or more of these principles is employed in the various types of mill. The energy required for grinding is a function of the fineness produced and the hardness of the coal.

Energy Required. There is no generally accepted theory of the relation of energy to fineness of crushing. However, Rittinger's law, "The work to produce material of a given size from a large size is proportioned to the new surface produced," is a closer approximation than Kick's law, which states, "The energy required to effect crushing or pulverizing is proportional to the volume reduction of the particle." The hardness of the coal, or grindability, is a relative measure of the energy required for crushing. For lack of an exact law governing the energy required for crushing and a simple method for determining new surface, it has been desirable to predict mill-grinding performance on the basis of grindability and 200-mesh sieve fineness rather than on the surface area produced.

The Hardgrove method of grindability determination has been generally used because it is a direct measure of the 200-mesh fineness produced by a standard unit of energy input. (See also Section 2.) Principal features of the Hardgrove apparatus for fineness determination are shown in Fig. 1. Grindability is determined by placing a 50-gram sample of air-dried coal, sized to minus 16 and plus 30 mesh, in the mortar of the test machine. After turning the machine through 60 revolutions the sample is removed and screened. The quantity passing a 200-mesh sieve is used to determine the Hardgrove grindability index by the following empirical formula:

\[ G = 6.93W + 13 \]

where \( W \) is the weight in grams of the sample that passes a 200-mesh sieve.

The grindability index of various coals permit the manufacturer to predict the performance of a particular type mill with the coals to be used. Table 1 shows the grindability of several typical coals and Fig. 2 shows the effect of grindability on the grinding capacity of a particular mill. For additional data on coals, see p. 2-26.

The ability of a particular mill to dry coal depends on the heat added in the form of hot air or gas, and the energy that is converted to heat due to grinding. To utilize properly the heat added by hot air or gas there must be intimate mixing of the air and coal, and the incoming feed should be rapidly mixed with the dryer coal being pulverized or contained in the circulating load. Different types of pulverizers show considerable variation in drying performance. The ability to use high gas or air temperatures entering the mill rather than large quantities of gas or air also varies considerably with mills of different types. To enable high inlet drying temperatures there must be high velocities of gas or air. Dead pockets within the mill are a fire hazard. High inlet temperatures and consequent low gas or air quantities are desirable because of lower power requirements for handling a smaller weight of air, lower tempering air requirements (thus higher efficiency on direct-fired units supplied with gas-air heaters), and lower primary air quantities (thus better burner performance on direct-fired units).

The extent to which drying must be accomplished in a mill depends primarily on the type of coal. In general, it is necessary to remove all surface moisture, leaving only the inherent moisture in the pulverized coal. Failure to remove the surface moisture limits the capacity of the mill considerably beyond the reduction of capacity that occurs due to moisture (see Fig. 2) in the raw coal, even when it is removed in the milling system.

The classification of coal size is determined by screening. The screens generally used and their corresponding sizes are shown in Table 2.

Classification of coal in mills generally is accomplished by means of air separation. Oversize particles are separated by a change in direction and returned to the grinding chamber. Means for application and control of this air separation principle differ for various mills. The classifier performance has a direct bearing on the grinding power required to obtain satisfactory combustion results. Satisfactory combustion results require
## Table 1. Grindability of Typical Coals

<table>
<thead>
<tr>
<th>State and County</th>
<th>Mining District or Seam</th>
<th>Grindability (Hardgrove)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alabama, Jefferson</td>
<td>Walker</td>
<td>Mary Lee</td>
</tr>
<tr>
<td>Arkansas, Franklin</td>
<td>Lee</td>
<td>Mary Lee</td>
</tr>
<tr>
<td>Colorado, El Paso</td>
<td>Las Animas</td>
<td>Deming</td>
</tr>
<tr>
<td>Illinois, Franklin</td>
<td>Williamson</td>
<td>Colorado Springs</td>
</tr>
<tr>
<td>Indiana, Clay, Greene, Vigo</td>
<td>Greene, Sullivan</td>
<td>Trinidad</td>
</tr>
<tr>
<td>Iowa, Appanoose, Wayne</td>
<td>Polk</td>
<td>Franklin</td>
</tr>
<tr>
<td>Kansas, Cherokee</td>
<td>Leavenworth</td>
<td>Williamson</td>
</tr>
<tr>
<td>Kentucky, Floyd, Letcher, Pike, Perry, Breathitt</td>
<td>Knott, Letcher</td>
<td>Springfield</td>
</tr>
<tr>
<td>Kentucky, Union, Webster</td>
<td>Harlan</td>
<td>Belleville-Sauton</td>
</tr>
<tr>
<td>Maryland, Allegany</td>
<td>Eastern Interior</td>
<td>Peoria</td>
</tr>
<tr>
<td>Michigan, Saginaw</td>
<td>Saginaw</td>
<td>Mystic</td>
</tr>
<tr>
<td>Missouri, Adair</td>
<td>Harlan</td>
<td>No. 3</td>
</tr>
<tr>
<td>Montana, Carbon</td>
<td>Hazard No. 4</td>
<td>No. 4</td>
</tr>
<tr>
<td>New Mexico, McKinley</td>
<td>Harlan</td>
<td>No. 5</td>
</tr>
<tr>
<td>North Dakota, Most Middle &amp; Western Counties</td>
<td>Harlan</td>
<td>No. 6</td>
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<tr>
<td>Ohio, Morgan, Noble, Washington, Harrison</td>
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<tr>
<td>Ohio, Morgan, Noble, Washington, Harrison</td>
<td>Eastern Interior</td>
<td>Mystic</td>
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</tbody>
</table>
a minimum quantity of plus 50-mesh material; a large quantity of minus 200-mesh and superfine material is not necessary, however. Good classification is thus measured by the retention of a minimum quantity of 50-mesh material with a given quantity of minus 200-mesh material. Figure 2 illustrates the grinding capacity variation with 200-mesh fineness on a particular mill.

Table 2. Standard Screen Sizes

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Inches</th>
<th>Millimeters</th>
<th>Mesh</th>
<th>Inches</th>
<th>Millimeters</th>
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</table>

Types of Pulverizer generally used are the ball, impact, ring roll, and ball race.

Ball mills consist of a horizontally rotating cylinder less than half full of balls of various diameters. The speed of rotation is approximately 20 rpm. Balls carried up the periphery in the direction of rotation continually cascade toward the center. Coal mixed with the ball charge is pulverized by impact, attrition, and crushing. Hot air passed through the mill dries the coal and removes the fines. A classifier is used in some designs to regulate the fineness of the finished product by returning the coarser particles. Low maintenance and quick response to change in output rate are characteristics of this mill. Power requirements, particularly at reduced capacity, are relatively high. Space requirements are relatively large for a given capacity, and with wet coal there is an extreme falling off in capacity. The large storage and heat capacity makes this type of mill poorly adapted for intermittent operation or quick starting.

Impact mills consist of a series of hammers or lugs revolving at high speed in an enclosed chamber. Grinding is by impact and attrition. Air passing through the mill dries the coal and carries away the fines. A means of classification is generally provided for returning oversize particles. This type is compact, low in cost, and may be built in very small sizes. It is well adapted to drying, because there is intimate mixing of air and coal. Maintenance and power consumption are relatively high, and it is difficult to maintain uniform fineness over the life of the wearing parts. The small storage capacity makes this type well adapted to quick starting and intermittent operation.

The ring-roll and ball-race mill principle is shown in Figs. 3 and 4. These mills pulverize by passing the coal between two surfaces, one rolling over the other. Grinding is accomplished by crushing and attrition. These types have low power consumption, are compact, maintain fineness over the life of the wearing parts, and handle wet coal with only a small reduction in capacity.

A unique application of this grinding principle is the Raymond Bowl Mill, shown in Fig. 5. The roll is restrained from coming in direct contact with the ring, so that there is no metal-to-metal contact. The grinding ring is rotated with the bowl at approximately 1200 ft per min. Stationary spring-loaded rolls are set with a travel-limit bar so that they do not come in contact with the grinding ring. A centrifugal-type classifier with adjustable inlet vanes is mounted directly over the center of the bowl. Raw coal fed to the bowl is thrown by centrifugal force to the face of the grinding ring, where it is passed under the spring-loaded rolls, which are free to rotate, thus grinding the coal as it is passed between the ring and the roll. As the coal is discharged over the rim of the grinding ring, it is thrown into an annular air passage where oversize particles are deflected back into the bowl. Pyrites and tramp iron are dropped to the bottom of the air inlet chamber, and the fines mixed with them are carried up by the hot air stream to the classifier inlet vanes. The coal deflected back into the bowl is again passed under the rolls for further grinding. Pyrites and tramp iron dropped to the bottom of the air chamber are discharged through
a discharge spout by sweeps. Fines entering the classifier are separated by the centrifugal motion imparted by the inlet vanes, the oversize particles being returned to the center of the bowl to be mixed with the raw coal. The final ground, dried, and classified product is taken from the classifier outlet to the exhauster inlet.

![Raymond Bowl Mill](image)

**Fig. 5. Raymond Bowl Mill.**

Evaluation of the mill best suited to a particular pulverized coal installation should consider the following factors: (1) Initial cost. (2) Power requirements at full and at partial load. (3) Maintenance cost and frequency. (4) Drying ability, limitation of drying temperatures. (5) Fineness and classification control. (6) Response to load changes. (7) Space requirements. (8) Quietness. (9) Ability to start and stop rapidly. (10) Physical limitations, consisting of (a) ease and reliability of raw coal supply to grinding elements, (b) lubrication ease and reliability, and (c) starting after trip-off. (11) Reliability. (12) Controllability—drying, fineness, output. (13) Accessibility and simplicity. (14) Rejection of pyrites. (15) Ability to handle foreign material.

### 28. PULVERIZED COAL SYSTEMS

Pulverized coal may be utilized in a *storage system* or in a *direct-fired system*. The latter system is characterized by immediate supply of coal to the burners and furnace as it is ground, with no part of it diverted to storage bins.

The *bin or storage system* has the advantage of operating flexibility, and permissible arrangement and location of equipment. This system permits preparation of coal during off-peak load hours at a constant (maximum) mill output rate. A few large-capacity pulverizers may be used without sacrificing flexibility. The ability to keep to a minimum the quantity of primary or carrier air used to convey the pulverized coal to the furnace is of distinct advantage in securing stability and range of operation of the fuel-burning equipment. These advantages are more than offset, in most cases, by the considerably higher equipment cost, the complication of venting the drying air or gas, transporting and storing the pulverized fuel, feeding the pulverized coal, and maintaining and operating the additional equipment. Justification of the storage system is difficult, except where low-volatile, hard-to-burn fuels, such as anthracite, must be utilized. A typical equipment arrangement for a storage system is illustrated in Fig. 6.
The direct-fired system, for which a typical equipment arrangement is illustrated in Fig. 7, is by far the most common arrangement. The advantages of this system are lower initial cost, simplicity of operation, and compactness of equipment. This system requires intelligent coordination in selecting milling, burning, and steam-generating equipment to obtain a reasonable degree of flexibility. The coordination of equipment must represent a satisfactory compromise between simplicity, operating range, types of coals (i.e., moisture, grindability, and volatile matter) to be burned, reliability, overall cost, efficiency, and excess milling and drying capacity.

FUEL-BURNING EQUIPMENT.

The function of fuel-burning equipment is to introduce fuel and air into a furnace in such a way that stability of ignition and substantially complete combustion with minimum excess air are obtained. For successful performance fuel-burning equipment must be designed in such a way that the following conditions can be obtained: (1) Uniform distribution of excess air and temperature at furnace outlet. (2) A means of ignition point and flame-shape control. (3) Freedom from localized slag deposition. (4) Protection against overheating and excessive wear of burner parts. (5) Accessibility for maintenance and adjustment.

Stability and Range. Design factors which control the stability of ignition are those which promote rapid supply of heat to fuel particles as they enter the furnace. Heat to evaporate moisture, distill off volatile matter, and raise the temperature to the kindling point must be supplied to each coal particle by the combustion process of the preceding fuel supply. Thus, to promote rapid ignition, stability, and wide range of operation, all essentially synonymous, the following design conditions can be utilized: (1) A furnace gas flow pattern to promote supply of heat to the incoming coal as it leaves the fuel nozzle. (2) A flow pattern of the fuel leaving the fuel nozzle which promotes low velocity eddies of some of the coal particles, allowing them to absorb heat before dilution by a large mass of combustion air. (3) A low relative quantity and high temperature of the air used to transport the fuel. (4) A high degree of fineness. (5) A high concentration of heat release close to the fuel nozzles. (6) A low heat absorption of the furnace surface in the vicinity of the fuel nozzles to permit a higher temperature for heating the incoming fuel.

In practice it is not usually considered necessary or desirable to obtain all these conditions; in some designs they may even be detrimental to the overall performance, because of deposition of an excessive amount of slag, production of a nonuniform furnace heat distribution, the necessity for utilizing uneconomically high fuel-air pressures, or high burner maintenance.

Completeness of Combustion. Design factors which control the completeness of combustion are those which contribute to the intimate mixing of burning fuel particles with the available oxygen in the combustion air. The importance of mixing is progressively
greater as the combustion process nears completion and the oxygen concentration becomes low, so that inert gas shields the oxygen from the fuel.

Design conditions that promote intimate mixing of fuel and oxygen are: (1) Utilization of high air velocities penetrating the burning fuel streams. (2) Impingement of burning fuel streams upon each other. (3) Utilization of numerous individual equi-quantity fuel and air streams. (4) Utilization of cyclonic gas movement within the furnace chamber.

Mixing. Most types of firing utilize several burners in a furnace to obtain better mixing and more effective utilization of furnace volume and heating surface. The effectiveness of mixing with multiple burners is dependent on the accuracy of metering fuel and air to each nozzle, unless there is a high degree of mixing of the various streams from each nozzle with each other. The use of riffler-type distributors is one commonly used method of subdividing a stream of primary air-coal mixture for more than one burner nozzle. Secondary-combustion air is subdivided by maintaining equal pressures across ports of equal area.

Types of Firing. Fundamentally there are two basic types of firing. In one, the furnace volume is used as a mixing chamber for the air and fuel from all burners. In the other, individual burner nozzles have individual air-supply systems, necessitating an accurate air- and fuel-metering means for each burner assembly.

The tangential burner (Fig. 8) is representative of the first type, whereas horizontal (Fig. 9) and vertical burners (Fig. 10) are examples of the second type. Horizontal burners, where installed in opposite walls, more nearly resemble the first type of firing. However, if more than one burner is installed in each wall, accurate control of the air and fuel to each opposed-burner couplet becomes as important as in the second type of firing.

The tangential-type burner, Fig. 8, is located in each of the four corners of a furnace, and its air and coal streams are directed tangent to a circle in the center of the furnace. The velocity of these streams produces a rotary motion of the gas within the furnace. Thus the gas-flow pattern from each burner nozzle aids ignition of fuel from succeeding burner nozzles. Scrubbing and impinging action of the gas, air, and fuel streams assures maximum turbulence, so that the entire furnace cross section is uniformly and effectively utilized. Because this type of burner is essentially a series of straight-shot nozzles, furnace outlet-temperature control can be accomplished by vertically adjusting the angle of nozzle discharge, thereby utilizing more or less of the furnace heating surface. The ability to control the furnace outlet gas temperature permits a more conservative furnace design for steam-temperature control at partial ratings. Figure 11 illustrates the application of this principle.

The horizontal-type burner, Fig. 9, consists of a central coal nozzle concentric with a throat and a series of adjustable air-admission vanes. Primary air and coal are admitted tangentially to the coal nozzle, and peripheral fuel distribution is obtained at the nozzle discharge by internal ribs and vanes. The secondary air, admitted through adjustable vanes, is given a whirling motion in the same direction as the primary air-coal mixture. Turbulent mixing of the primary and secondary air streams is obtained in the burner throat after partial ignition of the coal. Flame shape and ignition point are readily controlled by the adjustable-vane position.

The vertical type burner, Fig. 10, has a straight-shot burner nozzle firing vertically downward in the furnace, with the main supply of combustion air being furnished through wall ports along the path of the flame travel. The flow pattern in the furnace requires the gas to pass back up the furnace adjacent to the burner nozzle locations. Ignition point
and vertical flame travel are controlled by primary air pressure, quantity and velocity (of air admitted around the coal nozzles) and by selective elevation of the air admitted through the front wall ports.

Completeness of combustion for any burner is influenced by the fineness of pulverization and by the quantity of excess air. In practice the loss due to unburned fuel ranges from 0.1 to 1.5% of the fuel supplied to the furnace. The approximate rate at which this loss varies with excess air and fineness of pulverization is shown in Fig. 12.

The range of operation of pulverized fuel burners varies considerably with many design factors, as well as with the type of coal. The range generally obtained may be from 2 : 1 to 6 : 1, with all burners in service in a particular furnace. By shutting off some of the burners, the actual range of heat input into a furnace may be controlled successfully in some cases over a range as great as 20 : 1.

Fly-ash Removal (see also p. 7-94). The removal of fly ash from the gas before it is discharged into the atmosphere is sometimes necessary. In many cases, to meet local ordinances, fly-ash removal has paid for itself on the basis of the increased life and availability of the induced draft fan through elimination of abrasive material before the gas enters the fan. The quantity and the character of the ash discharge vary with the quantity of ash in the coal, the completeness of combustion, type of furnace, and fineness of pulverization. In general, approximately 50% of the ash initially in the coal will be collected in a wet- or slugging-bottom furnace, whereas approximately 20% of the ash is caught in a dry-bottom furnace. The boiler-pass hoppers collect approximately 5% of the total ash. The remaining ash is entrained in the gas with the unburned carbon left after incomplete combustion. This ash plus unburned carbon generally represents a dust loading of approximately 2 to 5 grains per cubic foot in the gas leaving the boiler.
A typical screen analysis of ash leaving the boiler would be:

<table>
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<tr>
<th>Size</th>
<th>Percentage</th>
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<tr>
<td>+100 microns</td>
<td>0.25%</td>
</tr>
<tr>
<td>+74</td>
<td>0.8</td>
</tr>
<tr>
<td>+44</td>
<td>5</td>
</tr>
<tr>
<td>+30</td>
<td>13</td>
</tr>
<tr>
<td>+20</td>
<td>29</td>
</tr>
<tr>
<td>+10</td>
<td>66</td>
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**Mechanical or electrical methods** may be used for removing this ash. The mechanical method utilizes the change-of-direction principle as obtained in a cyclone separator; the electrical method uses the attraction and repulsion of ionized particles.

The limitation of the mechanical collector is the inability to separate the small and submicron sizes. Thus a collection efficiency of 80 to 90%, is about the upper limit that can be anticipated for typical fly ash, using a gas pressure drop of approximately 4 in. of water. One limitation of the electrical precipitator is its inability to ionize carbon in the fly ash. The necessity for low velocities results in large sizes. Collection efficiencies up to 96% are obtainable with extremely low velocities and frequent rapping of plates.

**Fig. 12.** Combustion loss as a function of excess air and fuel fineness

In some instances it has been desirable to install both electrical and mechanical collectors in series. The larger particles, with high carbon content, are caught in the mechanical collector with a relatively low pressure drop and the smaller particles are caught in the electrical precipitator.

**EXPLOSION AND FIRE HAZARDS OF PULVERIZED FUEL.** In the handling, pulverisation, storage, and burning of pulverized fuel there are recognized hazards of fire and explosion that must be considered in design and operation of the equipment. These hazards have been acknowledged by the National Board of Fire Underwriters in a pamphlet which the manufacturers of pulverized fuel equipment use in determining design strength and arrangement of equipment.
FLY ASH

FLY-ASH COLLECTION

By R. B. Foley

29. FLY ASH

SOLID PARTICLES IN COMBUSTION GASES are classified as smoke, fumes, and dusts.

Smoke is unburned carbon particles of extremely small size, 0.001 to 0.25 micron. (Note: 0.25 micron = 0.00001 in., approximately.)

Fumes are condensed dispersoids 0.1 to 1.0 micron in size.

Dusts are solid particles larger than 1 micron in diameter. The dust resulting from combustion of solid fuels, consisting of ash and unburned carbon particles carried in the flue gases, is usually called fly ash. Although the larger carbon particles are sometimes called cinders, the term fly ash is commonly used to include all the solid particles larger than 1 micron.

COMPOSITION OF FLY ASH. The chemical components found most frequently in fly ash are listed in Table 1. The amount of unburned carbon varies from as low as 1% to more than 80%, depending on type of firing, furnace design, boiler load, and operating conditions. The percentages of various noncombustible oxides depend on the composition of the ash in the coal burned, but usually nearly half is silicon dioxide which, with ferric and aluminum oxide, accounts for more than 90% of the ash. In addition to the components listed, there are sometimes small quantities of alkalies and other metallic oxides (Ref. 1).

PHYSICAL CHARACTERISTICS of fly-ash particles, such as size, shape, and weight, are the properties which influence their behavior in the flue gases and the atmosphere. The carbon may be present as fairly large coke particles or as fine soot particles. The ash consists of both fused and crystalline particles. In the fly ash from pulverized-coal-fired boilers a large percentage of the ash particles is fused spheres, many of them hollow shell-like particles.

Some of the methods used to determine physical characteristics measure only the size of the particles. Standard sieves or screens are most commonly used to determine the size analysis of fly ash. Pulverized fuel fly ash, however, usually contains 60 to 90% of particles that pass through the finest standard sieve, which has 400 meshes per linear inch.

The two methods most commonly used for analyzing fly ash in the subsieve range, elutriation and sedimentation, depend on the aerodynamic characteristics of the particles, and thus measure the combined effect of all three of their physical properties on their behavior in flue gas or the atmosphere. In both these methods the fly-ash particles are separated into fractions dependent on their terminal velocity. The terminal velocity of a dust particle is the ultimate velocity it will obtain in free fall through a given quiescent fluid; it is generally expressed in terms of the particle velocity in standard air.

Although it has been common practice to express the results of analyses made by the elutriation and sedimentation methods in terms of particle diameters in microns, the transition from the terminal velocities as measured to particle size is based on the assumptions that the particles are solid spheres of uniform density. Since fly ash is a heterogeneous mixture of materials of widely different densities (see Table 1) and since only a

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<th>Table 1. Chemical Components of Fly Ash</th>
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<td>Iron oxide</td>
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<td>Ferric oxide</td>
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<td>Aluminum oxide</td>
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<td>Calcium oxide</td>
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<tr>
<td>Magnesium oxide</td>
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<tr>
<td>Sulfur trioxide</td>
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<tr>
<td>Phosphorous pentoxide</td>
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small percentage of the particles is solid spheres, these assumptions are not accurate. Moreover, it is terminal velocity of the particles, and not their size, which governs their movement in the flue gases, in the fly-ash collector, and in the atmosphere after they leave the stack.
It must be remembered, therefore, that when fly-ash analyses, made by these methods, are expressed in terms of micron size, the values are not the true size of the particles but the equivalent size of solid spherical particles of a specified density. Although the various particles have different densities the average specific gravity of most composite fly-ash samples is approximately 2.0; this value is usually used in computing the equivalent micron size. Figure 1 shows the relationship between terminal velocity and equivalent micron size for various specific gravities.

![Graph](image)

**Fig. 1.** Relationship between particle size and terminal velocity in standard air for solid spherical particles of various specific gravities. (Courtesy of American Blower Corp.)

**INFLUENCE OF TYPE OF FIRING.** Although many factors affect both quality and quantity of fly ash, the one of most influence is the method of firing the coal.

**Pulverized-coal-fired boilers** usually produce more and finer fly ash than any other type of firing. Since the coal is fine and burned in suspension in the furnace, a much higher percentage of the ash is carried out with the flue gases. At design rating, 70 to 85% of the ash may leave the furnace with the gases from the dry-bottom type of furnace; in wet-bottom or slag-tap furnaces 50 to 65% of the total ash is contained in the fly ash. The fly ash may contain up to 30 or 40% of unburned carbon, but generally the carbon content is much lower than that from stoker-fired furnaces. Although the degree of pulverization may materially affect the fineness of the fly ash, the total quantity is not affected as much as might be expected since the finer pulverization usually results in more complete burning and less unburned carbon. Depending on the percentage of ash in coal, furnace design, and operating conditions, the concentration of fly ash in the flue gases from a pulverized fuel-fired boiler may vary from 1 to 5 grains per cubic foot at the normal maximum boiler load. The analysis of the fly ash varies considerably with the boiler load and is also dependent on other factors such as fineness of the coal, type of coal, furnace velocities, and percentage of unburned carbon. Generally the unburned carbon particles are the largest; the finest particles are nearly all ash. Figure 2 shows graphically the usual ranges of analyses of pulverized fuel fly ash for maximum load operation. The center curve B represents an average analysis; curves A and C represent more extreme analyses.

**Stoker-fired Boilers.** Although the percentage of ash carried with the flue gas is usually considerably lower for stoker-fired boilers, there is generally more unburned carbon in the fly ash. The wide range of variables which affect the fly ash, such as type and size of coal, grate area, furnace volume and design, and operating conditions, causes such great differences in the fly ash that statements of analysis and concentration can be only very general.

With underfeed stokers, up to one-third of the ash may be carried out with flue gases at normal ratings. The fly ash concentration may vary from a few tenths of a grain per cubic foot when grate areas and furnace volumes are generous to two grains or more when the coal burned per square foot of grate area and furnace heat releases are high.
Since finer coals are often burned on chain-grate stokers the concentration may be somewhat higher, but the amount of fly ash varies considerably with the size of coal burned and the amount of unburned carbon.

With the spreader stoker, as much as 50% of the ash may be carried out of the furnace, because the fines are burned in suspension. The percentage of unburned carbon also materially affects the total fly-ash concentration. Owing to the wide variety of coals that may be burned on this type of stoker, and its flexibility in adaptation to furnace and boiler design, the concentration in the gases at the furnace exit under maximum operating loads may vary from about half a grain to as much as four grains per cubic foot.

In many stoker-fired installations, hoppers are provided under the rear boiler passes, and a large percentage of the coarser, heavier particles is trapped out of the gases at this point. The material caught in these hoppers is usually high in carbon content since 70 to 90% of the unburned carbon in the fly ash leaving the furnace is usually in the portion that stays on a 200-mesh screen.

![Fig. 2. Analysis of fly ash from pulverized coal fired boilers.](image)

![Fig. 3. Analysis of fly ash from stoker-fired boilers. A and B—underfeed stokers; C and D—spreader stokers.](image)

In many stoker-fired installations, especially when spreader stokers are used, fly ash is reinjected into the furnace. Since the carbon content in the material caught in the hoppers under the boiler passes is high, much of it can be reinjected without materially affecting the total concentration of solids in the flue gases. Fly-ash collectors, however, catch a much higher percentage of the fine ash particles, and when this ash is also reinjected the build-up of ash particles in the flue gases often results in doubling fly-ash concentrations.

Since it is difficult to give a typical analysis of the fly ash from stoker-fired boilers, those shown in Fig. 3 illustrate the wide range of analyses obtained.

**Influence of Type of Coal.** The type of coal burned, its sizing, and its ash content materially affect the fly ash produced. Other factors being equal, the finer the coal, the greater the quantity of fly ash and the finer its analysis. In a given furnace under
similar operating conditions the amount of fly ash in gases varies directly as the percentage of ash in the coal. As an illustration of the effect of the type of coal, tests on a spreader stoker installation showed a variation on the fly ash leaving the boiler of 12 to 40% of the total unburned solids when burning two different grades of coal. Fusion temperature of the ash affects the quantity retained in the furnace when the furnace temperature exceeds the lowest fusion point.

**INFLUENCE OF FURNACE AND BOILER DESIGN.** Design burning rates, furnace volumes, gas velocities in the furnace and through the boiler, and design and location of baffling are factors that influence the fly ash leaving the boiler. High furnace velocities not only increase the percentage of ash carried out but also result in a higher unburned carbon content in the fly ash. Under such conditions concentration is high and the analysis relatively coarse.

**INFLUENCE OF OPERATING CONDITIONS.** Smoke elimination is primarily a problem of proper operating conditions. The fly-ash problem can also be controlled to some extent by careful operation. By maintaining adequate furnace temperatures and properly controlling the fuel-air ratio, the quantity of unburned carbon and the fly-ash concentration (particularly in the nuisance range of coarser particles) are minimized.

Load variations have a pronounced effect on both the analysis and the concentration of the fly ash. Maximum loads result in highest concentration, coarsest analysis, and highest carbon content. Figure 4 illustrates the effect of load variations on the amount of fly ash produced with various types of firing. The effect of load variations is usually greater at the higher loads with underfed stokers. The increase at low loads with spreader stokers is due to the lower furnace temperatures.

Fly-ash concentrations during periods of soot blowing reach many times their values during normal operation. More frequent blowing periods and more care in the proper operation of the blowers often materially reduce the amount of fly ash produced during the soot-blowing periods.

**ORDINANCES GOVERNING FLY-ASH EMISSION.** Many communities have smoke ordinances which also limit the fly-ash emission; others are now considering such ordinances. In some of them the emission of fly ash, in such a manner as to cause a “nuisance,” is prohibited. In the majority of cases, however, definite limitations are set on the allowable concentration of fly ash that may be emitted from the stack. In addition to the limitation on the total fly-ash concentration, some ordinances also set a maximum allowable concentration of material retained on a 325-mesh sieve. This limitation on the coarser fly ash is desirable from the nuisance standpoint since these coarser particles are the greatest nuisance. The various ordinances now in effect limit the total fly-ash emission to 0.30 to 0.75 grain per cubic foot of flue gas, and most of them state that the flue-gas volume shall be determined at a temperature of 500 F with not to exceed 50% excess air. Where a limitation is also placed on the coarser material the allowable concentration of fly ash retained on a 325-mesh sieve is usually 0.20 grain per cubic foot.

A committee of The American Society of Mechanical Engineers has prepared a Model Ordinance which is being followed by many communities drawing up new laws or revising their present ones (Ref. 2). This model ordinance states that fly ash in the flue gas shall not exceed 0.85 lb per 1000 lb of gas adjusted to 50% excess air, except that it shall not be required that dust emitted to the atmosphere be less than 15% of the total dust entering the separating equipment. This limitation is equivalent to a concentration of 0.257 grain per cubic foot of gas at 500 F. It would be a difficult ordinance to meet in communities using high ash coal, were it not for the clause which waives this limitation as long as the dust collector is collecting at least 85% of the fly ash entering.

**30. FLY-ASH COLLECTORS**

**MECHANICAL COLLECTORS.** Baffle-type fly-ash collectors separate the fly ash from the flue gases by projecting the particles out of the gas stream when the gases make an abrupt change in direction. Usually the baffles are arranged in rows so that the gas stream is divided into a series of narrow ribbons. The draft loss is low, and, when necessary, collectors of this type can be designed to operate on natural draft installations or
with existing induced draft fans when the available draft is limited. The collection efficiency is best on the larger, heavier particles. Collectors of the baffle type are also used to concentrate the ash into a small portion of the total gases, from which the ash is separated in a centrifugal-type separator.

**Centrifugal separators**, used for collection of all types of fly ash, usually consist of a number of high-velocity-type cyclones arranged in a common casing with one or more common hoppers. Draft loss is generally 2 to 4 in. of water at design load. Higher collection efficiencies are obtained with higher pressure loss, and maximum efficiency is obtained with the higher boiler loads. The space required for their installation varies considerably, depending on the size of the individual units. The location of the gas inlets and outlets can be adapted to most boiler and fan arrangements. Figure 5 illustrates a collector of this type in which the cyclonic action is produced by a series of vanes.

**Centrifugal concentrators** concentrate fly ash into a small portion of the flue gas from which final separation is made in a secondary or auxiliary centrifugal separator. Fly-ash collectors of this type usually require the minimum amount of space. Several designs operating on this principle combine the fly-ash collector with the fan by replacing the fan inlet boxes with a scroll-shaped chamber in which the ash is thrown to the outer part of the scroll and drawn out with a small percentage of the gases through a scoop or shive-off located near the cut-off of the scroll. Figure 6 illustrates the principle of operation of one unit of this type. In some designs an auxiliary fan is used to draw off the secondary gases; in others the exhaust from the secondary separator is returned to center of the scroll opposite the fan inlet. The differential pressure between this point and the shive-off point produces the flow through the secondary circuit.

