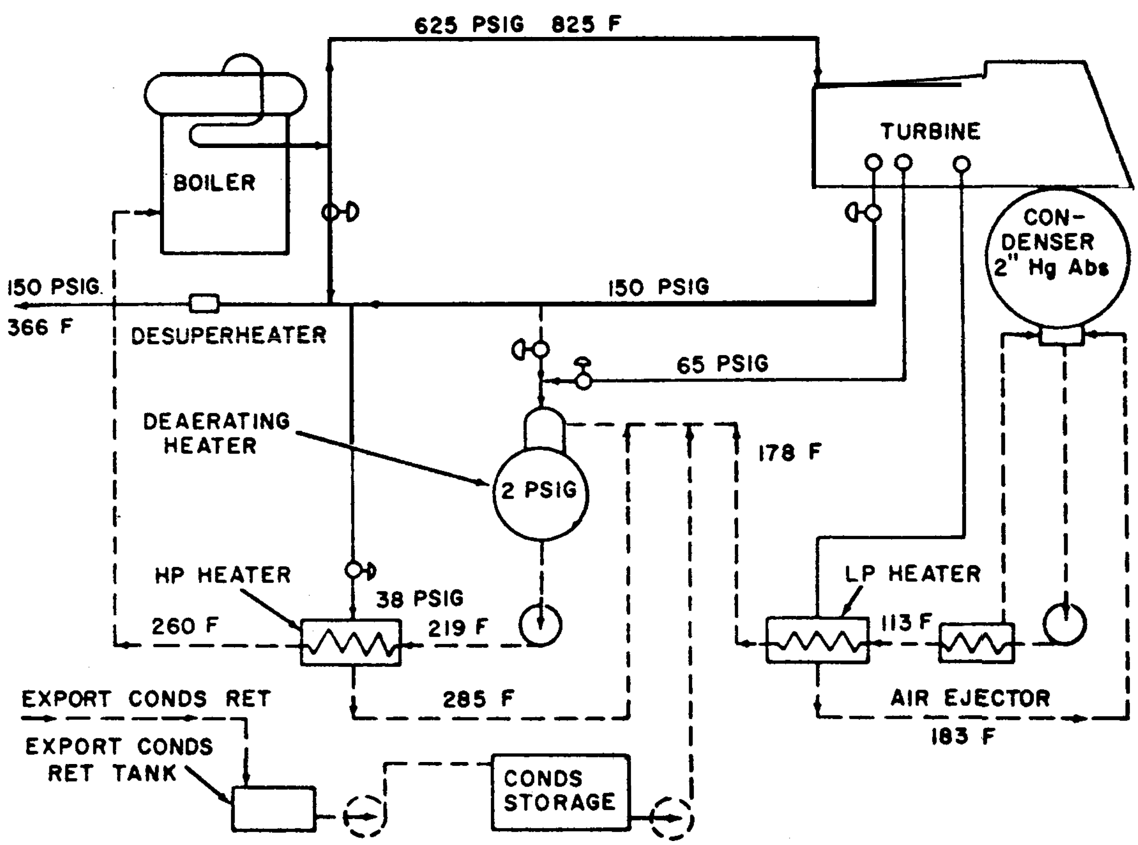

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Figure 3-3. Typical condensing-controlled extinction cycle.

e. Criteria for cogeneration. For minimum economic feasibility, cogeneration cycles will meet the following criteria:

(1) *Load balance.* There should be a reasonably balanced relationship between the peak and normal requirements for electric power and heat. The peak/normal ratio should not exceed 2:1.

(2) *Load coincidence.* There should be a fairly high coincidence, not less than 70%, of time and quantity demands for electrical power and heat.

(3) *Size.* While there is no absolute minimum size of steam power plant which can be built for cogeneration, a conventional steam (cogeneration) plant will be practical and economical only above some minimum size or capacity, below which other types of cogeneration, diesel or gas turbine become more economical and convenient.

(4) *Distribution medium.* Any cogeneration plant will be more effective and economical if the heat distribution medium is chosen at the lowest possible steam pressure or lowest possible hot water temperature. The power energy delivered by the turbine is highest when the exhaust steam pressure is lowest. Substantial cycle improvement can be made by selecting an exhaust steam pressure of 40 psig rather than 125 psig, for example. Hot water heat distribution will also be considered where practical or convenient, because hot water temperatures of 200 to 240° F can be delivered with exhaust steam pressure as low as 20 to 50 psig. The balance between distribution system and heat exchanger costs, and power cycle effectiveness will be optimized.

3-5. Selection of cycle steam conditions

a. Balanced costs and economy. For a new or isolated plant, the choice of initial steam conditions should be a balance between enhanced operating economy at higher pressures and temperatures, and generally lower first costs and less difficult operation at lower pressures and temperatures. Realistic projections of future fuel costs may tend to justify higher pressures and temperatures, but such factors as lower availability y, higher maintenance costs, more difficult operation, and more elaborate water treatment will also be considered.

b. Extension of existing plant. Where a new steam power plant is to be installed near an existing steam power or steam generation plant, careful consideration will be given to extending or paralleling the existing initial steam generating conditions. If existing steam generators are simply not usable in the new plant cycle, it may be appropriate to retire them or to retain them for emergency or standby service only. If boilers are retained for standby service only, steps will be taken in the project design for protection against internal corrosion.

c. *Special considerations.* Where the special circumstances of the establishment to be served are significant factors in power cycle selection, the following considerations may apply:

(1) *Electrical isolation.* Where the proposed plant is not to be interconnected with any local electric utility service, the selection of a simpler, lower pressure plant may be indicated for easier operation and better reliability y.

(2) *Geographic isolation.* Plants to be installed at great distances from sources of spare parts, maintenance services, and operating supplies may require special consideration of simplified cycles, redundant capacity and equipment, and highest practical reliability. Special maintenance tools and facilities may be required, the cost of which would be affected by the basic cycle design.

(3) *Weather conditions.* Plants to be installed under extreme weather conditions will require special consideration of weather protection, reliability, and redundancy. Heat rejection requires special design consideration in either very hot or very cold weather conditions. For arctic weather conditions, circulating hot water for the heat distribution medium has many advantages over steam, and the use of an antifreeze solution in lieu of pure water as a distribution medium should receive consideration.

3-6. Cycle equipment

a. General requirements. In addition to the prime movers, alternators, and steam generators, a complete power plant cycle includes a number of secondary elements which affect the economy and performance of the plant.

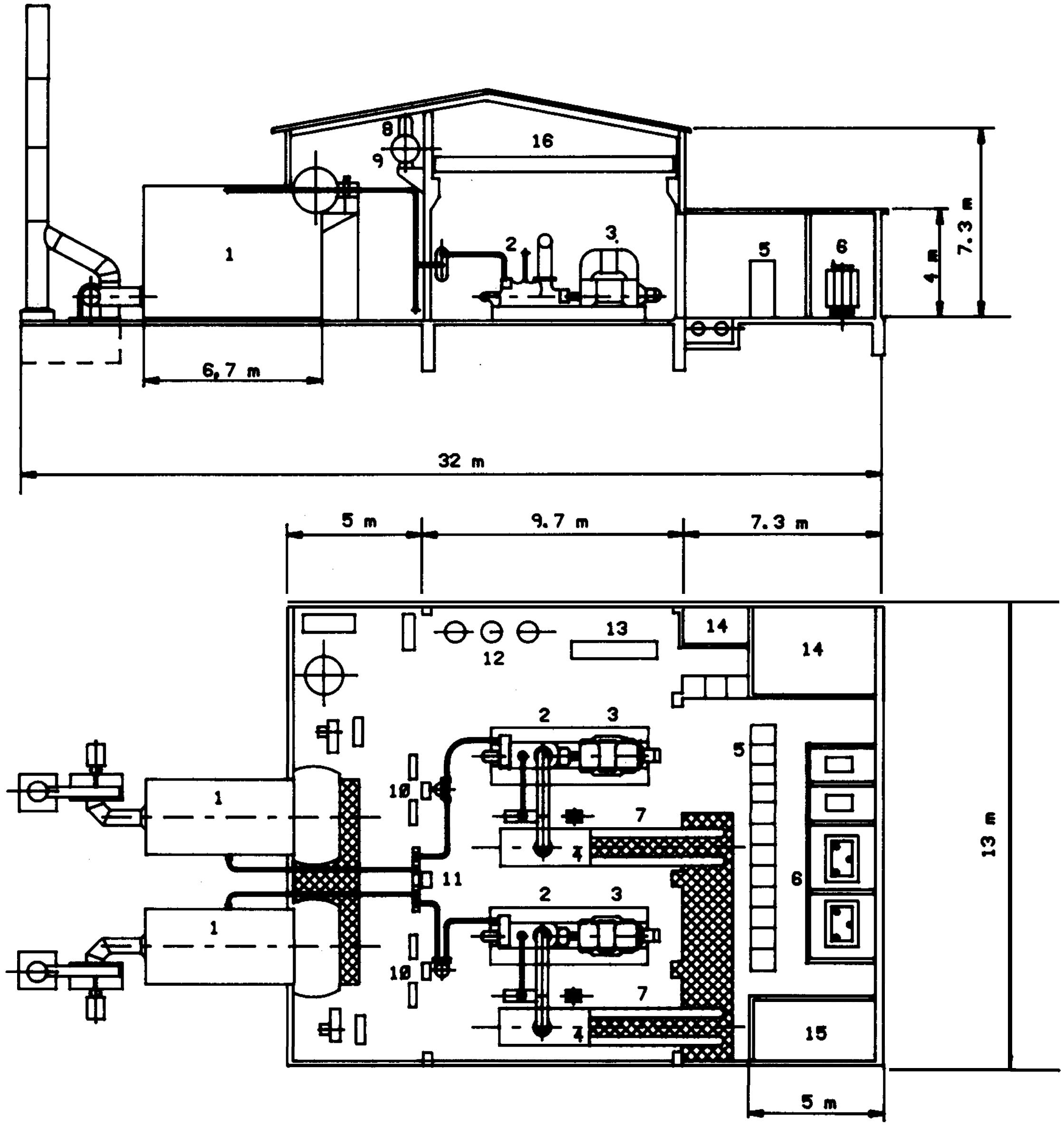
b. Major equipment. Refer to other parts of this manual for detailed information on steam turbine driven electric generators and steam generators.

c. Secondary cycle elements. Other equipment items affecting cycle performance, but subordinate to the steam generators and turbine generators, are also described in other parts of this chapter.

3-7. Steam power plant arrangement

a. General. Small units utilize the transverse arrangement in the turbine generator bay while the larger utility units are very long and require end-to-end arrangement of the turbine generators.

b. Typical small plants. Figures 3-4 and 3-6 show typical transverse small plant arrangements. Small units less than 5000 kW may have the condensers at the same level as the turbine generator for economy as shown in Figure 3-4. Figure 3-6 indicates the critical turbine room bay dimensions and the basic overall dimensions for the small power plants shown in Figure 3-5.



1 - BOILER, OIL-FIRED
2 - TURBINE
3 - GENERATOR
4 - CONDENSER
5 - SWITCHGEAR
6 - TRANSFORMERS
7 - COOLING WATER PIPES
8 - DEAERATOR
9 - FEEDWATER TANK
10 - BOILER FEED PUMPS
11 - STEAM MANIFOLD
12 - WATER TREATMENT
13 - START-UP DIESEL
14 - STORE ROOM
15 - OFFICE
16 - TURBINE ROOM CRANE

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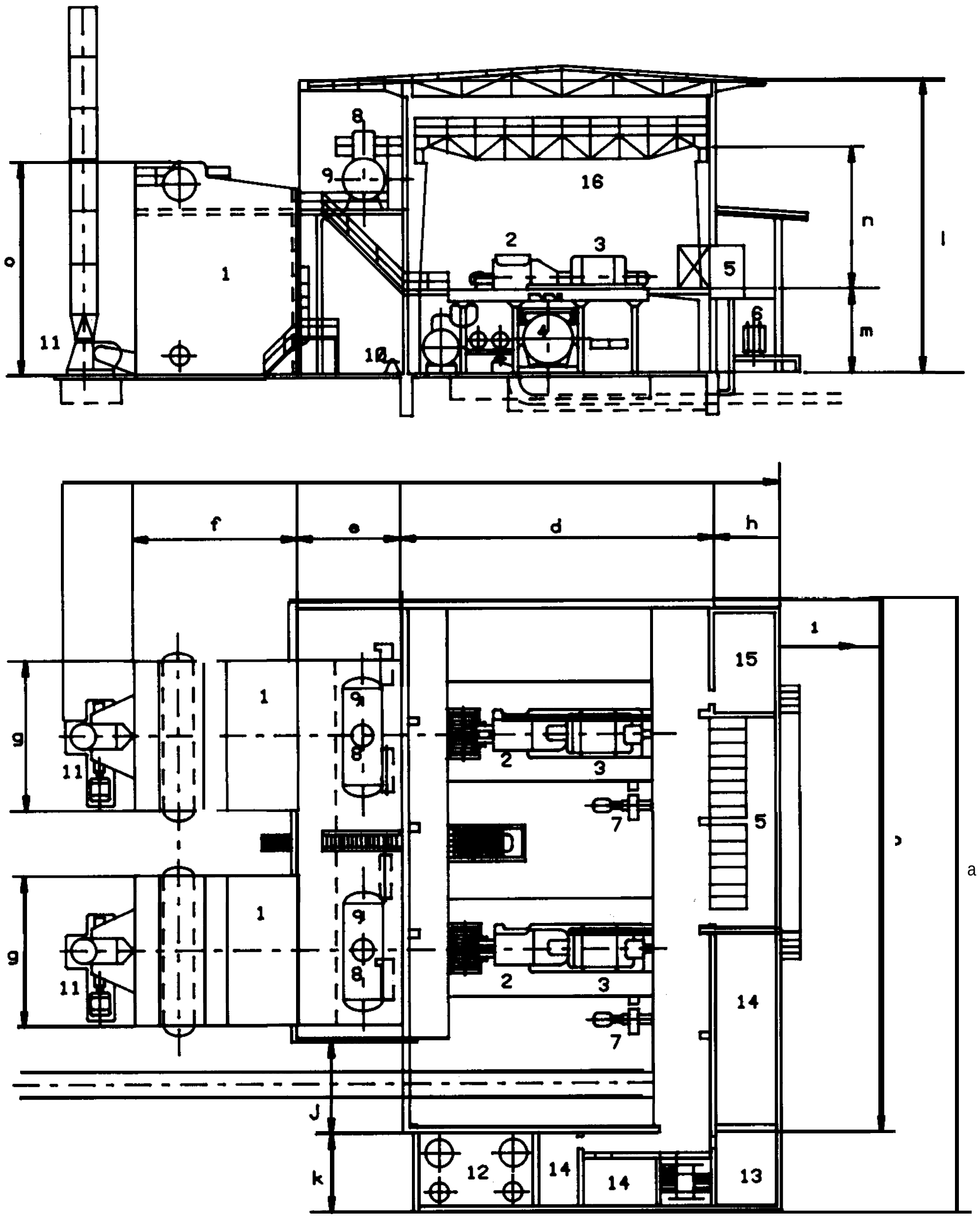
Figure 3-4. Typical small 2-unit powerplant “A”.



1 - BOILER, OIL-FIRED
2 - TURBINE
3 - GENERATOR
4 - CONDENSER
5 - SWITCHGEAR
6 - TRANSFORMERS
7 - COOLING WATER PUMP
8 - DEAERATOR
9 - FEEDWATER TANK
10 - BOILER FEED PUMPS
11 - INDUCED-DRAFT FAN
12 - WATER TREATMENT
13 - CHEMICAL LABORATORY
14 - OFFICES
15 - STORE ROOM
16 - TURBINE HOUSE CRANE

NOTE: SEE TABLE IN FIG. 3-6 FOR DIMENSIONS.

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Figure 3-5. Typical small 2-unit power plant "B".

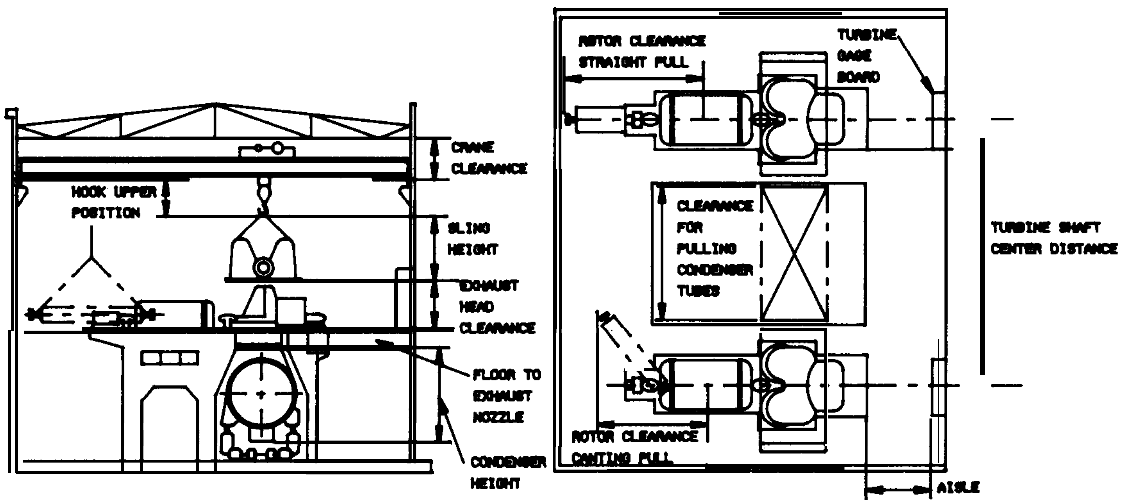
Section II. STEAM GENERATORS AND AUXILIARY SYSTEMS.

3-8. Steam generator conventional types and characteristics

a. Introduction. Number, size, and outlet steaming conditions of the steam generators will be as determined in planning studies and confirmed in the final project criteria prior to plant design activities. Note general criteria given in Section I of this chapter under discussion of typical plants and cycles.

b. Types and classes. Conventional steam generators for a steam power plant can be classified by type of fuel, by unit size, and by final steam condition. Units can also be classified by type of draft, by method of assembly, by degree of weather protection and by load factor application.

(1) *Fuel, general.* Type of fuel has a major impact on the general plant design in addition to the steam generator. Fuel selection may be dictated by considerations of policy and external circumstances



NOTE: ABOVE TURBINE ROOM BAY DIMENSIONS VARY ACCORDING TO TURBINE AND CONDENSER SUPPLIERS SELECTED.

