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The Combustion of Waste-Wood Products

By H. W. Beecher[1] and R. D. Watt,[2] Seattle, Wash.

This paper is not intended to be a rigorous argument or a scientific treatise advocating any particular procedure in producing steam from the combustion of wood waste, but rather the compilation of essential facts made apparent by many years of experience in dealing with this rather abstruse subject.

The waste products resulting from the manufacture of lumber, plywood, or cellulose for conversion into pulp are available as fuel. Manifestly, any portion of the log which can be economically converted into more valuable material should neither be classified as waste wood nor used for fuel. Sawdust and shavings can be handled for burning without further processing. Slabs, edgings, trimmings, and other waste products require further size reduction to prepare them for rapid combustion, easy transportation, and convenient handling. Such material is usually processed by a mechanical masticator, commonly known as a "hog." The product so obtained, together with sawdust and shavings, forms a mixed fuel called "hog fuel." The term "hog fuel" will be used to describe the waste-wood products dealt with in this paper, whether they be sawdust, shavings, or a mixture of these with hogged materials.

Economic Aspects

The cheapest power for operation of a sawmill is energy generated from steam produced by burning the resulting waste materials. This waste must be removed or destroyed and is available to the producing sawmill without transportation costs. Whenever sawmills are not located within economical transportation distance from external hog-fuel markets, large investments in refuse burners are necessary in order to dispose of excess waste fuel. In a country with an abundance of cheap hydroelectric power, fuel must be inexpensive to permit steam-plant competition, except where steam is required for process.

The mill production cost for the hogged portion of fuel is from 10 cents to 15 cents per unit, largely made up of power, operation, and maintenance expenses of the hogging equipment. The mill operator should receive a reasonable return on his investment and, if possible, secure a profit on his waste material. Prices charged for hog fuel vary from 50 cents to $1.50 per unit at the producing mill, depending upon the supply and demand and not upon the cost of production. If a mill is isolated from the market in a community where the other fuel-consuming activities also produce hog fuel as a by-product, there is competition for the consuming market and the mill operator must then be satisfied with smaller returns. Conversely, if the producing mill is located in an industrial center containing fuel-consuming plants which are not producers of fuel, an absorbing market is available, permitting the mill operator to secure a profit on his waste materials. In recent years the revenues from hog-fuel sales have represented a large part of the total profits of many mills.

The large volume and weight of hog fuel per available Btu makes the transportation cost loom large in the total cost to the consumer. The economical marketing zone is limited by transportation costs. Frequently, greater cost is involved in transportation and handling than the actual price paid by the consumer to the producer for the fuel at the point of manufacture. In spite of high transportation costs, hog fuel is generally available to the consumer at a cost per million Btu, comparing favorably with the costs of other types of fuel in the lowest-fuel-cost areas of the United States. Hog fuel is, therefore, the principal fuel used in the Pacific Northwest for steam production.

The high moisture content of hog fuel materially reduces the obtainable thermal efficiencies of boiler plants, as compared with the efficiencies secured with other fuels. This necessitates comparison of hog fuel with other fuels on the basis of their relative cost per available Btu. Many consumers of hog fuel pay as little as 50 cents a unit, delivered. In other Northwest plants the hog-fuel cost reaches $3.50 per unit. A unit of hog-fuel measurement occupies 200 cu ft. An average unit of hog fuel will contain approximately 20,000,000 Btu. Boiler-plant efficiencies with hog fuel vary, depending upon the type of installation, the percentage of rating at which the boiler plant operates, and whether air heaters are installed for recovery of additional heat from the boiler gases. These efficiencies vary from 45 per cent on the poorer installations to 65 per cent on the more modern and better equipped boiler plants.

To indicate the general low cost of hog fuel for the production of steam, it may be noted that with 60 per cent efficiency the available heat per average unit would be 12,000,000 Btu. At a cost of $1 per unit, the corresponding cost of steam production would be $8^1/_3$ cents per million Btu input. With fuel oil costing $1 per bbl and with 83 per cent boiler efficiency, the heat available in steam would be 5,200,000 Btu per bbl and the corresponding cost per million Btu input would be 19 cents. With coal having a heat value of 12,500 Btu per lb and costing $4 per ton and with an average boiler efficiency of 80 per cent, the corresponding cost of steam per million Btu input would be 20 cents. This comparison indicates that, in so far as a consideration of the combustion of hog fuel is concerned, a low-cost fuel is involved which has not yet encouraged the engineering research or the capital investment which would be warranted were it a higher-priced commodity.

Measurement of Hog Fuel

Neither buyers nor sellers of hog fuel have been willing to spend money to measure accurately fuel having such low cost per million available Btu. The seller has not had competition from fuels other than hog fuel from competing mills, as this fuel has been used principally in localities where neither oil nor coal has been competitive. The buyer, realizing the economic advantage in using hog fuel, as compared with other available fuels, has been unmindful of the advantages to be obtained by purchasing scientifically and determining the actual fuel values obtained for his dollar.

To compare and purchase on an available Btu basis one must purchase by weight, rather than by measurement, and make proper corrections for average moisture content. The simplicity and comparatively low cost of volumetric measurement has delayed the adoption of the more scientific system. The general adoption of volumetric measurement dates back 30 years when hog-fuel prices were yet lower than those prevailing today. The "unit" on which most hog fuel is purchased and sold contains 200 cu ft (material and voids) irrespective of the compacting of

[1] Consulting Engineer.

[2] Assistant to H. W. Beecher.

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the fuel brought about by the shape, size, or handling of the measuring containers. The process of mixing hog fuel and sawdust, in which the sawdust tends to fill the voids, provides greater fuel content per unit of measure. No allowance or credit is ordinarily given by the buyer for the greater number of Btu in a compacted unit. As a result, there is wide variation in the heating value per unit as sold. Another element of variation which is waived with the present measurement basis is moisture content with its serious effect on the availability of the heat units. With the present measurement basis, a buyer may obtain more heat per dollar from a unit for which he pays $1.50 than from another unit for which he pays only $1. The development of weighing-belt scales shortens the time until much of the hog fuel used in large plants will be weighed and, with proper moisture determination, be bought, sold, and compared on a Btu basis.

Availability of Waste-Wood Fuel

As long as lumber is cut from logs, there will be waste for which the primary market will be among the consumers of cheap fuel. A detailed survey by the Forest Service and reported by Allen H. Hodgson showed that, in the manufacturing of slightly over 10,000,000,000 fbm of green, rough-sawed lumber in the Douglas fir region in the year 1929, over 619,000,000 cu ft of solid-measure normal sawmill waste was produced. An analysis of the total volume of 1,354,000,000 cu ft of sound wood in the logs showed that approximately 67 per cent (911,000,000 cu ft) was converted into green, rough-sawed lumber; nearly 19 per cent (261,000,000 cu ft) became slabs, edgings, and trimmings; the balance, 14 per cent (191,000,000 cu ft) was sawdust. In addition to the sound wood there was approximately 167,000,000-cu ft solid measure of bark. This indicates that of the solid-wood material, inclusive of bark represented by the logs as delivered to the sawmill, 41 per cent is so-called "waste" and available for fuel.

We have been unable to find the result of any studies made to determine the percentage of solid wood in the tree represented by the solid wood in the saw logs. It is well known that the tops and branches cannot be economically handled, transported, and utilized for the production of lumber, as the available wood content, when converted into lumber, will not bring sufficient revenue to the mill to cover the cost of removing this material from the forest. The branches and tops left in the forest are a fire hazard and, if the cost of handling and removing them approached the price obtainable for the materials produced, they would be removed by the mills at a loss rather than be left in the forest as a menace. It is reasonable to assume that the material left in the forest is sufficient to make up the difference between 41 and 50 per cent, and to state broadly that, of the wood content of the average tree as logged and utilized in the Northwest, less than 50 per cent is converted into lumber and its allied products. The balance is an economic waste except for the value recovered as fuel.

While there may be some slight improvement in utilization, the greatest encouragement for the conservationist comes from the possibility of chemically treating selected portions of existing waste for production of cellulose products. Another field is fermentation and production of alcohol. A third utilization process is destructive distillation which would give charcoal, numerous by-products, and some steam for power production by burning the gas after cleansing of by-products. All these possibilities justify the statement that, as long as sawmills in the Northwest are operating at reasonable capacities, hog fuel in abundance will be available for industrial use.

There will remain a wide price range to the consumer at his place of use. As a consumer goes farther afield to secure his fuel in competition with other consumers, the more fortunately located sawmills with lower transportation costs will raise their prices and derive greater profits. Greater consuming markets will advance the prices so that small mills, beyond the zone of economical transportation at present fuel prices, will be provided with a market for materials now destroyed in refuse burners. Such destruction, while wasteful, is necessary without a market for this material.

Transportation of Hog Fuel

Where water transportation is available without rehandling, it is by far the cheapest method of moving hog fuel. Hog fuel is towed on barges as great a distance as 350 miles.

In certain localities, transportation must be by rail in specially fitted cars. Tariffs are either per car or per 100 lb weight. In either case it is desirable to have car contents a maximum. Cars are constructed with sides extending to maximum clearance height for the division in which they are to operate. A 50-ft car may be constructed to handle 35 units. However, most cars carry from 20 to 30 units.

The use of trucks for hog-fuel transportation is increasing. Special bodies carry from 4 to 6 units where regulations will permit. They load by gravity from bins, or conveyers, and are fitted with power dumps by which they are discharged into hoppers. Truck-transportation cost is dependent upon the length of haul which affects the portion of hourly truck and driver costs chargeable to each unit of hog fuel.

Each transportation problem must be independently treated to obtain the least possible cost. No set schedule of probable cost can be suggested as generally applying to any of the three methods outlined.

In rare cases, the hog-fuel-producing plant is situated near a consuming market and very low transportation costs can be obtained by use of belt conveyers between plants.

Except where necessary to elevate fuel steeply or to take off at several intermediate points, belt conveyers should be used. Even with multiple-point discharge, it frequently becomes desirable to use trough belts with traveling trippers or flat belts with unloading slices rather than to install scraper conveyers.

Belt conveyers are cheaper to operate than flight conveyers as they require less power, attention, and maintenance. Where belt conveyers cannot be conveniently installed, it is necessary to use special scraper or drag conveyers. All boiler-feed conveyers should be provided to convey more than current requirements and to return surplus to supply source, rather than to attempt regulation of the conveyed fuel supply in synchronism with fuel consumptions.

The amount of fuel storage required depends upon reliability of supply, possible interruption to transportation, and the necessity of avoiding plant outage. Plant consumptions, load factors, and operating intervals also affect the amount of storage which should be provided. The possibility of using oil or other more expensive fuel for emergency operation, with its attendant higher fuel cost, should be weighed against the fixed charges on average hog-fuel storage and recovery systems.

Heating Value

The high percentage of oxygen in wood reduces the heat content per pound as it is combined with carbon and hydrogen to form carbohydrates and, therefore, the total heat of combination of the combustibles is not all available. The manner in which these three elements are combined is not definitely known and the use of Dulong's formula, as applied to the ultimate analysis of wood, will not result in a Btu value corresponding to that obtained from calorimetric determinations. Hog fuel as normally delivered to the furnaces contains a high percentage of moisture. A portion of this is extraneous moisture, either resulting from wet logs, water lubrication of saws, or rain when fuel has been exposed. Most of the moisture, however, is in the cellular structure of the wood. The fuel, as received from the average sawmill, contains material with high moisture content from the sap wood of the log, material with a medium moisture content from the heartwood, and material with comparatively low moisture content from wood dried in commercial dry kilns.

In computing combustion results, moisture determinations are reported as the percentage of the total weight of wood and moisture represented by the moisture. This means that fuel containing 50 per cent moisture contains 1 lb of water per lb of dry fuel. With the high oxygen content of wood there would be $1^3/_4$ lb of water per lb of combustible. If the oxygen were combined with the hydrogen, as assumed in Dulong's formula, 50 per cent moisture in the fuel would correspond to approximately 2 lb of water per lb of ununited or available combustible. It is interesting to note, when the moisture content is increased from 50 to 60 per cent, the weight of moisture in the fuel is increased from 1 to $1^1/_2$ lb per lb of dry fuel. The hog fuel used in industrial plants of the Northwest will average from 25 per cent moisture, when principally kiln-dried material, to from 57 to 60 per cent moisture when largely green hemlock.

The available heat in hog fuel is a function of the moisture content. The heat necessary to raise the temperature of the wet fuel to 212 F, evaporate the water, and superheat the vapor to the exit-gas temperature is not available for steam production. This accounts for a substantial portion of the total losses in hog-fuel combustion.

All species of wood considered herein have approximately the same heating value on a bone-dry basis which will average 8900 Btu per lb of dry wood. However, some species are better fuels than others. Hemlock is not as good as fir. Spruce is better then hemlock, but poorer than fir. Cedar is a light fuel and requires a specially designed furnace for good results. Hemlock, as ordinarily available as fuel, has a high moisture content and does not readily part with its moisture. At least 20 to 25 per cent more capacity can be obtained from given furnaces and combustion chambers with fir fuel having about 45 per cent moisture content than with hemlock having 57 per cent moisture.

The heating value of stored hog fuel varies with the time in storage. Storage of hog fuel in the open decreases available Btu in fuel faster than storage under cover. This loss of heat value is attributed to slow oxidation which takes place at low temperatures. Cultures have been made from samples of hog fuel after storage over a considerable period which show an indication of molds and other wood-destroying fungi. These reactions are exothermic and the heat is lost.

Combustion

Coal and oil in age-long processes have both been formed from vegetable matter. All wood, coal, and oil contain the same elemental combustible materials; these elements are, however, combined in different ratios. A typical ultimate analysis of wood is as follows:

	Per cent
Carbon	50.31
Hydrogen	6.20
Oxygen	43.08
Nitrogen	0.04
Ash	0.37

Through the ages, during which coal and oil have been subjected to heat and pressure, much of the oxygen and moisture originally contained in the wood or other cellulose matter from which they were formed have been driven off, leaving a greater concentration of combustibles.

Nearly 45 per cent of the dry weight of wood, independent of the species, is oxygen. The hydrogen-to-carbon ratio in wood is of the same order as in oil and, therefore, for the same excess air, the percentage of water vapor as compared to dry gases will be approximately the same for these two fuels. Coals, as a rule, have much lower hydrogen-to-carbon ratios and, therefore, give combustion gases containing lower percentages of moisture. The heating value of the fixed carbon in wood fuel amounts to from 15 to 20 per cent of the total heat in the fuel. The high moisture and volatile contents of hog fuel delay combustion which proceeds as follows:

1 The driving off of the moisture content and raising the wood to a temperature at which volatiles will be driven off;

2 The actual distillation of volatiles;

3 The combustion of the fixed carbon.

The high oxygen content of wood with its low nitrogen content reduces the percentage of nitrogen in hog-fuel flue gas. Coal of typical analysis, if completely burned without excess air, would produce $18^1/_2$ per cent CO_2 in the combustion gases; similarly, oil of typical analysis, if burned without excess air, would produce $15^1/_2$ per cent CO_2; wood of the typical analysis quoted will give, if completely burned without excess air, approximately 20 per cent CO_2.

The wood itself contains but little noncombustible in the form of ash; however, hog fuel as normally fired may carry with it appreciable quantities of ash-forming material in the nature of extraneous matter embedded in the bark or wood fibers and not removed in preparation, transportation, and handling. This may consist of small pebbles, sand, and shells. Logs which have been transported in salt water give off gases containing salt fume which assists in lowering the fusion temperature of the noncombustible and accelerates the deposit of slag on boiler tubes.

With deep fuel beds, most of the fixed carbon, undoubtedly, leaves the fuel bed as carbon monoxide where it unites with additional oxygen to burn to the dioxide. The incandescent carbon adjacent to the grates burns to the dioxide and then, in passing further through the incandescent carbon, is reduced to the monoxide.

In the cellular type of furnace, it is important to provide secondary or overdraft air. The conical pile of fuel is too thick, except around its edges, to pass the necessary air for rapid combustion. The closing of secondary-air admission ports, resulting from too thick a fuel bed, is quickly evident in the smoking of furnaces. The admission of overdraft air through the grates in the front of the furnace with little resistance to the passage of such air decreases the negative pressure in the furnace and lowers the required average draft throughout the setting. Any decrease in required draft is desirable to avoid infiltration in the convection sections where excess air decreases the efficiency of the boiler.

The standard method of feeding fuel to flat-grate cellular-type furnaces is through feedhole openings located in the furnace roof, the fuel being transported to the furnace through chutes. Reasonable precautions are necessary to limit the amount of air entering the furnace through these chutes; any air so admitted decreases the air to the preheater and also results in furnace stratification. In spite of such precautions, the falling fuel produces an injector action and entrains considerable quantities of air.

Design of Furnaces to Burn Hog Fuel

The problems of proper combustion of hog fuel are greatly increased by the necessity of providing furnaces suitably designed for fuels varying in size from dust to pieces having 3 to 5 cu in. of content and for fuels of variable moisture content. Frequently, slugs of dry and highly combustible fuel are followed by slugs of wet fuel which form a damp blanket on the fuel pile. In the case of hopper-fired sloping-grate furnaces, one side of a hopper may contain dry fuel and the other side wet fuel.

The designer must provide furnaces to handle properly the wet fuel and, at the same time, not to punish unduly refractories during the periods in which only dry fuel is fed. To produce the best average combustion conditions, much study has been given to the use of sloping-grate furnaces where the fuel is admitted in a comparatively thin and uniform layer over a drying hearth, in which portion of the furnace reflected heat is utilized to drive off the moisture and start the distillation process necessary before the fixed-carbon content of the fuel can be ignited. Following this section of the furnace, the fuel flows over grates and, as the volatile content is driven off, combustion of the fixed carbon is maintained by the air passing through the grates and fuel bed.

Theoretically, such furnaces would be preferred to flat-grate, conical-pile furnaces with which by far the greatest number of hog-fuel-fired boilers are equipped. Practically, difficulties are encountered with sloping-grate furnaces caused by:

1 The fuel not being uniform in size and, therefore, containing streaks, or pockets, of greater density than adjacent areas, leading to the formation of blowholes through the fuel.

2 The fuel, not being of uniform moisture content, leads to the formation of areas in which distillation and ignition proceed more rapidly than in adjacent areas, thus resulting in the same formation of blowholes.

With a fuel as light as wood, particularly after the moisture and volatiles have been driven off, leaving charcoal cinders, these blowholes lift the cinders from the grates, depositing them at the foot of the sloping section and, in their formation, prevent the fuel from above the blowhole cascading to cover the hole. If too great ashpit pressures are used, this formation of blowholes is accentuated.

The accumulated charcoal cinders at the toe of the sloping grates affords such high resistance to the passage of air that insufficient air passes through this material. This prevents the combustion at the toe of the grate proceeding with sufficient rapidity to obtain high ratings per square foot of grate area, as the limiting rate for inflow of fuel over the drying-hearth section is the rate at which the fixed carbon can be consumed at the lower end of the grate. Even though sloping-grate furnaces have been tried in the Northwest with 15 to 16 ft of total length, obtainable capacities per foot of width of furnace have been less than those possible with well-designed furnaces of the so-called cellular type. As a result of the greater capacity obtainable in the latter furnace, most of the installations made in recent years have been of this design.

It is possible that extremely long sloping furnaces with special means for controlling the rate of feed and for cleaning the accumulated slag at the toe of the grate, with controlled and zoned air supply, could be developed to give results comparable with those obtained with a flat furnace. Such an installation would involve capital expenditures which do not appear to be commercially justified, as they could not improve materially upon the efficiencies obtained with the present flat cellular-type furnace. An advantage of the cellular type of furnace is the ability to operate a boiler at reduced rating while burning down and cleaning the slag from the grates in one of the multiple cells. Cell-type furnaces are constructed with widths for individual cells ranging from $6^1/_2$ to $8^1/_2$ ft, which appear to be the economical limits of conical piles to be covered by single feedholes.

The combustion-chamber volume, gas-travel length before convection surfaces, and the cross-sectional area of combustion space are related and important in hog-fuel combustion. In comparable installations, the gas weights with hog fuel are approximately 1.7 times the gas weights with oil, and approximately 1.25 times the gas weights with coal. This increased gas weight results in lower combustion-chamber temperatures which are further reduced by the high moisture content of the hog-fuel gases. The decreased temperature does not entirely offset the increased gas weights and larger cross-sectional areas are required when burning hog fuel to give comparable velocities in the combustion space. These factors make it essential to provide larger combustion spaces with hog fuel than with other fuels.

With the modern boiler installation, the increased capacity obtainable with preheated air has been largely responsible for the installation of preheaters rather than any gain in efficiency resulting from their use. When a preheater installation is charged with the extra capital, operating, and power costs, made necessary by the installation of forced- and induced-draft fans, and the necessary gas and air ducts, the low cost per Btu of the fuel precludes the justification of air preheaters on a strictly fuel-saving basis. With hog fuel it is impossible to obtain as low exit-gas temperatures from air preheaters as with other fuels. The high exit-gas temperatures, in part, result from the fact that only 75 to 80 per cent of the air required for combustion can be passed through the air preheater.

With the general introduction of water-cooled combustion chambers in an endeavor to reduce brickwork maintenance, the addition of the preheater has been found desirable in order to decrease the size of the combustion chamber and the length of gas travel necessary from the furnaces to the convection surfaces.

The use of preheaters has made it necessary to use water-cooled grates to avoid excessive grate maintenance. Water-cooled grates have also proved desirable to facilitate grate cleaning. The slag formed from the foreign matter brought in with the fuel does not adhere tenaciously to the water-cooled grates; whereas, with uncooled grates, it is removed with difficulty. Several designs of water-cooled grates have been developed for this service. The heat absorbed in the cooling water is low-potential heat and must be subtracted from the heat available for the production of steam. In many installations the heat obtained in grate-cooling water is used to heat condensate, or make-up water, in this manner supplanting heat which would otherwise be supplied by bled steam which had produced power or by the exhaust from noncondensing auxiliaries. It is, therefore, important in the design and construction of water-cooled grates to provide an arrangement for cooling which will extend the life of the grate, provide for easy cleaning, and, at the same time, extract from the grates and from the preheated air passing through them a minimum amount of this low-potential heat.

Driers

Hog-fuel driers offer attractive potential savings to the power-plant operator. The flue gas leaving an air heater at approximately 500 F contains sufficient heat to remove about $^1/_2$ lb of water per lb of dry wood without dropping the temperature of the gas so low that condensation difficulties will arise. In addition to the savings, the drying of hog fuel gives considerably increased capacity per square foot of grate.

Although hog-fuel driers seemingly have a broad field, the volume of fuel to be dried per available Btu makes necessary a drier of such large physical dimensions that the fixed charges and the operating and maintenance expenses make it difficult to justify the investment.

Several different types of hog-fuel driers have been proposed and tried in this country and abroad, but the authors do not know of any design which has proved completely satisfactory. There is a definite field for a satisfactory hog-fuel drier, but until all of the mechanical difficulties with the prevailing designs can be successfully solved their use will not become extensive.

Cinder Nuisance

The cinder nuisance from hog-fuel-burning plants has increased with higher firing rates required in the modern high-duty boilers equipped with forced- and induced-draft fans and air preheaters. Modern hog-fuel-burning plants are providing either mechanical separators or flue-gas washers for cinder removal. Starting with the use of large single cyclone dust separators, with relatively poor efficiencies on the fine light cinder particles, the necessity of securing better cinder elimination has led to a trial of various designs of mechanical devices, and to the development of special wet gas washers.

There is but little information available concerning the dust loadings of boiler-exit gases. At recent biddings by prominent boiler manufacturers, great divergence in the assumed percentages of unconsumed combustible showed that there was apparently no actual knowledge of the dust loadings to be expected. Bidders allowed as low as 0.25 per cent unconsumed combustible loss, while others allowed as much as 7.5 per cent. Some bidders gave constant percentages over the entire range of ratings, although it is evident that the unconsumed-carbon loss in the flue gas will increase with the rate of firing.

Recently, extensive and carefully conducted tests have been made by one of the country's foremost manufacturers of gas-cleansing equipment. Tests were made at various rates of operation, with and without preheat on distinctly different types of boilers. These tests will give the first authentic data on the dust loadings in flue gases of hog-fuel-fired boilers under variable conditions of firing, fuel, and rating. Unfortunately, this information is not as yet available to the public. The cinder problem when burning hog fuel at high firing rates is a real one. The industry may expect valuable information affecting the best means of cinder collection from such tests and the developments resulting therefrom.

Boiler Capacities

A high and narrow boiler is less expensive than a low and wide one of equal heating surface. The problems of boiler-plant design are not so much the provision of heat-absorbing surface as in obtaining suitable furnaces for combustion of the wet fuels at high rates per unit area of furnaces. Tandem furnaces do not operate as well as furnaces with a single feedhole per cell. Any boiler should have a minimum of two cells to permit carrying part load during grate-cleaning periods, while three cells permit carrying greater loads during such periods. Boiler plants with several boiler units can use fewer cells per unit without material loss of plant capacity. Many installations have furnaces splayed to a greater over-all width than the boiler to permit greater furnace capacity with narrow boilers.

Numerous factors affect the capacity obtained from hog-fuel-fired boilers. The following table is intended to indicate in a general way what capacity should be expected from a well-designed furnace cell of the general dimensions used in modern installations in the Northwest. Values are given for cells with and without preheat and with good fuels of different moisture content:

Moisture in fuel, per cent	Btu input per sq ft of grate area: Without preheat	With preheat
40	680000	850000
48	550000	690000
56	400000	500000

Caution should be exercised in the use of the capacities tabulated as they reflect what can be accomplished under good conditions and in properly designed furnaces.

Recent Developments in Burning Midwestern Coals on Water-Cooled Underfeed Stokers

By H. C. CARROLL,[1] CHICAGO, ILL.

The author points out that there is an increasing trend toward one-boiler-unit operation in a single plant. This is due to a better understanding of boiler and furnace design, feedwater treatment, heat-recovery devices, and fuel-burning equipment, all tending to promote increased efficiency, greater ease of operation, and greater reliability. The application of water-cooling to underfeed stokers can properly be compared to the application of water-cooling to refractory furnace walls; and it is the author's belief that it will prove fully as beneficial in protecting the grate structure of the stoker as has water-cooling proved its value in the protection of refractory furnace walls. The complications involved in applying water-cooling to underfeed stokers are outlined in the paper and numerous successful installations using Midwestern coals are described.

THE first attempt to water-cool multiple-retort underfeed stokers was made in 1929, where the tuyères reciprocated. Flexible hose was necessary to supply water to the cooling tubes, which proved impractical. In 1933, one Midwestern plant found it necessary to burn a cheap strip-mine Illinois coal with preheated air, or convert to gas. The multiple-retort stoker in this plant had stationary tuyères which were cooled by fixed tubes. These were subsequently extended to protect the entire air-emitting surfaces of the stoker.

These original units were protected by small tubes with forced circulation, obtained by taking water from the boiler drum through a booster pump, and pumping it through the stoker cooling tubes and back to the boiler drum. Later designs embodied natural-circulation tubes, connected in the same manner as conventional waterwalls.

At present there are about thirty installations of water-cooled stokers in commercial operation. These vary in size from 35,000 to 225,000 lb per hr, with a combined steam-generating capacity in excess of 3,000,000 lb of steam per hr. These units are distributed over a wide area, using a variety of coals from the eastern Atlantic fields to Nebraska, including Pennsylvania, Ohio, Indiana, Illinois, Kentucky, Iowa, and Kansas coals.

Midwestern Coals

Coal which moves into the Midwestern section of the country from mines located in this area comes from operations in the states of Ohio, western Kentucky, Indiana, Illinois, and Iowa. While the Ohio coals are not, strictly speaking, Midwestern with regard to geographical location, they bear a relation in rank to some of the coals from the Midwestern fields, and are likewise used for industrial purposes there. Table 1 shows in a general way the range of coals coming from these fields.

The approximate tonnages produced during the year 1938, from the states mentioned are as follows:

Ohio	17,920,000
Western Kentucky	7,275,000
Indiana	14,050,000
Illinois	40,650,000
Iowa	3,250,000
Kansas	2,560,000

It will be noted that in general the Midwestern coals are relatively high in moisture, high in ash, high in volatile matter, low in fixed carbon, high in sulphur (except the southern Illinois coals), low in fusion, and dry-heat values from 9800 Btu to 13,800 Btu. Midwestern coals do not stand weathering very well, structurally are not hard, and the grindability index is low. Most of it is classed as free-burning, noncoking, and, due to its high volatile content, easily ignited.

Burning Rates

One of the most important factors in the successful burning of Midwestern coals is the rate at which the coal is burned. We have adopted the projected area of the grate on the underfeed stoker as the area to be used in the calculation of fuel-burning rates.

Where the unit is designed for base-load conditions over the entire period of operation, definite burning rates can be established. However, as is often the case in industrial and isolated city plants which do not have connections with other plants, a range of burning rates must be established which will prove not only economical over the entire range of load, but also will prevent undue maintenance under peak-load conditions.

Keeping in mind the trend toward the single-boiler-unit operation, the water-cooled stoker presents a new possibility in this direction, especially where preheated combustion air is feasible.

Without water-cooling on underfeed grates, and using Midwestern coals with preheated air, continuous operation at a burning rate of 35 lb per sq ft per hr can be maintained conservatively with most coals without undue maintenance; and with possible periods of from 2 to 4 hr of 38 lb per sq ft per hr during peak-load conditions.

However, with water-cooled grates, this range is increased under the same operating conditions to 45 lb per sq ft per hr continuous operation; and for peak periods up to 60 lb per sq ft per hr of from 4 to 6 hr duration, or longer.

This means that a much wider range of loads can be economically carried without undue maintenance, as a smaller grate area can be employed and still have a good efficiency at the lower range of loads.

Where heat recovery is other than in the form of a preheater, such as an economizer, and air at normal temperatures is used for combustion, this range can be materially increased. In the case of the Richmond, Indiana, plant, to be described later, the range

[1] Director, Engineering Department, Commercial Testing & Engineering Company. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

TABLE 1 CHARACTERISTICS OF COALS FROM MIDWESTERN FIELDS

	Moisture, per cent	Dry ash, per cent	Dry volatile, per cent	Dry fixed carbon, per cent	Dry Btu	Dry sulphur, per cent	Fusion temp, F
Ohio	3.5– 9	7.0–12	36–42	50–54	12500–13800	2.5–4	2050–2500
Western Kentucky	6.0–10	5.0– 9	35–40	50–55	13300–13800	2.0–4	2000–2250
Indiana	6.0–15	6.5–15	37–44	46–55	12400–13400	1.0–5	1900–2500
Illinois	7.0–15	7.5–15	35–40	50–57	12400–13400	1.0–5	1900–2500
Iowa	15.0–20	12.0–20	35–40	35–40	9800–10500	4.0–6	1900–2200
Kansas	2.0–12	6.0–16	34–40	45–50	10500–13800	3.0–6	1900–2300

of load is from 30,000 to 100,000 lb of steam per hr, with a burning rate for the minimum load, using Indiana Fifth Vein coal of 11,250 Btu, as received, of 14 lb per sq ft per hr. At 100,000 lb of steam per hr, the burning rate is 50 lb per sq ft per hr. The established operating efficiency of this unit for its load range has been at least 83 per cent over long periods of operation.

Without water cooling and under the same load and conditions, an increase of 25 per cent of grate area would be required to hold the burning rates down to prevent undue maintenance. With the larger grate area required, the over-all operating efficiency would be materially lowered, due to very low burning rates at the light load conditions.

Thus it can be seen that the water-cooled stoker has a distinct advantage in covering a wider range of load more efficiently by being able to employ a decreased grate area.

An actual comparison of air-cooled and water-cooled stokers has been made in the power plant of the Pittsburgh Plate Glass Company, Barberton, Ohio, using Ohio coals. The No. 8 unit with a water-cooled stoker has been in operation 2 years, and the following tabulation gives the performance of this unit over that period:

Pressure, gage, psi	825
Total steam temperature, F	750
Air temperature to stoker, F	300
Continuous capacity (maximum), lb per hr	200000
Total steaming time, hr	14513
Coal burned, tons	143568
Steaming time, per cent	82.8
Capacity factor, per cent	72.4
Full-load capacity for actual steaming time, per cent of	87.4

The average combustion rate, covering the years 1938 and 1939, for the No. 8 water-cooled unit was 46 lb per sq ft per hr, as compared with 38.3 lb per sq ft per hr for the No. 7 air-cooled unit.

A new water-cooled unit, No. 3, which has been in service since April, 1940, will obviously show improved performance over the older units. The range of coals used on this unit is as follows:

Fairmont $^5/_8$-in. N & S, Btu	13500
Champion N & S (Pittsburgh Coal Co.), Btu	13000
Ohio No. 8 N & S, Btu	12500
Ohio No. 5 N & S (Metro), Btu	10500

No difficulty has been experienced in burning any of these coals up to a rate of 60 lb per sq ft per hr. The temperature of the preheated air is about 350 F.

It is the opinion of those responsible for the operation of this plant that "The water-cooled stoker is equally as satisfactory as the air-cooled stoker at approximately 20 per cent higher fuel-burning rate, using 350 F air preheat instead of cold air on the air-cooled unit."

Description of No. 3 Unit

No. 3 unit of the Pittsburgh Plate Glass Company at Barberton, designed for a continuous capacity of 180,000 lb of steam per hr at 860 psi gage pressure, 761 F final temperature, is a 4-drum Babcock & Wilcox bent-tube type, containing 15,015 sq ft of heating surface; a continuous-tube superheater having 5200 sq ft of surface; a continuous-loop Elesco economizer with 7700 sq ft of surface; and a tubular heater of 12,175 sq ft. All four sides of the furnace are water-cooled as follows: Front wall, 682 sq ft of plain area; rear wall, 290 sq ft of plain area, and 176 sq ft armored; side walls, 190 sq ft of plain area, with 360 sq ft armored; total water-cooling, plain area, 1162 sq ft, with 536 sq ft armored; furnace volume, 8650 cu ft.

The water-cooled, 13-retort, mechanical-drive stoker for No. 3 unit, equipped with a clinker-grinder-type dump, has an area of 404.3 sq ft, and is designed for a maximum of 350 F. The induced-draft and forced-draft fans are steam-turbine-driven. The stoker was designed for Ohio bituminous coal, having the following approximate analysis:

Moisture, per cent	2.39
Ash, per cent	11.16
Volatile, per cent	38.36
Fixed carbon, per cent	48.09
	100.00
Btu, as fired	12470
Btu, dry	12776
Sulphur, per cent	3.78
Fusion temperature, F	2200

Table 2 gives the results of tests, conducted on the No. 3 units described. The tests were made with Ohio No. 8 coal of about average quality. The ash has an initial deformation point of 2100 F, fusion 2200 F, and fluid 2300 F.

TABLE 2 DATA AND RESULTS OF ACCEPTANCE TESTS ON NO. 3 UNIT

Test number	2
Date of test	Feb. 14, 1940
Duration of test, hr	24
Steam generated, lb per hr	153556
Proximate Analysis of Coal Fired	Per cent
Moisture	3.56
Ash	11.90
Volatile matter	38.53
Fixed carbon	46.01
Heating value, Btu per lb of coal	12422
Combustible (unburned carbon) in refuse, per cent	8.46
Flue-Gas Analysis at Boiler Outlet	Per cent
CO_2	13.24
O_2	6.03
CO	0.20
N_2	80.53
CO_2 in flue gas at economizer outlet	13.01
CO_2 in flue gas at air-heater outlet	13.01
Temperatures	Deg F
Steam temperature at superheater outlet	779.8
Superheat	253.5
Air temperature entering air heater	79.5
Air temperature leaving air heater	302.8
Gas temperature leaving boiler	695.2
Gas temperature leaving economizer	573.5
Gas temperature leaving air heater	369.1
Feedwater entering economizer	222.8
Water temperature leaving economizer	288.7
Wet-bulb temperature of air entering system	53.2
Steam Pressure	
Drum pressure, psi, abs	888
Header pressure at superheater outlet, psi, abs	858
Hourly Quantities	
Coal burned, lb per hr	13458
Combustion rate, lb of coal per sq ft of grate per hr	45.7
Heat Balance	Per cent
Heat absorbed by steam and water in economizer, boiler, and superheater, including blowdown	84.45
Heat loss due to unburned combustible in refuse	1.30
Heat loss due to incomplete combustion of carbon (CO)	0.81
Heat loss due to sensible heat in dry gas	6.93
Heat loss due to evaporation of moisture in coal	0.35
Heat loss due to combustion of hydrogen in coal	4.12
Heat loss due to moisture supplied with combustion air	0.03
Radiation and unaccounted for losses	2.01
	100.00

TABLE 3 DUST WEIGHTS DETERMINED FROM TESTS CONDUCTED AT IOWA STATE COLLEGE

Date, 1939	Rating of unit, lb steam per hr	Run no.	Total weight, grains	Flue gas sampled, cu ft	325 mesh or above, per cent	Grains of 325 mesh or above	Total grains per cu ft of gas	Grains of 325 mesh or above per cu ft of gas
12–6	65000	1[a]	20.9801	337.38	23.39	3.15	0.0622	0.00818
					10.52			
12–6	65000	1[a]	39.1397	457.19	27.81	8.9259	0.0856	0.0915
					27.70			
12–7	80000	2	20.3707	140.61	38.05	7.75	0.1448	0.0551
12–7	80000	2	43.71	172.15	20.90	9.136	0.2539	0.0530
12–7	80000	3[b]	22.8896	168.35	29.88	6.84	0.1359	0.0406
12–7	80000	3[b]	58.00	264.25	41.55	24.10	0.2198	0.0912
					Average..........		0.15036	0.05659

[a] Two sets impinging bottles used during this run. Weights were added.
[b] Ashpit cleaned during this run.
NOTE: These results are with atmospheric conditions, 30 in. Hg and 60 F. The results disclose dust loading of 1/10 of the guarantee specified.

RESTRICTION OF FLY ASH

With increasing restrictions by the smoke ordinances regarding the emission of objectionable fly ash, in addition to smoke requirements, it is necessary to set a limit in specifying the emission of fly ash from any firing equipment. On underfeed-stoker installations, without provision other than the usual means provided in boiler units for the collection of fly ash, we are specifying that the unit shall have an efficiency of collection such that not more than 0.5 grain of dust + 44 mu in size shall be present per cu ft of flue gas, measured at 70 F, and 29.92 in. Hg barometer.

On a recent water-cooled underfeed-stoker installation, fly-ash tests over a range of loads were conducted. The coal burned had the following characteristics:

Moisture, per cent	15.44
Ash, per cent	18.62
Volatile, per cent	32.68
Fixed carbon, per cent	33.26
	100.00
Btu per lb	9282
Sulphur, per cent	4.48
Fusion temperature, F	1958

It is recognized that size consist of the fuel has considerable to do with emission of fly ash, especially on some types of firing.

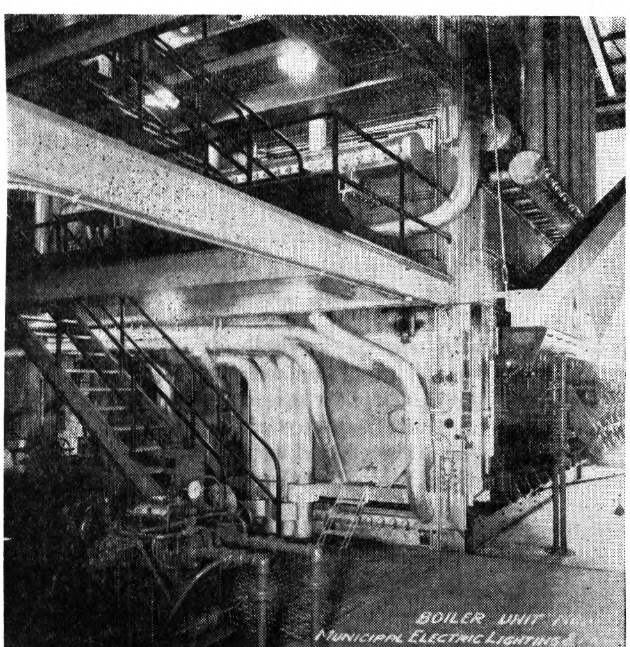


FIG. 1 BOILER UNIT NO. 4 OF THE MUNICIPAL ELECTRIC LIGHTING AND POWER PLANT, RICHMOND, INDIANA

While we cannot be definite in saying that the water-cooled underfeed stoker will emit less fly ash than the air-cooled stoker, yet, the fact remains that lower wind-box pressures prevail at the same burning rates due to a more homogeneous fuel bed, resulting in a more uniform air distribution, made possible by the cooling of the tuyères and the absence of adhering clinkers.