The principle of operation of another type of centrifugal concentrator is shown in Fig. 7. The centrifugal concentrator consists of a number of tubes assembled in a common casing. The gases are started swirling by means of a spinner element located at the entrance of each tube. Fly ash is concentrated along the outer wall of the tube while clean gases pass through the center outlet tube. The outer portion of the gas stream into which the fly ash has been concentrated is drawn from the dust chamber through the high-velocity-type cen-
trifugal separator by a secondary vent fan which discharges the cleaned gas back into the inlet of the unit. Since the secondary vent fan operates at constant speed and handles a nearly constant volume of gas, a higher percentage of the gases is handled at the lower loads, resulting in better efficiency characteristics at partial boiler loads. With fine pulverized fly ash, the design efficiency of this type of collector increases as the ratio of secondary gas volume to the total gases is increased. The tubes are assembled in a rectangular casing, and, within limits, the size of the unit for a given gas volume is governed by the available pressure drop, usually 2 to 3 1/2 in. water at design conditions.

**ELECTRICAL PRECIPITATORS.** Cottrell electrical precipitators used for collecting fly ash are usually of the horizontal-flow, plate type shown diagrammatically in Fig. 8. Flue gases are passed through an electrostatic field produced by a high-voltage d-c discharge between two sets of electrodes. The discharge electrodes, which must have a small radius of curvature, are usually wires supported from insulators and spaced between the vertical plates which form the collecting electrodes. The discharge electrodes are generally connected to the negative side of the high-voltage circuit and the collecting plates to the positive or grounded side. In passing through the corona discharge set up between the two electrodes, the fly-ash particles become ionized and are attracted to the collecting plates. Fly ash which adheres to the collecting electrodes is periodically removed by mechanically vibrating or rapping the plates.

The electrical equipment necessary to produce the high-voltage direct current consists of a transformer and either a mechanical or an electronic rectifier. The mechanical rectifier is usually driven by a synchronous motor, whereas the electronic type consists of a group of tubes arranged to give full-wave rectification. The power consumed is about the same for either type. Although the voltage is high, usually 50,000 to 75,000 volts, the current is low so that the power consumption is low.

Electrical precipitators can be designed for high collection efficiency, especially with the fine fly ash from pulverized coal-fired boilers. The collection efficiency increases as the gas velocity decreases and is a function of the length of time the gas remains in the active field. Precipitators designed for maximum efficiency thus have a large cross-sectional area and consist of several sections in series. Since the gas velocities are low, the draft loss is low. Uniform velocity distribution is important for proper operation, and guide vanes or perforated plates are often used at the inlet of the precipitator to equalize the gas distribution.

**COMBINATION MECHANICAL AND ELECTRICAL PRECIPITATORS.** Coarse carbon particles, the easiest to separate by mechanical collectors, are the most difficult to collect in the electrical precipitator. Maximum efficiency of the mechanical type is obtained with higher boiler loads whereas the efficiency of the electrical precipitator is best at lower loads (gas velocities are lower). A combination of the two is therefore ideal for obtaining maximum collection efficiency. When a mechanical collector is placed in series with the electrical precipitator, the lower concentration of fly ash gives a slower build-up of dust on the collecting plates, resulting in better operation with less frequent rapping of the plates. There is a trend toward the use of such combination collectors on large pulverized coal-fired installation where continuous collection efficiencies higher than 90% are desired.

**COST OF FLY-ASH COLLECTORS.** The cost of fly-ash collection equipment is somewhat unstable. For estimating purposes, however, cost of the mechanical type of fly-ash precipitator varies from approximately 8 to 14 cents per cubic foot per minute of flue gas at design conditions, for the equipment alone. The installed price ranges between 20 and 25 cents per cubic foot per minute. The cost of electric precipitators on an installed basis varies from 30 to 60 cents per cubic foot per minute of flue gases, depending on design conditions and desired efficiency of precipitation.

**DIMENSIONS OF FLY-ASH COLLECTORS.** In any specific instance the manufacturer of the equipment should be consulted for dimensions. The following data are useful, however, for estimating. The base or plan area of the multiple-centrifugal separator
type of unit is approximately 1 1/4 sq ft per 1000 cu ft per min of gas at design conditions. The height of the unit depends to a great extent on the storage capacity required in the hoppers. The centrifugal-concentrator type of unit illustrated in Fig. 7 is approximately 4 ft long in the direction of gas flow, and has a face area of approximately 1 sq ft per 1000 cu ft per min of gas flow.
SECTION 8

STEAM TURBINES AND ENGINES

By

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Johns Hopkins University.

W. TRINKS, Professor of Mechanical Engineering, Carnegie Insti-
tute of Technology.

THE STEAM TURBINE

By A. G. Christie

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THE STEAM ENGINE

By W. Trinks

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THE STEAM TURBINE

By A. G. Christie

A steam turbine is a form of heat engine in which two distinct changes of energy take place. The available heat energy of the steam first is converted into kinetic energy by the expansion of the steam in a suitably shaped passage, or nozzle, from which it issues as a jet. A portion of this kinetic energy then is converted into mechanical energy by directing the jet, at a proper angle, against curved blades mounted on a revolving disk or cylinder and by the reaction of the jet itself as it leaves the curved passage.

The pressure on the blades, causing rotary motion, is due solely to the change of momentum of the steam jet during its passage through these blades. Radiation and condensation losses in turbines are small. Leakage losses occur through clearances over the ends of reaction buckets and through labyrinths and glands. Friction of high-velocity steam jets through nozzle passages and across buckets, together with friction losses of high-speed revolving disks and idle buckets in steam-filled chambers, has considerable effect on the efficiency of the turbine.

1. TYPES OF TURBINE

THE SIMPLE IMPULSE TURBINE consists essentially of one or more nozzles supplied with high-pressure steam, with the discharge jet impinging, at a suitable angle, on a single row of buckets on a revolving disk. The steam expands in the nozzle to exhaust pressure, its velocity increasing during expansion. The resulting kinetic energy is partly converted into mechanical energy during the passage of steam across the buckets. The commonest type of simple impulse turbine is shown diagrammatically in Fig. 1.

For best efficiency of the simple impulse turbine the ratio \( \rho \) of wheel speed to steam speed of the jet issuing from the nozzle ought to be about 0.45 where the blading is short in radial length, and about 0.53 or higher for large radial lengths of blade, where pure impulse is seldom used. Available energy with these ratios cannot exceed 70 Btu and 55 Btu, respectively, with usual bucket speeds. Frequently the available energy exceeds these amounts, leading to a decrease in speed ratio \( \rho \), with resultant lower efficiency. Simple single-stage impulse turbines usually are built for small output, although they have been used in units up to 3000 hp.

VELOCITY-COMPONDED TURBINES utilize a very high-velocity jet from a nozzle more efficiently than the simple impulse turbine; for moderate velocities the efficiency of the simple-impulse stage is higher. The steam, after passing through the first row of moving blades, flows through a set of stationary curved buckets or passages. These reverse the direction of the jet and redirect it against a second row of moving blades. This reversal and reimpingement may occur several times before the steam finally escapes to the outlet.

When velocity compounding consists of two rows of revolving blades mounted on the same or parallel wheels, with intermediate stationary reversing blades, the form results in the well-known Curtis stage (Fig. 2), frequently called the two-row wheel, or “impulse” stage, even though the latter is not descriptive. This also is called a velocity stage. In another form, the reversing passages redirect the steam back against the same row of blades that the jet first crossed. This is known as the re-entry type of turbine (Fig. 3).

Helical flow Terry turbines (Fig. 4) employ a forged steel wheel in which semicircular buckets are milled in the rim at an angle of about 30 degrees with the tangent. The steam expands to exhaust pressure in the nozzle, which directs the steam into one side of the semicircular bucket. The steam gives up part of its energy in its first reversal through 180 degrees in the moving bucket but still retains considerable velocity. It then passes to a reversing chamber, which redirects the steam into the wheel buckets. This is repeated several times in additional reversing chambers until the kinetic energy of the steam is reduced to a low value. The whole operation occurs in a single wheel, frequently provided with several groups of nozzles and reversing chambers.

In all these velocity-compounded turbines, part of the kinetic energy is absorbed each time the steam passes through a revolving blade or bucket passage. More work per pound of steam is thus obtained than in a simple impulse turbine of the same bucket speed. Friction losses, however, increase with addition of the reversing blades or chambers. Al-
though the various forms of velocity compounding have their widest application in small steam turbines, particularly for noncondensing auxiliary services, such as driving pumps, blowers, exciters, and stokers, the two-row wheel stage frequently is used as the first stage of very large compound or multistage units.

Stage, in an impulse type of turbine, is a term which signifies that part of a machine in which a decrease of pressure occurs with the accompanying generation of kinetic energy (nozzle), together with succeeding passages where no further drop of pressure occurs (bucket). A stage, therefore, includes (1) nozzles, (2) moving buckets, and (3) reversing elements, or chambers, when used.

**THE MULTISTAGE IMPULSE TURBINE** consists of a series of simple impulse turbines on the same shaft; each of these forms a stage. It is so designed that the steam expands through only a portion of the total pressure range in the nozzles of the first stage. On leaving the buckets of the first wheel, steam enters the second-stage nozzles (which are carried in a diaphragm forming the wall of the stage) and expands through a further pressure drop. The jet impinges on a second row of revolving blades. The operation is repeated in every stage until the steam is fully expanded in the final stage to exhaust pressure.

With this construction (Fig. 5) it is possible to maintain the most efficient ratio of wheel speed to steam speed by properly apportioning the total available energy, from initial
conditions to final pressure, between a suitable number of stages. This is accomplished by proper choice of the nozzle areas, which determine the stage pressures. High efficiencies are possible with this type of turbine.

The total available energy from initial conditions to final pressure is divided in a particular manner, fixed by the manufacturer's construction, between the various stages. Steam speeds and wheel speeds are selected to give values of \( \rho \) (the ratio of wheel to steam speed) as near to the peak-efficiency value as commercially possible. Low-cost units with few stages have low values of \( \rho \) and consequent low efficiency. High-efficiency machines have high values of \( \rho \) (near 0.5) and many stages. Multistage impulse turbines can be built for the largest commercial ratings.

Another early form of turbine, chiefly of historical interest, comprised a series of velocity-compounded or two-row wheel stages. This construction was used on many early central-station turbines from the turn of the century throughout nearly two decades, but has been superseded by more efficient types, particularly the single-row stage.
Fig. 8. Multistage turbine. (Courtesy General Electric Co.)

Fig. 9. Steam path of an impulse-and-reaction turbine (frequently called simply a reaction turbine) illustrating similarity of moving and stationary rows.

Fig. 10. Reaction turbine. (Courtesy of Allis-Chalmers Manufacturing Co.)
Fig. 11. Reaction turbine, 30,000 kw–1800 rpm, single-cylinder condensing turbine; 850 psig–900 F–1.5 in. Hg; AIEE-ASME Preferred Standard. (Courtesy of Westinghouse)
The theoretically high efficiency of a small multistage impulse turbine with few stages is partially offset by the large windage losses of the disks and buckets of the higher-pressure stages, which revolve in a dense atmosphere of high-pressure steam, and which usually have nozzles delivering steam over only a portion of the periphery (partial-arc admission).

A velocity-compounded or two-row stage often is substituted for several of the early simple stages in multistage impulse turbines. The resultant compound impulse turbine, two stages of which are shown in Fig. 6, is shorter, more compact, costs less, and is nearly as efficient as the pure multistage impulse turbine of the same capacity. Figure 7 shows such a unit built by the DeLaval Steam Turbine Company, and Fig. 8 a unit built by General Electric Company. Although units of this type are truly "compound," because they have two types of turbine, the term is not accepted in practice. Rather it is reserved to mean groups of stages contained in separate casings, but using the same steam, e.g., a tandem-compound unit.

The pressure drop in early impulse turbines occurred wholly in the nozzles between stages. Radial clearances over the bucket tips were large, as no pressure difference existed across the buckets. Increased efficiency sometimes may be achieved by assigning a portion of the stage energy to the moving buckets, if clearances over the blade ends are small to reduce leakage due to pressure difference. Suitable labyrinth packings must be provided in all stages where the shaft passes through the diaphragms between stages.

No distinct line exists between impulse and reaction turbines, as the majority of so-called impulse turbines have more or less reaction. The term impulse applies to stages with no reaction, or with appreciably less than 50% reaction. In reaction turbines as much as 50% of the total heat drop per stage is expended, with a corresponding pressure drop, in passing through the moving blade, thereby increasing the relative outlet velocity.

THE IMPULSE-AND-REACTION TURBINE, typified by the steam path shown in Fig. 9, consists of a barrel- or conical-shaped drum placed inside a cylinder, with rows of blades attached alternately to the stationary cylinder and to the revolving spindle. The passages between blades of all rows form contracting orifices, hence there is a drop in pressure of the steam through every row of blades. A row of stationary blades and its following row of revolving blades are together known as a stage. Figure 10 typifies this turbine as built by Allis-Chalmers Manufacturing Company.

In this section reaction blading indicates approximately 50% reaction, i.e., half of the available energy for the stage is released in the stationary blade and half in the moving blade. For best efficiency in a reaction turbine, blade speed should be about 90% of steam speed at the nozzle exit. Theoretically twice as many stages are required as in the impulse turbine. To obtain this efficiency, low steam speeds and many moving rows of blades, and consequently a long spindle, must be used. In general, lower ratios of blade to steam speed prove more desirable in practice, giving fewer rows of blades and shorter turbines. Clearances in reaction turbines must be kept small to prevent excessive leakage. With standard reaction blading this necessitates small radial clearances over the ends of both stationary and moving blades.

Many reaction turbines have blading with shrouds for strength and for reduction of tip leakage. Such blades usually have radial clearance seals. Some reaction turbines are fitted with shrouds to provide an axial (rather than radial) clearance which can be accurately controlled by axial adjustment of the thrust bearing (end tightening).

On small reaction turbines using high steam pressure, full peripheral admission with the small steam volumes handled requires short blades, which are impractical and inefficient, so that a two-row impulse stage often replaces several reaction stages at the high-pressure end, as in the impulse type. The resulting machine (Fig. 11), sometimes known simply as the reaction turbine, is shorter and cheaper to build than the standard all-single-row unit. If a large pressure drop is allowed in the first-stage nozzles, casing temperatures are decreased and distortion troubles are lessened. This combination also permits reasonably long blades to be used on the first reaction row, which must have full peripheral admission.

A series of inlet valves on the first stage, with separate arcs for steam admission, provides better efficiency at partial loads than "throttling governing," which today is obsolete in

(Continued on p. 8-10)
FIG. 13. 40,000-kw Preferred Standard tandem-compound double-flow impulse steam turbine. (Courtesy of General Electric Co.)
Fig. 14. 80,000-kw tandem-compound double-flow condensing turbine, 1250 psig-950 F-1 in. Hg abs.  (Courtesy of Westinghouse)
large-turbine practice. Smaller and cheaper units use it because of its simplicity and low cost.

The Ljungstrom double-rotation turbine (Fig. 12), a radial-flow unit of the reaction type, consists of intermeshed sets of blading, each rotating in the opposite direction. Two generators, tied together electrically, are required. The relative velocity of the two sets of blades is twice that obtainable with a fixed casing and a single revolving spindle. This construction leads to high capacity and high efficiency for a given diameter of blade ring. The unit is compact and usually is placed above, and supported by, its surface condenser. Its construction permits the use of high-temperature steam and quick starting. The simple radial-flow design is applicable to back pressure, noncondensing, and the smaller sizes of condensing units. Large condensing units have double-flow axial blading in the exhaust end, as in Fig. 12.

APPLICATIONS OF STEAM TURBINES. The principal application of large turbines is to drive a-c generators through a solid or flexible couple. Turbines of various sizes, also direct connected, drive centrifugal pumps, small a-c generators, fans, blowers, etc.

For a given blade speed, turbines with small diameters, operating at high rpm, are most efficient. Such units may be connected through double helical reduction gearing to moderate-speed machinery such as fans, propeller shafts, compressors, stokers, pumps, and d-c and a-c generators of low ratings. Both driving and driven units then may operate under best conditions, and first cost of equipment is lower. Some geared turbines have operated rope and belt drives for factory machinery, as cotton mills, rolling mills, etc. Geared turbines also have been used to drive locomotives (see Section 14).

Single-cylinder condensing turbines are built in sizes up to 30,000 kw at 3600 rpm and 100,000 kw at 1800 rpm. Single-cylinder noncondensing (topping) units have been built for 65,000-kw output at 3600 rpm. Figure 13 shows a 40,000-kw Preferred Standard unit built by General Electric Company. Figure 14 shows a 80,000-kw unit built by Westinghouse Electric Corporation. Unidirectional, double-flow turbines in one cylinder provide large areas for the last blade rows. Larger units of either the tandem or cross-compound types often have double-flow low-pressure cylinders to obtain the desired low-pressure blade area. Compound units also are used with reduction gearing in marine and other service.

Impulse versus Reaction Turbines. The impulse type is best suited for use in the high-pressure region and for small steam quantities. The reaction type has advantages for the lower-pressure region, where large volumes of steam must be handled. Practice is tending toward the use of disks for low-pressure reaction blading at high blade speeds, rather than the "drum" construction. In commercial practice, there is no consistent difference in efficiency between the two types, as evidenced by test results and continuing sale of both types in large numbers to utilities. (See Ref. 1.)

Self-contained sets have a turbine mounted directly on or beside its condenser, and are generally connected to the generator through gearing. Exciter and condenser auxiliaries may be driven directly from the turbine through gearing, by connection to the main shaft, or by motors directly connected to the generator. Erection costs are low. Such sets are sometimes known as "packaged units."

STEAM CONDITIONS. Turbines can be built to operate at any steam pressure from a few inches of vacuum up to the highest steam pressures available. Central station pressures and temperatures generally are in the range shown in Table 1, although pressures up to 2400 psig are in use. Industrial turbines operate at all pressures from 100 to 1800 psig. The trend in initial temperature is distinctly upward, with 1050°F adopted for certain units. Temperature limits are fixed by available materials and required life. Increasing coal and labor costs will cause the adoption of even higher pressures and temperatures as suitable metals become available. (See Gas Turbines, Section 10.) Reheat cycles also become more attractive on certain systems with these conditions. Vacuum of 29 in. can be maintained with 57°F cooling water. With 70°F cooling water a vacuum of 28.5 in. can be obtained, using commercial condensers.

OPERATING CHARACTERISTICS. In the straight condensing turbine all the steam that enters the throttle, except some gland leakage in certain types, passes completely through the turbine to the condenser, in which a vacuum is maintained. Figure 15 shows a straight condensing turbine built by the Elliott Co.

Extraction or Regenerative Turbines. Steam is withdrawn from the turbine at intermediate stages and used to heat the feedwater in open or closed feedwater heaters. Because the extraction turbines give higher station economy, they are universally used in central station practice. (See Ref. 64.) Since steam is extracted at various points, the quantity of exhaust steam decreases, the size of condenser is smaller than otherwise, and an increase in the rating of a given casing is possible.
High initial pressures, with moderate initial steam temperature, results in excessive moisture in the low-pressure stages. To avoid this, and to improve station economy, steam can be withdrawn from the turbine at an intermediate stage, reheated by flue gases, live steam, or other hot fluid, and then returned to the turbine. This cycle requires a *reheat turbine* as shown in Fig. 16. (See also Article 17.) Reheat turbines invariably have extraction heaters to heat the feedwater. This type is known as a *regenerative-reheat turbine*.

Fig. 16. 80,000 kw–1800 rpm reheat turbine. (Courtesy Allis-Chalmers Manufacturing Co.)

**Topping Turbine-Generators** have been installed with new high-pressure, high-temperature steam generators in old stations. These turbines exhaust at relatively high back pressure directly into the low-pressure steam main, furnishing steam to older lower-pressure units. This combination, which appreciably improves the station heat rate, is widely used.

**Noncondensing turbines** exhaust at atmospheric pressure or above. They form the high-pressure units of reheat turbines. They also exhaust to heating systems, to industrial processes, or simply to atmosphere in a few cases. While many small noncondensing turbines for auxiliary use have comparatively low efficiencies, those used for power generation can be designed for higher engine efficiencies than condensing units of the same size.

In the *bleeder or extraction turbine*, steam is extracted at one or more intermediate stages, often at comparatively high pressures, for industrial use. Frequently the pressures at these bleeder stages must be maintained constant by a special regulating device forming a part of the turbine; it is then called an *automatic extraction turbine*. The steam not withdrawn at the bleeder points expands through the remainder of the turbine to the exhaust. This type of turbine may operate at a given load with nearly all the steam that enters at the throttle flowing out of the extraction openings or with all throttle steam passing to the condenser when no steam is bled, or with any condition intermediate between these extremes. Such turbines are widely used in industrial plants. Figure 17 shows an extraction turbine built by the Worthington Pump and Machinery Corporation.

**Turbine Speeds.** For 25 cycles, 1500 rpm; for 60 cycles, 3600, 1800, and 1200 rpm. European turbines for 50 cycles, 3000 and 1500 rpm. High-speed turbines are preferred for their lower weight and smaller floor space, although the efficiency is substantially equal to that of slow-speed units of equal ratings. Turbines direct-connected to pumps and blowers usually operate at the speed of the driven unit. Turbines for geared sets may run at any desired speed, and have been built for speeds of 6000 to 7200 rpm in the smaller sizes. This leads to low first cost and increased efficiency. Some marine turbines are built in speeds of 10,000 rpm and higher.

**NEMA Standards for Turbine-Generator Sets.** See Section 16.

**ASME-AIEE Joint Committee Preferred Standards** are given in Table 1 for 3600 rpm central-station turbines.
Table 1. AIEE-ASME Preferred Standard Large 3600-Rpm, 3-Phase 60-Cycle Condensing Steam-turbine Generators

<table>
<thead>
<tr>
<th></th>
<th>Air-cooled Generator</th>
<th>Hydrogen-cooled Generator Rated at 0.5 psig Hydrogen Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine generator rating, kw</td>
<td>11,500*</td>
<td>15,000, 20,000, 30,000, 40,000, 60,000</td>
</tr>
<tr>
<td>Turbine capability, kw †</td>
<td>12,650</td>
<td>16,500, 22,000, 33,000, 44,000, 66,000</td>
</tr>
<tr>
<td>Generator rating, kva</td>
<td>13,529</td>
<td>17,647, 23,529, 35,294, 47,058, 70,588</td>
</tr>
<tr>
<td>power factor</td>
<td>0.85</td>
<td>0.85, 0.85, 0.85, 0.85, 0.85, 0.85</td>
</tr>
<tr>
<td>short-circuit ratio</td>
<td>0.8</td>
<td>0.8, 0.8, 0.8, 0.8, 0.8, 0.8</td>
</tr>
<tr>
<td>Throttle pressure, psig</td>
<td>600</td>
<td>850, 850, 850, 850, 850, 850</td>
</tr>
<tr>
<td>Throttle temperature, °F</td>
<td>825</td>
<td>900, 900, 900, 900, 900, 900</td>
</tr>
<tr>
<td>Number of extraction openings</td>
<td>4</td>
<td>4, 4, 4, 4, 4, 4, 5</td>
</tr>
<tr>
<td>Saturation temperatures, °F at 25°C</td>
<td>36°F</td>
<td>25, 25, 25, 25, 25, 25</td>
</tr>
<tr>
<td>Generator rating †</td>
<td>325°F</td>
<td>25, 25, 25, 25, 25, 25</td>
</tr>
<tr>
<td>Steam flow capacity of turbine, lb per hr</td>
<td>123,000</td>
<td>147,000, 193,000, 302,000, 397,000, 595,000, 561,000</td>
</tr>
<tr>
<td>Generator capability at 0.85 psig, kva</td>
<td>20,294</td>
<td>27,058, 40,588, 54,117, 81,176</td>
</tr>
</tbody>
</table>

Notes:

1. A tolerance of plus or minus 10 °F shall apply to above saturation temperatures. (Tolerances shall be unilateral so as to reduce the spread in temperature between adjacent extraction openings.)
2. The turbine capacity is guaranteed continuous output at generator terminals when the turbine is clean and operating under specified throttle steam pressure and temperature and 2.5 in. Hg abs, exhaust pressure, with full extraction from all extraction openings.

* For data on efficiency of turbines of lower rating, see Table 8, p. 8-61.
† For additional data on capability, see p. 8-64.

2. STEAM-TURBINE CYCLES

If steam could be expanded in a turbine with no friction or other losses, expansion would be isentropic. Theoretically, steam turbines operate on the Rankine cycle or its modifications, such as the regenerative, reheating, or regenerative-reheating cycles. Turbine problems involving energy transformations in the steam are based on isentropic adiabatic expansion with the necessary modifying factors. These problems can be readily solved on a Mollier diagram. (See Art. 2, Section 4.)

In the Rankine cycle, the throttle steam is expanded isentropically from the initial steam condition to the exhaust pressure. The available energy measures the heat thus theoretically available for work (see p. 4-04). The engine efficiency expresses the ratio by which the actual turbine approaches the Rankine cycle in converting into work the energy available with isentropic expansion. The numerator may be the heat equivalent of either internal, coupling, or generator output in kilowatts. The denominator is the product of available energy in Btu per pound and total pounds of steam per hour. It is necessary, therefore, to state whether the calculated ratio is engine efficiency based on internal kilowatts, engine efficiency based on coupling kilowatts, or engine efficiency based on generator output. 1 kw·h = 3413 Btu. Let $I$ kw = internal kilowatts; $I_h$ = total heat per pound of steam at initial conditions before the throttle; $I_h$ = total heat per pound of steam after isentropic expansion to exhaust pressure; $W$ = total pounds of steam per hour. Engine efficiency, based on internal kilowatts, = $(3413 \times I) / W (I_h - I_h)$. Similar expressions can be written using coupling kilowatts or generator output. Where turbines
are sold to drive pumps, fans, or other equipment, the rating frequently is expressed as brake horsepower at the coupling. The numerator in the above equation then becomes \(2544 \times \text{bhp}\). Straight-condensing high-pressure and low-pressure turbines, with no extraction or reheating, follow the Rankine cycle, ideally.

The regenerative cycle, the reheating cycle, and the regenerative-reheating cycle (see Section 4) are used extensively for steam turbines. See ASME Power Test Code for Steam Turbines for efficiency calculation for the different cycles. (See Art. 5, Section 19.)

3. NOZZLES

Nozzle efficiency is of major importance in efficient turbine design. In general 1\% gain in nozzle efficiency has about four times as much effect on stage performance as 1\% gain in bucket efficiency. Nozzle efficiency should be tested experimentally before extensive application of a given design.

CRITICAL PRESSURE. The simplest form of nozzle consists of a circular hole with a well-rounded mouth. If installed in a chamber containing high-pressure steam at absolute pressure \(p_1\), the flow of steam increases as the pressure \(p_2\) on the discharge side of the nozzle becomes less than \(p_1\) until the critical pressure is reached. The flow will not increase with further decrease in the pressure on the discharge side of the nozzle. The critical pressure ratio of throat pressure to initial pressure \(p_1\) varies slightly with superheat. It can be used as 0.55 throughout the superheat range with no appreciable error. Its value can be found from the equation

\[
\text{Critical pressure}, \quad p_c = p_1 \left(\frac{2}{k+1}\right)^k
\]

where \(p_1 = \text{initial pressure}, \text{psia}\), and \(k = \text{exponent for isentropic expansion at constant entropy (}\rho v^k = \text{constant})\) of the steam at the stated conditions. The value of \(k\) varies but for superheated steam, an average value is \(k = 1.3\); for wet steam, \(k\) is variable, usually about 1.13, and critical pressure ratio is about 0.58. The velocity at critical pressure is that of sound in the gas or vapor at the pressure and density existing at the throat. (See Sections 1 and 3.)

The steam jet leaves the nozzle in practically straight lines as long as \(p_2\) equals or is greater than the critical pressure. Hence, for these expansions, nothing more is needed in a turbine than a convergent passageway, with the discharge directed at the desired angle toward the buckets. The orifice must be convergent, because the rate of increase of velocity exceeds the rate of increase of volume, until the critical pressure is reached. The most efficient convergent nozzle in turbines is short, curved, and has a thin trailing edge.

If \(p_2\) is less than the critical pressure, the pressure in the throat, or narrowest part of the orifice, remains at the critical pressure, and further expansion of the steam occurs after leaving the orifice. This causes the jet to expand in all directions. Walls are necessary to confine this further expansion with its accompanying increase in velocity and volume. To do useful work, the jet must be projected toward the buckets in a fixed direction, usually at an angle of 12 to 20 degrees to the plane of the wheel. Below the critical pressure the volume increases at a greater rate than the velocity. A diverging section, therefore, is added to the throat, forming a convergent-divergent nozzle, with mouth dimensions suitable for the range of expansion. The total angle of the divergent walls varies from 6 to 12 degrees. If the nozzles are of rectangular cross section, the sides continue to diverge for their full length. This also is done in some circular nozzles, and the result is a reamed nozzle.

Shapiro (Ref. 2) has shown that shock fronts occur in the diverging section of nozzles with straight diverging sides, because of the convergence of neighboring Mach lines, and appreciable losses result. A design of straight nozzle with curved diverging sides is proposed which obviates shock fronts. Analytical methods (e.g., Prandtl-Busemann) are available for design of such nozzles.

THEORETICAL NOZZLE VELOCITY. The available energy \((h_1 - h_2)\) between any initial conditions and a final pressure is represented by \(A-B\), in the Mollier diagram of Fig. 18.

As the steam expands in a nozzle, a portion of this available heat is transformed into kinetic energy and increases the velocity of the jet. The theoretical velocity \(V\), feet per second, resulting from complete transformation of this available energy is \(V = 223.9\sqrt{(h_1 - h_2)}\). This velocity is not obtained in an actual nozzle because of friction, eddy, and other losses.
Let $\phi$ = velocity coefficient for the nozzle, the fraction of the theoretical velocity actually generated at the nozzle exit.

The actual velocity $V_1$ of the steam jet leaving the nozzle is

$$V_1 = \phi V = 223.9\phi\sqrt{h_1 - h_2}$$

Nozzle efficiency, $\eta$, is the fraction of the available energy converted to useful kinetic energy. Then $V_1$ can be expressed as

$$V_1 = 223.9\sqrt{\eta(h_1 - h_2)}$$

The relation between $\phi$ and $\eta$ is

$$\phi = \frac{V_1}{V} = \sqrt{\eta} \quad \text{or} \quad \eta = \phi^2$$

Nozzle losses appear as reheat. The total heat at the nozzle mouth is

$$h_3 = h_3 + (1 - \phi^2)(h_1 - h_2)$$

This is represented by point C on Fig. 18. The specific volume of the steam can be determined from conditions at point C.

In multistage turbines steam may enter the nozzles of all but the first stage, with an appreciable velocity $V_o$, the carryover from the preceding stage, and with a kinetic energy $(V_o/223.9)^2$. The velocity of steam leaving the nozzle becomes

$$V_o' = 223.9\phi\sqrt{(h_1 - h_2) + (V_o/223.9)^2}$$

The total heat at the nozzle exit is

$$h_3' = h_3 + (1 - \phi^2)\left[(h_1 - h_2) + (V_o/223.9)^2\right]$$

Carryover is the kinetic energy represented by the absolute velocity leaving the preceding stage. Improvements in design of nozzle entrances permit recovery of about 85 to 95% of the carryover energy at most efficient load. Even higher recoveries are possible with further improvement in designs.

Many data have been published on nozzle experiments. (See Refs. 3 and 4.) Nozzle efficiency is greatly influenced by (1) form of approach to nozzle, (2) condition of steam entering nozzle, (3) degree of roughness of nozzle, (4) length of nozzle, (5) thickness of partitions in groups of nozzles, and (6) condition of outlet edges of partitions.

The Steam Nozzles Research Committee of the IME has published many valuable data on nozzle efficiency. The following data show the performance of a set of converging impulse nozzles with partitions 0.04 in. thick, with 20-degree angle, and with a parallel portion beyond the throat equal in length to twice the width at the throat.