DIMENSION	DIMENSIONS IN METERS			
	I	II	II	IV
	2x2.5 MW	2x4 MW	2x6.25 MW	2x12.5 MW
a	24.5	26.5	28	36
b	32	34	36	43
c	20	22	23	31
d	12	13	14	16
e	5	5	5	6
f	8.3	9	9.3	11.3
g	3.6	5	5.7	7.5
h	3.5	3.5	3.7	3.7
i	1.2	1.2	1.2	1.2
j	5.5	5.5	5.5	5.5
k	4.5	4.5	5	5
l	11	12	13	17.5
m	3.5	3.5	3.7	5
n	4.5	5	6	8
o	9	9	10	11

NOTE: DIMENSIONS IN TABLE ARE APPLICABLE TO FIG. 3-5

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Figure 3-6. Critical turbine room bay and power plant "B" dimensions.

unrelated to plant costs, convenience, or location. Units designed for solid fuels (coal, lignite, or solid waste) or designed for combinations of solid, liquid, and gaseous fuel are larger and more complex than units designed for fuel oil or fuel gas only.

(2) *Fuel coal.* The qualities or characteristics of particular coal fuels having significant impact on steam generator design and arrangement are: heating value, ash content, ash fusion temperature, friability, grindability, moisture, and volatile content as shown in Table 3-2. For spreader stoker firing, the size, gradation, or mixture of particle sizes affect stoker and grate selection, performance, and maintenance. For pulverized coal firing, grindability is a major consideration, and moisture content before and after local preparation must be considered. Coal burning equipment and related parts of the steam generator will be specified to match the specific characteristics of a preselected coal fuel as well as they can be determined at the time of design.

(3) *Unit sizes.* Larger numbers of smaller steam generators will tend to improve plant reliability and flexibility for maintenance. Smaller numbers of larger steam generators will result in lower first costs

Table 3-2. Fuel Characteristcs.

Characteristic	Effects
	Coal
Ultimate analysis	Heat balance.
Proximate analysis:	
Moisture	Handling and efficiency loss.
Volatile matter	Ignition and theoretical air.
Ash	Freight, storage, handling, air pollution.
Ash fusion temp	Slagging, allowable heat release, allowable furnace exit gas temperature.
Heat value	Heat balance, fuel cost.
Friability	Handling and storage.
Grindability	Crushing and pulverizing.
Size consist	Crushing, segregation, and spreading over fuel bed.
Sulphur	Allowable temp. of metal contacting flue gas; removal from flue gas.
	Oil
Ultimate analysis	Heat balance.
Grade	Fuel cost.
Viscosity	Preheating, pumping, firing.
Specific gravity	Pumping and metering.
Flash point	Vapor locking of pump suction.
Heat value	Heat balance, fuel cost.
Sulphur	Allowable temp. of metal contacting flue gas; removal from flue gas.
	Gas
Ultimate analysis	Heat balance.
Type	Pressure, firing, fuel cost.
Specific gravity	Metering.
Heat value	Heat balance, fuel cost.
Sulphur	Insignificant.

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per unit of capacity and may permit the use of design features and arrangements not available on smaller units. Larger units are inherently more efficient, and will normally have more efficient draft fans, better steam temperature control, and better control of steam solids.

(4) *Final steam conditions.* Desired pressure and temperature of the superheater outlet steam (and to a lesser extent feedwater temperature) will have a marked effect on the design and cost of a steam generator. The higher the pressure the heavier the pressure parts, and the higher the steam temperature the greater the superheater surface area and the more costly the tube material. In addition to this, however, boiler natural circulation problems increase with higher pressures because the densities of the saturated water and steam approach each other. In consequence, higher pressure boilers require more height and generally are of different design than boilers of 200 psig and less as used for general space heating and process application.

(5) *Type of draft.*

(a) Balanced draft. Steam generators for electric generating stations are usually of the so called "balanced draft" type with both forced and induced draft fans. This type of draft system uses one or more forced draft fans to supply combustion air under pressure to the burners (or under the grate) and one or more induced draft fans to carry the hot combustion gases from the furnace to the atmosphere; a slightly negative pressure is maintained in the furnace by the induced draft fans so that any gas leakage will be into rather than out of the furnace. Natural draft will be utilized to take care of the chimney or stack resistance while the remainder of the draft friction from the furnace to the chimney entrance is handled by the induced draft fans.

(b) Choice of draft. Except for special cases such as for an overseas power plant in low cost fuel areas, balanced draft, steam generators will be specified for steam electric generating stations.

(6) *Method of assembly.* A major division of steam generators is made between packaged or factory assembled units and larger field erected units. Factory assembled units are usually designed for convenient shipment by railroad or motor truck, complete with pressure parts, supporting structure, and enclosure in one or a few assemblies. These units are characteristically bottom supported, while the larger and more complex power steam generators are field erected, usually top supported.

(7) *Degree of weather protection.* For all types and sizes of steam generators, a choice must be made between indoor, outdoor and semi-outdoor installation. An outdoor installation is usually less expensive in first cost which permits a reduced general building construction costs. Aesthetic, environmental, or weather conditions may require indoor installation, although outdoors units have been used successfully in a variety of cold or otherwise hostile climates. In climates subject to cold weather, 30 "F. for 7 continuous days, outdoor units will require electrically or steam traced piping and appurtenances to prevent freezing. The firing aisle will be enclosed either as part of the main power plant building or as a separate weather protected enclosure; and the ends of the steam drum and retractable soot blowers will be enclosed and heated for operator convenience and maintenance.

(8) *Load factor application.* As with all parts of the plant cycle, the load factor on which the steam generator is to be operated affects design and cost factors. Units with load factors exceeding 50% will be selected and designed for relatively higher efficiencies, and more conservative parameters for furnace volume, heat transfer surface, and numbers and types of auxiliaries. Plants with load factors less than 50% will be served by relatively less expensive, smaller and less durable equipment.

3-9. Other steam generator characteristics

a. Water tube and waterwell design. Power plant boilers will be of the water welled or water cooled furnace types, in which the entire interior surface of the furnace is lined with steam generating heating surface in the form of closely spaced tubes usually all welded together in a gas tight enclosure.

b. Superheated steam. Depending on manufacturer's design some power boilers are designed to deliver superheated steam because of the requirements of the steam power cycle. A certain portion of the total boiler heating surface is arranged to add superheat energy to the steam flow. In superheater design, a balance of radiant and convective superheat surfaces will provide a reasonable superheat characteristic. With high 'pressure - high temperature turbine generators, it is usually desirable to provide superheat controls to obtain a flat characteristic down to at least 50 to 60 percent of load. This is done by installing excess superheat surface and then attemperating by means of spray water at the higher loads. In some instances, boilers are designed to obtain superheat control by means of tilting burners which change the heat absorption pattern in the steam generator, although supplementary attemperation is also provided with such a control system.

c. Balanced heating surface and volumetric design parameters. Steam generator design requires adequate and reasonable amounts of heating surface and furnace volume for acceptable performance and longevity.

(1) *Evaporative heating surface.* For its rated capacity output, an adequate total of evaporative or steam generating heat transfer surface is required, which is usually a combination of furnace wall radiant surface and boiler convection surface. Balanced design will provide adequate but not excessive heat flux through such surfaces to insure effective circulation, steam generation and efficiency.

(2) *Superheater surface.* For the required heat transfer, temperature control and protection of metal parts, the superheater must be designed for a balance between total surface, total steam flow area, and relative exposure to radiant convection heat sources. Superheaters may be of the drainable or non-drainable types. Non-drainable types offer certain advantages of cost, simplicity, and arrangement, but are vulnerable to damage on startup. Therefore, units requiring frequent cycles of shutdown and startup operations should be considered for fully drainable superheaters. With some boiler designs this may not be possible.

(3) *Furnace volume.* For a given steam generator capacity rating, a larger furnace provides lower furnace temperatures, less probability of hot spots, and a lower heat flux through the larger furnace wall surface. Flame impingement and slagging, particularly with pulverized coal fuel, can be controlled or prevented with increased furnace size.

(4) *General criteria.* Steam generator design will specify conservative lower limits of total heating surface, furnace wall surface and furnace volume, as well as the limits of superheat temperature control range. Furnace volume and surfaces will be sized to insure trouble free operation.

(5) *Specific criteria.* Steam generator specifications set minimum requirements for Btu heat release per cubic foot of furnace volume, for Btu heat release per square foot of effective radiant heating surface and, in the case of spreader stokers, for Btu per square foot of grate. Such parameters are not set forth in this manual, however, because of the wide range of fuels which can affect these equipment design considerations. The establishment of arbitrary limitations which may handicap the geometry of furnace designs is inappropriate. Prior to setting furnace geometry parameters, and after the type and grade of fuel are established and the particular service conditions are determined, the power plant designer will consult boiler manufacturers to insure that steam generator specifications are capable of being met.

d. Single unit versus steam header system. For cogeneration plants, especially in isolated locations or for units of 10,000 kW and less, a parallel boiler or steam header system may be more reliable and more economical than unit operation. Where a group of steam turbine prime movers of different types; i.e., one back pressure unit plus one condensing/extraction unit are installed together, overall economy can be enhanced by a header (or parallel) boiler arrangement.

3-10. Steam generator special types

a. Circulation. Water tube boilers will be specified to be of natural circulation. The exception to this rule is for wasteheat boilers which frequently are a . . special type of extended surface heat exchanger designed for forced circulation.

b. Fludized bed combustion. The fluidized bed boiler has the ability to produce steam in an environmentally accepted manner in controlling the stack emission of sulfur oxides by absorption of sulfur in the fuel bed as well as nitrogen oxides because of its relatively low fire box temperature. The fluidized bed boiler is a viable alternative to a spreader stoker unit. A fluidized bed steam generator consists of a fluidized bed combustor with a more or less conventional steam generator which includes radiant and convection boiler heat transfer surfaces plus heat recovery equipment, draft fans, and the usual array of steam generator auxiliaries. A typical fluidized bed boiler is shown in Figure 3-7.

3-11. Major auxiliary systems.

a. Burners.

(1) *Oil burners.* Fuel oil is introduced through oil burners, which deliver finely divided or atomized liquid fuel in a suitable pattern for mixing with combustion air at the burner opening. Atomizing methods are classified as pressure or mechanical type, air atomizing and steam atomizing type. Pressure atomization is usually more economical but is also more complex and presents problems of control, poor turndown, operation and maintenance. The range of fuel flows obtainable is more limited with pressure atomization. Steam atomization is simple to operate, reliable, and has a wide range, but consumes a portion of the boiler steam output and adds moisture to the furnace gases. Generally, steam atomization will be used when makeup water is relatively inexpensive, and for smaller, lower pressure plants. Air atomization will be used for plants burning light liquid fuels, or when steam reacts adversely with the fuel, i.e., high sulfur oils.

(2) *Gas and coal burners.* Natural gas or pulverized coal will be delivered to the burner for mixing with combustion air supply at the burner opening. Pulverized coal will be delivered by heated, pressurized primary air.

(3) *Burner accessories.* Oil, gas and pulverized

coal burners will be equipped with adjustable air guide registers designed to control and shape the air flow into the furnace, Some burner designs also provide for automatic insertion and withdrawal of varying size oil burner nozzles as load and operating conditions require.

(4) *Number of burners.* The number of burners required is a function both of load requirements and boiler manufacturer design. For the former, the individual burner turndown ratios per burner are provided in Table 3-3. Turndown ratios in excess of those listed can be achieved through the use of multiple burners. Manufacturer design limits capacity of each burner to that compatible with furnace flame and gas flow patterns, exposure and damage to heating surfaces, and convenience of operation and control.

(5) *Burner managerment systems.* Plant safety practices require power plant fuel burners to be equipped with comprehensive burner control and safety systems to prevent unsafe or dangerous conditions which may lead to furnace explosions. The primary purpose of a burner management system is safety which is provided by interlocks, furnace purge cycles and fail safe devices.

b. Pulverizes. The pulverizers (mills) are an essential part of powdered coal burning equipment, and are usually located adjacent to the steam generator and burners, but in a position to receive coal by gravity from the coal silo. The coal pulverizers grind

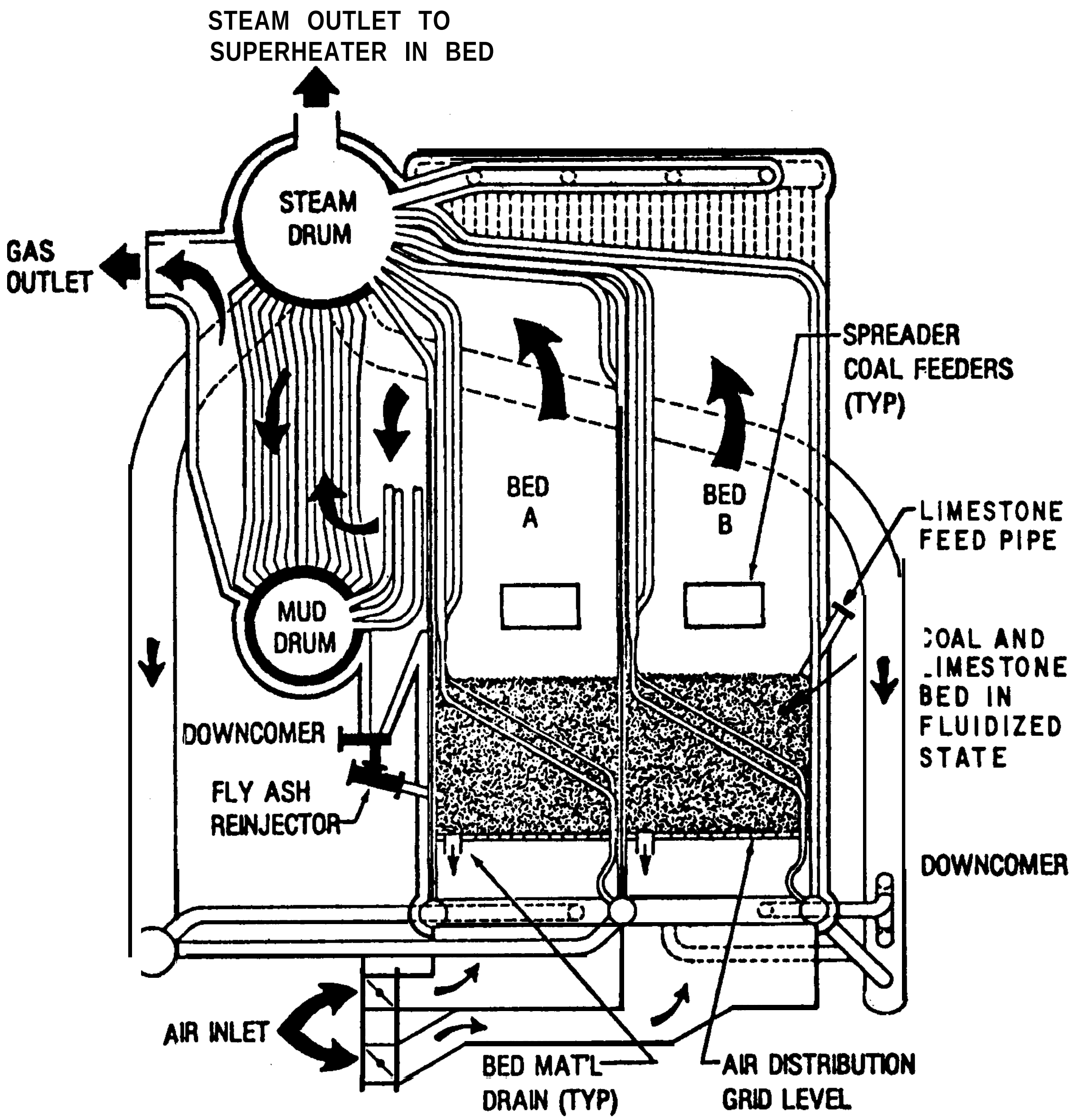

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Figure 3-7. Fluidized bed combustion boiler.

and classify the coal fuel to specific particle sizes for rapid and efficient burning. Reliable and safe pulverizing equipment is essential for steam generator operation. Pulverized coal burning will not be specified for boilers smaller than 150,000 lb/hour.

c. Stokers and grates. For small and medium sized coal burning steam generators, less than 150,000 lb/hour, coal stokers or fluidized bed units will be used. For power boilers, spreader stokers with traveling grates are used. Other types of stokers (retort, underfeed, or overfeed types) are generally obsolete for power plant use except perhaps for special fuels such as anthracite.

(1) Spreader stokers typically deliver sized coal, with some proportion of fines, by throwing it into the furnace where part of the fuel burns in suspension and the balance falls to the traveling grate for burnout. Stoker fired units will have two or more spreader feeder units, each delivering fuel to its own separate grate area. Stoker fired units are less responsive to load changes because a large proportion of the fuel burns on the grate for long time periods (minutes). Where the plant demand is expected to include sudden load changes, pulverized coal feeders are to be used.

(2) Grate operation requires close and skillful operator attention, and overall plant performance is sensitive to fuel sizing and operator experience. Grates for stoker fired units occupy a large part of the furnace floor and must be integrated with ash removal and handling systems. A high proportion of stoker ash must be removed from the grates in a wide range of particle sizes and characteristics although some unburned carbon and fly ash is carried out of the furnace by the flue gas. In contrast, a larger proportion of pulverized coal ash leaves the . furnace with the gas flow as finely divided particulate,

(3) Discharged ash is allowed to COOl in the ash hopper at the end of the grate and is then sometimes put through a clinker grinder prior to removal in the vacuum ash handling system described elsewhere in this manual.

d. Draft fans, ducts and flues.

(1) *Draft fans.*

(a) Air delivery to the furnace and flue gas re-

Table 3-3. Individual Burner Turndown Ratios.

Burner Type	Turndown Ratio
NATURAL GM	
Spud or Ring Type	5:1 to 10:1
HEAVY FUEL OIL	
Steam Atomizing	5:1 to 10:1
Mechanical Atomizing	3:1 to 10:1
COAL	
Pulverized	3:1
Spreader-Stoker	2:1 to 3:1
Fluidized Bed (single bed)	2:1 to 3:1

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moval will be provided by power driven draft fans designed for adequate volumes and pressures of air and gas flow. Typical theoretical air requirements are shown in Figure 3-8 to which must be added excess air which varies with type of firing, plus fan margins on both volumetric and pressure capacity for reliable full load operation. Oxygen and carbon dioxide in products of combustion for various amounts of excess air are also shown in Figure 3-8.

(b) Calculations of air and gas quantities and pressure drops are necessary. Since fans are heavy power consumers, for larger fans consideration should be given to the use of back pressure steam turbine drives for economy, reliability and their ability to provide speed variation. Multiple fans on each boiler unit will add to first costs but will provide more flexibility and reliability. Type of fan drives and number of fans will be considered for cost effectiveness. Fan speed will be conservatively selected, and silencers will be provided in those cases where noise by fans exceeds 80 decibels.

(c) Power plant steam generator units designed for coal or oil will use balanced draft design with both forced and induced draft fans arranged for closely controlled negative furnace pressure.