Through the courtesy of the Chicago Smoke Department we were permitted to use their equipment on this test. This equip-

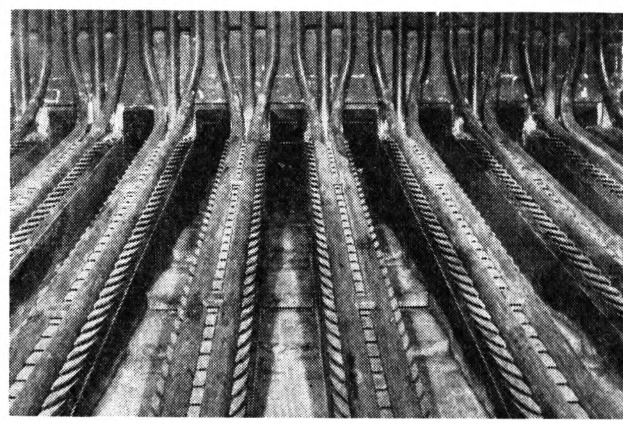


FIG. 2 INTERIOR VIEW, SHOWING SIDE WATERWALLS OF WATER-COOLED STOKER, RICHMOND PLANT

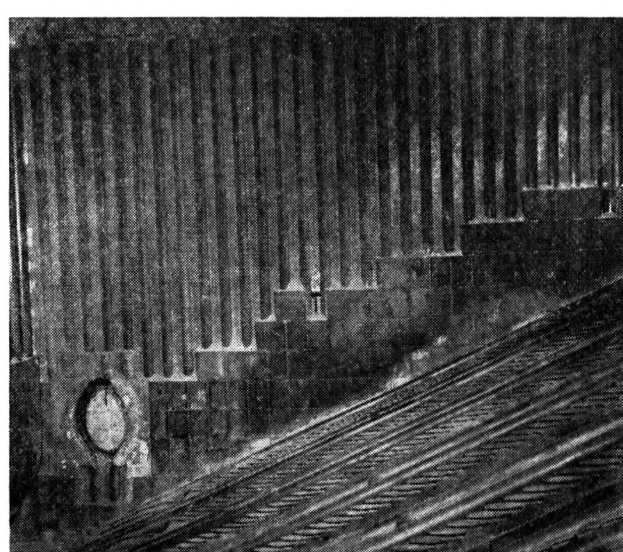


FIG. 3 ANOTHER VIEW OF THE WATER-COOLED STOKER AND THE SIDE WATERWALLS

ment has been described in a recent article.[2] The results are given in Table 3.

Description and Results of Water-Cooled Stoker Unit at Richmond, Indiana, Municipal Plant

Boiler. This Richmond unit, designed for a continuous capacity of 100,000 lb of steam per hr at 430 psi gage pressure and 825 F final temperature, is a Babcock & Wilcox 4-drum, bent-tube type, containing 10,112 sq ft of heating surface, with a pendant-type superheater having 2218 sq ft of surface, and a return-bend economizer of 3350 sq ft. All four sides of the furnace are water-cooled, as follows: Front wall, 440 sq ft; rear wall 142 sq ft with 95 sq ft armored; side walls partially water-cooled, 126 sq ft, and 306 sq ft armored; the remainder of the upper walls is refractory. The total plain surface is 708 sq ft and the total armored surface, 401 sq ft. The furnace volume is 5140 cu ft.

Hagan automatic control is installed, together with complete instrumentation of the unit.

Stoker. The stoker is a continuous-dump, 9-retort, hydraulic-drive, Taylor water-cooled unit with an area of 255 sq ft, designed for a maximum air temperature of 80 F, to burn Indiana coal. The induced-draft fan has two motors on the same fan for 2-speed control.

Overfire air from a plenum box, extending across the entire furnace width and supplied from the main mud box, is distributed above the fuel bed by a series of nozzles inserted through the front waterwall tubes.

This unit was placed in service in March, 1939. The record made during a continuous run from August 1, 1939, to January 22, 1940, when the unit was taken out of service for inspection, is given in Table 4.

TABLE 4 PERFORMANCE OF RICHMOND UNIT DURING CONTINUOUS RUN AUG. 1, 1939, TO JAN. 22, 1940

Total continuous service, hr	4184
Total steam generated, lb	256,935,500
Total coal burned, lb	30,473,400
Average evaporation, lb	8.43
Average heat value of Indiana coal (as fired), Btu	11,266
Total heat per lb of steam generated (435 lb gage; 700 F total temp), Btu	1358
Heat in feedwater to economizer above 32 F (270 minus 32), Btu	238
Btu added to steam generated in unit	1120
Combined efficiency (entire period), per cent	83.8
Average steam per hour, lb	61,500
Minimum loads as low as, lb	25,000
Maximum loads as high as, lb	120,000
Guaranteed performance at 75,000 lb of steam per hr, per cent	82.15

TABLE 5 CHARACTERISTICS OF FIFTH VEIN INDIANA $1^1/_2$-IN. SCREENINGS

Name of coal	Hercules	Knox	Patoka	Elberfeld	Panhandle	Enos
Moisture, per cent	11.41	8.66	9.91	8.75	10.15	10.18
Ash, per cent	10.72	11.67	8.95	10.78	9.94	10.18
Btu	11873	11493	11950	11440	11205	11462
Sulphur, per cent	2.90	4.56	3.59	3.50	2.79	3.98
Fusion temperature of ash, F	2200	2076	2200	2210	2230	2110

The commercial performance of this unit over the entire operating period approximates very closely the guaranteed and predicted performance set up in the contract.

A careful inspection of the unit at the conclusion of this run, after having burned 23,000 tons of coal since March, 1939, revealed no indication of any burning of either tuyères or tuyère supports in any of the nine retorts. The kicker bars and overfeed mechanism were practically in their original condition. Some indication of overheating was noted on the steplike ends of the bars. The dump plates, with the exception of two regions about 4 in. diam where they were burned, were in perfect condition.

The slagging on the first bank of boiler tubes, which is protected by a slag screen, was not enough to interfere with the draft loss or efficiency. No new stoker parts were installed at this inspection, or at any previous inspection. The slagging was cleaned up, and the unit returned to service. It has been in continuous service ever since, with equal efficiency.

Six different grades of Indiana Fifth Vein screenings were used during this run, and no difficulty was experienced with any of these fuels. The characteristics of these coals are given in Table 5.

The availability factor for the period from August 1, 1939, to April 30, 1940, was 98.4 per cent; the efficiency for the entire period was 83.1 per cent.

TABLE 6 ACCEPTANCE TEST ON RICHMOND UNIT; HEAT BALANCE AT GUARANTEED POINT

	75,000 Lb per Hr Btu	Per cent
Heating value of fuel, dry basis	12892.00	100.00
Heat absorbed by water in economizer	520.84	4.04
Heat absorbed by water and steam in boiler	8644.08	67.05
Heat absorbed by steam in superheater	1482.58	11.50
Heat absorbed by steam-generating unit	10647.50	82.59
Heat loss due to moisture in coal	168.89	1.31
Heat loss due to water from combustion of hydrogen	555.64	4.31
Heat loss due to moisture in air	21.92	0.17
Heat loss due to dry chimney gases	1063.59	8.25
Heat loss due to unburned gaseous combustible	....	..
Heat loss due to unconsumed combustible in refuse	198.54	1.54
Heat loss due to unconsumed hydrogen, hydrocarbons, radiation, and unaccounted for	235.92	1.83
Total, dry basis	12892.00	100.00

TABLE 7 POWER CONSUMPTION; MEASURED HORSEPOWER INPUT TO AUXILIARIES

Load, lb of steam per hr	50000	75000	100000
Forced-draft fan	29.8	32.0	38.2
Induced-draft fan	23.0	30.83	67.44
Hele-Shaw stoker drive	7.0	7.0	7.0
Total hp input	59.8	69.83	112.64
Equivalent kw	44.9	52.5	84.6
Tons coal fired per hr	2.80	4.3	5.79
Input per kw per all auxiliaries per ton of coal as fired	16.0	12.2	14.6

TABLE 8 ANALYSIS OF ATKINSON COAL

	As fired, per cent	Dry, per cent
Moisture	18.27	...
Ash	10.01	12.25
Volatile	34.12	41.75
Fixed carbon	37.60	46.00
	100.00	100.00
Sulphur	2.87	3.50
Btu per lb	10053	12300
Fusion temperature F	1930 F	

All of the auxiliaries are electrically driven. The induced-draft fan has a low- and high-speed motor.

The 50,000- and 75,000-lb[3] per hr loads on the unit were carried by the low-speed motor; and the effect of the high-speed motor on the 100,000-lb load is reflected in the higher consumption of current per ton of coal fed to the stoker.

Reports From Other Plants Using Midwestern Coals

Municipal Light & Power Plant, Fremont, Neb. This plant has a water-cooled crusher-roll-type stoker, and is designed to generate 75,000 lb of steam per hr at 400 psi gage, and 750 F total steam temperature.

[2] "Low-Cost Flue-Dust Sampler," *Power*, April, 1940, pp. 70–72.

[3] As this unit was designed for maximum over-all economy at 75,000 lb, it is interesting to note that the actual results reflected this economy at the desired point.

The unit is burning Kansas coals originating in the Pittsburgh district. Although these coals carry a relatively high Btu, from 12,400 to 12,700 the fusion temperature of the ash consistently ranges between 2000 and 2100 F.

This unit recently completed a continuous run of 237 days, or 5670 hours. During that period the combined efficiency was 83.4 per cent when burning the Kansas coals of approximately 2000 F ash-fusion point.

Municipal Light Plant, Greenwood, Mississippi. This plant has a 6-retort, 41-tuyère, water-cooled C.A.D. stoker, having the same retort design as now used at Richmond, Ind. Coal originating in Webster County, Kentucky, has been used. The author has been advised that 11,000 tons of coal have been burned without replacement of a single part.

Iowa Electric Light & Power Company, Cedar Rapids, Iowa. This plant has probably burned more Midwestern coal on water-cooled underfeed stokers than any plant in the country with results reported as follows:

"Regarding the underfeed stokers referred to, four of these stokers were started on Atkinson coal in 1934, and four in 1935. This report will show the total amount of coal burned from the time the stokers were first started on Atkinson coal until the first of the year 1940. The total coal on the eight stokers is 574,772 tons.

"In 1935, we installed one stoker of the crusher type, water-cooled; and until January, 1940, this stoker burned 82,450 tons. The total cost on this particular stoker is 2.226 cents per ton of coal burned.

"The balance of the stokers had considerable work done on them last year, which was all on the outside of the stokers, such as ram boxes, crankshafts, crankshaft bearings, and gears. We did not feel that the total amount of this expenditure should be charged against the Atkinson coal, as these stokers were installed at least 10 years previous.

"On these stokers we have tabulated the amount of the expense on the inner parts of the stokers, such as pushers, tuyères, tuyère supports, and other items on the inside of the firebox. This cost is 4.758 cents per ton, made up as follows: Pushers, 1.108 cents; tuyères, 1.827 cents; remainder of stokers, such as side frames, dump plates, and other internal parts, 1.823 cents.

"Some of this cost also includes experimental work with the water-cooling system, as we made about four changes before we actually achieved the desired results.

"The average coal-burning rate on the fuel-burning portion of the stoker is 42 lb per hr; we have run on a week's test as high as 60 lb per hr.

"In regard to the date we first started experimenting with water-cooling, this was the early part of 1933."

Libbey-Owens-Ford Glass Company, Rossford, Ohio. "The Libbey-Owens-Ford Glass Company had three bent-tube boilers of 12,500 sq ft area at the Rossford, Ohio, plant. These boilers were equipped with water-cooled side and rear walls, and multiple-retort underfeed stokers.

"In order to widen the range of fuels that could be burned, it was decided in 1937, to modify these three stokers to include water-cooled grates. As a result of these changes, we have widened the range of coals that can be burned to advantage, and thereby placed ourselves in a better position to purchase any coals.

"We have burned on these stokers, since water-cooling them, over 75,000 tons of coal. At times we have operated only one boiler, and at other times three boilers, according to the steam demand.

"The ratings normally vary from 75 to 225 per cent; the average rating is approximately 175 per cent. The average over-all operating efficiency is better than 80 per cent.

"The stokers installed under boilers *B* and *C* have just been overhauled at a cost of less than $300 each for material and labor. The stoker installed under *A* boiler will be repaired at a later time.

"Approximately 50 per cent of the coal burned was Ohio No. 8 fusing at 2100 F. The remaining 50 per cent was West Virginia and Kentucky coal containing ash which fuses from 2400 to 2600 F. The temperature of the preheated air varies from 280 to 310 F. We have not had to take a boiler off the line in 2 years as a result of stoker trouble."

Advantages of Water-Cooled Grates on Midwestern Coals

Some of the advantages of water-cooled-stoker installations burning Midwestern coal, as compared to air-cooled stokers using the same fuel, have been noted as follows:

1 There is more uniform flow of coal, with a consequent ease of distribution.

2 A more homogeneous fuel bed has been noted than with straight air-cooled stokers; and better air-distribution results.

3 The size of clinkers, with proper burning rates for coal, has not been objectionable, and they have been easily discharged automatically under the bridge wall with continuous-dump design.

4 There is every indication that the maintenance will be low.

5 Indications are that the power consumption is lower.

6 The range of fuels with low ash-fusion characteristic is greater; and the flexibility of fuels is greater.

7 Test efficiencies are more closely approximated in regular plant operation than on the air-cooled stoker because of the ease in obtaining high CO_2 in water-cooled units.

8 The availability of the unit is materially increased.

9 With crusher-type water-cooled stokers less artificial cooling of the ash is required, and, consequently, less moisture put into the flue gases. The quick chilling of molten ash particles assures a continuous and even flow of fuel.

10 The water-cooled stoker provides an additional factor of safety against careless operation.

11 A distinct purchasing advantage is obtained in that fusion temperature and sulphur content are not limiting factors, as is often the case without water-cooling.

12 Greater latitude of operation.

Discussion

T. C. Cheasley.[4] There certainly can be no doubt as to the care exercised by the author in securing data to present in this paper. Also, I think all of us will agree, the statements he has made are conservative and will stand close scrutiny.

Unfortunately, there has been comparatively little opportunity for most of us to see water-cooled tuyères in operation and we must, therefore, draw on our imaginations a good deal to visualize the results reported.

Certain it is that the burning of most Midwestern coals has presented problems to the designers of furnaces and boilers as well as to operating engineers unless, and except, the burning rates have been held at fairly low maximums.

It is generally conceded that the washing of Midwestern coals has done much to improve operation and to reduce stoker and furnace maintenance, but it is the opinion of some of us that the average coal in the average plant is not being given the opportunity to produce its highest combustion efficiency.

[4] Fuel Engineer, Sinclair Coal Company, Kansas City, Mo. Mem. A.S.M.E.

By this is meant the frequent departure from fundamentals in speaking of stoker operation, and also in actual operation. We speak of the "pounds of coal burned per square foot of grate area" and, of course, there is no such operation as burning coal in an underfeed stoker, i.e., if the stoker is being properly operated. A stoker does not burn coal, it converts coal to coke, which is burned in progression. The fact that we talk and think in "pounds of coal burned" is, in the writer's opinion, at least in part responsible for some of the high maintenance costs mentioned. At the same time this condition causes some rather good coals to become discredited.

Extremely high excess air "at the point of volatile release," is probably largely responsible for this, and the "average" CO_2, shown at the usual point of taking gas samples, does not tell the whole story. Undoubtedly, the water cooling of tuyère areas, which are exposed to high temperatures, will reduce the build up of temperature in the stoker parts, and thus should decrease maintenance costs by prolonged life of these parts. However, as the writer views the situation, the chief benefits, as stated by the author, should be the opening up of greater selections of coal with the advantage in many cases of very much lower delivered prices and lower steam costs.

F. S. Scott.[5] Definite and marked improvements have been made in underfeed stokers in recent years. It will satisfactorily meet the present more exacting requirements of steam-generating plants.

This paper describes the water-cooled stoker and its contribution to the art. The idea of water cooling a stoker is not new. It is an advancement of the art. Its development has been retarded by the physical difficulties which are reflected in the cost.

The Westinghouse Electric & Manufacturing Company has developed the underfeed stoker along different lines from water cooling. Our development of the link-grate stoker has made a definite improvement in the ability of the underfeed stoker to handle Midwestern coals. High efficiency, low maintenance, and good reliability are obtained.

This link-grate underfeed-type stoker was first installed in 1928. At first it was only applied with a clinker-grinder-type stoker. The continuous-ash-discharge type without a clinker grinder was first installed in 1931. At present there are more than 150 continuous-ash-discharge link-grate stokers and over 85 link-grate clinker-grinder stokers in operation, using all types of bituminous coals, from those mined in Iowa to the Eastern low-volatile or Pocahontas-type coals.

There are 72 link-grate stokers in operation on Midwestern coals with a base-load capacity in excess of 5,000,000 lb of steam per hr. The peak-load capacity is considerably above this figure.

Burning Rates

The author expresses the opinion that 35 lb of coal per sq ft per hr of projected grate area is conservative for continuous operation with air-cooled underfeed stokers using Midwestern coals. The writer's experience indicates that, if such rates are applied to link-grate underfeed stokers, it unnecessarily increases the first cost and causes difficulties at low loads.

It is very important that serious consideration be given to burning rates on these stokers. The difficulties can be as great with a stoker that is too large as with one that is too small. The maximum burning rates on underfeed stokers are determined by the ability of the fuel bed to stay on the grates. High air velocities cause the coal to "blow" or lift from the fuel bed, thereby limiting the maximum burning rates. The velocity of the air is determined by the quantity flowing through the fixed area of tuyère openings. The quantity of air flowing, when the excess air is constant (or CO_2), is in turn determined by the "heat duty" or the combustible burned. Expressing "heat duty" in other terms, it is the Btu released from the coal per square foot of projected stoker area per hour.

For many years we have based our engineering on Btu release per unit area with due allowance for the agglutinating quality of coal which is a measure of its resistance to being blown away. Without exception, the predicted combustion rates have been readily maintained.

There are two parts which require cooling when a stoker is burning coal, i.e., the grates and the ash adjacent to the grates. If the ash near the grates is cooled well below the "sticky" state and the grates are also cool, there will be no difficulty with the movement of the fuel bed and no burned stoker parts. It is a known fact that high velocities of fluids flowing past solids cause better heat transfer than low velocities. Therefore, the writer believes it to be logical to state that high air velocities through the stoker tuyères, but within the practical limits, will cool both the grates and the adjacent ash better than very low velocities. This fact has been demonstrated in practice. Where stokers have been operated at very low rates or on "bank" for long periods of time, the fuel bed becomes loaded with large clinkers, the stoker iron can be observed "red hot," and parts are burned. These large clinkers are then difficult to move when the load is increased. This indicates some of the dangers inherent in selecting a stoker that is too large.

Stokers can also be selected which are too small. The lack of induced draft or insufficient "black" surface exposed to the fire in the furnace is the more frequent cause of difficulties with fuel beds when the load on the boiler has been increased. It is indicated that burning rates should not be considered in pound of coal burned per sq ft per hr because a stoker is primarily machine for converting the potential heat in coal to sensible hea in gases. Burning rates should be measured by the heat duty or the Btu release per square foot per hour. We are interested in materials handling only in so far as it affects the efficient production of heat for use by the heat-exchanging apparatus.

It is thought to be generally considered that 40 lb of coal per sq ft per hr of projected grate area is conservative practice for continuous loads when using Eastern coal of 14,000 Btu per lb as received. This is 560,000 Btu heat release per sq ft per hr of projected area. If the same rate in pounds per square foot is used for Midwestern coals of 11,000 Btu per lb as for Eastern coals, the heat release would be 440,000 Btu per sq ft per hr or a reduction of 21.5 per cent in the heat-release duty per square foot of projected stoker area. If the same rate of heat-release duty is used for Midwestern coals, as for Eastern coals of 560,000 Btu per sq ft per hr, the rate in pounds of 11,000-Btu coal would be about 51 lb per sq ft per hr. This shows an increase in material passed over the stoker of about 27.5 per cent with no increase in the heat-release duty.

The nature of the constituents of the ash in Midwestern coals and their behavior when subjected to furnace conditions indicate that the heat duty should be reduced on stokers using them as against Eastern coals. Long experience indicates that 21.5 per cent reduction in heat duty is considerably greater than is necessary or economical.

A 10.7 per cent reduction in the heat-duty rate of 560,000 Btu per sq ft per hr to 500,000 Btu per sq ft per hr of projected stoker-grate area is conservative. Even though the heat-duty rate has been dropped to 500,000 Btu per sq ft per hr, the amount of material passed over this stoker area is increased from 40 lb per sq ft per hr to 45.5 lb per sq ft per hr. These figures are quite conservative in that there is a considerable number of stokers which are operating satisfactorily at base loads above this point.

[5] Stoker Engineer, N. W. District, Westinghouse Electric & Manufacturing Company, Chicago, Ill.

This brief discussion leads to the conclusion that the basic and most important item in determining the desirable burning rates for stokers is the Btu heat release per square foot per hour of projected grate area.

It has been found that, with Indiana, Illinois, and similar coals, air-cooled link-grate underfeed stokers operate best, throughout their load range, with a continuous maximum burning rate of 500,000 to 550,000 Btu heat release per square foot per hour of projected area. If the coal has 11,000 Btu per lb, this is a rate in pounds of coal material passed over the stokers of 45.5 to 50 lb per sq ft of projected stoker area.

Loads of from 4 to 6 hours in duration can be satisfactorily carried up to 825,000 Btu per square foot per hour or 75 lb of 11,000-Btu coal per sq ft per hr.

Our experience with many stokers indicates that the link-grate underfeed stoker can conservatively burn more coal per square foot per hour, with all the advantages named by the author, than he feels can be justified with the special water-cooled stoker. To demonstrate the foregoing point the following examples are given:

One installation of two 7-retort link-grate continuous-ash-discharge stokers in Iowa burns Iowa coal of 8500 Btu per lb to 10,500 Btu per lb. The stokers having 178 sq ft of projected grate area have been in operation more than 2 years. During the last year the maintenance has been less than 2 cents per ton of coal burned. The normal continuous load with 10,500-Btu coal is 40 lb per sq ft per hr or 420,000 Btu heat release per sq ft of projected area. The same normal continuous steam output is carried when this plant burns 8500-Btu coal. The rate with 8500-Btu coal is 49.5 lb of coal per sq ft or a heat release of 420,000 Btu per sq ft of projected area per hr.

No difficulty has been experienced up to the limit of the induced-draft fans or 20 per cent above these figures. The efficiencies being obtained are approximately the same as those guaranteed.

Another set of installations consisting of five identical stokers in three different plants burns various kinds and grades of Indiana and Illinois coals of approximately 12,000 Btu per lb as received. The normal continuous load carried on these stokers of 153 sq ft of projected area is about 46 lb per sq ft or 550,000 Btu heat release per sq ft per hr. The induced-draft fans limit the maximum load to a rate of approximately 59 lb per sq ft per hr or 710,000 Btu per hr of projected stoker area.

These units have been installed more than 2 years. The maintenance for these stokers has been less than 1 cent per ton of coal burned.

Efficiency

Another point the writer would like to comment upon briefly is that of boiler-unit efficiency. A comparison of boiler-unit efficiencies of units carrying different pressures and of different designs means but little when the discussion concerns firing equipment only. The efficiency of firing equipment, or the combustion efficiency when the coal is the same, depends upon three factors which may be varied (1) the excess air in the products of combustion in the furnace, as measured by the per cent CO_2, (2) the combustible loss to the ashpit, (3) the soot and cinder loss. The greatest of these losses is that in the dry gases or that due to excess air, as measured by the per cent CO_2 in the products of combustion.

The author has shown that both the ashpit loss and the soot and cinder loss are low with underfeed-stoker firing. He has shown in one place the percentage of CO_2 at the boiler outlet. This is for the water-cooled stoker at Barberton, Ohio—it is 13.24 per cent.

In the first example previously mentioned in this discussion, the CO_2 averages 13.5 to 14 per cent at the boiler outlet, in daily operation.

The second set of installations using Indiana and Illinois coals averages 14 to 15 per cent CO_2 at the boiler outlet, in daily operation.

A modern boiler unit, fired by a link-grate underfeed stoker, will produce in excess of 14 per cent CO_2 at the boiler outlet without stack smoke and no use of secondary air at continuous burning rates up to 550,000 Btu per sq ft per hr of projected stoker area.

Advantages Claimed for Link-Grate Stokers

The author indicates that the water-cooled underfeed stoker shows certain advantages over similar air-cooled stokers. It is claimed that the link-grate underfeed stoker shows advantages over other types of underfeed stokers burning Midwestern coals as follows:

1 More uniform flow of coal and, therefore, more even distillation of gases and burning of coke.

2 Absence of large clinkers at the rear of the stoker makes it possible to burn out more of the combustible before the refuse enters the ashpit.

3 Maintenance is low.

4 The range of fuels is greater so that the ash-fusion point of the coal is relatively unimportant. It is possible to handle coal satisfactorily with an ash content of 25 per cent or more.

5 Tests efficiencies can be closely approximated in regular operations.

6 Reliability is increased.

7 High continuous capacities are easily obtained.

R. L. Swinney.[6] The author mentions the trend toward one-boiler-unit operation in a single plant. This trend is the result of the increasing number of steam-generating units which are demonstrating their ability to stay in continuous service over long periods of time.

The unit at the Richmond, Indiana, municipal plant is mentioned as having been in continuous service for a period of 175 days or approximately 6 months.

There are many other similar records of units burning pulverized coal, oil, and gas, some of these plants having 100 per cent feedwater make-up.

Progress in the art of feedwater treatment and improvements in the design of water-cooled furnaces and fuel-burning equipment have made these records possible, but much credit is due the operators of plants where such records have been established. It is predicted that properly designed steam-generating units will be more generally accepted as having dependability for continuous service approaching that of turbines.

A. W. Thorson.[7] The author is to be commended for bringing the literature up to date on the subject of his paper. It is gratifying to know that numerous installations of this type of equipment are performing satisfactorily as indicated by the wealth of operating data included in the paper.

The text and part of the discussion have specified maximum combustion rates either in pounds or Btu per square foot per hour. The writer would suggest that the safe maximum burning rate depends not only upon the heating value of the fuel but to a considerable extent upon the ash content and the fusion characteristics of the ash.

Does the use of the water-cooled stoker bring about any reduction in investment costs? It appears that it might, inasmuch, as steam-generating capacity is increased by the stoker-cooling tubes

[6] Sales Engineer, The Babcock & Wilcox Company, Chicago, Ill.

[7] Assistant Fuel Service Engineer, Chesapeake and Ohio Railway Company Mem. A.S.M.E.

and the cost of the stoker itself is probably not much higher; and also because the maximum possible combustion rate appears to be higher with water cooling than without for the same fuel.

Author's Closure

Mr. Scott's discussion has introduced some interesting information concerning the developments of non-water-cooled stokers. It is the author's opinion, however, that the information furnished by Mr. Scott is largely the commercial viewpoint of one manufacturer.

As brought out in the evidence presented in the paper, the experience of users is that claims for the link-grate-stoker performance, when burning Midwestern coals, are not as consistently maintained as where the water-cooled grate is employed, particularly at the higher coal-burning rates when using preheated air. Therefore, it is logical to conclude, from the experience of users, that water cooling will afford more consistent and reliable protection for the grate than when the sole grate-cooling medium is air. The flow of water through the cooling tubes is not affected by fuel-bed thickness, porosity, or characteristics of the coal; while uniform and desired air flow through the grate is influenced by these factors.

The matter of burning rates mentioned by the discusser is one of proper application to the particular coal under consideration, and it is doubtful whether a fixed rate of heat release can be used to cover the characteristics of all Midwestern coals. The rates set up by the author were determined from long experience in connection with the average coals of the Midwest.

The burning characteristics of Midwestern coals are too greatly diversified to establish any definite percentage of reduction in heat duty when burning these coals, and it is a matter of careful decision on the part of the engineer in setting grate areas.

The Fuel-Bed Tests at Hell Gate Generating Station, 1937-1938

By M. A. Mayers,[1] W. H. Dargan,[2] Joseph Gershberg,[3] B. C. Dalway,[4] M. J. Williams,[5] and E. R. Kaiser[6]

This paper is a final report on fuel-bed tests at Hell Gate Generating Station and includes some of the conclusions reached in the authors' study of the data up to this time. The fuel bed of a multiple-retort underfeed stoker is shown to consist of vertical strata parallel to the center lines of the retorts and tuyère stacks in which coking and ignition processes occur along the entire length of the stoker. The actual burning of the carbonized fuel occurs largely by overfeed action in the burning lanes where the temperature and pressure gradients are determined by the rate of primary-air flow, and the gases, rising from the burning lanes, may require the addition of secondary air for complete combustion. It is shown that control of the contour of the fire in operating practice may affect not only the amount of coke blown from the fire and the degree of gas stratification in the boiler passes, but also the severity of treatment to which the stoker iron is subjected.

Introduction

This report gives the results of measurements of the temperatures, gas analyses, and the air and gas pressures in the fuel bed of a typical large commercial underfeed stoker. The work continued the studies of the Coal Research Laboratory of the Carnegie Institute of Technology on the nature of combustion processes, and arose from a desire to apply the results of previous work to the fuel beds which exist in commercial equipment. The Consolidated Edison Company of New York expressed its interest in such measurements and offered its cooperation, if a practicable program could be worked out. The Coal Research Laboratory invited the participation in the project of the staff of Bituminous Coal Research, Inc., at Battelle Memorial Institute, because of their interest and experience in the investigation of combustion problems. The research organizations submitted a tentative program on March 17, 1937, which called for the investigation of the following main variables: (*a*) The rates of flow of coal and air; (*b*) temperatures; and (*c*) gas compositions at various points in the fuel bed. The temperatures and gas pressures were to be observed, and gas samples for analysis drawn by the use of probes of refractory material inserted into the fuel bed from the wind box through holes in the stoker iron. The influence on the three variables of the following operating factors was to be determined: the rate of burning, the amount of excess air, the contour of the fuel bed, the size and size consist of the coal, and the kind of coal. It was estimated that the studies would require at least 6 months if only a few coals were studied.

The tentative program was approved by the Consolidated Edison Company on April 9, 1937. Hell Gate Generating Station was chosen for the tests because the existence of an isolated bunker at that station would facilitate the use of different coals during the tests. It became apparent on trial that it would be unsafe for men to work in the stoker wind box principally because of the danger of serious injury in the event of a furnace-wall tube failure. Inspection of the stokers installed in the station showed that only on the seventh-row stokers was it possible to insert probes into the retorts from below without interference with the links operating the secondary rams. Hence, an isolated working place from which the probes could be handled was built into the wind box of boiler No. 73 while it was out of service for routine maintenance in July, permitting preliminary tests of the probes during the two-week period, August 19 to September 4.

These tests showed that the isolated working place, later known as the doghouse, provided adequate safety and convenience for the test crew, and that the routine of the fuel-bed tests did not interfere with operation of the boiler. They also showed that it was feasible to insert probes into the fuel bed from below, but indicated that certain changes in their design were desirable. New probes designed in accordance with these indications were made and testing was started November 1. By January 1, 1938, the equipment had been tried out and a single series of measurements at a low burning rate had been finished. These data were embodied in a first progress report distributed to the parties to the investigation at the end of February.

On January 1, 1938, Bituminous Coal Research, Inc., withdrew from cooperation in the project, because the funds it had allotted to the work were exhausted. The program was continued by the Consolidated Edison Company and the Coal Research Laboratory with the assistance of the Pittsburgh Coal Company. Three additional series of tests were made in which the load, the percentage of excess air, and the fire contour were varied separately, but in all of which the same coal, one normally used in the station, was fired. In these series, traverses were made at three sections in the stoker but, as examination of the data indicated the action of the stoker to be much the same at all sections from the neck to the pit, measurements in the last three series, in which special coals were fired, were made at only one section in the center of the stoker. Active testing was finished by April 29, 1938, and the motion pictures of fuel-bed behavior made as a part of this project were completed during the week ending May 14.

The results of the last six series of tests were submitted to the sponsors in a second progress report, distributed in October, 1938. A report embodying an outline of the results and tentative conclusions subject to revision was presented orally before the annual meeting of this Society in December, 1938. In addition to these reports, the apparatus used for the fuel-bed motion pictures has been described in a paper by Markson and Dargan (22)[7] and the

[1] Coal Research Laboratory, Carnegie Institute of Technology, Pittsburgh, Pa. Mem. A.S.M.E.

[2] Research Bureau, Consolidated Edison Company of New York, New York, N. Y.

[3] Technical Service Department, Consolidated Edison Company of New York, New York, N. Y. Mem. A.S.M.E.

[4] Contract Control and Inspection Department, Consolidated Edison Company of New York, New York, N. Y.; formerly Technical Service Department, Consolidated Edison Company of New York.

[5] Pittsburgh Coal Company of Wisconsin, Duluth, Minn.; formerly Pittsburgh Coal Company, Pittsburgh, Pa.

[6] Fuels Division, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., December 5-9, 1938, of The American Society of Mechanical Engineers.

[7] Numbers in parentheses refer to the Bibliography at the end of the paper.

method of correcting temperatures has been reported by Mayers (25).

Conclusion and Summary

This work represents an initial survey of the conditions under which combustion takes place on multiple-retort underfeed stokers. The data are not as precise or as detailed as might be desired, they apply specifically only to the particular piece of equipment on which the tests were made, and they do not cover a wide range of fuels. It is desirable that further tests of this nature be made on other stokers of different construction, burning many more of the hundreds of coals available for underfeed-stoker firing in this country. Additional tests could be made with much less expenditure of time and money and the experience gained in those reported herewith would be of great value in developing equipment by the use of which more accurate and more detailed information could be obtained. More data on the extremely interesting region in the walls of the burning lane should be secured by inserting probes along diagonals passing from the retort into the burning lane. This was not attempted in these tests because the importance of this region was not appreciated.

Despite its limitations, this investigation is not without immediately applicable results. It shows, for instance, that a change in the contour of the fire may have beneficial results in the uniformity of operation of the stoker, and that, in this installation, the use of the "short" fire leads to less severe treatment of the iron of the stoker. It also shows that what apparently is a thinner fire may actually have more fuel over the tuyères and so be more stable than a thicker fire, with its accompanying narrow burning lanes; and finally, it repeats the demonstration that the gases rising from the active portions of the bed may require additional air for combustion. Another result of importance in the design of stokers is the elucidation of the structure of the fuel bed obtained by these tests, which showed that the mechanism of heat flow into the coal in the retort is similar to that governing heat flow into a slot-type by-product coke oven. This makes available for the use of stoker designers a considerable amount of information concerning heat transfer which is directly applicable to their problem.

The principal facts brought out by this investigation may be summarized as follows:

1 The fuel beds of multiple-retort underfeed stokers exhibit a considerable degree of uniformity of structure relative to the axes of the burning lanes, the positions of which, however, are not fixed with respect to the center lines of the tuyère stacks, but may vary within limits. Fluctuations of temperatures and pressures from the averages at points fixed with respect to these axes are relatively small, but the deviations of gas composition are large, apparently because of the slow speed of diffusion by comparison with the rates of heat transfer and of the transmission of hydrostatic pressure in the system.

2 The fuel beds consist of burning lanes through which most of the air for combustion passes, separated by heaps of coal in the retorts through which very little air flows.

3 The coal in the retorts is retained between walls of coke, at the cooler boundary of which rapid carbonization of the coal occurs and through which heat for carbonization and ignition of the coal is conducted transversely across the stoker from the burning lanes.

4 The fuel has a component of flow transversely across the stoker from the retorts to the burning lanes approximately equal to that calculated from the burning rate and the dimensions of the fuel bed.

5 At the bottom of the burning lanes an extremely high-temperature coke is burned and gasified.

6 The temperature and pressure gradients in the main primary-air stream are determined by the rate of air flow, but the temperatures attained just above the tuyères depend also upon the fire contour.

7 The maximum temperatures observed, 2800 to 3000 F, appear close to the top of the bed, and are nearly the same in all series, regardless of the load, the fuel, or the fire contour.

Results of Tests

A summarized tabulation of the data on the conditions of operation in all the runs, averaged for each series, is given in Table 1. Series 1 and series 2 differ only in load; series 2 and 3 in excess air; and series 3 and 4 mainly in fire contour (refer to paragraph on "Stoker Operation"), although there was an increase in excess air in series 4 over series 3. Series 5, 6, and 7 were run at the same load and excess air as series 2 but with three different coals; series 5 with coal from Pocahontas No. 3 seam; series 6 with

TABLE 1 SUMMARY OF DATA ON CONDITIONS OF OPERATION

Series No.	1	2	3	4	5	6	7
Runs No.	21–44	45–63	64–71, 76	72–75 77–83	84–88	89–93	94–97
Dates	11/23/37 –12/17/37	1/19/38 –3/9/38	3/17/38 –3/24/38	3/25/38 –4/ 7/38	4/14/38 –4/15/38	4/20/38 –4/21/38	4/26/38
Coal	Lower Kittaning, run-of-mine				Pocahontas	Sized, Lower Kittaning	Mallory Gas
Bailey-meter readings, average:							
Steam	149	217	209	206	207	209	207
Air	152	219	236	238	211	211	215
Fire contour	Long	Long	Long	Short	Short	Medium	Long
Average temperatures, F:							
Steam, superheater outlet	618	652	644	662	651	655	660
Feedwater entering boiler	287	292	327	322	315	309	309
Flue gas, boiler outlet	550	619	609	607	597	608	597
Pressure and draft, in. water:							
Wind-box pressure	1.5	2.9	2.7	2.9	3.0	2.7	2.5
Furnace draft	0.12	0.15	0.30	0.21	0.12	0.14	0.11
Boiler-outlet draft	0.30	0.94	1.18	0.97	0.66	0.68	0.67
Excess air, per cent:							
Bailey-meter setting, pens together	37	44	37	49	42	40	37
Actual	40	45	54	72	45	41	42
Hourly rates, 1000 lb per hr:							
Steam flow	82.7	120.3	115.8	112.0	114.5	115.6	114.5
Coal burning (estimated)	8.0	12.2	11.8	11.5	11.3	11.9	11.5
Air flow (estimated)	116	185.5	190.3	208.5	176.4	174.0	173.5
Unit rates, lb per sq ft per hr:	Coal / Air	Coal / Air	Coal / Air	Coal / Air	Coal / Air	Coal / Air	Coal / Air
(a)	20.9	31.9	30.8	30.0	29.4	31.0	30.0
(b)	26.2 / 380	39.9 / 606	38.5 / 621	36.6 / 682	36.5 / 576	38.9 / 569	37.6 / 567
(c)	44.0 / 640	67.0 / 1020	64.9 / 1045	61.5 / 1145	61.5 / 970	65.4 / 957	63.2 / 954

(a) Referred to total projected area, 382 sq ft.
(b) Referred to actual area, omitting ashpit, 306 sq ft.
(c) Referred to air-admission area, 182 sq ft.