<table>
<thead>
<tr>
<th>Theoretical velocity, $V$ ft per</th>
<th>400</th>
<th>600</th>
<th>800</th>
<th>1000</th>
<th>1200</th>
<th>1400</th>
<th>1600</th>
<th>1800</th>
</tr>
</thead>
<tbody>
<tr>
<td>Corresponding Btu per lb</td>
<td>3.2</td>
<td>7.2</td>
<td>12.8</td>
<td>20.1</td>
<td>28.8</td>
<td>39.2</td>
<td>51.2</td>
<td>64.8</td>
</tr>
<tr>
<td>Velocity coefficient, $\phi$</td>
<td>0.976</td>
<td>0.955</td>
<td>0.947</td>
<td>0.946</td>
<td>0.944</td>
<td>0.944</td>
<td>0.945</td>
<td>0.946</td>
</tr>
<tr>
<td>Nozzle efficiency, $\eta$</td>
<td>0.953</td>
<td>0.915</td>
<td>0.897</td>
<td>0.895</td>
<td>0.892</td>
<td>0.892</td>
<td>0.894</td>
<td>0.895</td>
</tr>
</tbody>
</table>

The Nozzle Research Committee also noted that better coefficients were obtained when the nozzle partitions were well chamfered at the outlet edge. The angle of discharge is decreased by this chamfer and may be less than the nominal angle. These coefficients include any influences of supersaturation.

Tests indicate that nozzle performance is improved by omitting any parallel portion beyond the throat of converging nozzles.

The formula for theoretical nozzle discharge is

$$W = 3600 \cdot 144 \cdot \sqrt{2g \times \frac{2^{k - 1}}{k - 1} \cdot \frac{144 \cdot p_1}{v_1} \cdot \left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{k}} - \left(\frac{p_3}{p_1}\right)^{\frac{k+1}{k}}\right]}$$

where $W$ = discharge, pounds per hour; $g$ = 32.2 ft per sec$^2$; $p_1$ and $p_3$ = pressure before and after the nozzle, psi; $v_1$ = specific volume at $p_1$, cubic feet per pound; $k$ = ratio of specific heats; and $A$ = area of orifice in square inches. For pressure ratios less than critical (see p. 8-15) use the critical pressure ratio.
Horstman has a simplified but quite accurate approximation of this formula for superheated steam, particularly applicable to small pressure drops:

\[ W = 2580A \sqrt{\frac{p_i}{V_i}} - [0.13 + (1.13p_i/p_T) - (p_i/p_T)^2] \]

Wirt (Ref. 5) gives the results of tests of convergent nozzles by means of impact tubes using air. Warren and Keenan (Ref. 6), Keenan (Ref. 7), and Kraft (Ref. 8) give further data on tests of steam nozzles. The coefficients disclosed in Refs. 5 and 6 are:

<table>
<thead>
<tr>
<th>Mach. Number (Ratio of Theoretical Velocity to Sound Velocity)</th>
<th>( \phi_r ) Reaction Test Steam (Warren and Keenan)</th>
<th>( \phi_i ) Impact Test Air (Wirt)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.86</td>
<td>98.3%</td>
<td>98.4%</td>
</tr>
<tr>
<td>0.67</td>
<td>98.3%</td>
<td>98.0%</td>
</tr>
<tr>
<td>0.50</td>
<td>98.6%</td>
<td>98 65%</td>
</tr>
</tbody>
</table>

Modern converging nozzles of curved airfoil section with blunt entrance edges show high velocity coefficients comparable to those found by Warren and Keenan.

**NOZZLE EFFICIENCY.** Many factors influence nozzle efficiency. The end effects or secondary flows in nozzles of small radial height require correction coefficients varying from 0.95 for nozzles 0.5 in. high to 1.00 for nozzles 3 in. or more high. The total angle of turning in the nozzle influences efficiency. The greater the turning, the lower the velocity coefficient. Smoothness of all nozzle surfaces affects wall losses. The contour of the nozzle partitions is an important factor in nozzle efficiency, as shown by Kraft and Berry (Ref. 9). Also see New (Ref. 10). These authors outline modern methods of nozzle and blade testing and their application.

**Shock** frequently occurs in improperly designed converging-diverging nozzles, the result of overexpansion. It leads to nozzle losses when the pressure at the mouth is greater than that for which the nozzle is designed. The steam expands as though conditions at the mouth were those for which the nozzle was designed, until at a certain point in the nozzle the pressure becomes less than that at the mouth. Recompression to the pressure at the mouth then begins, and the volume decreases. This causes the jet to detach itself from the wall, and often results in setting up pressure pulsations. The detached jet may be no longer in the direction of the nozzle axis, and so will have an unfavorable angle of discharge. (See Ref. 3, p. 88, Steam Shock.)

When steam expands rapidly from a slightly superheated condition, it does not begin to condense when the saturated condition is reached but continues to expand as in the superheated region, thus becoming supersaturated. Supersaturation has been shown to exist in simple nozzles (Refs. 11, 12, and 13). Supersaturation, with its lesser steam volumes, causes greater nozzle discharges than saturated steam. Supersaturation also tends to lessen the efficiency of nozzles because of energy loss when drops start to form and the steam mass seeks equilibrium.

**Nozzle coefficients** \( \phi \) for convergent-divergent designs are less than those for convergent nozzles because of losses in the divergent section. The coefficient is affected by nozzle form, dryness of steam, and final steam velocity. Values of \( \phi = 0.96 \) may be expected with superheated steam in good designs with proper expansion ratios, but \( \phi \) may decrease to 0.93 with wet steam. Supersaturation increases the flow coefficient. Commercial nozzles may not have efficiencies as high as given above, because of poor entry conditions, too wide flare beyond the throat, or too thick and unchamfered partitions at the mouth. Gains of 5% in nozzle efficiency have resulted from the redesign of nozzles in certain cases.

**FLOW OF STEAM IN NOZZLES.** The design of all nozzles is based on the continuity equation:

\[ w = A_tV_t + 144v_t = A_mV_m + 144v_m \]

where \( w \) = weight of steam flowing, pounds per second; \( A_t \) = area at throat, square inches; \( V_t \) = velocity at throat, feet per second; \( v_t \) = specific volume of the steam at throat conditions, cubic feet per pound; \( A_m, V_m, \) and \( v_m \) are similar conditions at the mouth, or exit.

The pressure at the throat of a convergent-divergent nozzle is always the critical pressure, if the discharge pressure is lower than critical. Hence, the flow of steam through such a nozzle is constant, regardless of the value of \( p_t \), the discharge pressure, provided it is less than the critical pressure. The converging portion of such nozzles generally is short, particularly when the nozzle axis is straight. Expansion in the converging portion can be assumed to take place with a velocity coefficient \( \phi = 0.99 \) for straight nozzles and 0.98 if the approach to the throat is curved. The following equation applies to the flow of
STEAM TURBINES AND ENGINES

steam through a convergent-divergent nozzle if the back pressure is less than critical:

\[ w_* = 0.3155A_t \sqrt{p_i + v_i} \]

where \( w_* \) = pounds of superheated steam flowing per second; \( A_t \) = area of throat, square inches; \( p_i \) = initial absolute steam pressure, psia; \( v_i \) = specific volume of steam at pressure \( p_i \) and the stated superheat, cubic feet per pound. Velocity in the throat, feet per second = \( V_t = 72.24\sqrt{p_i v_i} \). There is evidence that supersaturation of initially saturated steam persists through a pressure ratio of about 4 (Wilson limit) with steam initially dry. The discharge \( w_* \) of a nozzle, pounds of steam per second, is given by the same equation as for superheat, but supersaturation coefficients, ranging from 1.0 for dry steam with low pressure drop to 1.06 for wet steam at critical pressure ratio, must be applied.

Converging nozzles formed of bent plates sometimes are used in low-pressure stages requiring long blades. Shorter nozzles are of the built-up type with curved airfoil partitions. This construction is being extended even to the lower-pressure stages.

When the mouth (exit) of the nozzle is too large the pressure in the nozzle falls below \( p_i \) and the nozzle has "overexpansion." Serious eddies, shocks and other losses resulting from recompression are set up, and the loss from this cause may be serious. If the mouth is too small, the steam will not be fully expanded until after leaving the mouth of the nozzle, and is thus undersuppressed. Stoney states that the loss in jet velocity \( M \) in percentage for various ratios \( R \) of actual nozzle mouth areas to theoretical mouth area is:

<table>
<thead>
<tr>
<th>( R )</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.8</th>
<th>0.9</th>
<th>1.0</th>
<th>1.1</th>
<th>1.2</th>
<th>1.26</th>
</tr>
</thead>
<tbody>
<tr>
<td>( M )</td>
<td>6.8</td>
<td>4.2</td>
<td>2.4</td>
<td>1.1</td>
<td>0.3</td>
<td>0</td>
<td>1.2</td>
<td>4.0</td>
<td>7.0</td>
</tr>
</tbody>
</table>

Note that the losses resulting from undersuppression are only about one-third of the losses from overexpansion, and not of large magnitude for usual values of \( R \).

Flügel (Ref. 14, p. 72) states that, if in converging-diverging nozzles set at an angle to the axis of the blade row the actual back pressure exceeds the design pressure, compression shock takes place, so that the jet leaves the nozzle mouth at an angle smaller than the nominal nozzle angle. On the other hand, if the back pressure is less than design pressure, the jet tends to "turn the corner" so that it leaves the nozzle at an angle greater than the nominal nozzle angle; at the same time there is a tendency to start pulsation waves in the jet as it continues to expand beyond the nozzle mouth.

The lower efficiencies of converging-diverging nozzles have led many designers to use only converging nozzles in turbines designed for high efficiency, using divergent sections only for very large pressure ranges.

AREA OF NOZZLES. To direct the jet, a series of moderately small nozzles, spaced around the arc of the wheel, is used instead of one large nozzle. The total nozzle discharge area of a full circle of convergent nozzles is given by

\[ A = \pi D l e \sin \alpha \]

where \( A \) = nozzle flow area, square inches; \( D \) = pitch diameter of nozzle ring, inches; \( l \) = nozzle radial height (range: 0.5 to 40.0 in.); \( e \) = edge thickness factor (range: 0.88 to 0.95); \( \alpha \) = nozzle discharge angle (range: 12° to 20°).

CROSS SECTION OF NOZZLES. Converging-diverging nozzles may be of circular cross section at throat and mouth, or they may have rectangular cross section throughout. The former are used on single-stage turbines and sometimes in the first stage of multistage turbines of moderate efficiency. Frequently, these are pitched slightly towards the center of the shaft to cause the jet to enter the blades with less spilling at the shroud. Simple converging orifices in diaphragms are of rectangular cross section throughout, normal to the axis of flow.

The discharge angle \( \alpha \) of convergent orifices must frequently be increased in the last stage of impulse turbines to provide passageway for the large volume of steam present, without excessive nozzle length. The pitch of nozzles is fixed arbitrarily. Usually it equals \( 1 \frac{1}{2} \) to 3 times the blade pitch. Another rule is not to exceed 1 in. in width at the throat when measured at right angles to the axis of the jet at the mean diameter.

NOZZLE MATERIALS. Converging-diverging nozzles on some of the smallest turbines are made of brass or bronze, for low-pressure, low-temperature steam. Nozzles are formed of alloy steels (usually stainless, 12% chromium) when steam temperatures exceed 400 °F or where the steam is wet and may corrode the metal.

Convergent nozzles for high pressures and temperatures usually are made of 12% chromium steel, machined and welded into the diaphragms. Large low-pressure nozzles are sometimes formed of bent plates of 12% chromium steel cast into diaphragms, or formed partitions may be used.

Variable entrance and exit angle nozzles and buckets are used to an increasing extent in low-pressure stages, to conform to the so-called vortex theory.
First-stage nozzles, of 12% chromium steel, are sometimes built up in sections. They are bolted or welded to the inlet steam belt of the casing.

4. BUCKETS

The steam leaving the nozzle in impulse turbines is directed against the revolving buckets at an angle of 12 to 16 degrees with the plane of the wheel for the early stages, increasing to 30 degrees in some low-pressure stages. The relative entrance velocity of the jet and its entering angle can be found from the velocity diagram, as in Fig. 19. The exit angle of the buckets is made the same as the entering angle in some small units, particularly those of the re-entry type. On larger turbines the exit angle is almost always less than the inlet angle.

LOSSES IN BUCKETS are due to secondary flows, to leakage over the tips, and to friction in the passages. Secondary flows are caused by compression and re-expansion of the steam jet on the curved face of the passage, and by the bottom wall of the passage on the disk and the top wall formed by the bucket cover. These wall losses are known as end effects; the longer the bucket, the less the influence of end effects upon efficiency. Some designers use the aerodynamicist's term aspect ratio, expressed as bucket height/bucket width. With a given value of \( \rho \), efficiency increases with an increase of aspect ratio.

Other losses include an aspirating loss (sometimes called nozzle-end loss) due to mixing of the jet and dead steam in buckets leaving an idle arc or in the clearance between nozzle and bucket, and shock losses when the jet strikes the bucket at the wrong angle.

Because of these losses the relative velocity \( V_{r1} \) leaving the bucket is less than the entering relative velocity \( V_{r1} \). The ratio of leaving to entering relative velocity is called the bucket velocity coefficient, i.e., \( V_{r1}/V_{r1} = \psi \). This coefficient is influenced by the width of bucket, the form of its rear flank, the total angle through which the steam is turned, the relative velocity of the steam, and the smoothness of the passage.

COMPRESSION IN BUCKET PASSAGES. Stodola remarks that the greater part of bucket losses results from compression and re-expansion in the curved passages. Belluzo (Ref. 15, pp. 74–78) develops the principles of shock and recompression in curved passages. In a curved passage, recompression results from centrifugal force on steam molecules traveling at high relative velocity through the curved passage. If the curve of the passage is assumed to be an arc of a circle of radius \( R \) feet, and of width, \( h \), feet (\( h \) is usually small); \( d_1 \) = density of steam, pounds per cubic foot; \( g \) = 32.2; \( V_r \) = mean velocity of steam crossing blade, feet per second; \( P_a \) = absolute pressure of steam at inlet and outlet of bucket (stage pressure), psia; \( P_x \) = absolute pressure that the steam assumes as a result of recompression due to centrifugal force, psia; then

\[
144(P_x - P_a) = \frac{d_1}{g} \times \frac{h}{R} \times V_r^2 \quad \text{or} \quad P_x = P_a + \frac{d_1}{g} \times \frac{h}{R} \times V_r^2
\]

If recompression were the only consideration, wide passages with wide buckets having curves of large radius would be best. Such buckets have large losses due to increased end effects or, in other words, to decreased aspect ratios. Hence widths have been a compromise, determined largely as a result of test and experience.

Velocity Coefficients. Zietemann (Ref. 16, p. 72) concludes that bucket velocity coefficients are principally influenced by the total angle through which the steam is turned. He presents the following values for commercial blading where \( \beta_1 \) and \( \beta_2 \) are the entering and leaving angles of the blade.

\[
\frac{\beta_1 + \beta_2}{2} \quad 10 \quad 20 \quad 30 \quad 40 \quad 50 \quad 60 \quad 70 \quad 80
\]

\[
\psi \quad 0.77 \quad 0.85 \quad 0.89 \quad 0.918 \quad 0.938 \quad 0.953 \quad 0.962 \quad 0.966
\]

Zietemann's values depend only on the total angle of deflection. Stodola (Ref. 3, p. 179) presents data from Brown Boveri experiments which indicate that the coefficient \( \psi \), with buckets having 30-degree inlet and outlet angles, varies with relative entering steam velocity \( V_{r1} \). For a given bucket, the coefficient curve is hyperbolic. The mean of the maximum values of \( \psi \) is:

\[
V_{r1} \quad 400 \quad 800 \quad 1200 \quad 1600 \quad 2000 \quad 2400 \quad 2800 \quad 3200
\]

\[
\psi \quad 91.2 \quad 91 \quad 90.8 \quad 90.4 \quad 89.9 \quad 89 \quad 87.8 \quad 86.2
\]

A correction factor for end effects in the bucket passage should be applied to the above values. The following correction factors have been used:

<table>
<thead>
<tr>
<th>Bucket height, in.</th>
<th>0.5</th>
<th>0.75</th>
<th>1.00</th>
<th>1.50</th>
<th>2.00</th>
<th>2.50</th>
<th>3.00 and longer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Correction factor</td>
<td>0.95</td>
<td>0.97</td>
<td>0.98</td>
<td>0.99</td>
<td>0.995</td>
<td>0.999</td>
<td>1.00</td>
</tr>
</tbody>
</table>
The above data on the coefficient $\psi$ may be used to calculate velocity diagram efficiencies that agree fairly well with practice.

Aerodynamic test methods provide correct values for $\psi$ for various buckets. In some cases lift and drag coefficients are found. Sometimes the blade coefficient is related to Reynolds' and Mach numbers. Dollin presents values for reaction blade efficiencies as a function of Reynolds' numbers (Ref. 17).

**VELOCITY DIAGRAMS.** Figure 19A is the velocity diagram for one stage of a multi-stage turbine, which operates under the following conditions: pressure and temperature before nozzle (no carryover), 65 psia; 380°F; expands to 38 psia; nozzle angle, $\alpha = 14^\circ$; nozzle coefficient, $\phi = 95.9\%$; wheel speed, $u = 650$ ft per sec; available energy, $h_1 - h_2 = 45.4$ Btu. Velocity leaving nozzle, $V_1 = 223.9 \times 0.959 \sqrt{45.4} = 1446$ ft per sec.

From the diagram, the relative entering velocity $V_{ri} = 830$ ft per sec, and blade entrance angle $\beta_i = 25^\circ$. Assume $\beta_e = 25^\circ$ and $\psi = 0.87$. Relative leaving velocity $V_{re} = 0.87 \times 830 = 722$ ft per sec. Absolute velocity $V_o = 304$ ft per sec.

**Fig. 19.** Velocity diagram for impulse stage. Framed values illustrate the effect of using a smaller bucket exit angle.

The work done, in foot-pounds, is the product of the wheel speed $u$ and the total (algebraic) change in tangential velocity, $V_w$, divided by gravity $g$. This can be readily determined from Fig. 19B, the usual form of combined velocity diagram. The lower triangle of Fig. 19A is placed on the same base $u$ as the upper triangle, and $V_w = (V_{re} + V_{wo})$ equals 1407.

$$\text{Work done in Btu} = \frac{uV_w}{778 \times g} = \frac{650 \times 1407}{778 \times 32.2} = 36.5 \text{ Btu}$$

Combined nozzle-bucket efficiency $= 30.5/45.4 = 80.4\%$.

The volume of steam can be assumed to remain constant during its passage through the buckets. The length on the inlet side is usually about $1/16$ in. longer than the height of the nozzle exit.

**Velocity diagrams for a 2-row wheel** are shown in Fig. 20, $A$ and $B$, based on the following assumptions: initial conditions, 140 psia; 450°F; back pressure, 20 psia; nozzle coefficient $\phi = 95.8\%$; nozzle angle, $\alpha = 14$ degrees; wheel speed, 525 ft per sec; available energy $(h_1 - h_2) = 155.7$ Btu per lb. $\psi_{bl}$, $\psi_{br}$, and $\psi_{tu}$ for first moving, reversing, and second moving rows respectively, are 0.83, 0.84, and 0.87. Usually only the diagram Fig. 20B is drawn. The velocities of whirl from the diagram are 3712 ft per sec for the first row, and 912 ft per sec for the second row, a total of 4624 ft per sec.

$$\text{Work done} = \frac{uV_w}{778g} = \frac{525 \times 4624}{778 \times 32.2} = 96.9 \text{ Btu per lb}$$

The combined nozzle-bucket efficiency $= 62.2\%$. Three rows of moving blades are used on some wheels which operate at low wheel speeds, but these designs are rare.
BUCKETS

REACTION. Stages are said to have reaction when the relative velocity across the bucket is increased by the expenditure in the blade of a portion of the total available energy of the stage. This portion may vary from 5% in high-pressure stages to 50% in the last stage. This last row then becomes a full-reaction stage. Since the available energy released results from a pressure difference across the bucket row, provision must be made to reduce steam leakage over the tip. Radial sealing strips extending from the casing, small clearances over the tips, if unshrouded, or axial sealing strips (end tightening) are used to reduce leakage. Axial seals are used on some high-efficiency impulse units and on two-row stages of some large turbines.

When reaction is added to an impulse stage the optimum value of the ratio of wheel speed to jet velocity at the nozzle exit, \( \rho \), changes. The following values (practical compromises) may be used on impulse designs when reaction is expressed as a percentage of stage available energy:

<table>
<thead>
<tr>
<th>% Reaction</th>
<th>( \rho )</th>
<th>% Reaction</th>
<th>( \rho )</th>
<th>% Reaction</th>
<th>( \rho )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.430</td>
<td>20</td>
<td>0.530</td>
<td>40</td>
<td>0.685</td>
</tr>
<tr>
<td>5</td>
<td>0.450</td>
<td>25</td>
<td>0.565</td>
<td>45</td>
<td>0.735</td>
</tr>
<tr>
<td>10</td>
<td>0.475</td>
<td>30</td>
<td>0.600</td>
<td>50</td>
<td>0.800</td>
</tr>
<tr>
<td>15</td>
<td>0.500</td>
<td>35</td>
<td>0.640</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

All values of \( \rho \) given tend to increase as the nozzle height increases to large values. The values quoted are for small to medium heights.

Zietemann's data show that impulse stage efficiency is influenced by the turning angle of the steam in the bucket. For a given total stage available energy, use of a portion of this available energy to give reaction in the buckets decreases the angle of turning, thus decreasing losses and increasing stage efficiency. Hodgkinson (Ref. 18) gives data on blade width-curve relationships.

The inlet angles of commercial blades usually are increased several degrees over that found on the velocity diagram. This allows the steam to enter without striking the back of the bucket. Less loss occurs if steam strikes the face of the bucket at a slight angle than if it strikes the back. The former condition is splashing, the latter is blunting; both are undesirable.

BUCKET VIBRATION. Buckets are subject to bending stresses because of the tangential forces and because of the pressure difference across them. If there is any irregularity in the driving force due to interruption of steam flow from nozzles as at the horizontal flange, from buckets passing through the steam jet at partial load, or from nozzle partitions, vibrations are set up in the buckets which may lead to early fatigue failure. The amplitude of vibration is lessened by the damping properties of the bucket material. Studies have been made on relative damping properties of available materials, and one of the best, a 13% chromium steel, is frequently used. It not only has high internal damping, but also good stainless properties. (See Refs. 19 and 20.)

Various means are provided to lessen bucket vibration. Lashing or tie wires or other fastenings at intermediate positions on the bucket length have been used. Riveted covers on the bucket tips, tying several buckets together as a unit, also decrease vibration. Double bucket covers with overlapping joints are also used, but only in unusual designs. The longer buckets should be tuned away from their resonant frequencies by careful design, to avoid breakage by fatigue.

BUCKET WIDTH varies from 1/2 to 3/4 in. in small single-stage impulse turbines, and from 3/4 to 2 in., or even larger, in multistage units. The wider buckets are used in the larger machines. The maximum length should not exceed 8 to 12 times the width which is basically fixed by the stresses.

In multistage machines the maximum bending stress in the first-stage buckets occurs at light load, when the first open nozzles expand with a large pressure drop and high velocity. This condition gives the largest number of kilowatts per bucket, and therefore must be the basis of design. This condition may require a relatively wide bucket to withstand the tangential force, as well as vibrations that may be set up.

BUCKET DESIGN. When inlet and outlet angles and width have been chosen, the buckets can be designed. The curve forming the face of the bucket is drawn tangent to the lines forming inlet and exit angles. The pitch usually is chosen as 0.5 to 0.6 of the width, but seldom should exceed 1 in. With equiangular blades Kearton gives the pitch \( P = b/2 \sin 2\beta_1 \), where \( b \) = width, inches, and \( \beta_1 \) = entrance angle. The rear flanks are made parallel to the inlet and exit angles, and the curve of the back of the bucket is fixed by experience. The outlet edges are made as thin as manufacturing considerations will permit. The inlet edges of high-pressure buckets are made thin. In low-pressure blading, however, inlet edges are now thick and rounded, to reduce erosion and to improve efficiency.
VELOCITY RATIO. An important criterion of turbine performance is the velocity ratio \( \rho \), the ratio of the mean wheel speed to the steam speed leaving the nozzle. Turbine efficiency depends on this ratio alone, and not on the individual values of the velocities. For maximum theoretical efficiency of impulse wheels with no bucket losses and no carry-over, \( \rho = \cos \alpha / 2 \), and for velocity compounded wheels, \( \rho = \cos \alpha / 2n \), where \( \alpha \) = angle that the nozzle axis makes with the plane of the blades and \( n \) = number of moving rows in the stage.

Stodola indicates that highest efficiencies in a stage of a multistage impulse unit occur when \( \alpha = 12 \) to 15 degrees. Assuming \( \alpha = 12 \) degrees, the maximum theoretical efficiency occurs at \( \rho = 0.489 \) for a single-row impulse stage and approximately 0.25 for a two-row stage. Consideration of stage losses and carryover lead to the use of higher values of \( \rho \) for maximum efficiency.

Experience has shown that a two-row stage is subject to less decrease in efficiency than a single-row stage when the available energy across the stage is increased as in the first stage of large units at light load. Hence, two-row stages are more frequently used than single-row stages for the first stage of large highly efficient turbines. When reaction is added to either two-row or single-row stages, the optimum value of \( \rho \) increases. Depending on the degree of reaction, \( \rho \) may increase to 0.3 for two-row and 0.7 for single-row stages.

The carryover energy of the absolute exit velocity of the previous moving row is added to the available energy in the nozzle. When carryover is large, larger values of \( \rho \) than \( \cos \alpha / 2 \) may give higher stage efficiencies when \( \rho \) is calculated from the static isentropic available energy nozzle.

REACTION BLADING consists essentially of a series of converging nozzles. Figure 21 is a typical velocity diagram for a reaction stage of one stationary and one moving row.

![Figure 21: Velocity diagram for reaction stage.](image)

Provision is made for 50% reaction in usual designs, i.e., half the stage available energy is expended in the stationary guide blade (nozzle) and half in the moving blade. With this construction, \( V_1 = V_{re} \); \( V_{ri} = V_o \), and the discharge angle \( \alpha \) is the same for both rows, varying from 14.5 to 21 degrees in high-pressure and intermediate-pressure blading. The usual angle in reaction blading is about 18 degrees. Speed ratio = \( \rho = u/V_1 \). The diagram work per stage = \( 2(V_1/223.9)^2 \times (2p \cos \alpha - \rho^2) \) Btu. The efficiency of this stage is proportional to \( (2p \cos \alpha - \rho^2) \). It depends principally on \( \rho \), as the outlet angle usually is a fixed quantity, and reaches a maximum when \( \rho = \cos \alpha \). In practice, \( \rho \) varies from 0.6 to 0.95 in land turbines and seldom is less than 0.85 in large units. The higher values of \( \rho \) give more expensive but more economical turbines. Lower values of \( \rho \) give smaller, and less efficient machines. The work done = \( uV_{w}/778g \) Btu per lb, where \( V_{w} \) is found as indicated in Fig. 21B.

LOSSLSS IN REACTION BLADING. One of the principal losses in reaction blading is leakage over the blade tips, due to the difference in pressure across the blade row. The leakage passes directly across the clearance, while the working steam in the blades is turned through the angle \( \alpha \). The percentage loss from leakage through the clearance is

\[
L = 100C + (LM \sin \alpha + C)
\]

where \( L \) = loss, percentage; \( C \) = clearance, inches; \( L \) = length of the blade, inches; \( \alpha \) = exit angle of blade; and \( M \) = thickness coefficient (unity for most reaction blading).

Shrouded blades with radial seal strips inserted in the casing may have only 60 to 75% of the leakage calculated by the above equation.

The steam passing through the blades \( W_1 = W/C_1 \) where \( W \) = total steam per hour in pounds flowing through the turbine, and the clearance factor \( C_1 = (1 + C/2 \sin \alpha) \). The bucket efficiency \( \eta_b = \eta_d/C_1 \), where \( \eta_d \) = the diagram efficiency as calculated from a diagram similar to Fig. 21, using appropriate velocity coefficients.

Leakage losses are greatest with short blades and large clearances, and smallest for the long blades of the last rows. A two-row stage frequently replaces the high-pressure section of reaction turbines, where leakage and other losses are high. In such stages the
clearance over the tip of the shortest blade is never less than 0.025 in. With no leakage the efficiency of reaction blades would be substantially that of nozzles, from 92 to 97%. Leakage, interstage, and rotation losses must be considered, however, in calculating the stage efficiency.

The face and rear flanks of reaction blading are made up of a series of curves. One effective form is made from the intersection of two ellipses. The entrance edge of reaction blades usually is quite blunt, to reduce erosion.

Widths of reaction blades vary from \( \frac{1}{2} \) in. for blades 4 in. long to \( 1\frac{1}{2} \) in. for blades 30 in. long. The pitch of blade rows is fixed somewhat by the type of shroud used to secure end packing, as it must be possible to shift the spindle toward the exhaust to clear these shrouds in lifting the rotor.

**Area Through Last Row of Buckets.** A major consideration in turbine design is the large volume of steam to be passed through the last row of buckets, particularly at high vacuum. If the area of the passage through the last row is too small, the steam will have a high absolute leaving velocity, involving a considerable energy loss. This area, therefore, should be as large as possible, while still retaining sufficiently small blade angles to insure good diagram efficiency with safe bucket lengths.

Kearton (Ref. 21, p. 303) gives the centrifugal stress, psi, in steel blades of uniform cross section as

\[
f_c = 4.06l \left( \frac{d}{100} \right) \times \left( \frac{N}{100} \right)^2 = 0.215u^2m
\]

where \( d \) = mean diameter, inches; \( N \) = rpm; \( l \) = length of blades, inches; \( u \) = wheel speed, feet per second; and \( m \) = ratio of blade length to mean diameter. It is also shown that \( f_c = 1.88A(N/100)^2 \), where \( A \) = area of annulus of last blade row, square feet. From this it is evident that the stress in the blades of the last row is directly proportional to the annular area of the bucket ring for a given speed. These stresses should be multiplied by 1.074 for brass or bronze blades.

**Leaving Loss.** The absolute velocity from the last row of buckets represents kinetic energy generated from the available energy. This kinetic energy \((V/223.9)^2\), known as the "leaving loss," is one of the principal losses of the turbine. It cannot contribute useful work after leaving the last blade row. Blades as long as commercially justified are used to increase the exhaust annulus and to reduce this loss.

**Hood Loss.** Pressure drop through the exhaust hood due to friction and eddies causes a decrease in available energy known as hood loss. When added to the leaving loss, the total is known as exhaust loss.

**Vortex Theory and Bucket Design.** It is general practice to put some degree of reaction in almost all stages, particularly in the low-pressure section. Many low-pressure stages have as much as 50% average reaction. This division of stage energy between nozzle and bucket lessens the velocity in the nozzle, but even so the velocity frequently exceeds the critical. In some cases limited last-stage area due to stress limits on blade length forces the use of reaction.

Radial pressure differences exist in the space between the stationary and moving elements; this difference is more marked in the greater bucket lengths. *Free vortex theory* is therefore used in design. Under the precepts of this theory the flow across the nozzle-blade passage is constrained to have a vortex velocity distribution by proper variation in angle (and passage area) of the nozzle and blade. Such distribution requires that the tangential components of the absolute velocities vary along the blade height in inverse proportion to the radii at which they occur. As a consequence of this vortex flow, motion of the steam particles exerts a centrifugal force which results in an increase in pressure from the inner to the outer radii of the blade row, maintaining equilibrium in the passage, and preventing establishment of radial velocity components.

As a result of this vortex action, varying degrees of expansion occur in the stationary nozzle and in the following bucket. The greater part of the pressure change of the stage takes place in the stationary element at the inner radius, and less pressure change occurs at this radius in the revolving bucket. As a result of this energy distribution, the bucket contour of long blades at the inner radius approaches that of an impulse bucket with its discharge angle slightly closed, whereas near the tip the configuration is more like that of a reaction stage. Thus the net result of vortex design is a stage which has radial variation of the degree of reaction.

The build-up of pressure at the outer radius of a long bucket leads to a smaller pressure drop in the nozzle and a larger pressure drop at the moving bucket tip. The buckets are warped from base to tip, with varying inlet angles, increasing from base to tip while discharge angles decrease. Some turbines are designed with variable-angle warped-surface nozzles in the stationary element of the stage to provide for the varying degree of expand-
sion throughout its height. Other designers adhere to a constant discharge angle on the stationary element, depending on the radial spread at increasing diameters to give the increased nozzle flow area required by the increase in reaction.

Vortex design reduces the probability of radial flow in the space between stationary and moving elements in a stage. This theory has been applied to the last stage of many turbines, to all stages of some high-efficiency units built abroad and to at least one unit in this country. The stage efficiency is increased only a small amount on short buckets but may be increased by a measurable quantity on the last stages. No test evidence of an overall improvement has yet been published.