(2) *Ducts and flues.* Air ducts and gas flues will be adequate in size and structural strength and designed with provision for expansion, support, corrosion resistance and overall gas tightness. Adequate space and weight capacity will be allowed in overall plant arrangement to avoid awkward, noisy or marginal fan, duct and flue systems. Final steam generator design will insure that fan capacities (especially pressure) are matched properly to realistic air and gas path losses considering operation with dirty boilers and under abnormal operating conditions. Damper durability and control characteristics will be carefully designed; dampers used for control purposes will be of opposed blade construction.

e. Heat recovery. Overall design criteria require highest fuel efficiency for a power boiler; therefore, steam generators will be provided with heat recovery equipment of two principal types: air preheater and economizers.

(1) Efficiency effects. Both principal types of heat recovery equipment remove relatively low level heat from the flue gases prior to flue gas discharge to the atmosphere, using boiler fluid media (air or water) which can effectively absorb such low level energy. Such equipment adds to the cost, complexity and operational skills required, which will be balanced by the plant designer against the life cycle fuel savings.

(2) *Air preheater.* Simple tubular surface heaters will be specified for smaller units and the regenerative type heater for larger boilers. To mini-

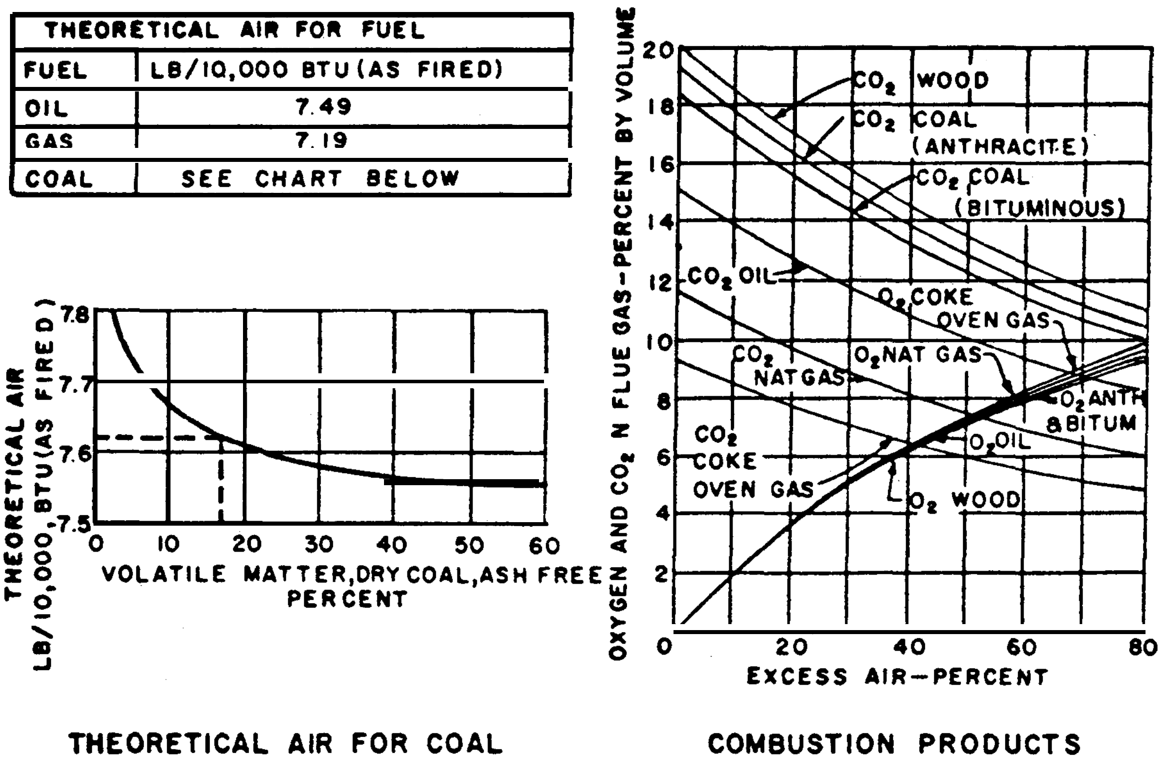

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Figure 3-8. Theoretical air and combustion products.

mize corrosion and acid/moisture damage, especially with dirty and high sulphur fuels, special alloy steel will be used in the low temperature heat transfer surface (replaceable tubes or "baskets") of air preheater. Steam coil air heaters will be installed to maintain certain minimum inlet air (and metal) temperatures and thus protect the main preheater from corrosion at low loads or low ambient air temperatures. Figure 3-9 illustrates the usual range of minimum metal temperatures for heat recovery equipment.

(3) *Economizers.* Either an economizer or an air heater or a balanced selection of both as is usual in a power boiler will be provided, allowing also for turbine cycle feedwater stage heating.

f. Stacks.

(1) Delivery of flue gases to the atmosphere through a flue gas stack or chimney will be provided.

(2) Stacks and chimneys will be designed to discharge their gases without adverse local effects. Dispersion patterns and considerations will be treated during design.

(3) Stacks and chimneys will be sized with due regard to natural draft and stack friction with

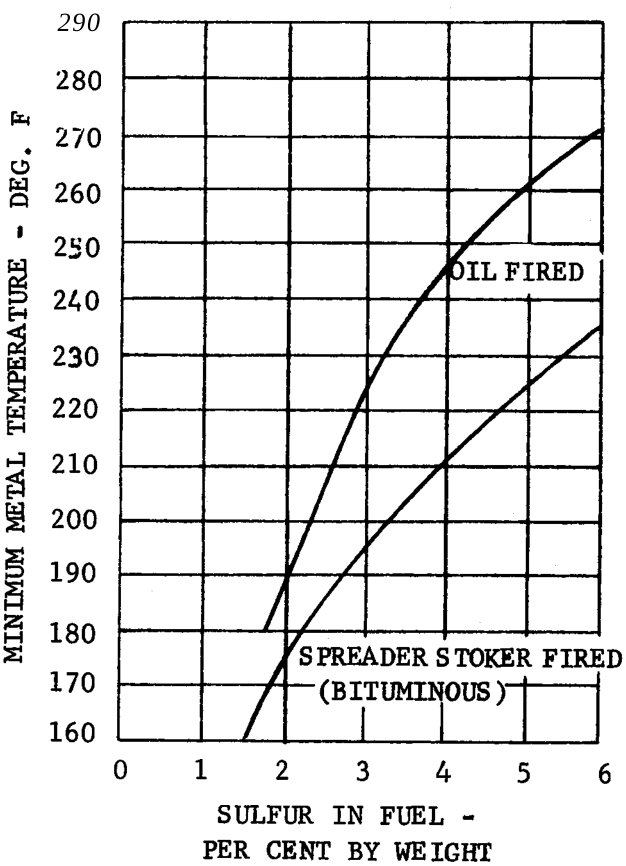

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Figure 3-9. Minimum metal temperatures for boiler heat recovery equipment.

height sometimes limited by aesthetic or other noneconomic considerations. Draft is a function of density difference between the hot stack gases and ambient air, and a number of formulas are available for calculating draft and friction. Utilize draft of the stack or chimney only to overcome friction within the chimney with the induced draft fan(s) supplying stack or chimney entrance. Maintain relatively high gas exit velocities (50 to 60 feet per second) to eject gases as high above ground level as possible. Reheat (usually by steam) will be provided if the gases are treated (and cooled) in a flue gas desulfurization scrubber prior to entering the stack to add buoyancy and prevent their settling to the ground after ejection to the atmosphere. Insure that downwash due to wind and building effects does not drive the flue gas to the ground.

g. Flue gas cleanup. The requirements for flue gas cleanup will be determined during design.

(1) *Design considerations.* The extent and nature of the air pollution problem will be analyzed prior to specifying the environmental control system for the steam generator. The system will meet all applicable requirements, and the application will be the most economically feasible method of accomplishment. All alternative solutions to the problem will be considered which will satisfy the given load and which will produce the least objectionable wastes. Plant design will be such as to accommodate future additions or modifications at minimum cost. Questions concerning unusual problems, unique applacations or marginal and future requirements will be directed to the design agency having jurisdiction over the project. Table 3-4 shows the emission levels allowable under the National Ambient Air Quality Standards.

(2) *Particulate control.* Removal of flue gas particulate material is broadly divided into mechanical dust collectors, electrostatic precipitators, bag filters, and gas scrubbing systems. For power plants of the size range here considered estimated uncontrolled emission levels of various pollutants are shown in Table 3-5. Environmental regulations require control of particulate, sulfur oxides and nitrogen oxides. For reference purposes in this manual, typical control equipment performance is shown in Table 3-6, 3-7, 3-8, 3-9, 3-10 and 3-11. These only provide general guidance. The designer will refer to TM 5-815-l/AFR 19-6/NAVFAC DM-3.15 for details of this equipment and related computational requirements and design criteria.

(a) *Mechanical collectors.* For oil fired steam generators with output steaming capacities less than 200,000 pounds per hour, mechanical (centrifugal) type dust collectors may be effective and economical depending on the applicable emission standards. For a coal fired boiler with a spreader stoker, a mechanical collector in series with an electrostatic precipitator or baghouse also might be considered. Performance requirements and technical environmental standards must be carefully matched, and ultimate performance warranties and tests require careful and explicit definitions. Collected dust from a mechanical collector containing a large proportion of combustibles may be reinfected into the furnace for final burnout; this will increase steam generator efficiency slightly but also will increase collector dust loading and carryover. Ultimate collected dust material must be handled and disposed of systematically to avoid objectionable environmental effects.

(b) Electrostatic *precipitators.* For pulverized coal firing, adequate particulate control will require electrostatic precipitators (ESP). ESP systems are well developed and effective, but add substantial capital and maintenance costs. Very high percent-

CONTAMINANT	1-YEAR	24-HOURS(1)	MAXIMUM CONCENTRATION AVERAGED OVER (1) 8-HOURS	3-HOURS	1-HOUR
Sulfur dioxide as SO_2					
primary	80 ug/m^3	80 ug/m^3	—	—	—
secondary	—	—	—	300 ug/m^3	—
Particulates					
primary	75 ug/m^3	260 ug/m^3	—	—	—
secondary	60 ug/m^3	150 ug/m^3	—	—	—
Carbon monoxide primary and secondary	—	—	10 mg/m^3	—	40 mg/m^3
Photochemical oxidants primary and secondary		—	—	—	40 mg/m^3
Hydrocarbons primary and secondary		—	—	160 ug/m^3	—
Nitrogen dioxide primary and secondary	100 ug/m^3	—	—	—	—
Lead Primary and secondary	1.5 ug/m^3(2)	—	—	—	—

Notes: (1) Not to be exceeded more than once per year.
(2) Averaged over a calendar quarter.

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Table 3-4. Emission Levels Allowable, National Ambient Air Quality Standards.

Table3-5. Uncontrolled Emissions.

	COAL FIRED (Lb of Pollutant/Ton of Coal)		OIL FIRED (Lb of Pollutant/1000 Gal)		NATURAL GAS (Lb of Pollutant/10^6 Ft^3)
Pollutant	Pulverized Coal Boilers	Stokers or FBC Boilers	Residual Oil	Distillate Oil	
Particulate	16A[1]	13A[2]	$10S^3$ + 3	2	5-15
Sulfur Oxides	$38S^3$	$38S^3$	$159S^3$	$144S^3$	0.6
Nitrogen Oxides	18	15	60		

1. The letter A indicates that the weight percentage of ash in the coal should be multiplied by the value given. Example: If the factor is 16 and the ash content is 10 percent, the particulate emissions before the control equipment would be 10 times 16, or 160 pounds of particulate per ton of coal.

2. Without fly ash reinfection. With fly ash reinfection use 20A.

3. S equals the sulfur content, use like the factor A (see Note 1 above) for estimate emissions.

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Table 3-6. *Characteristics of Cyclones for Particulate Control.*

Type	Body Diameter (Feet)	Gas Flow Ft /min	Pressure Drop (In. H O)	Inlet Velocity Ft/Sec	Collection Efficiency (%)	Application	Other
Conventional	4-12	1,000-20,000	.5-2	20-70	50-80	Material handling. Exhaust gas pre-cleaner.	Large headroom require-ments. Limited to large coarse particles large grain loadings.
High efficiency single cyclone	Less than 3	100-2,000	2-6	50-70	80-85	Industrial boiler particulate control.	Smaller space require-ment. Parallel arrangement, inlet vane flow controls needed continuous dust removal system purge operation.
Multicyclones	.5-1	30,000-100,000	3-6	50-70	90-95	Industrial and utility boiler Particulate control.	Plenums required. Problems: gas recir-culation fouling. Continuous dust removal system, flow control.
Irrigated cyclone (wet) high effi-ciency single unit	less than 3	100-2,000	2-6	50	90-95	Boiler application (low sulfur fuel) (low gas temp.).	Water rate 5-15 gal/ 1,000 ft^3/min. Corrosion resistant materials.

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Table 3-2! Characteristics of Scrubbers for Particulate Control.

Scrubber Type	Energy Type	Pressure Drop In. H O	Gas Flow Ft /Min	Internal Velocity Ft/Sec	Particle Collection Efficiency	Water Usage Per 1000 Gal/Min
Centrifugal Scrubber	Low Energy	3-8	1,000-20,000	50-150	80	3-5
Impingement & Entrainment	Low Energy	4-20	500-50,000	50-150	60-90	10-40
Venturi	High Energy	4-200	200-150,000	200-600	95-99	5-7
Ejector Venturi	High Energy	10-50	500-10,000	200-500	90-98	70-145

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Table 3-8. Characteristics of Electrostatic Precipitators (ESP) for Particulate Control.

Type	Operating Temperature °F	Resistivity at 300° F ohm-cm	Gas Flow Ft/Min	Pressure Drop In. of Water	
Hot ESP	600+	Greater Than 10^{12}	100,000+	Less Than 1"	
Cold ESP	300	Less Than 10^{10}			
Wet ESP	300-	Greater Than 10^{12} below 10^{4}			

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Table 3-9. Characteristics of Baghouses for Particulate Control.

System Type	Pressure Loss (Inches of Water)	Efficiency	Cloth Type	Filter Ratio (cfm/ft Cloth Area)	Recommended Application
Shaker	3-6	99+%	Woven	1-5	Dust with good filter cleaning properties, intermittent collection.
Reverse Flow	3-6	99+%	Woven	1-5	Dust with good filter cleaning properties, high temperature collection (incinerator fly-ash) with glass bags.
Pulse Jet	3-6	99+%	Felted	4-20	Efficient for coal and oil fly ash collection.
Reverse Jet	3-8	99+%	Felted	10-30	Collection of fine dusts and fumes.
Envelope	3-6	99+%	Woven	1-5	Collection of highly abrasive dust .

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Table 3-10. Characteristics of Flue-Gas Desulfurization Systems for Particulate Control.

System Type	SO Removal Efficiency (%)	Pressure Drop (Inches of Water)	Recovery and Regeneration	Operational Reliability	Retrofit to Existing Installations
1) Limestone Boiler Injection Type	30-40%	Less Than 6“	No Recovery of Limestone	High	Yea
2) Limestone, Srubber Injection Type	30-40%	Greater Than 6“	No Recovery of Lime	High	Yea
3) Lime, Scrubber, Injection Type	90%+	Greater Than 6“	No Recovery of Lime	Low	Yea
4) Magnesium Oxide	90%+	Greater Than 6“	Recovery of MgO and Sulfuric Acid	Low	Yea
5) Wellman-Lord	90%+	Greater Than 6“	Recovery of $NaS0_3$ and Elemental Sulfur	Unknown	
6) Catalytic oxidation	85%	May be as high as 24”	Recovery of 80% H_2S0_4	Unknown	No
7) Single Alkali Systems	90%+	Tray Tower Pressure Drop 1.6-2.0 in. H_2O/tray, w/Venturi add 10-14 in. H_2O	Little Recovery of Sodium Carbonate	Unknown	Yea
8) Dual Alkali	90-95%+		Regeneration of Sodium Hydroxide and Sodium Sulfites	Unknown	Yea

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Tabble 3-11. Techniques for Nitrogen Oxide Control.

Technique	Potential NO Reduction (%)	Advantages	Disadvantages
Load Reduction		Easily implemented; no additional equipment required; reduced particulate and SO_x emissions.	Reduction in generating capacity; possible reduction in boiler thermal thermal efficiency.
Low Excess Air Firing	15 to 40	Increased boiler thermal efficiency; possible reduction in particulate emissions may be combined with a load reduction to obtain additional NO_x emission decrease; reduction in high temperature corrosion and ash deposition.	A combustion control system which closely monitors and controls fuel/air ratios is required.
Two Stage Conbustion			
Coal	30	---	Boiler windboxes must be designed for this application.
Oil	40	---	Furnace corrosion and particulate emissions may increase.
Gas	50	- --	
Off-Stoichiometric Combustion			Control of alternate fuel rich/and fuel lean burners may be a problem during transient load conditions.
Coal	45	---	
Reduced Combustion Air Preheat	10-50	---	Not applicable to coal or oil fired units; reduction in boiler thermal efficiency; increase in exit gas volume and temperature; reduction in boiler load.
Flue Gas Recirculation	20-50	Possible improvement in combustion efficiency end reduction in particulate emissions.	Boiler windbox must be modified to handle the additional gas volume; ductwork, fans and Controls required.

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ages of particulate removal can be attained (99 percent, plus) but precipitators are sensitive to ash composition, fuel additives, flue gas temperatures and moisture content, and even weather conditions. ESP's are frequently used with and ahead of flue gas washing and desulfurization systems. They may be either hot precipitators ahead of the air preheater in the gas path or cold precipitators after the air preheater. Hot precipitators are more expensive because of the larger volume of gas to be handled and temperature influence on materials. But they are sometimes necessary for low sulfur fuels where cold precipitators are relatively inefficient.

(c) Bag filters. Effective particulate removal may be obtained with bag filter systems or bag houses, which mechanically filter the gas by passage through specially designed filter fabric surfaces. Bag filters are especially effective on very fine particles, and at relatively low flue gas temperatures. They may be used to improve or upgrade other particulate collection systems such as centrifugal collectors. Also they are probably the most economic choice for most medium and small size coal fired steam generators.

(d) Flue gas desulfurization. While various gaseous pollutants are subject to environmental control and limitation, the pollutants which must be removed from the power plant flue gases are the oxides of sulfur (SO_2 and SO_3). Many flue gas desulfuriztion (FGD) scrubbing systems to control SO_2 and SO_3 stack emission have been installed and operated, with wide variations in effectiveness, reliability, longevity and cost. For small or medium sized power plants, FGD systems should be avoided if possible by the use of low sulfur fuel. If the parameters of the project indicate that a FGD system is required, adequate allowances for redundancy, capital cost, operating costs, space, and environmental impact will be made. Alternatively, a fluidized bed boiler (para. 3-10 c) may be a better economic choice for such a project.