TABLE 2 COAL ANALYSES

Series No.	1	2	3	4	5	6	7
Coal	Eureka, run-of-mine				Pocahontas, nut and slack	Eureka, sized	Mallory
Supplier	Berwind-White Coal Mining Company				Pocahontas Fuel Company, Inc.	Berwind-White Company	Hunt's Point Gas Plant
Source:							
County and state	Cambria, Pa.				West Virginia	Cambria, Pa.	Logan, W. Va.
Seam	Lower Kittaning				Pocahontas No. 3	Lower Kittaning	Eagle
Proximate analysis, per cent:							
Moisture, as received	3.6	3.3	3.3	3.2	4.3	3.3	4.5
Volatile matter, dry	16.4	17.1	16.7	16.6	17.8	17.0	32.5
Fixed carbon, dry	77.1	75.5	75.2	76.4	76.1	75.2	63.2
Ash, dry	6.5	7.4	8.1	7.0	6.1	7.8	4.3
Sulphur, dry	1.13	1.43	1.46	1.38	0.60	1.36	0.57
Heating value:							
As received	14100	14080	13970	14100	14170	13940	14080
Swelling index in V.M. det., per cent	...	...	...	85	95	90	65
Ash-fusion temperatures, F:							
Initial	2560	2450	2610	2510	2550	2630	2650
Softening	2620	2510	2670	2570	2600	2680	2700
Size, per cent:							
On $1^1/_2$-in. round hole	0.4	1.1	0.8		0.4	0.9	0.4
Through $1^1/_2$-in. on 1-in.	1.6	3.4	1.6		2.9	3.5	2.6
Through 1-in. on $^3/_4$-in.	2.5	0.7	2.3		3.5	3.1	3.4
Through $^3/_4$-in. on $^1/_2$-in.	..	6.7	5.9		5.4	7.2	8.0
Through $^1/_2$-in. on $^3/_8$-in.	10.8[a]	..	6.2		5.4	8.1	8.1
Through $^3/_8$-in. on No. 4 square mesh	11.1	8.5[b]	9.9		10.4	16.6	17.1
Through No. 4 on No. 8	..	47.9[c]	17.9		18.1	24.0	18.5
Through No. 8 on No. 14	..	14.8[e]	18.9		18.4	16.2	13.8
Through No. 14 on No. 20	44.5[d]	5.1[f]	7.5		7.4	5.2	5.8
Through No. 20 on No. 60	17.1[g]	6.2[h]	19.1		19.4	10.7	11.7
Through No. 60 on No. 100	5.5[j]	1.6	1.4		2.5	0.3	1.8
Through No. 100	6.5	4.0	8.5		6.2	4.2	8.8

[a] Through $^3/_4$-in. on $^3/_8$-in.
[b] Through $^1/_2$-in. on $^1/_4$-in. square mesh.
[c] Through $^1/_4$-in. square on $^1/_8$-in. square.
[d] Through No. 4 on No. 20.
[e] Through $^1/_8$-in. square on No. 20.
[f] Through No. 20 on No. 30.
[g] Through No. 20 on No. 50.
[h] Through No. 30 on No. 60.
[j] Through No. 50 on No. 100.

Lower Kittaning coal, the same as that used in the first four series, except that it had been sized at the washing plant before shipment by the removal of fines passing a $^5/_{32}$-in. screen; and series 7 with a high-volatile-gas coal from the Hunt's Point Gas Plant of the Consolidated Edison Company. Reference to Table 2 and Fig. 1 shows that the "sized" Lower Kittaning coal used in series 6 had been so degraded during shipment that its size consist was not significantly different from the nut and slack ordinarily used. Thus, this series cannot be expected to show any effect which might be produced by a change in the sizing of the coal fired. No trouble was experienced due to smoke emission in series 7 using high-volatile coal. The flame in the furnace in this series was very dense and hugged the bed very closely so that it was difficult to see the fuel bed through it, but no secondary combustion was observed at the top of the first pass at any time with either a long or a short fire.

The values of steam flow and air flow in Table 1 are averages of the readings recorded on the boiler-room floor at 15-min intervals but, since the load was held very steady during the tests, slight error is expected on this account. Only the total steam readings were averaged; the readings of the meter on the west-side superheater outlet (refer to paragraph on "Test Apparatus") were generally 10 to 20 points lower during series 1 to 3, but differed very little from the totalizing meter in subsequent series. This is believed to be due to the somewhat more severe stratification which existed during the tests with the long fire than in those with short fires. Superheated-steam temperatures at the two outlets differed by amounts up to 30 F, the east side usually being higher. This behavior was irregular but appeared somewhat less frequently with short than with long fires. In calculating the average steam temperatures, a straight arithmetic average of all the readings in each series was taken, as the weighted average differed from this by only 1 to 2 F. The flue-gas temperature leaving the boiler is not accurate, as it was based on the reading of the Bailey-meter gas thermometer which was not checked.

Of the quantities given under the heading "Unit Rates," the first fuel-burning rate, referred to the projected area of the stoker and ashpit, 382 sq ft, is the quantity usually cited in comparisons of stokers. The others are based on measurements of the various elements of the stoker, taken by the test crew, which resulted in

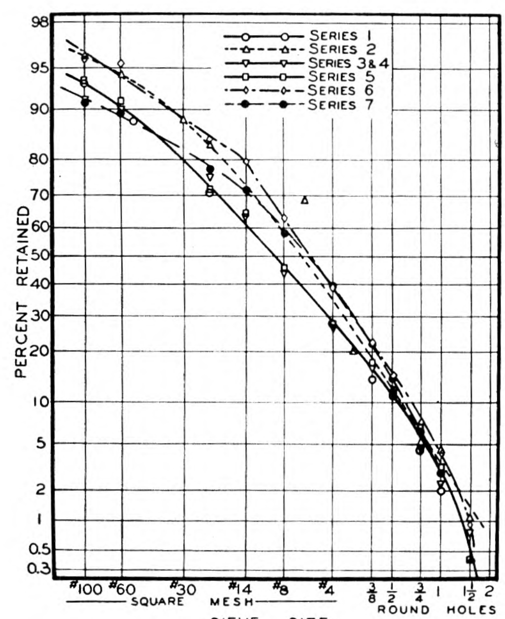


FIG. 1 SIZE DISTRIBUTION OF COALS PLOTTED ON LOG PROBABILITY COORDINATES

the following areas: Total stoker area, omitting ashpit, 306 sq ft; projected area of stoker, omitting ashpit, 274 sq ft; and total air-admission area, including area of tuyère stacks, projected area of side-wall tuyères, and extension grate, 182 sq ft. Fuel-burning and air-flow rates, based on these areas, are useful in the consideration of the significance of the measurements.

As the results of the tests include data on temperatures, pressures, and gas compositions at each of ten or more levels in a total of 79 test runs, it is obviously out of the question to present the complete experimental data in this paper. For this reason, only samples of the tabulated results will be given here; complete tabulations of the data have been prepared and deposited in the archives of the Society and may also be obtained, either as photo-

TABLE 3 TEST RESULTS; RUN NO. 72, NO. 2 TUYÈRE, EAST

Distance above tuyère, in.	Temp, F	Corrected temperatures, F	Gas-pressure differential, in. water	Gas analysis, per cent CO_2	O_2	CO	H_2	CH_4
—1	385	...	2.05	...	...	...	...	...
— 1/2	450	...	1.0	...	...	...	...	...
0	450	...	0.44	...	...	...	...	...
1/2	1055	1910	0.44	9.8	10.2	3.5	1.4	...
1	1400	2030	0.43	3.9	16.0	1.3	...	...
1 1/2	1900	2080	0.47	7.6	12.8	...	...	...
2	2295	2500	0.55	16.1	4.6	...	...	...
2 1/2	2490	...	0.60	16.0	3.2	...	...	...
3	2505	...	0.59	11.9	0.4	9.5	2.4	1.5
3 1/2	2475	...	0.90	14.6	0.4	7.6	...	...
4	2535	...	0.80	11.2	0.0	7.0	4.2	0.9
4 1/2	2550	...	0.90	11.9	4.2	3.5	0.6	0.8
5	2580	...	1.1	11.9	0.2	7.6	...	...
6	2655	...	1.25	12.2	0.5	7.2	1.3	0.5
7	2655	...	1.25	12.4	0.6	5.5	0.8	0.4
8	2550	...	1.55	9.2	0.8	8.4	8.8	1.8
9	2750	...	2.1	11.6	0.8	6.8	1.1	0.9
10	2700	...	2.9	13.9	5.7	0.0	...	...
11	2780	...	3.0	...	...	...	...	...
12	2685	...	2.7	...	...	...	...	...
12[c]	2445[a]	...	3.25	16.0	0.1	2.6	...	...
13	2475[b]	...	2.9	15.4	1.9	0.9	...	...
14	2625	...	2.8	17.9	0.0	1.6	0.1	0.1

[a] Momentary temperatures as high as 3025 F.
[b] Probe seen in hole about 2 in. below top of burning lane.
[c] Taken during withdrawal of probe.

bed being built up between parts (refer to paragraph on "Testing Procedure").

Parentheses around corrected temperatures indicate that these readings are unreliable because the observed rate of temperature rise was too great.

Discussion and Interpretation

Structure of the Fuel Bed. In order to facilitate visualizing the results of the tests, Figs. 2 to 6 have been prepared, showing, on cross sections of the fuel bed, contours of equal temperature, constant gas composition, and equal pressure drop. It must be borne in mind that these figures do not represent either averages or a steady state; they are, rather, the authors' best guess, based on plots of the observations, as to what might be observed, if it were possible to take simultaneous readings throughout the region tested at some instant. Fig. 2 shows isothermals on transverse cross sections of the bed at the three positions tested, taken in the planes, differing slightly from the vertical (refer to Fig. 13), in which the probes

TABLE 4 TEST RESULTS; RUN NO. 61, NO. 2 RETORT, WEST

Distance above ram, in.	Time after stopping stoker, min	Maximum temperature observed, F	Time after reaching position, min	Corrected temperature, F	Gas-pressure differential, in. water	Gas analysis, per cent CO_2	O_2	CO	H_2	CH
0	..	95	...	..	2.9	..	...	...	..	..
5	..	95	...	..	2.2	..	...	...	..	..
10	..	95	...	..	2.2	..	...	...	..	..
15	..	95	...	..	2.5	..	...	...	..	..
17.5	..	95	...	..	2.6	..	...	..	..	..
18.5	..	95	...	..	2.6	..	...	...	..	..
19.5	..	95	...	..	2.65	..	...	...	..	..
20.5	..	95	...	..	2.65	..	...	...	..	..
21.5	..	95	...	..	2.7	..	...	...	..	..
22.5	..	95	...	..	2.8	..	...	...	..	..
23.5	..	95	...	..	2.8	..	...	...	..	..
24.5	7.0	570	1.0	1250	2.8	5.2	6.2	4.2	10.2	12.2
..	..	..	1.5	1460	...	..	...	...	..	..
25.5	10.0	1195	1.0	1860	2.9	5.2	5.0	5.7	12.7	10.7
..	..	..	2.0	1990	...	..	...	...	..	..
				Part 2						
23.5	2.5	250	...	..	3.1	3.6	7.9	...	..	..
24.5	5.5	875	1.0	1500	2.95	5.0	2.5	5.2	14.6	14.4
..	..	..	2.0	1665	...	..	...	...	..	..
25.5	8.5	1835	1.0	(2390)	2.8	5.3	2.0	6.8	17.6	15.1
..	..	..	2.0	(2615)	...	..	...	...	..	..
26.5	11.0	2220	Steady		2.8	8.4	0.9	6.6	..	..
27.5	..	2280	Steady		2.85	14.1	4.1	0.0	1.3	1.2
28.5[a]	..	2325	Steady		3.1	10.1	0.6	7.1	12.3	6.5

[a] Probe 3 in. above fuel bed.

prints or as microfilm, from The American Documentation Institute, Washington, D. C. Details of the procedure and discussions of the accuracy and precision of the results are given in later paragraphs. The results obtained are exemplified by Tables 3 and 4.

The data from a typical run over the tuyère stack are given in Table 3, in which the first column gives the depth of penetration into the fuel bed; the second the average temperature if a steady reading was reached, otherwise, the maximum observed; column three, the corrected temperature, if a steady reading was not attained, when the correction referred to in the section on "Supplementary Data" was applied; the fourth column shows the pressure drop of the combustion gases through the stoker and fuel bed to that point; and columns 5 to 9 give the percentage composition, dry, of the gas sample taken at that point. In Table 4, which presents the data from a typical retort run, the first column gives the distance above the secondary ram; the second, the time after stopping the stoker; and the third, the uncorrected temperature at that time; while the fourth and fifth columns give the time and the corrected temperature at intervals during the period the probe was held at that point. The sixth column gives the pressure differential, and the remaining columns the analysis of the dry gas. The retort runs were completed in two parts, the fuel were advanced through the bed, at a load of 120,000 lb of steam per hr with 45 per cent excess air and a long fire (series 2). The sections show a portion of the bed, consisting of one full retort and tuyère stack and one half of the next adjacent elements; the lines of insertion of the probes are indicated by dot-and-dash lines, with scales showing the distance above the stoker iron in inches. Position No. 1 is near the head of the stoker, while No. 3 is near the extension grate (refer to paragraph on "Equipment and Construction"). The contours are nearly vertical at the sides of the retorts and the temperature increases very rapidly across a narrow region above the walls of the retorts in the direction of approach to the center line of the tuyère stack. The coal in the retort does not rise above room temperature until it approaches the walls of the retort or a thin skin on top, although the width and height of the unheated heap of coal become less as the distance from the head end of the stoker increases. The region of very high temperatures above the tuyères is coincident with the burning lanes which are so prominent a feature of stoker fires. It is evident that the principal direction of heat flow into raw coal normal to the contours is horizontally across the stoker from each burning lane toward the center line of the adjacent retort. It should also be noted that coal in the retort, at the middle of the stoker longitudinally, may rise 23 in. above the floor of the

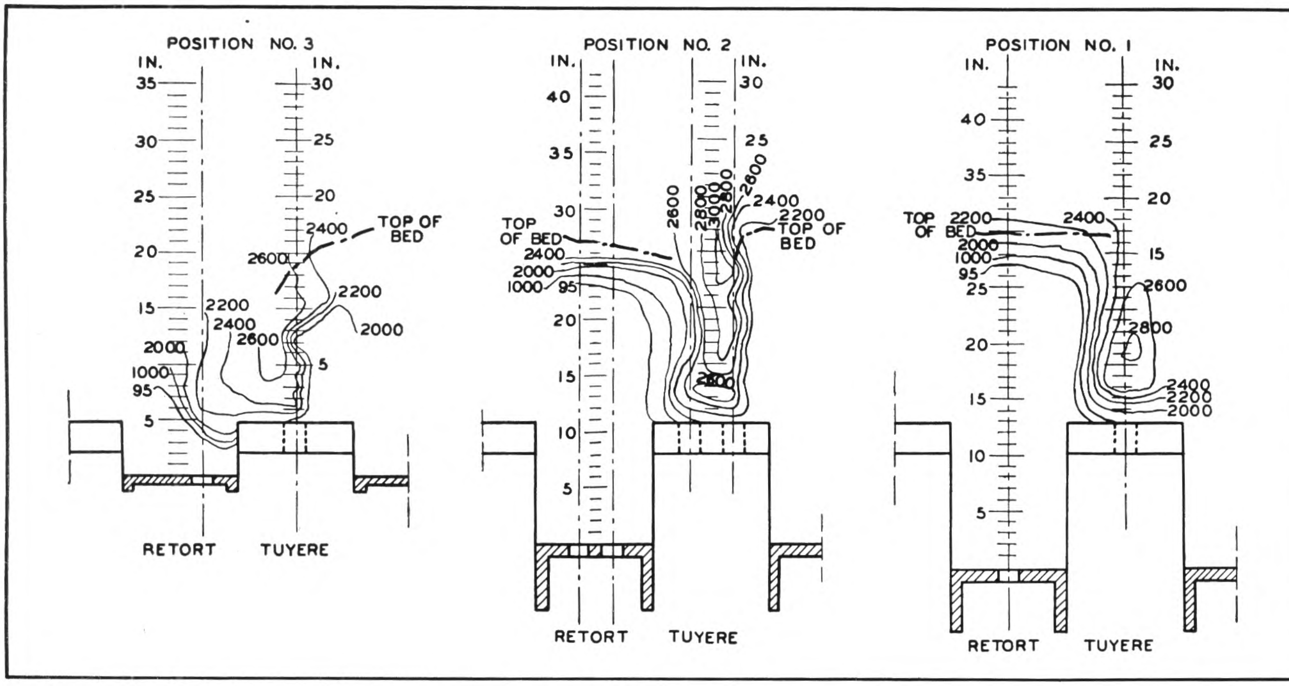


FIG. 2 TEMPERATURE CONTOURS IN FUEL BED

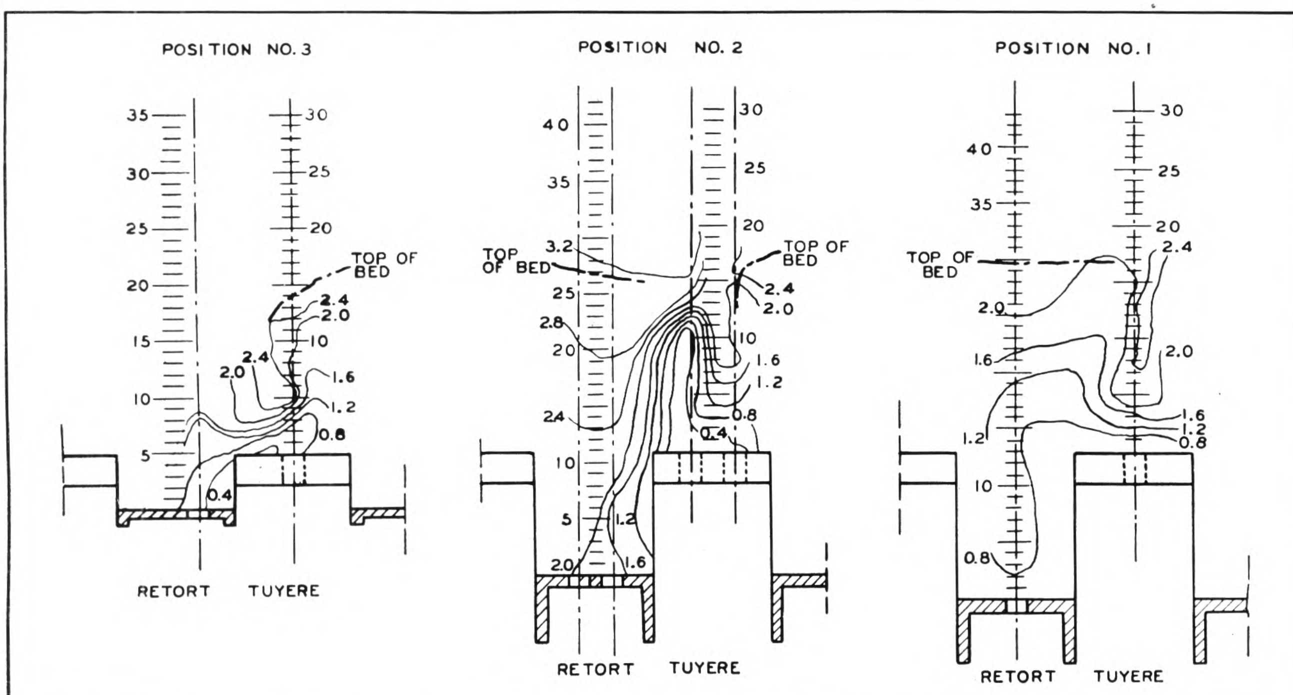


FIG. 3 PRESSURE CONTOURS IN FUEL BED

retort, or 12 in. above the level of the tuyères, without being heated above room temperature, even though at the same level less than 12 in. away, across the stoker, the temperature in the burning lane may have reached 3000 F.

Fig. 3 shows contours of equal drop in gas pressure on the same cross sections and under the same conditions of firing. These contours suggest that the principal flow of air is upward along the burning lanes and that there is comparatively little flow through the retort. In fact, the nearly vertical course of the contours in the walls of the burning lanes indicates that these walls may be almost completely impervious to air flow. That there is comparatively slight air flow up through the retorts is shown by the low rate of pressure drop through this rather densely packed heap.

Fig. 4 shows lines of constant oxygen, and Fig. 5 lines of constant carbon monoxide and of constant hydrogen-plus-methane concentrations. The air in the retort is not consumed until it approaches the region of very rapid temperature rise shown in

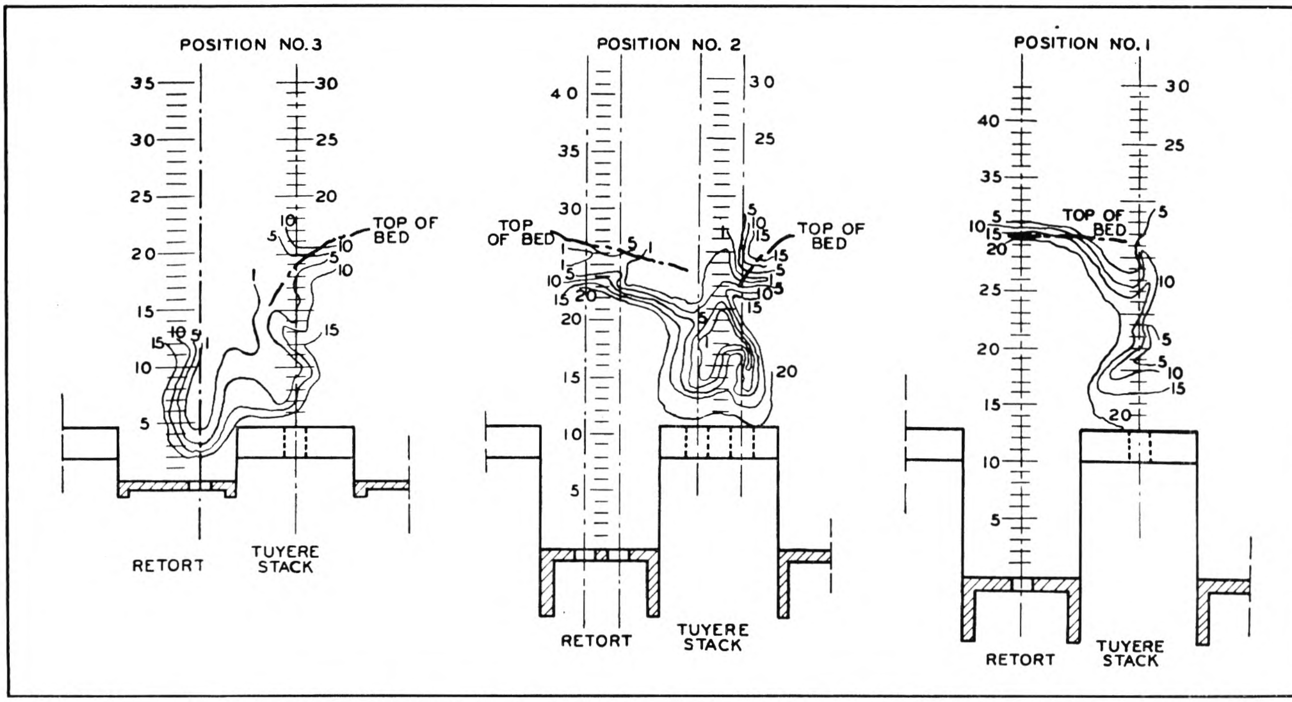


FIG. 4 CONTOURS OF EQUAL OXYGEN CONCENTRATION

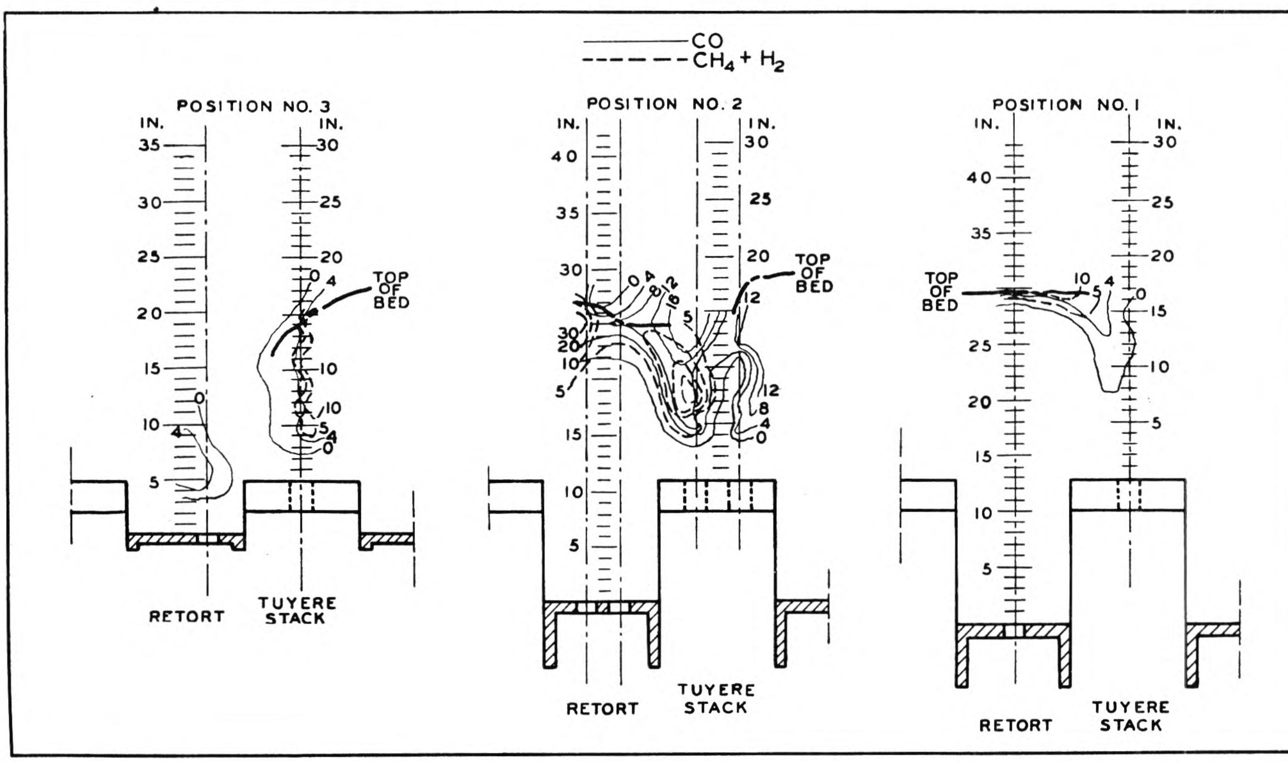


FIG. 5 CONTOURS OF CONSTANT CONCENTRATIONS OF CARBON MONOXIDE AND OF OTHER COMBUSTIBLE GASES

Fig. 2. In the burning lane, the oxygen penetrates further into the bed, the closer the center line of the lane is approached; along the walls of the lane it decreases practically to zero at only 3 in. above the tuyères while, at the center of the lane, the air may still contain 5 per cent of oxygen 6 in. above the tuyères. This result accords with the data on the pressure drop, which indicated that the gas velocity varies greatly in a section across the burning lane, from very small values at the walls of the lane to very large values at the center. It is also to be noted that the oxygen concentration appears to increase again toward the top of the lane. This effect is quite marked, and is accompanied by a decrease in the concentration of the combustible constituents, as shown in Fig. 5, and usually by an increase in carbon dioxide, although the latter may be masked by excessive dilution by the fresh air entering at

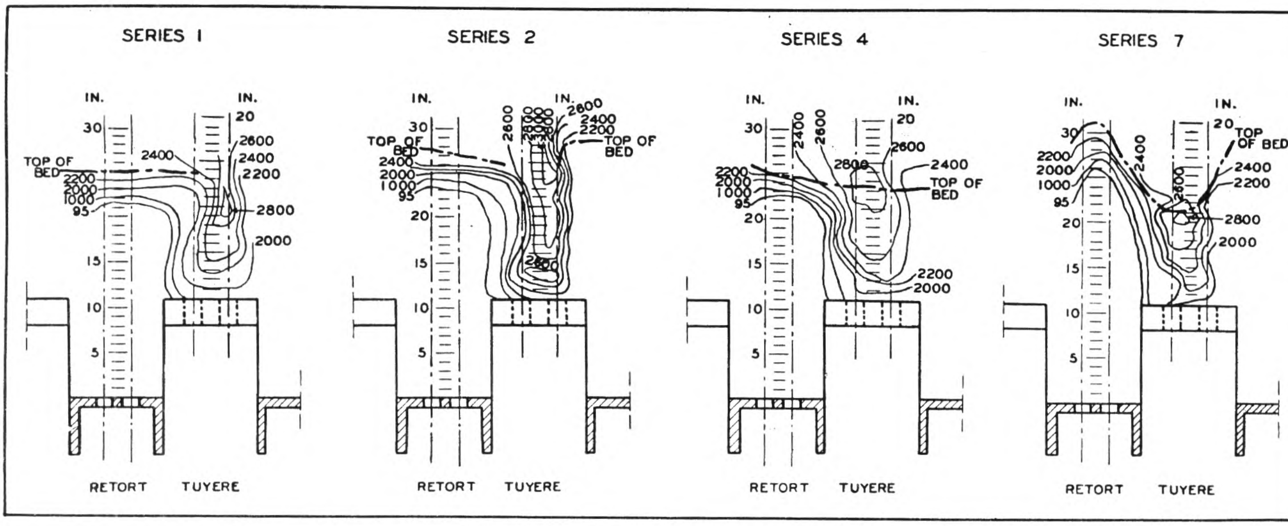


FIG. 6 TEMPERATURE CONTOURS FOR DIFFERENT COALS AND CONDITIONS OF OPERATION

this point. The authors have been unable to agree on an interpretation of this secondary increase in oxygen.

The greatest concentrations of combustible gases appear in the walls of the burning lanes at about the same level as the maximum temperatures in the lanes themselves. In this region, the sum of methane plus hydrogen may reach values above 25 per cent. It can be seen by reference to Fig. 2, that the temperature in this region ranges from 1500 to 2200 F, and from Fig. 3, that it is just this region which appears to be most nearly impervious to gas flow. All these facts point to the conclusion that this is the region in which carbonization of the fuel is occurring at the most rapid rate.

Comparison of charts similar to those just referred to for different parts of the bed, and for different conditions of operation, shows quantitative differences but no qualitative change in the pattern, as indicated in Fig. 6, the isothermals at No. 2 position for a number of different conditions of operation. All these sections show the same pattern of nearly vertical isotherms in the walls of the burning lanes connecting regions low in the burning lanes with regions high in the retorts. Indeed, almost the same maximum temperatures at almost the same levels were observed in all the runs. Plots of the other variables investigated show similar uniformity under different conditions and so are not presented here.

These charts indicate that the structure of the fuel beds of underfeed stokers differs materially from the conventional picture, which consists of a bed of horizontal strata, the bottom one being green coal and the top one being freely burning coke with all gradations occurring in between. These results show that, actually, the stratification is mainly along vertical planes running from the head of the stoker to the end of the underfeed section along the walls of the burning lanes. A section through the fuel bed perpendicular to these planes, that is, across the stoker, shows green coal in the center of the retort extending almost to the top of the bed, then coal being heated, coked, and ignited as the burning lane is approached, with free burning of prepared fuel in the burning lane approximately along the center line of the tuyère stack. The flow of heat proceeds mainly in a horizontal direction, the coke layer forming first as a thin skin on the wall of the burning lane at the head end of the stoker and progressing across the retort toward its center line with increasing distance from the front wall. Thus the retort of such a stoker is similar to a by-

TABLE 5 AVERAGE TEMPERATURES AT NO. 2 TUYÈRE

Series	1		2		3		4
Distance above tuyères, in.	Temp, F		Temp, F		Temp, F		Temp, F
0	1830		..		1660		..
0.5	1920		2190		1885		2030
1.0	1960		2325		1715		1850
1.5	2165		2420		2280		2075
2.0	2050		2475		2261		2065
2.5	2030		2570		2310		2155
3.0	2140		2620		2465		2410
σ	231		203				
	a	*b*	*a*	*b*	(*a*)	(*b*)[2]	
3.5	2225	2140	2600	2545	2490	2325	2465
4.0	2305	2105	2620	2385	2730	2400	2480
4.5	2390	2120	2670	2320	2690	2340	2550
5.0	2445	2175	2725	2250	2700		2585
5.5	2430	2235	2695	2235		2265	
6.0	2460	2120	2780	2265		2235	2650
6.5	2505	2060	2810	2090			
7.0	2565	2090	2770	2200		2295	2700
7.5	2590	2020	2880	2255			
8.0	2620	2020		2250		2340	2650
8.5	2730	1960		2310			
9.0	2815	1850		2380		2685	2695
9.5	2825	1740		2415			
10.0	2820	1695					2755
10.5		1695					
11.0		1710					2795
11.5		1725					
12							2755
13							2680
14							2700
σ	133	122	41	195	249[1]		167[3]

[1] σ calculated for series 3 and 3*a* from 0.5 in. to 5 in.
[2] One run only; σ not calculated.
[3] σ calculated for series 4 from 0.5 in. to 14 in.

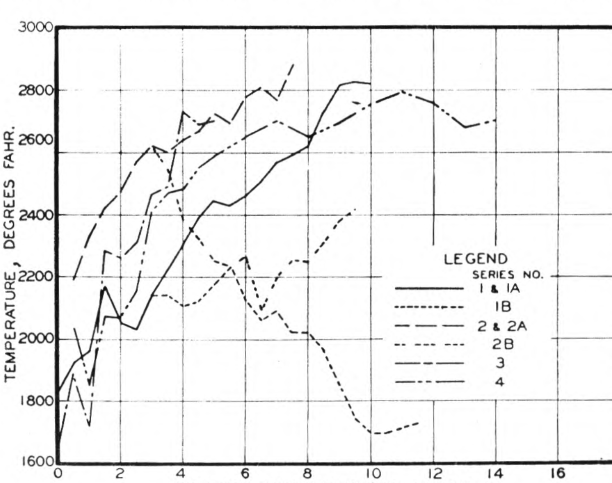


FIG. 7 AVERAGE TEMPERATURES AT NO. 2 TUYÈRE POSITION

product coke oven except that the confining walls are absent and combustion takes place in direct contact with the outside layer of coke; the laws (6) which govern heat flow in a by-product oven may be expected to govern the flow of heat into a stoker retort.

Carbonized fuel breaks off from the coke walls because of the formation of shrinkage cracks and the agitating action of the secondary rams, and falls down into the burning lanes where the level of the fuel is considerably lower than it is in the retorts. These lanes may cover only a portion of the width of the tuyère stack; they consist of channels with more or less irregular coke walls, not necessarily continuous along the length of the stoker; and they generally have a more or less densely packed continuous fuel bed with, perhaps, some ash at the bottom, in the lower part of the channel. The primary air flows up through these lanes, burning and gasifying the bed of coke through which it passes. The air velocity varies within wide limits in the burning lane, depending upon the density of the fuel bed through which each filament of flow passes, and the distance required to consume the oxygen in any filament increases with the local flow velocity.

While this description is based upon measurements made on only one stoker using but a limited number of coals, it is probable that, qualitatively at least, it applies quite generally to all stokers of this type. It is evident that the cases will be quite rare in which the air flow from the tuyère stacks spreads with equal intensity through all sections of the coal, both above the tuyères and in the retort; only in those cases, however, will there be any marked deviation from the structure outlined, in which the stratification is along vertical planes and the heat flow into the raw coal is horizontal. It appears also that the high combustion rates, compared to the rates of ignition observed in pure underfeed burning, which may be attained in multiple-retort stokers, are due to the fact that there is no air flow through the planes of ignition to remove the heat conducted into this region from the hotter parts of the fire, and to the possibility that the aggregate of all the zones of ignition, that is, the sum of the areas of the burning-lane walls may be considerably greater than the projected area of the stoker.

Temperatures. The average temperatures at No. 2 tuyère position for the first four series of runs are given in Table 5 and in Fig. 7. The precision (1) of the measurements within each series is indicated by the magnitudes of the standard deviations σ given at the bottom of each column in Table 5.

These standard deviations indicate that the procedure of averaging the runs by series is probably justified, since the probable error, equal for large numbers of observations to $0.6745\ \sigma$, is never greater than about 170 F, approximately the order of accuracy which was expected before the tests were made, and is usually much less. It will be observed that the precision is often greater than the accuracy of the measurements (2), estimated as being about 90 F (refer to section on "Supplementary Data"). Table 6, the average differences between various pairs of groups of runs, shows that these differences are statistically significant (11), excepting for the upper portion of the traverses in series 2*a* and 3, since the probability that such differences could occur by chance is always less than 0.05, and usually less than 0.01.

These data indicate that the magnitude of the temperature gradient immediately above the tuyères is proportional to the air-flow rate, as shown in Table 7. The data of tests with special coals are not extensive enough to justify statistical analysis but they indicate that this conclusion holds true also for series 5 to 7 as well. On the other hand, the temperatures attained at $^1/_2$ in. and 3 in. above the tuyères depend not only upon this gradient, but also upon the condition of the fire, as can be seen by comparison of the data for series 2, 3, and 4, all of which have nearly the same air-flow rate. In series 2, which was run at an excess air of 45 per cent with a long fire, the temperature rose to about 2600 F at 3 in. above the tuyères, while the temperature $^1/_2$ in. above the tuyères was about 2200 F. In series 3, with a somewhat greater air flow and correspondingly greater gradient, the temperature 3 in. above the tuyères was only 2460 F, showing the effect of an increase of excess air to 54 per cent and the correspondingly more open fire. In series 4, however, in which the short fire was used, the temperature 3 in. above the tuyères dropped still further to 2400 F.

In the upper part of the bed, it is necessary to differentiate between traverses which passed up through the burning lanes, denoted by the suffix *a* in Tables 5 and 6, and those which were in the walls of the lane, denoted by the suffix *b*. Considering first the *a* runs, it will be observed that the maximum temperature attained in all of those with the long fire was about the same. In series 1, this maximum of above 2800 F was reached at about 9 in. above the tuyères, while in series 2 and 3 the same value was reached somewhat lower, at about 6 in. above the tuyères, even though series 3 was run with high excess air. Data for distances above 5 in. are not given in Table 5 for series 3*a*, as not enough runs were available to justify statistical analysis. In series 4, however, with the short fire, the temperatures were substantially lower throughout the upper part of the bed, and the maximum of just under 2800 F was reached only at 11 in. above the tuyères. The data of series 5 and 6 in which short fires were carried with special coals confirm this trend.

At the top of the bed, when the probe rose into the gas space, the temperature indication fell off quite sharply. This is shown in the averages for series 4, the only one in which a large enough

TABLE 6 SIGNIFICANCE OF DIFFERENCES BETWEEN TEMPERATURES IN DIFFERENT SERIES

Series	Range, in.	Average difference of means	P	
2-1	0.5–3	389	<0.01	Significant
2*a*-1*a*	3.5–16	122	<0.05	Significant
2*b*-1*b*	3.5–11.5	320	<0.01	Significant
4-2*a*	0.5–9	224	<0.01	Significant
3-2*a*	0.5–3	263	<0.01	Significant
3-2*a*	3.5–9	15	<0.6	Not significant
3-1*a*	1.5–5	273	<0.01	Significant
4-1*a*	2–8	148	<0.01	Significant

TABLE 7 TEMPERATURES AND MAXIMUM TEMPERATURE GRADIENTS NEAR THE TUYÈRES AT NO. 2 POSITION

Series	Air rate through tuyères, lb per sq ft per hr	Temperatures at $^1/_2$ in., F	Temperatures at 3 in., F	Maximum temperature gradient, deg per in.	Ratio, Temperature gradient / Air-flow rate
1	640	1915	2140	195	0.304
2	1020	2185	2620	235	0.230
3	1045	1885	2465	395	0.378
4	1145	2030	2410	350	0.306
Average					0.304 ± 0.023

TABLE 8 TEMPERATURE DROP AT TOP OF BURNING LANE
(Part of doghouse log; run No. 94; No. 2 tuyère, west)

Time, p.m.	Position, in.	Temperature mv	Temperature F	Bed differential, in. water	Sample No.	Notes
9:05	27	14.2		1.45		
06		14.8	2625		49	
07	28	15.2		1.45		
08		14.9	2640		50	
09	29	15.2		2.35		Probe pushed hard
10		15.4				
11		15.4	2715		51	
12	30	15.7		2.70		
13		15.9	2795		52	
14	31	16.0	2810	2.68		
15		14.9	2640		53	
16	32	13.8		3.00		
17		13.1	2370		54	
18		13.1				
19	33	13.3		2.68		
20		13.8	2475		55	
21	34	14.4		2.70		
22		14.2				
23		14.4	2565		56	
24	35	14.5		2.75		
25		14.5	2580		1	Probe reported 6 in. out of fuel in burning lane

number of probes penetrated the bed to justify carrying the averages that far. This effect was observed, however, in every individual run in which the probe was seen from the rear door of the furnace. A good example is given by run No. 94, series 7, of which the readings for the last 8 in. are given in Table 8. The observation from the boiler-room floor that the probe extended about 6 in. above the level of the fuel bed in the burning lane at level 35 shows that the beginning of the precipitous drop in temperature at level 31 was close to the top of the bed.