Various methods have been proposed to permit calculation of vortex stages, but they have not been published. (See also Axial Flow Compressors, Section 1.) One method is given by Pochobradska (Ref. 22). In general all vortex designs are based on the relation

\[ V_{at} = \text{constant} \]

where \( V_w \) = (tangential) velocity of whirl at any radius, feet per second, and \( u \) = wheel speed at the same radius, feet per second.

**BUCKET FABRICATION AND DESIGN.** Buckets may be precision-cast, rolled, drop-forged, or machined from rectangular bars. Alloy steels generally are used for blading, consisting of 12 to 14% chromium and 0.10 to 0.15% carbon.

Some buckets are drop-forged with an axial bulb-shaped end on a straight shank at the blade base. After machining, they are driven axially into similar-shaped slots in the outer rim of the disk. The inverted T base with one or more sets of shoulders is a common type of fastening. Such buckets are inserted into rotor grooves which are closed by special devices when the last bucket has been inserted. Frequently T-base buckets are held tight against their shoulders by a small wedging strip driven into a shallow slot below the T base. Many long buckets have a straddle base which fits over a T- or tooth-shaped rim on the disk.

Spacing of most buckets is secured by machining the integral base to the proper width, thus also determining the discharge angle. One large manufacturer cuts the shorter buckets in low-temperature turbines from a rolled strip of copper-nickel alloy. They are set in a jig to insure uniform spacing and correct angle, and an alloy foundation ring is cast around their bases. A shroud strip is silver-soldered to the outer ends and the foundation ring is then machined, with one or more projections to match grooves in the spindle. These segments are balanced, inserted in grooves, and held in place by a caulking strip at one side. Serrated grooves are used with some forms of bucket root.

No sharp corners or edges are permissible on any part of the bucket, particularly at the base, since fatigue failures may start at sharp edges or corners. Side-entry buckets are sometimes used on large low-pressure stages with each bucket row on a separate disk. In this design serrated bucket roots are driven into grooves milled either axially or in curved form across the rim of disks integral with or fastened to the shaft. The roots are locked in place by locking pins after being driven in place.

Buckets are examined carefully at each overhaul for signs of fatigue cracks, either by visual examination with a magnifying glass or by magnafux.

Recent designs are curved airfoil sections, with well-rounded entrances. This construction is less sensitive to the angle of the entering steam jet, has a greater cross section, and is more rugged than buckets with finer entrances. A larger tenon through the shroud can be provided, which is highly desirable when it is riveted over the shroud.

**Taper.** Long low-pressure revolving buckets may have a tapering form to provide greater strength at the base, without excessive weight. C. A. Parsons and Company in England makes a hollow blade for low-pressure use, with increasing metal cross section from tip to base.

**Lacing wires** are silver-soldered to some of the larger buckets to reduce vibration. Long alloy steel buckets may have reinforcing projections, either welded to each individual bucket or forged integral with it. If welded, the buckets are heat treated before finishing. These projections on adjacent buckets can then be welded together after assembly without undue heating of the bucket. Thus a reinforcement is provided against vibration, at a midpoint in the blade length.

**Gaging of reaction blades** is the ratio of the net area for steam flow on the mean diameter measured at the blade outlet at 90 degrees to the direction of the jet, to the annular area occupied by the blade ring. Thus 25% gaging means a net steam flow area of 25% of the blade-ring annulus.

**Shrouds.** Some buckets, particularly small ones, are formed with projections at their tips to form their own shroud rings. Shroud strips may be held by riveting over the tenons machined at the ends of the buckets. Shrouds may consist of flat strips in impulse turbines with a clearance of 0.03 in. between adjacent groups, each containing six or eight buckets.

Sometimes flat shrouds, which project on one side toward the base of an adjacent row,
are used for end tightening. The clearance between shroud and blade base can be adjusted to a much lesser amount than the permissible end clearance of similar blades. The reduced leakage with such shrouds results in better turbine economy.

Methods of Attaching Shrouds. Sometimes large cylinder blades are assembled in jigs, and the shroud is riveted on. The group is then inserted as a unit in the cylinder grooves. Cylinder blades are often inserted before the shroud is added and riveted. Flat shrouds may be riveted or silver-soldered to the blades and clearances maintained by radial sealing strips fastened to the casing. Channel and angle-shaped shrouds are also used, with the projecting strip serving as a seal.

Strength of Roots. All bucket roots must withstand the stresses due to centrifugal force. Methods of calculating such stresses can be found in Refs. 4 or 23. The centrifugal force on a bucket is

$$F = 0.00002342 N^2 W r$$

where $N$ = rpm; $W$ = weight, pounds, of bucket and shroud ring beyond the section of the base carrying the load; $r$ = radius, inches, from the center of the shaft to the center of gravity of the bucket and shroud beyond the section under stress. At normal speeds, buckets should not be stressed higher than 0.5 of their elastic limit. This allows the turbine a certain overspeed without risk of bucket trouble.

Bucket roots must be designed to withstand stresses due to vibration. Ryan and Rettaliau (Ref. 24) analyzed stresses in bucket roots and corners by means of plastic models and polarized light. Calibration of these models permitted calculations to be made of stresses in actual buckets.

Corrosion. Certain bucket materials corrode badly if the feedwater is not deaerated to free it of oxygen and carbon dioxide. Corrosion often is rapid in idle turbines into which steam leaks through the throttle valve. Aluminum-bronze blades have corroded badly in turbines receiving wet steam carrying magnesium and calcium chlorides.

Bucket failures have sometimes been attributed to corrosion fatigue due to the presence of salt deposits which accelerate cracking of the metal. Stages near the dew point in the turbine steam path are strengthened for greater corrosion fatigue strength.

Erosion of the inlet edges of low-pressure buckets in turbines with high tip speeds has sometimes required the renewal of buckets in 3 to 7 years. This is due to impact of water drops formed as a result of expansion. F. W. Gardner (Ref. 25) shows that the drops of water must be extremely fine, that they tend to concentrate on the outer ends, that the force of impact of the drops on buckets moving 1000 ft per sec is about 90,000 psi, that it is impossible to remove by any separating device all the water that condenses, and that hard bucket material and hard sheaths on blades (e.g., stellite) are necessary to withstand this erosion.

Expansion in most condensing turbines is carried to conditions where the moisture content in the exhaust steam ranges from 10 to 12%. This limit is fixed by energy losses due to mechanical interference between moisture drops and steam, and to the rapid erosion of the inlet edges of unprotected buckets caused by impingement of moisture drops. See Ref. 26 for curves showing braking losses caused by moisture in buckets, given in percentage of the useful output at different pressures and temperatures. Curves are shown of improvements resulting from drainage grooves.

Christie and Colburn (Ref. 27) found that erosion was most serious in turbine buckets with tips speeds over 1000 ft per sec, that it is more pronounced in reaction than in impulse buckets, and that erosion is most severe about 1 to 1 1/2 in. from the tip and around lacing wires. They also show how steam conditions at the entrance to the last row may be estimated.

Hardened strips fastened or brazed to the inlet edges of the buckets, coatings of hard alloys as stellite, tungsten tool steel, tantalum, etc., fused onto the flank, drainage systems to remove interstage moisture, alloy steel material, and combinations of these, reduce erosion.

Stainless steel buckets with 11 to 13% chromium are widely used in low-pressure buckets. In addition, they are protected on the inlet edge by a hard (stellite) shield one-half to two-thirds of the outer bucket length.

Water-drainage grooves are provided around the casing of most units to draw off water formed by condensation. These water-drainage devices remove 25 to 30% or more of the moisture present. The low-pressure bleeder opening also serves to draw off such water. Caldwell has proposed that hollow partitions be used for nozzle or bucket passages in low-pressure stages so that steam from a higher pressure stage can be provided to reheat the passing steam and to keep it relatively dry.

Fabrication. It is becoming general practice to mill buckets individually from stress-relieved bars or forgings. The roots are milled to serve as spacers. Necessary shoulders or serrated holding teeth are milled accurately on the sides of the base.
Some hollow low-pressure stationary blades for reaction turbines have been made by folding an alloy steel sheet over a form and welding the trailing edge, thus decreasing machine weight. Precision cast blades for complex shapes are under trial.

Material of many kinds has been tried for turbine buckets. Present practice favors an alloy steel with 11.5 to 13% chromium, 0.10 to 0.15% carbon, ultimate strength 90,000 to 100,000 psi, yield strength of 80,000 psi, and proof strength of 70,000 psi. A steel of 19% chromium, 9% nickel, and 0.5% tungsten has been used for 1000 F. Austenitic steels are used for higher temperatures. Monel metal (ultimate strength 85,000 psi and elastic limit 50,000 psi) is used in small units with moderate steam temperatures and stresses because it does not readily corrode. Special steel alloys and pure nickel are used in industrial turbines where corrosion may be serious.

Clearance. When the same pressure exists on both sides of impulse buckets, the clearance over their ends may be large. The axial clearance between nozzle exit and bucket entrance on small impulse turbines varies from $1/32$ to $1/16$ in. Large machines have axial clearances up to $1/2$ in.

The radial clearances over the ends of reaction or impulse buckets with moderate reaction must be kept small. Consideration also must be given to the rigidity of the casing. Various formulas are used by builders of reaction turbines for determining this clearance. Some formulas are based on distance between bearings and others on blade length and mean diameter of row, with consideration of the taper on the ends of the blades.

One formula for radial clearance of buckets without taper at the top is

$$C = 0.015 + 0.003D + 0.005l_b$$

where $C =$ clearance, inches, $D =$ mean diameter of blade ring, feet, and $l_b =$ length of blades, inches.

This radial clearance, even when sealing strips are used, is seldom less than 0.020 in.

Stodola (Ref. 28) quotes one builder of reaction turbines as allowing on large single-cylinder turbines a clearance of $0.001 \times$ mean diameter, and on short two-cylinder units $0.0008 \times$ mean diameter.

The clearance of end-tightened blading can be set at substantially that of the dummy pistons. This usually will be less than half the end clearance given by the above formula, particularly when axial adjustments of the turbine rotor relative to the cylinder can be made with the unit under load. When measuring devices are installed, the operating clearance may be as close as 0.005 in.

Bucket length usually is limited to about 35% of the mean diameter of the bucket row. Maximum bucket lengths on 3600 rpm turbines are: 23 in. on a 42.5 in. disk, giving mean diameter of 65.5 in., mean blade speed of 1029 ft per sec, and tip speed of 1390 ft per sec; on 1800-rpm turbines, 40 in. long buckets on an 80-in. disk, mean diameter 120 in., mean blade speed 942 ft per sec, and tip speed 1257 ft per sec.

Minimum bucket length is seldom less than 2% of the mean diameter. This minimum is $1/2$ in. on small impulse turbines and generally 1 in. on reaction turbines.

**EFFICIENCY AT THE WHEEL PERIPHERY OF IMPULSE UNITS** is found by combining the nozzle efficiency $\eta_n$ with the diagram efficiency $\eta_d$. Stodola (Ref. 3, p. 222) makes these statements on the efficiency at the wheel periphery: (1) Efficiency depends only on $\rho$, the ratio of wheel speed to steam speed, and not on the individual values of the velocities. (2) Efficiency varies with the peripheral velocity according to a parabolic law. (3) With a small nozzle angle $\alpha$ the maximum value of efficiency at the wheel periphery is attained when the peripheral velocity is nearly half the steam velocity. (4) The best efficiency for constant blade coefficient $\psi$ is higher the smaller the nozzle angle $\alpha$. (5) Energy loss in the nozzle is nearly four times as detrimental as energy loss in the buckets.

**NOZZLE AND BUCKET EFFICIENCY.** Data on nozzle and bucket losses are not complete. Many designers start with internal efficiencies based on tests of similar units, from which stage efficiencies can be calculated. Then nozzle and bucket efficiencies can be deduced with considerable reliability. Such deductive design based on test performance is preferable to synthetic design using assumed efficiency factors. Detailed methods of proportioning nozzles and blading will be found in Refs. 3, 4, 14, and 16.

**DEPOSITS IN TURBINES.** The steam supplied to turbines from boilers should be free from moisture, dust, acid, and corroding chemicals and as nearly pure as possible. Any corrosive element in the steam will rapidly destroy turbine blading, starting at the dew point. The surfaces are continually swept clear by the high-velocity steam, thereby accelerating corrosive attack.

Although feedwater in modern plants consists of condensate and evaporated make-up water, impurities may be carried into boilers from condenser leakage. Chemicals must be added to maintain the desired sulfate-alkalinity ratio so that the average concentration
in the boiler drums may range from 1000 to 3500 parts per million. Because of the evaporation of any moisture in the superheater, some impurities are carried over as dust or vapor and deposited on the governor valves, nozzles, and buckets of the turbine, closing up the passages and decreasing the capacity. The decreased capacity may range from 15 to 50% in a few weeks, in some cases. (See Refs. 62 and 63.)

5. ROTORS

Rotors of small high-speed impulse turbines usually consist of a disk or wheel, carrying the buckets, pressed on a shaft and held against a shoulder by a lock nut. Some velocity compound turbines with low wheel speeds use two or more disks, each with a single row of buckets, instead of one disk with several rows. Small turbines generally are designed with shafts that operate well below their critical speed.

Multistage impulse turbines have a series of disks mounted on the shaft, with intermediate diaphragms between to carry the nozzles and the labyrinth packing. Surfaces of the disks should be smooth, and preferably polished.

Impulse-turbine rotors for high-pressure and topping units are frequently made of a single forging with disks formed by removing the steel between the disks; such construction is known as the "solid-rotor" type. The short rigid rotor permits operation below the calculated critical speed. Because of its extended surface the rotor is readily heated. Closeness of diaphragms prevents the rotor's cooling off faster than the shell, on shutting down. All rotors are now heat-stabilized before final machining, by heating in a furnace to about 900°F. This removes residual stresses, and has been found to give more smoothly operating machines.

Disks of two-row wheels have a rapidly tapering section and a heavy hub. Bucket speed at the mean diameter usually varies from 200 to 700 ft per sec.

Disks for multistage impulse turbines are made of uniform thickness, or of a hyperbolic tapering section. With only a relatively light rim, the stresses are less than in two-row wheels. They operate at rim speeds varying from 400 to 650 ft per sec.

DISK MATERIAL. The maximum working stress in disks at 25% overspeed should not exceed the elastic limit. At normal speed, the working stress of low-pressure, low-temperature disks should not exceed 16,000 to 20,000 psi. The working stress of disks used at high temperature are even lower, depending on the elastic limit at working temperature. Stresses at normal speed are designed to give a factor of safety of 2 or higher.

Disks of small turbines with low wheel stresses and low steam temperatures are of medium carbon steels with ultimate strength of 65,000 to 75,000 psi; elastic limit 30,000 to 40,000 psi. Larger disks at moderate stresses and temperatures are of carbon steel forgings with about 0.45% carbon, normalized and drawn, with ultimate strength 75,000 psi; yield point at room temperature, 40,000 psi. Where stresses are high, carbon-molybdenum disks, forged and heat treated, ultimate strength 100,000 psi, yield point 70,000 psi, are used.

A wide variety of alloy steels has been used for highly stressed disks operating at high temperatures. Among them are nickel steels, chromium-nickel, chromium-molybdenum, nickel-molybdenum, and molybdenum-vanadium. In general their ultimate strength varies from 85,000 to 105,000 psi, with yield points of 55,000 to 80,000 psi. (See ASTM A294 for details of composition.)

In general, chromium is added for greater strength and molybdenum for higher temperature resistance. Cobalt may be required with temperatures above 1000°F. Vanadium and columbium are stabilizing elements.

Certain manufacturers overstress the disks by operating them at overspeeds. The disks then are allowed to age, resulting in an increase in the elastic limit of the material.

Care must be taken in forging the disks to work all the metal thoroughly. This can be assured in forging by heavy blows which penetrate the whole metal and thus reduce grain size. Light forging or rolling work only the outside portions of the metal. No subsequent heat treatment can make up for lack of proper working.

DISK DESIGN. Disks usually are designed by assuming a thickness of disk under the rim, such that it will not buckle or bend during machining and erection. This thickness may be \( \frac{3}{8} \) in. on small wheels, increasing to 1 in. on some two-row wheels and large-diameter impulse disks. The thickness at other points in a disk of hyperbolic profile is found from the equation \( t = cr^2 \), where \( t \) = thickness, inches; \( r \) = radius, inches; and \( c \) = a constant found for the conditions under the rim. The exponent \( a \) varies from \((-0.4)\) to \((-0.8)\) in multistage impulse wheels and is taken as \((-1)\) for two-row wheels.
Wide variations exist in the disk designs of various builders. Some disks have been made too thin and have given trouble in service because of nodal vibration.

The first step in disk design is to assume the thickness under the rim. A value of exponent \( a \) is then chosen, depending on the type of disk desired. Since \( r \) is known, the value for the constant \( c \) can be found when the thickness is chosen. The thickness at any other radius can be readily calculated from the equation. Present practice on large turbines is to make the disks rather heavy, hence less liable to vibration.

The theoretical hyperbolic curve is modified near the hub to one or more arcs of a circle, with much greater curvature than the hyperbola, to provide a heavy hub in which a keyway may be cut without weakening the disk section. The proportions of the rim are determined by the size, number of rows of blades, and the methods of fastening the blades to the rim. The rim is connected to the narrowest part of the disk by a section of curved profile, frequently consisting of arcs of circles of short radius. The flanks of the disks between the curved sections at hub and rim are sometimes made straight taper or wedge shaped for easier machining. Thin form is slightly stronger than the hyperbolic profile, and is more easily made. Departures from true hyperbolic form make the solution of disk stresses complicated and tedious.

**DETERMINATION OF DISK STRESSES.** After preliminary designs are finished, calculations are made to determine stresses. Three stresses, radial, tangential, and axial, may act at any given point. The axial stress is of negligible value if there is no sudden change in axial thickness, as at a hub. Radial and tangential stresses can be calculated by neglecting axial stress. Stodola (Ref. 3) developed formulas for determining these stresses, but they are complicated in the form presented.

S. H. Weaver (Ref. 29) describes a method of calculation which is simpler and more readily applied. For machining purposes radial sections of disks usually consist of straight lines and arcs of circles. The equations of these lines present mathematical difficulties in calculating stresses. Hence the section is assumed to consist of one or more hyperbolas of the equation, \( l = cr^n \). In this equation the exponent \( a \), the shape constant of the profile, has a negative value when the thickness decreases with a larger radius, a zero value for a constant or uniform thickness, and a positive value when the thickness increases with radius. For a given disk profile, Fig. 22, the value of \( a \) may be found from

\[
a = \log_{10} \frac{t_2}{t_1} + \log_{10} \frac{r_2}{r_1}
\]

Stodola's equations for tangential and radial stress are given below. **Notation.** \( m_1, m_2, \) and \( p \) are algebraic quantities as given in the equations of Group II, below; \( a \) = shape constant of profile of the particular portion of the disk section; \( V \) = Poisson's ratio of deformation = 0.3 for steel; \( E_1 \) = Young's modulus of elasticity; \( R \) = radial stress at radius \( r \), psi; \( T \) = tangential stress at radius \( r \), psi; \( r \) = any radius in disk section, inches; \( b_1 \) and \( b_{11} \) = boundary condition constants; \( \omega \) = angular velocity of rotation in radians; \( u \) = mass of disk material per unit of volume = 0.2815 pounds per cubic inch for average steel; and \( v \) = radial elongation, inches.

\[
R = 1 - \nu^2 \left[ (3 + V)pr^2 + (m_1 + V)b_1r^{m_1-1} + (m_2 + V)b_1r^{m_2-1} \right]
\]

\[
1 - \nu^2 \left[ (1 + 3V)pr^2 + (1 + m_1 V)b_1r^{m_1-1} + (1 + m_2 V)b_1r^{m_2-1} \right]
\]

It will be necessary to know two stresses in order to determine the values of the condition constants, \( b_1 \) and \( b_{11} \) and to transform these equations. Known radial stresses \( R_1 \) at radius \( r_1 \) and \( R_2 \) at radius \( r_2 \) are assumed. The tangential stresses at \( r_1 \) and \( r_2 \) are

\[
\begin{align*}
T_1 &= Ar_1^3 - BR_1 + CR_1^2 \\
T_2 &= Dr_2^3 - ER_2 + FR_2^2
\end{align*}
\]

where \( A, B, C, D, E, \) and \( F \) have the values given in equations of Group II.

The stresses due to the external centrifugal load and the weight of the disk itself vary as the square of the speed. If all stress values are calculated for 1000 rpm, the stresses at any other speed can be found by multiplying by the square of the speed ratio.
The following formulas (Group II) based on 1000 rpm assist in solving for the various stresses:

\[
K \frac{r_1}{r_2}, \quad a = \frac{\log_{10} \left( \frac{t_2}{t_1} \right)}{\log_{10} (1/K)} \quad \text{or} \quad -\frac{\log_{10} (t_1/t_2)}{\log_{10} (1/K)}
\]

\[
m_1 = -(a/2) - \sqrt{a^2/4 - 0.3a + 1}
\]

\[
m_2 = -(a/2) + \sqrt{a^2/4 - 0.8a + 1}
\]

\[
p = \frac{m_1K^{m_2-1} - m_2K^{m_1-1}}{K^{m_2-1} - K^{m_1-1}}
\]

\[
E = \frac{m_1 - m_2}{K^{m_2-1} - K^{m_1-1}} \quad \text{or} \quad \frac{E}{K^{a+1}}
\]

\[
F = B + a
\]

\[
A = \frac{3.3(C - K^2B) - 1.9K^2}{1 + 0.4125a}
\]

\[
D = \frac{3.3(F - K^2E) - 1.9}{1 + 0.4125a}
\]

(II)

The factors \( A, B, C, D, E, \) and \( F \) are functions only of the shape constant \( a \) and the ratio of radii \( K \).

These factors become relatively simple for the portions of the disk where the wheel section is of constant thickness, as at the hub and sometimes at the rim. At 1000 rpm the functions reduce to these equations (Group III):

\[
a = 0 \quad K = \frac{r_1}{r_2} \quad C = \frac{x}{1 - K^2}
\]

\[
B = F = C - 1 \quad E = C - 2
\]

\[
A = 6.6 + 1.4K^2 \quad D = 6.6K^2 + 1.4
\]

(III)

Since determination of the values of the functions takes time in making a stress calculation, Weaver has prepared the following approximate equations (Group IV) for the more rapid determination of these functions by the use of common logarithms. The error in these equations is about 0.7% as a maximum.

\[
B = \frac{\log_{10} (1/K)}{5.43} \left( a^2 - 1.2a - (a/2) + \left( \frac{2}{1 - K^2} - 1 \right) \right)
\]

\[
F = B + a
\]

\[
E = \frac{\log_{10} (1/K)}{10} \left( a^2 + 10a - (a/2) + \left( \frac{2}{1 - K^2} - 2 \right) \right)
\]

between the limits 0.8 and 0.1 for \( K \) and \((-5)\) and 0 for \( a \).

\[
E = \frac{\log_{10} (1/K)}{7} \left( a^2 + 10a - (a/2) + \left( \frac{2}{1 - K^2} - 2 \right) \right)
\]

between the limits 0.97 and 0.8 for \( K \) and 40 and 0 for \( a \).

\[
C = \frac{\log_{10} (1/K)}{3.33} \left( a^2 + 4.8a + (a/2) + \frac{2}{1 - K^2} \right)
\]

between the limits 0.8 and 0.4 for \( K \) and \((-5)\) and 0 for \( a \).

\[
C = \frac{\log_{10} (1/K)}{4.65} \left( a^2 + 6a + (a/2) + \frac{2}{1 - K^2} \right)
\]

between the limits 0.97 and 0.8 for \( K \) and 40 and 0 for \( a \).

\[
A = 3.1(1 - K)^{0.89}a + (6.6 + 1.4K^2)
\]

\[
D = 1.25(1 - K)^{1.75}a + (6.6K^2 + 1.4)
\]

(IV)

These functions can be plotted in simple alignment charts with negligible errors (see Refs. 30 and 31).

Example. This example shows the method of applying these formulas. Figure 23 is a half section of a two-row wheel disk designed to run at 3600 rpm. The disk is divided into five sections, 1, 2, 3, 4, and 5. The curved profiles are assumed to be portions of hyperbolas whose value can be found from equation of Group II for each section. All sections are assumed to have the same thickness where each joins the adjacent section. Rings 1 and 5 are of uniform thickness, hence \( a = 0 \). Ring 4 has a positive value of \( a \), since thickness increases rapidly with the radius; \( a \) is negative in sections 2 and 3, since the thickness decreases as the radius increases.
Two of the radial stresses must be known. The radial stress at the bore may be taken as zero, as the shrink fit is supposed to be almost neutralised at normal speed by the centrifugal expansion of the bore, and at some overspeed the radial stress is zero. The outer radial stress of the blade load equals the centrifugal force of blades, shrouds, etc. The centrifugal force is

\[ CF = 0.00001421 N^2 d_1 \]

where \( N = \text{rpm} \); \( d_1 = \text{diameter, inches, to center of gravity of blades; } w = \text{total weight of blades, shrouds, etc., pounds.} \) The centrifugal force per inch of circumference at diameter \( d_2 \) at the edge of the disk \( = CF \pi d_2 \). This is assumed in the problem at 172 lb per in., at radius \( r = 15 \frac{3}{4} \) in.

The unknown radial stresses at the lines dividing the imaginary rings are taken as \( e \) between rings 1 and 2, as \( f \) between rings 2 and 3, as \( g \) between rings 3 and 4, and as \( h \) between rings 4 and 5. The data may now be collected in Table 2. Constants \( A \) to \( F \) may be calculated from equations of Groups II and III; approximate results may be calculated from equations of Groups III and IV or may be read from the alignment charts referred to above. The values given in Table 2 are calculated from equations of Groups III and IV.

**Table 2. Data on Stresses at Lines Dividing Imaginary Rings in Fig. 23**

<table>
<thead>
<tr>
<th>Ring No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_1 )</td>
<td>3.50</td>
<td>5.00</td>
<td>8.5</td>
<td>13.375</td>
<td>14.875</td>
</tr>
<tr>
<td>( r_2 )</td>
<td>5.00</td>
<td>8.5</td>
<td>13.375</td>
<td>14.875</td>
<td>15.75</td>
</tr>
<tr>
<td>( t_1 )</td>
<td>4.00</td>
<td>4.00</td>
<td>1.25</td>
<td>0.625</td>
<td>3.00</td>
</tr>
<tr>
<td>( t_2 )</td>
<td>4.00</td>
<td>1.25</td>
<td>0.625</td>
<td>3.00</td>
<td>3.00</td>
</tr>
<tr>
<td>( K = r_1/r_2 )</td>
<td>0.70</td>
<td>0.588</td>
<td>0.863</td>
<td>0.899</td>
<td>0.944</td>
</tr>
<tr>
<td>( a )</td>
<td>0</td>
<td>-2.19</td>
<td>-1.53</td>
<td>14.78</td>
<td>0</td>
</tr>
<tr>
<td>( R_1 )</td>
<td>0</td>
<td>e</td>
<td>f</td>
<td>g</td>
<td>h</td>
</tr>
<tr>
<td>( R_2 )</td>
<td>e</td>
<td>f</td>
<td>g</td>
<td>h</td>
<td>172</td>
</tr>
<tr>
<td>( A )</td>
<td>7.29</td>
<td>6.02</td>
<td>6.59</td>
<td>8.11</td>
<td>7.85</td>
</tr>
<tr>
<td>( B )</td>
<td>2.92</td>
<td>3.47</td>
<td>3.27</td>
<td>3.73</td>
<td>17.35</td>
</tr>
<tr>
<td>( C )</td>
<td>3.92</td>
<td>1.57</td>
<td>2.30</td>
<td>20.86</td>
<td>18.35</td>
</tr>
<tr>
<td>( D )</td>
<td>4.64</td>
<td>3.11</td>
<td>3.74</td>
<td>7.06</td>
<td>7.28</td>
</tr>
<tr>
<td>( E )</td>
<td>1.92</td>
<td>1.74</td>
<td>1.87</td>
<td>3.43</td>
<td>16.35</td>
</tr>
<tr>
<td>( F )</td>
<td>2.92</td>
<td>1.28</td>
<td>1.74</td>
<td>18.51</td>
<td>17.35</td>
</tr>
</tbody>
</table>

There is only one thickness at any one radius, hence, at the dividing line between any two imaginary rings there can be only one radial stress and one tangential stress. Take the line between rings 1 and 2, at radius 5 in. The outer tangential stress of ring 1 must equal the inner tangential stress of ring 2. Hence,

\[ T_2 \text{ (ring 1)} = T_1 \text{ (ring 2)} \]

or

\[ (D r_2^2 - E R_1 + F R_2) \text{ (ring 1)} = (A r_2^2 - B R_1 + C R_2) \text{ (ring 2)} \]

Similar equations can be written for each imaginary line at the given radius separating rings, substituting the values of radial stresses assumed above. These four equations can next be solved for the unknown radial stresses \( e, f, g, \) and \( h \). The equations for the equal tangential stress at the various radii are:

At radius 5 in.:

\[ 4.04 \times (5.0)^2 - 1.92 \times 0 + 2.92 \times e = 6.02 \times (8.5)^2 - 3.47 \times e + 1.57 \times f \]

At radius 8.5 in.:

\[ 3.11 \times (8.5)^2 - 1.74 \times e + 1.28 \times f = 6.59 \times (13.375)^2 - 2.37 \times f + 2.3 \times g \]

At radius 13.375 in.:

\[ 3.74 \times (13.375)^2 - 1.87 \times f + 1.74 \times g = 8.11 \times (14.875)^2 - 3.73 \times g + 20.86 \times h \]

At radius 14.875 in.:

\[ 7.06 \times (14.875)^2 - 3.43 \times g + 18.51 \times h = 7.85 \times (15.75)^2 - 17.35 \times h + 18.35 \times 172 \]

These equations can be solved easily by a substitution method as follows.

From equation for 5 in. radius,

\[ e = 0.246 f + 49.9 \]

Substituting this value of \( e \) in the equation for 8.5 in. radius and solving,

\[ f = 0.558 g + 322.5 \]

Substituting this value of \( f \) in the equation for 13.375 in. radius and solving,

\[ g = 4.71 h + 360.9 \]

When this value of \( g \) is substituted in the last equation it is found that

\[ h = 243 \]

Substituting in the three preceding equations, the following values are found:

\[ g = 1503 \quad f = 1091 \quad e = 318 \]
The tangential stresses at the various radii can be found by substituting in either side of the foregoing equations the values of $e, f, g,$ and $A$:

At radius 5 in.:  
$$T_5 = 4.64 \times (5.0)^2 + 2.92 \times 318 = 1045$$

At radius 8.5 in.:  
$$T_{8.5} = 3.11 \times (8.5)^2 - 1.74 \times 318 + 1.28 \times 1091 = 1068$$

At radius 13.375 in.:  
$$T_{13.375} = 3.74 \times (13.375)^2 - 1.87 \times 1091 + 1.74 \times 1503 = 1244$$

At radius 14.875 in.:  
$$T_{14.875} = 7.85 \times (14.875)^2 - 17.35 \times 243 + 18.35 \times 172 = 894$$

(Note. Some decimals have been dropped to simplify the solutions.)

The tangential stress at the bore is found to be:  
$$T_b = Ar_2^2 - BR_1 + CR_2 = 7.29 \times (5)^2 - 2.92 \times 0 + 3.92 \times 318 = 1430$$

At the rim:  
$$T_2 = Dr_2^2 - ER_1 + FR_2 = 7.28 \times (15.75)^2 - 16.35 \times 243 + 17.35 \times 172 = 817$$

The stresses at 1000 rpm are:

<table>
<thead>
<tr>
<th>$R$</th>
<th>$3.5$</th>
<th>$5$</th>
<th>$8.5$</th>
<th>$13.375$</th>
<th>$14.875$</th>
<th>$15.75$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T$</td>
<td>$318$</td>
<td>$1091$</td>
<td>$1503$</td>
<td>$243$</td>
<td>$172$</td>
<td></td>
</tr>
</tbody>
</table>

The stresses at intermediate points in any ring can be computed from the known values of radii and thickness at these points by equations of Group I.

At 3600 rpm the stresses above should be multiplied by the square of the ratio of the speeds, i.e., by $(3600/1000)^2 = 12.96$.