(1) Wet scrubbers utilize either limestone, lime, or a combination of lime and soda ash as sorbents for the SO_2 and SO_3 in the boiler flue gas stream. A mixed slurry of the sorbent material is sprayed into the flue gas duct where it mixes with and wets the particulate in the gas stream. The $S0_2$ and $S0_9$ reacts with the calcium hydroxide of the slurry to form calcium sulfate. The gas then continues to a separator tower where the solids and excess solution settle and separate from the water vapor saturated gas stream which vents to the atmosphere through the boiler stack. Wet scrubbers permit the use of coal with a sulfur content as high as 5 percent.

(2) Dry scrubbers generally utilize a diluted solution of slaked lime slurry which is atomized by compressed air and injected into the boiler flue gas stream. SO_2 and SO_3 in the flue gas is absorbed by the slurry droplets and reacts with the calcium hydroxide of the slurry to form calcium sulfite. Evaporation of the water in the slurry droplets occurs simultaneously with the reaction. The dry flue gas then travels to a bag filter system and then to the boiler stack. The bag filter system collects the boiler exit solid particles and the dried reaction products. Additional remaining SO_2 and SO_3 are removed by the flue gas filtering through the accumulation on the surface of the bag filters, Dry scrubbers permit the use of coal with a sulfur content as high as 3 percent.

(3) *Induced draft fan requirements.* Induced draft fans will be designed with sufficient capacity to produce the required flow while overcoming the static pressure losses associated with the ductwork, economizer, air preheater, and air pollution control equipment under all operating (clean and dirt y) conditions.

(4) *Waste removal.* Flue gas cleanup systems usually produce substantial quantities of waste products, often much greater in mass than the substances actually removed from the exit gases. Design and arrangement must allow for dewatering and stabilization of FGD sludge, removal, storage and disposal of waste products with due regard for environmental impacts.

3-12. Minor auxiliary systems

Various minor auxiliary systems and components are vital parts of the steam generator.

a. Piping and valves. Various piping systems are defined as parts of the complete boiler (refer to the ASME Boiler Code), and must be designed for safe and effective service; this includes steam and feedwater piping, fuel piping, blowdown piping, safety and control valve piping, isolation valves, drips, drains and instrument connections.

b. Controls and instruments. Superheater and 'burner management controls are best purchased along with the steam generator so that there will be integrated steam temperature and burner systems.

c. Soot blowers. Continuous or frequent on line cleaning of furnace, boiler economizer, and air preheater heating surfaces is required to maintain performance and efficiency. Soot blower systems, steam or air operated, will be provided for this purpose. The selection of steam *or* air for soot blowing is an economic choice and will be evaluated in terms of steam and makeup water vs. compressed air costs with due allowance for capital and operating cost components.

Section III. FUEL HANDLING AND STORAGE SYSTEMS

3-13. Introduction

a. Purpose. Figure 3-10 is a block diagram illustrating the various steps and equipment required for a solid fuel storage and handling system.

b. Fuels for consideration. Equipment required for a system depends on the type of fuel or fuels burned. The three major types of fuels utilized for steam raising are gaseous, liquid and solid.

3-14. Typical fuel oil storage and handling system

The usual power plant fuel oil storage and handling system includes:

a. Unloading and storage.

(1) Unloading pumps will be supplied, as required for the type of delivery system used, as part of the power plant facilities. Time for unloading will be analyzed and unloading pump(s) optimized for the circumstances and oil quantities involved. Heavier fuel oils are loaded into transport tanks hot and cool during delivery. Steam supply for tank car heaters will be provided at the plant if it is expected that the temperature of the oil delivered will be below the 120 to 150°F. range.

(2) Storage of the fuel oil will be in two tanks so as to provide more versatility for tank cleanout inspection and repair. A minimum of 30 days storage capacity at maximum expected power plant load (maximum steaming capacity of all boilers with maximum expected turbine generator output and maximum export steam, if any) will be provided. Factors such as reliability of supply and whether

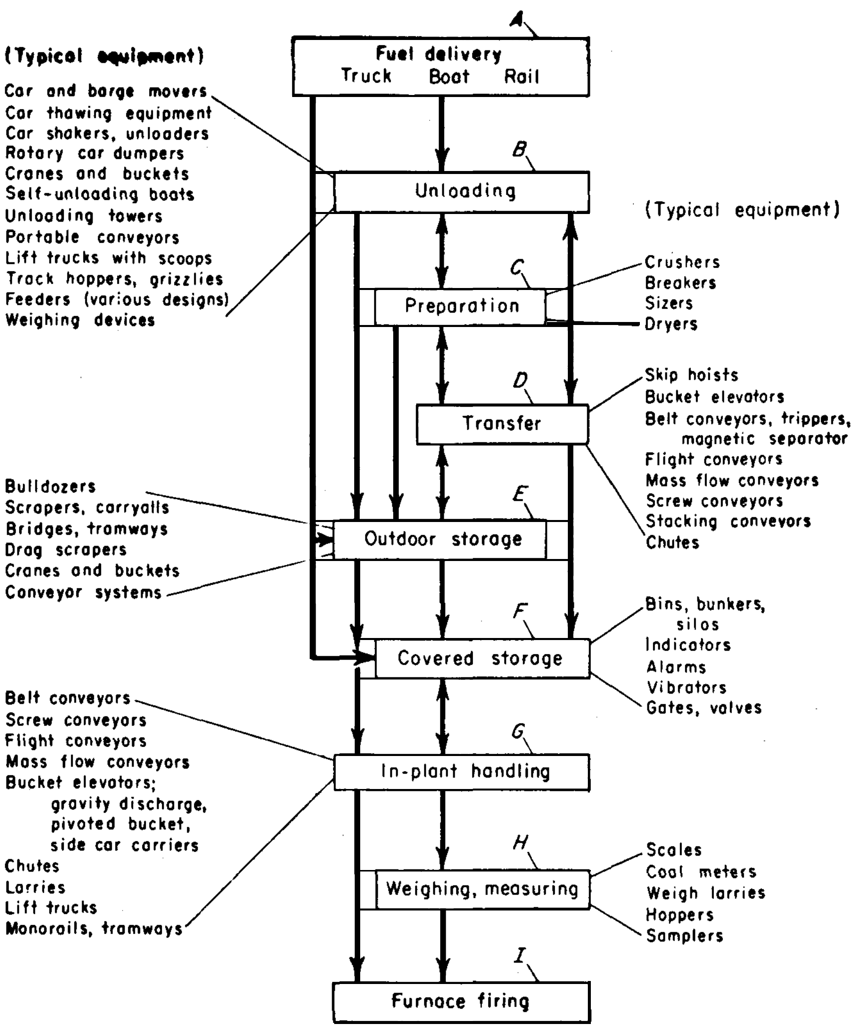

Figure 3-10. Coal handling system diagram.

backup power is available from other sources may result in additional storage requirements. Space for future tanks will be allocated where additional boilers are planned, but storage capacity will not be provided initially.

(3) Storage tank(s) for heavy oils will be heated with a suction type heater, a continuous coil extending over the bottom of the tank, or a combination of both types of surfaces. Steam is usually the most economical heating medium although hot water can be considered depending on the temperatures at which low level heat is available in the power plant. Tank exterior insulation will be provided.

b. Fuelpumps and heaters.

(1) Fuel oil forwarding pumps to transfer oil from bulk storage to the burner pumps will be provided. Both forwarding and burner pumps should be selected with at least 10 percent excess capacity over maximum burning rate in the boilers. Sizing will consider additional pumps for future boilers and pressure requirements will be selected for pipe friction, control valves, heater pressure drops, and burners. A reasonable selection would be one pump per boiler with a common spare if the system is designed for a common supply to all boilers. For high pressure mechanical atomizing burners, each boiler may also have its own metering pump with spare.

(2) Pumps may be either centrifugal or positive displacement. Positive displacement pumps will be specified for the heavier fuel oils. Centrifugal pumps will be specified for crude oils. Where absolute reliaability is required, a spare pump driven by a steam turbine with gear reducer will be used. For "black starts, " or where a steam turbine may be inconvenient, a dc motor driver may be selected for use for relatively short periods.

(3) At least two fuel oil heaters will be used for reliability and to facilitate maintenance. Typical heater design for Bunker C! fuel oil will provide for temperature increases from 100 to 230° F using steam or hot water for heating medium.

c. Piping system.

(1) The piping system will be designed to maintain pressure by recirculating excess oil to the bulk storage tank. The burner pumps also will circulate back to the storage tank. A recirculation connection will be provided at each burner for startup. It will be manually valved and shut off after burner is successfully lit off and operating smoothly.

(2) Piping systems will be adapted to the type of burner utilized. Steam atomizing burners will have "blowback" connections to cleanse burners of fuel with steam on shutdown. Mechanical atomizing burner piping will be designed to suit the requirements of the burner.

d. Instruments and control. Instruments and controls include combustion controls, burner management system, control valves and shut off valves.

3-15. Coal handling and storage systems

a. Available systems. The following principal systems will be used as appropriate for handling, storing and reclaiming coal:

(1) *Relatively small to intermediate system;* coal purchases sized and washed. A system with a track or truck (or combined track/truck) hopper, bucket elevator with feeder, coal silo, spouts and chutes, and a dust collecting system will be used. Elevator will be arranged to discharge via closed chute into one or two silos, or spouted to a ground pile for moving into dead storage by bulldozer. Reclaim from dead storage will be by means of bulldozer to track/truck hopper.

(2) *Intermediate system;* coal purchased sized and washed. This will be similar to the system described in (1) above but will use an enclosed skip hoist instead of a bucket elevator for conveying coal to top of silo.

(3) *Intermediate system alternatives.* For more than two boilers, an overbunker flight or belt conveyor will be used. If mine run, uncrushed coal proves economical, a crusher with feeder will be installed in association with the track/truck hopper.

(4) *Larger systems, usually with mine run coal.* A larger system will include track or truck (or combined track/truck) unloading hopper, separate dead storage reclaim hoppers, inclined belt conveyors with appropriate feeders, transfer towers, vibrating screens, magnetic separators, crusher(s), overbunker conveyor(s) with automatic tripper, weighing equipment, sampling equipment, silos, dust collecting system(s), fire protection, and like items. Where two or more types of coal are burned (e.g., high and low sulphur), blending facilities will be required.

(5) *For cold climates.* All systems, regardless of size, which receive coal by railroad will require car thawing facilities and car shakeouts for loosening frozen coal. These facilities will not be provided for truck unloading because truck runs are usually short.

b. Selection of handling capacity. Coal handling system capacity will be selected so that ultimate planned 24-hour coal consumption of the plant at maximum expected power plant load can be unloaded or reclaimed in not more than 7-1/2 hours, or within the time span of one shift after allowance of a 1/2-hour margin for preparation and cleanup time. The handling capacity should be calculated using the worst (lowest heating value) coal which may be burned in the future and a maximum steam capacity boiler efficiency at least 3 percent less than guaranteed by boiler manufacturer.

c. Outdoor storage pile. The size of the outdoor storage pile will be based on not less than 90 days of the ultimate planned 24-hour coal consumption of the plant at maximum expected power plant load. Some power plants, particularly existing plants which are being rehabilitated or expanded, will have outdoor space limitations or are situated so that it is environmentally inadvisable to have a substantial outdoor coal pile.

d. Plant Storage.

(1) For small or medium sized spreader stoker fired plants, grade mounted silo storage will be specified with a live storage shelf above and a reserve storage space below. Usually arranged with one silo per boiler and the silo located on the outside of the firing aisle opposite the boiler, the live storage shelf will be placed high enough so that the spout to the stoker hopper or coal scale above the hopper emerges at a point high enough for the spout angle to be not less than 60 degrees from the horizontal. The reserve storage below the live storage shelf will be arranged to recirculate back to the loading point of the elevator so that coal can be raised to the top of the live storage shelf as needed. Figure 3-11 shows a typical bucket elevator grade mounted silo arrangement for a small or medium sized steam generating facility.

(2) For large sized spreader stoker fired plants, silo type overhead construction will be specified. It will be fabricated of structural steel or reinforced concrete with stainless steel lined conical bottoms.

(3) For small or medium sized plants combined live and reserve storage in the silo will be not less than 3 days at 60 percent of maximum expected load of the boiler(s) being supplied from the silo so that reserves from the outside storage pile need not be drawn upon during weekends when operating staff is reduced. For large sized plants this storage requirement will be 1 day.

e. Equipment and systems.

(1) *Bucket elevators.* Bucket elevators will be chain and bucket type. For relatively small installations the belt and bucket type is feasible although not as rugged as the chain and bucket type. Typical bucket elevator system is shown in Figure 3-11.

(2) *Skip hoists.* Because of the requirement for dust suppression and equipment closure dictated by

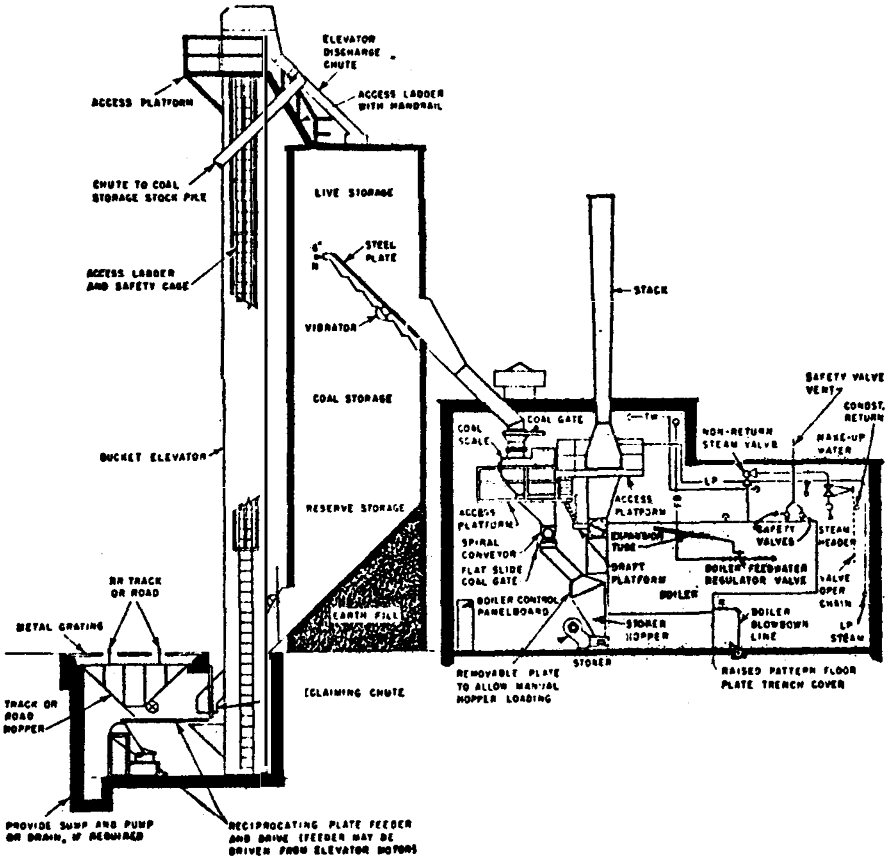

Figure 3-11. Typical coal handling system for spreader stoker fired boiler (with bucket elevator).

environmental considerations, skip hoists will not be specified.

(3) *Belt conveyors.* Belt conveyors will be selected for speeds not in excess of 500 to 550 feet per minute. They will be specified with roller bearings for pulleys and idlers, with heavy duty belts, and with rugged helical or herringbone gear drive units.

(4) *Feeders.* Feeders are required to transfer coal at a uniform rate from each unloading and intermediate hopper to the conveyor. Such feeders will be of the reciprocating plate or vibrating pan type with single or variable speed drive. Reciprocating type feeders will be used for smaller installations; the vibrating type will be used for larger systems.

(5) *Miscellaneous.* The following items are required as noted

(a) Magnetic separators for removal of tramp iron from mine run coal.

(b) Weigh scale at each boiler and, for larger installations, for weighing in coal as received. Scales will be of the belt type with temperature compensated load cell. For very small installations, a low cost displacement type scale for each boiler will be used.

(c) Coal crusher for mine run coal; for large installations the crusher will be preceded by vibrating (scalping) screens for separating out and by-passing fines around the crusher.

(d) Traveling tripper for overbunker conveyor serving a number of bunkers in series.

(e) One or more coal samplers to check “as received” and' ‘as fired” samples for large systems.

(f) Chutes, hoppers and skirts, as required, fabricated of continuously welded steel for dust tightness and with wearing surfaces lined with stainless steel. Vibrators and poke holes will be provided at all points subject to coal stoppage or hang-up.

(g) Car shakeout and a thaw shed for loosening frozen coal from railroad cars.

(h) Dust control systems as required throughout the coal handling areas. All handling equipment—hoppers, conveyors and galleries-will be enclosed in dust tight casings or building shells and provided with negative pressure ventilation complete with heated air supply, exhaust blowers, separators, and bag filters for removing dust from exhausted air. In addition, high dust concentration areas located outside which cannot be enclosed, such as unloading and reclaim hoppers, will be provided with spray type dust suppression equipment.

(i) Fire protection system of the sprinkler type.

(j) Freeze protection for any water piping located outdoors or in unheated closures as provided for dust suppression or fire protection systems.

(k) A vacuum cleaning system for maintenance of coal handling systems having galleries and equipment enclosures.

(l) System of controls for sequencing and monitoring entire coal handling system.

Section IV. ASH HANDLING SYSTEMS

3-16. Introduction

a. Background.

(1) Most gaseous fuels burn cleanly, and the amount of incombustible material is so small that it can be safely ignored. When liquid or solid fuel is fired in a boiler, however, the incombustible material, or ash, together with a small amount of unburned carbon chiefly in the form of soot or cinders, collects in the bottom of the furnace or is carried out in a lightweight, finely divided form usually known loosely as “fly ash.” Collection of the bottom ash from combustion of coal has never been a problem as the ash is heavy and easily directed into hoppers which may be dry or filled with water,

(2) Current ash collection technology is capable of removing up to 99 percent or more of all fly ash from the furnace gases by utilizing a precipitator or baghouse, often in combination with a mechanical collector. Heavier fly ash particles collected from the boiler gas passages and mechanical collectors often have a high percentage of unburned carbon content, particularly in the case of spreader stoker fired boilers; this heavier material may be reinfected into the furnace to reduce unburned carbon losses and increase efficiency, although this procedure does increase the dust loading on the collection equipment downstream of the last hopper from which such material is reinfected.

(3) It is mandatory to install precipitators or baghouses on all new coal fired boilers for final cleanup of the flue gases prior to their ejection to atmosphere. But in most regions of the United States, mechanical collectors alone are adequate for heavy oil fired boilers because of the conventionally low ash content of this type of fuel. An investigation is required, however, for each particular oil fired unit being considered.

b. Purpose. It is the purpose of the ash handling system to:

(1) Collect the bottom ash from coal-fired spreader stoker or AFBC boilers and to convey it dry by vacuum or hydraulically by liquid pressure to a temporary or permanent storage terminal. The latter may be a storage bin or silo for ultimate transfer to rail or truck for transport to a remote disposal area, or it maybe an on-site fill area or storage pond for the larger systems where the power plant site is

adequate and environmentally acceptable for this purpose.