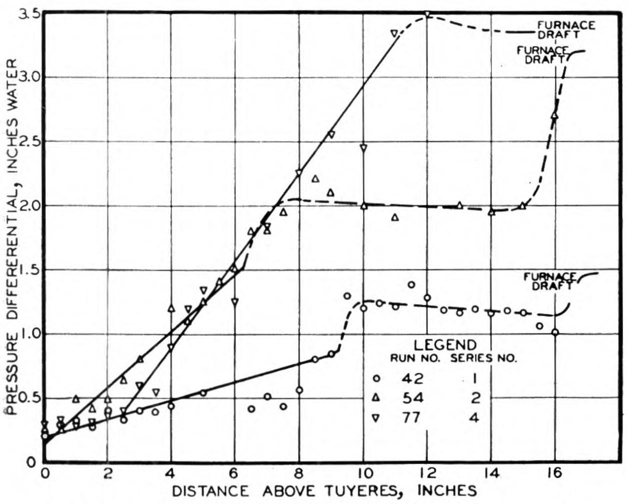


FIG. 8 PRESSURE LOSS AS A FUNCTION OF DISTANCE ABOVE TUYÈRES

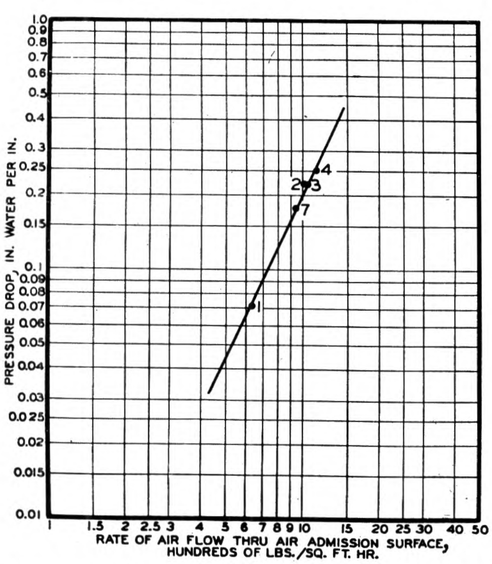


FIG. 9 PRESSURE GRADIENT AS A FUNCTION OF AIR-FLOW RATE

In runs in which the probe was raised, not through the burning lane, but through its walls, the temperature dropped from a maximum in the neighborhood of 3 in. above the tuyères, up to which point it had followed the same course as in runs in the lane, and fell off continuously to as low as 1700 F at 10 in. above the tuyères. The negative gradient in this region appeared to be independent of the load and excess air, while the length of the region of decreasing temperature was quite unpredictable and may have depended only upon the distance from the line of traverse to the center of the burning lane. The fact that these traverses differed from those just described, in being off the burning lane, was substantiated by the observations of the other quantities, gas pressure and analysis, and also by direct observation in many runs of the position of the probe when it had penetrated the fuel bed and was visible from the boiler-room floor. In every case, the region of declining temperature was terminated by an abrupt transition to a region in which temperatures of the same magnitude as those prevailing in the burning lane at these levels were observed. This transition was accompanied by characteristic changes in gas analysis and pressure (refer to section "Pressure Differential") and was often coupled with a momentary great increase in the resistance to motion of the probe through the bed, followed by complete freedom of motion. This phenomenon was interpreted as meaning that at this point the probe broke through the wall and entered the burning lane or emerged from the top of the bed.

Pressure Differential. Plotting pressure differential against distance above the tuyères leads to quite different results for runs in the burning lanes and those out of the lanes. The results of the runs in the burning lanes are readily interpreted, as shown by the examples, from runs at No. 2 position, Fig. 8. The pressure drop is linearly related to the distance above the tuyères beyond a small distance of the order of 1 to $1^1/_2$ in. and extending to 8 to 12 in. above the tuyères. While the absolute values of pressure drop to any point do not agree very well in different tests under the same conditions, the values of the pressure gradients do. These are plotted, to logarithmic scales, against the air-flow rates for the different series in Fig. 9, where it will be observed they fall on a reasonably straight line with a slope of 2.17. The values of air-flow rate used in this calculation are based on the indication of the air-flow pen of the boiler meter, and are referred to the total air-admission surface of the stoker. This calculation is admittedly a rather unsatisfactory approximation; the errors in it arise from two main sources (*a*) the air-flow meter, which is not a precision instrument at best, measures the air flow through the boiler, not that through the stoker, (*b*) there is no assurance that the air flow through different sections of the air-admission surface of the stoker is uniform. No correction has been made for air admitted through the front-wall secondary-air ports.

In spite of these qualifications, the data show that the relation given in Fig. 9 is significant. The result may be compared with the formula given by Diepschlag (8) for the pressure loss through beds of spheres. To obtain the pressure drop observed, the fuel bed would have to be equivalent to a bed of spheres about $^1/_4$ in. diam. Diepschlag's measurements, however, were made using cold air. Applying the corrections for temperature indicated by Carman's analysis (7) the size of the particles would be increased to about 0.63 in. to produce the pressure gradient of 0.196 in. water per in. of fuel-bed depth at 1000 lb air per sq ft per hr, given by Fig. 9. The slope of the line in Fig. 9 is greater than that found by Diepschlag and by Arbatsky (3), in tests using cold air, who obtained values of the exponent of 1.6 to 1.7. This discrepancy, as well as the small particle size indicated by the calculations, may be ascribed to the errors previously noted, but it might also be due to the difference in temperatures since, in a test on a chain-grate stoker burning anthracite, Arbatsky observed a value of the exponent of 2.1 for small values of air-flow rate.

Certain other significant conclusions may be drawn from these data. The linear portion of the curves in Fig. 8 for runs in series 1 to 3 are always terminated by a sharp break upward, followed by a practically flat portion, which is not however at a pressure drop corresponding to the furnace draft but from 0.3 to 0.5 in. water less. It has been shown by Hirst (15) in a discussion of the principles of coal cleaning by pneumatic tables that the maximum pressure gradient which can be attained in a porous bed of broken solids is that equivalent to the weight of the bed per unit thickness. Above this gradient, which for coal is about 0.6 in. water per in. of bed and is nearly independent of particle size, the bed becomes disrupted. The sharp break referred to, which appears in the data for series 1 to 3, always involves a gradient materially exceeding this limiting value, so it is evident that a continuous fuel bed does not exist above it. It appears at levels of from 8 to 10 in. above the tuyères, which may be 6 to 8 in. below the level of the top of the fuel bed in the retorts. In series 4, however, when the short fire was used, no break appears. The linear portion extends to a distance of 12 in. above the tuyères, at which level the pressure has dropped very nearly to that in the furnace, showing that, in spite of the thin fuel bed of the short fire, there is more fuel over the tuyères in this fire than in the long one.

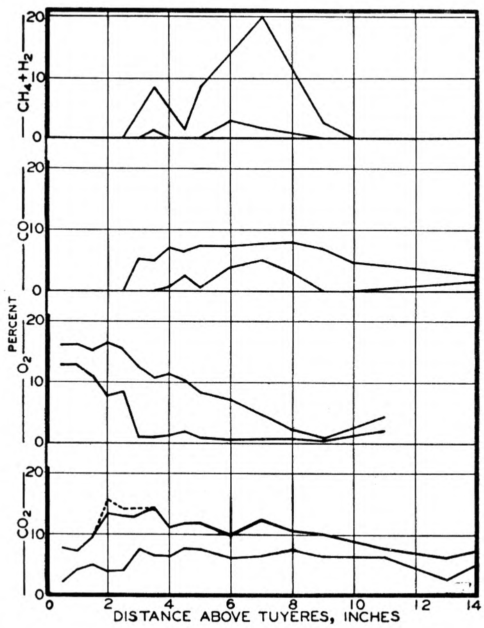


FIG. 10 DISTRIBUTION OF GAS ANALYSES AT NO. 2 TUYÈRE POSITION

The pressure gradients in the burning lanes at other positions show that in series 1, at low load, the air-flow rate per unit area is much greater at No. 1 position, at the neck of the stoker, and somewhat less at No. 3 position, just above the extension grates, than it is at No. 2. In series 2, the same effect was observed but to a much less marked degree, while in series 3 not enough data are available to permit a comparison. In series 4, the pressure gradients and, hence the rates of air flow per unit area, are very nearly the same at all three positions.

The pressure drops in runs in which the probe did not pass up through the burning lane show none of the regularity just described. They usually show a short section of not over 3 to 4 in. in which the gradient is somewhat less than that observed in the lane, followed by a portion, corresponding in position to the region of decreasing temperature mentioned in the previous paragraph, where the pressure remains very nearly constant. This region is believed to be in close proximity to, or in the interior of a, relatively impervious wall, since the absence of pressure gradient indicates negligible air flow in the direction of motion of the probe. The presence of such a wall can, of course, be accounted for by the presence of unbroken coke or of a plastic mass of coal undergoing carbonization. At the top of this region, in the same place that the abrupt increase of temperature was found, the pressure drops abruptly to a value below the furnace pressure. In the retort runs, the same values of pressure just above the top of the bed, from 0.05 to 0.15 in. of water below the furnace pressure, were observed. These results indicate that there is little flow into the furnace from the space immediately above the retorts, since, if there is no pressure drop due to flow upward from this level, a lower pressure than that observed in the furnace would be measured here, because the furnace-draft connection used for the operating meters is at a level about 10 ft above the fuel bed. Thus, at the level of the bed, the chimney effect of a 10-ft column of hot gas would be added to the furnace draft observed with the regular operating furnace draft gage.

Gas Composition. The gas compositions show much less regularity than the quantities just considered and their analysis does not lead to the type of quantitative results obtained in the consideration of temperatures and pressure drops. This is probably due to the fact that the gas analysis is much more subject to transient variations than are the other two quantities, since diffusion in the gas stream is a much slower process than heat transfer by radiation at the temperatures under consideration, or than the mechanical transmission of pressure through the gas which takes place with the speed of sound. For the same reason, it would obviously be necessary for the sampling point to reach exactly equivalent positions with respect to the filaments of flow and the proximity of surfaces of burning fuel in different runs to attain a high degree of precision of the measurements. Since the fuel bed, by its very nature, represents a single structure only in a statistical fashion, it is obvious that only a statistical approach to an average picture of the gas compositions at various points in the bed is available. The average values at different points might be more closely approximated by samples taken from each position over rather long periods of time but, in view of the comparatively short life of the probes in the fuel bed, it was not practicable to take such samples.

The limits, within which one half of the analyses taken at various levels at No. 2 position in all the series lie, are shown in Fig. 10. This figure was constructed by plotting all the data and then passing the lines through the groups of points at each level in such a way that one quarter of the points would be above the upper line, and one quarter below the lower one. These curves enclose the region within which the analysis of a sample taken at that position would probably lie.

The dispersion of these data is so great that significant differences between the results for the different series of runs cannot be found, but the general trends for all series can be observed in Fig. 10. The oxygen concentration falls off more or less rapidly to small values at distances 5 to 6 in. above the tuyères, remains at this low level up to about 10 in. above the tuyères, and then begins to increase again. This behavior is observable, not only in this statistical representation of the data, but also in each individual run. Carbon dioxide increases, more or less as a mirror image of oxygen concentration, to a maximum at 2 to 5 in. above the tuyères and then decreases, being replaced by increasing concentrations of carbon monoxide. Hydrogen and methane begin to appear at distances of 3 to 6 in. above the tuyères, but never reach very high concentrations in the tests in the burning lanes. In runs out of the lanes, the hydrogen and methane may reach very high values, a typical composition from a point 7 in. above the tuyères (run No. 55) being CO_2, 8.2; O_2, 1.2; CO, 7.3; H_2, 15.1; and CH_4, 4.8. The high ratio of hydrogen to methane, which is typical of the data from this region, indicates that these gases arise either from the later stages of the process of carbonization of the coal (18), or from the cracking of methane. Gas analyses in the retort, on the other hand, usually show a preponderance of methane over hydrogen, which is characteristic of the earlier stages of carbonization.

At levels in the bed above those where methane and hydrogen first appear, diluting the gas stream, the combustible gases burn out. At first, the water-gas reaction appears to account for most of the changes in analysis, as the hydrogen and carbon-dioxide concentrations decrease with a corresponding increase in carbon monoxide and but little change in methane. Later, however, methane and hydrogen disappear at parallel rates, but carbon monoxide is consumed much less rapidly, invariably being the last combustible gas to vanish. In the initial gasification region, below 6 in. above the tuyères, the fuel is almost pure carbon, that is, a very high-temperature coke, as shown by the fact that the sum of the concentrations of oxygen and carbon dioxide, when the monoxide is absent, always amounts to more than 20 per cent and, when the monoxide is present, frequently reaches 20 per cent + $^1/_2$ (CO), corresponding to the combustion of a pure carbon fuel. This result suggests that combustion in the burning lane takes place in accordance with the laws governing "pure-overfeed" action (24, 29).

Previous Investigations

Current descriptions of the fuel beds of underfeed stokers, with the exception of that of Barnes, appear to be based upon knowledge born of long experience in the operation of this equipment, without the control of exact measurements of conditions in such fuel beds. The work of Barnes (5) was concerned with single-retort stokers of domestic sizes, which differ so greatly from the large multiple-retort type investigated in this study, it could not be predicted that the results of his work would be applicable to these stokers. It appears, however, that if due allowance is made for the differences in the geometry of the system his results may be transferred with little change to a description of the beds of multiple-retort stokers, which usually operate in the region described by Barnes as "black-center" burning, and show many of the characteristics of such operation.

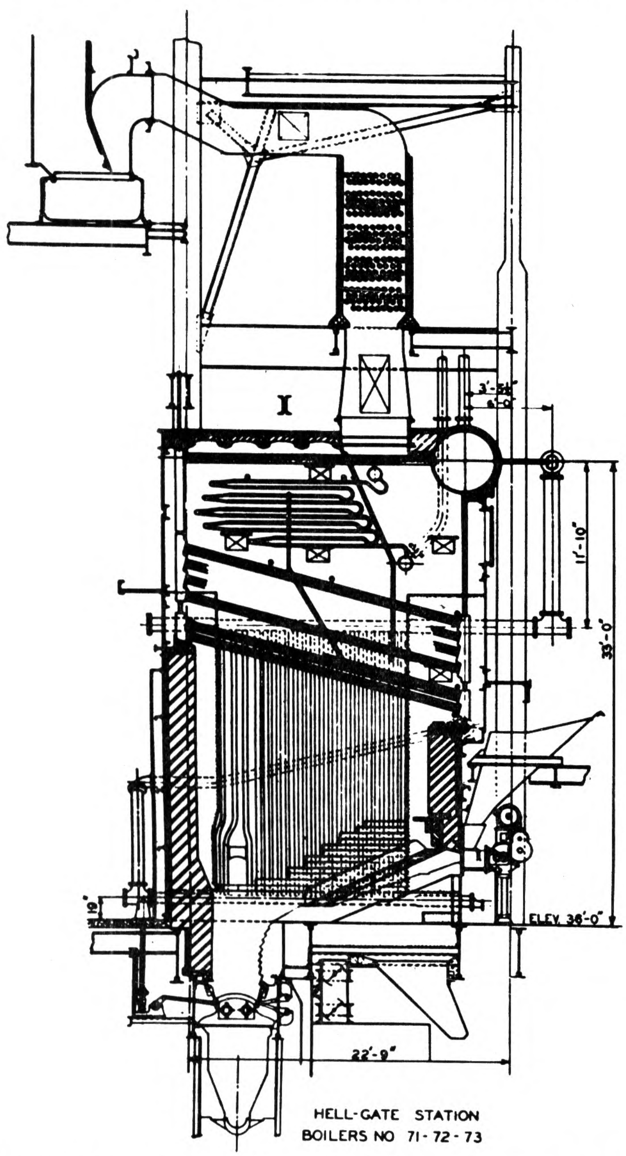


Fig. 11 Cross Section of Boiler and Stoker; Boiler No. 73, Hell Gate Generating Station

R. A. Foresman (12) has given a description of multiple-retort-stoker fuel beds which agrees quite well with that developed from this study, but no supporting data were given and conclusions which might be drawn from the geometry of the bed were not stated. In particular, he showed a deep pile of uncoked coal in the retort extending to a considerable height above the tuyères, but his description does not suggest the function of the coke wall in delimiting the "burning lane," at least with coking coals, or consider the implications of the horizontal heat flow in determining the rate of ignition. Tobey (33) has also described fuel beds which have many points of similarity to those described here. Houghton (16) has used the results of analyses of the fuel at different points in a fuel bed, which had been suddenly quenched, in an attempt to describe the progress of combustion in multiple-retort stokers, but he appears to share the impression (17) that ignition in such stokers proceeds by a mechanism analogous to that which controls "pure underfeed burning." His emphasis on the desirability of carrying thin fuel beds in burning low-volatile coal is supported by the results of this investigation, which demonstrate the paradox that a thinner fuel bed may actually have more fuel over the tuyères.

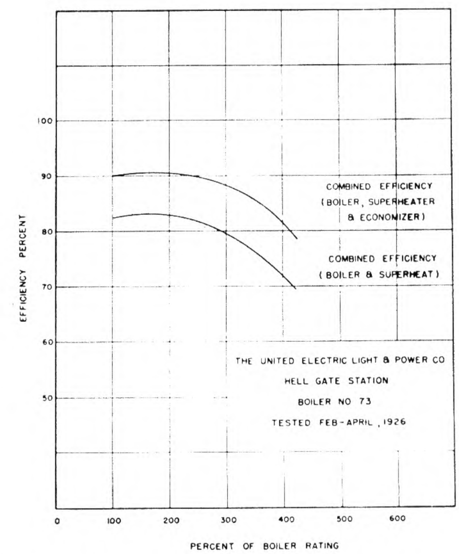


Fig. 12 Boiler Efficiency at Various Loads; Boiler No. 73
(From acceptance tests by United Electric Light & Power Company.)

In "pure underfeed burning," as described by Nicholls (27, 28), ignition of incoming fuel proceeds in the direction opposite to that of the air flow and takes place, as shown by Mayers (24), if the rate of heat conduction from the zone of free combustion is greater than the rate at which heat is convected back into that zone by the primary air. This type of burning leads to a fuel bed which is stratified in horizontal layers. The upper layers are in a condition similar to that described by Kreisinger, Ovitz, and Augustine (19), except that the temperatures, in the case of underfeed burning, are lower than those found by the last-mentioned authors in agreement with calculation (24). The rate of ignition in such a

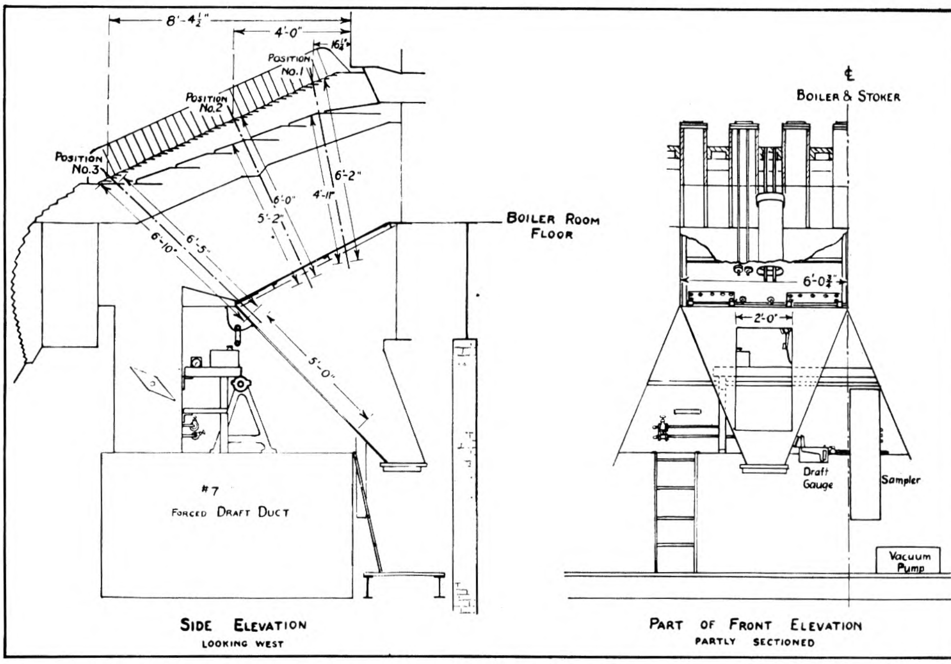


Fig. 13 Arrangement of Test Place or "Doghouse"

system can be calculated (26) from the characteristics of the fuel and the rate of air flow; it has also been determined experimentally (30, 31) as a technical characteristic of the fuel.

In Europe, where traveling-grate stokers are extensively used, it has been shown (14, 21, 23) that the concept of "pure underfeed burning" describes the processes occurring on such stokers, and this has been made use of in setting up model fuel beds (9, 13, 20, 32) for the study of fuel beds on traveling-grate stokers under controlled conditions. These investigations were of little use as a guide to the development of methods of measurement in the present project. In the investigations of traveling-grate stokers of commercial sizes, only grate temperatures and gas analyses over the fuel bed were determined, while the others, in which temperatures within the bed were determined, were carried out on model fuel beds in which equipment could be used which would not stand the conditions of operation in a full-size multiple-retort stoker.

Equipment and Construction

The tests were run on the stoker of boiler No. 73 at Hell Gate Generating Station of the Consolidated Edison Company. The boiler has 12,560 sq ft of heating surface and is a straight-tube, sectional-header type with an overdeck superheater. The furnace is water-cooled on the side walls and has 10 small secondary-air admission ports in the front wall. It has a 14-retort 37-tuyère Taylor stoker with high side-wall tuyères and a clinker-grinder ashpit. The unit is rated at 165,000 lb of steam per hr at 280 psi abs and 700 F steam temperature. A cross section of the boiler and stoker is shown in Fig. 11. The results of acceptance tests, made by the United Electric Light & Power Company in 1926, are shown in Fig. 12. Coal consumption during the tests was calculated from the steam flow indicated by the boiler meter, on the basis of the efficiencies found in these tests, divided by an operating factor of 1.06.

Although the stoker is rated at 37 tuyères, the original tuyères ($1^1/_2$ in. thick) have been replaced by thinner ones, 1 in. thick, so that there are actually 49 tuyères in each stack. There are 5 secondary rams in each retort, all bolted together so that they move as a unit, although provision had been made in the design of the stoker to allow certain amounts of lost motion between adjacent pushers; and the extension grates also move with the secondary rams and have the same stroke. The stroke of the set of rams serving each retort may be controlled from the front of the stoker by movement of an adjustable shoe. The furnace has two large inspection doors in the side walls above the ends of the ashpit. For these tests, an additional small inspection door was provided in the rear wall directly opposite the retort and tuyère stack in which the traverses were made.

Fig. 13 shows the construction of the doghouse, which was built into the sifting hopper to the east of the center line of the stoker, there being 4 hoppers under the stoker. A hole, 2 ft wide and 5 ft long, was cut in the rear sloping wall of the hopper and was provided with a cover plate which could be bolted over it when tests were not being carried on. The top of the hopper was roofed over with $^1/_4$-in. plate, reinforced by 2-in. angles, while the rear 6-in. section of the roof was built in the form of a trap door which could be operated from outside the hopper.

Water, compressed-air, and power connections were brought down along the wall of the wind-box connection, and a table was set up between that wall and the wind-box damper-operating shaft, as shown in Fig. 13. An electric fan standing on the table and directed into the doghouse was used to insure adequate air circulation, and a field-artillery telephone set with a line from the doghouse to the boiler-room floor near the panel of boiler No. 73 allowed ready communication between the two principal test stations.

Eight test points in the stoker were selected, four in the sixth retort from the east side and four in the tuyère stack adjacent to it on the east. These were divided into three groups: Position No. 1, 16 in. from the front wall, having one point in the tuyère stack and one in the retort; position No. 2, 4 ft from the front wall, having two points in each; and position No. 3, 8 ft $2^1/_2$ in. from the front wall, having one in each. It was originally planned to place the two points in both tuyère stack and retort at No. 2 position unequal distances from the center lines, but this proved to be impracticable because of constructional difficulties. In order to clear the operating link for the extension grate, it was necessary to place the point in No. 3 retort position 2 in. east, that is, toward the test-tuyère stack, of the center line of the re-

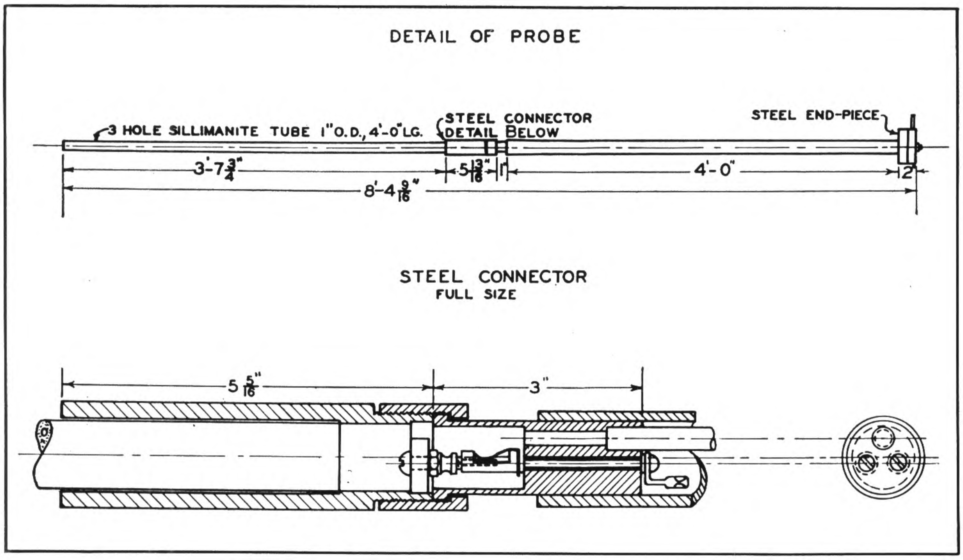


Fig. 14 Uncooled Probe

tort. At each test point, the stoker iron was burned out to accommodate a length of $1^1/_2$-in. standard pipe welded in place, which extended down through the roof of the doghouse. The guide tubes from the tuyère stacks passed through packing caps, welded to the roof, while those from points in the retort passed through 3-ft lengths of flexible steel tubing, welded to the roof at their lower ends and to plates welded to the guide tubes at their upper ends. All of the guide tubes terminated below the roof of the doghouse in square-cut surfaces which were used as reference planes to determine the distance of penetration of the probes into the fuel bed.

Test Apparatus

The probes used to measure temperatures and to take gas samples were required to withstand temperatures up to 3150 F, temperature gradients of 1000 deg per in., and the mechanical stresses incident to being forced through fuel beds 1 to 3 ft deep, containing coal in various stages of carbonization, coke, and occasional clinker, but still had to be small enough so that they did not seriously disturb the fuel bed and so that they might reach temperature equilibrium with the bed within a reasonable time. It was anticipated that each traverse of the fuel bed would probably be terminated by failure of the probe, but it was hoped that many of the traverses might extend to the top surface of the bed.

The water-cooled probe had a cooled section 8 ft long and a 12-in. mullite tip, $^7/_{16}$ in. diam. It was found that the tip was not strong enough to withstand the mechanical stresses to which it was subjected in tests over the tuyère stack, so its use was discontinued for such tests after run No. 20, the last of the preliminary group. Until run No. 48, the water-cooled probe was used in retort runs to locate the region of rapid temperature rise before stopping the stoker, but this procedure, which involved changing probes during a run, was abandoned after it was discovered that the uncooled probes were sensitive enough for this purpose. In run No. 43 the water-cooled probe was used with a shortened tip to withdraw gas samples from the upper levels of the fuel bed and to attempt to observe the probe from the boiler-room floor above the tuyères, which had not previously been done in regular runs. No temperature measurements were made during this run, as it was felt that the close proximity of water-cooled surfaces to the measuring junction would invalidate the observations. The attempt to observe the probe was unsuccessful, but this was accomplished in regular runs in the later series.

With the exceptions noted, uncooled probes, shown in Fig. 14, were used for all the measurements reported. These probes consisted of 1-in. mullite tubes, 4 ft long, with three holes, two $^1/_{16}$-in-bore for thermocouple leads, and one $^1/_8$-in-bore for gas sampling. The bottom six inches of the mullite were covered with a sprayed coating of copper so that the mullite tube could be soldered into the $1^1/_2$-in-outside-diam sleeve by which the probe end was connected to the 1-in. pipe extension or carrier. This sleeve, shown in detail in the lower part of Fig. 14, was connected to the carrier by a copper-gasketed joint and carried a bakelite junction piece on which the thermocouple wires were connected by binding posts to pins which mated with the jacks in the carrier from which compensating lead wire ran to a polarized porcelain receptacle in the hexagonal base of the carrier. A copper tube, $^3/_8$ in. diam, connected the gastight space in the connecting pieces with a nipple on the hexagonal base. A sleeve of $^1/_4$-in. standard pipe, welded to one side of the hexagonal base of the carrier parallel to the axis of the probe, was drilled every $^3/_8$ in. so that a chain passed through it to support the probe could be fixed in position by a pin passing through a set of holes and the chain. The pipe extension was marked by grooves extending one quarter of the distance around the pipe at every inch and by numbered grooves extending all the way around the pipe at every 5 in.

Thermocouples were made of No. 24 gage platinum and platinum 10 per cent rhodium wire. Silica sleeves, 24 in. long and $1^1/_2$ in. outside diam, were cemented to the measuring ends of probes which were to be used in the burning lanes so that they projected $^1/_8$ in. or somewhat less beyond the end of the probe, and cemented caps of Alundum (commercial alumina) and Insa-lute, a proprietary sodium-silicate cement, were laid over the thermocouple junctions in the cups so formed. After drying for 3 to 4 days, these caps were baked in a muffle furnace at a temperature of 800 F for 8 hr and allowed to cool slowly. The development of this procedure took until about the middle of the third series; prior to this time, the caps had not been baked and only those that had been stored for at least 3 weeks penetrated far into the fuel bed. Probes which were to be used in the retorts were not equipped with sleeves or caps after run No. 58, as the conditions in this region were not severe enough to require them. A view of a probe end, showing the sleeve and cap, is given in Fig. 15.

The apparatus for drawing gas samples, designed and built by the Research Bureau of the Consolidated Edison Company and illustrated in Fig. 16, was arranged to evacuate the single-opening

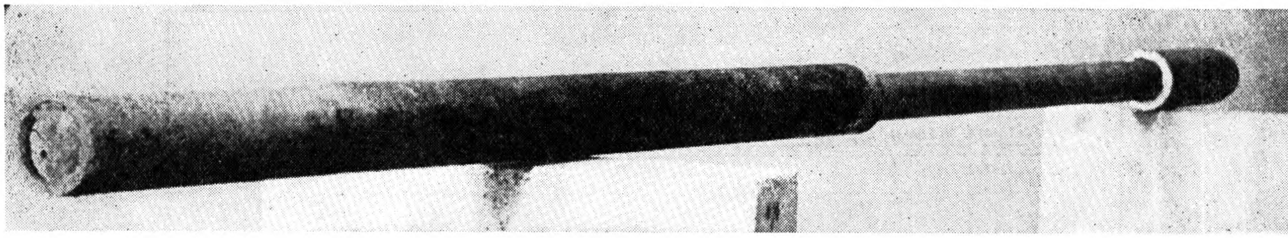


Fig. 15 End of Uncooled Probe, Showing Sleeve and Cap

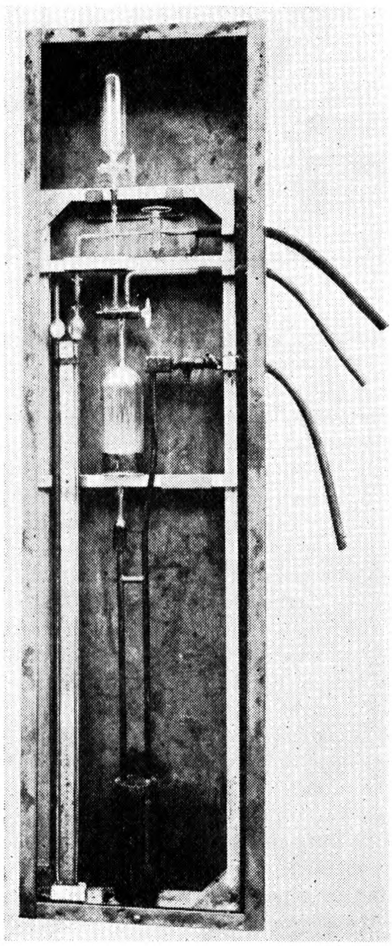


Fig. 16 Gas Sampler

sample bottles, flush the sampling line, withdraw a spot sample of the gas over mercury, and compress it into the sample bottle at a pressure 20 to 30 mm of mercury above atmospheric. The copper-tube sampling line ran from a convenient point in the doghouse, where a rubber-tube connection to the probe was wired to it, to a point near the sampler to which it was also connected by rubber tubing. It contained a tee, isolated by plug cocks, through which it was connected to the low-pressure side of a double-range oil-filled draft gage. The high-pressure side of the gage was connected to the wind-box section immediately below the portion of the stoker which was being tested, so that the gage read directly the pressure drop through the stoker and fuel bed. This arrangement was used to reduce the influence of minor fluctuations of wind-box pressure on the readings.

Gas samples were analyzed by means of a special Ellison gas analyzer having a 50-ml burette graduated to 31 per cent and with 250-ml solution containers, and by a large mercury-sealed apparatus, similar to that described by Evans and Davenport (10). All samples were run first in the small analyzer to determine CO_2, O_2, and CO. If the total of these compounds fell below 18.5 per cent plus the amount of CO, the samples were transferred to the large apparatus, in which the absorption of CO was completed, O_2 was added, and the sample burned on a platinum filament. The results of the combustion were calculated on the assumption that the gas consisted only of H_2 and CH_4. No attempt was made to analyze for unsaturated hydrocarbons. The speed of analysis was limited by the time required for combustion. When a long series of samples containing combustibles was being analyzed, the combustion analysis of every second or third sample was sometimes omitted, as each sample required approximately 40 min. Determination of CO_2 and O_2 on the small analyzer could be made continuously at the rate of about 15 samples per hr.

Motion pictures of the fuel bed were taken, following the completion of the tests. Plans had been made for taking these pictures during the tests themselves, but the apparatus was not completed in time to make this possible. The piece of equipment which made successful pictures possible was the pyroscope which has already been described (22). It was designed and constructed by the research bureau of the Consolidated Edison Company in cooperation with the Bausch & Lomb Optical Company, Rochester, N. Y. In these tests, the pyroscope and camera were usually mounted behind the boiler, and pictures were taken through the special observation door at a rate of one frame every 3 sec for periods of several hours. Shots were also taken from the side door of the furnace, looking across the fuel bed, using a camera speed of 64 frames per sec (slow motion) in order to observe the action of "popcorn," or flycoke.

In addition to the instruments just described which were developed especially for these tests, the regular operating instruments were also used and these were supplemented in some cases by calibrated test instruments. Steam flow was read from the Bailey boiler meter, which, in this installation contains two steam-flow elements, one for the west-side superheater outlet, and one, connected with the west-side meter by a totalizing linkage, for the

TABLE 9 BAILEY AIR-FLOWMETER CHECKS; BOILER NO. 73 HELL GATE GENERATING STATION

Date	11/18/37		2/24/38		3/11/38		4/6/38	
Steam-flow reading	148		224		220		207	
Air-flow reading	143		231		232		215	
	Gas-analysis traverse, per cent CO_2							
Distance from side wall, ft	E	W	E	W	E	W	E	W
1	8.0	9.7	9.9	9.8	10.6	10.0	8.6	8.0
3	11.6	10.8	11.0	10.6	11.4	10.2	11.0	11.1
5	15.6	13.6	11.4	12.3	13.3	11.9	13.4	12.5
7	16.8	15.1	12.6	13.0	13.8	12.7	14.0	12.7
9	16.7	15.6	14.7	13.6	14.4	14.2	14.4	12.7
11	15.8	16.6	14.6	14.2	15.3	14.2	14.1	14.0
Average CO_2, per cent	13.8		12.3		12.7		12.2	
Corresponding total air, per cent	135		152		147		153	
Total air with pens together, per cent	140		147		140		148	
CO_2 with pens together, per cent	13.4		12.7		13.4		12.7	

Note: Boiler out for overhaul, thoroughly cleaned January 25 to February 24. External cleaning only, March 19–20.

east side. These meters were checked against a static water column several times during the course of the tests and at no time was any correction found necessary. Air flow was read from the Bailey-meter air-flow pen, the setting of which was checked against Orsat traverses four times during the course of the tests, with the results shown in Table 9. Both steam-flow- and air-flow-meter checks were run by the Station Service Bureau of the Technical Service Department, using their standardized procedure. The air-flow-pen setting was not changed during the course of the tests, so that the variations in excess air with pens together are due to changes in the cleanness of the boiler. These variations were taken into account in calculating air flows from the observations. Furnace- and boiler-outlet drafts were read on an inclined draft gage, and the wind-box pressure under the test section was read on a vertical draft gage. The latter reading was compared with the operating wind-box pressure-gage reading, taken below the division plates between sections, to assure uniform operation during the tests. Steam temperatures were read on two engraved-stem test thermometers set in wells in the two superheater outlet headers, and feedwater temperature to the boiler was read on a test thermometer in the economizer outlet. A Ranarex CO_2 recorder was installed with a sampling line drawing from the third pass of the boiler directly above the test section, in order to observe the effect of changes in operating conditions on the gas stratification in the boiler passes.

The speed of the stoker was observed by timing a revolution with a stop watch and the length of the strokes of the secondary rams was measured with a foot rule. A log of all the ram strokes was kept after run No. 68; prior to this time only the stroke in the test retort had been recorded.

Coal Used in Tests

Coal is delivered to the station by barges from which it is hoisted to unloading towers containing screens and crushers which may be by-passed. From the tower it is dropped into cars operating on a cable railway which distribute it to the various bunkers. Samples for proximate analysis and heating-value determination are taken from the stream flowing from the coal tower into the cars. Coal for these tests was delivered to bunker No. 6, of about 350 tons capacity. The lorry serving the sixth and seventh rows of boilers could be filled from No. 6 bunker by the use of a transfer feed screw. Throughout the first four series of tests, the bunker was kept filled with Lower Kittaning run-of-mine coal, excepting for a period of about 2 days around March 15 (just before series 3), when a special coal under test by the operating department was in the bunker. For each of the last three series of tests in which special coals were used, the bunker was carefully cleaned before the special coals were dumped. When the Pocahontas No. 3 and "sized" Lower Kittaning coals were being hoisted, the crusher was by-passed, but at all other times the coal passed through the crushers as in normal operation. The stoker hopper was filled at hourly intervals from the weigh lorry, the weight of each dump being recorded. The lorry was completely emptied before drawing coal from No. 6 bunker for the test boiler.