The resultant stresses are as follows:

<table>
<thead>
<tr>
<th>$R$</th>
<th>$3.5$</th>
<th>$5$</th>
<th>$8.5$</th>
<th>$13.375$</th>
<th>$14.875$</th>
<th>$15.75$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T$</td>
<td>$318$</td>
<td>$1091$</td>
<td>$1503$</td>
<td>$243$</td>
<td>$172$</td>
<td></td>
</tr>
</tbody>
</table>

Disks should be able to withstand an overspeed of 20% without exceeding the elastic limit. The stresses at this speed would be $(1.2)^2 = 1.44 \times$ stresses at normal load.

The maximum stress in the above table is 19,479 psi at radius 13.375 at 3600 rpm and at 4320 rpm (20% overspeed) it would be 28,050 psi, which is within the usual elastic limit of steel used in disks. If this stress is considered excessive, the design may be modified by thickening the metal at this point and allowing a smaller taper on the disk. The radial elongation in inches at any radius $r$ in inches is

$$y = (T - 0.3R)r + E_1$$

where $E_1$ = modulus of elasticity for steel = 29,000,000 for usual disk material. The radial elongation at 3600 rpm is:

<table>
<thead>
<tr>
<th>Radius</th>
<th>3.25</th>
<th>5</th>
<th>8.5</th>
<th>13.375</th>
<th>14.875</th>
<th>15.75</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elongation, $y$</td>
<td>0.00212</td>
<td>0.00212</td>
<td>0.00281</td>
<td>0.00474</td>
<td>0.00546</td>
<td>0.00639</td>
</tr>
</tbody>
</table>

Weaver states that the approximate method generally gives stresses about 1% too high. The great advantage of this method is the reduction of the time required for computation.

H. Iaerle (Ref. 32) presents a simple method for the determination of disk stresses from a single diagram, shown in Fig. 24, which can be applied to any disk profile, and yield results sufficiently accurate for all practical purposes. The general formulas for disk stresses given by Stodola, and stated above in discussing Weaver's methods, form the basis of Iaerle's method. These formulas reduce to much simpler expressions when applied to a disk of constant thickness, in which case $a = 0$ in the expression $t = Ct^a$, where $t$ = thickness of disk, inches, at radius $r$, inches, $C$ = a constant, $a$ an exponent governing the curvature and found by means of equations of Group II on page 8–29.

Let $T$ = tangential stress, psi, and $R$ = radial stress, psi.

Iaerle's methods are based on the assumption that

$$S = T + R = \text{sum of principal stresses}$$

and

$$D = T - R = \text{difference of principal stresses}$$

The following equations, derived from those of Stodola, apply to a disk of constant thickness:

$$S = (1 + V) \frac{u}{2g} (-U^2 + K_1)$$

$$D = (1 - V) \frac{u}{4g} (U^2 + K_2U - 2)$$

$$K_1 = \frac{4gEh_1}{(1 - V^2)u} \quad \text{and} \quad K_2 = \frac{8gE\omega^2h_2}{(1 - V^2)u}$$
where \( V \) = Poisson's ratio (\( = 0.3 \) for steel); \( u \) = weight of disk material, pounds per cubic inch; \( g \) = \( 32.2 \times 12 \) (in.); \( \omega \) = angular velocity, radians per second; \( U \) = tangential velocity of the disk, inches per second; \( E \) = Young's modulus (\( = 29,000,000 \) for steel); and \( b_1 \) and \( b_2 \) are constants depending on stress conditions at bore and rim as in Stodola's formulas.

The only variables at a given radius in the equations for \( S \) and \( D \) are \( K_1 \) and \( K_2 \). Series of curves as represented by the above equations are plotted in Fig. 24, each curve being based on a different value of \( K_1 \) and \( K_2 \), respectively. The problem is now simplified to selecting the proper curve or set of curves according to the specific details of the disk under consideration and finding \( S \) and \( D \) from the curves.

If \( S \) and \( D \) can be determined from the diagram, then

\[
T = S + D \quad \text{and} \quad R = S - D
\]
Figure 24 is plotted with stresses as abscissae and tangential velocities as ordinates. The heavy line on the diagram curving toward the left is the stress curve for a thin ring.

Examples. These examples show the application of this diagram. In the case of a disk of uniform thickness with a concentric bore and no load on the rim, the radial stress \( R \) at both bore and rim must be zero. Then

\[
S = T + R = T \quad \text{and} \quad D = T - R = T \quad \text{or} \quad S = D = T
\]

at both rim and at bore. That is, the \( S \) and \( D \) curves must intersect at both bore and rim, and these must be the same curves on the diagram at both places, for, since the boundary conditions are fixed, the values of \( K_1 \) or \( K_2 \) are the same throughout the disk. For example, let the rim speed be 500 ft per sec, and the bore speed 100 ft per sec. The same \( S \) and \( D \) curves must intersect at the 500-ft and 100-ft ordinates, as shown in Fig. 25, at 5500 lb. and at 22,000 lb., respectively, which must be the tangential stress at these points, since the radial stress at both points is zero. Tangential stresses at other speeds may be plotted by bisecting the distance between the \( S \) and \( D \) curves at each speed. Radial stresses can be found by halving the difference between the \( S \) and \( D \) curves and plotting from the zero ordinate.

The ease and rapidity with which problems relating to disks of uniform thickness can be solved by the \( S \) and \( D \) chart suggest the application of this method to disks of other than parallel profile. If the hyperbolic or other profile is replaced by a stepped disk consisting of a number of concentric rings, each of constant thickness as in Fig. 26, this method can be applied.

The assumption is made that, provided the steps are comparatively small, the stresses in adjacent concentric layers on either side of the step are inversely proportional to the axial dimensions or thickness \( t' \) and \( t \), as in Fig. 26. Hence

\[
\frac{t}{t'} = \frac{R'}{R} = \frac{T'}{T}
\]

Let \( \Delta R \) denote the increment (positive or negative) of the radial stress at the step. Then

\[
\Delta R = R - R' = R [1 - (t'/t)]
\]

The following expression is derived for \( \Delta T \):

\[
\Delta T = V \Delta R = VR [1 - (t'/t)]
\]

where \( V = \) Poisson's ratio = 0.3 for steel.

Combining these,

\[
\Delta S = S - S' = 1.3 \Delta R
\]

and

\[
\Delta D = D - D' = -0.7 \Delta R
\]

for the change in the \( S \) and \( D \) curves at the step. It is now quite easy to determine the \( S \) and \( D \) values at all points on the irregular disk profile, as shown in the example (Fig. 27), when applied to a disk with tapered sides, for a two-row wheel.

The mean diameter is 48 in., and the normal speed, 3000 rpm. The stepped disk, in substitution for the actual profile, is shown by the fine lines. Assumed peripheral velocity of the disk proper, 615 ft per sec. The net weight of blades, shrouts, etc., is assumed as 65 lb at normal speed, exerting a centrifugal pull of 2860 lb per in. of circumference. The net width of the disk at the periphery, after
deducting for dovetail grooves for blades is $t' = 2.2$ in. Hence, radial stress due to blade load $R$ (at periphery) $= 2860/2.2 = 1300$ psi. The tangential stress at the periphery may be assumed; let $T = 9000$ psi. Hence, at the periphery,

$$S = 9000 + 1300 = 10,300 \text{ psi}$$

$$D = 9000 - 1300 = 7,700 \text{ psi}$$

These values constitute the point of origin of the $S$ and $D$ curves across the outermost step. It is assumed that $R = 0$ at the bore. Hence $S = D = T$ at the bore, i.e., the curves intersect at the ordinate of the bore. If forced fit must be allowed for, $R$ at the bore may be chosen with either a positive or negative value to suit the particular conditions.

![Figure 27](image)

**Table 3. Stresses and Dimensions of Turbine Disk, Fig. 27**

<table>
<thead>
<tr>
<th>$U$</th>
<th>$t$</th>
<th>$t'$</th>
<th>$1 - \frac{t}{t'}$</th>
<th>$R^*$</th>
<th>$\Delta R$</th>
<th>$\Delta S$</th>
<th>$\Delta D$</th>
<th>$T$ Actual</th>
<th>$R$ Actual</th>
</tr>
</thead>
<tbody>
<tr>
<td>615</td>
<td>...</td>
<td>2.2</td>
<td>........</td>
<td>1,300</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>9,000</td>
<td>1,300</td>
</tr>
<tr>
<td>600</td>
<td>2.2</td>
<td>3.26</td>
<td>0.325</td>
<td>2,120</td>
<td>690</td>
<td>900</td>
<td>-484</td>
<td>9,400</td>
<td>1,300</td>
</tr>
<tr>
<td>590</td>
<td>3.26</td>
<td>1.80</td>
<td>-0.81</td>
<td>1,960</td>
<td>-1580</td>
<td>-2060</td>
<td>-1150</td>
<td>9,800</td>
<td>2,500</td>
</tr>
<tr>
<td>580</td>
<td>1.80</td>
<td>0.68</td>
<td>-1.65</td>
<td>4,100</td>
<td>-6760</td>
<td>-8800</td>
<td>-4750</td>
<td>10,900</td>
<td>6,800</td>
</tr>
<tr>
<td>570</td>
<td>0.68</td>
<td>0.47</td>
<td>-0.44</td>
<td>11,450</td>
<td>-5030</td>
<td>-6580</td>
<td>-3550</td>
<td>13,600</td>
<td>16,300</td>
</tr>
<tr>
<td>560</td>
<td>0.47</td>
<td>0.55</td>
<td>0.15</td>
<td>17,050</td>
<td>2480</td>
<td>3220</td>
<td>-1740</td>
<td>14,100</td>
<td>16,600</td>
</tr>
<tr>
<td>550</td>
<td>0.55</td>
<td>0.73</td>
<td>0.246</td>
<td>18,250</td>
<td>4500</td>
<td>5650</td>
<td>-3150</td>
<td>13,500</td>
<td>16,200</td>
</tr>
<tr>
<td>420</td>
<td>0.73</td>
<td>0.92</td>
<td>0.206</td>
<td>18,170</td>
<td>3740</td>
<td>4860</td>
<td>-2620</td>
<td>13,100</td>
<td>16,200</td>
</tr>
<tr>
<td>340</td>
<td>0.92</td>
<td>1.18</td>
<td>0.220</td>
<td>18,440</td>
<td>4060</td>
<td>5300</td>
<td>-2850</td>
<td>11,700</td>
<td>16,000</td>
</tr>
<tr>
<td>300</td>
<td>1.18</td>
<td>1.90</td>
<td>0.380</td>
<td>16,280</td>
<td>6200</td>
<td>8060</td>
<td>-4530</td>
<td>10,000</td>
<td>13,600</td>
</tr>
<tr>
<td>260</td>
<td>1.90</td>
<td>3.20</td>
<td>0.406</td>
<td>11,500</td>
<td>4670</td>
<td>6100</td>
<td>-3270</td>
<td>8,300</td>
<td>8,800</td>
</tr>
<tr>
<td>220</td>
<td>3.20</td>
<td>5.11</td>
<td>0.375</td>
<td>7,800</td>
<td>2920</td>
<td>3800</td>
<td>-2050</td>
<td>7,700</td>
<td>5,900</td>
</tr>
<tr>
<td>185</td>
<td>5.11</td>
<td>6.30</td>
<td>0.189</td>
<td>5,150</td>
<td>970</td>
<td>126</td>
<td>-680</td>
<td>8,300</td>
<td>4,200</td>
</tr>
<tr>
<td>82</td>
<td>6.30</td>
<td>6.30</td>
<td>0.0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>14,800</td>
<td>0</td>
</tr>
</tbody>
</table>

*Values at inner diameter of steps.

Beginning at the tangential velocity of the periphery, follow the $S$ and $D$ curves across the radial extension of the first step, i.e., from tangential velocity, 615 ft per sec, to tangential velocity, 800 ft per sec. At step 600, read $R = (S - D)/2$, find ratio $t'/t$ from the substituted profile, and calculate $\Delta R$, $\Delta S$, and $\Delta D$. Plot the new $S'$ and $D'$ for the second step and continue the $S$ and $D$ curves across it. The procedure is the same for all other steps. If tangential stress $T$ at the periphery has been chosen correctly, the $S$ and $D$ curves for the innermost step must intersect at the bore of the disk. If they do not, another value of $T$ at the periphery may be chosen and the process repeated until intersection occurs at the bore.

The axial thickness of the several concentric rings which form the substituted disk section correspond to the thickness of the true profile at points midway between the steps, and the stresses as determined by the $S$ and $D$ curves at these points coincide with the actual stresses in the original profile. Hence, if the horizontal distance between the $S$ and $D$ curves in the center of the stepped rings is bisected, points on the true tangential stress curve are obtained. A smooth curve through this series of points rep-
represents graphically the distribution and magnitude of the tangential stress $T$ throughout the disk. The radial stress $R$ at the middle of each step can be determined by measuring the distance of the tangential stress curve $T$ from the $S$ or $D$ curves, and plotting this distance from the zero axis at the various radii. Another smooth curve through these points indicates the magnitude and distribution of the radial stress $R$. Both these curves are clearly shown in Fig. 27. Values of $R$ and $T$ have been scaled off from these curves at radii corresponding to the various peripheral speeds $U$ in the first column of Table 3, and are tabulated in the last two columns for general information. They could be scaled at any other point if desired. The maximum stress in this disk is a radial stress of 16,600 psi and it exists just below the rim.

M. G. Driessen (Ref. 33) discusses Haerle’s method as outlined above and presents a larger chart for $S$ and $D$ curves than Fig. 24. The purpose of this paper is (1) to shorten the steps necessary to pass from one element of the disk to the next; (2) to provide an alternative to this cut-and-try method; and (3) to indicate a manner in which the results once obtained can be used for all other conditions under which this disk is used, i.e., for different loads at the circumference and for different speeds. The suggestions simplify Haerle’s method.

As in Haerle’s solution, the change of stress at a change of section is assumed inversely proportional to the thickness. Driessen proposes to find the new $S$ and $D$ curves for the new section as follows: $\Delta S = 1.3\Delta R$, and $\Delta T = -0.7\Delta R$. For the first section $S - D = 2R$ and for the second section $S' - D' = 2R'$. The change is $(S' - D') = (S - D) = 2(R' - R) = 2\Delta R$. Hence $\Delta S = (1.3/2)\Delta R$ and $-\Delta D = (0.7/2)\Delta R$. If $(S - D)$ and $(S' - D')$ are measured in inches on the chart, the distances $\Delta S$ and $-\Delta D$ in inches can be readily found, and the new points for starting the $S$ and $D$ curves for the changed sections easily are located on the chart and the quantities in psi can be read from the diagram.

In Haerle’s method the radial and tangential stresses at one diameter, usually the hub, were assumed, and the rim stresses determined. If the radial rim stress did not fit actual conditions, the disk was recomputed for other assumed tangential stresses at the hub until the desired radial rim stress results from the calculation. Driessen shows that if the tangential stress at the hub, $T_h$, is plotted against the radial stress at the rim, $R_h$, it follows a straight line. Hence when two points in this relationship are found, a straight line can be drawn through these points and any other $T_h$ can be determined from this line for the actual $R_h$ that prevails. This greatly shortens the computations.

Driessen points out that with large shrink fits, the stresses no longer are proportional to the squares of the speeds. He then outlines a method for computing and recording the limiting stresses of any disk, so that, if an earlier design of disk is considered for a new turbine, reference to these limiting stress values will determine its suitability.

Other methods for calculating disk stresses will be found in Refs. 3, 4, 21, and 34.

**SHRINK FITS FOR DISKS.** Experience has shown that disks tend to work loose on shafts if shrinkage allowance in the bore is insufficient. This may be due to enlargement of the bore from centrifugal stress at speed, or from relaxation due to high steam temperatures. Robinson (Ref. 35) and others discuss the first cause. Disk materials should be tested for relaxation at working temperature. Disks are usually designed for 100,000-hr operation without exceeding 80% of ultimate relaxation. The bore is proportioned to produce a stress in the cold state of 80 to 100% of the yield strength.

Disks frequently are secured to the shaft by a conical bushing. When a keyway is used on a disk, the key should serve only as a safety member, and exert neither radial nor lateral pressure. Sometimes two keyways are cut at opposite ends of a diameter to equalize stresses and to maintain balance. Disks of some turbines are held on the shaft by four fitted keys 90 degrees apart, by means of which expansion can take place equally in all directions and shrink fits are of less moment.

Often disks are bored with an allowance for pressing on the shaft of 0.0015 in. per in. of bore diameter, requiring 1 1/4 to 2 tons pressure per inch of shaft diameter to force each disk in place. Such press fit compensates for the increase in bore of the hub of the disk when under stress and prevents it from creeping along the shaft. Many builders shrink disks in place by heating the disk and chilling the shaft. This has advantages over forcing the disks into place by means of a hydraulic press.

In turbines having separate wheels (as opposed to the solid-rotor construction) it has been found necessary to be sure that the wheels stay central and drive the shaft under all temperature and stress conditions. This is accomplished by a pin bushing, keyed to the shaft with the wheel hub attached to the bushing by a number of radial pins. A separate key and bushing is provided for each wheel. When packing sleeves are placed between wheels, they are undercut to reduce heat transfer to the shaft should a rub occur. A lock or locating ring for each disk may be fixed on the shaft to prevent axial movement of the disk along the shaft.
DISK VIBRATION. (See Ref. 36.) Failures of some of the disks and buckets of early turbines led to intensive studies of vibration. These developed the fact that stationary disks and buckets vibrated harmonically in an even number of segments, between which were radial lines of quiet, called nodes. They appear as 4, 6, 8, 10, 12, or even higher numbers of nodes. Between these given frequencies the wheel was comparatively quiet. The higher the number of nodes, the higher the frequency of vibration and the less easily is vibration started. Both disk wheel and buckets vibrate as a continuous disk and must be considered as unit in this type of vibration. The frequency of vibration is determined by two factors: \( a \), the stiffness, and \( b \), the mass of the vibrating body. The stiffer the body, the faster it vibrates, and the more massive it is, the slower it will vibrate. Centrifugal force exerts a stiffening effect on a disk and increases the frequency at which the nodes appear.

The combined frequency of a particle, \( f_n \), due to the combined effects of stiffness and centrifugal force is \( f_n = \sqrt{f^2 + BN^2} \), where \( f_n \) = natural frequency of a particle of mass \( m \), with an elastic support of such stiffness that a force \( R \) is required to produce unit deformation; \( f_n = (1/2\pi)\sqrt{R/m} \); \( N \) = revolutions per second; \( B \) = a speed coefficient which varies with the design of the wheel and the type of vibration. \( B \) has a low value when vibrating sectors extend well into the wheel, and higher values as the number of nodes increases. \( B \) usually varies from 2 to 3.

The critical speed \( N_{cr} = f_n/\sqrt{(n/2)^2 - B} \), where \( n \) = number of nodes. Minor resonant speeds occur, the equation for the first of which is \( N_{cr1} = f_n/\sqrt{(1 + n/2)^2 - B} \). (See Ref. 36 for other values.)

When disks are revolved, their circumferences develop, at certain speeds, a form of wave motion which travels around the wheel circumference in the opposite direction to rotation. This wave motion has an even number of nodes which move around the wheel with the wave. Every part of the wheel rim thus vibrates over a period of time during each revolution. If the number of nodes is the same, the frequency of vibration of every particle along the edge of a disk wheel is the same either for standing vibration or for traveling waves. The speed of the traveling wave per second equals the number of complete vibrations of the corresponding standing wave per second multiplied by the length of a complete wave. For a traveling wave all particles vibrate through the same amplitude, but their time phases vary successively along the wheel edge.

When these backward traveling waves have the same speed backward that the wheel has forward, a standing wave results. This condition has been most conducive to disk and blade failure. The speeds at which the standing wave forms are called the wheel critical speeds. Wave trains in disks may be started by the application of a small extra force at a given point due to uneven nozzle dimensions, thick partitions, or other lack of symmetry. These waves persist after formation if the speed is suitable for the wave.

The effects of vibration are: (1) The wheel may burst. (2) The wheel may rub. (3) Buckets may fail from fatigue. Mathematics has been developed to predict disk stresses, and disks are made heavier than formerly, but one can be sure of a wheel only after a direct test. For this purpose, machines have been developed in which to test disks for vibration under working temperatures and speeds. Safe limits between the operating speed and the critical speed are, for 4 nodes, 15% above or below the operating speed, and for 6 nodes, 10% above or below.

When disk wheel critical speeds fall within these limits, the wheel is tuned, that is, metal is removed either from the disk itself or from the buckets until the critical speed has been shifted beyond the specified limits. Disks are smooth finished throughout so that no tool marks may form starting points for fatigue cracks. All sharp edges are rounded off for the same reason.

BUCKET VIBRATION. In addition to disk vibrations, the buckets themselves may be excited into vibration. These vibrations may be in the direction of minimum or maximum stiffness, or they may be torsional vibrations of a bucket group. They are started by irregularities in the steam flow path which result from obstructions such as nozzle partitions and struts in the exhaust chamber. The vibrations set up stresses in the buckets, bucket fastenings, shrouds, and lacing wires which add to the stresses already imposed by centrifugal force and steam load.

Because the magnitude and wave form of the exciting force are unknown, the absolute value of vibration stresses cannot be calculated. Therefore, blades must be designed by allowing a margin above centrifugal and steam stress, in which vibration may increase the stress but not exceed the endurance limit. A method of making this calculation is given by Timoshenko (Ref. 37).

The amplification factor (margin by which vibration may increase the stress) is determined largely from experience, but in buckets whose lowest natural frequencies are rela-
tively high harmonics of running speed (12 or more), vibration is difficult to excite, and vibratory stresses usually are small. For natural frequencies between the sixth and twelfth harmonic of running speed, a larger margin of stress must be provided for vibration. At harmonics lower than 6, large enough margin of stress cannot be provided, and it is necessary to design the bucket so that none of its lower natural frequencies coincides with a harmonic of running speed. To do this the first mode frequencies of the blades in the direction of minimum stiffness and of maximum stiffness may be calculated by the Stodola method (Ref. 38, p. 188). Calculations of the torsional mode and of the higher modes of bending vibration can be made but they are complicated, and tests are usually made for their determination. To adjust values of the various frequencies so that resonance is avoided at all harmonics below 6 requires adroit handling of bucket mass distribution, stiffness, root strength, number, position, and size of lashing wires and shroud, and selection of number of buckets in a group.

**Centrifugal Stiffening Effect.** In making frequency calculations, account must be taken of the increase in natural frequency due to the stiffening effect of centrifugal force. An approximation of this may be calculated (Ref. 38, p. 309) but most reliable information is obtained by test. The frequency at running speed may be obtained from

\[ f_{\text{rotating}} = f_{\text{stationary}} + K \left( \frac{\text{rpm}}{60} \right)^2 \]

where \( f \) is frequency and \( K \) is a coefficient which depends on bucket design and the type of the vibration. Values of \( K \) obtained from rotating frequency tests vary from 1 to 15, depending on the frequency and mode of vibration.

Tangential vibration may be set up in the buckets themselves at certain frequencies. Long slender blades must be so designed and tuned that the critical speed falls outside the limits above noted. Blades can be tuned by affecting their stiffness by: (1) soldering or welding the ends to a shroud; (2) adding an intermediate stiffening member between tip and base (lashing wire); and (3) decreasing the mass of the buckets. The position of the lashing wire has a very great effect on the resistance of blades to vibration.

The frequency of vibration of a reed is \( f = IC/r^2 \), where \( f \) = frequency, cycles per second; \( I \) = length, inches; \( t \) = thickness, inches; and \( C \) = constant of proportionality. For turbine buckets a factor \( S \) represents the scale of equivalent thickness. Comparing two buckets \( S_2/S_1 = f_2^2/f_1^2 \). Thus a test bucket is 14 in. long, with a frequency of 62. The scale of thickness for a similar bucket 24 in. long with a frequency of 37 is \( S_2/S_1 = 37 \times 24^2/62 \times 14^2 = 1.755 \). That is, the required thickness of the new bucket is 1.755 that of the one tested.

Buckets that may be subject to resonant vibration, which leads to fatigue failure, are heavier and somewhat wider than earlier forms and have both inlet and outlet edges rounded. Where possible, long buckets are made sufficiently stiff without the use of lashing wires.

For further details and the mathematics of the subject, see Refs. 36 and 39-44.

**Compound Vibration.** While the critical speed can be calculated by the preceding formula, and the expected vibration will occur at that speed, it often is noted in practice that the turbine goes through several critical frequencies before reaching the calculated critical speed, owing to the elastic scale of the foundations and the masses of the turbine parts. The foundation is subject to periodic forces induced by the turbine speed, to the influence of the superimposed mass, and to the elasticity of the supports. If deflection in the supports is large, vibration is increased. On the other hand, steel columns supporting the unit may be too stiff. The completed unit, therefore, frequently is studied by a vibrometer, and adjustment made both in machine balance and in foundations to secure the desired quiet operation at normal speed.

**DISK LOSSES.** Formulas for disk losses give conflicting values. Research has not definitely fixed these data. “Idle blades” are those revolving blades that are not receiving steam at the moment.

The following formulas may be used for disk and idle blade losses:

\[ L_D = \text{disk loss, kilowatts} = 0.042D^2 \left( \frac{u/100}{v} \right)^{1.9} \]

\[ L_B = \text{idle blade loss, kilowatts} = 0.19D^{1.28} \left( \frac{u/100}{v} \right)^{2.98} \]

where \( u \) = wheel speed at mean diameter, feet per second; \( D \) = mean diameter, feet; \( l \) = blade length, inches; \( v \) = specific volume of steam, cubic feet per pound; and \( s \) = fraction of mean circumference not receiving steam from nozzles. For a two-row wheel, a correction factor of 1.23 is applied to \( L_B \).
Another formula for disk loss is

\[ L_D \text{ (in kilowatts)} = 0.03D^3(u/100)^4/v \]

where \( D \) = diameter of disk at base of bucket feet; \( u \) and \( v \) are as above defined.

The idle bucket loss \( L_B \) can be diminished to 0.25–0.50 of the above value by enclosing the idle section by a channel-shaped ring or shield fastened to the casing.

The loss \( L_B \) so given above for rotation in the normal direction is approximate. For backward rotation, as in the reverse elements of marine turbines, \( L_B \) must be multiplied by factors ranging from 10 for short blades to 30 for blades 8 in. long.

Data and other formulas on disk and blade losses may be found in Refs. 3, p. 201; Ref. 4, p. 529; Ref. 16, pp. 77–78; Ref. 28; and Ref. 45.

Displacement loss or nozzle-end loss is the energy required to sweep out inert steam from idle bucket passages when they come into the active arc of steam admission. This loss \( L \), expressed as a fraction of the total steam flow, is

\[ L = \frac{nK_d}{AD} \frac{b}{\text{AD}} \]

where \( n \) = number of separate groups of nozzles interrupting continuous flow; \( b \) = bucket width of first rotating row, inches; \( A \) = percentage of circumference covered by nozzles; \( D \) = mean diameter of stage, inches; and values of \( K_d \) are:

<table>
<thead>
<tr>
<th>Velocity ratio</th>
<th>0.05</th>
<th>0.10</th>
<th>0.15</th>
<th>0.25</th>
<th>0.30</th>
<th>0.40</th>
<th>0.50</th>
<th>0.60</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_d ) (two-row)</td>
<td>0.02</td>
<td>0.04</td>
<td>0.07</td>
<td>0.12</td>
<td>0.14</td>
<td>0.20</td>
<td>……</td>
<td>……</td>
</tr>
<tr>
<td>( K_d ) (single-row)</td>
<td>……</td>
<td>0.03</td>
<td>0.05</td>
<td>0.09</td>
<td>0.11</td>
<td>0.15</td>
<td>0.19</td>
<td>0.235</td>
</tr>
</tbody>
</table>

SHAFT DIAMETERS for impulse turbines are a compromise between small diameters, which give low diaphragm-packing leakage, and the need for a reasonably stiff shaft, to insure operation above critical speed. Shaft sizes tend to larger diameters, favoring safe operation at some sacrifice of efficiency. With spherically seated bearings, the maximum deflection of shafts carrying disks varies from 0.005 to 0.030 in. Shafts with solid bearings are stiffer, and have about one half of these deflections.

Kearton suggests as a first approximation of shaft diameter: \( d = \frac{4\sqrt{ND}}{K} \), where \( d \) = mean diameter of middle portion of spindle, inches; \( l \) = bearing span, inches; \( N \) = rpm; \( D \) = average diameter of disks, inches; \( K \) = a constant; for land turbines \( K = 6,000,000 \) or higher.

Rotors of reaction turbines consist of solid shafts throughout, of solid shafts in the high-pressure section with rings or disks on the low-pressure end, of a hollow cylinder fastened rigidly to the spindle ends, or of other modified constructions. In reaction turbines with small radial clearances of blades, larger and stiffer shafts are used than with impulse turbines. With spherically seated, self-adjusting bearings, the maximum deflection varies from 0.001 to 0.005 in. When high mean blade speeds are necessary in the low-pressure section of a reaction turbine, disks of hyperbolic or conical cross section, either integral with the spindle end or placed as rings on the shaft, are used. These disks carry one to seven rows of blading.

A bore hole concentric with the finished rotor, with a maximum eccentricity of about 0.02 in., is drilled through each large shaft for complete periscopic inspection of its interior metallurgical structure. This hole assists in attaining uniform heating and in relieving stresses.

In one design, solid disks are welded together at their peripheries, and annealed to eliminate welding stresses, no through shaft being provided. The advantages claimed are that the stiff rotor runs far below its critical speed; the distribution of material is excellent; the spindle heats quickly and uniformly; the wheels cannot work loose; no keys or keyway are needed; the stresses are small; and the weight is low.

STRESSES IN DRUM AND OTHER SOLID ROTORS. Solid rotors are used in many central station and main propulsion marine turbines. For the reaction-type machine, these rotors are grooved only for the rotating blade fastenings. For the impulse type machine a deep groove is cut between the wheels to provide space for the nozzle diaphragm, thus giving a rotor that is essentially a shaft with integral disks.

Mean tangential stress is the principal design criterion in either construction. Stresses computed on the basis of elastic theory are of secondary importance. This is particularly true of the tangential stress at the inspection bore of such rotors. Plastic flow redistributes the stress in the relatively ductile material which is used, and the rotor approaches a condition where the tangential stress is uniform from periphery to bore. This redistribution takes place rapidly when a rotor is overspeeded. It takes place more slowly, but just as surely, when a rotor is operated at design speed and elevated temperature. The calculated
mean tangential stress then becomes the basis for the prediction of the overspeed bursting strength and the creep of the rotor at normal operating speed.

The mean tangential stress $T_m$, psi, is computed from the formula:

$$T_m = \frac{KI}{A} + \frac{P}{2\pi A}$$

where $K = 8 \times \left( \frac{\text{rpm}}{1000} \right)^2$ (for steel, where density = 0.28 lb per cu in.); $I = \text{rectangular}$

moment of inertia about axis of rotation of cross-sectional area of solid of revolution (one side of axis only); $A = \text{net cross-sectional area capable of carrying tangential stress on one side of axis of rotation only, square inches;}$ and $P = \text{total centrifugal force of buckets and any other parts not included in the section from which } I \text{ is determined, pounds.}$

The limit for this mean tangential stress is then decided on two considerations. First, from the standpoint of a reasonable factor of safety for overspeeding, it is limited to one-third the yield strength of the material used at the normal operating speed. Second, it is limited to a value which will give an acceptable creep during the expected life of the machine at the temperature for which the machine is to be used.

**STEEL FOR ROTORS.** Carbon steel is still employed where stress and temperatures permit its use. Its service record has been good. Its heat treatment is known. However, increased stress and temperatures of 825°F and above require the use of alloy steels.

Mochel (Ref. 46) points out that the rotor must have physical characteristics to enable it to withstand rotation stress and to transmit load; at the same time it must operate smoothly for many years. The latter requirement can be met only by such heat treatment as will relieve all internal stress from the forging. Otherwise slow relaxation at operating temperatures may cause deformations which lead to rough operation. These careful heat treatments are specified in Mochel’s paper and attention is called to factors influencing stress relief.

Mochel offers specifications for rotor steels, some of which are given in Table 4.