(2) Collect fly ash and to convey it dry to temporary or permanent storage as described above for bottom ash. Fly ash, being very light, will be wetted and is mixed with bottom ash prior to disposal to prevent a severe dust problem.

3-17. Description of major components

a. Typical oil fired system. Oil fired boilers do not require any bottom ash removal facilities, since ash and unburned carbon are light and carried out with the furnace exit gas. A mechanical collector may be required for small or intermediate sized boilers having steaming rates of 200,000 pounds per hour or less. The fly ash from the gas passage and mechanical collector hoppers can usually be handled manually because of the small amount of fly ash (soot) collected. The soot from the fuel oil is greasy and can coagulate at atmospheric temperatures making it difficult to handle. To overcome this, hoppers should be heated with steam, hot water, or electric power. Hoppers will be equipped with an outlet valve having an air lock and a means of attaching disposable paper bags sized to permit manual handling. Each hopper will be selected so that it need not be evacuated more than once every few days. If boiler size and estimated soot/ash loading is such that manual handling becomes burdensome, a vacuum or hydraulic system as described below should be considered.

b. Typical ash handling system for small or inter mediate sized coal fired boilers;

(1) Plant fuel burning rates and ash content of coal are critical in sizing the ash handling system. Sizing criteria will provide for selecting hoppers and handling equipment so that ash does not have to be removed more frequently than once each 8-hour shift using the highest ash content coal anticipated and with boiler at maximum continuous steaming capacity. For the smaller, non-automatic system it may be cost effective to select hoppers and equip ment which will permit operating at 60 percent of maximum steam capacity for 3 days without removing ash to facilitate operating with a minimum weekend crew.

(2) For a typical military power plant, the most economical selection for both bottom and fly ash disposal is a vacuum type dry system with a steam jet or mechanical exhauster for creating the vacuum (Figure 3-12). This typical plant would probably have a traveling grate spreader stoker, a mechanical collector, and a baghouse; in all likelihood, no on-site ash disposal area would be available.

(3) The ash system for the typical plant will include the following for each boiler:

(a) A refractory lined bottom ash hopper to receive the discharge from the traveling grate. A clinker grinder is not required for a spreader stoker although adequate poke holes should be incorporated into the outlet sections of the hopper.

(b) Gas passage fly ash hoppers as required by the boiler design for boiler proper, economizer, and air heater.

(c) Collector fly ash hoppers for the mechanical collector and baghouse.

(d) Air lock valves, one at each hopper outlet, manually or automatically operated as selected by the design engineer.

(4) And the following items are common to all boilers in the plant:

(a) Ash collecting piping fabricated of special hardened ferro-alloy to transfer bottom and fly ash to Storage.

(b) Vacuum producing equipment, steam or mechanical exhauster as may prove economical. For plants with substantial export steam and with low quality, relatively inexpensive makeup requirements, steam will be the choice. For plants with high quality, expensive makeup requirements, consideration should be given to the higher cost mechanical exhauster.

(c) Primary and secondary mechanical (centrifugal) separators and baghouse filter are used to clean the dust out of the ash handling system exhaust prior to discharge to the atmosphere. This equipment is mounted on top of the silo.

(d) Reinforced concrete or vitrified tile overhead silo with separator and air lock for loading silo with a “dustless” unloader designed to dampen ashes as they are unloaded into a truck or railroad car for transport to remote disposal.

(e) Automatic control system for sequencing operation of the system. Usually the manual initiation of such a system starts the exhauster and then removes bottom and fly ash from each separator collection point in a predetermined sequence. Ash unloading to vehicles is separately controlled.

Section V. TURBINES AND AUXILIARY SYSTEMS

3-18. Turbine prime movers

The following paragraphs on turbine generators discuss size and other overall characteristics of the turbine generator set. For detailed discussion of the generator and its associated electrical accessories, refer to Chapter 4.

a. Size and type ranges. Steam turbine generators for military installations will fall into the fol-

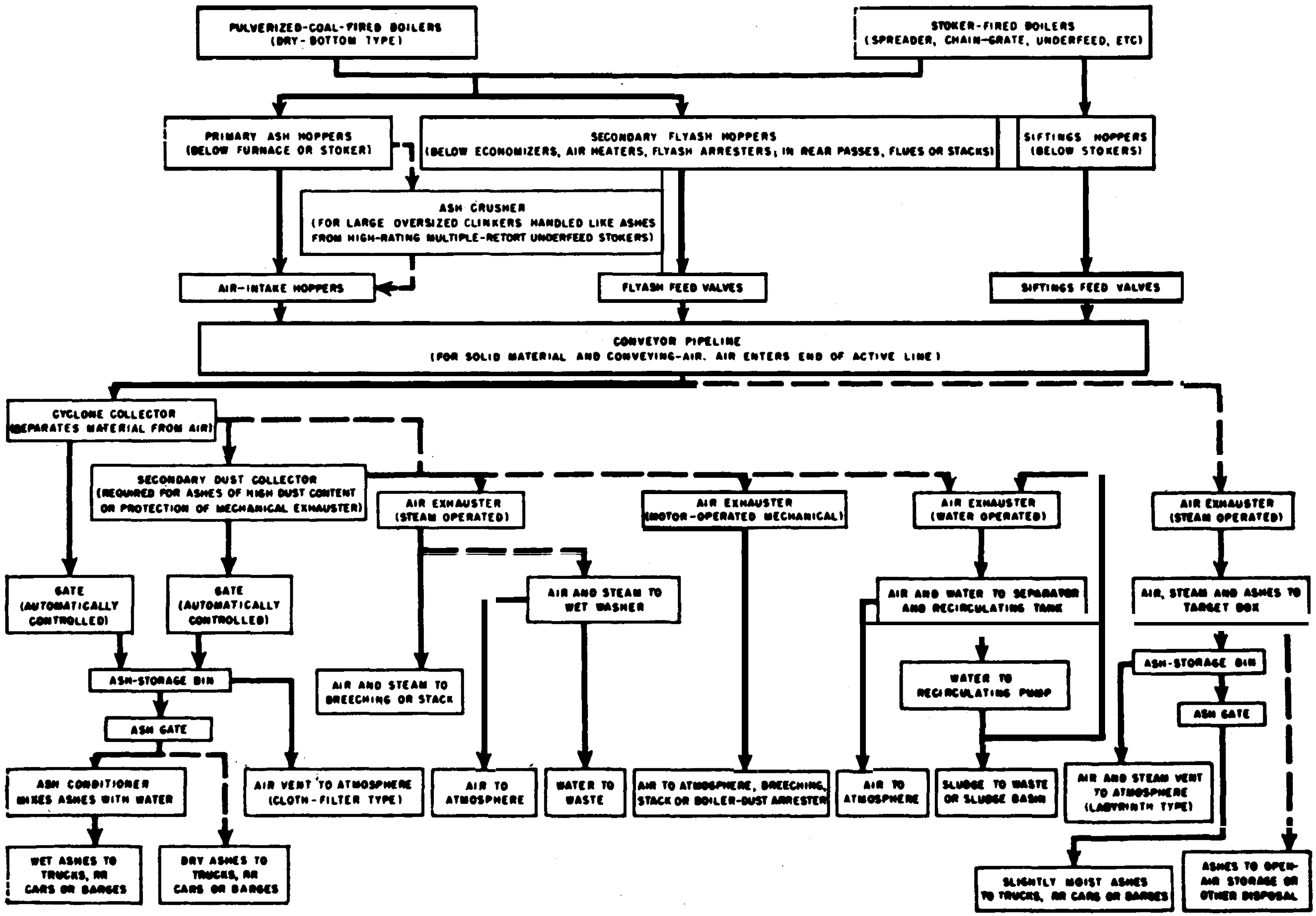

Figure 3-12. Pneumatic ash handling systems—variations.

lowing size ranges:

(1) *Small turbine generators.* From 500 to about 2500 kW rated capacity, turbine generators will usually be single stage, geared units without extraction openings for either back pressure or condensing service. Rated condensing pressures for single stage turbines range from 3 to 6 inches Hga. Exhaust pressures for back pressure units in cogeneration service typically range from 15 psig to 250 psig.

(2) *Intermediate turbine generators.* From about 2500 to 10,000 kW rated capacity, turbine generators will be either multi-stage, multi-valve machines with two pole direct drive generators turning at 3600 rpm, or high speed turbines with gear reducers may also be used in this size range. Units are equipped with either uncontrolled or controlled (automatic) extraction openings. Below 4000 kW, there will be one or two openings with steam pressures up to 600 psig and 750°F. From 4000 kW to 10,000 kW, turbines will be provided with two to four uncontrolled extraction openings, or one or two automatic extraction openings. These turbines would have initial steam conditions from 600 psig to 1250 psig, and 750°F to 900°F. Typical initial steam conditions would be 600 psig, 825º For 850 psig, 900°F.

(3) *Large turbine generators.* In the capacity range 10,000 to 30,000 kW, turbine generators will be direct drive, multi-stage, multi-valve units. For electric power generator applications, from two to five uncontrolled extraction openings will be required for feedwater heating. In cogeneration applications which include the provision of process or heating steam along with power generation, one automatic extraction opening will be required for each level of processor heating steam pressure specified, along with uncontrolled extraction openings for feedwater heating. Initial steam conditions range up to 1450 psig and 950 "F with condensing pressures from 1 1/2 to 4 inches Hga.

b. Turbine features and accessories. In all size ranges, turbine generator sets are supplied by the manufacturer with basic accessories as follows:

(1) Generator with cooling system, excitation and voltage regulator, coupling, and speed reduction gear, if used.

(2) Turbine and generator (and gear) lubrication system including tank, pumps, piping, and controls.

(3) Load speed governor, emergency overspeed governor, and emergency inlet steam trip valve with related hydraulic piping.

(4) Full rigid base plate in small sizes or separate mounting sole plates for installation in concrete pedestal for larger units.

(5) Insulation and jacketing, instruments, turning gear and special tools.

3-19. Generators

For purposes of this section, it is noted that the generator must be mechanically compatible with the driving turbine, coupling, lubrication system, and vibration characteristics (see Chapter 4 for generator details).

3-20. Turbine features

a. General. Turbine construction may be generally classified as high or low pressure, single or multistage, back pressure on condensing, direct drive or gear reducer drive, and for electric generator or for mechanical drive service.

(1) *Shell pressures.* High or low pressure construction refers generally to the internal pressures to be contained by the main shell or casing parts.

(2) *Single us. multi-stage.* Single or multi-stage designs are selected to suit the general size, enthalpy drops and performance requirements of the turbine. Multi-stage machines are much more expensive but are also considerably more efficient. Single stage machines are always less expensive, simpler and less efficient. They may have up to three velocity wheels of blading with reentry stationary vanes between wheels to improve efficiency. As casing pressure of single stage turbines are equal to exhaust pressures, the design of seals and bearings is relatively simple.

(3) *Back pressure vs. condensing.* Selection of a back pressure or a condensing turbine is dependent on the plant function and cycle parameters. (See Chapter 3, Section I for discussion of cycles.) Condensing machines are larger and more complex with high pressure and vacuum sealing provisions, steam condensers, stage feedwater heating, extensive lube oil systems and valve gear, and related auxiliary features.

(4) *Direct drive vs. geared sets.* Direct drive turbines generators turn the turbine shaft at generator speed. Units 2500 kW and larger are normally direct connected. Small, and especially single stage, turbines may be gear driven for compactness and for single stage economy. Gear reducers add complexity and energy losses to the turbine and should be used only after careful consideration of overall economy and reliability.

(5) *Mechanical drive.* Main turbine units in power plants drive electrical generators, although large pumps or air compressors may also be driven by large turbines. In this event, the turbines are called "mechanical drive" turbines. Mechanical drive turbines are usually variable speed units with special governing equipment to adapt to best economy balance between driver (turbine) and driven machine. Small auxiliary turbines for cycle pumps,

fans, or air compressor drives are usually single stage, back pressure, direct drive type designed for mechanical simplicity and reliability. Both constant speed and variable speed governors are used depending on the application.

b. Arrangement. Turbine generators are horizontal shaft type with horizontally split casings. Relatively small mechanical drive turbines may be built with vertical shafts. Turbine rotor shaft is usually supported in two sleeve type, self aligning bearings, sealed and protected from internal casing steam conditions. Output shaft is coupled to the shaft of the generator which is provided with its own enclosure but is always mounted on the same foundation as the turbine.

(1) *Balance.* Balanced and integrated design of the turbine, coupling and generator moving parts is important to successful operation, and freedom from torsional or lateral vibrations as well as prevention of expansion damage are essential.

(2) *Foundations.* Foundations and pedestals for turbine generators will be carefully designed to accommodate and protect the turbine generator, condenser, and associated equipment. Strength, mass, stiffness, and vibration characteristics must be considered. Most turbine generator pedestals in the United States are constructed of massive concrete.

3-21. Governing and control

a. Turbine generators speed/load control. Electrical generator output is in the form of synchronized ac electrical power, causing the generator and driving turbine to rotate at exactly the same speed (or frequency) as other synchronized generators connected into the common network. Basic speed/load governing equipment is designed to allow each unit to hold its own load steady at constant frequency, or to accept its share of load variations, as the common frequency rises and falls. Very small machines may use direct mechanical governors, but the bulk of the units will use either mechanical-hydraulic governing systems or electrohydraulic systems. Non-reheat condensing units 5000 kW and larger and back pressure units without automatic extraction will be equipped with mechanical-hydraulic governing. For automatic extraction units larger than 20,000 kW, governing will be specified either with a mechanical-hydraulic or an electro-hydraulic system.

b. Overspeed governors. All turbines require separate safety or overspeed governing systems to insure inlet steam interruption if the machine exceeds a safe speed for any reason. The emergency governor closes a specially designed stop valve which not only shuts off steam flow but also trips various safety devices to prevent overspeed by flash steam induction through the turbine bleed (extraction) points.

c. Single and multi-valve arrangements. Whatever type of governor is used, it will modulate the turbine inlet valves to regulate steam flow and turbine output. For machines expected to operate extensively at low or partial loads, multi-valve arrangements improve economy. Single valve turbines, in general, have equal economy and efficiency at rated load, but lower part load efficiencies.

3-22. Turning gear

a. General. For turbines sized 10,000 kW and larger, a motor operated turning gear is required to prevent the bowing of the turbine rotor created by the temperature differential existing between the upper and lower turbine casings during the long period after shutdown in which the turbine cools down. The turbine cannot be restarted until it has completely cooled down without risk of damage to interstate packing and decrease of turbine efficiency, causing delays in restarting. The turning gear is mounted at the exhaust end of the turbine and is used to turn the rotor at a speed of 1 to 4 rpm when the turbine is shut down in order to permit uniform cooling of the rotor. Turning gear is also used during startup to evenly warm up the rotor before rolling the turbine with steam and as a jacking device for turning the rotor as required for inspection and maintenance when the turbine is shut down.

b. Arrangement and controls. The turning gear will consist of a horizontal electric motor with a set of gear chains and a clutching arrangement which engages a gear ring on the shaft of the turbine. Its controls are arranged for local and/or remote starting and to automatically disengage when the turbine reaches a predetermined speed during startup with steam. It is also arranged to automatically engage when the turbine has been shut down and decelerated to a sufficiently slow speed. Indicating lights will be provided to indicate the disengaged or engaged status of the turning gear and an interlock provided to prevent the operation of the turning gear if the pressure in the turbine lubrication oil system is below a predetermined safe setting.

3-23. Lubrication systems

a. General. Every turbine and its driven machine or generator requires adequate lubricating oil sup ply including pressurization, filtration, oil cooling, and emergency provisions to insure lubrication in the event of a failure of main oil supply. For a typical turbine generator, an integrated lube oil storage tank with built in normal and emergency pumps is usually provided. Oil cooling may be by means of an external or internal water cooled heat exchanger. Oil temperatures should be monitored and controlled, and heating may be required for startup.

b. Oil Pumps. Two full capacity main lube oil pumps will be provided. One will be directly driven from the turbine shaft for multi-stage machines. The second full size pump will be ac electric motor driven. An emergency dc motor driven or turbine-driven backup pump will be specified to allow orderly shutdown during normal startup and shutdown when the shaft driven pump cannot maintain pressure, or after main pump failure, or in the event of failure of the power supply to the ac electric motor driven pumps.

c. Filtration. Strainers and filters are necessary for the protection and longevity of lubricated parts. Filters and strainers should be arranged in pairs for on line cleaning, inspection, and maintenance. Larger turbine generator units are sometimes equipped with special off base lubrication systems to provide separate, high quality filtering.

3-24. Extraction features

a. Uncontrolled extraction systems. Uncontrolled bleed or extraction openings are merely nozzles in the turbine shell between stages through which relatively limited amounts of steam may be extracted for stage feedwater heating. Such openings add little to the turbine cost as compared with the cost of feedwater heaters, piping, and controls. Turbines so equipped are usually rated and will have efficiencies and performance based on normal extraction pressures and regenerative feedwater heating calculations. Uncontrolled extraction opening pressures will vary in proportion to turbine steam flow, and extracted steam will not be used or routed to any substantial uses except for feedwater heating.

b. Automatic extraction. Controlled or automatic extraction turbines are more elaborate and equipped with variable internal orifices or valves to modulate internal steam flows so as to maintain extraction pressures within specified ranges. Automatic extraction machine governors provide automatic self-contained modulation of the internal flow orifices or valves, using hydraulic operators. Automatic extraction governing systems can also be adapted to respond to external controls or cycle parameters to permit extraction pressures to adjust to changing cycle conditions.

c. Extraction turbine selection. Any automatic extraction turbine is more expensive than its straight uncontrolled extraction counterpart of similar size, capacity and type; its selection and use require comprehensive planning studies and economic analysis for justification. Sometimes the same objective can be achieved by selecting two units, one of which is an uncontrolled extraction-condensing machine and the other a back pressure machine.

3-25. Instruments and special tools

a. Operating instruments. Each turbine will be equipped with appropriate instruments and alarms to monitor normal and abnormal operating conditions including speed, vibration, shell and rotor expansions, steam and metal temperatures, rotor straightness, turning gear operation, and various steam, oil and hydraulic system pressures.

b. Special took. Particularly for larger machines, complete sets of special tools, lifting bars, and related special items are required for organized and effective erection and maintenance.

Section VI. CONDENSER AND CIRCULATING WATERSYSTEM

3-26. Introduction

a. Purpose.

(1) The primary purpose of a condenser and circulating water system is to remove the latent heat from the steam exhausted from the exhaust end of the steam turbine prime mover, and to transfer the latent heat so removed to the circulating water which is the medium for dissipating this heat to the atmosphere. A secondary purpose is to recover the condensate resulting from the phase change in the exhaust steam and to recirculate it as the working fluid in the cycle.