Samples of coal for size analysis were taken from the stoker hopper by means of a long-handled scoop with a capacity of about 3 lb. Four scoopfuls were taken at points spaced approximately evenly across the stoker hopper to give gross samples of about 300 lb. The size analysis was determined by the use of a Tyler 12-in-square rocking-sieve shaker for sieves coarser than $^3/_8$-in. round hole, and 8-in. round hand screens for finer sizes. No measurements were made of segregation in the stoker hopper, as visual observation indicated this to be negligible as is to be expected in view of the use of a lorry for filling. The analyses of the coals are given in Table 2; the size analyses are plotted to log probability coordinates in Fig. 1. The size distributions of the coals used in series 1, 3, 4, and 5 are so much alike that they may be represented by a single curve, which is not, however, a continuous straight line. Series 2 has a coarser size distribution than that of the other series using Lower Kittaning run-of-mine coal, because the crusher usually used was out of service during a part of the period. The coal of series 6 was sized at the cleaning plant by removal of the $^5/_{32}$-in. fines, and has a somewhat coarser distribution than the other coals. There was a considerable amount of degradation in transit so that almost 45 per cent of it is finer than $^1/_8$ in., as received. The high-volatile coal, used in series 7, is almost as coarse in the large sizes as the coals of series 2 and 6 but contains an excessive amount of superfines, about 9 per cent passing a No. 100 screen.

All analytical data excepting sizing were determined by the technical service department of the company, using standardized procedures.

Testing Procedure

A full crew for the tests consisted of seven men, four being assigned to the doghouse, and one each to the boiler-room floor, to the chemistry, laboratory, and to relief and supervision. Of those assigned to the doghouse, two handled the probes within the doghouse itself, one operated the gas sampler and read the pressure differential, and one read the potentiometer, handled communications with the boiler-room floor, and recorded data. Operation of the gas sampler required the development of a considerable degree of skill, so that every effort was made to keep the same man on this job. The man on the boiler-room floor read the boiler-operating and test instruments at 15-min intervals, took coal samples from the stoker hopper each time it was filled, supervised the operation of the stoker, and kept the crew in the doghouse informed of the conditions of operation. The man assigned to the chemistry laboratory, who operated the gas-analysis apparatus with the intermittent assistance of the man on relief, had to develop a high degree of skill, and continuity in this assignment was essential for satisfactory analytical results. The relief man generally supervised the entire procedure, transported samples and probes, assisted in the gas analysis, and relieved the men in the doghouse at intervals. Relief was especially necessary for the men assigned to handling the probe, as the doghouse was usually hot and sometimes was rather gassy because of fumes blown down through the guide tubes from the fuel bed. Tests could be run for short periods with only six men, but this was avoided whenever possible, since it invariably resulted in the gas analysis falling behind.

Different procedures were used in tuyère tests and tests over the retort. In a tuyère test, the probe was inserted in the guide tube and raised until the tip was about 10 in. below the level of the tuyère plates. It was then blown out from the compressed-air line and the gas-sampling line and thermocouple-extension leads were connected. At this time the potentiometer, gas sampler, and draft gage were checked. Then the probe was moved up to 1 in. below the level of the tuyère. The chain was passed through the sleeve and pinned in place, the temperature and draft readings were started. The man on the boiler-room floor, who had previously notified the doghouse that conditions were sufficiently steady and representative for testing, was informed by telephone of the beginning of the test. The probe was advanced by $^1/_2$-in. steps, temperatures and pressure differentials being read at each step, and from $^1/_2$ in. above the level of the tuyère gas samples were taken at each position. Beyond 5 in. above the level of the tuyères the probe was advanced by 1 in. at each step after run No. 64. The probe was held in each position for at least $2^1/_2$ min, which was sufficient to allow reading the draft gage, a 40 to 50-sec flushing of the gas-sampling line, and drawing a gas sample,

TABLE 10 SAMPLE LOG OF TUYÈRE TEST; RUN NO. 72

(No. 2 tuyère, east; zero = 25¹/₂ in.; March 25, 1938)

Time	Position, in.	Potentiometer reading, mv	Temp, F	Bed differential, in. water	Sample no.
2:19	24¹/₂	0.8		2.05	
		1.1			
		1.3			
		1.4	385		
2:21	25	1.7		1.0	
		1.7			
		1.7	450		
2:22	25¹/₂	1.6		0.44	
		1.6			
		1.7	450		
2:23¹/₂	26	2.1		0.44	43
		3.3			
		4.2			
		4.6			
		4.9	1055		
2:25¹/₂	26¹/₂	5.0		0.43	16
		5.4			
		6.0			
		6.4			
		6.8			
		6.9	1400		
2:28	27	7.1		0.47	17
		7.6			
		8.4			
		8.8			
		9.3			
		10.0	1900		
2:31	27¹/₂	10.5		0.55	18
		10.7			
		12.2			
2:32¹/₂	27¹/₂	12.5			
		12.6	2295		
2:33¹/₂	28	12.9		0.60	19
		13.2			
		13.9			
		13.8	(13.9)		
		13.9	2490		
		14.0			
2:36	28¹/₂	14.2		0.59	20
		14.2			
		14.2	(14.0)		
		13.9	2505		
		13.8			
		13.7			
2:38¹/₂	29	13.6		0.90	21
		13.6			
		13.6			
		13.7	(13.8)		
		14.1	2475		
		14.1			
		14.1			
2:42	29¹/₂	13.9		0.80	22
		14.1			
		14.2			
		14.3	(14.2)		
		14.4	2535		
		14.2			
		14.1		0.90	
2:46	30	13.8			23
2:47		14.3	(14.3)		
		14.2	2550		
		14.3			
2:48¹/₂	30¹/₂	14.3		1.10	28
		14.4			
		14.4	(14.5)		
		14.6	2580		
		14.6			
2:51	31¹/₂	14.4			
		14.9		1.25	29
		14.9	(15.0)		
		15.1	2655		
		15.2			
		15.1			
2:54	32¹/₂	15.3		1.25	30
		15.4			
		15.1	(15.0)		
		15.1	2655		
		14.8			
		14.8			
		14.8			
2:57	33¹/₂	14.8		1.55	31
		14.7			
		14.3	(14.3)		
		14.2	2550		
		13.8			
		14.0			
		14.3			
3:00¹/₂	34¹/₂	14.7		2.10	32
		14.9			
		15.1			
		15.3	(15.6)		
		15.6	2750		
		15.7			
		15.6			

while temperatures were read every ¹/₂ min. The probe was advanced until it broke, as indicated by open circuit of the thermocouple, or until the probe was sighted above the level of the fuel bed. A portion of the log from run No. 72 is shown in Table 10,

TABLE 11 SAMPLE LOG OF RETORT TEST; RUN NO. 61

(No. 2 retort, west; zero = 7¹/₂ in.; March 9, 1938)

Time	Position, in.	Potentiometer reading, mv	Temp, F	Bed differential, in. water	Sample no.
		Stoker stopped at 10:09			
	7¹/₂	0.2		2.9	
	12¹/₂	0.2	95	2.2	
	17¹/₂	0.2		2.2	
	22¹/₂	0.2		2.5	
	25	0.2		2.6	
	26	0.2		2.6	
	27	0.2		2.65	
	28	0.2		2.65	
	29	0.2		2.7	
	30	0.2		2.8	
	31	0.2		2.8	
10:14	32	0.2	95	2.8	
10:14¹/₂		0.6	200		
15		1.1	320		
		1.8	470		
16		2.3	570		50 Slow
	33	3.6	820	2.9	
17		4.0	890		
		4.7	1020		
18		5.1	1090		
		5.5	1160		
19		5.7	1195		51 Slow
		Pulled probe out			
		Stoker stopped at 10:53			
10:54	31	0.5	175	3.1	
55		0.6	200		
		0.8	250		52
56	32	1.1	320	2.95	
		1.7	450		
57	32	2.3	570		
		2.9	685		
58		3.4	780		
		3.9	875		53
59	33	4.4	965		
		5.5	1160	2.8	

NOTE: No sleeve and no Alundum on tip of probe.

The procedure in the retort developed as the tests were carried on. The problem that necessitated a difference in procedure between the retort and tuyère tests was that introduced by the motion of the stoker. Over the tuyères, the fuel did not move enough to break the probe but, in the retort, the relative motion between fuel and pusher was sufficiently great to shear the probe off at every stroke. Preliminary tests showed that the probe could be kept in the fuel bed during the time the pusher was moving down the stoker, that is, toward the bridge wall, without breakage, but on the return stroke the probe was broken every time it was allowed to remain in. Since a time longer than the period between strokes was required for taking readings at each position, it was obvious that the stoker, or at least the section of it under test, must be stopped while the readings were taken. About 10 min prior to the test, the test section of the stoker was put in high speed by means of the gear-change box. The probe was inserted to a position several inches below the level of the secondary ram and the instruments checked. Then the man on the boiler-room floor was instructed to stop the test section and inform the doghouse the instant this was done. The prob was raised to the level of the top of the ram and advanced by 5-in. steps as rapidly as readings of temperature and pressure differential could be made. This was continued up to the neighborhood of the region of rapid temperature rise, known from previous experience to be more than 25 in. above the level of the ram at No. 1 position, and more than 20 in. at No. 2 position. From here on the probe was advanced by 2-in. and then by 1-in. steps, holding it at each position for about ¹/₂ min, in order to make certain that any rise in temperature would be observed. As soon as the galvanometer needle moved off zero, the advance was stopped and the temperature was recorded at ¹/₂-min intervals, while the pressure differential was read and a gas sample drawn. From this point, the probe was advanced 1 in. at a time, taking 2¹/₂ min at each position, until not more than 13 min had elapsed after stopping the stoker, when it was withdrawn. The section of the stoker was again started in high speed and run for about 15 to 20 min to build up the fuel bed to its original condition. If the

TABLE 12 SAMPLE OF BOILER-ROOM FLOOR LOG; RUN NO. 72; MARCH 25, 1938

Chart time	Steam Total	Steam West	Air	Draft Furnace, in. water	Draft Third pass, in. water	Wind-box pressure, in. water	B.O. gas temp, F	Superheat temperature East, F	Superheat temperature West, F	Feed-water temp, F	Pusher stroke, in.	Time for 1 revolution, min-sec	Ranarex CO_2, per cent
				0 = 0.10									
2:15	195	160	237	0.27	1.03	2.80	595	655	658	323	5/8	1-11	13.0
30	200	180	237	0.28	1.035	2.80	598	656	658	326	5/8	1-11	12.5
45	195	180	237	0.275	1.045	2.80	600	661	657	328	5/8	1-8	12.5
3:00	200	180	237	0.275	1.01	2.70	600	662	655	327	5/8	1-8	12.5
15	196	182	236	0.295	1.035	2.70	595	660	654	325	5/8	1-6	13.0

Fire Conditions

Chart time

2:15—Fuel bed appears even over entire stoker from west door, but from east door appears heavier on west side from neck down to 1/2 stoker length. Fire light but not much "popcorning." Considerable amount of small clinkers on extension grate, and under fuel in retorts and on tuyères; fuel rather well burned out on extension grates. Furnace clear except at neck where it is quite smoky; flame short; no secondary combustion top of first pass.

Chart time

2:38—Speeded up stoker on rheostat to 1 revolution in 1 min 8 sec. Half-moon effect on extension grate.

2:45—Burning lane at test position veering off to right, viewed from rear door.

3:00—Fire light.

3:10—Saw probe in center of burning lane. Tip of probe about 2 in. below top of burning lane and about 7 to 8 in. below top of fuel bed.

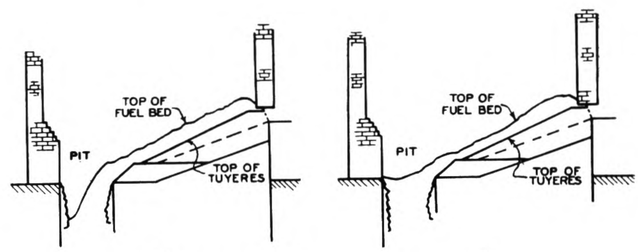
probe had not been observed during the previous part of the run the stoker was stopped again and the probe was again inserted and advanced by 1-in. steps from 1 in. below the last previous position. The probe was invariably sighted from the boiler-room floor during the second half of the test before 13 min had elapsed.

In tests in No. 3 retort position, the procedure was yet different since at this position the guide tube terminated in the dead plate just above the extension grate, where the motion of the fuel was not sufficient to break probes when they were less than 5 in. above the level of the dead plate. In tests at this position, the probe was inserted and moved upward by 1-in. steps to 5 in. without stopping the stoker. When high temperatures were observed, gas samples were taken at each level. When the probe reached 5 in. above the dead plate, the stoker was stopped and the run was continued in just the same way as in any other retort test. A sample log of a retort run (No. 61) is given in Table 11, and a sample of the boiler-room floor log is given in Table 12.

Stoker Operation

During the tests, more uniform fires than those required for normal operation had to be maintained to permit duplication of conditions in the various tests in each series, and to allow the calculation of the air flow through the test section as the average air-flow rate through the stoker. For this reason, it was necessary to place the operation of the boiler on a special schedule. Over week ends the doghouse was closed so that siftings could be discharged through the trap in the doghouse roof and the boiler was in normal operation on automatic control. From Monday to Friday the fire was cleaned as early as possible on the 12 to 8 watch, but the pit was not ground down after cleaning. The fire was normally banked during and after cleaning, and was operated on automatic control in accordance with load requirements after coming off bank until 8 a.m., when it was placed on hand control and run at the test rating during the remainder of the day. At the end of the tests, it was put back on automatic control and was in normal operation for the remainder of the 4 to 12 watch. The doghouse was opened at 8 a.m. on Monday and was left open, siftings being allowed to accumulate on the roof, until Friday night. Slagging of the first pass of the boiler with hand water lances, which had to be done three times a week on the day watch, was carried out the first thing in the morning on Tuesdays, Thursdays, and Saturdays, thus interfering with testing on only 2 days, when the testing day was shortened by about 1 hr.

During normal operation, the output of the boiler is governed by an automatic regulator (Smoot combustion control) which controls the air flow through the boiler in accordance with impulses sent out by a master controller, actuated by the main steam pressure. This regulator controls the draft at the boiler outlet; the wind-box pressure is controlled by another regulator,



(a) (b)

FIG. 17 FIRE CONTOURS
(a, Normal or long fire; b, short fire.)

FIG. 18 LONG FIRE; SINGLE FRAME FROM MOTION PICTURES TAKEN MAY 13, 1938
(Load 124,000 lb of steam per hr; excess air 36 per cent.)

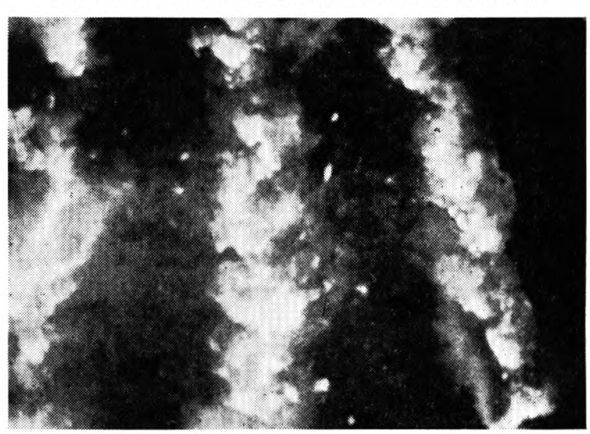


FIG. 19 SHORT FIRE; SINGLE FRAME FROM MOTION PICTURES TAKEN MAY 14, 1938
(Load 119,000 lb of steam per hr; excess air 53 per cent.)

not subject to loading by the master controller, which maintains a constant furnace draft. Stoker speed is determined by the voltage generated by a separate motor generator set for each row of boilers, not usually operated on the automatic control, modified by individual stoker-motor field rheostats at each boiler. During tests, the boiler outlet draft was taken off automatic control so that the regulator did not receive the impulses from the master controller, and was set by hand at a constant load. The windbox pressure regulator was left in service and the stoker-motor motor generator set was adjusted to give a voltage corresponding to the test load.

The stoker was operated by the regular operators with the advice and assistance of the test crew. This assistance became more extensive as the tests advanced and the conditions of operation departed more and more from normal. During the first two series, while the test crew was relatively unacquainted with the standard of operations and when it was desirable that the operating conditions be as nearly normal as possible, the test crew exercised but little supervision beyond recording changes in the settings of the various controls. With the beginning of series 3, which was run with high excess air, the necessity of operating with a thinner fuel bed than normal made it necessary for the test crew to take part in the adjustment of the stoker to a greater extent than previously; this necessity continued throughout the remaining series, especially when coals which were unfamiliar to the station operators were being fired.

During the first three series of runs, the normal fire contour, sketched in Fig. 17(*a*), was maintained. The sketch shows the approximate profile of the general level of the fire, set by the level of the fuel in the retorts. It could be seen from the rear observation door that the burning lanes were open channels in which a fuel level could not be observed. When the fire was thinned down to secure high excess-air operation in series 3, it was found extremely difficult to keep fuel on the tuyères at the head end of the stoker and to keep blowing from the front wall within bounds. Drifts of "popcorn" built up along the bridge wall and on one occasion the fuel was blown completely away from 3 or 4 ft of the upper end of the stoker. To overcome this difficulty, the contour of the fire was changed to that shown in Fig. 17(*b*), referred to as the short fire, by shortening the strokes of the secondary rams, and testing was continued as series 4.

Short-fire operation was continued throughout the remainder of the tests except with the high-volatile coal, both with high and with normal excess air, as it was found not only to decrease the amount of "popcorning," but also to decrease slightly the degree of stratification in the flue gas, as can be observed by comparing the traverse shown in Table 11 on April 6, run with a short fire, with the others in the table run with a normal fire. It caused a marked decrease in secondary combustion, observed at the top of the first pass, and produced a fire which was more completely burned out on the tail than the normal practice. In fact, it was necessary to carry the pit at a much higher level than the usual practice in this station in order to prevent the appearance of holes in the lower end of the fire. The differences in the general appearance of the long and short fires are shown in Figs. 18 and 19, made from single frames of the motion pictures taken after the tests. Part of the increased clearness of Fig. 19 is due to the increased excess air, as this picture was taken under conditions approximating those of series 4, while Fig. 18 was typical of series 2. The increase in the width of the burning lanes with the short fire is, however, quite apparent.

When special coals were burned for the tests, the new coal was burned for 24 hr before any tests were run, in order to assure that only the special coal would be in the fire during the tests and to give the test crew time to work out the proper conditions for firing the new coal.

Preliminary Tests

The results of the tests run in August and September, 1937, have already been referred to as showing the feasibility of the proposed test procedure and indicating the need for redesign of the original probes to the forms described. In addition to these, the first 20 runs made after the beginning of intensive testing on November 1 were in the nature of preliminary runs and were devoted mainly to a comparison of the compositions of gas samples drawn by the water-cooled and uncooled probes and to the temperature indications of the two kinds of probes.

When the uncooled probes were first used in these runs, they were not equipped with sleeves or caps and it was found that they broke, apparently because of the thermal shock, after penetrations of only 2 to 3 in. H. W. Russell, chief physicist of Battelle Memorial Institute, suggested, on hearing of this difficulty, that the mullite tubes be provided with silica sleeves which, although they could not stand the maximum temperatures encountered, would slow up the rate of heat penetration in the early stages enough to prevent spalling. When this was done, it was found that the probes penetrated well into the bed without breaking; but that the thermocouples were attacked by slag in the region of high temperature, leading to erroneous temperature indications. This difficulty was eliminated by the provision of the cemented caps described in the section on "Test Apparatus." These provisions made the uncooled probes quite massive, causing their indications of temperature to lag in the early stages of each run. The method of correcting for this is described in the next section on "Supplementary Data."

The water-cooled probes used in the preliminary tests were also provided with silica sleeves and caps, but failed because of mechanical breakage, so their use was discontinued. There was, however, some question as to whether gas samples taken with the uncooled probes would be cooled rapidly enough to quench the gas reactions and thus be representative of conditions in the fuel bed. The data in Table 13 show a comparison of gas samples taken from the same levels at No. 2 tuyère, west position, by both the water-cooled and uncooled probes. While these early runs

TABLE 13 COMPARISON OF GAS SAMPLES DRAWN WITH COOLED AND UNCOOLED PROBES AT NO. 2 TUYÈRE, WEST

Height above tuyères, in.	Water-cooled probe					Uncooled probe				
	Run No.	CO_2 per cent	O_2 per cent	CO per cent	Total per cent	Run No.	CO_2 per cent	O_2 per cent	CO per cent	Total per cent
1	4	2.2	15.6	0.3	18.1	3	0.0	15.8	0.0	15.8
	8	3.6	15.4	0.2	19.2	9	2.8	15.9	0.1	18.8
	11	1.7	18.2	0.0	19.7	14	5.2	14.4	0.1	19.7
	15	6.9	12.4	2.5	21.8					
1 1/2						7	0.9	18.4	0.0	19.3
						10	6.9	10.6	...	..
3	2	7.9	6.2	2.9	17.0	3	0.7	19.1	0.7	20.5
	4	6.7	5.1	3.7	15.5	7	8.1	11.8	0.2	20.1
	8	13.8	4.5	3.1	21.4	9	8.1	6.8	4.0	18.9
	11	0.7	20.3	0.3	21.3	10	11.9	5.6	2.1	19.6
	15	13.1	5.8	0.4	19.3	14	14.0	1.2	3.8	19.0
5	11	1.5	18.8	0.4	20.7	7	8.5	11.2	0.0	19.7
	15	5.7	7.7	6.2	19.6	14	12.3	3.1	4.4	19.7
7	11	0.5	19.9	0.2	20.6	7	1.0	18.2	0.2	19.4

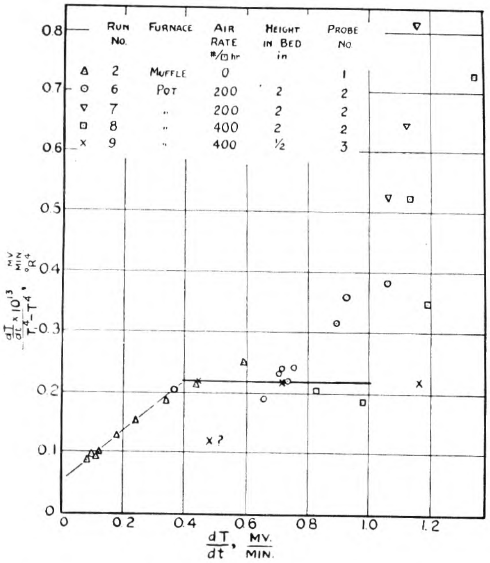


FIG. 20 DATA FOR CORRECTING TEMPERATURE INDICATIONS OF CAPPED PROBES

rarely penetrated beyond 3 in. into the bed, and the analysis for H_2 and CH_4 in these samples was not completed, the data show that the differences between samples taken by the water-cooled and by the uncooled probes were less than the differences among samples taken by either one. In particular, the samples taken in run No. 9 with the uncooled probe are parallel to those taken in run No. 8 with the cooled probe, those from No. 3 with the uncooled probe to those from No. 11 with the water-cooled probe, and so on.

These results indicated that the water-cooled probe did not quench the gas reactions any more rapidly than the uncooled probe, as might be anticipated from the fact that the top ten inches of the water-cooled probe were uncooled. On the other hand, samples taken with the water-cooled probe and a tip only 2 in. long (run No. 43, when no temperature measurements were made) do not differ consistently from samples taken at the same points with the uncooled probe. Finally, many samples taken with the uncooled probe, from levels more than 4 in. above the tuyères, had analyses showing the presence of up to 10 per cent of combustible gases mixed with 3 per cent of oxygen. These facts indicated that the gas reactions may be rapidly enough quenched even in the uncooled probe to introduce only negligible errors in the gas compositions observed.

SUPPLEMENTARY DATA

(a) *Correction of Temperature Indications.* Tables 10 and 11 show that the temperatures, recorded when entering the fuel bed above the tuyères and when passing through the hot layer above the retorts, did not reach steady values. It was recognized that the probes were so heavy as to raise the question of temperature equilibrium between them and their surroundings, so in the earlier tests they were held in position until the rate of rise of tempera-

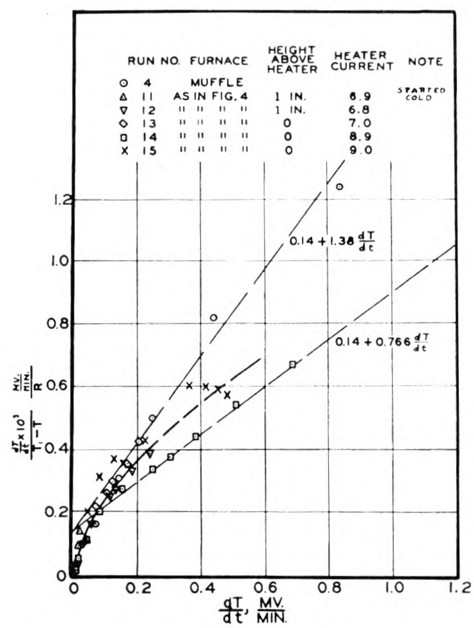


FIG. 21 DATA FOR CORRECTING TEMPERATURE INDICATIONS OF UNCAPPED PROBES

TABLE 14 RESULTS OF TESTS CALCULATED FROM MOTION-PICTURE FILMS

Set	Film speed: Projected frames per sec	Film speed: Exposed frames per min	Transverse distance as fraction of c to c distance of tuyères	Time of motion as projected, sec	Transverse velocity, fph	Calculated	
					Long fire		
1	12.6	18.0	0.075	6	1.85	Rating, 1000 lb per hr:	
			0.076	14	0.81	Steam	125
						Coal	12.7
2	14.9	18.0	0.106	10	1.33	Burning rate referred to air admission surface	
			−0.057	21	−0.34		69.8
			0.161	28	0.72		
3	13.6	18.0	0.034	7	0.67	Per foot length of tuyères	61.1
			0.068	11	0.85		
			−0.155	18	−1.18	Fuel bed depth, in.	14
			0.106	10	1.45		
	Average				0.68 ±0.22	Coal flow rate	0.58
					Short fire		
1	14.8	18.0	0.101	12	1.06		
			−0.089	9	−1.25		
			0.153	11	1.76		
			0.072	13	0.70	Rating, 1000 lb per hr:	
2	13.9	18.0	0.095	14	0.96	Steam	119
			0.095	7	1.83		
			0.057	12	0.65	Coal	12.2
3	14.4	18.0	0.078	8	1.26	Burning rate referred to air admission surface	
			0.070	13	0.70		67.2
			0.039	10	0.51		
			0.068	16	0.55		
			0.072	10	0.94	Per foot length of tuyères	58.8
4	13.9	18.0	0.082	8	1.38		
			−0.028	9	−0.42	Fuel-bed depth, in.	12
			0.037	7	0.52		
	Average				0.75 ±0.14	Coal flow rate	0.65

ture fell to 0.2 mv (32 F) per min in the belief that this would assure a discrepancy between the temperature observed and that of the surroundings less than the expected error (150 F). This required 5 min or more and thus shortened the traverses which could be made within the normal life of the probes, so this procedure was abandoned. It became necessary, therefore, to estimate, from the observed data, the temperature which would have been reached by the probe if left for an indefinite time at each of the lower levels.

In order to do this, tests of capped probes were made in one of the pot-type furnaces in the Fuel Laboratory of the Pittsburgh Experiment Station of the U. S. Bureau of Mines, and of the uncapped probes in a special setup in the Coal Research Laboratory. The details of procedure and of analysis of the data are reported elsewhere (25), but the results of the tests are embodied in Figs. 20 and 21, applying to capped and uncapped probes, respectively. These curves give data by the use of which the temperature of the surroundings, referred to as T_1, may be calculated when both the indicated temperature T and its rate of rise at the time of observation $\frac{dT}{dt}$ are known. Thus, suppose the indicated temperature with the capped probe is 1385 F and the rate of rise is 0.63 $\frac{\text{mv}}{\text{min}}$. From Fig. 20, at $\frac{dT}{dt} = 0.63 \frac{\text{mv}}{\text{min}}$, the ratio $\frac{dT}{dt}/(T_1^4 - T^4)$ is 0.233×10^{-13}, giving $T_1^4 - T^4 = 27 \times 10^{12}$. Since $(1385 + 460)^4 = 11.56 \times 10^{12}$, $T_1^4 = 38.6 \times 10^{12}$ and $T_1 = (2490 - 460) = 2030$ F. Fig. 21 is used in the same way except that only the first powers appear and absolute temperatures need not be calculated. It is estimated that corrected temperatures calculated in this way are within ± 90 F of the radiant mean temperature at the point with the capped probes, and within ± 150 F with uncapped probes.

The temperatures at points, at which a continuous change was observed, have been corrected by the method thus described. These include the points less than 3 to $3^1/_2$ in. above the tuyères and those in the thin (about 2 in. thick) heated skin at the top of the retorts. At all other points, the observed temperatures were averaged, omitting from the average any continuous series which may have appeared at the beginning of the observation period, in which successive readings differed by more than 0.1 mv (15 F). It is estimated that these temperatures are in error by not more than 100 F, and considered probable that, if a consistent error exists, it is such as to make the recorded temperatures too low.

(*b*) *Estimate of the Flow Velocity of the Fuel.* The magnitude of the transverse component of the fuel flow was estimated from the motion pictures taken at the end of the tests. This component is responsible for the delivery of coked fuel from the retorts to the tuyères and has the direction of the horizontal perpendicular to the center lines of the tuyère stacks and retorts. It is considered positive in the direction from the center line of the retort toward the center line of the next adjacent tuyère stack.

Films taken through the auxiliary door in the rear wall of the furnace were projected through a glass balance case on a screen of tracing paper mounted inside the case, thus making it possible to mark the position on the back of the tracing paper of any prominent object at any time. Using a metronome as a timer, the progress of several bodies of fuel shown in each portion of the film was marked as points at $^1/_2$-sec intervals, while the film was being projected at a known rate. The center lines of the tuyère stacks were also shown on the tracing by passing a line through the average position of the burning lane. The line of dots, representing the progress of each body of fuel, usually sloped downward toward the center lines of the tuyères, although a few sloped away from them. From the slope of the line, making allowance for perspective, the number of dots, representing the time of projection, and the ratio of film speed as projected to that during exposure, the transverse component of the velocity of the fuel could be calculated.

Some results calculated in this way from films taken at a load of 125,000 lb of steam per hr with both long and short fires are shown in Table 14. The films exposed at a lower load were too cloudy to permit calculation. The results have only a low precision, but the values observed are close to those calculated from the dimensions of the bed. Moreover, the observed difference between the flow velocities in the long and short fires is parallel to that calculated.

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Parallel Versus Individual Operation of Multicyclones

By L. C. WHITON, JR.,[1] PORT CHESTER, N. Y.

In this paper, the author discusses the problem of determining the comparative efficiencies of individual cyclone-type dust collectors and cyclones grouped with interconnecting ducts and hoppers. The test setup and procedure followed are described fully, while the results are completely tabulated. The conclusion is reached that, with a damper-equipped cyclone of the size used in actual practice and operated under identical conditions of resistance and temperature, performance of the individual cyclone is representative of the group in the field.

THE increasing adoption of cyclone-type dust collectors in series for parallel operation indicates the importance of determining whether an individual cyclone or a group of cyclones with interconnecting ducts and hoppers would have the same dust-collection-efficiency characteristics. This is partly due to the fact that extremely accurate collection efficiencies may be determined for an individual cyclone, since a single unit handles a relatively small quantity of gas and, therefore, can be tested for its efficiency, other than by merely sampling the dust in the gas. The possibility of experimental error in the sampling method of determination is necessarily considerable.[2] However, by utilizing an individual cyclone, a far more accurate measure of the efficiency of this cyclone can be made by any one of several methods. For example, a given quantity of dust can be introduced into an otherwise dust-free gas, and the amount collected used for determining collection efficiency.

Another accurate method of test can be arrived at by connecting a single cyclone to a flue, weighing the dust caught in an individual hopper for the collector, and then collecting the escaping dust in a cloth bag and determining the increase in the weight of the bag. By comparing the absolute weight of dust caught and the increase in the weight of the cloth collector, a greater degree of accuracy of the efficiency determination may be attained than by sampling at the inlet and outlet of a cyclone collector, due to the greater experimental error inherent in the sampling method.

Such results have been studied, and field tests on a group of multicyclones of the same design have been shown to correspond accurately with observations made when one cyclone was tested.[3]

Nevertheless, it is important to observe the operation of individual or multiple cyclones under precisely the same conditions, in order to determine whether possible recirculation through certain of the cyclones destroys the efficiency of any of them because of varying static conditions, thus causing an over-all collection with multiple units which would not be true with an individual unit.

[1] Prat-Daniel Corporation.

[2] "Testing Dust Collectors," by J. E. Watson, *Power*, November, 1939, pp. 68–70.

[3] "Dust-Collection Tests at New Power Plant of the Industrial Rayon Corporation," by C. B. McBride, *Combustion*, September, 1939, pp. 36–38.

Contributed by the Fuels Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TESTS OF SINGLE- AND MULTIPLE-CYCLONE OPERATION

To check the matter a test setup was prepared as shown in Fig. 1. The arrangement consisted of three commercial-size cyclones with a common inlet and outlet duct, and a common bin. At the bottom of each cyclone and within the bin, a threaded connection was made so that small individual bins could be attached to the cyclones, although they would still have common inlet and outlet ducts. These individual bins prevented any circulation between cyclones via the bin.

All cyclones were of Thermix design, the important feature of which is that each cyclone is equipped with a vertical damper at the inlet. Although in practice these vertical dampers are interconnected by an outside mechanism so that they are all in the same relative position, it was possible in connection with the second series of tests to be described to place them in variable positions so that one cyclone would handle a considerably greater quantity of gas than another. Thus the worst possible conditions were reproduced; hence, if any circulation were to occur with a resultant drop in collection efficiency, this fact would be noted.

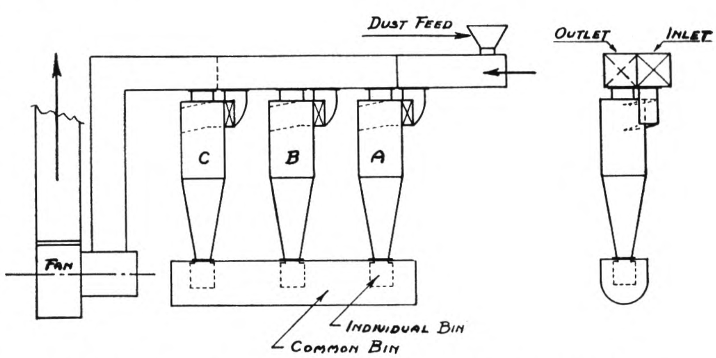


FIG. 1 TEST SETUP OF THREE MULTICYCLONES IN PARALLEL

The dust used for testing was collected by an electrostatic precipitator and was unusually fine due to some loss of coarser particles in the gas which escaped therefrom. The analysis of this dust follows: On 100 mesh, 2.86; through 100 on 200 mesh, 14.00; through 200 on 325 mesh, 10.38; through 325 mesh, 72.76.

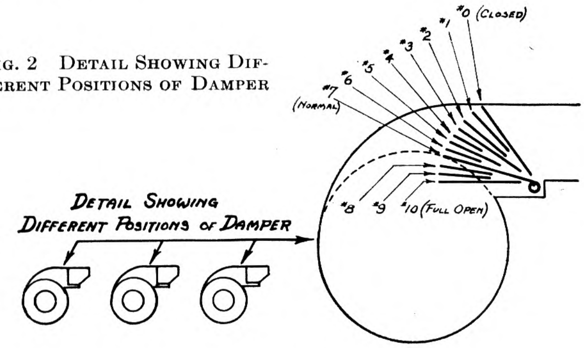
No attempt was made to use hot gases, since a comparison in operation rather than an absolute determination of cyclone efficiency was to be made.

The dust, of which several barrels were received, was carefully mixed in a large bin in order to obtain a mixture that so far as possible would be uniform.

In the test, air was drawn through the cyclones at a predetermined draft loss by means of an induced-draft fan at the outlet. The dust was fed through a 15-ft duct to the immediate inlet of the cyclones at an approximate rate of 2.5 to 3.5 grains per cu ft of air. The feeding mechanism was a special hopper, equipped with screens and a rotating device, which supplied an even load of dust. The mixture of air and dust was passed through an egg-crate guide in order to even the flow into the inlet duct immediately above the cyclones.

Although, as stated, the dust was carefully mixed, it was felt that it might still be possible for certain portions in the bin to be coarser than others. In order to diminish the chance of errors occurring from this cause, the tests were run first with a common and then with an individual bin, rather than entire series of tests with a common bin, and then again with an individual bin. After each test, dust settling on the bottom of the 15-ft inlet duct was stirred with an air lance and allowed to pass through the collector.

FIG. 2 DETAIL SHOWING DIFFERENT POSITIONS OF DAMPER



As is shown in Fig. 2, the damper in the cyclone could be adjusted to ten different positions. Position 7 was tangential with the cylindrical body of the cyclone and was the maximum opening permissible to obtain good dust collection and similar to the normal design of the cyclone inlet with the damper omitted.

CONDITIONS AND RESULTS OF TESTS

In the first series of tests, the results of which are given in Table 1, all three dampers were set in position 7. The draft loss of the cyclones was maintained at 2 in. There was a slight variation in temperature caused by a near-by furnace which was operating in the test laboratories. Also, in this first series, there was a variation in grain loading, due to the operation of the loader, which was not as uniform as desired. However, it is to be noted that the variation between 2 and 20 grains per cu ft does not noticeably affect the collection efficiency.[4]

[4] "Operating Variables of Cyclone Dust Collectors," by L. C. Whiton, Jr., *Chem. & Met. Engr.*, March, 1932, p. 150.

The average collection efficiencies, using either the common bin or the individual bin, were within 0.56 per cent, which is within the limit of tolerance for experimental error. Therefore, it should not be assumed that the common bin was necessarily more effective in the operation than the individual bin. However, the results may be considered as conclusive proof that, with the dampers set in such a position as to obtain approximately the same static conditions in each cyclone, there is no decrease in collection efficiency when using the common bin.

The second series of tests was conducted with the dampers of the cyclones set in various positions with the express purpose of creating different static conditions in the different units. It must be pointed out that positions 8, 9, and 10, as well as 1 and 2, are not indicated for the highest percentage of collection possible with this design of cyclone. Therefore, it is to be expected that the collection efficiencies will be somewhat less than in a more normal range from positions 3 to 7. The practical object in having available positions 8, 9, and 10 is to make it possible to pass 25 per cent more gas through the units at position 10, and thus provide the capacity safety factor for temporary operating conditions which is desirable in connection with boiler-plant operation. In other words, without increasing the draft loss, it is thus made possible to reduce the number of cyclones to 80 per cent of what would otherwise be required.

In Table 2, the results are the more interesting in that there was such a wide variance in the quantity of gas handled by each cyclone without any appreciable difference in the collection efficiency. This table actually indicates 0.26 per cent better collection with a common bin, which is due to experimental error.

It is well known that a group of cyclones in parallel can be thrown out of balance statically so that recirculation and reduced efficiency will result. From this series of tests, it can be concluded that, by use of a common bin, considerable variation in positioning of the inlet-control dampers, with resulting variations in cyclone capacity, can take place with no decrease in collection.

Provided the individual cyclone tested is equipped with inlet-control dampers, and is of the size to be used in actual practice and operated under identical conditions of resistance and temperature, it may be concluded that it is representative of the performance of the group in the field.