**Table 4. Rotor Steels**

<table>
<thead>
<tr>
<th>Type of Steel</th>
<th>Carbon</th>
<th>Alloy 1</th>
<th>Alloy 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>0.48 max</td>
<td>0.43 max</td>
<td>0.43 max</td>
</tr>
<tr>
<td>Manganese</td>
<td>0.40-0.80</td>
<td>0.40-0.80</td>
<td>0.40-0.80</td>
</tr>
<tr>
<td>Phosphorus</td>
<td>0.04 max</td>
<td>0.05 max</td>
<td>0.04 max</td>
</tr>
<tr>
<td>Sulfur</td>
<td>0.045 max</td>
<td>0.05 max</td>
<td>0.045 max</td>
</tr>
<tr>
<td>Nickel</td>
<td>.........</td>
<td>2.50 min</td>
<td>2.50 min</td>
</tr>
<tr>
<td>Chromium</td>
<td>.........</td>
<td>0.30-0.60</td>
<td>0.60 max</td>
</tr>
<tr>
<td>Molybdenum</td>
<td>.........</td>
<td>0.20 min</td>
<td>0.30 min</td>
</tr>
<tr>
<td>Vanadium</td>
<td>.........</td>
<td>.........</td>
<td>0.25 max</td>
</tr>
<tr>
<td>Tensile strength, psi</td>
<td>75,000</td>
<td>90,000</td>
<td>95,000</td>
</tr>
<tr>
<td>Yield strength, psi</td>
<td>40,000</td>
<td>65,000</td>
<td>65,000</td>
</tr>
<tr>
<td>Elongation in 2 in., %</td>
<td>20.0</td>
<td>18.0</td>
<td>18.0</td>
</tr>
<tr>
<td>Reduction in area, %</td>
<td>35.0</td>
<td>40.0</td>
<td>35.0</td>
</tr>
</tbody>
</table>

See also ASTM A293 covering Carbon-steel and Alloy-steel Forgings for Turbine Rotors and Shafts and ASTM A394 covering Carbon-steel and Alloy-steel Forgings for Turbine Bucket Wheels.

**DYNAMIC BALANCE.** All rotors must be in dynamic balance to avoid excessive vibration at high speed. Lack of balance may be due to nonhomogeneous disk or drum material, to slight eccentricity of the rotating masses, or to errors in workmanship. Static balance first is obtained by mounting the shaft on carefully leveled knife edges and applying counterweights until it remains at rest in any position. Disk wheels are balanced similarly by mounting them on true arbors.

Static balance of the assembled shaft and its disk or drum is no assurance that it will be in good dynamic balance, as two heavy masses in the completed rotor may be placed so as to form a static couple. This will cause severe vibration at high speed, because of the dynamically unbalanced centrifugal forces resulting from these masses. They may be balanced by providing additional masses to set up an equal and opposite couple.

Dynamic balancing of high-speed rotors is required where the center of weight does not coincide with the axis of rotation. Balancing usually is accomplished by adding weights in tapped holes in a disk or in the shaft periphery in a position opposite the heavy point. For rotors that operate below the first critical speed, a balancing plane at each end of the rotor is usually sufficient to correct unbalance. Rotors that operate above the first critical speed may require additional planes if accurate balancing is necessary. The use of additional planes permits balance weights to be placed more nearly in the same axial plane as the unbalance, thus eliminating the introduction of internal bending moments which tend
to bow the shaft at high speed. Balancing machines indicate the resultant of unbalances within the rotor but cannot show the distribution of the individual unbalances. When such unbalance distribution occurs in long rotors, the distribution of the balance weights may be estimated by analysis based on observation of the vibration at several speeds. In a well-balanced rotor, the mass center should be within 0.00025 in. of the true axis of rotation.

All manufacturing plants now have dynamic balancing machines in their plants where rotors, partly or completely assembled, can be quickly and accurately balanced. When it is necessary to balance a rotor in the field, it is run up to the operating speed, the high spot is marked, and counterweights are added by a cut-and-try method until balance is obtained. This operation is tedious and difficult, and final balance depends on the skill and experience of the operator.

**Critical Speed.** As every horizontal rotor deflects under its own weight, it is never possible to have the center of mass and the true center line of the shaft coincide. As the rotor speeds up, this eccentricity of mass results in an increasing centrifugal force tending to bend the shaft. At a certain speed this unbalanced centrifugal force neutralizes the elasticity of the shaft which resists deflection. The shaft deflection increases progressively, and, if unrestrained, failure would result. In an actual turbine the shaft will rub before this happens, causing considerable damage. The speed causing indefinitely large deflection of the rotor for a small initial eccentricity is the critical speed of the shaft. If the speed is increased above the critical speed, the shaft begins to straighten and tends to revolve about its true center of mass. Operation may be very smooth under these conditions, although other critical speeds may be encountered at still higher speeds.

The calculation of critical speed is difficult, except in the simplest forms of shafts and wheels. Methods of finding critical speeds will be found in Refs. 4, 21, and 47. Some engineers use this formula for a rough approximation:

\[
\text{Critical speed, rpm} = 188/\sqrt{y}
\]

where \( y \) = maximum shaft deflection, inches. The deflection \( y \) depends on bearing and coupling conditions, being less with solid bearings and couplings than with flexible ones.

Some small turbines run above their critical speed. Hence, in starting, they pass through this speed. To avoid serious deflections they are brought to speed quickly, passing through the critical speed so fast that no extreme vibration can occur.

**Running Speed** should be 25% below the calculated critical speed, for "stiff-shaft" machines. For flexible shaft machines (which normally run above the critical speed) the running speed should be 15% above the calculated critical speed. In general the actual critical speed is less than the calculated and tends to equalize these margins. Balance also affects the actual critical speed.

Rotor shafts often are bent if a rub occurs from any cause. This is due to local overheating on the side that rubs. The resulting expansion makes the rub worse, and finally the overheating causes a permanent set in the shaft. Danger from rubs of such character is being reduced by the use of thin metal labyrinth packings in diaphragm glands and on shrouds and the use of thin-tipped blades in reaction blading when shrouds are not used.

### 6. Turbine Details

**Thrust Bearings.** There is little end thrust in impulse turbines. In small units, thrust bearings frequently consist of shaft collars on each side of one of the bearings. Ball bearings are used as thrust bearings in other types of small turbine.

Large turbines of early design frequently had a marine-type collar thrust bearing. Most modern impulse turbines have a single runner on the shaft which revolves between babbit-faced bearing rings. Kingsbury thrust bearings are also widely used on large turbines.

The direction of thrust in operation is under control of the designer. End play of 0.010 to 0.015 in. may be permitted on the shaft with Kingsbury bearings. Thrust toward the exhaust tends to increase the clearances of end-tightened blades. Some builders, therefore, provide for thrust towards the steam end, where the thrust bearing is located, and provide position indicators to check shaft location. This insures minimum leakage, with end tightening.

**Kingsbury Thrust Bearings.** Reynolds' theory of lubrication has been applied to thrust bearings by Albert Kingsbury in America, and A. G. M. Michell in England, and their names designate forms of bearings widely used by turbine builders.

The Kingsbury bearing for steam turbines usually is designed to carry a load of 250 to 425 psi to allow for dirty or worn oil and to provide a wide margin of safety. Under ideal
conditions it could carry 3000 psi. In the fixed type of Kingsbury thrust bearing, liners are placed between the thrust bearing cage and the pedestal to fix the spindle position in the turbine. The bearing sometimes is mounted in a cage which can be adjusted axially from the outside to locate the moving blades relative to the casing. Thrust collars of large turbines are integral with the shaft.

The mean speed on Kingsbury thrust blocks may be over 200 ft per sec. However, to keep the diameter small, lower speeds generally are used. Ample quantities of oil at low velocities must be supplied. Thermocouples are sometimes placed in thrust shoes to check incipient failure.

**COUPLINGS.** The claw-type coupling, which has a certain amount of flexibility, consists of two halves, each in two parts. The inner sleeve, keyed to the shaft, has jaws on the outer flange. The outer sleeve has a plain flange to bolt to the other outer sleeve on the other shaft. A set of claws, cut on the other end of the outer sleeve, fits into the jaws of the fixed sleeve. The two halves of the coupling are held together by fitted bolts. Lubrication of the bearing surfaces of the jaws is usually insured by an oil catcher and holes through the jaws to the bearing faces. Hardened steel plates are used on the jaw wearing surfaces. Various forms of pin-type and other couplings are used on small turbines. Falk couplings are also in service.

**Flexible couplings** of the Fast, Waldron, and Poole types comprise two hubs, each keyed to its respective shaft. Each hub has external spur teeth cut on it, at the maximum distance possible from the shaft end of the hub. A sleeve surrounding these hubs is flanged and split vertically at its center for disconnecting the two shafts. The two halves are bolted together through the flanges. Each half of the sleeve has internal spur teeth cut on its bore at its outside end, which engage the external teeth of the hub. The sleeve is carried at each end by an oil-tight supporting ring. The error in alignment of the two shafts can be about ten times the clearance between the external and internal teeth, which are in an oil bath when in operation. These couplings have proved very satisfactory where expansion from heat, as on turbo drives for auxiliary equipment, makes it difficult to maintain correct alignment.

**Solid couplings** are extensively used on large turbines. They stiffen the shafts of both turbine and generator but require careful alignment. The coupling flanges may be integral with the shaft or shrunk on.

Various plans for securing true alignment include leveling pads on the bedplates, squaring and leveling coupling faces, stretching piano wire over the centers and checking up, and the use of surface gages. Allowance must be made in noncondensing units for expansion above the bedplate on heating up.

Turbines may be thrown out of alignment by pipe strains. Piping must have bends so arranged that no strains are transmitted to the turbine casing.

If the condenser is bolted to the turbine exhaust, consideration must be given to moments due to cooling water piping and to any eccentricity due to condenser water loading. If the condenser is not bolted to the exhaust, loadings due to the expansion joint may affect alignment.

**DUMMY PISTONS.** Balance, or dummy, pistons are used on reaction turbines to equalize the thrust toward the exhaust due to difference of pressure between the inlet and outlet of each row of moving blades and also to unbalanced pressure on annular surfaces when the drum is stepped-up in size. Two-step dummy pistons are used on large turbines, and single-step pistons on small units. Pipes fitted outside the casing equalize the pressures on the pistons with corresponding pressures on the bladed sections. Any unbalanced thrust is taken by the Kingsbury bearing.

**Dummy packing** in many designs consists of several axial knife edges projecting from the stationary elements which extend toward radial lands on the revolving pistons, thereby increasing the number of throttlings in a given axial length and reducing the distance between shaft bearings. Experimental data are used as a basis for calculating losses from such dummies. Dummies must provide space between throttling points to act as expansion chambers, must dissipate heat readily should contact occur, and one material must wear away rapidly with little heat generation. With radial clearances, dummies frequently consist of plain radial strips alternately deep and shallow, sealing against alternately low and high lands on the spindle dummy piston. Radial dummies depend on throttling through the small radial clearance at the tips of the projecting teeth for reducing the leakage. Both radial and axial sealing strips have been used. Sealing strips of nickel ribbon, chromium stainless steel, and other alloys are used.

**Area of Dummy Pistons.** Goudie (Ref. 4) says that the dynamic thrust on the blades in an axial direction is usually less than 1% and never exceeds 2% of the total thrust in
reaction turbines, and may be neglected. For the annular area \( A_d \) of the balance piston for each cylinder of reaction blading, he gives on p. 424, the formula

\[
A_d = \frac{P_2 A_1}{P_1 - P_c} + \frac{\sum(P_i - P_o)a}{2(P_1 - P_2)}
\]

where \( P_i \) = pressure on front of dummy, psia; \( P_2 \) = difference of pressure, psia, on any annular drum area at \( A_1 \); \( A_1 \) = annular area of any step-up in drum at entrance; \( P_c \) = condenser or back pressure, psia; \( (P_i - P_o) \) = drop in pressure in a group of blades of constant mean diameter; \( a \) = annular area between drum and casing at a group of blades; pressures are psia; areas are in square inches.

Leakage of Steam through Dummy Pistons. H. M. Martin (Ref. 48) discusses the leakage of steam through dummy pistons and submits a formula which is claimed to check within 1% of the actual loss. From this formula the following equation is derived:

\[
w = 0.4722 A \sqrt{\frac{P_1}{v_s}} \left( 1 - \frac{1}{N} + \frac{1}{\log r} \right)
\]

where \( w \) = steam leakage, pounds per second, through whole dummy; \( A \) = area available for flow of steam at any dummy constriction, square inches; \( P_1 \) = initial pressure, psia; \( v_s \) = specific volume of steam at pressure \( P_1 \), cubic feet per pound; \( N \) = number of throughings in the dummy; \( r \) = ratio of initial to absolute final pressure over the dummy = \( P_1/P_2 \), where \( P_2 \) = the pressure on the rear side of the piston, psia. This is the formula generally used by American turbine builders. When the high-low type is used with sealing strips, the coefficient is reduced from 0.4722 to 0.40. Other values, based on tests, are used by some manufacturers.

Clearance of Dummies. Radial dummies must be used where the balance piston is distant from the thrust and considerable expansion can occur. Radial clearances on such seals are about 0.001 in. per in. of diameter. Side-contact dummies are ground to fit by revolving the spindle slowly and drawing up on the thrust until contact occurs. They are afterwards set, when thoroughly heated, by drawing up on the thrust, the spindle revolving very slowly, until first contact is heard by listening on the casing. The thrust then is moved to obtain the desired running clearance, which varies from 0.004 in. on small reaction turbines up to 0.012 to 0.015 on large turbines. Balance pistons near the middle of the spindle require somewhat larger clearance. When finally set, all thrust blocks are locked after allowing sufficient end play for lubrication. This adjustment should be checked periodically to detect wear in the thrust collars due to clogged oil supply or dirty oil.

Labyrinth seals. Glands must be provided in all turbines where the shaft leaves the casing. Impulse turbines also require glands where the diaphragms between stages encircle the shaft.

Carbon ring glands comprise several carbon rings, each in its own compartment of a cast-iron or steel case. The several segments of the ring are pressed together by a garter spring or an arched flat spring. The rings usually are divided into three or four segments. Clearances on the shaft diameter are from zero to approximately 0.006 in., depending on the size of the shaft.

Dummy piston glands are used in certain reaction turbines.

Labyrinth glands for diaphragms consist of cut or inserted teeth projecting from the shaft toward a smooth stationary casing, from the casing toward the shaft, or from both with the teeth alternating, similar to radial clearance dummy pistons. The ends of these teeth are knife-edged, and usually clearance of 0.002 in. per foot of bearing span is allowed in design. These constrictions throttle the steam into a larger space, where it forms eddies and restricts flow. Alternate high and low lands break up steam flow and reduce leakage.

The high-pressure labyrinth gland, where the shaft leaves the casing, can be made in two sections. The longer inner part seals against the internal pressure, the outer part against atmosphere. Steam is withdrawn from or supplied to the gland at this middle point. Steam above atmospheric pressure must be supplied at the intermediate point in the exhaust-end gland to seal it against air leakage, which would destroy the vacuum. An alternative common arrangement uses a water seal impeller capable of sealing up to 5 to 10 psig pressure to avoid the necessity for sealing steam at the outer end. Marine turbines (variable speed) must, however, use the steam sealing arrangement, because the water seal is not effective at reduced speed.

High-pressure gland leakage either is used in a low-pressure gland, or led from an intermediate pressure take-off to a feedwater heater. Provision usually is made to admit live steam to both low-pressure and high-pressure glands to seal them on starting. The clearance between the teeth of these glands and the shaft or casing is usually 0.005 to 0.020 in. Materials used for labyrinth glands are babbitt, aluminum, brass, and bronze, and for
TURBINE DETAILS

high temperatures, stainless or other alloy steels. An eductor is provided on these glands to prevent steam leakage into the room.

Some designs of labyrinth glands are planned so that any rubbing causes the parts automatically to separate. Glands made of packing rings of anti-friction metal are used on some small turbines which hardly warrant the expense of a more elaborate type. Relief grooves cut in the shaft on each side of labyrinth packings prevent bowing of shafts by localizing the heating if rubbing occurs.

**WATER GLANDS**, where the spindle leaves the casing, consist of a small impeller or paddle wheel, fastened to a long sleeve or hub on the shaft, which revolves in a gland casing. This gland is supplied with water under a pressure of 10 to 15 psig. The water is unable to leak along the shaft, as the action of the impeller holds it in a solid ring against the outer casing. On the other hand, air cannot leak into the turbine because of this solid ring of water under pressure much greater than atmosphere. Several forms of combined water and labyrinth glands can be steam sealed when the turbines are run at speeds too low to maintain the water-gland seal. The governor may regulate the supply of water or steam to the glands. Proper leak-off passages are provided for steam and water when needed.

Water glands generally are used outside the labyrinth glands on high-pressure ends of impulse turbines, to prevent steam leakage into the turbine room. On account of the high temperature at this gland, condensate must be circulated through it, discharging into the feed system.

**Power Required by Water Gland.** Guy and Jones (Ref. 49) state that experiments indicate that the power required by a water gland with the paddle completely immersed in water is \( P = 6u^3D^2/10^6 \), where \( P \) = horsepower required; \( u \) = peripheral velocity of paddle wheel, feet per second; and \( D \) = diameter of paddle wheel, feet. Under actual operating conditions the paddle is not fully immersed on both sides, and tests indicate that the power required is about 50% of that given by the formula.

The water required by water glands varies from 0.5 to 2% of the condensate, but little of this is lost, since at the low-pressure end the vapor from the gland enters the exhaust and is recovered in the condenser. At high-pressure glands, the water must circulate, and the heat it absorbs can be fully recovered. Water glands usually seal at any speed above one-half of normal speed.

**Steam required by casing glands** can be estimated by a chart to solve problems in labyrinth packing presented in *Engineering*, Vol. 128, p. 65, 1929. Martin's formula is used by many builders to compute diaphragm gland leakage. (See also Ref. 50.) Some builders use coefficients which depend on clearances, arrangement of lands, eddy chambers, etc., and which vary from 0.35 to 0.472 in the Martin formula.

**BEARINGS.** Turbine bearings may be divided into two classes, self-oiling and forced lubrication. Self-oiling bearings, used only on small turbines, generally consist of babbit-lined cast-iron or cast-steel shells, with oil supplied by oil rings revolved by the shaft. Bearings for large units and for reduction gears always have forced lubrication from the main oiling system.

Large turbines may have spherically seated, self-aligning bearings. In reaction turbines, the spherical seats take the form of three or four pads under which are placed steel-adjusting shims of varying thickness. Clearances at the ends of the blades can be equalized by changing the shims and thus shifting the position of the shaft relative to the casing. A clearance of 0.008 to 0.012 in. is provided above the top pad to prevent the bearing being pinched, and to allow the shaft to be self-aligning.

Because impulse turbines do not require close clearances over the ends of the buckets, their bearings have plain spherical seats. When light shafts are used in some forms of impulse turbines, it is desirable to decrease the deflection and increase the critical speed, by using solid parallel bearings. Such bearings also are used for reduction gearing where accurate alignment is essential.

Large bearings consist of cast-iron or cast-steel shells split in half horizontally and lined with babbit. They are relieved for 20 to 30 degrees at the sides above and below the joint, except for 3/4 in. at each end, by reboring slightly oversize with a separating plate between the halves. The lower halves are scraped to fit the journal for an arc of 120 degrees at the bottom. Bearings usually are bored 0.001 to 0.003 in. large per inch diameter of journal. With forced lubrication, oil usually is supplied both at the top and sides of the bearings. Oil throwers either are turned on the shaft, or attached to it at the outer end of the bearing, to prevent escape of oil. Oil guards are provided on the bearing cover for the same purpose. Provision is made for the escape of entrained air from the bearing pedestals.

The design of the spindle usually fixes the size of the bearing. The rubbing velocity of the journal should not exceed 150 ft per sec. The bearing pressure is found by dividing
the total load on the bearing by the product of its length and diameter. A safe limit of this pressure is up to 200 psi. The ratio of bearing length to diameter varies from 0.75 to 1.5. Bearings are made shorter than formerly, as this reduces the total length of the turbine. The design should be such as to reduce spattering and splashing of oil, which lead to oxidation troubles and acid formation.

Pressures may reach 1000 psig at contact surfaces of gears driving the main oil pump and governor, because of minute errors in tooth pitch and profile that result in momentary increases in tooth loads. Ample lubrication of such gears is a necessity.

Oil whip has been experienced on turbine bearings at high rpm. While this is still under investigation, it appears desirable to maintain bearing loads on such turbines above 100 psig, to prevent oil whip. Special pressure bearings also are used, in some instances.

Bearings on some small turbines are heated by conduction through the casings and through the shaft. Such bearings may heat to 250 °F, and a heavy oil is required. Usual bearing temperatures range from 125 °F to 160 °F; occasionally the oil leaves at 175 °F.

Bearings depend for their proper functioning on the supply of a thick wedge-shaped oil film on the side of the bearing where the shaft turns downward. This film spreads and maintains a separation of the two metal surfaces. There is no true coefficient of friction, but the shearing action of the oil offers a resistance to motion which is the so-called coefficient of friction. Kraft (Ref. 1) states that this factor is 0.008 for a bearing of good workmanship. The heat, Btu per minute, generated in a bearing, \( H = \pi d N \mu W + (12 \times 778) \), where \( d \) = bearing diameter, inches; \( N \) = rpm; \( \mu \) = so-called mean coefficient of friction between journal and bearing; \( W \) = total load on the bearing, pounds.

The oil required per bearing can be estimated from the formula:

\[
\text{Kilowatt loss per bearing} = \frac{0.26zN^2LD^2}{M \times 10^8}
\]

where \( z \) = absolute viscosity of oil in centipoises (usually 14); \( M \) = clearance ratio of bearing expressed as clearance in inches per inch of diameter (\( M \) is usually 0.001 to 0.003 in.); \( N \) = rpm; \( L \) = bearing length, inches; and \( D \) = diameter, inches.

Oil required in gallons per minute = 0.8064 \times \text{kilowatt loss}

This quantity is based on assumptions of specific heat of oil = 0.5; temperature rise in bearing = 30 °F; specific gravity of oil = 0.9; and an arbitrary multiplier 1.6 to allow for journal heating, etc.

Hot turbine bearings may be caused by insufficient oil, when oil pipes or oil grooves are plugged up or the supply fails. Sometimes bearing clearances are insufficient to admit the proper amount of oil, particularly at the sides of the bearings. Heating also may be due to too heavy an oil, to emulsified oil, or to old contaminated oil.

7. REDUCTION GEARING

See Design and Production Volume of Kent's Mechanical Engineers' Handbook for design of reduction gearing. See also Marine Engineering, Section 15 of this book, for additional data on marine turbine gearing.

The efficiency of reduction gears is difficult to determine by mechanical methods. A common method of calculating gearing efficiency is to measure the friction heat carried away by the lubricating oil and to allow for radiation from the gear casing. Carefully made laboratory tests by both input and output measurements, and by heat measurement on single-reduction gears, show practically the same losses. Large single-reduction gears have shown efficiencies of 98 to 99%. The efficiency of small single-reduction gear sets ranges from 96 to 98%. Double-reduction gearing has given efficiencies varying from 88 to 97% on test. During World War II nearly all high-power combatant vessels were propelled by cross-compound steam turbines driving double reduction geared propellers. Such units had both light weight and small space as prime advantages, and were highly reliable in service.

8. TURBINE LUBRICATION

OILING SYSTEMS. Small turbines provided with ring-oiled bearings require only the maintenance of a suitable supply of pure mineral oil, changed at frequent intervals, in the reservoir below the bearings. The outer surfaces of the pedestals dissipate the heat generated by friction.

Large turbines have a completely self-contained oiling system, including fine wire screens to remove particles, an oil pump, an oil cooler, suitable piping systems, and an oil reservoir.
Some turbines have a centrifugal pump on the main shaft of the unit. It delivers oil at high pressure to the governor system and also to an ejector which induces additional oil flow at lower pressures for shaft bearings and thrust bearing.

Sometimes the oil pump is driven by gearing from the main turbine shaft or, if reduction gearing is used, sometimes from the slower speed shaft. Separate motor-driven oil pumps are in use with some turbines. The pump, usually of the rotary gear type, supplies oil at pressures between 50 and 200 psig pressure. The volumetric efficiency of gear-type pumps ranges from 70 to 80% with hot oil. Oil pressures to the bearings range from 5 psig to 25 psig, usually obtained through a reducing valve. Relief valves discharge any surplus oil to the reservoir.

An auxiliary turbine- or motor-driven centrifugal oil pump is placed on medium-size and large turbines to circulate oil through the bearings before starting the unit or when the main unit is on the turning gear. Such pumps are equipped with regulators which start them automatically when the oil pressure drops below a minimum safe value. Low oil pressure alarms are often installed. Many turbines are automatically stopped by a tripping device on the throttle valve when oil pressure fails.

Oil coolers, built in many forms with brass, admiralty, or copper cooling coils, should be readily accessible for cleaning if raw water is used. Frequently, condensate is used as the cooling medium so that less cleaning is necessary.

Oil may circulate through the tubes or outside of them with shell baffles. The heat transfer coefficient in oil coolers is low, varying from 10 to 20 Btu per sq ft per hr per °F temperature difference. Coolers lower the oil temperature about 20 to 30 °F. Oil should pass to the bearings at temperatures between 105 and 140 °F. Temperatures leaving the bearings range from 130 to 160 °F. It is frequently specified that sufficient oil-cooler capacity shall be installed to keep the maximum temperature of the oil below 150 °F when using cooling water at the maximum temperature specified by the purchaser for summer conditions. Only enough water should be circulated through the oil cooler to maintain the desired minimum bearing oil temperature. Cooling water pressure should be lower than the pressure of the oil passing through the cooler.

Oil-pump capacity is fixed by the total amount of oil required by the bearings and governing system, together with a liberal margin to provide for pump slip, air vents, etc. Oil-pump capacities furnished by one builder for 3600 rpm units varies from 50 gal per min on 1000 kw units to 1200 gal per min on 100,000 kw turbines. On 1800 rpm units, capacities range from 290 gal per min on 50,000 kw units to 1500 gal per min on 165,000 kw turbines. (See Ref. 51.)

Oil reservoirs vary in size with the type of turbine and with the several manufacturers. A frequent requirement for units of 5000 kw and over is that the capacity of the oil reservoir at the turbine shall be such that it will take 5 to 10 min to circulate a quantity of oil equal to the tank capacity. Reservoir capacities range from 100 gal on a 500 kw unit to 3000 gal on a 100,000 kw, 3600 rpm turbine, and 4000 gal on an 1800 rpm, 165,000 kw unit. With smaller turbines the period for complete circulation shall be not less than 5 min. Special precautions must be taken to minimize the danger of fires in oil reservoirs. Emergency drain lines to reservoirs, CO₂ and other nonflammable flooding equipment, and location of the reservoir in a fireproof room below the turbine are used. Oil temperatures in reservoirs should exceed ambient to prevent condensation of moisture in the inside of the reservoir.

Oil Piping. Seamless copper tubing, with brazed flanges and cast-iron or brass fittings, is sometimes used for oil piping. Steel piping and steel tubing are extensively used, with welded nipples for branches and joints. This material should be well pickled before assembly to remove mill scale and rust. Threads are made tight against oil pressure by shellac or similar material, or by seal welding.

Oil velocities through pressure oil piping vary from 5 to 15 ft per sec. All drain lines should be sloped to drain back to the oil reservoir by gravity. Drain pipes are proportioned to run only half full, to allow for natural venting. All oil piping is placed on the opposite side of the turbine from the steam piping and stop valve, to lessen fire risk in case of pipe failure.

These recommendations for the construction of oil piping lessen dangers from fire.

Cleanliness and good housekeeping are of primary importance. Avoid accumulations of combustible material. Wipe up spilled or dripping oil. Prevent oil collection in pits or other depressions. Arrange piping for ready inspection, but enclose sufficiently so that it is not used as a ladder or for hanging things on. Design with good supports and as nearly free from stress as possible.

Main oil lines should be of standard-weight pipe, or seamless tubing of equal wall thickness, with extra heavy steel flanges and steel fittings. Use welds as extensively as possible. Erect barriers around all oil joints when they must be used. Weld all threaded joints. Provide lock washers on all flange bolts. Male and female flange joints are preferred. Use metal-to-metal unions.
STEAM TURBINES AND ENGINES

No connections should be made with less than 1/4-in. pipe size, with shut-off valves as close as possible to the main pipe. Screen reservoir vents and locate them at a distance from the steam line. Reservoir covers should be self-closing.

Avoid gage glasses and use a mechanical indicator for oil level. Nipples for pressure gages must have restricted openings.

Keep oil system, particularly oil storage or reservoir, as remote as possible from high-temperature steam connections.

TURNING GEAR. A motor-driven turning gear to rotate the shaft at slow speed when the turbine is taken out of service is frequently furnished for units of 10,000 kw rating and higher. Rotation prevents shaft distortion due to uneven cooling, keeps the rotor and casing at more uniform temperatures, and permits more rapid starts. The turning gear is also used when warming up a cold machine. Turning-gear speeds vary from 1 to 40 rpm on the main shaft with some preference for 25 to 30 rpm. The higher speeds maintain a better oil film in the bearings. Full oil circulation is desirable, even with the turning gear in operation, to prevent journals and bearings from overheating by conduction from the turbine or governor rotor when shutting down, since bearing metal tends to become plastic around 300 F. Oil to bearings should be at nearly normal operating temperature before the unit is brought up to speed.

LUBRICATING OIL FOR TURBINES must be a properly refined, highly filtered, pure mineral oil, free from alkali or acid, and inhibited against rust, corrosion, and oxidation. While organic acids form from oxidation of unsaturated compounds or those containing sulfur, oxygen, or nitrogen, in the presence of water, that oil should be chosen which has the lowest organic acidity, not only when fresh but also after continued use. Organic acidity, expressed as the amount of potassium hydroxide required to neutralize 1 gram of oil, should not exceed 0.10 mg, preferably 0.05 mg. This is known as the neutralization number. (See Ref. 52.)

Water may enter the oil system from steam leaks at glands, from condensation of atmospheric moisture on the walls of reservoir, pedestals, or oil piping, or by other means. Oil should be able to separate rapidly from water and not form emulsions. If, however, emulsions are formed, they should be quickly separated on heating. Oil must be free from components of low boiling point, to maintain constant viscosity. It should have little tendency to break down and form sludge, when agitated at the actual operating temperature and mixed with air and water. The ideal lubricating oil should have maximum adhesion and minimum cohesion.

Foaming oil may be caused by water in the system or it may be due to air entering through leaks in the oil-pump suction piping or strainer. Air becomes mixed with oil in the bearings and in return oil piping. Foaming can be corrected by removing the water and preventing the entrance of the air. Turbine oils should have nonfoaming characteristics and should permit the entrapped air to separate quickly from the oil in the reservoir before it is recirculated.

Oils oxidize at various rates. Oxidation increases rapidly above 150 F. Water in oil causes oxidation of iron surfaces. Some oils have natural oxidation inhibitors added. The addition of about 10% of old oil to a new batch of oil has been found effective in inhibiting oxidation, and is commonly practiced.

Oils which decompose and age rapidly in service deposit a hydrocarbon sludge in bearings, piping, and oil cooler. Besides plugging passages in bearings, thrusts, and couplings, it forms an insulation on the cooling coils. Higher oil temperatures result, causing still more rapid aging. Such oils should be removed and filtered, the system thoroughly cleaned, and new, higher-grade oil purchased. Cotton waste must not be used to clean the oil system, as lint causes trouble later by stopping up the oil passages. Helpful experiences with lubricating systems are presented by Lowe (Ref. 53).

Reduction gears have higher tooth pressures per square inch than the usual bearing pressures and require a heavier oil than bearings. In some cases, these gears have their own oiling systems, using a heavy oil. When both gears and turbine bearings are on the same system, either a medium or a heavy oil may be used. Hot oil may be supplied to turbine bearings and only the gear supply circulated through the oil cooler.

FILTRATION AND PURIFICATION OF OILS are necessary with moderate and large-size turbines, usually accomplished in one of these ways: (1) continuous by-pass system; (2) batch system; and (3) continuous by-pass batch system.

The continuous by-pass system continuously takes out a certain fraction of the oil for treatment. The batch system takes out a large amount at intervals. The continuous by-pass batch system is a combination of the other two methods. Separation of water and sludge may be secured by: (1) Passing the oil through a centrifugal separator or filter such as the DeLaval and Sharpley, which separate the substances on the basis of their specific gravities. This removes water and dirt, but not alkalis and acids or soluble
sludge. (2) Bag filters, which remove insoluble sludge but will not remove acids or soluble sludge. (3) Separating tanks, in which the oil is cooled and where water and sludge can settle out and be withdrawn. Usually oil reservoirs are provided with piping which allows water that gets into the oil to overflow automatically when it settles in the reservoir.