(2) Practically, these purposes are accomplished in two steps. In the first step, the condenser is supplied with circulating water which serves as a medium for absorbing the latent heat in the condensing exhaust steam. The source of this circulating water can be a natural body of water such as an ocean, a river, or a lake, or it can be from a recirculated source such as a cooling tower or cooling pond. In the second step, the heated circulating water is rejected to the natural body of water or recirculated source which, in turn, transfers the heat to the atmosphere, principally by evaporative cooling effect.

b. Equipment required—general. Equipment required for a system depends on the type of system utilized. There are two basic types of condensers: surface and direct contact.

There are also two basic types of cooling systems:

Once through; and

Recirculating type, including cooling ponds, mechanical draft cooling towers, natural draft cooling towers, or a combination of a pond and tower.

3-27. Description of major components

a. Surface condensers.

(1) *General description.* These units are designed as shell and tube heat exchangers. A surface condenser consists of a casing or shell with a chamber at each end called a "water box. " Tube sheets separate the two water boxes from the center steam space. Banks of tubes connect the water boxes by piercing the tube sheets; the tubes essentially fill the shell or steam space. Circulating water pumps force the cooling (circulating] water through the water boxes and the connecting tubes. Uncontaminated condensate is recovered in surface condensers since the cooling water does not mix with the condensing steam. Steam pressure in a condenser (or vacuum) depends mainly on the flow rate and temperature of the cooling water and on the effectiveness of air removal equipment.

(2) *Passes and water boxes.*

(a) Tubing and water boxes may be arranged for single pass or two pass flow of water through the shell. In single pass units, water enters the water box at one end of the tubes, flows once through all the tubes in parallel, and leaves through the outlet water box at the opposite end of the tubes. In two pass units, water flows through the bottom half of the tubes (sometimes the top half) in one direction, reverses in the far end water box, and returns through the upper or lower half of the tubes to the near water box. Water enters and leaves through the near water box which is divided into two chambers by a horizontal plate. The far end water box is undivided to permit reversal of flow.

(b) For a relatively large cooling water source and low circulating water pump heads (hence low unit pumping energy costs), single pass units will be used. For limited cooling water supplies and high circulating water pump heads (hence high unit pumping energy costs), two pass condensers will be specified. In all cases, the overall condenser-circulating water system must be optimized by the designer to arrive at the best combination of condenser surface, temperature, vacuum, circulating water pumps, piping, and ultimate heat rejection equipment.

(c) Most large condensers, in addition to the inlet waterbox horizontal division, have vertical partitions to give two separate parallel flow paths through the shell. This permits taking half the condensing surface our of service for cleaning while water flows through the other half to keep the unit running at reduced load.

(3) *Hot well.* The hot well stores the condensate and keeps a net positive suction head on the condensate pumps. Hot well will have a capacity of at least 3 minutes maximum condensing load for surges and to permit variations in level for the condensate control system.

(4) *Air removal offtakes.* One or more air offtakes in the steam space lead accumulating air to the air removal pump.

(5) *Tubes.*

(a) The tubes provide the heat transfer surface in the condenser are fastened into tube sheets, usually made of Muntz metal. Modern designs have tubes rolled into both tube sheets; for ultra-tightness, alloy steel tubes may be welded into tube sheets of appropriate material. Admiralty is the most common tube material and frequently is satisfactory for once through systems using fresh water and for recirculating systems. Tube material in the "off gas" section of the condenser should be stainless steel because of the highly corrosive effects of carbon dioxide and ammonia in the presence of moisture and oxygen. These gases are most concentrated in this section. Other typical condenser tube materials include:

(1) Cupronickel
(2) Aluminum bronze
(3) Aluminim brass
(4) Various grades of stainless steel

(b) Condenser tube water velocities range from 6 to 9 feet per second (Table 3- 12). Higher flow rates raise pumping power requirements and erode tubes at their entrances, thus shortening their life expectancy. Lower velocities are inefficient from a heat transfer point of view. Tubes are generally installed with an upwardly bowed arc. This provides for thermal expansion, aids drainage in a shutdown condenser, and helps prevent tube vibration.

b. Direct contact condensers. Direct contact condensers will not be specified.

c. Condenser auxiliaries.

(1) *General.* A condenser needs equipment and conduits to move cooling water through the tubes, remove air from the steam space, and extract condensate from the hotwell. Such equipment and conduits will include:

(a) Circulating water pumps.
(b) Condensate or hotwell pumps.
(c) Air removal equipment and piping.
(d) Priming ejectors.
(e) Atmospheric relief valve.
(f) Inlet water tunnel, piping, canal, or combination of these conduits.
(g) Discharge water tunnel, piping or canal, or combination of these conduits.

(2) *Circulating water pumps.* A condenser uses 75 to 100 pounds of circulating water per pound of steam condensed. Hence, large units need substantial water flows; to keep pump work to a minimum, top of condenser water boxes in a closed system will

Table 3-12. Condenser Tube Design Velocities.

Material	Design Velocities fps		
	Fresh Water	Brackish Water	Salt Water
Admiralty Metal	7.0	(1)	(1)
Aluminum Brass[2]	8.0	**7.0**	**7.0**
Copper-Nickel Alloys:			
90-10	8.0	8.0	7.0 to 7.5
80-20	8.0	8.0	7.0 to 7.5
70-30	9.0	9.0	8.0 to 8.5
Stainless Steel	9.0 to 9.5	8.0[3]	8.0[3]
Aluminum[4]	8.0	7.0	6.8

NOTES:

(1) Not normally used, but if used, velocity shall not exceed 6.0 fps.

(2) For salt and brackish water, velocities in excess of 6.8 fps are not recommended.

(3) Minimum velocity of 5.5 fps to prevent chloride attack.

(4) Not recommended for circulating water containing high concentration of heavy metal salts.

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not be higher than approximately 27 feet above minimum water source level which permits siphon operation without imposing static head. With a siphon system, air bubbles tend to migrate to the top of the system and must be removed with vacuum-producing equipment. The circulating pumps then need to develop only enough head to overcome the flow resistance of the circulating water circuit. Circulating pumps for condensers are generally of the centrifugal type for horizontal pumps, and either mixed flow or propeller type for vertical pumps. Vertical pumps will be specified because of their adaptability for intake structures and their ability to handle high capacities at relatively low heads. Pump material will be selected for long life.

(3) *Condensate pumps.* Condensate (or hotwell) pumps handle much smaller flows than the circulating water pumps. They must develop heads to push water through atmospheric pressure, pipe and control valve friction, closed heater water circuit friction, and the elevation of the deaerator storage tank. These pumps take suction at low pressure of two inches Hg absolute or less and handle water at saturation temperature; to prevent flashing of the condensate, they are mounted below the hotwell to receive a net positive suction head. Modern vertical “can” type pumps will be used. Specially designed pump glands prevent air leakage into the condensate, and vents from the pump connecting to the vapor space in the condenser prevent vapor binding.

(4) *Spare pumps.* Two 100 percent pumps for both circulating water and condensate service will

be specified. If the circulating water system serves more than one condenser, there will be one circulating pump per condenser with an extra pump as a common spare. Condensate pump capacity will be sized to handle the maximum condenser load under any condition of operation (e.g., with automatic extraction to heating or process shutoff and including all feedwater heater drains and miscellaneous drips received by the condenser.)

(5) *Air removal.*

(a) Non-condensable gases such as air, carbon dioxide, and hydrogen migrate continuously into the steam space of a condenser inasmuch as it is the lowest pressure region in the cycle. These gases may enter through leakage at glands, valve bonnets, porous walls, or may be in the throttle steam. Those gases not dissolved by the condensate diffuse throughout the steam space of the condenser. As these gases accumulate, their partial pressure raises the condenser total pressure and hence decreases efficiency of the turbine because of loss of available energy. The total condenser pressure is:

$$P_c = P_s + P_a$$

where P_s = steam saturation pressure corresponding to steam temperature

P_a = air pressure (moisture free)

This equation shows that air leakage must be removed constantly to maintain lowest possible vacuum for the equipment selected and the particular exhaust steam loading. In removing this air, it will always have some entrained vapor. Because of its subatmospheric pressure, the mixture must be compressed for discharge to atmosphere.

(b) Although the mass of air leakage to the condenser may be relatively small because of its very low pressure, its removal requires handling of a large volume by the air removal equipment. The air offtakes withdraw the air-vapor moisture from the steam space over a cold section of the condenser tubes or through an external cooler, which condenses part of the moisture and increases the air-to-steam ratio. Steam jets or mechanical vacuum pumps receive the mixture and compress it to atmosphere pressure.

(6) Condenser cleanliness. Surface condenser performance depends greatly on the cleanliness of the tube water side heat transfer surface. When dirty fresh water or sea water is used in the circulating water system, automatic backflush or mechanical cleaning systems will be specified for on line cleaning of the interior condenser tube surfaces.

d. Circulating water system–once through

(1) *System components.* A typical once through circulating water system, shown in figure 3-13, consists of the following components:

(a) Intake structure.

(b) Discharge, or outfall.

(c) Trash racks.

(d) Traveling screens.

(e) Circulating water pumps.

(f) Circulating water pump structure (indoor or outdoor).

(g) Circulating water canals, tunnels, and pipework.

(2) *System operation.*

(a) The circulating water system functions as follows. Water from an ocean, river, lake, or pond flows either directly from the source to the circulating water structure or through conduits which bring water from offshore; the inlet conduits discharge into a common plenum which is part of the circulating water pump structure. Water flows through bar trash racks which protect the traveling screens from damage by heavy debris and then through traveling screens where smaller debris is removed. For large systems, a motor operated trash rake can be installed to clear the bar trash racks of heavy debris. In case the traveling screens become clogged, or to prevent clogging, they are periodically backwashes by a high pressure water jet system. The backwash is returned to the ocean or other body of water. Each separate screen well is provided with stop logs and sluice gates to allow dewatering for maintenance purposes.

(b) The water for each screen flows to the suction of the circulating water pumps. For small systems, two 100-percent capacity pumps will be selected while for larger systems, three 50-percent pumps will be used. At least one pump is required for standby. Each pump will be equipped with a motorized butterfly valve for isolation purposes. The pumps discharge into a common circulating water tunnel or supply pipe which may feed one or more condensers. Also, a branch line delivers water to the booster pumps serving the closed cooling water exchangers.

(c) Both inlet and outlet water boxes of the main condensers will be equipped with butterfly valves for isolation purposes and expansion joints. As mentioned above, the system may have the capability to reverse flow in each of the condenser halves for cleaning the tubes. The frequency and duration of the condenser reverse flow or back wash operation is dictated by operating experience.

(d) The warmed circulating water from the condensers and closed cooling water exchangers is discharged to the ocean, river, lake, or pond via a common discharge tunnel.

(3) *Circulating water pump setting.* The circulating water pumps are designed to remain operable with the water level at the lowest anticipated eleva-

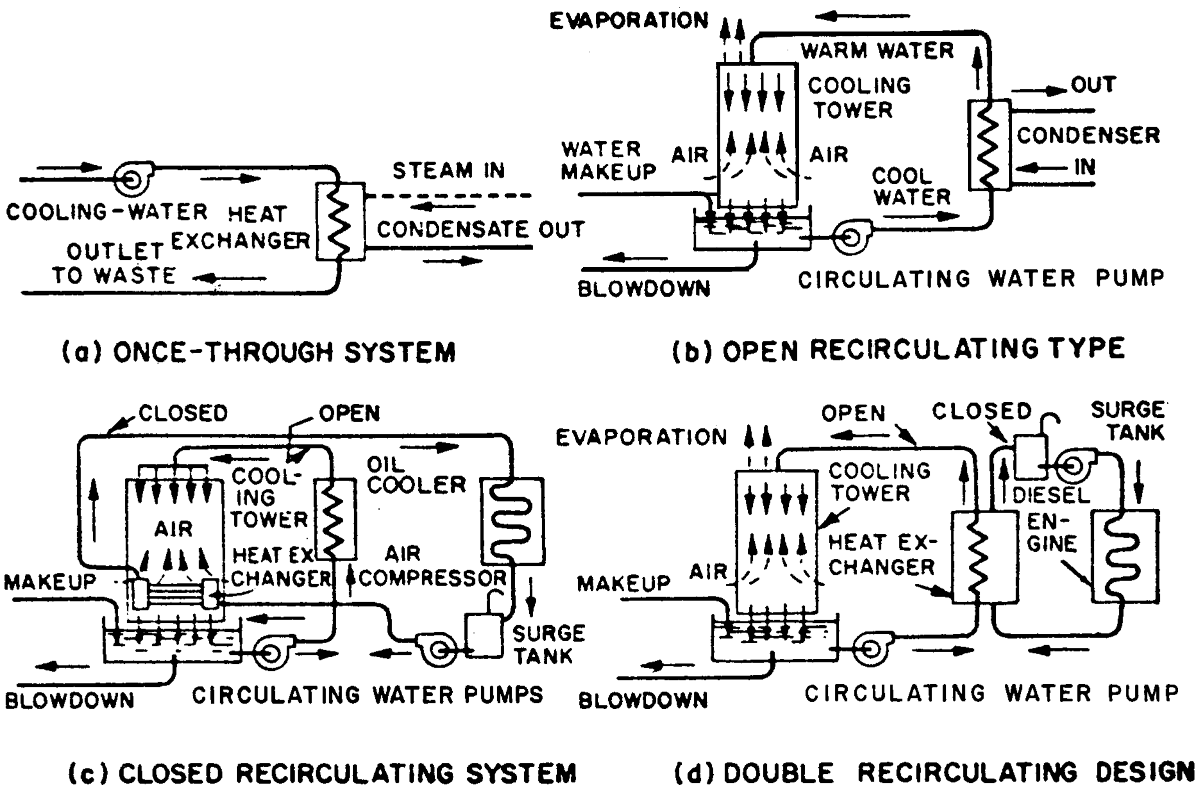

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Figure 3-13. Types of circulating water systems.

tion of the selected source. This level is a function of the neap tide for an ocean source and seasonal level variations for a natural lake or river. Cooling ponds are usually man-made with the level controlled within modest limits. The pump motors and valve motor operators will be located so that no electrical parts will be immersed in water at the highest anticipated elevation of the water source.

(4) *System pressure control.* On shutdown of a circulating water pump, water hammer is avoided by ensuring that the pumps coast down as the pump isolation valves close. System hydraulics, circulating pump coastdown times, and system isolation valve closing times must be analyzed to preclude damage to the system due to water hammer. The condenser tubes and water boxes are to be designed for a pressure of approximately 25 psig which is well above the ordinary maximum discharge pressure of the circulating water pumps, but all equipment must be protected against surge pressures caused by sudden collapse of system pressure.

(5) *Inspection and testing.* All active components of the circulating water system will be accessible for inspection during station operation.

e. Circulating water system—recirculating type

(1) *General discussion.*

(a) With a once-through system, the evaporative losses responsible for rejecting heat to the atmosphere occur in the natural body of water as the warmed circulating water is mixed with the residual water and is cooled over a period of time by evaporation and conduction heat transfer. With a recirculation system, the same water constantly circulates; evaporative losses responsible for rejecting heat to the atmosphere occur in the cooling equipment and must be replenished at the power plant site. Recirculating systems can utilize one of the following for heat rejection:

(1) A natural draft, hyperbolic cooling tower.

(2) A mechanical draft cooling tower, usually induced draft.

(3) A spray pond with a network of piping serving banks of spray nozzles.

(b) Very large, man-made ponds which take advantage of natural evaporative cooling may be considered as "recirculating" systems, although for design purposes of the circulating water system

they are once through and hence considered as such in paragraph *d* above.

(c) To avoid fogging and plumes which are characteristic of cooling towers under certain atmospheric conditions in humid climates, so called wet-dry cooling towers may be used. These towers use a combination of finned heat transfer surface and evaporative cooling to eliminate the fog and visible plume. The wet-dry types of towers are expensive and not considered in this manual. Hyperbolic towers also are expensive and are not applicable to units less than 300-500 M W; while spray ponds have limited application (for smaller units) because of the large ground area required and the problem of excessive drift. Therefore, the following descriptive material applies only to conventional induced draft cooling towers which, except for very special circumstances, will be the choice for a military power plant requiring a recirculating type system.

(2) *System components.* A typical recirculating system with a mechanical draft cooling tower consists of the following components:

(a) Intake structure which is usually an extension of the cooling tower basin.

(b) Circulating water pumps.

(c) Circulating water piping or tunnels to condensers and from condensers to top of cooling tower.

(d) Cooling tower with makeup and blowdown systems.

(3) *System operation.*

(a) The recirculating system functions as follows. Cooled water from the tower basin is directed to the circulating water pump pit. The pit is similar to the intake structure for a once through system except it is much simpler because trash racks or traveling screens are not required, and the pit setting can be designed without reference to levels of a natural body of water. The circulating water pumps pressure the water and direct it to the condensers through the circulating water discharge piping. A stream of circulating water is taken off from the main condenser supply and by means of booster pumps further pressurized as required for bearing cooling, generator cooling, and turbine generator oil cooling. From the outlet of the condensers and miscellaneous cooling services, the warmed circulating water is directed to the top of the cooling tower for rejection of heat to the atmosphere.

(b) Circulating water pump and condenser valving is similar to that described for a typical once-through system, but no automatic back flushing or mechanical cleaning system is required for the condenser. Also, due to the higher pumping heads commonly required for elevating water to the top of the tower and the break in water pressure at that point which precludes a siphon, higher pressure ratings for the pumps and condensers will be specified.

(4) *Cooling tower design.*

(a) In an induced draft mechanical cooling tower, atmospheric air enters the louvers at the bottom perimeter of the tower, flows up through the fill, usually counterflow to the falling water drop lets, and is ejected to the atmosphere in saturated condition thus carrying off the operating load of heat picked up in the condenser. Placement and arrangement of the tower or towers on the power station site will be carefully planned to avoid recirculation of saturated air back into the tower intake and to prevent drift from the tower depositing on electrical buses and equipment in the switchyard, roadways and other areas where the drift could be detrimental.

(b) Hot circulating water from the condenser enters the distribution header at the top of the tower. In conventional towers about 75 percent of the cooling takes place be evaporation and the remainder by heat conduction; the ratio depends on the humidity of the entering air and various factors.

(5) *Cooling tower performance.* The principal performance factor of a cooling tower is its approach to the wet bulb temperature; this is the difference between the cold water temperature leaving the tower and the wet bulb temperature of the entering air. The smaller the approach, the more efficient and expensive the tower. Another critical factor is the cooling range. This is the difference between the hot water temperature entering the tower and the cold water temperature leaving it is essentially the same as the circulating water temperature rise in the condenser. Practically, tower approaches are 8 to 15°F with ranges of 18 to 22°F. Selection of approach and range for a tower is the subject for an economic optimization which should include simultaneous selection of the condensers as these two major items of equipment are interdependent.