TABLE 1 TEST RESULTS ON THREE CYCLONES IN PARALLEL SERIES WITH DAMPERS SET UNIFORMLY IN POSITION 7

Test no.	Type bin	Cyclone resistance, in. w.g.	Temp, F	Duration of test, min	Total air, cu ft	Weight charged, oz	Loading, grains per cu ft	Weight collected, oz	Efficiency per cent
1	Common	2.0	85	20	69300	336	2.12	306.25	91.2
2	Individual	2.0	79	16	54900	317	2.52	291	91.8
3	Common	2.0	81	15	51600	311.5	2.64	281	90.2
4	Individual	2.0	79	16	54900	316	2.52	280	88.5
5	Common	2.0	85	20	69300	317	2.00	294.75	93.1
6	Individual	2.0	75	10 1/2	35700	317	3.88	290.5	91.7
7	Common	2.0	77	9	30750	317.75	4.52	288.25	90.8
8	Individual	2.0	86	14	48600	314	2.82	282.5	90.0
9	Common	2.0	78	12	41100	316	3.36	291	92.2
10	Individual	1.94	85	14	47700	311.5	2.85	284.25	91.3
11	Common	2.0	77	17 1/4	58950	318	2.36	286.5	90.1
12	Individual	2.0	76	27	92100	318	1.51	289	90.9
Avg	Common	2.0	..	..	...	....	2.833	....	91.26
Avg	Individual	1.99	..	..	...	....	2.683	....	90.7

TABLE 2 TEST RESULTS ON THREE CYCLONES IN PARALLEL SERIES WITH DAMPERS SET IN VARIOUS POSITIONS

Test no.	Type bin	Damper position in cyclone A	B	C	Cyclone resistance, in. w.g.	Temp, F	Duration of test, min	Total air, cu ft	Weight charged, oz	Loading, grains per cu ft	Weight collected, oz	Efficiency, per cent
101	Common	3	5	7	1.94	79	15 1/2	52200	316.75	2.65	285	90.0
102	Individual	3	5	7	1.94	78	14 1/2	48700	313	2.81	279.5	89.4
103	Common	7	5	3	1.94	83	14 1/2	49100	316.5	2.82	287	90.7
104	Individual	7	5	3	2.0	82	13	44800	318	3.10	284.5	89.5
105	Common	2	5	8	2.0	76	15 1/4	52000	315	2.65	276	87.7
106	Individual	2	5	8	2.0	77	14	47900	316	2.88	274.5	87.0
107	Common	1	5	9	2.0	81	15 1/4	52400	316	2.64	272	86.1
108	Individual	1	5	9	2.0	82	14 1/2	49900	316	2.77	270	85.5
109	Common	7	1	7	2.06	80	15 1/4	52300	318	2.66	276	86.8
110	Individual	7	1	7	2.0	92	16	56200	318	2.47	282	88.7
Avg	Common	..	..	..	1.988	..	..	...	....	2.684	...	88.28
Avg	Individual	..	..	..	1.988	..	..	...	....	2.806	...	88.02

Discussion

H. H. Bubar.[5] Mr. Whiton's paper presents extremely interesting observations on the performance of an individual cyclone, tested under the conditions stated, as a fair criterion of the field operation of a multiple-cyclone unit.

From these observations it is deduced that the method outlined is more accurate than the commonly used sampling method of testing, because the possibility of experimental error in the sampling method is necessarily considerable. These observations are amplified by the statement that extremely accurate tests can be run on individual cyclones and that the extreme accuracy is partially due to the relatively small gas volume used.

It is believed the accuracy of these and other conclusions made in the discussion, are open to question. Nothing in the paper justifies the conclusions drawn. In the tests as outlined, both in arrangement of apparatus and method of conducting the tests, hypothetical points are established which do not conform to field conditions. No proof is offered that measurement of small gas volumes is more accurate than measurement of large gas volumes.

The error in testing is not experimental and it is not necessarily considerable. It is true that in a laboratory setup a given quantity of dust can be introduced into the gas stream and accurate data obtained. However, in field installations, the dust is already in the gas stream and the relative quantity must be determined. How otherwise than by the sampling method are the initial dust loading and gas volume to be obtained and checked in field installations?

It is stated that field tests on a group of multicyclones of the same design have shown corresponding accuracy with observations made on a single unit. While it is not stated how the field tests were made, this statement appears to be contradictory to the previous statement of "necessarily considerable" errors experienced in testing a multiple unit in the field.

It is stated that it is important to observe the operation of individual or multiple cyclones under precisely similar conditions. The results of the laboratory tests, which do not indicate conditions at all similar to field conditions, either in setup of apparatus or method of testing, are then presented as conclusive proof to the effect that testing an individual cyclone will give the same results as testing a multiple-cyclone installation.

As a further basis for the statement of conclusive results obtained, the screen analysis of the dust used is stated as being "unusually" fine, as the dust was obtained from an electrostatic precipitator. This unusually fine dust is stated to have 72.76 per cent through a 325-mesh screen. Recorded experience with dust from electrostatic precipitators on power plants would indicate this dust as being comparatively coarse, as such dust usually runs close to 90 per cent through a 325-mesh screen and often finer.

It is stated that it is well known that a group of cyclones in parallel can be thrown off balance statically so that recirculation and reduced efficiency will result. If such is the case, then any test of a single cyclone unit should be considered to be inconclusive and the standard sampling method of efficiency determination used to establish the over-all performance of a multicyclone collector.

It is stated that an egg-crate guide was used in order to even the flow into the inlet duct immediately above the cyclones. In field installations of any size, we have yet to find any case of an approximately even flow of inlet gas to dust-collecting apparatus.

It is stated that, although the dust was carefully mixed, it was felt that it might still be possible for certain portions in the bin to be coarser than others. In extracting samples from various parts of flue or breeching on field installations, how often does it occur that the dust in these various samples is of uniform screen analysis?

No draft differentials are given between various points in the apparatus to permit a check of operating conditions or to compute gas volumes accurately.

No dimensions are indicated whereby size of cyclones, ducts, common bins, etc., can be determined so as to establish velocities at different points in the apparatus.

In Fig. 1 a common bin is shown which does not conform in cross-sectional area with bins used in field installations. Because of this difference it is possible that air velocities and turbulences in this common test bin would be much higher than in the bins designed for field conditions. Assuming this possibility, then the draft differential in the test bin would be higher than that in the field installation, resulting in a definite variation between the test and field results.

Also, the results of tests on three cyclones set up in parallel are deemed conclusive proof of what would be secured with, say, fifteen cyclones set up in parallel. This is questionable.

The data submitted do not justify the conclusions drawn in the paper, either in the substitution of tests on a single collector for tests on an entire installation, or in the substitution of the method offered in preference to the sampling method as outlined in the tentative code of the A.S.M.E.

P. H. Hardie.[6] The author has presented interesting data on the controversial subject of whether the performance of a single unit of a multicyclone dust separator can be accepted as representative of the combined units. The data obtained on the experimental setup used would seem to prove that it can. However, such a limited investigation should not be accepted as final proof. Are three cyclones in parallel representative of forty, or even twelve? Did the methods used to create an unbalanced condition produce the same type of disturbance to proper operation as might be encountered in service?

The failure of some installations of multicyclones to duplicate the performance of a single unit tested individually is thought to be due to poor distribution of flow at the entrance to the cyclones or to the disturbances set up in the outlet duct at the points where the escaping gases leave the cyclones. Either, or both, of these conditions could cause unequal pressures in the different units which would result in the flow of dust-laden gas from the bin through some of the units to the outlet duct. This condition is somewhat similar to that which exists when a bin is not properly sealed.

The first group of tests would seem to indicate merely that, with these particular arrangements and dimensions of inlet duct, outlet duct, and bin, there is no difference in performance when operating with the common bin or with individual bins. In this connection it is noted that the flow in the inlet duct was straightened. The results would be more helpful if poorer rather than better distribution at entrance had been produced.

The conditions imposed during the second group of tests were probably more severe than would be encountered in actual practice. The results so obtained are highly informative; but does unequal flow through the different units, when produced by setting the dampers in different positions, reproduce the same type of unbalancing as sometimes exists when the dampers are in the same position?

Even if we accept the author's conclusion that one cyclone can be used to determine the performance of a large number of similar

[5] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[6] Research Engineer, Department, Technical Development and Research, Consolidated Edison Company of New York, Inc., Brooklyn, N. Y. Mem. A.S.M.E.

ones in parallel, there would seem to be good reasons for preferring the over-all test after installation as proposed in the A.S.M.E. Power Test Code for Dust-Separating Apparatus. The apparent advantages of the single-unit test materialize only if the tests can be carried out in the manufacturer's testing laboratory. This means that a large representative sample of the dust must be collected and shipped to the factory. The collection of this dust is in itself almost as much of a job as running the over-all test. Even if a similar steam-generating unit, equipped with an electrostatic precipitator, is available from which a representative sample of dust can easily be obtained, there still remains the problem of reproducing the original finely divided mixture of dust and gas in the laboratory.

Dust particles, especially the very fine ones, tend to stick together once the dust has been collected and allowed to pack. By very fine particles is meant those which are considerably smaller than 44 mu (325 mesh). It is in this range of particle sizes that the efficiency by size of cyclone separators plunges toward zero as the particle size is reduced. Above 44 mu 100 per cent efficiency is easily attained. To the writer's knowledge no satisfactory means has been devised for separating these conglomerate particles and reproducing the original finely divided mixture of suspended dust. When using an elutriator for size analysis, it takes many hours and often days of operation before all of these conglomerate particles are finally broken up. Therefore, the greater accuracy of measurement obtained when testing a single cyclone loses its significance when there is no certainty that the dust-gas mixture being used is truly representative.

H. E. Macomber.[7] The author has proposed the idea that the performance of a dust-collector installation in the field, consisting of a number of collecting units operating in parallel, each of essentially identical size and design, will duplicate the individual performance of any one of the units of the assembly, or of a similar unit outside the assembly. The data presented for the conditions stated in the paper bear support to the idea.

The writer, however, looking at the problem from the viewpoint of the user of dust-collection equipment, wishes to bring the following thoughts into the picture:

1 A shop test or a field test of an individual unit should not supplant a field test of the entire equipment in position, even though the field test may be more difficult of attainment as to a comparable degree of accuracy. That is, the purchaser wants to know the degree of performance he is receiving from the assembled apparatus in his plant, regardless of the performance similar equipment may have shown to be possible in the shop, or which might be expected following the results obtained from tests of an individual unit of like size and design in the field.

2 Conditions under which dust-collection equipment is actually required to operate in some instances may vary considerably from day to day, as well as from the conditions previously set up as the typical basis upon which the guarantee is computed.

3 Further, every dust collector is affected to some extent by variation in the density and size distribution of the dust particles, volumes of gas or air to be passed, arrangement of ducting preceding the collector, etc.

Thus, briefly, the situation as it concerns the ability of a collector to meet a specified value of removal efficiency seems to be that, generally, the lower the value of the design-point efficiency, the greater will be the deviation from that value if the actual dust sizing or consistency as found is such that it contains an increased or decreased fraction of fines.

The point which the writer wishes to emphasize is that, in the purchase of a collector, the field test, which would probably accompany its operation before acceptance apparently assumes greater importance as the guaranteed-performance ability of the collector to be purchased is decreased.

4 Even so, some manufacturers may state a guarantee on the basis of total collection efficiency and pressure loss for a typical condition, without regard to possible variation in field conditions, and some purchasers may buy the equipment with the understanding that the guarantee value of total collection efficiency will be maintained regardless of field conditions. On the other hand, other manufacturers definitely state any guarantee of total collection to be subject to adjustment according to the actual conditions and make-up of the gas- or air-borne dust encountered in the field.

Considering the points here outlined, it is the writer's opinion that, although the performance of an individual cyclone unit was indicative of the performance of multicyclones, when handling caught dust under the stated conditions, he would not, as a user, be satisfied that the results so determined should be considered as applicable to the performance of field-assembly installations in position.

[7] Engineer, Production Department, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

R. F. O'Mara.[8] Since the subject of this paper is one in which we have been interested and on which we have been carrying a continuing research program over the last 12 years, we feel that the tests as discussed represent a good beginning of such an investigation, but that they are quite academic and too sketchy to justify the broad conclusions which have been drawn. On the whole, however, our many tests both in the laboratory and in the field substantiate the broad conclusions outlined in the paper.

In carrying work from the laboratory to the field or when making tests on an installation in the field on which to base selection of the final equipment, it has been our experience that the same size collector in physical dimensions as is to be used in the final installation must be tested and that, if this is done, the results from a single tube can be reasonably well checked with an installation consisting of a large number of identical small tubes. The following results are taken from check tests on a single tube followed later by actual installations:

Ventilating Air From Clinker Grinders

Preliminary Test. One 9-in. tube; efficiency 93.8 per cent at 2.8 in. pressure drop

Installation. 18 tubes; efficiency 92.4 per cent at 2.9 in. pressure drop

Asphalt-Mixing Plant

Preliminary Test. One 9-in. tube; efficiency 94.7 per cent at 2 in. pressure drop

Installation. 20 tubes; efficiency 94.5 per cent at 3 in. pressure drop

Pulverized-Coal-Fired Boiler

Preliminary Test. One 9-in. tube; efficiency 85 per cent at 2 in. pressure drop

Installation. One hundred and eighty 9-in. tubes; efficiency 85.6 per cent at 2 in. pressure drop

Cement-Kiln Gases

Preliminary Test. One 9-in. tube; efficiency 86.6 per cent at 3.3-in. pressure drop

Installation. Two hundred and eighty-eight $10^1/_4$-in. tubes; efficiency 85.5 per cent at 4 in. pressure drop

Gases From Fuller's-Earth Treating Furnace

Preliminary Test. One 6-in. tube; efficiency 92.8 per cent at 2 in. pressure drop

Installation. Thirty-six 6-in. tubes; efficiency 92.5 per cent at 2 in. pressure drop

[8] Western Precipitation Corporation, Los Angeles, Calif. Mem. A.S.M.E.

In multiple operation of cyclonic separators study of the header design to provide good distribution both of the gases and the suspended solids is equally as important as the hopper construction.

The discussion in the paper regarding sampling technique indicates lack of knowledge or investigation of the refinements in sampling technique that have been developed in other industries, particularly the metallurgical industry, both for ferrous and nonferrous metals. Here, guarantees are required on recovery of gold, silver, and precious metals from smelter gases and 1 per cent more or less in efficiency may mean losses of thousands of dollars a year. Efficiencies are not based on particle size but on a basis of the percentage recovery of the actual values in the escaping dust. Ofttimes these values lie in the finer rather than in the coarser material.

Power-plant engineers could take a leaf from the metallurgical engineers' refinements in sampling techniques for gases carrying suspended matter in the solid or liquid state. The following bibliography is given on the measurement of suspended solids and gases:

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7 "Determination of SO_2 and SO_3 in Flue Gases," by R. J. Nestell and E. Anderson, *Industrial and Engineering Chemistry*, vol. 8, 1916, pp. 258–260.

8 "Determination of Suspended Matter in Gases by Collection on Filter Paper," by S. H. Katz and G. W. Smith, U. S. Bureau of Mines Reports of Investigations, Serial 2378, 1922.

J. M. DallaValle.[9] The characteristics and limitations of multicyclones have never been treated comprehensively. This has been due in part to their relatively recent introduction in the field of dust collection, and perhaps also to inability to generalize performance in terms of tests carried out with a so-called "standard" dust. Mathematically, the behavior of a particle in a cyclone is easily determined, but anyone having even slight experience with particulate matter knows well that its true dynamical behavior depends upon other considerations than a knowledge of accelerating forces alone. A discussion of some of the factors involved in the development of an ideal cyclone was presented by Lissman[10] several years ago. In particular, Mr. Lissman discusses the design arrangements necessary to reduce turbulence in the dustbin in order to assure a minimum of interference with other cyclones (in parallel) connected to it. This paper emphasizes what is actually achieved in this connection under operating conditions.

The fact that a common dustbin with cyclone control dampers set in various positions does not materially affect the efficiency of dust collection (as compared with individual bins) is heartening to dust engineers who have heretofore possessed no definite information in this connection and in fact have often questioned its use. More striking than this, however, is the fact that changes in the control damper setting in no way appreciably affect the collection efficiency of the cyclone. This, as the author notes, is an important consideration in boiler plants equipped with multicyclone collectors where air volumes handled vary greatly, but draft losses must not be increased.

The writer expresses the hope that the author and others interested in the performance of multicyclones will make available their knowledge of the behavior of other dusts than fly ash, and perhaps decide upon a uniform performance test to be used in the laboratory. The testing of equipment after instead of before installation is not only costly to all parties concerned in event its performance is unsatisfactory, but also confuses the prospective purchaser who as a rule cannot judge the merits of various designs tested in a score of different ways.

[9] In charge, Environmental Sanitation Section, U. S. Public Health Service, Bethesda, Md.

[10] "An Analysis of Mechanical Methods of Dust Collection," by M. A. Lissman, *Chemical and Metallurgical Engineering*, vol. 37, 1930, pp. 630–634.

Author's Closure

Mr. Bubar's observations appear to be mainly intent upon the necessity of dust sampling in the field in preference to testing one cyclone.

It was not the intent in this research to make such comparisons, and it might be stated that the author is highly in favor of field test. It was intended in this research to give the results in a specific instance of testing one cyclone or testing three cyclones under accurate test conditions. If all of the information supplemental to this simple series of observations were included, as Mr. Bubar feels would be of interest, the amount of data presented in this report would have been far beyond its intended scope, or the time allotted for the presentation.

Mr. Macomber's observations also bring out the point concerning field tests compared to single cyclone tests. As observed previously, this was not the intent of the report.

The author thoroughly agrees with the point brought out by Mr. Macomber that an adjustment should be included in guarantees for variations in dust fineness that may be encountered in practice, over that assumed before the installation is installed. The author's company, however, has had some difficulty in getting people to agree to such a plan, and consequently in general has offered lower operating guarantees than would be expected from their own experience, in order to cover such contingencies as mentioned by Mr. Macomber.

Mr. O'Mara's observations appear to bear out the conclusions of the report, and it is interesting to have such confirmation from another manufacturer.

Mr. Hardie has brought out some important points which add considerably to the information contained in the original paper.

He asks perfectly correctly whether it may not be possible that three cyclones with a common hopper operate successfully, whereas 40, or even 12, may not. This is a very just observation, and the author cannot say from experience whether, for example, 40, cyclones would operate the same as the 3 in this test. In commercial practice, however, the number of cyclones connected with one hopper is necessarily limited. In view of the results of this test, therefore, any group connected with the hopper may be considered the unit. Four to nine cyclones are generally used per hopper. In one instance we know of twenty 2-foot cyclones per hopper, but this was unusual and exceptional, and is not to be recommended, especially as the hopper is extremely deep in such an instance because of the necessary angles of its sides.

Mr. Hardie states that tests of some installations of parallel operated cyclones do not always duplicate the performance of single units tested individually. There have been frequent occasions to compare field tests with tests of a single cyclone, and as far as can be observed, the rule is that they show a similar collection, and it is rather the exception that they do not. In such instances this may be due to duct layout, the settling of dust in the flues, and various other factors which sometimes make it difficult to approach the ideal from the standpoint of field testing, even with the greatest of skill being exercised.

As Mr. Hardie points out, the variable positioning of the cyclone dampers in the second series of tests is a condition more severe than in actual practice, but questions whether or not this reproduces the condition of different static conditions being created in the flue connecting a series of cyclones. He is probably correct that it is not precisely comparable. However, it has been the custom of the author's company to connect the inlets and outlets of the cyclones to large rectangular overhead ducts that are practically plenum chambers themselves. Many static readings have been taken at various parts of such ducts, and it has been found that the static condition is practically uniform throughout. It is, therefore, probable that the eccentric positioning of dampers in the test observations is considerably more severe than is ecountered in practice.

The author agrees with Mr. Hardie that it is preferable to operate a field test in accordance with the A.S.M.E. proposed code. The only advantage of utilizing a single unit test is that considerable information can be obtained by a prospective user, before the final installation is made, by attaching such a unit directly to his flue. Furthermore, cases may present themselves where it is an impossibility to obtain a field test because of the arrangement of ducts, the settling of dust in the flues, and other factors. In such an instance a single unit test might be valuable.

Testing of dust in the manufacturer's plant necessarily requires dust that has been previously collected. It has been found that if dust from the electrostatic precipitator is used, it is generally somewhat finer than the dust in the gas itself, and such dust appears to give about the same results as when a unit is hooked up directly to the flue. It may be that the fact that it is harder to collect, because of its greater fineness, is compensated for by some conglomeration.

In a series of tests run at the Stamford Gas and Electric Company a cyclone was attached to the flue. At 2 in. draft loss and an average of 289 deg an over-all collection of 88.1 per cent was obtained with pulverized-fuel dust. Dust was then taken from the electrostatic precipitator at the same plant, which analyzed 77.9 per cent through 325 mesh, and, at an average of 2.6 in. draft loss, 91.3 per cent was the efficiency with the same cyclone previously used at Stamford. The draft loss was somewhat higher; therefore a better collection should be expected. Furthermore, the gas temperature was atmospheric; hence about 220 deg less than at Stamford, which would further increase efficiency.

A further series of tests was then run in the research plant, using hot flue gas to carry the electrostatic dust. At an average draft loss of 2.1 in., and an average temperature of 433 deg, an average collection of 90.1 per cent was obtained with slightly finer dust, which averaged 80.01 per cent through 325 mesh. These tests are given primarily because they are on the same plant with the same dust, and might further be compared with seven hour tests, extending over 14 consecutive days, at the plant of the Industrial Rayon Corporation,[11] which had 30 cyclones of the same design per boiler. Here the draft loss was less, namely, 1.45 in., the temperature averaged 345 deg, and the collection showed 88.4 per cent. The screen analysis of the dust in the flue was especially fine, however, being an average of 88.6 per cent through a 325-mesh screen. The reason for this was that the boiler which was designed for 90,000 lb of steam per hour was averaging 52,000 lb per hour during these tests.

The author believes, therefore, that although the research reported in the original paper is limited necessarily, the observations elsewhere seem to bear out the conclusions with the particular cyclone tested.

[11] "Dust Collection Tests at New Power Plant of Industrial Rayon Corporation," by C. B. McBride, *Combustion*, September, 1939, pp. 36–38.

Penstocks for the Grand Coulee Dam

By P. J. Bier,[1] Denver, Colo.

The ultimate power development for the Grand Coulee Dam includes eighteen main units of 150,000 hp each and three station-service units of 14,000 hp. Individual penstocks of 18 ft diam were provided for the main units and of 6 ft diam for the station-service units. The penstocks are of welded-plate steel construction, fabricated and X-rayed in accordance with the API-ASME Code. The 18-ft penstocks were fabricated in a plant erected near the dam, and the 6-ft penstocks were produced in the subcontractor's shop. All penstocks were installed in octagonal tunnels provided for that purpose through the dam, and were embedded in reinforced concrete after installation.

General Description

The penstocks for the Grand Coulee Dam on the Columbia Basin project, Washington, were provided for the release of water under pressure for the production of power. The total estimated flow through the penstocks, at a rated head of 330 ft for the turbines, is approximately 82,000 sec-ft, based on the ultimate development of eighteen main power units and three station-service units. The main-unit penstocks are 18 ft in diam, serving turbines of 150,000 hp each, and the station-service penstocks are 6 ft in diam, serving turbines of 14,000 hp each, making a total ultimate capacity of 2,742,000 hp. The generating equipment will be installed in two separate buildings, one on the east bank of the river called the right powerhouse, and one on the west bank of the river called the left powerhouse. The two powerhouses are located at the downstream toe of the dam, and will contain nine main power units each, the left powerhouse including, in addition, the three station-service units.

All penstock intakes will be protected by trash racks arranged in semicircular form around the openings, and reaching up to within 21 ft of the crest of the dam. Hydraulically operated coaster gates will be installed on the face of the dam to shut off the flow from any of the main-unit penstock intakes until the turbines have been installed. The upstream ends of the main-unit penstocks were provided with hemispherical bulkheads to exclude the reservoir water from the penstocks during construction, and until the turbines are installed. After the installation, the intake will be closed with the coaster gate, and dewatered through a drain valve in the bulkhead, which latter is then flame-cut into small pieces and removed through the manhole.

Installation and Contracts

While the initial installation includes only three main units and two station-service units, the penstocks were installed for the full development. The Grand Coulee Dam is being built in two stages and under two separate contracts. The first contract included the foundation to an approximate elevation of 1000, and the second contract calls for the completion of the dam to elevation 1311. As it was not feasible to install the penstocks under the first contract, tunnels were provided which permitted the installation of the penstocks at a later date. These tunnels were of octagonal shape, 24 ft in size for the main-unit penstocks and 12 ft for the station-service penstocks. They were reinforced to resist the loads on the dam due to its dead weight and live water load, and for temperature effect.

The tunnels were started under the first contract, and completed to the upstream face of the dam under the second contract. The contract for the fabrication and installation of the penstocks, exclusive of the installation of the upstream sections containing the bulkheads, was awarded to the Western Pipe and Steel Company of San Francisco, at a bid price of $1,456,624 which included twelve 14-ft-diam pump inlet pipes. This sum, however, did not include the cost of transporting the pipes and pipe materials to the fabricating plant and to the dam. The estimated weight of all pipes furnished under the contract was approximately 8300 tons. The contractor sublet the fabrication of the station-service penstocks and the preparation and rolling of the plates for the main-unit penstocks to the Chicago Bridge and Iron Company.

The 18-ft-diam penstock, because of the impossibility of shipping pipe of this size, made shop fabrication impracticable, and it was necessary to erect a field-fabricating plant near the dam. Erection of this plant was started in April, 1938, and shop fabrication was completed in March, 1940. The installation of the penstocks was completed in May, 1940.

DESIGN OF PENSTOCKS

Main-Unit Penstocks

The penstocks are on a slope of $20^1/_2$ deg, starting with elevation 1041 at the upstream face of the dam and dropping down to elevation 938 at the turbine inlet, as shown in Figs. 1 and 2. The penstocks have an inside diameter of 18 ft which is reduced to 15 ft with a reducing bend near the turbine. The 15-ft-diam end section is connected to the turbine-scroll case with a double-acting expansion joint, as shown in Fig. 3, which will permit both axial and lateral displacements of the turbine in relation to the penstock. Axial displacements will be caused by temperature changes in the pipe and turbine, and lateral displacements by deflections of the dam and deformation of the foundation.

The inlet transitions are 30 ft long, and were designed to reduce inlet losses. They were formed in the concrete, and are provided with 30-in. air pipes, to facilitate the escape of entrained air when in operation or filling, and the inflow of free air when draining the penstock. The air pipe will also serve as a vent when the penstock is empty.

Each penstock is 329 ft long from the face of the dam to the scroll case, and is embedded in the tunnel backfill except for 10 ft around the expansion joint, where an access chamber was provided for the maintenance of the joint packing. This chamber can be reached by a passageway from the powerhouse. The penstocks were designed for a maximum head of 420 ft, including a static head of 360 ft and a water-hammer head of 60 ft. The head due to water hammer was computed in accordance with the rigid and elastic water-column theories on a basis of a minimum closure time of 4 sec at full gate, the turbine specifications calling for governors adjustable to a rate of turbine-gate movement of from 4 to 12 sec for a full-closing stroke.

The design of the penstocks was based on the use of firebox-quality steel plate, grade B, A.S.T.M. designation A-89, which has a minimum yield point of 27,000 lb, a minimum tensile

[1] Senior Engineer, Bureau of Reclamation, United States Department of the Interior. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

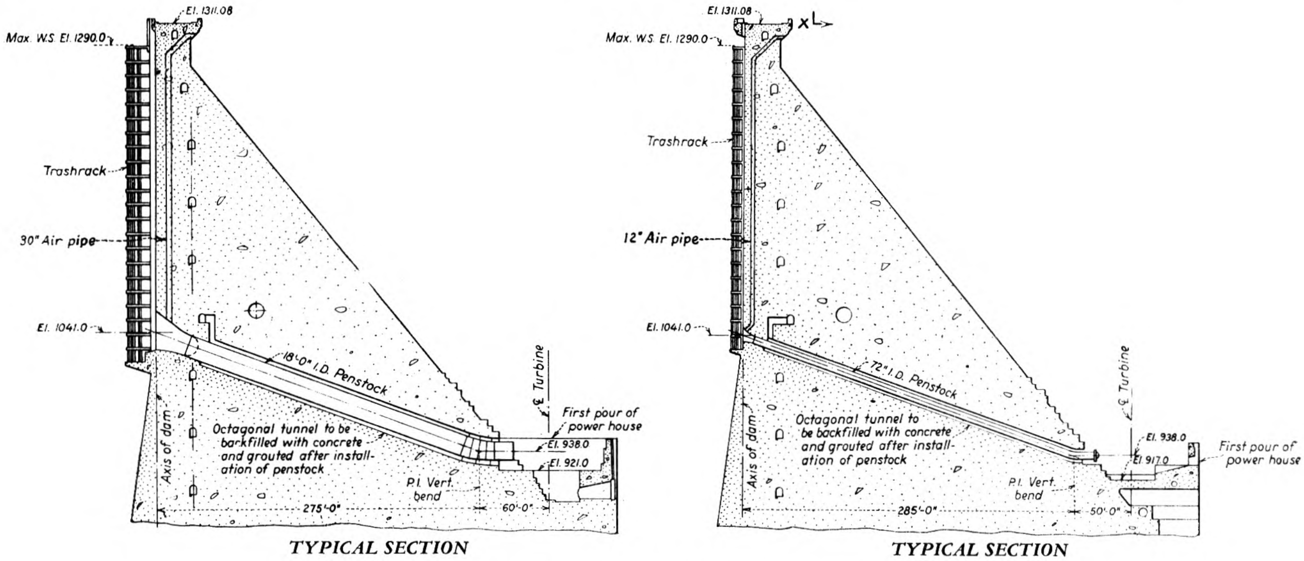

FIG. 1 PROFILES OF DAM AND PENSTOCKS

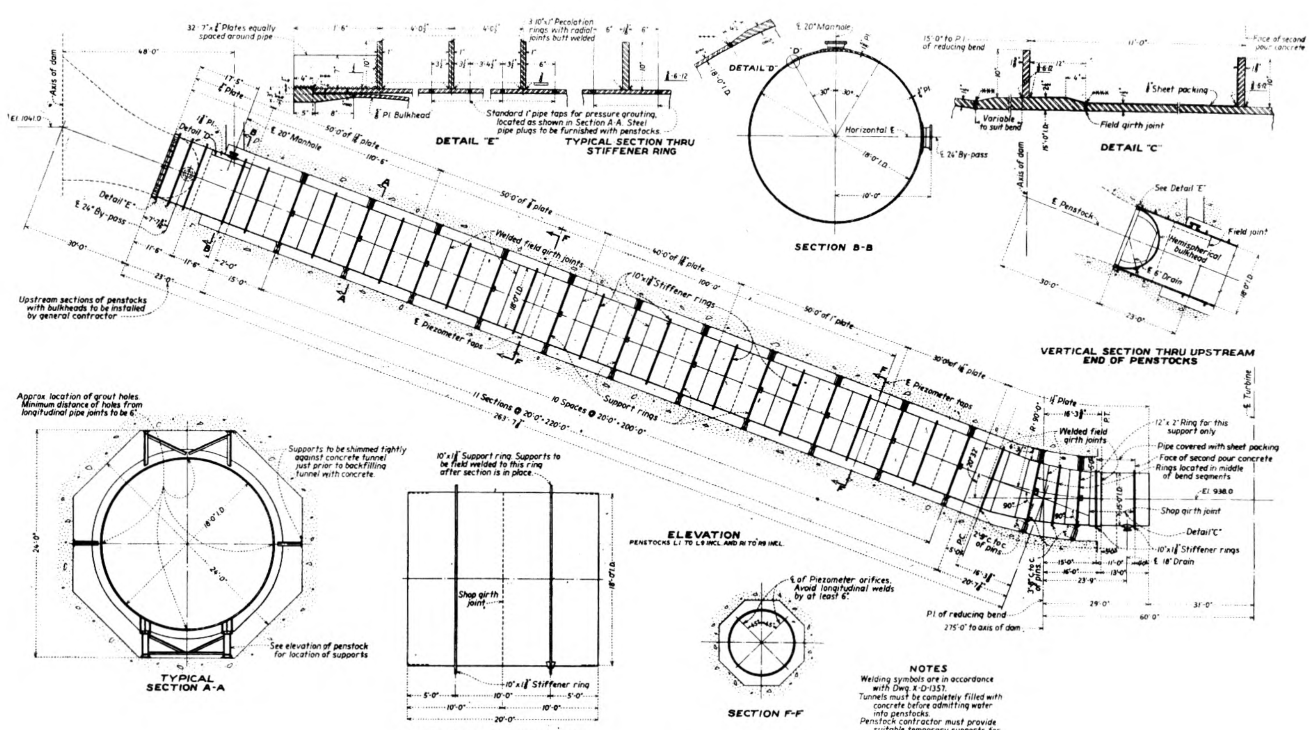

FIG. 2 DESIGN OF 18-FT PENSTOCKS

strength of 50,000 psi, and an elongation in 2 in. of 32 per cent. This steel was selected because of good welding qualities due to a low carbon content (0.20 to 0.22 per cent), which reduces the tendency of air hardening and produces ductile welds. Shocks due to water hammer and surge waves induce impact stresses which may be the cause of failure, especially in penstocks with brittle welds. With the use of this ductile steel, and due to the fact that the penstocks were to be embedded in reinforced concrete, thermal stress relieving of the pipe sections was not considered necessary. Radiographic examination of the welded joints, on the other hand, seemed desirable because it provides an effective check on the quality of welding, and permits an increase in joint efficiency from 80 per cent to 90 per cent, in accordance with the API-ASME Code for Fusion-Welded Unfired Pressure Vessels, which was adopted for this work. As an additional safeguard, a hydrostatic-pressure test was specified for all completed pipe sections.

Several schemes of installation were studied, and the scheme adopted consisted of embedding the penstocks in the tunnel backfill and painting the outside of the pipe and stiffeners to prevent a bond between the steel and concrete, thus permitting the pipe to "breathe" between empty and pressure conditions. A design stress equal to two thirds of the yield point was used for the fully embedded pipe having ample concrete coverage and

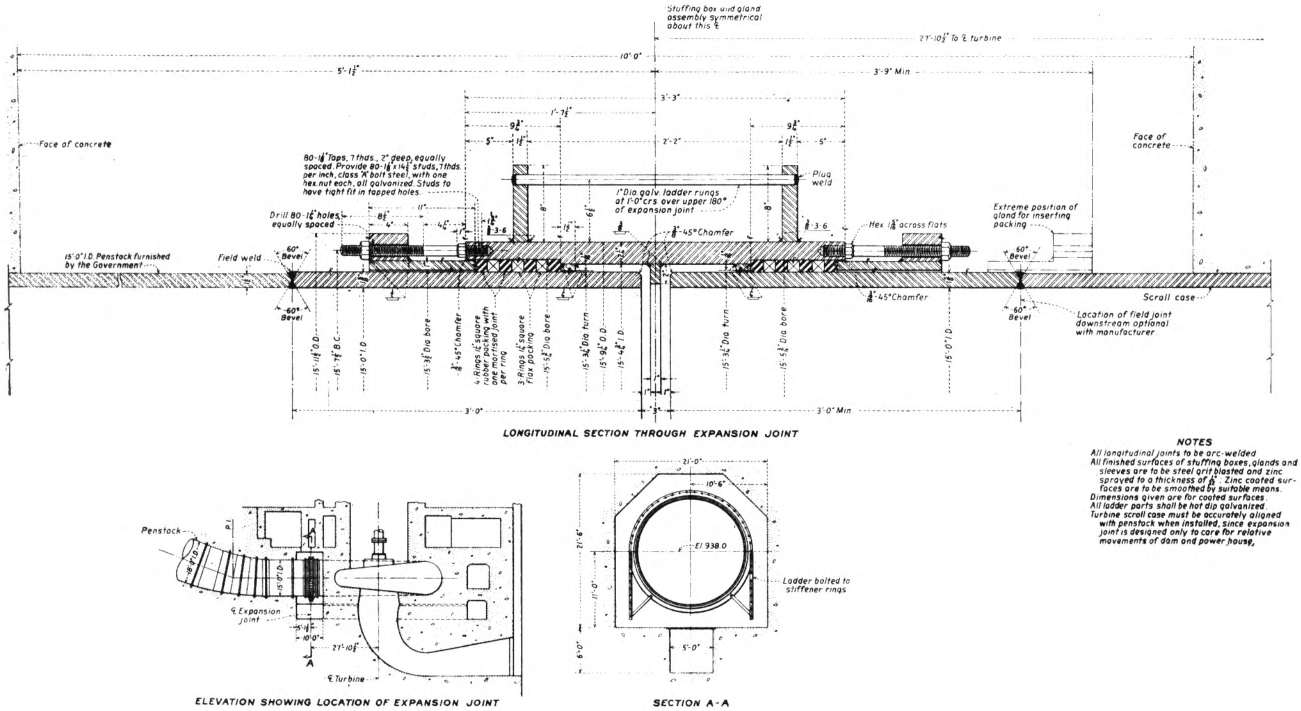

Fig. 3 Expansion Joint 15 Ft Diam

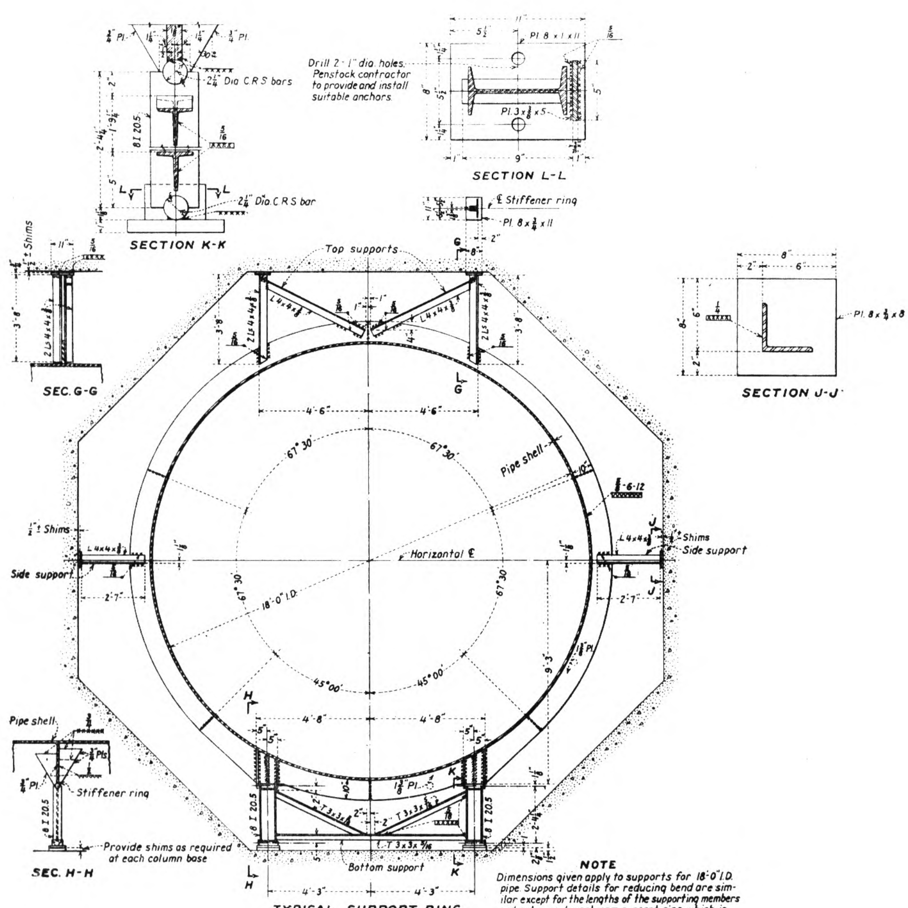

Fig. 4 Typical Stiffener Ring and Supports

reinforcement, and one half of the yield point for the pipe near the downstream toe of the dam where the concrete cover was not considered sufficient to aid the shell in resisting the internal pressure. The shell thickness was computed at $^3/_4$ in. at the upper end increasing by $^1/_{16}$-in. increments to $1^1/_{16}$ in. at the lower end of the tangent, and to $1^1/_2$ in. in the bend and lower end section.