Turbine builders furnish purchasers with general specifications for turbine oils. Non-inflammable oils for lubricating systems have been used but are not popular.

**CHARACTERISTICS OF OIL.** Oil must be free from inorganic acid, asphaltum pitch, resinous substances, animal and vegetable oils, and soap or other substances added to give body to the oil. It should contain no dirt, grit, or other suspended matter. The specific gravity should be between 0.86 and 0.88 at 60 F.; flash point, not below 334 F.; fire test, 375 F. The higher the values of flash and fire point, the better, particularly in plants using highly superheated steam. Acids can be detected by blue litmus, which turns red in the presence of acids. Animal and vegetable oils, used as adulterants, can be detected by a milk-white emulsion which forms when the oil is shaken with a strong solution of borax. After standing in a cool place, the clear mineral oil will be at the top, the borax solution at the bottom, and the emulsion in between.

**Sludge** in oil plugs piping, coats oil cooler tubes, and causes hot bearings by reducing the lubricating value of the oil. Sludge forms as a result of agitation of impure oil in the presence of water and air. To test the sludging properties of oils, a sludge accelerator has been developed which produces sludge in 1% of the time in actual service. For a description of this equipment and information on its use, see Ref. 54. This report also contains detailed instructions for making flash, fire, viscosity, pour, total acidity, alkalinity, corrosion, carbonization, and demulsibility tests on lubricating oils.

Table 5 gives representative oil specifications for several services.

<table>
<thead>
<tr>
<th>Table 5. Oil Recommendations for Various Types of Service</th>
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<tbody>
<tr>
<td>Properties</td>
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<td>----------------------------------------------------------</td>
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<tr>
<td>Saybolt Viscosity, Seconds</td>
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<td>at 100 F</td>
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<td>at 130 F</td>
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<td>at 210 F</td>
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<td>Flash point, °F</td>
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<td>Corrosion resistance test</td>
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<tr>
<td>Normal temperature of oil to bearings</td>
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<tr>
<td>Minimum oil temperature when starting</td>
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</tbody>
</table>

**Notes:**
1. For marine service, where it is sometimes desirable to have only one grade of oil for both main propulsion and auxiliary units, an oil having a viscosity of 375 to 425 seconds Saybolt at 100 F can be used with the understanding that power losses in bearings will be increased when higher viscosity oils are used.
2. In some installations, normal bearing temperatures may exceed 180 F because of high ambient temperature, restricted ventilation, etc. For bearing temperatures higher than 180 F, consult turbine oil supplier for recommendation.
9. GOVERNORS AND CONTROL

Turbines may be governed by throttling or by nozzle control. In _throttling governing_, the main steam is throttled to some pressure lower than line pressure before it enters the first-stage nozzles, thus reducing the flow to a value just sufficient to maintain the load. In _nozzle governing_, a series of valves is provided to admit steam under control of the governor to each of a number of separate first-stage nozzle groups. The only valve under throttling action is the last one being opened; the others are wide open, with only a small pressure drop across them. Nozzle governing gives a better steam rate than throttling at partial loads and is widely used on large units.

_Throttle governing_ is used on turbines having full peripheral admission, as on reaction turbines. The steam-rate curve with throttle governing can be made to approximate that of nozzle governing by selecting a _most efficient load_ at a relatively low rating percentage and by providing additional capacity by secondary, tertiary, and even quaternary valves admitting steam to lower stages of the turbine.

_Governors for Small Turbines_. The speed of most small turbines is controlled by a powerful centrifugal governor mounted directly on the end of the rotor shaft. Movements of the governor spindle are transmitted through a single lever to a regulating throttle valve, usually a balanced valve. The governor weights are carried on levers by tool-steel knife edges in tool-steel bearings requiring no lubrication. In some types, ball bearings are used instead of knife edges. One type has weights which roll on flat tempered steel springs with no pins, bushings, bearings, or knife edges. Sometimes the lever connecting the governor spindle to the regulating valve also is provided with knife edges. These governors must be examined frequently to see that no parts are loose or worn. Speed regulation of these governors is about ±2%.

_MECHANICAL OIL RELAY GOVERNORS_ are used on most moderate-sized and large turbines both condensing and noncondensing. They include a centrifugal governor geared to the turbine shaft, and an oil pump which may also be geared to the shaft, mounted on the shaft itself, or motor driven. The governor operates a relay or pilot valve which admits oil at pressures between 100 and 200 psig to a servomotor that controls the steam-governing valves. A restoring mechanism provides _feedback_ to reset the relay control after each servomotor movement. The servomotor acts directly on a single balanced _steam-control valve_ or a series of valves through a cam and lever mechanism. Overload valves also are opened by the servomotor.

When a series of steam-inlet valves are used, they may be lifted by a bar to which the valves are loosely attached. The valve stems are of various lengths to assure opening in sequence. This is known as the _bar-lift system_.

Some industrial turbines, where a range of speed is desired and close regulation over the load range is not necessary, have hydraulic orifice governors, using oil as the operating medium.

_Oil governors_ are used which depend for their functioning on the variation in oil pressure supplied by a centrifugal pump mounted on the turbine shaft. This oil pressure varies as the square of the turbine speed. A pressure transformer changes the small pressure variations due to speed changes into relatively large pressure changes on the operating relay. The basic impeller pressure is balanced by a spring adjustment which can bias the turbine speed. If the speed changes the change in oil pressure moves the relay, operating ports which control the flow of oil to the operating piston of the steam-valve servomotor. A compensating follow-up linkage insures stability. Various additional control devices can be incorporated readily in this type of _governing system_, and they frequently are added on special request.

_Pressure-regulating governors_ are frequently applied to turbines driving pumps, etc., for controlling extraction pressures on single- or double-extraction turbines and sometimes for exhaust pressure control. These are usually rather simple regulators that admit sufficient steam to the turbine to maintain a given pressure or pressure differential at the desired point, such as at the discharge of a pump, the extraction line, or in the exhaust. Pressure-regulating systems must be arranged to coordinate with the regular speed and overspeed control governors.

_Governors on turboalternators_ have their regulation fitted to the service for which the unit is designed. In general, 4% variation between no load and rated load is allowed for electric lighting service. Greater variation may be desirable where the load changes are large. Too close regulation is not necessary or desirable, as it may cause the load to surge between machines that are in electrical parallel. Surging often can be reduced by increasing the regulation of the governors. Turbines operating in parallel with hydro units usually do not require close regulation. (See Refs. 55 and 56.)
FREQUENCY CONTROL. Since the introduction of electric clocks, constant frequency must be maintained on the power system of public utilities, regardless of load. With major generating systems interconnected in power pools, one station fixes frequency for the pool. Frequency, system, and tie line loads can be maintained by hand operation of the governor auxiliary speed control, but automatic frequency regulators are used on many generating systems.

REGULATING VALVES. Balanced valves are used in nearly every case for regulating valves of the throttling type, particularly on small units with direct-connected governors. These were formerly of the double-seated poppet type. The difficulty of keeping the two seats tight when using superheat, led some builders to adopt the balanced single-seated valve. Valves and seats are generally of hardened steel. Such valves must be kept absolutely steamtight when closed. Stainless or other alloy steel, with stellite trimmings, is generally used for high-temperature steam valves.

CONTROL VALVES. In large units, a multiplicity of control valves is provided to give better efficiency over a wide range of load. Single-seated valves, streamlined to reduce eddy losses, are used on nozzle-controlled turbines. They are bar- or cam-operated by single or multiple servomotors. Control valves usually are streamlined to reduce losses. Some are of inverted mushroom form. The seat is generally well rounded and of venturi shape to regain a substantial portion of the velocity head of the flow through the valve. The bearing portion of the valve on its seat is frequently spherical in shape to insure line contact on the seat.

STEAM STRAINERS in front of the throttle valve prevent grit, pipe scale, or other foreign substances from reaching the turbine. They are made of stainless steel wire mesh for small units and drilled stainless steel plate for large units, with holes 1/16 to 1/8 in. in diameter, and generally in basket form, although flat strainer cages are used on some units. The combined area of these openings should be two to three times the valve area to reduce throttling losses. The steam chest may be integral with the turbine casing or may be separate. When separate it is bolted rigidly to the foundation and connected to the casing by several flexible seamless steel pipes. The latter construction, sometimes adopted for high temperature units, isolates the hot valve chest from the cooler casing. In general, the main steam line should be flexible enough to impose only negligible stress on the turbine casing.

OVERSPEED GOVERNOR. All turbines have some form of automatic overspeed governor. A small centrifugal governor sometimes is provided in the end of the shaft. At 10% above normal speed, it automatically trips either the stop valve, the governor valve, a separate overspeed valve, or combinations of these. The usual form of overspeed governor consists of a bolt-headed pin in the shaft end, at right angles to its axis. The centrifugal force on the unbalanced bolt head is opposed, up to 10% overspeed, by a spring. At that speed, the bolt flies out, striking a trigger, which releases the spring-loaded stop or other valve and permits it to close. On some large machines, the trigger operates an oil relay valve which causes a second small piston to unlatch the main stop or other valve and allows it to close. A modification of this form of governor is an eccentric ring on the governor spindle or main shaft, the centrifugal force of the eccentric part being balanced by a spring. At a given overspeed, this spring is compressed, the eccentric ring strikes a trigger, and the control valve is closed either by a spring or by an oil relay.

LOAD RELEASE. An anticipator device, sometimes provided on large turbines, causes the main stop valve or the governor valves to close on complete and sudden loss of load, before the rotor has speeded up sufficiently to operate the overspeed governor. See Ref. 57 for a description of supervisory instruments and their operation.

AUTOMATIC EXTRACTION PRESSURE CONTROL. Present governing specifications for automatic extraction turbines generally call for compensated control systems with interconnections between the control mechanisms of both the speed-governing and pressure-regulating systems. Simultaneous action of both systems takes place in response to a change in either speed or controlled steam pressure. With an increase in load, the controls open the control valves to supply additional steam, and simultaneously adjust the pressure-regulating valve to maintain extraction pressure. An increased demand for extraction steam causes the valves controlling steam to the portion of the turbine beyond the extraction point to close partially, thus tending to reduce the power. The slight decrease in power is simultaneously compensated for by an approximate correction of the main control valves, immediately followed by an accurate correction by the speed governor. Thus there is no change in total load.

THROTTLE VALVES, now going out of general use (see Stop Valves, below), often were opened against a heavy spring. The handle for opening the valve is fastened to the valve stem by a latch which may be released, usually by an oil relay, when the overspeed governor acts, thus forming a combined throttle and trip valve. It serves as a throttling
valve only when coming up to speed. The unit always should be shut down by tripping
this throttle valve. Some throttle valves are controlled by an oil-operated piston. Over-
speed governors on most large units can be reset when the turbine speed drops to about
2% above rated speed. On smaller units the speed may drop considerably below rated
speed before the trip mechanism can be set.

Other overspeed governors on small turbines release a flap valve, which closes of its
own weight. Still others control a butterfly valve in the steam supply.

STOP VALVES. The old-fashioned throttle valve, applied to hundreds of central
station turbines, now is largely superseded by an oil-operated stop valve. This valve,
installed at the steam inlet to the turbine, is normally held open by a spring-loaded hy-
draulic cylinder. Upon overspeed, the oil is vented to a drain line, and the valve closes
almost instantaneously. The stop valve has no intermediate position; it is always wide
open or closed. Usually the control valves must be closed, and a small by-pass line around
the stop valve opened to bleed steam into the steam chest, before the hydraulic cylinder
has enough power to open the stop valve. This is a safety precaution which prevents
accidental overspeeding in starting up the unit. The prime function of the stop valve is
to prevent overspeeding and to isolate the turbine from the main steam line.

SPECIAL CONTROLS. Central station turbines sometimes are supplied with one or
more of these controls, among others: Steam flow limit device to control the maximum
output fixed for the turbine; low-steam-pressure limit device to reduce the load in order to
permit the boiler to restore pressure; low-frequency block load device to provide for the
rapid increase of steam flow should frequency drop.

10. CASINGS, DIAPHRAGMS, AND EXHAUST

CASING MATERIALS. Casings of turbines for low steam pressures and temperatures
frequently are made of cast iron, split horizontally, with all steam and exhaust connections
in the lower half. The steam chest sometimes is carried around one end in the form of a
cored passage, with hand-controlled valves to the nozzles. When high-temperature steam
is used, casings must be made of alloy steel. Casings for pressures of 1200 psig and higher
have been made of forgings.

Casings for large turbines usually are split horizontally, so that the upper half can be
removed to permit examination and repairs. Many single-cylinder casings have a bolted
vertical joint between the high- and low-pressure sections. High-pressure sections of
turbines carrying temperatures over 425 °F are made of cast steel.

For steam temperatures above 825 °F, casings are cast of chromium-molybdenum steel.
Above 1000 °F the casings are of cast austenitic stainless or other alloy steel. Many indus-
trial turbines, marine turbines, and the low-pressure casings of central station turbines
are fabricated from rolled flat steel plate.

CASING CONSTRUCTION. While strength is an important consideration in casing
design, rigidity under varying temperature conditions is imperative. Hence, they usually
are cast without ribs on the outside and with the smallest flanges consistent with strength
requirements. They approach the smooth barrel form, and all abrupt changes in diameter
are avoided. If circumferential flanges or ribs are provided, they must not be too deep
or distortion will occur because of slow adjustment to changes in temperature. Cored
passages are avoided.

Stationary dummy rings are generally cast separately and fastened to the inside of the
casings. Stationary blade rings of reaction turbines are cast separately and are registered
in a barrel-shaped casing in different ways to provide free expansion under rapid tempera-
ture changes. Double-shell construction frequently is used in high-pressure elements
to lessen the pressure and temperature differences across the flanged joints, by confining
the high pressure and high temperature to the inner casing.

Horizontal flanges of casings are lapped or scraped to make them steamtight in opera-
tion. Sometimes the joint is coated with thin linseed oil before closing.

Flange joints of high-pressure high-temperature turbines are difficult to keep tight.
Boils creep and relax. Flange faces sometimes are undercut at the bolt diameter to in-
crease the moment effect of the outer section. Narrow flange faces are used to increase
unit compressive stress on the joint and to increase responsiveness to temperature changes.
These flanges are thick, usually about 2 1/2 times the bolt diameter. Sometimes slots
are cut through the outer flanges to permit adjustment of temperature after sudden changes
of load. (See Refs. 58 and 59.)

The high-pressure end of casings often is supported on sliding feet, located near the
horizontal center line, to allow free expansion. Transverse alignment is obtained by ver-
tical keys. In some designs, a flexible support, in the form of a steel I beam or channel
under the high-pressure end, holds the unit in sidewise and vertical alignment but permits longitudinal expansion through flexure of its web. With such construction, the cylinder is bolted solidly to the pedestal which rests on the I section.

Casings must be designed to withstand bursting pressures 50% above normal working load. For low-pressure cylinders, normal pressure may be taken as 30 to 50 psig. The deflection between supports should be a minimum. The pull of the vacuum on the exhaust outlet, when an expansion joint is used, must be considered in calculating this deflection. The lifting force of a spring-supported condenser also must be considered.

Cylinders should be blanked off and subjected to about 25 psig steam pressure for about 24 hr before final boring, to season the metal and relieve casting stresses. Cylinder supports should be as near to the center line of the machine as possible, to avoid vertical changes in shaft alignment from temperature changes. Usually the generator end of the casing is fastened rigidly to the bedplate with provision for expansion at the opposite end.

The thickness of the metal casing is usually calculated by the thin cylinder formula, 
\[ t = \frac{pd}{2f}, \]\nwhere \( t \) = thickness, inches; \( p \) = internal pressure, psig; \( d \) = internal diameter, inches; and \( f \) = allowable stress in the material, psig. See Stodola (Ref. 3) for further details.

Drainage grooves to remove moisture are provided in the last stages of large turbines. They are capable of removing about 25% of the moisture present.

**BOLTS.** Flange bolts are put as close to the inner diameter of the casing as possible, leaving sufficient metal (at least 1 in.) between the inner wall and the bolt hole. This requires a deep flange and long through bolts, closely spaced. Sometimes these bolts are turned down to a diameter slightly less than the bottom of the thread to allow uniform extension. Usually, such bolts are hollow. They are heated electrically to a predetermined extension, the nut is set up snug, and the bolt is allowed to cool. The nuts are usually cylindrical with a small hex on top to permit closer spacing of bolts. Some manufacturers mill the threads on these bolts with a larger clearance between nut and bolt threads than standard practice. Threads are sometimes tapered to give an even bearing on all threads, when stretch occurs under load. (See Ref. 60.)

Various alloys have been used for bolts. ASTM B-14 specifies a chrome-molybdenum-vanadium steel, tensile strength 100,000 psi, yield point 80,000 psi. The bolts are cold stressed to about 55,000 psi, equivalent to a stretch of 0.001 in. per inch of bolt length. Some use similar alloys heat treated to 125,000 psi, ultimate strength. Hardened steel washers are used under the nuts. Threads are lubricated by various materials, such as mixtures of graphite, white lead, and penetrating oils.

Bolts are proportioned on their relaxation stress at the steam temperature. After operation for a period at high temperature, creep and relaxation lower the bolt stress to its relaxation stress, where it remains indefinitely. (See Ref. 59.)

**DIAPHRAGMS**, when cast in halves, must be made sufficiently strong and rigid to avoid undue deformations due to temperature and pressure differences. They may be dished towards the high-pressure side to increase strength. Diaphragms in large machines are of cast steel, fitting on centralizing supports in turned circumferential slots in the casing. These diaphragms sometimes carry a shoulder projecting almost to the adjoining diaphragms and just outside the blade ring.

Diaphragms for high-pressure high-temperature units are built up by welding. Nozzle partitions are usually of stainless material. Their inner and outer bounding walls, whether integral or separate, are highly finished. They are welded to an inner half disk and to an outer half ring; this results in robust rigid nozzles of high efficiency, in strong diaphragms. The welded parts are then finish turned to size.

Diaphragms for low-pressure stages with long radial nozzles are made with steel plate nozzle partitions of suitable form, cast in position in the diaphragm, or of formed partitions. Other methods of constructing both high- and low-pressure diaphragms are used by the various manufacturers.

Design constants for diaphragms are largely based on experience, since split diaphragms are not easily analyzed mathematically. They must be designed to have a limited deflection, to avoid gland rubs, and frequently are pressure-tested under design conditions to confirm their suitability.

**CASING CORROSION**, often in the form of grooves in the interior casing walls between rows of blading, has occurred in certain units. This grooving often leads to a considerable loss of metal which adversely affects the structural integrity of the machine. Corrosion is worst in the neighborhood of the dew point of the turbine. The cause of this corrosion has been traced to corrosive gases which attack the metal in the presence of moisture. The attack first occurs at the dew point. Oxygen, carbon dioxide, sulfur dioxide, and hydrogen sulfide may contribute to corrosive action. These gases may enter
the boiler with the feedwater or may result from the action of boiler compounds. Corrosion can be prevented by proper deaeration and chemical treatment of feedwater.

**EXHAUST HOODS** are made of cast iron or of welded steel plate. Exhaust outlets of large turbines are stiffened and reinforced against vacuum by cast-in or bolted partitions, or by heavy staybolts with pipe spacers. These partitions or struts must not be placed close to the last blade row since they may cause undue blade vibration. Exhaust outlets usually are designed to have about twice the annulus area of the last stage blading. Where possible, these outlets should diffuse the leaving steam to produce the lowest absolute pressure at the discharge of the last blade row.

Kraft (Ref. 1) states that the design of the exhaust should fulfill the following conditions:
1. The transformation into pressure of the kinetic energy of the steam leaving the blades must begin as soon as possible after the last wheel; it must take place as quickly as possible yet it must be gradual. (2) The curvature must be gentle; the area of the steam passage in a direction perpendicular to the flow must increase continuously, causing the steam velocity to decrease gradually. (3) The jets of steam escaping from the last wheel must not interfere with each other and cause eddies. (4) As few guiding surfaces as possible should be provided in the exhaust casing, to keep down the friction losses against the walls.

(See Ref. 61 for data on exhaust hood losses.)

**ATMOSPHERIC EXHAUST.** Provision may be made for an atmospheric exhaust on condensing turbines, to discharge exhaust steam to atmosphere through a relief valve. These valves are difficult to keep tight. The valve and piping are expensive and large. In modern operation, where machines are started and shut down under vacuum, it is inadvisable to operate noncondensing, particularly when high superheat is used; hence relief valves are seldom, if ever, used. Many modern turbines are installed with a lead blow-out or explosion diaphragm set at 2 to 5 psig. These turbines have a special vacuum-control device which trips the stop valve if the vacuum drops below a certain value, thus avoiding rupture of the blow-out diaphragm in all but the most extreme situations.

# 11. ERECTION AND OPERATION

**TURBINE FOUNDATIONS.** Satisfactory turbine operation can be assured only when proper foundations are provided, particularly for large units. Turbine builders provide customers with detailed instructions covering the design and construction of satisfactory foundations.

The foundation should have sufficient weight and mass to hold the turbine rigid against vibration. The subfoundations are determined by the character of underlying material, but must be designed so that the concentrated weights of the turbogenerator are spread over an ample area to prevent springing or settling. A reinforced solid mat 2 to 5 ft thick is desirable except when the foundation is on rock.

The turbine foundation and its base should be independent of main building or other foundation structures. Floor beams should not rest on turbine foundations. Overhanging cantilever supports should be avoided. Large mass is desirable to insure smooth operation. Space must be provided at the generator end for removal of the field. The floor for this section must be strong enough to carry the field. Space for dismantling should be provided on floors of sufficient strength.

**Preferences for foundation materials** are, in the order named (1) steel heavily encased in concrete; (2) reinforced concrete; and (3) bare steel.

The foundation generally consists of heavy columns tied together by beams. Necessary openings for condenser connections, extraction and main steam lines, water and oil pipes, electrical conduits, etc., should be provided, with necessary reinforcing around these openings.

Manufacturers differ in their methods for calculation of loadings. In general, the vertical load includes the weight of the turbine and generator plus the reaction of the condenser or the pull of vacuum if a vertical expansion joint is used, plus 25 to 50% allowance for auxiliary parts and impact. Turbine outline drawings show the distribution of these weights.

To design bracings, horizontal loads at the shaft center line are assumed equal to 50% of the rotating weights on four-pole units and 100% on two-pole units. A longitudinal load of 10% of the total weight of the turbine generator is also assumed to act at the center line. Provision must be made for sliding forces when one end of the turbine slides on pads, and for thrust forces when thermal expansion is taken by flexing plates. In areas subject to earthquakes, an additional horizontal thrust must be assumed in the design.

**Resonance deflections** must be avoided. They are given by the various builders, and can be obtained before foundations are designed. In general, deflections of beams and
columns should not exceed 0.020 in. for 1800 rpm units and 0.010 in. for 3600 rpm machines.

A concrete mixture is recommended that has a compressive strength of not less than 3000 psi when 28 days old. This corresponds roughly to a 1-2-3 mixture. Solid walls are preferred to columns. Care must be taken in pouring to get a monolithic structure. Temperature effects from hot exhaust pipes of superposed units must receive special consideration.

Structural steel foundations are sometimes used where subsoil conditions are bad, and loadings must be the minimum. Generally, the space between beams is filled with concrete; steel columns should be adequately braced; beams must have ample gusset plate stiffening; and main foundation members must be protected from radiant heat. Cantilever beams should be avoided where possible. Deflections should not exceed those given for reinforced concrete.

THE DESIGN AND CONSTRUCTION OF TURBINE FOUNDATIONS usually is carried out by the purchaser. While the turbine builder will assume no responsibility for such plans, it is desirable to submit them to the builder for his approval to avoid mistakes, omissions of openings for drains, ducts, etc., and for suggestions as to resonance, rigidity, etc.

Some small turbines are leveled by steel shims on the steel work, and bolted in place. In other cases, the steel is designed so that a concrete top is provided and 1 to 1 grout is poured around the bedplates in the usual way, after leveling with wedges.

Floors between adjacent turbines and between foundations and walls should be supported on steel work independent of the turbine foundations, to prevent vibration being carried to other structures. In many large stations only an operating platform about 6 ft wide is built around the turbine bedplate, leaving the remainder of the engine room open to the basement floor. This provides better light around the condenser auxiliaries, and the condenser equipment is more accessible to the crane. This platform should be carried on independent supports as cantilevered walkways may be subject to vibration. Dismantling and repairs can be carried out on the basement floor. If carried out on a floor at the turbine elevation, heavy steel or concrete flooring and floor supports would be required.

Structure Vibration. In case of vibration trouble, a study of the structure by means of a vibrometer may locate the cause of the difficulty, provided the revolving elements are in dynamic balance.

ERECION OF STEAM TURBINES. In erection, if turbine and generator are on one shaft, it is necessary only to level the unit carefully. If the set consists of a turbine with a separately driven unit, as a generator or pump, or if gears are used, care must be taken to insure not only correct levels but also accurate alignment, particularly at the couplings. Leveling pads are furnished on many bedplates. Levels should be checked with the turbine heated to operating temperature, as the alignment of some designs is affected by expansion due to the heating of the exhaust end. Leveling is done by steel wedges at frequent intervals under the edge of the bedplate, allowing a space of 2 in. under the bedplate for grouting.

There should be sufficient wedges to insure that no deformation occurs between them. After the grout has set for a day, they can be withdrawn. Some engineers raise the outboard bearings of both turbine and generator slightly to obtain more accurate alignment at the coupling by allowing for the deflection due to weight of the turbine rotor and revolving field. Leveling always should be done with all weights in place. Detailed information on erection will be found in the instruction books of manufacturers. Kearton's Steam Turbine Operation also contains many data and suggestions for alignment methods, erection, etc.

When leveling is completed, the bedplate is grouted in place, using one part of high-grade portland cement and one part of sharp sand, mixed in a thin grout and well rammed to prevent air bubbles under the base. That should cover the foot of the bedplate at least 3 in. Provision usually is made to grout in bedplates supported on steel. Occasionally lead, 1/2 to 1 in. thick, is placed in place of grout on structural steel foundations. Grouting should be done before any steam, exhaust, or extraction piping is connected. Foundation bolts are drawn up tight after the grout has set.

Oil piping generally is blown out with steam to remove dirt that may have got in during shipment, and is allowed to dry thoroughly before assembly. Steam lines should be blown out with full steam pressure, to remove any pipe scale, welding shot, or dirt.

STEAM PIPING. Piping to turbines must provide for expansion and contraction of the turbine and of the pipe line, so that the least possible stress will be imposed on the turbine. Pipe vibrations also must be avoided.

The steam chest containing the control valves is either (1) bolted directly to the turbine casing, (2) cast integral with the cylinder, or (3) fastened firmly to the foundation with
flexible pipe connections from it to the turbine casing. The stop valve, usually of the automatic spring-closing type operated by the overspeed governor, is usually separately supported.

It is desirable that there should be minimum stress on the stop valve from piping. Allowance should be made for the maximum turbine and piping expansion from cold to working temperature. This expansion may be halved by cold-springing the piping into position for half the total expansion. Torsional stresses should be avoided. See Section 6 for data on piping.

Welded pipe joints are generally used on high-pressure piping. Several forms of weld are used, but all are stress relieved by electrical heating on completion. Welding icicles must not be allowed in steam lines as they eventually break loose. High-temperature steam piping of alloy steel has heavy walls, and consequently provision for expansion is difficult. Special care is necessary to avoid excessive thrust on the turbine. (See Arts. 11 and 12, Section 6.)

Drains must be provided for all low points in piping, steam chests, bends, etc. The manufacturer's drawings show the location of these drains.

Exhaust piping must be absolutely airtight. When an expansion joint is used, consideration must be given to the collapsing effect of the vacuum, and provision made by suitable brackets, anchor bolts, or other devices to prevent distortion.

PLANT DESIGN. Turbogenerator plants are arranged in either of two ways. (1) The turbogenerators are set with their axes at right angles to the boiler-room wall. Condensers are placed directly under the turbines. The units are spaced to give ample room between them for operating and for dismantling. This distance often is fixed by the space required by auxiliaries, usually located in the basement. Space must be provided at the ends of the units to permit the field to be withdrawn from the generator. With large units, this results in a wide turbine room and long crane span. (2) Large turbines often are placed with their axes parallel to the boiler-room wall. The distance between turbines is fixed by the space necessary to withdraw the generator field. The condenser usually is set at right angles to the turbine axis. Space must be allowed for withdrawing condenser tubes, and also on the side of the turbine for dismantling.

Unit sets have been developed in the smaller outputs where turbine, generator, exciter, condenser, and pumps form a simple unit.

The condenser should be placed as close to the turbine exhaust as possible. In small sizes, a bellows-type copper expansion joint may be placed between turbine exhaust and condenser inlet. Large units are built with no expansion joint, the condenser being bolted directly to the exhaust nozzle. The supporting pads on the condenser shall rest on springs. They are adjusted to carry all, or a major portion of, the weight of the condenser when full of water. They provide a certain amount of flexibility to care for expansion from temperature changes. All water, air pump, condensate pump, and atmospheric exhaust connections should have expansion joints when the condenser is bolted firmly to the turbine in this manner. Condensers on some large units are fastened to their foundations with a flexible connection to the turbine exhaust outlet.

Headroom under the crane above engine-room floor must be sufficient to permit lifting off the turbine upper-half casing and removing the rotor for repairs or inspection.

Piping. The purchaser furnishes piping for cooling water to and from the oil cooler, to and from the generator air cooler, for water glands, for steam to and from auxiliary oil pump, and for drain connections. Manufacturers' drawings show where these connections can be made. The manufacturer usually supplies all oil piping, piping to steam-sealed glands, and water-cooling piping to bearings, when used.

Cooling air for small generators may be taken from below the unit and, after passing through the generator, may be discharged into the engine room. In this method, much dirt enters the generator and adheres to the windings. In case of a short circuit, a fire in the windings usually results from the continued supply of fresh air. Larger generators, up to 10,000 kw capacity, now are furnished with a closed system of air circulation and with air coolers supplied with condensate or with cooling water from other sources. These eliminate the necessity of cleaning generators. In case of a short circuit the fire is smothered by exhaustion of the limited oxygen supply in the closed system. Small breathers with viscous air filters provide for changes in volume in the closed circuit.

HYDROGEN COOLING. Standard units of 15,000 kw and above, as shown in Table 1 (p. 8-14), are furnished with hydrogen-cooled generators built into the closed casing of the generator. This casing is explosion-proof, being designed for an internal pressure of about 80 psig. Normal hydrogen pressure is 0.5 psig. When hydrogen pressure is increased to 15 psig the generator load may be increased about 15%, with the same temperature rise as at normal load and 0.5 psig hydrogen pressure. Special seals prevent hydrogen leakage where the shaft passes out of the casing. These seals employ oil as a sealing medium, and
this oil absorbs hydrogen. Separate equipment dehydragenates the oil and returns it to the main system. Provision must be made for purging the generator of air before admitting hydrogen. This purging is usually done by carbon dioxide which displaces the air and, in turn, is displaced by hydrogen.

**CARE AND OPERATION OF STEAM TURBINES.** Manufacturers issue complete instructions for the care and operation of their respective units. The following notes direct attention to a few operating considerations:

The logical steps in starting a turbine are: First, the condenser circulating pump and the other condenser auxiliaries should be started; next, the auxiliary oil pump on the turbine, and a full oil supply furnished to all bearings. Cooling water should be turned on the oil cooler. Any new oil added to the system should be poured through several thicknesses of cheese-cloth to remove any chips, cuttings, etc.