(6) *Cooling tower makeup.*

(a) Makeup must be continuously added to the tower collecting basin to replace water lost by evaporation and drift. In many cases, the makeup water must be softened to prevent scaling of heat transfer surfaces; this will be accomplished by means of cold lime softening. Also the circulating water must be treated with bioxides and inhibitors while in use to kill algae, preserve the fill, and prevent metal corrosion and fouling. Algae control is accomplished by means of chlorine injection; acid and phosphate feeds are used for pH control and to keep heat surfaces clean.

(b) The circulating water system must be blown down periodically to remove the accumulated solid concentrated by evaporation.

3-28. Environmental concerns

a. Possible problems. Some of the environmental concerns which have an impact on various types of power plant waste heat rejection systems are as follows:

(1) Compatibility of circulating water system with type of land use allocated to the surrounding area of the power plant.

(2) Atmospheric ground level fogging from cooling tower.

(3) Cooling tower plumes.

(4) Ice formation on adjacent roads, buildings and structures in the winter.

(5) Noise from cooling tower fans and circulating water pumps.

(6) Salts deposition on the contiguous countryside as the evaporated water from the tower is absorbed in the atmosphere and the entrained chemicals injected in the circulating water system fallout.

(7) Effect on aquatic life for once though systems:

(a) Entrapment or fish kill.

(b) Migration of aquatic life.

(c) Thermal discharge.

(d) Chemical discharge.

(e) Effect of plankton.

(8) Effect on animal and bird life.

(9) Possible obstruction to aircraft (usually only a problem for tall hyperbolic towers).

(10) Obstruction to ship and boat navigation (for once through system intakes or navigable streams or bodies of water).

b. Solutions to problems. Judicious selection of the type of circulating water system and optimum orientation of the power plant at the site can minimize these problems. However, many military projects will involve cogeneration facilities which may require use of existing areas where construction of cooling towers may present serious on base problems and, hence, will require innovative design solutions.

Section VII. FEEDWATER SYSTEM

3-29. Feedwater heaters

a. Open type—deaerators.

(1) *Purpose.* Open type feedwater heaters are used primarily to reduce feedwater oxygen and other noncondensable gases to essentially zero and thus decrease corrosion in the boiler and boiler feed system. Secondarily, they are used to increase thermal efficiency as part of the regenerative feedwater heating cycle.

(2) *Types.*

(a) There are two basic types of open deaerating heaters used in steam power plants—tray type and spray type. The tray or combination spray/tray type unit will be used. In plants where heater tray maintenance could be a problem, or where the feedwater has a high solids content or is corrosive, a spray type deaerator will be considered.

(b) All types of deaerators will have internal or external vent condensers, the internal parts of which will be protected from corrosive gases and oxidation by chloride stress resistant stainless steel.

(c) In cogeneration plants where large amounts of raw water makeup are required, a deaerating hot process softener will be selected depending on the steam conditions and the type of raw water being treated (Section IX, paragraph 3-38 and 3-39).

(3) *Location.* The deaerating heater should be located to maintain a pressure higher than the NPSH required by the boiler feed pumps under all conditions of operation. This means providing a margin of static head to compensate for sudden fall off in deaerator pressure under an upset condition. Access will be provided for heater maintenance and for reading and maintaining heater instrumentation.

(4) *Design criteria.*

(a) Deaerating heaters and storage tanks will comply with the latest revisions of the following standards:

(1) ASME Unified Pressure Vessel Code.

(2) ASME Power Test Code for Deaerators.

(3) Heat Exchanger Institute (HE I).

(4) American National Standards Institute (ANSI).

(b) Steam pressure to the deaerating heater *will* not be less than three psig.

(c) Feedwater leaving the deaerator will contain no more than 0.005 cc/liter of oxygen and zero residual carbon dioxide. Residual content of the dissolved gases will be consistent with their relative volubility and ionization.

(d) Deaerator storage capacity will be not less than ten minutes in terms of maximum design flow through the unit.

(e) Deaerator will have an internal or external oil separator if the steam supply may contain oil, such as from a reciprocating steam engine.

(f) Deaerating heater will be provided with the following minimum instrumentation: relief valve, thermometer, thermocouple and test well at feedwater inlet and outlet, and steam inlet; pressure gauge at feedwater and steam inlets; and a level control system (paragraph c).

b. Closed type.

(1) *Purpose.* along with the deaerating heater, closed feedwater heaters are used in a regenerative feedwater cycle to increase thermal efficiency and thus provide fuel savings. An economic evaluation will be made to determine the number of stages of feedwater heating to be incorporated into the cycle. Condensing type steam turbine units often have both low pressure heaters (suction side of the boiler feed pumps) and high pressure heaters (on the discharge side of the feed pumps). The economic analysis of the heaters should consider a desuperheater section when there is a high degree of superheat in the steam to the heater and an internal or external drain cooler (using entering condensate or boiler feedwater) to reduce drains below steam saturation temperature.

(2) Type. The feedwater heaters will be of the U-tube type.

(3) Location. Heaters will be located to allow easy access for reading and maintaining heater instrumentation and for pulling the tube bundle or heater shell. High pressure heaters will be located to provide the best economic balance of high pressure feedwater piping, steam piping and heater drain pip ing.

(4) Design criteria

(a) Heaters will comply with the latest revisions of the following standards:

(1) ASME Unfired Pressure Vessel Code.

(2) ASME Power Test Code for Feedwater Heaters.

(3) Tubular Exchanger Manufacturers Association (TEMA).

(4) Heat Exchanger Institute (HE I).

(5) American National Standards Institute (ANSI).

(b) Each feedwater heater will be provided with the following minimum instrumentation: shell and tube relief valves; thermometer, thermocouple and test well at feedwater inlet and outlet; steam inlet and drain outlet; pressure gauge at feedwater inlet and outlet, and steam inlet; and level control system.

c. Level control systems.

(1) *Purpose.* Level control systems are required for all open and closed feedwater heaters to assure efficient operation of each heater and provide for protection of other related power plant equipment. The level control system for the feedwater heaters is an integrated part of a plant cycle level control system which includes the condenser hotwell and the boiler level controls, and must be designed with this in mind. This paragraph sets forth design criteria which are essential to a feedwater heater level control system. Modifications may be required to fit the actual plant cycle.

(2) *Closed feedwater heaters.*

(a) Closed feedwater heater drains are usually cascaded to the next lowest stage feedwater heater or to the condenser, A normal and emergency drain line from each heater will be provided. At high loads with high extraction steam pressure, the normal heater drain valve cascades drain to the next lowest stage heater to control its own heater level. At low loads with lower extraction steam pressure and lower pressure differential between successive heaters, sufficient pressure may not be available to allow the drains to flow to the next lowest stage heater. In this case, an emergency drain valve will be provided to cascade to a lower stage heater or to the condenser to hold the predetermined level.

(b) The following minimum instrumentation will be supplied to provide adequate level control at each heater: gauge glass; level controller to modulate normal drain line control valve (if emergency drain line control valve is used, controller must have a split range); and high water level alarm switch.

(3) *Open feedwater heaters-deaerators. The* following minimum instrumentation will be supplied to provide adequate level control at the heater: gauge glass, level controller to control feedwater inlet control waive (if more than one feedwater inlet source, controller must have a split range); low water level alarm switch; "low-low" water level alarm switch to sound alarm and trip boiler feed pumps, or other pumps taking suction from heater; high water level alarm switch; and "high-high" water level controller to remove water from the deaerator to the condenser or flash tank, or to divert feedwater away from the deaerator by opening a diverting valve to dump water from the feedwater line to the condenser or condensate storage tank.

(4) *Reference.* The following papers should be consulted in designing feedwater level control systems, particularly in regard to the prevention of water induction through extraction piping

(a) ASMD Standard TWDPS-1, July 1972, "Recommended Practices for the Prevention of Water Damage to Steam Turbines Used for Electric Power Generation (Part 1- Fossil Fueled Plants)."

(b) General Electric Company Standard GEK-25504, Revision D, "Design and Operating Recommendations to Minimize Water Induction in Large Steam Turbines."

(c) Westinghouse Standard, "Recommendation to Minimize Water Damage to Steam Turbines."

3-30. Boiler feed pumps.

a. General. Boiler feed pumps are used to pressurize water from the deaerating feedwater heater or deaerating hot process softener and feed it through any high pressure closed feedwater heaters to the boiler inlet. Discharge from the boiler superheated steam in order to maintain proper main steam ternperature to the steam turbine generator.

b. Types. There are two types of centrifugal multi-stage boiler feed pumps commonly used in steam power plants—horizontally split case and barrel type with horizontal or vertical (segmented) split inner case. The horizontal split case type will be used on boilers with rated outlet pressures up to 900 psig. Barrel type pumps will be used on boilers with rated outlet pressure in excess of 900 psig.

c. Number of pumps. In all cases, at least one spare feed pump will be provided.

(1) For power plants where one battery of boiler feed pumps feeds one boiler.

(a) If the boiler is base loaded most of the time at a high load factor, then use two pumps each at 110-125 percent of boiler maximum steaming capacity.

(b) If the boiler is subject to daily wide range load swings, use three pumps at 55-62.5 percent of boiler maximum steaming capacity. With this arrangement, two pumps are operated in parallel between 50 and 100 percent boiler output, but only one pump is operated below 50 percent capacity. This arrangement allows for pump operation in its most efficient range and also permits a greater degree of flexibility.

(2) For power plants where one battery of pump feeds more than one boiler through a header system, the number of pumps and rating will be chosen to provide optimum operating efficiency and capital costs. At least three 55-62.5 percent pumps should be selected based on maximum steaming capacity of all boilers served by the battery to provide the flexibility required for a wide range of total feedwater flows.

d. Location. The boiler feed pumps will be located at the lowest plant level with the deaerating heater or softener elevated sufficiently to maintain pump suction pressure higher than the required NPSH of the pump under all operating conditions. This means a substantial margin over the theoretically calculated requirements to provide for pressures collapses in the dearator under abnormal operating conditions. Deaerator level will never be decreased for structural or aesthetic reasons, and suction pipe connecting deaerator to boiler feed pumps should be sized so that friction loss is negligible.

e. Recirculation control system.

(1) To prevent overheating and pump damage, each boiler feed pump will have its own recirculation control system to maintain minimum pump flow whenever the pump is in operation. The control system will consist off

(a) Flow element to be installed in the pump suction line.

(b) Flow controller.

(c) Flow control valve.

(d) Breakdown orifice.

(2) Whenever the pump flow decreases to minimum required flow, as measured by the flow element in the suction line, the flow controller will be designed to open the flow control valve to maintain minimum pump flow. The recirculation line will be discharge to the deaerator. A breakdown orifice will be installed in the recirculation line just before it enters the deaerator to reduce the pressure from boiler feed pump discharge level to deaerator operating pressure.

f. Design criteria.

(1) Boiler feed pumps will comply with the latest revisions of the following standards:

(a) Hydraulics Institute (HI).

(b) American National Standards Institute (ANSI).

(2) Pump head characteristics will be maximum at zero flow with continuously decreasing head as flow increases to insure stable operation of one pump, or multiple pumps in parallel, at all loads.

(3) Pumps will operate quietly at all loads without internal flashing and operate continuously without overheating or objectionable noises at minimum recirculation flow.

(4) Provision will be made in pump design for expansion of

(a) Casing and rotor relative to one another.

(b) Casing relative to the base.

(c) Pump rotor relative to the shaft of the driver.

(d) Inner and outer casing for double casing pumps.

(5) All rotating parts will be balanced statically and dynamically for all speeds.

(6) Pump design will provide axial as well as radial balance of the rotor at all outputs.

(7) One end of the pump shaft will be accessible for portable tachometer measurements.

(8) Each pump will be provided with a pump warmup system so that when it is used as a standby it can be hot, ready for quick startup. This is done by connecting a small bleed line and orifice from the common discharge header to the pump discharge inside of the stop and check valve. Hot water can then flow back through the pump and open suction valve to the common suction header, thus keeping the pump at operating temperature.

(9) Pump will be designed so that it will start safely from a cold start to full load in 60 seconds in

an emergency, although it will normally be warmed before starting as described above.

(10) Other design criteria should be as forth in Military Specification MIL-P-17552D.

g. Pump drives. For military plants, one steam turbine driven pump may be justified under certain conditions; e.g., if the plant is isolated, or if it is a cogeneration plant or there is otherwise a need for substantial quantities of exhaust steam. Usually, however, adequate reliability can be incorporated into the feed pumps by other means, and from a plant efficiency point of view it is always better to bleed steam 'from the prime mover(s) rather than to use steam from an inefficient mechanical drive turbine.

3-31. Feedwater supply

a. General description.

(1) In general terms, the feedwater supply includes the condensate system as well as the boiler feed system.

(2) The condensate system includes the condensate pumps, condensate piping, low pressure closed heaters, deaerator, and condensate system level and makeup controls. Cycle makeup may be introduced either into the condenser hotwell or the deaerator. For large quantities of makeup as in cogeneration plants, the deaerator maybe preferred as it contains a larger surge volume. The condenser, however, is better for this purpose when makeup is of high purity and corrosive (demineralized and undeaerated). With this arrangement, corrosive demineralized water can be deaerated in the condenser hotwell; the excess not immediately required for cycle makeup is extracted and pumped to an atmospheric storage tank where it will be passive in its deaerated state. As hotwell condensate is at a much lower temperature than deaerator condensate, the heat loss in the atmospheric storage tank is much less with this arrangement.

(3) The feedwater system includes the boiler feed pumps, high pressure closed heaters, boiler feed suction and discharge piping, feedwater level controls for the boiler, and boiler desuperheater water supply with its piping and controls.

b. Unit vs. common system. Multiple unit cogeneration plants producing export steam as well as electric will always have ties for the high pressure steam, the extraction steam, and the high pressure feedwater system. If there are low pressure closed heaters incorporated into the prime movers, the condensate system usually remains independent for each such prime mover; however, the deaerator and boiler feed pumps are frequently common for all boilers although paralleling of independent high pressure heater trains (if part of the cycle) on the feedwater side maybe incorporated if high pressure bleeds on the primer movers are uncontrolled. Each cogeneration feedwater system must carefully be designed to suit the basic parameters of the cycle. Level control problems can become complex, particularly if the cycle includes multiple deaerators operating in parallel.

c. *Feedwater controls.* Condensate pumps, boiler feed pumps, deaerator, and closed feedwater heaters are described as equipment items under other headings in this manual. Feedwater system controls will consist of the following

(1) Condenser hotwell level controls which control hotwell level by recirculating condensate from the condensate pump discharge to the hotwell, by extracting excess fluid from the cycle and pumping it to atmospheric condensate storage (surge) tanks, and by introducing makeup (usually from the same condensate storage tanks) into the hotwell to replenish cycle fluid.

(2) Condensate pump minimum flow controls to recirculate sufficient condensate back to the condenser hotwell to prevent condensate pumps from overheating.

(3) Deaerator level controls to regulate amount of condensate transferred from condenser hotwell to deaerator and, in an emergency, to overflow excess water in the deaerator storage tank to the condensate storage tank(s).

(4) Numerous different control systems are possible for all three of the above categories. Regardless of the method selected, the hotwell and the deaerator level controls must be closely coordinated and integrated because the hotwell and deaerator tank are both surge vessels in the same fluid system.

(5) Other details on instruments and controls for the feedwater supply are described under Section 1 of Chapter 5, Instruments and Controls.

Section VIII. SERVICE WATER AND CLOSED COOLING SYSTEMS

3-32. Introduction

a. Definitions and purposes. Service water supply systems and subsystems can be categorized as follows:

(1) For stations with salt circulating water or heavily contaminated or sedimented fresh circulating water.

(a) Most power stations, other than those with cooling towers, fall into this category. Circulating water booster pumps increase the pressure of a (small) part of the circulating water to a level ade-

quate to circulate through closed cooling water exchangers. If the source is fresh water, these pumps may also supply water to the water treating system. Supplementary sources of water such as the area public water supply or well water may be used for potable use and/or as a supply to the water treating system. In some cases, particularly for larger stations, the service water system may have its pumps divorced from the circulating water pumps to provide more flexibility y and reliability.

(b) The closed cooling water exchangers transfer rejected heat from the turbine generator lube oil and generator air (or hydrogen) coolers, bearings and incidental use to the circulating water sidestream pressurized by the booster pumps. The medium used for this transfer is cycle condensate which recirculates between the closed cooling exchangers and the ultimate equipment where heat is removed. This closed cooling cycle has its own circulating (closed cooling water) pumps, expansion tank and temperature controls.

(2) For stations with cooling towers. Circulating water booster pumps (or separate service water pumps). may also be used for this type of power plant. In the case of cooling tower systems, however, the treated cooling tower circulating water can be used directly in the turbine generator lube oil and generator air (or hydrogen) coolers and various other services where a condensate quality cooling medium is unnecessary. This substantially reduces the size of a closed cooling system because the turbine generator auxiliary cooling requirements are the largest heat rejection load other than that required for the main condenser. If a closed cooling system is used for a station with a cooling tower, it should be designed to serve equipment such as air compressor cylinder jackets and after coolers, excitation system coolers, hydraulic system fluid coolers, boiler TV cameras, and other similar more or less delicate service. If available, city water, high quality well water, or other clean water source might be used for this delicate equipment cooling service and thus eliminate the closed cooling water system.

b. Equipment required—general. Equipment required for each system is as follows:

(1) Service water system

(a) Circulating water booster pumps (or separate service water pumps).

(b) Piping components, valves, specialities and instrumentation.

(2) Closed cooling water system.

(a) Cosed cooling water circulating pumps.

(b) Closed cooling water heat exchangers.

(c) Expansion tank.

(d) Piping components, valves, specialities and instrumentation. Adequate instrumentation (thermometers, pressure gages, and flow indicators) should be incorporated into the system to allow monitoring of equipment cooling.

3-33. Description of major components

a. Service water systerm.

(1) *Circulating water booster (or service water) pumps.* These pumps are motor driven, horizontal (or vertical) centrifugal type. Either two 100-percent or three 50-percent pumps will be selected for this duty. Three pumps provide more flexibility; depending upon heat rejection load and desired water temperature, one pump or two pumps can be operated with the third pump standing by as a spare. A pressure switch on the common discharge line alarms high pressure, and in the case of the booster pumps a pressure switch on the suction header or interlocks with the circulating water pumps provides permissive to prevent starting the pumps unless the circulating water system is in operation.

(2) *Temperature control.* In the event the system serves heat rejection loads directly, temperature control for each equipment where heat is removed will be by means of either automatic or manually controlled valves installed on the cooling water discharge line from each piece of equiment, or by using a by-pass arrangement to pass variable amounts of water through the equipment without upsetting system hydraulic balance.

b. Closed cooling water system.

(1) *Closed cooling water pumps.* The closed cooling water pumps will be motor driven, horizontal, end suction, centrifugal type with two 100-percent or three 50-percent pumps as recommended for the pumps described in *a* above.

(2) *Closed cooling water heat exchangers.* The closed cooling water exchangers will be horizontal shell and tube test exchangers with the treated plant cycle condensate on the shell side and circulating (service) water on the tube side. Two 100-percent capacity exchangers will be selected for this service, although three 50-percent units may be selected for large systems.