Each penstock was provided with three percolation rings near the upstream end to retard the seepage of water along the outside of the shell. Single stiffener rings, $1^3/_8$ in. thick $\times$ 10 in. high, were welded onto the shell every 10 ft to maintain the circularity of the pipe section during fabrication, handling, and installation. Structural-steel supports, as shown in Fig. 4, were provided at every other stiffener ring, at the bottom and the top, the former to support the weight of the empty pipe during installation and the latter to protect the pipe against uplift

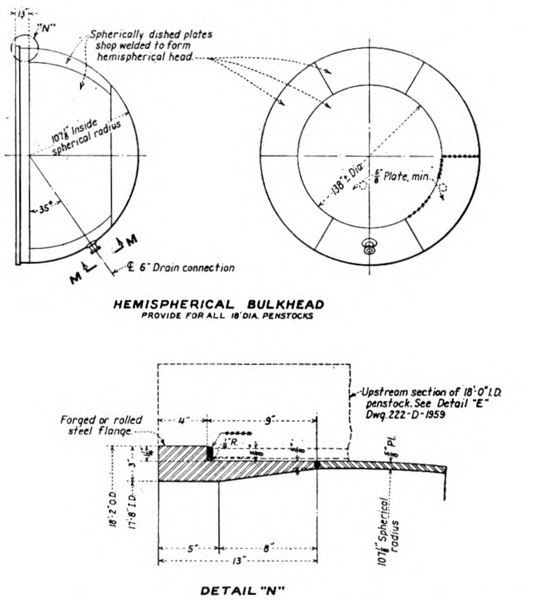

FIG. 5 HEMISPHERICAL BULKHEAD FOR 18-FT PENSTOCKS

during concreting in the tunnel. Lateral struts were provided at the supporting points to maintain the pipe in alignment while concreting. The lower supports were pinned at top and bottom to permit temperature movements in the line during erection.

The reducing bend was designed with a long radius and small deflection angles to reduce turbulence and head losses. A 24-in-diam by-pass pipe was connected to the upstream end of each penstock to provide means for filling the penstock, and to make it possible to operate the coaster gates under balanced pressure. The by-pass lines take water from the reservoir through a grated inlet and are controlled with double valves located in valve chambers near the 1050-ft gallery. The penstock is made accessible with a 20-in-diam manhole, placed on top of the pipe near the upstream end, which can be reached through a shaft from the gallery. Each penstock can be drained through an 18-in. drain at the downstream end of the pipe. The manhole and by-pass openings were reinforced to compensate for the loss in metal cut from the shell. The reinforcement consists of a plate ring welded to the shell, the nozzle being made from a pipe nipple and welding-neck flange. The manhole opening was streamlined with a plug welded from light-gage steel provided with drain holes to prevent unbalanced pressures. As the manhole is located in an open shaft, the pipe shell around the concrete will be subjected to secondary bending stresses; therefore, it was increased in thickness by inserting a $1^5/_8$-in. plate for the length of the course and for one sixth of the circumference.

The bulkheads placed in the upstream sections are hemispherical, and were welded from spherically formed plates and a central closing plate, as shown in Fig. 5. Special flanges were welded to the bulkhead shells for attachment to the upstream-pipe sections. These flanges were designed to resist the bending stresses due to the deformation of the shell when under pressure. The pipe shell opposite the flanges was stiffened with a number of ribs placed radially along the outside diameter. The hemispherical shape was selected as being the most efficient from the standpoint of stress, also because ellipsoidal heads for this diameter were not available, and could not be transported in one piece. The bulkheads were designed for a static head of 270 ft, requiring a $^1/_2$-in. shell, which was increased to $^5/_8$ in. to allow for corrosion and to provide additional safety in view of the importance of this water stop.

Two sets of piezometric orifices of four each, located 100 ft apart, were provided for the purpose of performing flow and pressure tests. The stainless-steel orifices were placed at quarter points of the shell. Individual lines of $^3/_4$-in. extra-heavy steel pipe were run from each orifice to the terminal box in the powerhouse, with one static-pressure line from the reservoir to the terminal. Grout holes consisting of 1-in. pipe taps, were provided in the penstocks, for the purpose of back-grouting and consolidating the concrete backfill. They were placed on the inside of the pipe at horizontal and vertical centers, on each side of the stiffeners and percolation rings.

STATION-SERVICE PENSTOCKS

The station-service penstocks are similar in design to the main-unit penstocks. The intake is located at elevation 1041, and the penstocks slope toward the turbines on an angle of about 20 deg, as shown in Fig. 1. Each turbine will be protected with a ring-seal gate. Penstock No. S3, which is to serve the future turbine, is provided with a welded bulkhead at the downstream end. A bulkhead gate will be used on the upstream face to close any of the three intakes when the turbines and ring-seal gates are installed. The three intakes were closed with temporary timber bulkheads before the reservoir level reached the intake elevation during construction. Between the lower ends of the penstocks and the ring-seal gates, double-acting expansion joints are installed, for similar reasons as for the main-unit penstocks.

The penstocks have a uniform diameter of 6 ft and are 332 ft long, from the face of the dam to the ring-seal gate. They were designed for a total head, including water hammer, of 400 ft, which resulted in plate thicknesses of $^3/_8$ in. in the tunnel and $^9/_{16}$ in. at the downstream end near the toe of the dam. The penstocks were fabricated in the subcontractor's Birmingham plant, in 20- and 40-ft lengths, ready for installation in the tunnels. Each penstock was provided with a manhole-and-drain connection, also with piezometer orifices, grout holes, percolation rings, stiffeners, and supports, similar to the 18-ft-diam penstocks.

FABRICATION OF MAIN-UNIT PENSTOCKS

FIELD-FABRICATING PLANT

The field-fabricating plant was located near the construction railroad in a settlement called Electric City, located about $3^1/_2$ miles from the dam. The plant consisted of a storage yard for incoming pipe materials, a welding shop, and a yard for testing, painting, and the storage of completed pipe sections. The welding shop was of frame and corrugated-steel construction, 70 ft wide, 260 ft long, and 45 ft high under the roof trusses. Adjoining the shop building on one side was an annex containing transformers and shop equipment, also offices for the shop superintendent and the government inspectors; on the other side of the building was an office for the X-ray technicians, and a de-

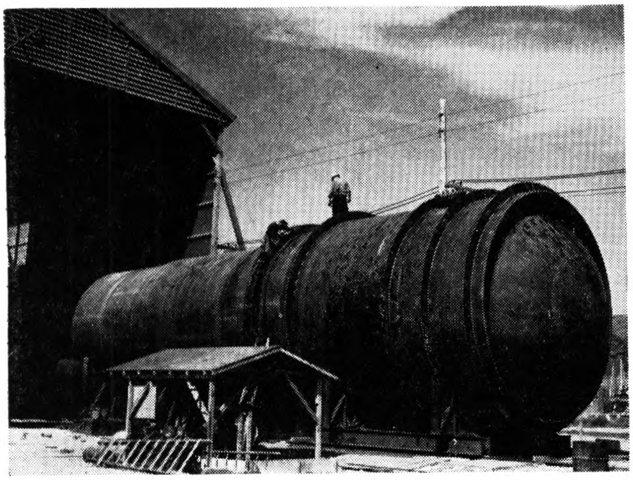

FIG. 6 TESTING MACHINE FOR MAIN-UNIT PENSTOCKS

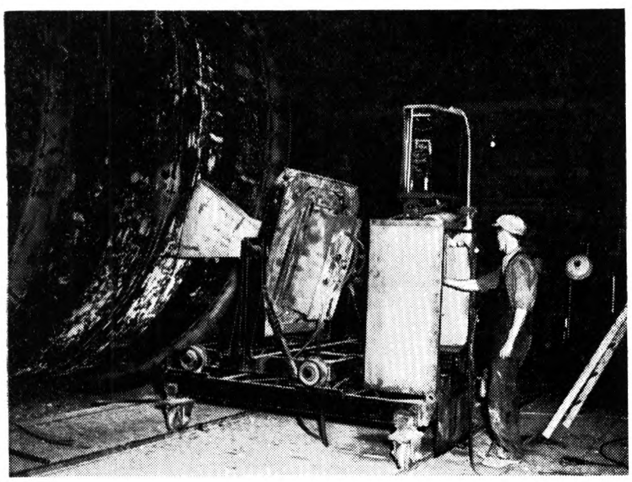

FIG. 7 PORTABLE X-RAY APPARATUS USED IN FABRICATING PLANT

veloping room for the X-ray films. The entire plant, including yards and shop, was served by a 35-ton gantry crane having a span of 66 ft 6 in. and a runway of about 1100 ft.

SHOP EQUIPMENT

The shop equipment consisted of four setting-up bases for the fitting and welding of the 18-ft pipe shell and stiffeners. Each base consisted of a series of concrete piers arranged in a circle and provided either with level steel platforms or with steel brackets or telescoping pipes on each pier, depending upon the fitting work to be done. A bulldozer, supported on a structural-steel traveler, was used to hold the two halves of the pipe or the pipe and stiffener together when tack-welding preparatory to the final welding. A motor-driven trolley and a roller base were installed for the movement and rolling of the pipe, during the welding of the longitudinal and circumferential joints, respectively. For the automatic-welding work, a welding machine was used which was supported from a structural-steel tower. A specially built testing machine, as shown in Fig. 6, was provided for the hydrostatic-pressure tests. The X-ray apparatus used is shown in Fig. 7; it was of portable construction with a capacity of 200 kv at 8 ma. A number of adjustable, rounding-out spiders, made of round bars with a central plate, were used for the pipe sections during construction and until the stiffeners were welded on. The auxiliary equipment in the shop annex included an air compressor and receiver, a lathe, planer, milling machine, drill press, and a Riehle testing machine.

SHOP PROCEDURE

The 18-ft penstocks were fabricated in sections or erection lengths of 20 ft, weighing up to 28 tons each, except the upstream-end sections containing the hemispherical bulkheads, which were produced in 23-ft lengths, weighing about 37 tons each. The pipes were made from two 120-in-wide plates which were beveled for welding, and rolled in the subcontractor's Chicago shop, then shipped to the field-fabricating plant for completion. The fabrication procedure for a 20-ft pipe section was as follows:

1 Two rolled plates were set up and aligned on one of the fitting bases and tack-welded into a cylinder.

2 A rounding-out spider was inserted into the 10-ft tack-welded pipe course, which then was taken to the trolley under the automatic welding machine, and welded along the longitudinal seam.

3 The stiffener rings, having been flame-cut in six segments from $1^3/_8$-in. plates and assembled on a special platform, were fitted to the 10-ft pipe course. Intermittent fillet welds, 6 in. long and 12 in. on centers, alternated on both sides, were used between the pipe and the stiffener ring.

4 Two 10-ft pipe courses were tack-welded together, then placed on the roller base, and completely welded on the automatic machine.

5 The welded seams were radiographed in 16-in. sections, marked along the joint. If no welding defects were disclosed in the films, the pipe section was ready for the pressure test.

6 The pipe section, after passing the visual inspection and X-ray examination, was placed in the testing machine, and subjected to a hydrostatic-pressure test of about 150 per cent of the design pressure, stressing the shell to approximately 21,000 psi. The test pressure was applied and relieved three successive times, and held a sufficient length of time to permit a thorough inspection of all joints for signs of leakage or failure. This pressure test proved its value as cracks from 4 to $9^1/_2$ in. long were discovered in five pipe sections, which apparently eluded detection by X-raying. Four of the cracks were near the junction of the longitudinal and girth seams where stress concentrations due to welding are more prevalent.

7 After the pressure test, the pipe section, with the exception of areas within 10 in. of the joints to be welded, was cleaned on the outside of all rust, dirt, grease, and loose scale, and painted with two coats of coal-tar paint applied cold. After the paint was dry, the section was ready for transport to the dam, and erection in the tunnels.

WELDING PROCESS

The contractor used the single-pass process of welding for all machine welds in the pipe shell, for plate thicknesses up to and including $1^1/_{16}$ in.; for all heavier plates, manual welding was used. Single-V welding grooves were used for the automatic welds, and double-V grooves for the manual welds. In the automatic welding process, the joint was welded in one pass on the outside of the shell, the welding head remaining stationary while the pipe was moved along the trolley for the longitudinal seam or rotated on the roller base for the circumferential seam. During welding, a water-cooled grooved copper backing-up bar was pressed against the underside of the shell opposite the welding head by means of a bracket attached to a pipe-post. The flux was deposited ahead of the welding operation and a coiled, bare electrode was fed in automatically, the welding speed and electric current having been regulated to suit the plate thickness. The lower layer of the flux covering the joint was fused into solid slag which protected the deposited weld metal against the atmosphere, and effected some annealing of the weld. Preparatory to making a longitudinal-seam weld, small plates of the same thickness as the shell were welded to both ends of the pipe course to form extensions of the welding seam. The starting plate, about 6 in. square, was in one piece and the finishing plate in two pieces, each $4^1/_2$ in. wide $\times$ $7^1/_2$ in. long, both beveled for welding and tacked together at the bottom of the groove. By this means, the tendency to crack, especially at the finishing ends of the longitudinal joints, was practically eliminated. After

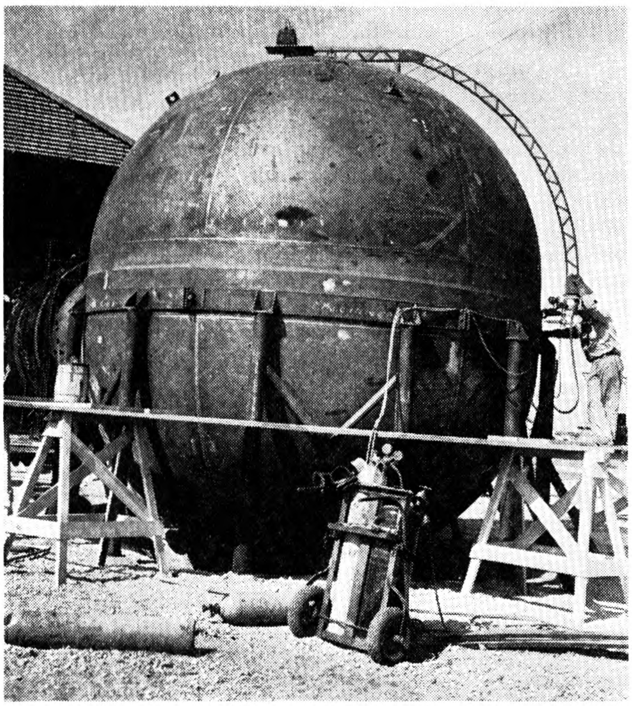

Fig. 8 Two Hemispherical Bulkheads Welded Together for Pressure Test and Being Separated by Flame-Cutting

welding, the extension plates were removed by flame-cutting, and the welding edges restored by grinding. The undersides of the single-pass welds were chipped to sound metal, and back-welded by hand, producing a double-welded butt joint as required in the specifications.

Before the automatic- and manual-welding processes were authorized, the contractor was required to furnish proof of their adequacy. Accordingly, two $^3/_4$-in. plates and two $1^1/_{16}$-in. plates were welded up by the contractor with the single-pass process, and two $1^1/_2$-in. plates with the manual process. Test specimens machined from the $^3/_4$-in. and $1^1/_{16}$-in. plates showed, for the weld metal, yield points and tensile strengths well in excess of the plate values, although the elongations, at 30 per cent, were somewhat lower. Similar results were obtained with test specimens machined from the $1^1/_2$-in. manually welded plates, which indicated average elongations of 28 per cent and strength values also well in excess of the parent plate.

The hemispherical bulkheads were pressure-tested by welding two bulkheads into a closed vessel, which was facilitated by the fact that the bulkhead flanges were rolled in pairs requiring only the welding of a head to each end of the double flange. A hydrostatic pressure of 240 psi, or about double the design pressure, was applied and relieved three successive times similar to the pipe tests. After testing, the two flanges were separated by flame-cutting, as shown in Fig. 8, and the bulkheads were welded into the upstream pipe sections of each 18-ft penstock, using a heavy butt weld between the end of the pipe and the underside of the flange.

Qualification of Welders and Weld Inspection

All welders engaged on manual welding were tested and qualified in accordance with the API-ASME Code. The contrator was required to furnish welded test plates both for the qualification of welders and for routine tests on production welding by the manual and automatic processes. For plates from $^3/_4$ in. to $1^1/_{16}$ in., inclusive, one test plate was furnished for each 200 ft of welded seam, and for the $1^1/_2$-in. plates, one test plate for each 100 ft of welded seam. The test specimens were machined and tested by the contractor, under the supervision of government inspectors. The tests included reduced-section-tension, free-bend, reverse-bend, and nick-break tests, and in some cases also all-weld-metal tension tests.

All shop and field fabrication and welding work was performed in the presence of government inspectors stationed at the plant, and all completed pipe sections were inspected and approved by the inspectors before being released for installation in the tunnels. The radiographic work was done by the contractor, using cellulose-acetate films of the slow-burning type, $4^1/_2$ in. wide $\times$ 17 in. long, with an effective length of 16 in., requiring 43 exposures for each girth joint. The exposure time was 20 sec for the $^3/_4$-in. plate, and $1^3/_4$ min for the $1^1/_2$-in. plate, requiring currents of 140 kv at 5 ma, and 175 kv at 7 ma, respectively. The developed films were examined by the contractor's X-ray technician and the government's chief welding inspector, for defects such as slag inclusions, cracks, porosity, or unfused areas. Defective sections of welds were traced on narrow strips of paper, located on the seam, and marked with yellow crayon for chipping. The chipped areas were rewelded, and reradiographed to prove the quality of the repair welding.

Installation of Penstocks

The specifications for the construction of the dam provided that the upstream sections of the penstocks be installed first to form a water stop for protection during the installation of the lower sections. Therefore, the upstream sections were installed and embedded by the dam contractor during concreting of the dam. Fig. 9 shows several of these with their hemispherical bulkheads welded in place preparatory to embedment in the dam.

The pipe sections for the right or east-side powerhouse were transported from the fabricating plant by the construction railroad, as shown in Fig. 10, to the government warehouse where they were unloaded onto a special 60-ton, 20-wheel trailer drawn by a truck in front, and on the downgrades braked by a truck in the rear. They were then hauled to a point on the riverbank where the trailer was run onto a barge, which was moved across the tail bay to the base of the powerhouse by a winch on the barge. The pipe was transferred from the barge to a tunnel trolley by a barge derrick, as shown in Fig. 11. In the tunnel, the pipe section and the trolley were pulled into place on a 6-ft gage track by an electric hoist placed at the downstream end of each tunnel. The trolley was equipped with jacks and rollers with which the section was aligned, centered, and set to grade from the bench marks previously established by government field parties. Before the tunnel trolley was withdrawn, the pipe section was placed on structural-steel supports which were hinged at their connections with the pipe and with base plates. The bases were shimmed to the proper elevation, and anchored against lateral movement, after which sufficient tack-welding was completed.

The pipe sections for the left or west-side powerhouse were transported by the construction railroad directly into the power-

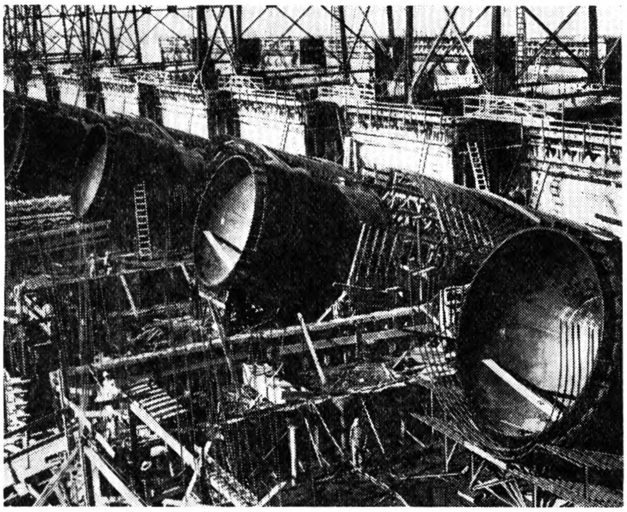

FIG. 9 UPSTREAM SECTIONS OF 18-FT-DIAM PENSTOCKS IN PLACE IN DAM

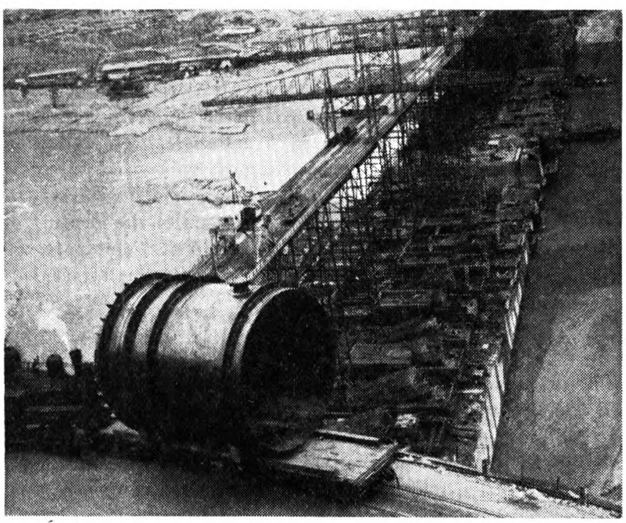

FIG. 10 RAILROAD TRANSPORT OF MAIN-UNIT PENSTOCK SECTION

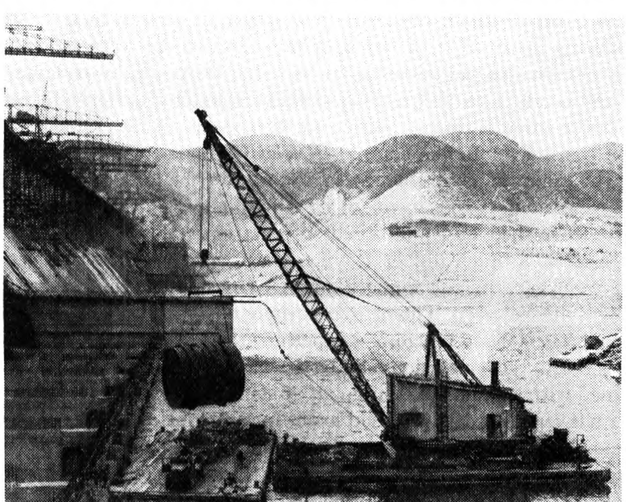

FIG. 11 HOISTING MAIN-UNIT PENSTOCK SECTION FROM TRANSPORT BARGE BY BARGE DERRICK

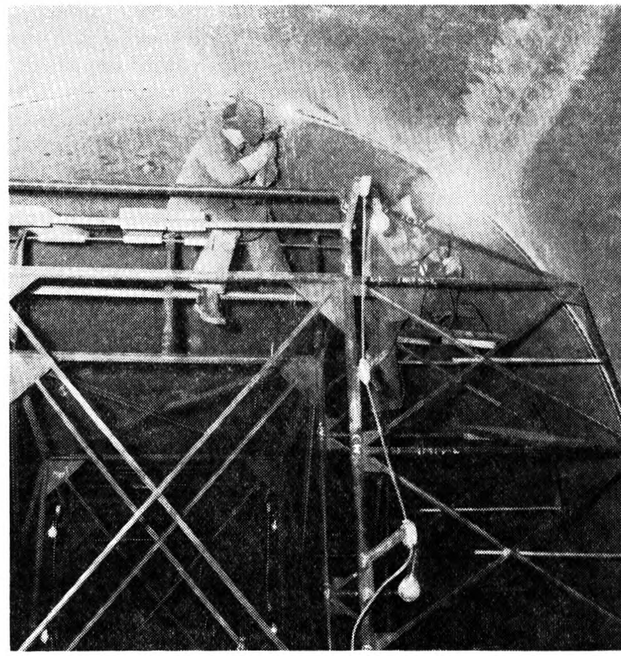

FIG. 12 WELDING OF ERECTION JOINTS FOR 18-FT-DIAM PENSTOCKS IN TUNNEL

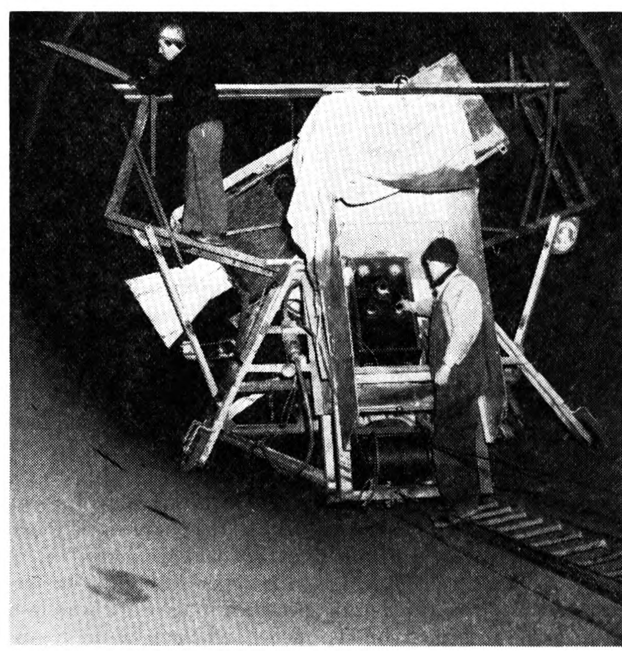

FIG. 13 X-RAYING TUNNEL WELDS IN 18-FT-DIAM PENSTOCKS

house itself, where they were lifted off the railroad cars by an electric overhead traveling crane which moved them to their respective tunnel trolleys, at which point they were routed in the same manner as the sections in the east powerhouse.

All tunnel joints were double-V butt welds, no butt straps being used at any seams. For the welding work on the inside of the pipe a portable steel scaffold as shown in Fig. 12 was used. With the exception of the final bead, each was cleaned and peened, the average peening time being 1 hr per bead around the circumference.

All the tunnel welds were radiographed in a manner similar to the shop welds, using a portable X-ray machine mounted on a rubber-tired trolley, as shown in Fig. 13. Generally speaking, all welding in each tunnel was completed before moving an X-ray machine into it. However, as the work approached completion, X-ray equipment and welding equipment were used in the same tunnel, particular care being taken to protect the workmen from scattered radiation. In all, 51,000 ft of welding were X-rayed, including the shop welds; and a total of 60,000 ft of X-ray film was used. There were approximately 300,000 lb of electrodes used.

After the completion of welding and X-raying, the paint coating on the outside of the pipe was repaired, the top and side supports were welded to the stiffeners, the piezometer piping was installed, and the reinforcing steel placed before the concrete backfill was started. After the backfill was placed, it was pressure-grouted from the inside of the pipe, using tapped grout holes for the purpose. Additional grout holes along the penstock were drilled as required. The penstocks were painted on the inside with two coats of coal-tar paint, applied cold, which completed the installation.

The station-service penstocks were installed first, followed by the main-unit penstocks for the right powerhouse, and then the left powerhouse units were installed. Installation started May 22, 1939, and exactly one year later, on May 22, 1940, all installation had been completed, six months ahead of the scheduled completion date.

Organization

The penstocks were designed under the direction of the author. All penstock designs are made under the general direction of W. C. Beatty, mechanical engineer, and L. N. McClellan, chief electrical engineer. All engineering designs are under the general direction of J. L. Savage, chief designing engineer, and all engineering and construction work is under the general direction of S. O. Harper, chief engineer, Denver, Colo. All activities of the Bureau are under the general charge of John C. Page, commissioner, Washington, D. C.

Discussion

C. L. Barker.[2] This paper presents in an interesting manner the design, manufacture, and installation of the huge pipes which will carry the flow of the Columbia through the Grand Coulee Dam.

Hydraulic model tests were made of the entrance but no tests were made on models of the penstocks. It might have been wise to make tests of the linning surface of the penstocks, and perhaps of the bends and the expansion joints. If the head loss through the penstock could have been reduced 1 ft, the power saved would have been 9000 hp. Tests of this type would also have been of value in comparing model results with the prototype. Piezometer connections were made in the penstock to measure head loss. This makes it possible to build a model and compare the results between model and prototype. Too frequently, tests of this type are not made.

The radiographic method of material examination is interesting and important. For many years, the only tests which could be applied to materials were tests of strength. These were applied to specimens and destroyed the material. The hardness test was one of the first tests of material which did not destroy; the radiographic examination is the second method. Its use in industry has spread rapidly in the last 3 years.

The five failures which developed in the course of the hydrostatic tests are important. The author states that after the five failures had occurred, due to the hydrostatic tests, those portions of film covering the failed sections were re-examined to see if the difficulty lay in studying the film or in the method. In every case the film showed no sign of crack or failure. It is rather safe to presume in every case that these failures, since they failed to show in the film, must have been hairline cracks, the plane of which was in line with the X-ray beam.

Messrs. Townsend and Abbott, of the Bell Laboratories, have reported studies[3] on hairline cracks. This type of crack is common in castings as well as in welding plates. Their report states: "It was found that the limiting cross-sectional area of a crack that could be distinguished in $^1/_2$-in. steel plate (the thickness most frequently encountered in telephone work) under the best radiographic conditions was 0.000035 sq in., representing a crack having a depth of 0.007 in. and a width of 0.005 in. Because of the importance of the orientation of these very fine shrinkage cracks, upon their detection by X-ray methods, it is sometimes desirable and often imperative to take two or more radiographs at different angles, in order that the defects presenting too small a difference in thickness for detection in one of them may be detected in the other, where the path of the radiation through the defect may have happened to coincide more nearly with its longitudinal axis."

This method, which by the way has been used in Germany, suggests the necessity for a slightly different method of radiographic examination which would eliminate the difficulty referred to. It should be possible to build an X-ray machine, using two tubes so spaced that the radiation paths would cross through the weld or section to be examined, at an angle. The radiation would be received on a film, perhaps slightly wider than the present film, but in the same manner. This would give a stereoscopic effect and make possible determination of the location of the defect in the weld or casting. It would also eliminate the possibility of failing to record the hairline cracks due to accidental alignment with the radiation path.

Author's Closure

The discussion presented by Mr. Barker is of timely interest as it touches upon such important features as model tests as an aid to a more efficient design and the use of the proper X-ray technique as a nondestructive test of the welding procedure.

While extensive hydraulic model experiments were necessary for the Boulder penstocks because of the complexity of the system, they were not considered of any particular value in the design of the Grand Coulee penstocks, as the latter consist only of straight runs of pipe without branch outlets or complicated fittings. The reducing bends at the lower ends of the penstocks were the only feature where head losses could be reduced by improved design. These bends were designed with small deflection angles producing a fairly smooth interior, using an R/D ratio of $5^1/_2$ which, on the basis of numerous bend loss experiments, is considered to be about the most favorable ratio, resulting in the lowest hydraulic loss. A model experiment for this specific installation could not have furnished any additional information unless it were performed at a much higher Reynolds number than that used for all bend experiments made to date. Considering the diversity of results between Thoma's experiments at a Reynolds number of 225,000 and Gregorig's experiments at a Reynolds number of 750,000, the conclusion may be drawn that bend loss values obtained with small-scale equipment at such low Reynolds' numbers are not representative of bend losses for large installations having Reynolds' numbers correspondingly higher.

The estimated friction loss through the penstock for a flow of 4500 cfs, at the rated head of 330 ft, is only 1.1 ft. This cannot

[2] Assistant Professor of Hydraulic Engineering, State College of Washington, Pullman, Wash.

[3] "Some Applications of X Rays to Industrial Problems," by J. R. Townsend and L. E. Abbott, *Metal Progress*, vol. 29, Feb., 1936, pp. 64–70 and 86.

be reduced unless the velocity is reduced by increasing the diameter which, however, has been determined from an economic study evaluating hydraulic losses and the cost of construction. The head loss in the bend, according to Thoma's experiments, is estimated at 0.3 ft. This, added to the friction loss in the pipe, would bring the total hydraulic loss to 1.4 ft, which is equivalent to 665 hp per unit. Any economies in head which may be possible by a different design of the bend would be so small as not to effect a worth-while saving.

It is well realized that the cracks which were discovered during the hydrostatic-pressure tests could have been detected by a double X-ray exposure from two different angles. Such procedure, however, would have greatly increased the already high cost of radiographic inspection on this job. In order to get the full benefit of this method of inspection, it would be necessary to use two exposures for every foot of weld, which would nearly double the cost. By checking the pipe sections with an excess pressure test at two thirds of the yield point of the metal, any deficiencies in X-ray examination were detectable, as proved by the cracks discovered during the tests. It may be of interest to mention here that this double-exposure examination was used on some of the repair welds for both the Boulder and Grand Coulee penstocks. The method was useful in such cases to prove the quality of the repair welds, also to disclose the depth of the defect below the surface. This made it possible to determine from which side the defect should be chipped out, reducing thereby the amount of chipping and rewelding necessary in the repair work.

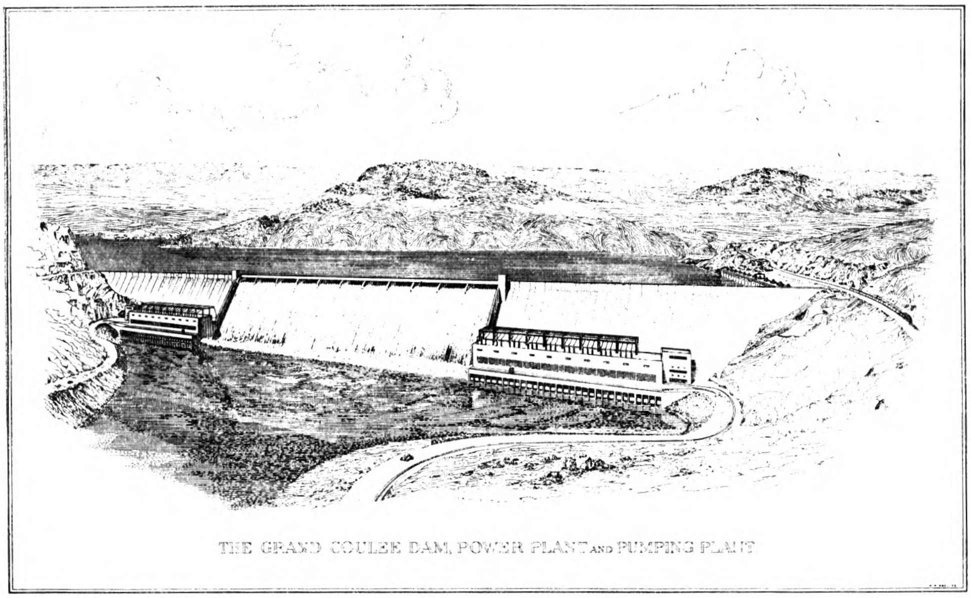

FIG. 1 GENERAL ARRANGEMENT OF GRAND COULEE DAM, POWER PLANT, AND PUMPING UNIT

Turbines for Grand Coulee Dam

By JAMES J. BURNARD,[1] DENVER, COLO.

The Grand Coulee power plant, when fully equipped, will be the largest hydroelectric power plant in existence, containing eighteen 150,000-hp main generating units and three 14,000-hp station-service units, or a total capacity of 2,742,000 hp. The turbines are of the vertical-shaft single-runner Francis type with spiral casing, supplied with water through individual plate-steel penstocks connecting with the upstream face of the Grand Coulee Dam. This paper describes in some detail the mechanical features of the main hydraulic turbines for this plant.

THE Grand Coulee power plant is located at the downstream toe of the Grand Coulee Dam on the Columbia River, about 94 miles north and west of Spokane, in the State of Washington. It consists of two powerhouses, one at each end of the dam, and is designed for the ultimate installation of eighteen 150,000-hp main generating units and three 14,000-hp station-service units.

Nine 150,000-hp turbines, driving 60-cycle main-unit generators of 108,000 kva each, will be installed in each powerhouse and, in addition, space is provided in the left or west powerhouse for three 14,000-hp turbines, driving 60-cycle service generators of 12,500 kva each, and for a control bay. The initial development includes the completion of the left powerhouse and the installation of three 150,000-hp main units, two 14,000-hp service units, and common station facilities.

The general arrangement of the project is shown in Fig. 1. Water is supplied to the turbines through a concrete and steel trash-rack structure at the upstream face of the dam. An individual 18-ft-diam welded-steel penstock with an 18-ft to 15-ft reducing bend at the turbine connection is provided for each main generating unit. A coaster gate, mounted at the upper end of each penstock and operated by a vertical oil-actuated hydraulic hoist, controls the water to the penstock and turbine. Each station-service unit is supplied by an individual 6-ft-diam welded-steel penstock with a shutoff gate of the ring-seal design at the inlet to the scroll case. The inlets to all penstocks are at elevation 1041, and the center lines of the turbine distributors are at elevation 938.[2]

HYDRAULIC CONDITIONS

The Columbia Basin project, when fully developed, will reclaim 1,200,000 acres of land, regulate the flow of the Columbia River, and develop electrical energy to be used for pumping for irrigation and for industrial purposes. Waters of the upper Columbia River are impounded from a drainage area of 74,100 sq miles into a reservoir 151 miles long, having a total capacity of approximately 10,000,000 acre-ft. The upper 80 ft of the reservoir, containing approximately 5,000,000 acre-ft, are available for power production and for the regulation of the river flow for the improvement of navigation and benefit of future power developments downstream from the dam.

The power generated from six of the proposed eighteen 150,000-hp units will be used during the high-water season to pump water into a balancing reservoir in the Grand Coulee where, by means of a system of canals and laterals, it will be distributed for irrigation purposes. High-water periods occur at such times that secondary power will take care of pumping needs. Therefore, all primary and some secondary power will be available for power

[1] Engineer, Bureau of Reclamation, United States Department of the Interior.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[2] "Penstocks for the Grand Coulee Dam," by P. J. Bier, published on page 219 of this issue of the TRANSACTIONS.

purposes for use at the project or for transmission to other consumers.

The flow of the Columbia River at the dam site varies from a minimum of 17,000 cfs to a maximum of 492,000 cfs, with an average flow of 109,000 cfs corresponding to an annual runoff of approximately 79,000,000 acre-ft. The elevation of the water surface in the reservoir fluctuates between elevation 1290, with flood storage capacity completely full, to a minimum at about elevation 1208. The surface of the water in the river immediately below the dam site, with a low-water flow of 17,000 cfs, is at about elevation 932, and with a discharge of 492,000 cfs is at about elevation 985. The center lines of the turbine distributors are set at elevation 938, and the elevation of the water surface in the tailrace varies from about 47 ft above to 6 ft below this elevation, the average being from 5 ft to 10 ft above it, Fig. 2.

The turbines operate under a net effective head varying from a minimum of 263 ft to a maximum of 355 ft; and for 90 per cent of the time the net effective head will be between 310 and 345 ft, with the weighted average net head of 330 ft.