The most difficult part of starting is the proper warming-up of spindle and casing. If a stationary turbine rotor is allowed to cool down from high temperature, cooling takes place unevenly, and the rotor may become bent. It cannot be operated in this condition without extreme vibration and rubs on sealing strips, thus increasing clearances and leakage areas. Careful warming-up is necessary to straighten the rotor. When the turbine has no turning gear, with steam-sealed glands, the vacuum pumps are started and steam turned into the glands. A small amount of steam allowed to pass the throttle valve will, on account of its density, rise to the top of the casing, causing it to heat and expand. The lower half remains filled with cold air and does not change in temperature or form. The result is a distorted cylinder. Disks also distort if one-half only is heated. It is evident that warming with a small amount of steam may cause undesirable distortions, which may result in blade rubs. A better method is suddenly to admit enough steam through the throttle valve to revolve the spindle, and to then close the throttle until only sufficient steam enters to keep the spindle turning over slowly. The rotating blades carry the steam around the casing, causing it to heat more evenly and rapidly than in any other way. This should continue until the turbine is evenly heated, before allowing the machine to speed up. All drains must be kept open during the warming process. A blade rub developed in starting probably is due to local distortion, and often may be relieved by allowing the turbine to stand for a short time while the heat in the casing diffuses through the whole body. Another careful start then may indicate that the blades are clear. It is unwise to bring a turbine up to speed without warming, as severe stresses undoubtedly are produced in certain parts.

**Starting up and loading** are important considerations on large turbines. The time required to bring turbines of 20,000 kw and larger from cold up to speed varies from 1/2 to 4 hr, and load may be added at the rate of 1000 to 3000 kw per min. If possible, large turbines ought not to be operated at less than 20% of rating unless designed for such light load operation.

When turbines are connected to generators in parallel with water-power plants, and must operate for considerable periods at no load, sufficient cooling steam must be admitted to cool the rotor by removing heat generated by windage.

As the turbine speeds up, the glands begin to seal and vacuum builds up. It is well to ascertain if the valves are sufficiently tight to prevent overspeeding with no load and full vacuum. The turbine next can be synchronized and load applied.

**Shutting Down.** The turbine is shut down by reversing the above processes in the regular order. The machine usually is stopped by tripping the valve, to see that it is free and acting properly, or by speeding up the turbine by holding the governor lever until the emergency governor acts, at 10% above normal speed.

When a unit with turning gear is shut down, it should be kept rolling by the turning gear until all parts are at room temperature to prevent bowing of the spindle. This may take 72 hr on large units. No steam should be allowed to leak into any turbine during a shutdown.

**Inspection.** Every turbine should be opened and inspected periodically, usually once a year, to note any wear of parts or other troubles. Corrosion of blades in rows at the dew point generally is due to impure feedwater or to poor deaeration. Wet steam may cause erosion of the last rows of blades. Oil reservoirs require cleaning. Water glands scale up, unless condensate only is used. Clearances require checking. Wear on thrust blocks should be noted and corrected. Leakage in stop and control valve seats should be stopped. Care must be used in reassembling to avoid damage to the blading.

The highest vacuum should be maintained at all times in the condenser exhaust. Frequent use of heavy asphaltum paint on all exhaust joints is a good preventive against air leaks.

**REMOVAL OF TURBINE DEPOSITS.** Continued carry-over from the boiler results in accumulated deposits on strainer, valves, steam passages, and blading. Increased
deposition is indicated by increased stage pressures at a given load, or by reduction in maximum capacity. These deposits consist of soluble and insoluble material. The cause of carry-over should be investigated and the difficulty overcome at the boiler.

Soluble deposits may be removed by washing the affected parts with water when shut down, or by use of saturated steam when operating at light load or low speed. In the latter case, superheat is slowly reduced by water injection in the steam header. Reduction of steam temperature at the boiler also helps. Wet steam sometimes dissolves the deposits when steam washing is done at no load and at low rpm. Removal may be checked by the conductivity of the condensate. After removal, superheat can be slowly restored.

Insoluble deposits, often principally silicas, are difficult to remove. Sometimes the turbine is opened and the deposit removed by an air blast, with powdered coal ash used as an abrasive. Or the turbine can be washed with caustic soda solution while the rotor is turned at a slow speed. All traces of soda must be removed when washing is completed. Turbine builders issue instructions for use of these processes on their turbines. (See Refs. 62 and 63.)

ACCIDENTS to steam turbines generally are due to one of these causes: failure of the oil supply; overspeeding due to failure of the overspeed governor to act; failure of buckets due to fatigue of material from vibration; fouling of blades with foreign parts in the casing; failure of the disks or drum from internal defects; or starting quickly an unevenly heated and distorted turbine. Many of these conditions can be foreseen and prevented by proper vigilance on the part of the operating engineer.

TURBINE OPERATING DATA for a period of years will be found in Turbines, NELA and EEI. The following terms have been adopted as standards in stating turbine performance (see also Art. 32, Section 16):

- **Period Hours.** Total hours per year (8760 in a normal year).
- **Service Demand Factor.** The ratio of demand hours to period hours.
- **Service Demand Availability Factor.** The ratio of service hours to demand hours.
- **Unit Capacity Factor.** The ratio of kilowatt-hours generated to the product of the unit rating and period hours.
- **Unit Output Factor.** The ratio of kilowatt-hours generated to the product of the unit rating and service hours.
- **Unit Operation Factor.** The ratio of service hours to period hours.
- **Maximum Possible Unit Operation Factor.** The ratio of the sum of the service hours and the reserve hours to the period hours.

Turbine design and reliability have been improved until the service demand availability factor has reached values of 95%.

Availability and reliability are important factors in turbine construction and operation, since outages not only cost money but also may lead to shutdowns of the entire electrical system. Gains in economy sometimes have been sacrificed to obtain more rugged and reliable operating units.

SYNCHRONOUS CONDENSER OPERATION of steam turbogenerator sets is often practiced. Generally the manufacturer refuses to accept responsibility for such an operating condition, although few failures have been assigned to this cause. If the turbine is coupled to the generator during such operation, particular care must be taken to ventilate the unit with steam to remove heat generated by windage. This is done by admitting steam through an oriﬁce into the ﬁrst stage. This steam, while supplying some of the energy to overcome the mechanical losses of the set, is less than the normal no-load steam flow. The heat generated and the location of the hottest stage depend on the size, type, number of stages, and density. The condenser usually is kept in service to maintain a high vacuum. Thermometers are placed on the turbine casing to indicate the rise in temperature when operated as a synchronous condenser. Limiting temperatures are about 500°F for a steel casing and 400°F for a cast-iron casing for small units. In large turbines, manufacturers prescribe even lower limits.

When the turbine is uncoupled and the generator operated alone as a synchronous condenser, it is started as a motor by being tied in electrically with another operating unit.

**12. CORRECTION FACTORS FOR TURBINE DATA**

An increase in steam pressure increases the available energy, which tends to reduce steam consumption at a given load and leads to lower leaving losses. If initial temperature is constant, moisture increases in the low-pressure stages, tending to decrease efficiency. \( p \), the ratio of wheel to steam speeds, decreases on the average, but \( R \), the reheat factor, increases. The net result is a slight decrease in turbine efficiency, together with lower steam consumption, and better heat rate.
An increase in vacuum increases the available energy and decreases steam consumption and heat rate. \( \rho \) decreases, tending to lower efficiency. The leaving loss increases. The net result is lowered efficiency, but also decreased steam consumption and heat rate.

An increase in initial steam temperature increases efficiency. It increases the available energy and decreases \( \rho \), which tends to decrease efficiency. This often is partly offset by the decreased moisture in the low-pressure section, although leaving losses tend to increase. The net effect is an increase in efficiency, accompanied by a decrease in steam consumption and heat rate.

**Correction Factors.** As it is impossible to reproduce on a plant test exactly the standard conditions specified in the contract, every steam turbine guarantee should contain the contract corrections for such variations from standard conditions as may occur on test. The corrections will vary with the various types and sizes of turbine, and with certain assumptions in their design. They should cover variations in initial pressure, vacuum, and load.

Only the manufacturer can state, with any degree of assurance, reasonable correction factors for his particular design. The purchaser, however, can check these by noting whether there is any appreciable change in the engine efficiency when the corrections are applied. (See Fig. 60 for correction data on a 60,000-kw turbine.) Correction factors furnished by builders are generally based on tests of units of the size and type in question. For extraction turbines, the exhaust pressure correction is expressed most conveniently as gain or loss in output, expressed as a function of the steam flow to the condenser.

**Rules for Steam Turbine Tests** are given in the ASME power test code on steam turbines (see Section 19). This code covers instruments, methods of measurement, and computation of results, and in an appendix gives examples of the various computations necessary to obtain the actual performance.

### 13. TURBINE PERFORMANCE

Steam-turbine performance usually is stated in tables giving size, speed, initial steam and exhaust conditions, pounds of steam per kilowatt-hour, and net Btu per kilowatt-hour. These tables, while of interest to engineers for reference, have three limitations. (1) It is impossible in a small space to quote tests for every possible steam condition under which turbines of many sizes may operate. (2) Builders seldom allow any tests to be published except the best records made by their equipment, and such tests obviously do not represent average conditions. (3) Turbine builders are prepared to furnish different types of the same size of unit, and frequently of different efficiencies, built to meet quite different conditions. For instance, a simple, low-cost unit, with a low ratio of wheel speed to steam speed and high leaving losses, may be best suited for certain commercial conditions. For other conditions, a more expensive turbine, of more refined construction, with a high ratio of wheel speed to steam speed, and with low leaving loss, may be most desirable. The latter will have a much lower heat consumption and higher cost than the first unit.

The following data will serve as a guide in estimating turbine performance or in checking tests.

**Engine Efficiency.** Operating conditions, particularly on small turbines, vary over such a wide range that tables covering every condition are beyond space limitations. The engine efficiency is used in the following paragraphs as a measure of performance of the smaller units.

The steam consumption of these units can be readily determined by dividing 3413 (the heat equivalent of 1 kw-hr) by the product of the engine efficiency, \( \eta_e \), at the generator terminals and the available energy from initial steam conditions to exhaust pressure. Thus pounds per kilowatt-hour = \( \frac{3413}{[\eta_e(h_1 - h_2)]} \).

The available energy \( (h_1 - h_2) \) can be found from a Mollier diagram or from the Theoretical Steam Rate Tables (see Section 4). The steam consumption in terms of brake-horsepower can be calculated by substituting 2544 for the horsepower and \( \eta_b \) for the engine efficiency at the coupling. Thus pounds per brake-horsepower-hour = \( \frac{2544}{[\eta_b(h_1 - h_2)]} \).

Leaving loss also may be considered at the same time as engine efficiency, as it is a measure of the unutilized velocity leaving the last row of blades.

Since turbines operate best under the conditions for which they are designed, the efficiency ratio under any other set of operating conditions will vary from the ratio at the specified conditions. Because of the leaving loss at the last row of blades of high-vacuum turbines, it usually happens that a higher efficiency ratio may be obtained at a lower vacuum than specified, even though the steam consumption per kilowatt-hour may be higher.
HEAT CONSUMPTION of a turbine is expressed in Btu per kilowatt-hour. British practice expresses turbine performance in terms of thermal efficiency. This is the ratio of 3413 (the heat equivalent of 1 kwhr) to the heat consumption of the unit, expressed in Btu per kilowatt-hour.

LAMBDA OF A TURBINE. Another British practice is to state the lambda (λ) value for a unit where

\[
\lambda = \left( \frac{\Sigma d^2}{100} \right) \times \left( \frac{\text{rpm}}{100} \right)^2
\]

where \( \Sigma d^2 \) is the sum of the squares of the mean diameters, inches, of all rows. The higher the value of \( \lambda \), the greater the engine efficiency as a rule. Another similar factor, called by Kraft the Parsons coefficient or quality factor \( q = \Sigma u^2/h_0 \), where \( \Sigma u^2 \) = the sum of the squares of the various wheel speeds, feet per second, and \( h_0 \) = isentropic available energy, Btu per lb, from initial conditions to final pressure. \( \Sigma u^2 \) gives an idea of the bulk of a turbine and consequently of the price of the unit. Kraft gives the following values of engine efficiency at the coupling for various values of \( q \):

\[
\begin{align*}
q & = 5000 \quad 7500 \quad 10,000 \quad 12,500 \quad 15,000 \quad 17,500 \\
\eta_e & = 69.8\% \quad 77.2\% \quad 81.5\% \quad 84.2\% \quad 85.5\% \quad 86\%
\end{align*}
\]

He points out that an increase in \( q \) means an increased number of stages, and that a large increase in \( q \) gives only a comparatively small increase in efficiency in the higher ranges. Also, the quality factor \( q \) must be higher for reaction turbines than for impulse units for the same efficiency.

PERFORMANCE OF MECHANICAL-DRIVE UNITS.*

In setting up a preliminary heat balance it is necessary that the designer have some source of information with respect to the approximate efficiency of auxiliary drives. Where mechanical-drive turbines are used for driving auxiliaries they may be either

"direct-connected" or geared. In the larger sizes of turbine used for driving moderately slow-speed auxiliaries considerable advantage frequently can be gained by using geared sets rather than direct drive. Mechanical-drive turbines may be subdivided further into single-stage and multistage units. Although it is impossible to generalize with respect to the suitability of either type without consideration of the specific application, a broad general dividing line will fall in the neighborhood of 100 hp. Below this rating, single-stage turbines are commonly used; above this rating the multistage turbine is more popular. Occasionally, where space is at a premium and simplicity essential, the single-stage unit will be used for the larger ratings.

In accord with this general philosophy of mechanical drive of auxiliaries by steam turbines, Fig. 28 presents approximate turbine coupling efficiencies for estimating purposes. The efficiencies in Fig. 28 indicate the approximate level of currently quoted efficiencies on auxiliary-drive turbines. In using this curve, it must be realized that considerable latitude

* Contributed by the Editor and used, by permission, from Chapter I, "Heat Engineering and Thermodynamics," by J. Kenneth Salisbury, of Volume II, Marine Engineering, edited by H. L. Seward; published by the Society of Naval Architects and Marine Engineers, 1944.
is available to the purchaser in the direction of either higher costs and better efficiencies or lower costs and poorer efficiencies. If any departure from the ordinary line of equipment is specified, it usually will be necessary that special designs be made, which may increase the cost.

In Fig. 28 turbine efficiencies based on the output at the turbine coupling have been plotted against rating in horsepower for various rotational speeds. All these efficiencies are based on steam conditions of 400 psig, 0°F superheat, and 15 psig exhaust pressure. For other steam conditions correction factors are provided to take care of inherent variation in efficiency with steam conditions. These corrections are shown in Figs. 29 and 30. Figure 29 presents the correction for initial pressures other than 400 psig. This correction is primarily a function of the rating of the unit; hence the parameter shown in the correction curves is horsepower rating. In Fig. 29 is shown an additional curve from which the superheat correction may be obtained. Large turbines are insensitive to moderate changes in available energy because they operate at or near their peak-efficiency speeds; superheat correction is intended to take care of the changes in efficiency resulting from change in moisture and supersaturation loss in the exhaust end of the turbine. Ordinarily mechanical-drive turbines, because of their somewhat poorer efficiency and higher exhaust pressures, have less moisture loss. They are sometimes "underspeeded," especially in the smaller sizes, for economic reasons; hence the superheat
correction in this case takes care of the effect on efficiency of changes in velocity ratio that result from change in the initial superheat, at a given pressure.

Figure 30 provides correction factors resulting from changes in ratio of absolute exhaust pressure to absolute initial pressure. To cover this correction adequately it is necessary that two correction factors be used, one of them depending on the rating of the unit and the other on the design speed. These corrections, valid for exhaust pressures up to 50 psig, should not be used for units having an exhaust pressure higher than this value.

To obtain the coupling efficiency of an auxiliary-drive turbine the basic efficiency is read from Fig. 28 at the proper rating and design rotational speed. This basic efficiency is then multiplied by an initial pressure-correction factor taken from Fig. 29. The resulting efficiency is then corrected for turbine pressure ratio, using the data given in Fig. 30. The product represents the full-load efficiency of the turbine based on coupling output and referred to the Rankine cycle.

EFFICIENCY OF HIGH-SPEED AUXILIARY TURBINE-GENERATOR SETS.*

The trend in auxiliary turbine-generator sets is toward efficient high-speed geared turbines. These turbines generally operate at about 10,000 rpm and are connected through gearing to low-speed (about 1200 rpm) generators. When built in this pattern the auxiliary turbine-generator sets have a level of efficiencies which is approximately 10 to 15% higher than the slower-speed sets. In some cases the improved efficiency will not be warranted, and the slower speed set may be chosen. On the other hand, since the high-speed geared set is more modern in design and shows promise of being the type of equipment that will be most widely used in the future, the curves of auxiliary turbine-generator set efficiencies shown in Fig. 31 are for this type of design.

In Fig. 31 are shown the efficiencies referred to the Rankine cycle of a-c and d-c turbine generator sets in a range of powers from 150 to 500 kw. These efficiencies, based on the output at the auxiliary-generator terminals, are predicated on initial steam conditions of 400 psig, 200 F superheat, and 2 in. Hg abs exhaust pressure. While the basic steam conditions for which the curves are drawn are representative, correction factors are provided in Fig. 32 by which these efficiencies may be adjusted for the inherent variation in efficiency which results from changed steam conditions. In Fig. 32 three correction factor curves are shown. These factors correct for initial pressure, initial superheat, and exhaust pressure. Because the initial pressure-correction factor depends on the size of the unit under consideration, two curves are drawn, for 150 kw and 500 kw. For intermediate ratings the pressure-correction factor may be linearly interpolated. Superheat and exhaust pressure-correction factors are independent of the rating.

To determine the full-load efficiency of an auxiliary turbine-generator set the efficiency is read from Fig. 31 at the proper rating and then multiplied successively by the correction factors for initial pressure, superheat, and exhaust pressure.

* Contributed by the Editor and used, by permission, from Chapter I, "Heat Engineering and Thermodynamics," by J. Kenneth Salisbury, of Volume II, Marine Engineering, edited by H. L. Seward; published by the Society of Naval Architects and Marine Engineers, 1944.
EXAMPLE. Find the efficiency of an auxiliary a-c turbine-generator unit rated 250 kw, having steam conditions of 300 psig, 100 F superheat, and 3 in. Hg abs exhaust pressure.

Rated load efficiency (from Fig. 31) = 57.5%
Initial pressure-correction factor (from Fig. 32) = 1.02
Initial superheat-correction factor (from Fig. 32) = 0.982
Exhaust pressure-correction factor (from Fig. 32) = 1.010
Rated load efficiency = 57.5 (1.02) (0.982) (1.010) = 58.2%

EFFICIENCIES OF NONCONDENSING TURBINES, NEMA SIZES, at the generator terminals, are given in Table 6. Most efficient load is generally full rating.

Table 6. Engine Efficiencies of Noncondensing Turbines in Per Cent
(Various initial pressures; 200 F superheat; atmospheric exhaust pressure)

<table>
<thead>
<tr>
<th>Rating, kw</th>
<th>Initial Pressure, psig</th>
<th>Rating, kw</th>
<th>Initial Pressure, psig</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>200</td>
<td>400</td>
<td>600</td>
</tr>
<tr>
<td>500</td>
<td>59.6</td>
<td>55.1</td>
<td>50.9</td>
</tr>
<tr>
<td>700</td>
<td>62.9</td>
<td>58.5</td>
<td>54.4</td>
</tr>
<tr>
<td>1000</td>
<td>66.0</td>
<td>62.1</td>
<td>58.4</td>
</tr>
<tr>
<td>1500</td>
<td>69.3</td>
<td>65.8</td>
<td>62.2</td>
</tr>
<tr>
<td>2000</td>
<td>71.4</td>
<td>68.2</td>
<td>65.1</td>
</tr>
<tr>
<td>2500</td>
<td>72.9</td>
<td>70.1</td>
<td>67.1</td>
</tr>
</tbody>
</table>

Note. In Tables 6, 8, and 11 the 500- to 1500-kw values are for high-speed turbines geared to 1200 rpm a-c generators. Turbine speeds are chosen to suit the steam conditions. Data for units 2000 kw and above are for 3000-rpm direct-connected turbine alternators.

Correction factors to noncondensing engine efficiencies for partial loads vary with initial conditions. Table 7 gives typical correction factors to engine efficiencies of the noncondensing units of Table 6 for one set of initial conditions.

Table 7. Correction to Engine Efficiencies of Noncondensing Turbines for Partial Load Operation
(400 psig, 200 F, superheat; atmospheric exhaust pressure)

<table>
<thead>
<tr>
<th>Rating, kw</th>
<th>Load, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>50</td>
</tr>
<tr>
<td>500</td>
<td>.698</td>
</tr>
<tr>
<td>700</td>
<td>.710</td>
</tr>
<tr>
<td>1000</td>
<td>.718</td>
</tr>
<tr>
<td>1500</td>
<td>.726</td>
</tr>
<tr>
<td>2000</td>
<td>.730</td>
</tr>
<tr>
<td>2500</td>
<td>.733</td>
</tr>
<tr>
<td>3000</td>
<td>.735</td>
</tr>
<tr>
<td>3500</td>
<td>.736</td>
</tr>
<tr>
<td>4000</td>
<td>.738</td>
</tr>
<tr>
<td>5000</td>
<td>.739</td>
</tr>
<tr>
<td>6000</td>
<td>.740</td>
</tr>
<tr>
<td>7500</td>
<td>.741</td>
</tr>
</tbody>
</table>

EFFICIENCIES OF CONDENSING TURBINES, NEMA SIZES, at the generator terminals, at rated load are given in Table 8. (See note above regarding unit speeds.)

Table 8. Engine Efficiencies of Condensing Turbines in Per Cent
(Various initial pressures; 200 F superheat; 2 in. Hg abs exhaust)

<table>
<thead>
<tr>
<th>Rating, kw</th>
<th>Initial Pressure, psig</th>
<th>Rating, kw</th>
<th>Initial Pressure, psig</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>400</td>
<td>600</td>
<td>200</td>
</tr>
<tr>
<td>500</td>
<td>61.2</td>
<td>58.5</td>
<td>56.3</td>
</tr>
<tr>
<td>700</td>
<td>63.4</td>
<td>61.7</td>
<td>59.1</td>
</tr>
<tr>
<td>1000</td>
<td>66.0</td>
<td>63.8</td>
<td>61.8</td>
</tr>
<tr>
<td>1500</td>
<td>68.6</td>
<td>66.3</td>
<td>64.7</td>
</tr>
<tr>
<td>2000</td>
<td>70.4</td>
<td>68.4</td>
<td>66.7</td>
</tr>
<tr>
<td>2500</td>
<td>71.6</td>
<td>69.8</td>
<td>68.1</td>
</tr>
</tbody>
</table>
Correction factors to condensing efficiencies for partial loads vary with initial conditions. Table 9 gives correction factors for the units of Table 8 when operating at the specified initial conditions.

**Table 9. Correction to Engine Efficiencies of Condensing Turbines for Partial Load Operation**

(400 psig; 200 F superheat, 2 in. Hg abs)

<table>
<thead>
<tr>
<th>Rating, kw</th>
<th>Load, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>25</td>
</tr>
<tr>
<td>500</td>
<td>.754</td>
</tr>
<tr>
<td>700</td>
<td>.769</td>
</tr>
<tr>
<td>1000</td>
<td>.783</td>
</tr>
<tr>
<td>1500</td>
<td>.796</td>
</tr>
<tr>
<td>2000</td>
<td>.803</td>
</tr>
<tr>
<td>2500</td>
<td>.809</td>
</tr>
<tr>
<td>3000</td>
<td>.812</td>
</tr>
<tr>
<td>3500</td>
<td>.814</td>
</tr>
<tr>
<td>4000</td>
<td>.816</td>
</tr>
<tr>
<td>5000</td>
<td>.819</td>
</tr>
<tr>
<td>6000</td>
<td>.821</td>
</tr>
<tr>
<td>7500</td>
<td>.822</td>
</tr>
</tbody>
</table>

Superheat corrections for smaller turbines are given for the noncondensing units (Table 6) and the condensing units (Table 8) in Table 10.

**Table 10. Superheat Correction to Engine Efficiencies**

<table>
<thead>
<tr>
<th>Superheat, °F</th>
<th>Noncondensing</th>
<th>Condensing</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>0.988</td>
<td>0.980</td>
</tr>
<tr>
<td>125</td>
<td>0.992</td>
<td>0.985</td>
</tr>
<tr>
<td>150</td>
<td>0.996</td>
<td>0.990</td>
</tr>
<tr>
<td>175</td>
<td>0.998</td>
<td>0.995</td>
</tr>
<tr>
<td>200</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>225</td>
<td>1.001</td>
<td>1.004</td>
</tr>
<tr>
<td>250</td>
<td>1.002</td>
<td>1.009</td>
</tr>
<tr>
<td>275</td>
<td>1.003</td>
<td>1.013</td>
</tr>
<tr>
<td>300</td>
<td>1.003</td>
<td>1.017</td>
</tr>
</tbody>
</table>

**GEARED TURBINES** are sometimes furnished for industrial use. The gear losses lower the efficiencies of these sets. Expected engine efficiencies of certain larger multistage geared condensing turbines operating at 400 psig–200 F superheat–2 in. Hg abs back pressure, 3600 rpm, and above are shown in Table 11.

**Table 11. Engine Efficiencies of Geared Turbines**

<table>
<thead>
<tr>
<th>Brake Horsepower</th>
<th>Engine Efficiency, Including Gears, %</th>
<th>Brake Horsepower</th>
<th>Engine Efficiency, Including Gears, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>65.6</td>
<td>10,000</td>
<td>73.4</td>
</tr>
<tr>
<td>3000</td>
<td>68.6</td>
<td>10,000</td>
<td>74.3</td>
</tr>
<tr>
<td>5000</td>
<td>71.5</td>
<td>10,000</td>
<td>74.3</td>
</tr>
</tbody>
</table>

**AIEE-ASME PREFERRED STANDARD TURBINES.** The performance data in Table 1, p. 8-14, applies to large condensing turbine generator units (3600 rpm, 3 phase, 60 cycle) built according to the AIEE-ASME preferred standard ratings.

**Maximum Capacity.** AIEE-ASME preferred standard turbines have a guaranteed capability of 110% of rated kilowatts when operated with the initial pressures and temperatures shown in Table 1 (p. 8-14) and an exhaust pressure of 2 1/2 in. Hg abs. The steam-flow capacities stated in Table 1 correspond to guaranteed kilowatt capabilities of the turbines when operated with the standard number of feedwater heaters.

**Effect of Vacuum.** Capability of turbines when operated with rated initial steam pressure and temperature and with extraction for feedwater heating in the standard number of heaters, but with exhaust pressure other than 2.5 in. Hg abs, will change by the percentage given in Table 12.
Table 12. Effect of Vacuum on Capability

<table>
<thead>
<tr>
<th>Initial Steam Pressure, psig</th>
<th>Initial Steam Temperature, °F</th>
<th>Per Cent Change in Kilowatt Capability</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Exhaust Pressure, in. Hg abs</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.0</td>
</tr>
<tr>
<td>600</td>
<td>825</td>
<td>+3.9</td>
</tr>
<tr>
<td>850</td>
<td>900</td>
<td>+3.3</td>
</tr>
<tr>
<td>1250</td>
<td>950</td>
<td>+2.6</td>
</tr>
</tbody>
</table>

Effect of Initial Pressure. The kilowatt capability of turbines when operated at initial temperature, and with extraction for feedwater heating in the standard number of heaters, will be reduced by a 50-psig drop in initial pressure, as shown:

<table>
<thead>
<tr>
<th>Initial Steam Pressure, psig</th>
<th>Initial Steam Temperature, °F</th>
<th>Reduction in Kilowatt Capability, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>600</td>
<td>825</td>
<td>9</td>
</tr>
<tr>
<td>850</td>
<td>900</td>
<td>6</td>
</tr>
<tr>
<td>1250</td>
<td>950</td>
<td>4</td>
</tr>
</tbody>
</table>

Effect of Initial Temperature. The kilowatt capability will be reduced 2% for 50°F reduction in initial steam temperature at rated initial pressure, and with full extraction for feedwater heating.

Effect of Bypassing Top Heater. When operated with rated initial steam conditions and an exhaust pressure of 2.5 in. Hg abs or lower, and with extraction for feedwater heating but with no steam extracted from the highest pressure extraction outlet provided in the standard turbine, the kilowatt capability of the turbine will be increased by 4%.

Extraction Openings. Table 1, p. 8-14, shows saturation temperatures at the extraction openings, with all openings in service.

14. PERFORMANCE CALCULATIONS

Steam-turbine calculations are made for the purposes of design, of estimating turbine performance with extraction heaters or bleeder connections, and of checking test results. Methods of making calculations are varied, since the procedure is not standardized. Many assumptions are made to simplify the work. The following methods yield results satisfactory for estimating and checking purposes. For an extensive discussion, see Ref. 64.

CHARACTERISTIC CURVES. Three characteristic curves are needed to analyze turbine performance: the Willans line, the shell pressure curves, and the state-line, or condition curve. All three are discussed in this article. The first of them, the Willans line, indicates the relation between total steam per hour and load. This line may be assumed to be straight between no load and most efficient load. Above most efficient load, the form of the line depends on the overload valve arrangement. The slope of the line may differ from that below most efficient load. With nozzle governing, the true Willans line consists of a series of scallops, concave down, each having a height of only about 0.5 to 3% of the steam flow. For all practical purposes a straight line up to most efficient load may be substituted.

When guarantees of steam consumption are not available, the Willans line may be readily estimated by methods given herein. The available energy from initial conditions to final pressure can be found using the Mollier chart, or the theoretical steam rate may be found (see Section 4). Warren and Knowlton (Ref. 65) present efficiency data representative of present practice in large steam turbines.

Figure 33 from Ref. 65 presents engine efficiencies for large turbines of various capacities, with 300°F initial superheat, at various initial steam pressures, and given generator efficiencies, with assumed mechanical and exhaust losses. The engine efficiency at other superheats can be found by dividing the engine efficiency found from Fig. 33 by the values given in Table 13.

Table 13. Superheat Corrections to Engine Efficiency

<table>
<thead>
<tr>
<th>Superheat at throttle, °F</th>
<th>0</th>
<th>100</th>
<th>200</th>
<th>300</th>
<th>400</th>
<th>500</th>
<th>600</th>
</tr>
</thead>
<tbody>
<tr>
<td>Correction factor</td>
<td>.060</td>
<td>1.044</td>
<td>1.018</td>
<td>1.000</td>
<td>.987</td>
<td>.978</td>
<td>.970</td>
</tr>
</tbody>
</table>
Having an engine efficiency from Fig. 33 and the theoretical steam rate, the actual steam rate may be found:

$$\text{Actual steam rate} = \frac{\text{Theoretical steam rate}}{\text{Engine efficiency}}$$

Steam flow at rating is found by multiplying steam rate by the rated kilowatts. This flow is plotted as one point on the Willans line. For condensing units at normal exhaust pressures the no-load steam flow is approximately 0.5 lb per kw of rated capacity. This yields a second point. Connecting the two by a straight line gives a good first approximation of the Willans line. An approximate steam rate curve may be found by dividing various flows by corresponding loads. More accurate methods are given on p. 8-67.

**CAPABILITY AND RATED LOAD.** Some large units still are nominally rated at 80% power factor. Such units, however, are required to develop full kva at 100% power factor. The nominal rating thus is only 80% of the capability of the turbine-generator set.

When standardization of turbines was undertaken, the difference between nominal rating and capability was reduced. Capability is now 110% of nominal rating on standard units, a practice that may later be extended to all large units. Since capability is a true measure of the turbine's ability to deliver power, it is considered by some authorities as the true rating of the unit for power-generating purposes.

Capability of turbines rated less than 10,000 kw is generally 25% greater than the nominal rated capacity. The generator kva rating is at 80% power factor for nominal kilowatt capacity.

The maximum total steam required, in pounds per hour, is found by multiplying maximum output by the steam rate. The steam rate at maximum load may be larger than that at most efficient load by 0 to 4%.

**NO-LOAD TOTAL STEAM FLOW** with generator at full voltage depends on the design of turbine. Data collected from tests indicate that its value may be approximated

<table>
<thead>
<tr>
<th>Name plate rating, kw</th>
<th>1000</th>
<th>2000</th>
<th>3000</th>
<th>4000</th>
<th>5000</th>
<th>10,000</th>
<th>15,000</th>
<th>20,000</th>
<th>25,000</th>
<th>30,000</th>
<th>40,000</th>
<th>50,000 and over</th>
</tr>
</thead>
<tbody>
<tr>
<td>% of total steam at most efficient load</td>
<td>13.8</td>
<td>12.5</td>
<td>11.8</td>
<td>11.3</td>
<td>10.9</td>
<td>9.9</td>
<td>9.2</td>
<td>8.8</td>
<td>8.6</td>
<td>8.4</td>
<td>8.2</td>
<td>8.0</td>
</tr>
</tbody>
</table>