(3) *Temperature control.* Temperature control for each equipment item rejecting heat will be similar to that described above for the service water system.

3-34. Description of systems

a. Service water system.

(1) The service water system heat load is the sum of the heat loads for the closed cooling water system and any other station auxiliary systems which may be included. The system is designed to maintain the closed cooling water system supply temperature at 950 For less during normal operation

with maximum heat rejection load. The system will also be capable of being controlled or manually adjusted so that a minimum closed cooling water supply temperature of approximately 55 ‘F can be maintained with the ultimate heat sink at its lowest temperature and minimum head load on the closed cooling water system. The service water system will be designed with adequate backup and other reliability features to provide the required cooling to components as necessary for emergency shutdown of the plant. In the case of a system with circulating water booster pumps, this may mean a crossover from a city or well water system or a special small circulating water pump.

(2) Where cooling towers are utilized, means will be provided at the cooling tower basin to permit the service water system to remain in operation while the cooling tower is down for maintenance or repairs.

(3) The system will be designed such that operational transients (e.g., pump startup or water hammer due to power failure) do not cause adverse effects in the system. Where necessary, suitable valving or surge control devices will be provided.

b. Closed cooling water system.

(1) The closed cooling water coolant temperature is maintained at a constant value by automatic control of the service water flow through the heat exchanger. This is achieved by control valve modulation of the heat exchanger by-pass flow. All equipment cooled by the cooling system is individually temperature controlled by either manual or automatic valves on the coolant discharge from, or by by-pass control around each piece of equipment. The quantity of coolant in the system is automatically maintained at a predetermined level in the expansion tank by means of a level controller which operates a control valve supplying makeup from the cycle condensate system. The head tank is provided with an emergency overflow. On a failure of a running closed cooling water pump, it is usual to provide means to start a standby pump automatically.

(2) The system will be designed to ensure adequate heat removal based on the assumption that all service equipment will be operating at maximum design conditions.

3-35. Arrangement

a. *Service water system.* The circulating water booster pumps will be located as close as possible to the cooling load center which generally will be near the turbine generator units. All service water piping located in the yard will be buried below the frost line.

b. Closed cooling water system. The closed cooling water system exchangers will be located near the turbine generators.

3-36. Reliability of systems

It is of utmost importance that the service and closed cooling water systems be maintained in service during emergency conditions. In the event power from the normal auxiliary source is lost, the motor driven pumps and electrically actuated devices will be automatically supplied by the emergency power source (Chapter 4, Section VII). Each standby pump will be designed for manual or automatic startup upon loss of an operating pump with suitable alarms incorporated to warn operators of loss of pressure in either system.

3-37. Testing

The systems will be designed to allow appropriate initial and periodic testing to:

u. Permit initial hydrostatic testing as required in the ASME Boiler and Pressure Vessel Code.

b. Assure the operability and the performance of the active components of the system.

c. Permit testing of individual components or subsystems such that plant safety is not impaired and that undesirable transients are not present.

Section IX. WATER CONDITIONING SYSTEMS

3-38. Water Conditioning Selection

a. Purpose.

(1) All naturally occuring waters, whether surface water or well water, contain dissolved and possibly suspended impurities (solids) which may be injurious to steam boiler operation and cooling water service. Fresh water makeup to a cooling tower, depending on its quality, usually requires little or no pretreatment. Fresh water makeup to a boiler system ranges from possibly no pretreatment (in the case of soft well water used in low pressure boiler) to ultra-purification required for a typical high pressure boiler used in power generation.

(2) The purpose of the water conditioning systems is to purify or condition raw water to the required quality for all phases of power plant operation. Today, most high pressure boilers (600 psig or above) require high quality makeup water which is usually produced by ion exchange techniques. Toreduce the undesirable concentrations of turbidity and organic matter found in most surface waters, the raw water will normally be clarifed by coagulation and filtration for pretreatment prior to passing to the ion exchangers (demineralizers). Such pretreat-

ment, which may also include some degree of softening, will normally be adequate without further treatment for cooling tower makeup and other general plant use.

b. Methods of conditioning.

(1) Water conditioning can be generally categorized as' 'external" treatment or' 'internal" treatment. External treatment clarifies, softens, or purifies raw water prior to introducing it into the power plant fluid streams (the boiler feed water, cooling tower system, and process water) or prior to utilizing it for potable or general washup purposes. Internal treatment methods introduce chemicals directly into the power plant fluid stream where they counteract or moderate the undesirable effects of water impurities. Blowdown is used in the evaporative processes to control the increased concentration of dissolved and suspended solids at manageable levels.

(2) Some of the methods of water conditioning are as follows:

(a) Removal of suspended matter by sedimentation, coagulation, and filtration (clarification).

(b) Deaeration and degasification for removal of gases.

(c) Cold or hot lime softening.

(d) Sodium zeolite ion exchange.

(f) Choride cycle dealkalization.

(g) Demineralization (ultimate ion exchange).

(h) Internal chemical treatment.

(i) Blowdown to remove sludge and concentration buildups.

c. Treatment Selection. Tables 3-13, *3-14,* and 3-15 provide general guidelines for selection of treatment methodologies. The choice among these is an economic one depending vitally on the actual constituents of the incoming water. The designer will make a thorough life cycle of these techniques in conjunction with the plant data. Water treatment experts and manufacturer experience data will called upon.

Section X. COMPRESSED AIR SYSTEMS

3-39. Introduction

a. Purpose. The purpose of the compressed air systems is to provide all the compressed air requirements throughout the power plant. The compressed air systems will include service air and instrument air systems.

b. Equipment required-general. Equipment required for a compressed air system is shown in Figures 3-14 and 3-15. Each system will include

(1) Air compressors.

(2) Air aftercoolers.

(3) Air receiver.

(4) Air dryer (usually for instrument air system only).

(5) Piping, valves and instrumentation.

c. Equipment served by the compressed air systems.

(1) Service (or plant) air system for operation of tools, blowing and cleaning.

(2) Instrument air system for instrument and control purposes.

(3) Soot blower air system for boiler soot blowing operations.

3-40. Description of major components

a. Air compressors. Typical service and instrument air compressor? for power plant service are single or two stage, reciprocating piston type with electric motor drive, usually rated for 90 to 125 psig discharge pressure. They may be vertical or horizontal and, for instrument air service, always have oilless pistons and cylinders to eliminate oil carryover. Non-lubricated design for service air as well as instrument air will be specified so that when the former is used for backup of the latter, oil carryover will not be a problem. Slow speed horizontal units for service and instrument air will be used. Soot blower service requirements call for pressures which require multi-stage design. The inlet air filter-silencer will be a replaceable dry felt cartridge type. Each compressor will have completely separate and independent controls. The compressor controls will permit either constant speed-unloaded cylinder control or automatic start-stop control. Means will be provided ***in*** a multi-compressor system for selection of the' 'lead" compressor.

b. Air aftercooler. The air aftercooler for each compressor will be of the shell and tube type, designed to handle the maximum rated output of the compressor. Water cooling is provided except for relatively small units which may be air cooled. Water for cooling is condensate from the closed cooling system which is routed counter-flow to the air through the aftercooler, and then through the cylinder jackets. Standard aftercoolers are rated for 95 "F. maximum inlet cooling water. Permissive can be installed to prevent compressor startup unless cooling water is available and to shut compressor down or sound an alarm (or both) on failure of water when unit is in operation.

c. Air receiver. Each compressor will have its own receiver equipped with an automatic drainer for removal of water.

d. Instrument air dryer. The instrument air dryer

Table 3-13. General Guide for Raw Water Treatment of BoilerMakeup

St earn Pressure (psig)	Silica reg./l.	Alkalinity reg./l. (as $CaCO_3$)	Water Treatment
up to 450	Under 15	Under 50	Sodium ion exchange.
		Over 50	Hot lime-hot zeolite, or cold lime zeolite, or hot lime soda, or sodium ion exchange plus chloride anion exchange.
	Over 15	Over 50	Hot lime-hot zeolite, or cold lime-zeolite, or hot lime soda.
450 to 600	Under 5	Under 50	Sodium ion exchange plus chloride anion exchange, or hot lime-hot zeolite.
		Over 50	Sodium plus hydrogen ion exchange, or cold lime-zeolite or hot lime-hot zeolite.
	Above 5		Demineralizer, or hot lime-hot zeolite.
600 to 1000	------- 'Any Water	-------	Demineralizer.
1000 & Higher	------- Any Water	-------	Demineralizer.

NOTES :

(1) Guide is based on boiler water concentrations recommended in the American Boiler and Affiliated Industries "Manual of Industry Standards and Engineering Information."
(2) Add filters when turbidity exceeds 10mg./l.
(3) See Table 3-15 for effectiveness of treatments.
(4) reg./l. = p.p.m.

Source: Adapted from NAVFAC DM3

Table 3-14. Internal Chemical Treatment.

Corrosive Treatment Required	Chemical
Maintenance of feedwater pH and boiler water alkalinity for scale and corrosion control.	Caustic Soda Soda Ash Sulfuric Acid
Prevention of boiler scale by internal softening of the boiler water.	Phosphates Soda Ash Sodium Aluminate Alginates Sodium Silicate
Conditioning of boiler sludge to prevent adherence to internal boiler surfaces.	Tannins Lignin Derivatives Starch Glucose Derivatives
Prevention of scale from hot water in pipelines, stage heaters, and economizers.	Polyphosphates Tannins Lignin Derivatives Glucose Derivatives
Prevention of oxygen corrosion by chemical deaeration of boiler feedwater.	Sulfites Tannins Ferrus hydroxide Glucose Derivatives Hydrazine Ammonia
Prevention of corrosion by protective film formation.	Tannins Lignin Derivatives Glucose Derivatives
Prevention of corrosion by condensate.	Amine Compounds Ammonia
Prevention of foam in boiler water.	Polyamides Polyalkylene Glycols
Inhibition of caustic embrittlement.	Sodium Sulfate Phosphates Tannins Nitrates

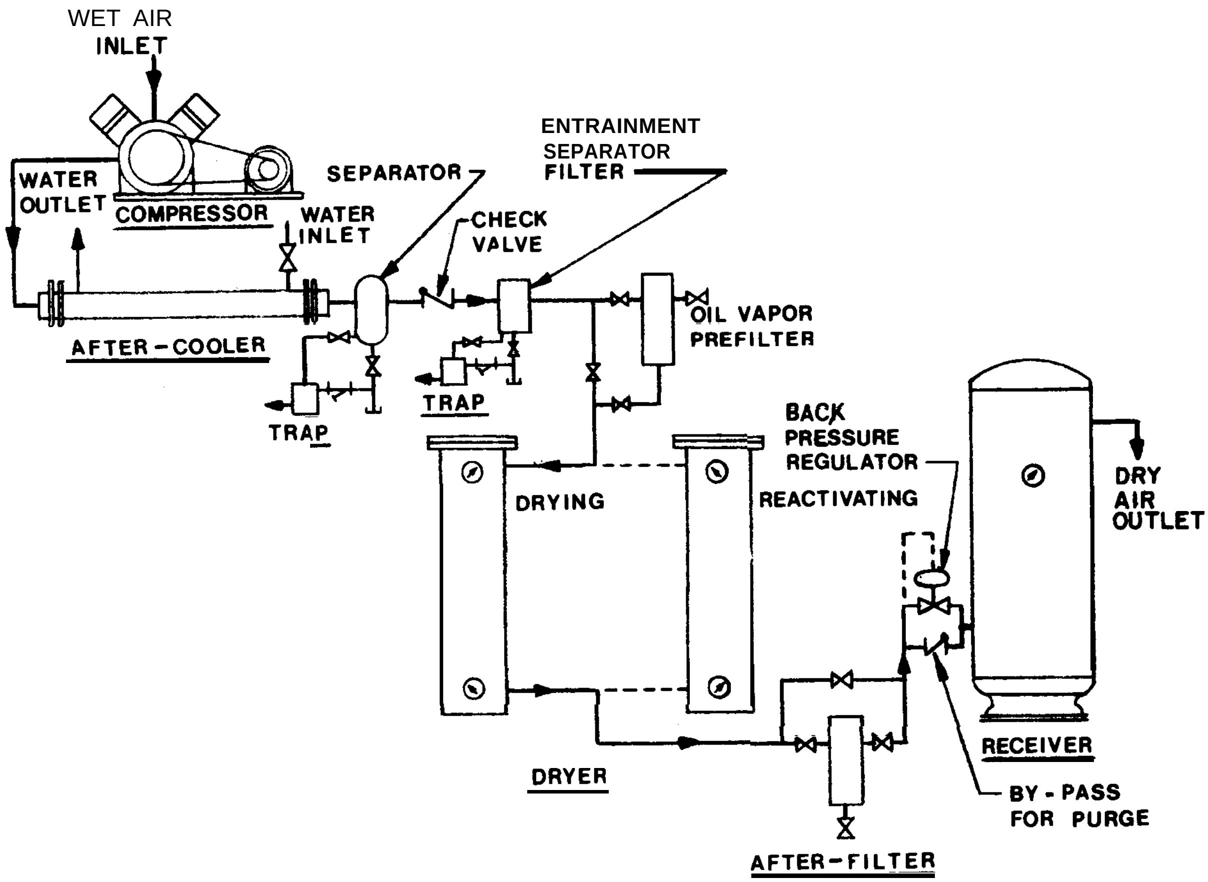
U.S. Army Corps of Engineers

Table 3-15. Effectiveness of Water Treatment

	Average Analysis of Effluent				
Treatment	Hardness (as CaCO) mg./l.	Alkalinity (as CaCO) mg./1.	co mg./l. in Steam	Dissolved Solids mg./l.	Silica mg./l.
Cold Lime-Zeolite	o to 2	75	Medium High	Reduced	8
Hot Lime Soda	17 to 25	35 to 50	Medium High	Reduced	3
Hot Lime-Hot Zeolite	o to 2	20 to 25	Low	Reduced	3
Sodium Zeolite	o to 2	Unchanged	Low to High	Unchanged	Unchanged
Sodium Plus Hydrogen Zeolite	o to 2	10 to 30	Low	Reduced	Unchanged
Sodium Zeolite Plus Chloride Anion Exchanger	o to 2	15 to 35	Low	Unchanged	Unchanged
Demineralizer	o to 2	o to 2	o to 5	o t o 5	Below 0.15
Evaporator	o to 2	o to 2	o to 5	o t o 5	Below 0.15

NOTE: (1) reg./l. = p.p.m.

Source: NAWFAC DM3



Courtesy of Pope, Evans and Robbins (Non-Copyrighted)

Figure 3-14. Typical compressed airsystem.

will be of the automatic heat reactivating, dual chamber, chemical desiccant, downflow type. It will contain a prefilter and afterfilter to limit particulate size in the outlet dried air. Reactivating heat will be provided by steam heaters.

3-41. Description of systems

a. General. The service (or plant) air and the instrument air systems may have separate or common compressors. Regardless of compressor arrangement, service and instrument air systems will each have their own air receivers. There will be isolation in the piping system to prevent upsets in the service air system from carrying over into the vital instrument air system.

b. Service air system. The service air system capacity will meet normal system usage with one compressor out of service. System capacity will include emergency instrument air requirements as well as service air requirements for maintenance during plant operation. Service air supply will include work shops, laboratory, air hose stations for maintenance use, and like items. Air hose stations should be spaced so that air is available at each piece of equipment by using an air hose no longer than 75 feet. Exceptions to this will be as follows:

(1) The turbine operating floor will have service air stations every 50 feet to handle air wrenches used to tension the turbine hood bolts.

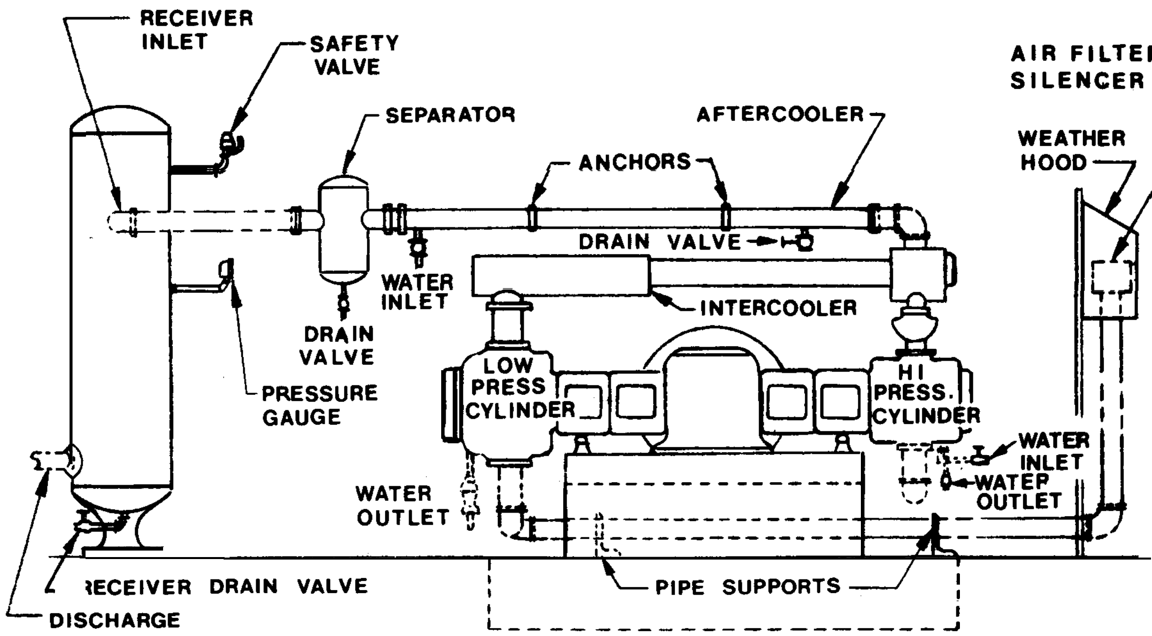
(2) No service air stations are required in the control room and in areas devoted solely to switchgear and motor control centers.

(3) Service air stations will be provided inside buildings at doors where equipment or supplies may be brought in or out.

c. *Instrument air sys tern.* A detailed analysis will be performed to determine system requirements. The analysis will be based on:

(1) The number of air operated valves and dampers included in the mechanical systems.

(2) The number of air transmitters, controllers and converters.



Courtesy of Pope, Evans and Robbins (Non-Copyrighted)

Figure 3-15. Typical arrangement of air compressor and accessories.

(3) A list of another estimated air usage not in-eluded in the above items.

d. Piping system.

(1) *Headers.* Each separate system will have a looped header to distribute the compressed air, and for large stations a looped header will be provided at each of the floor levels.

(2) *Instrument air reserve.* In instances where short term, large volume air flow is required, local air receivers can be considered to meet such needs and thereby eliminate installation of excessive compressor capacity. However, compressor must be sized to recharge the receivers while continuing to supply normal air demands.