Hydraulic Turbines

A study of reservoir operations during the critical periods of low runoff shows that, with the reservoir drawn down 80 ft, there will be sufficient flow to maintain a uniform power output of

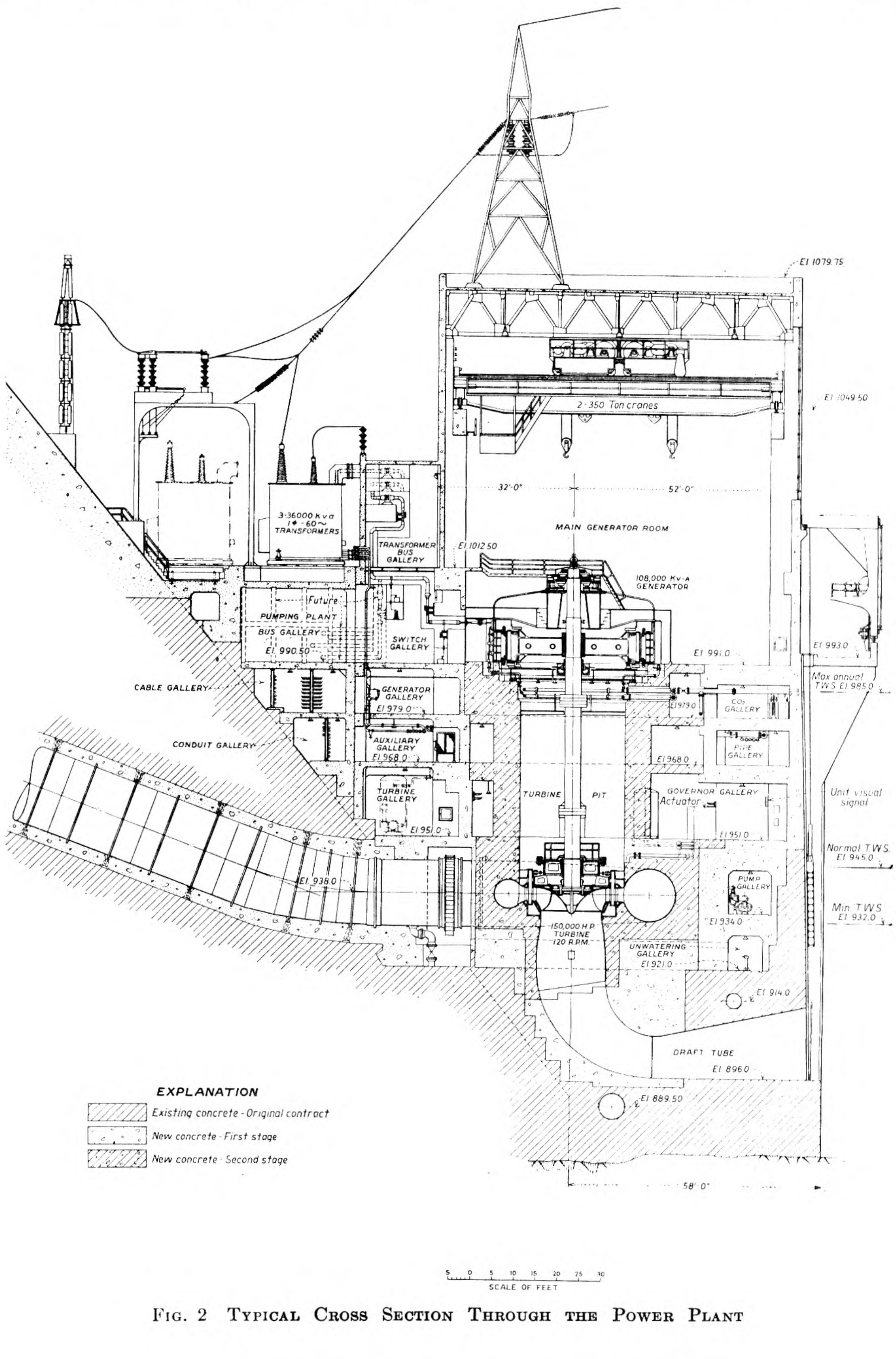

Fig. 2 Typical Cross Section Through the Power Plant

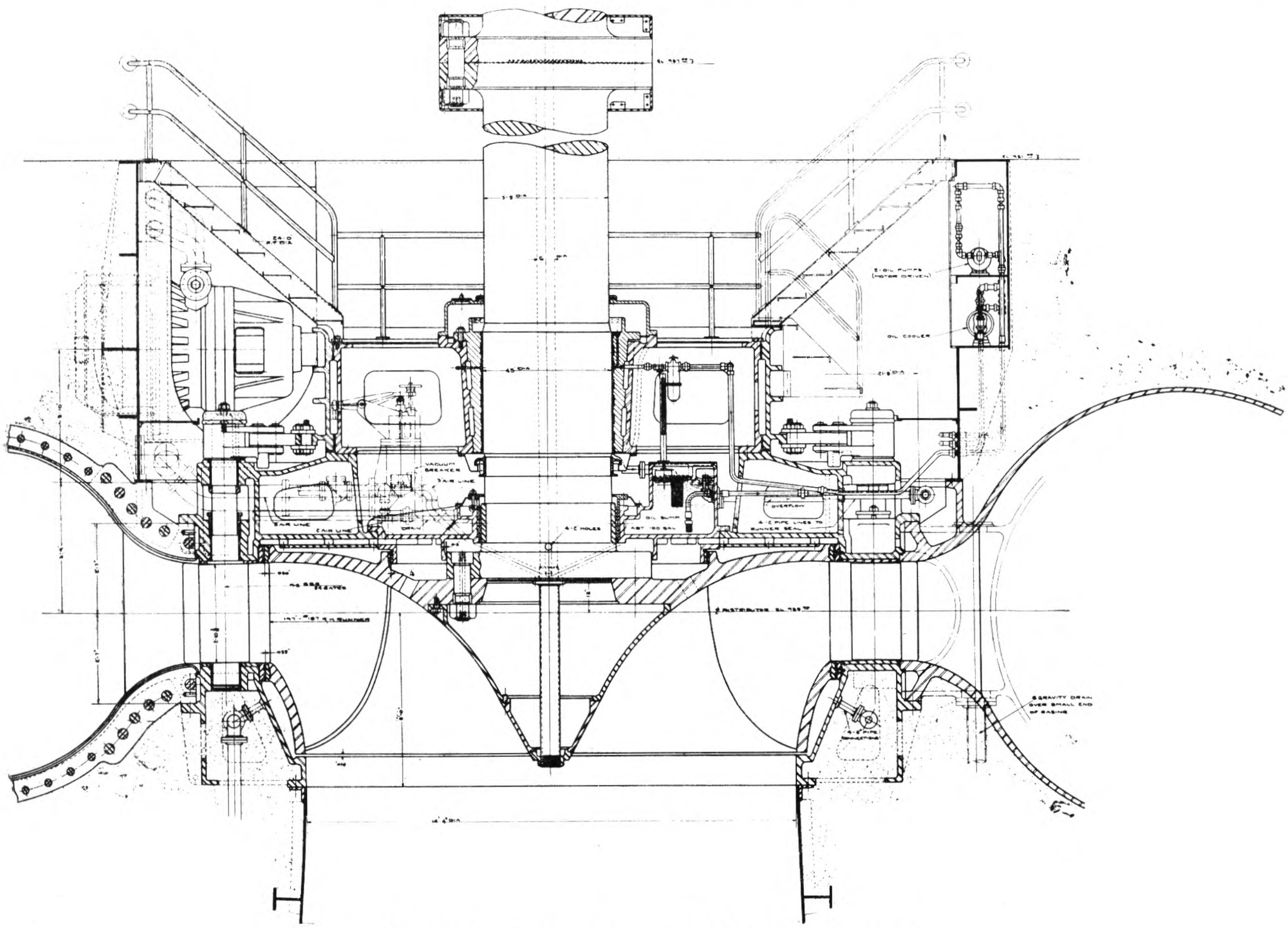

FIG. 5 CROSS SECTION THROUGH 150,000-HP TURBINE

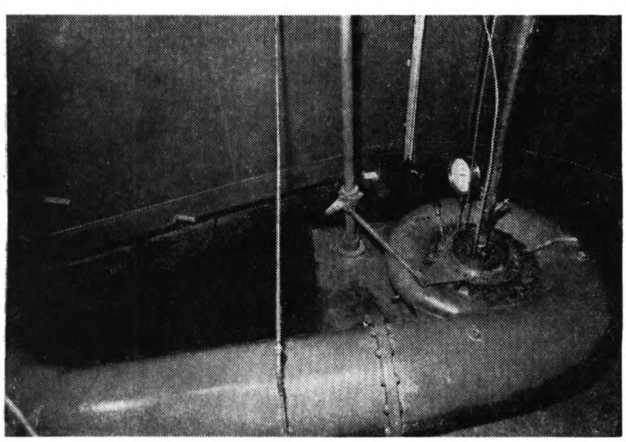

FIG. 4 TURBINE MODEL IN MANUFACTURER'S LABORATORY

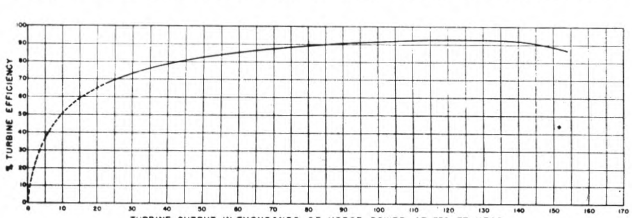

FIG. 3 CURVE SHOWING EXPECTED PERFORMANCE OF 150,000-HP TURBINE AT THE RATED HEAD

920,000 kw of which 800,000 kw will be used for the generation of firm continuous power, and the balance for secondary power for pumping for irrigation and for stand-by service. Load conditions for this power require the installation of sufficient units to bring the ultimate turbine installation to 2,700,000 hp.

The turbines selected for the Grand Coulee power plant are of the vertical-shaft single-runner Francis type with spiral casing. The three initial 150,000-hp turbines operate at a speed of 120 rpm and, with respect to horsepower capacity, are the largest hydraulic prime movers in existence. As shown in Fig. 3, they are designed for a maximum efficiency of about 93 per cent at a power output of 125,000 hp under the designed head of 330 ft. A model of the turbine is shown in Fig. 4. This model is homologous with the prototype (except for being of left-hand rotation to suit the manufacturer's laboratory). The homologous parts include the runner, casing, wicket gates, and draft tube. The runner diameter is 18.9 in. as compared with 197 in. for the prototype. Tests on the model were made under heads varying from 25 to 50 ft.

The arrangement and setting of the 150,000-hp turbine is shown in Fig. 5. A cast-steel spiral casing, cast integral with the speed ring, is embedded in the concrete substructure of the powerhouse, with the center line of the distributor at elevation 938. A plate-steel pit liner, to which the servomotors are bolted, lines the turbine pit and extends from the top of the casing to elevation 951. The upper and lower covers of the turbine are bolted to the speed ring, and contain the stationary wearing rings and plates and the bearings for the wicket gates which control the water to the runner. The upper cover also supports the single main bearing, stuffing box, and gate-shifting ring. The runner, of cast steel, is attached to the main shaft by means of a bolted, flanged connection. The draft-tube liner is bolted to the lower cover and extends to a point 22 ft below the center line of the turbine distributor.

Hydraulic Design

The velocity of the water entering the spiral casing is 29 fps, or approximately 20 per cent of the spouting velocity under the designed head, Fig. 6. This velocity remains approximately constant throughout the length of the casing, as the cross-sectional area of the casing is decreased proportionately to the increment of flow into the speed ring. The speed ring forming the throat of the spiral casing is spanned by fourteen fixed vanes, so designed as properly to direct the water to twenty-four movable wicket-type vanes or gates. The water, after passing through these wicket gates, enters the runner with a velocity of approximately 53 per cent of the spouting velocity under the designed head. The runner has nineteen vanes the curved surfaces of which, by changing the direction of the water, convert the energy of the water into power and produce a final discharge velocity at the throat of the draft tube of approximately 29 fps. The discharge of water from this point to the tailrace is through a flattened elbow type of draft tube where the decrease in velocity is such as to produce a residual velocity at the end of the draft tube of 6.8 fps. This residual velocity represents a final rejection 0.2 per cent of the total available energy.

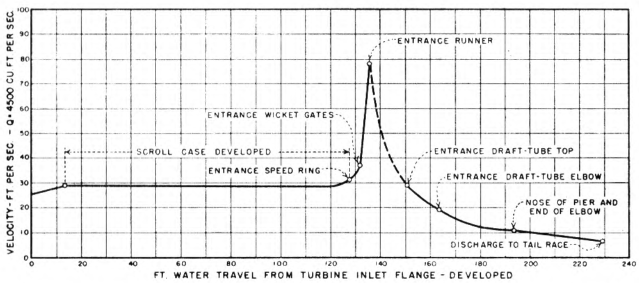

Fig. 6 Curve Showing Velocity of Water Through 150,000-Hp Turbine

Leakage of water between the crown of the runner and the turbine cover plate, and around the lower shroud of the runner, is minimized through the use of two plate-steel wearing rings which act as seals, Fig. 5. Means are provided, should the unit be operated as a synchronous condenser with the turbine gates closed and the water depressed below the runner, to supply the runner-seal chambers with penstock water. The essential use of this water is to act as a cooling medium to prevent heating and consequent expansion and possible seizing of the runner wearing rings to the stationary wearing rings located in the upper and lower covers of the turbine.

Hydraulic losses in the seal chambers are materially reduced by the elimination of all projections and pockets. A plate-steel baffle is provided on the upper cover just above the runner to direct any leakage water through the seal clearances to cored holes in the runner hub.

If future operation of the units as synchronous condensers is found desirable, provision has been made for the installation of float-operated air valves which will admit compressed air below the runner. The level of the water in the draft tube will be depressed to a point about 3 ft below the bottom of the runner, and thus materially decrease the power required to drive the runner.

Provision has been made for the admission of atmospheric air to the turbine cover plate under conditions of small gate openings to act as a draft-tube vacuum breaker and to remove water from the runner. This is accomplished by means of a poppet-type valve, automatically operated from the gate-shifting ring. An 8-in. pipe line connects the valve with the downstream face of the powerhouse so as to prevent the objectionable noise from an air inlet inside the turbine pit.

The penstock and scroll case are initially filled by means of a 24-in. by-pass, located at the upstream end of the penstock. The entrained air is discharged through a 30-in. pipe, embedded in the concrete, terminating at elevation 1302 at the downstream face of the dam.

Mechanical Design

The physical dimensions of the spiral casings for the Grand Coulee turbines are the largest ever attempted for cast-steel construction. In order to transport a casing of this size from the manufacturer's shop to the project, it was necessary to sectionalize it into fourteen parts with radial flanged bolted joints. The average section is 15 ft 6 in. × 14 ft 6 in. × 7 ft 6 in. and weighs approximately 41,000 lb. The total weight of the castings for the 14 casing sections of the turbine is 582,000 lb. The heaviest section weighs 59,000 lb. Most of the casing sections, the runner, and many other parts, will have to be shipped on special cars. A view of the casing assembled at the contractor's plant is given

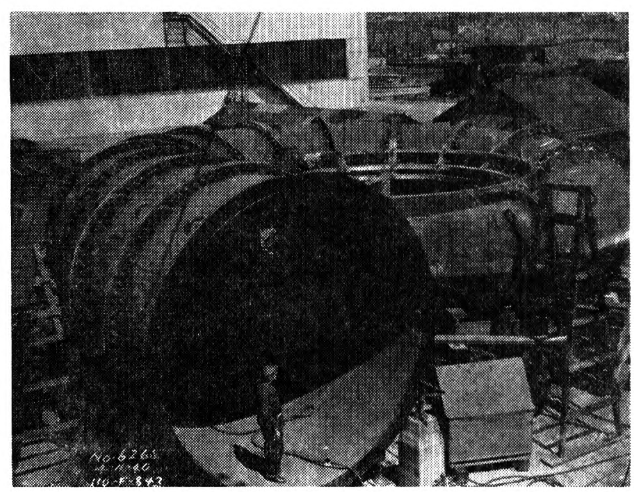

Fig. 7 Spiral Casing Assembled at Manufacturer's Plant

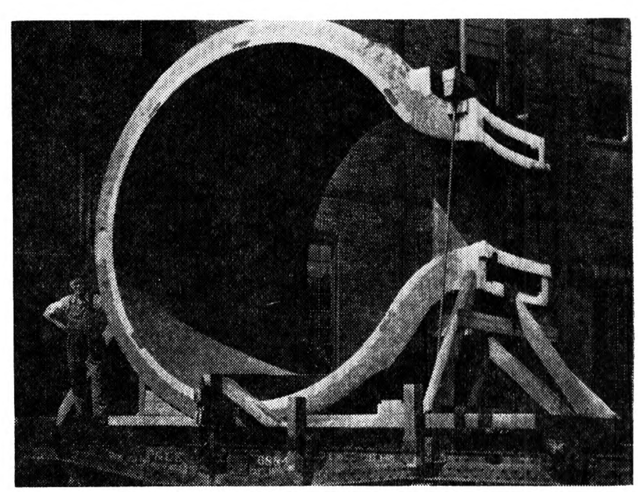

Fig. 8 Casting for Casing Section No. 7; Weight 38,000 Lb

in Fig. 7. A view of one of the sections being transported is shown in Fig. 8.

Owing to the large number of bolted sections, the support and anchorage of the casing, while filled with water under pressure during erection and embedding in concrete, presented a very interesting problem. After much study of various methods, it was finally agreed that the support by means of screw jacks bearing against steps cast on the flanges of the various sections would prove the most suitable. Turnbuckle rods, attached at one end to ears welded on the flanges and at the other end to loop bars embedded in the surrounding concrete, are prestressed to 12,000 psi to prevent any possible movement of the casing during the pouring operation. This method of support and anchorage is shown in Fig. 9.

All of the essential parts of the turbines are made of cast steel with an allowable design stress not to exceed 10,000 psi under normal operating conditions. Under emergency conditions, stresses up to one third the yield point of the material were allowed. The flanges of the scroll case, however, are an exception, having an allowable bending stress of 15,000 psi. It was found that, if the bending stresses were kept below the 10,000 psi, required by the specifications, the flange thicknesses would be so great that unsound castings would probably result.

Turbine Runner

The turbine runner is of the Francis type of cast steel made in one piece and is 197 in. diam. It has sufficient strength to support its own weight plus the weight of the turbine shaft, up to the connection with the generator shaft, with the runner resting on a finished ledge in the lower cover or foundation ring. It is designed and constructed to withstand safely the stresses resulting from operation at a runaway speed of 220 rpm under conditions of maximum head, with the turbine gates wide open and with no load on the generators.

The runner is provided with two steel wearing rings, one on the lower shroud and one on the crown. The wearing rings are shrunk on and machined on the outside diameter for a close running fit with stationary wearing rings in the top and bottom covers. Cored passages in the hub reduce the downward thrust of the water on the top of the runner, and the water passages are finished to a smooth surface to minimize friction and cavitation. A cast-steel runner tip with a removable cast-steel cap will allow inspection of the runner and shaft connection. The finished runner, complete with wearing rings attached, is to be statically balanced in the contractor's shop before shipment to the project. The runner casting being transported to the manufacturer's shop is shown in Fig. 10.

Shaft for the Main Turbine

The turbine shaft is made in two sections of forged heat-treated open-hearth carbon steel with coupling flanges forged integrally with the shaft. The material of the shaft has an ultimate strength of 70,000 psi and a yield point of 35,000 psi and, under normal operating conditions, is stressed to not more than 4700 psi. It is capable of operating at any speed up to a full runaway speed of 220 rpm without vibration or objectionable distortion. It is 44 in. in diam at the coupling end and extends 43 ft above the center line of the turbine distributor, where it connects to the generator shaft. The flanges at the ends of the shaft form a male and female bolted coupling with body-bound forced-fit coupling bolts. Axial holes are provided in the bolts to permit the use of a cooling medium for shrinking them for assembly and disassembly. A bolt cover made in halves encloses the nuts on each side of the coupling.

A 6-in-diam hole is provided throughout the length of the shaft for inspection purposes, with four 2-in-diam radial holes adjacent to the lower flange connecting with the central hole for

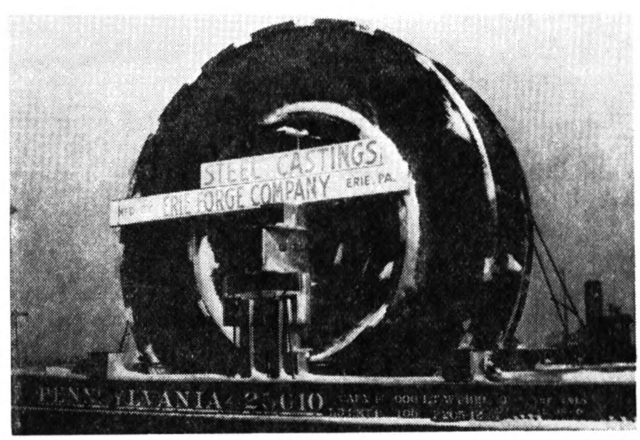

Fig. 10 Runner Casting for 150,000-Hp Turbine

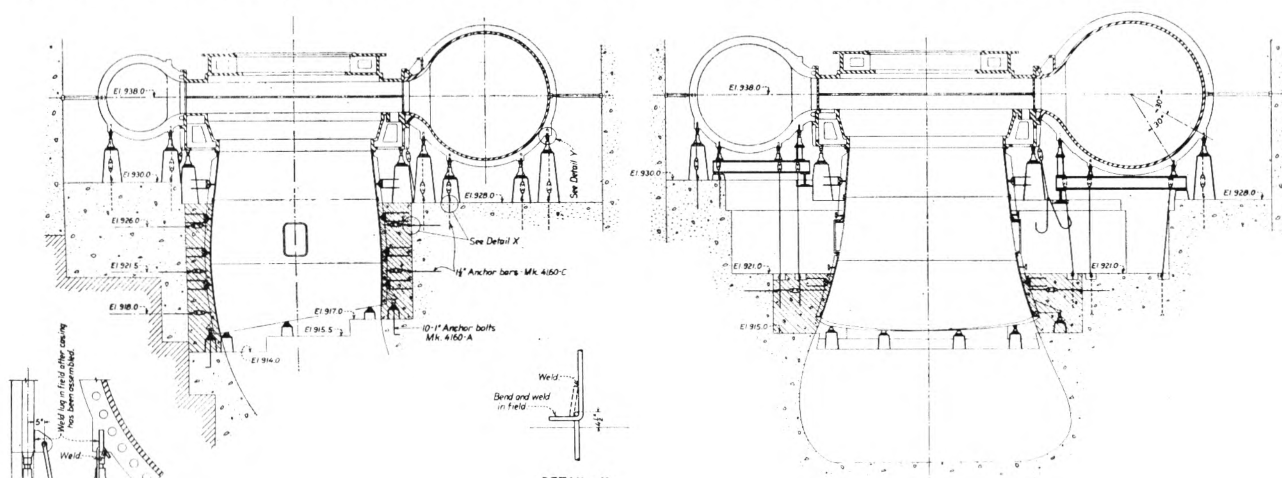

Fig. 9 Arrangement of Supports and Anchors for Turbine Casing

the purpose of admitting air from the cover plate to the tip of the runner. A renewable stainless-steel sleeve in halves is secured to the shaft, by means of shrink links, where it passes through the packing box.

Guide Bearing

The main-shaft bearing is of the babbitt, oil-lubricated type, 45 in. diam, with a length of $41^1/_2$ in., and consists of a semisteel shell, made in two sections, bolted to a bearing support. The bearing support is of cast steel in two pieces and is bolted to the upper cover. It will support the main bearing and act as a guide for the gate-operating ring. A pressure switch located in the oil groove in the bearing directly opposite the oil inlet will sound an alarm on low oil pressure, and a mercury-contact-type thermal relay with bulb located in the hot-oil discharge from the bearing oil pan will sound an alarm upon excessive temperature rise. A cast-steel oil deflector provided with an oil collector is secured to the shaft immediately below the bearing and drains the oil into the turbine oil sump which is formed in one side of the packing-box support.

Two duplicate lubricating-oil pumps, for circulating the oil through the bearing, are mounted in an alcove in the pit liner. The main pump is driven by a 60-cycle 440-v a-c motor, and the stand-by pump is driven by a 250-v d-c motor. Normally, the oil will be circulated by the a-c motor pumping unit, the d-c motor pumping unit being arranged to start automatically and supply oil to the bearing upon failure of the oil pressure.

Casing and Speed Ring

The casing is of spiral form, made of cast steel in fourteen sections, and is cast integrally with the speed ring which forms a tie across its throat. The sections are joined together by means of flanged, bolted joints of ample strength and rigidity, Fig. 7.

The speed ring containing fourteen fixed vanes is designed to support the weight of the superimposed parts and also to resist the upward thrust due to a pressure of 175 psi in the casing with no superimposed load.

The casing is substantially circular in cross section to prevent deformation under pressure and is designed so that all internal parts of the turbine may be removed from above, Fig. 5. It has an inlet diameter of 15 ft and extends upstream to the face of a finished flange a distance of 20 ft from the center line of the turbine, where it connects to an expansion joint. A 24 × 36-in. manhole with a hinged cover is provided in the casing for access to the interior.

The expansion joint made of plate steel connects the turbine casing to the penstock and consists of two separate pipe sections and an outside ring which form a packing box at the joint between the sections. The packing box is provided with alternate rings of rubber and flax packing held in place by steel glands.[2]

Covers

The top and bottom covers are each made of cast steel in two pieces to facilitate shipment and handling. The top cover is designed to support the weight of the main-shaft-bearing housing, the stuffing box, and the gate-shifting ring, Fig. 5. It contains two bronze-bushed bearings for each wicket-gate stem. Provision is made to carry the weight of the wicket gates and levers and any unbalanced hydraulic thrust by means of thrust bearings in the upper cover. The lower and inner faces of the top cover adjacent to the gates and runner are provided with finished surfaces for attaching renewable plate-steel wearing plates and rings.

An adjustable packing box, which may be repacked without disturbing the main bearing, is provided where the main shaft passes through the top cover. The packing box is lubricated by means of water supplied from the penstock through a reducing valve in an amount sufficient to effect lubrication without leakage into the turbine pit. Packing boxes are also provided where the wicket-gate stems pass through the top cover.

The bottom cover is combined with the foundation ring. It is bolted to the spiral casing and draft-tube liner and contains the lower bronze-bushed bearings for the wicket-gate stems. The upper and inner faces of the bottom cover are provided with finished surfaces for attaching renewable plate-steel wearing plates and rings, as in the top cover.

Wearing Rings and Plates

Each runner wearing ring is made from three carbon-steel bars with the ends welded together to form a ring. The ring is shrunk on the runner and in addition is secured by means of fillister-head screws with heads countersunk flush with the outside diameter. The stationary wearing rings in the top and bottom covers are also of carbon steel, but in addition each wearing ring has two inserts, of hard brass, calked into dovetail slots in the ring. The hard-brass inserts were decided upon as it was desired that two metals of different characteristics be used in the event of accidental contact between the rotating and stationary rings. The stationary rings are machined on the outside diameter and sufficient stock is left on each ring seat in the top and bottom covers to machine to a true circle by means of a boring rig during assembly at the project.

Gates and Operating Mechanism

The turbine gates are of the balanced-wicket type of cast steel with stems cast integral. Each stem is provided with three bronze-bushed bearings, one located in the bottom cover and two in the upper cover, one below and one above the stuffing box. The stem is bored throughout its entire length to supply grease to the lower bearing.

Each gate has a cast-steel lever keyed to its upper end which, by means of a steel link, is connected to the gate-shifting ring. The link is connected to the lever by a semisteel shear pin designed to be the weakest element in the gate mechanism. This pin is strong enough to withstand the maximum operating forces but will break and protect the rest of the mechanism from injury in the event that one or more of the gates becomes blocked.

The gates are so designed that in case of breakage of the shear pin the movement of the gate is limited so as to prevent interference with the operation of other gates or the runner. Each gate is also provided with a thrust collar to carry any upward thrust due to hydraulic pressure, and is suspended in mid-position between the upper and lower covers by means of a thrust washer.

The gate-shifting ring is of cast steel in one piece, of rigid design, and is guided by renewable bronze guide strips on the top cover and bearing housing. It is connected at diametrically opposite points, through adjustable forged-steel connecting rods, to the servomotors.

Pit Liner

The pit liner is of welded construction, made of $^1/_2$-in. steel plate, with an inside diameter of 24 ft. The bottom of the liner is bolted to a flange on the spiral casing and the liner extends up from the top of the casing to the governor gallery floor at elevation 951. The liner is designed to withstand hydrostatic pressure with tail water at elevation 985 without severe distortion. It has rigid circular flanges for mounting the servomotors. Alcoves are formed in the liner for the main-bearing oil pumps and instruments. Two checkered steel-plate walkways are provided in the turbine pit, one outside of the gate stems and one on top of the bearing housing. Steps leading to the turbine pit are mounted on the shifting ring, Fig. 5.

Servomotors

The turbine is provided with two oil-pressure-operated double-acting hydraulic cylinders or servomotors which are mounted on heavy circular flanges in the walls of the pit liner. They have semisteel cylinders and cast-steel heads and stuffing boxes. The servomotors have a combined capacity sufficient to exert a torque in the shifting ring of 2,900,000 ft-lb, with an oil pressure of 250 psi. Under this oil pressure and with an adequate supply of oil, the servomotors are capable, under maximum operating-head conditions, of moving the turbine gates a full opening or closing stroke in 4 sec. The servomotors are provided with adjustable by-pass connections, whereby the rate of closure may be retarded from slightly below the speed-no-load position for maximum head to the fully closed position, so as to minimize pressure rises in the penstock. Provision is made on the piston rods for locking the gates positively in the opened or closed position, or for limiting the movement of the gates.

Draft Tube

The draft tube is formed in the concrete substructure of the powerhouse, the upper part being lined to a point 22 ft below the center line of the turbine distributor. The liner is made of $^3/_4$-in-thick steel plate with welded joints and is heavily reinforced on the outside by means of suitable ribs. These ribs also provide means for anchoring the liner to the concrete of the powerhouse by the use of turnbuckle rods. Jack pads, attached to the lower edge of the liner, are provided for supporting it during the pouring of concrete. The top of the liner is bolted to the foundation ring by means of a flanged connection. Two 24 $\times$ 36-in. manholes with cast-steel hinged covers, opening outward into the access passageways, are provided diametrically opposite each other to permit access to the draft tube and to the under part of the turbine runner.

The liner was assembled in the shop and match-marked, then shipped completely knocked down for welding in the field.

The draft tube is unwatered by gravity through valves provided in the draft-tube dividing piers. These valves connect through a common unwatering header to a sump, from which the water is pumped by a motor-driven, deep-well type of pump to the tailrace. During the unwatering operation, the downstream end of the draft tube is closed by means of steel stop logs.

Governors

Governors for the 150,000-hp units are of the oil-pressure actuator type with motor-driven speed-responsive elements actuated from 3-phase permanent-magnet generators connected to the tops of the generator shafts. They control the gates of the turbines by means of the servomotors connected to the gate-shifting ring. They are adjustable to provide variable rates of actions for completely opening or closing the turbine gates in from 4 to 12 sec. The speed-responsive elements are sensitive to turbine-speed variations of 0.01 per cent. The speed change from no load to full load is adjustable from 0 to 6 per cent.

Each governor is complete with an oil-pressure system, including two pressure-controlled 40-hp motor-driven pumps and a sump tank located within the actuator, and a pressure tank located adjacent to the actuator. The actuator is located at one end of and forms a part of the control board in the governor gallery at elevation 951, Fig. 11. The pressure tank is located just back of the actuator and connected to it by means of a 6-in. pipe below the floor level. The system is designed to operate with an oil pressure ranging from 250 to 300 psi.

The actuators are equipped with the following features which may be operated manually at the actuator or electrically by remote control from the control board:

(*a*) A gate-limit-control device which will operate two adjustable 1.5-amp 250-v d-c ungrounded limit switches. Each switch is independent and adjustable for circuit opening or closing over the range of travel of the gate-limit device.

(*b*) A speed-level-controlling device. Speed controls are from 85 per cent of rated speed at no load and zero speed droop to 115 per cent of rated speed at rated load and maximum speed droop.

(*c*) A device for opening and closing the turbine gates at the normal rate of movement. This device will be used for manual starting and stopping of the turbine and for automatic shutdown by means of automatic protective features incorporated in the main generator, governor equipment, or transformers.

(*d*) A device for controlling the speed droop of the turbine. The amount of speed droop is adjustable from zero to 5 per cent.

In addition, the following devices are furnished:

(*e*) An electrically operated speed indicator mounted on the actuator column. This speed indicator is driven from a magneto-type generator on the generator shaft. In addition to indicating turbine speed, it will also indicate when rotation starts and stops.

(*f*) Two gate-limit and gate position indicators of the dual type, one mounted on the actuator and the other mounted on the benchboard in the main control room. These instruments indicate the position of the governor gate-limit device and the position of the turbine gates.

(*g*) An overspeed switch, mounted on and forming a part of the governor-drive generator, arranged to shut down the turbine and sound an alarm upon overspeed.

(*h*) A combination automatic and hand-operated air valve for controlling the operation of the generator brakes. The air valve is controlled by means of a low-speed switch, located in the housing of the permanent-magnet generator, so adjusted that the brakes cannot be applied until the turbine gates are fully closed and the speed of the unit has been reduced to 30 rpm. Brake application is intermittent with the time periods adjustable, for a selected number of cycles. After that, the brakes are applied constantly until the unit is brought to a stop.

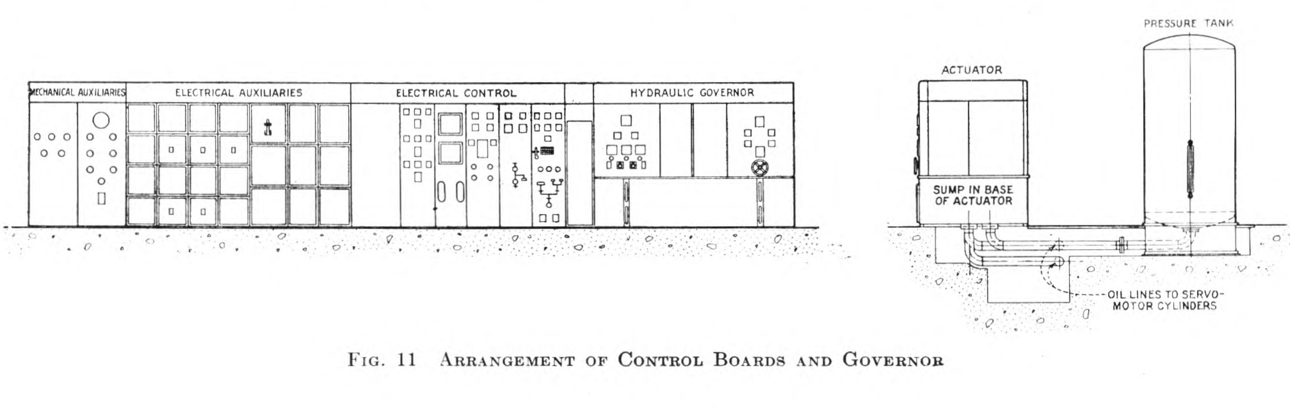

Fig. 11 Arrangement of Control Boards and Governor

(*i*) Manual control of the turbine gates at the actuator by means of oil pressure from the governor oil-pressure system.

(*j*) An airplane-type flexible-steel restoring cable, operating over sheaves and enclosed in a metal conduit, connecting the governor-restoring mechanism and the turbine gate-operating mechanism. This eliminates lost motion between the servomotor piston travel and the governor pilot valve.

The two oil pumps in the governor system have a combined capacity per minute of 3 times the total oil volume of the servomotors. They will supply an adequate quantity of oil to the servomotors so as to operate the turbine gates through a complete closing or opening stroke in 4 sec with an oil pressure of 250 psi, and with an operating head on the turbine of 355 ft. They are arranged to start and stop at predetermined oil pressures in the pressure tank.

The two pumps are interconnected so that they can be operated independently, or together. When operating together, the interconnection and automatic control is such that either pumping unit may be used for normal operation with the other unit serving as a stand-by unit, arranged to start automatically either on failure of the electric-power supply to the operating pump or upon the oil pressure falling below a predetermined amount.

The pressure tank has a total volume of 20 times the volume of the servomotor cylinders and will supply five complete servomotor strokes of the turbine with a drop in pressure from 300 to 250 psi without the operation of the pumps. Float valves are provided in the bottom of the pressure tank to close automatically in case the oil level drops, so as to prevent air, from the pressure tank, entering the governing system.

The oil sump tank for the governor system is located in the base of the actuator, thus eliminating the necessity of having sump tanks below the governor floor.

Acceptance Tests

After the turbines have been installed in the powerhouse and placed in satisfactory operation, they will be tested to determine whether or not the contractor's guarantees of horsepower, water discharge, and efficiency have been fulfilled. These tests will be conducted in accordance with the testing code for hydraulic turbines recommended by The American Society of Mechanical Engineers.

Pumping Plant

Some of the power developed at the Grand Coulee Dam is to be used for the operation of pumps for irrigation purposes; therefore, a short description of the pumping plant may be of interest.

All irrigation water for the Columbia Basin project will be pumped approximately 295 ft from the reservoir behind the Grand Coulee Dam into a balancing reservoir formed in the Grand Coulee by the construction of earth dams at each end. The Grand Coulee Dam will raise the water from a minimum of 275 ft to a maximum of 355 ft, and the pumps will lift it the remaining distance to the balancing reservoir. Ordinarily, pumping will be against a 295-ft head, i.e., from a full storage reservoir behind the dam to a full balancing reservoir in the Grand Coulee.

The pumping plant is located along the shoreline of the reservoir just upstream from the left abutment of the dam, as shown in Fig. 1.

Twelve pumping units are proposed for the ultimate installation, each unit consisting of a single-stage vertical-shaft centrifugal pump having a capacity of 1600 cfs when operating under a total head of 295 ft, direct-connected to a 65,000-hp synchronous motor. The motor capacity is of such size that, with a full reservoir, one main generating unit in the power plant will have sufficient capacity to operate two pumping units.

The water is supplied to each pump through an individual 14-ft-diam welded plate-steel intake pipe which has a bellmouth inlet at elevation 1191.75 in the upstream face of the pumping-plant dam. The center lines of the pump casings are located at elevation 1203 and from this point the water is discharged through 12-ft-diam welded plate-steel discharge pipes about 800 ft long into a canal leading to the balancing reservoir in the Grand Coulee 1.7 miles away.

Through a canal of approximately 15,000 cfs capacity, water will be carried about 10 miles from the balancing reservoir to other canals from which it will be distributed through numerous laterals for irrigation purposes.

Acknowledgments

The 150,000-hp turbines for the Grand Coulee power plant were designed by the Newport News Shipbuilding and Dry Dock Company, Newport News, Va., and the governors were designed by the Woodward Governor Company, Rockford, Ill., in accordance with specifications prepared by the United States Bureau of Reclamation.

The author is indebted to the engineers of the Bureau of Reclamation for valuable aid in connection with the preparation of this paper.

The mechanical installations at the Grand Coulee power plant were designed under the supervision of the author. All mechanical installations are designed under the general direction of I. A. Winter, senior engineer, and L. N. McClellan, chief electrical and mechanical engineer.

All engineering designs prepared by the bureau are under the general direction of J. L. Savage, chief designing engineer; all engineering and construction work is under the direction of S. O. Harper, chief engineer, with headquarters at Denver, Colo.; and all activities of the bureau are under the general charge of J. C. Page, Commissioner of Reclamation, with headquarters at Washington, D. C.

Discussion

R. V. Terry.[3] This paper has been written primarily from the viewpoint of the designer of the power plant. Perhaps a few remarks on these turbines from the viewpoint of the turbine designer and manufacturer would be of interest.

Among the first problems to be attacked was that of splitting up the parts, especially the cast-steel casing, so that they could be handled by rail shipments. The final design evolved for the casing provided for splitting the combined casing and speed ring into 14 pieces, which included two inlet rings. Longitudinal joints were avoided, all joints being placed at right angles with the flow. This reduced the loading of joints per unit length and facilitated manufacture. The angular spacing of the segments varied from 18 deg at the large end to a maximum of 63 deg near the smaller end. The 14 speed-ring vanes are irregularly spaced to suit each casing segment. Scale models were made of the largest pieces, of the special available railroad cars, and of the composite railroad clearances, as a final check on the drawings. High-tensile-strength bolt material, heat-treated, was used for the joints to reduce flange dimensions, and the bolts were prestressed to 23,000 psi, as determined by micrometer measurements.

Due to its size and weight, the casing was assembled on a special foundation outside the shop, similar to that used in the field for boring, for assembly with other parts to be embedded in concrete, and for the hydrostatic test. Boring was accomplished with a portable mill previously employed for battleship-turret work. The complete assembly for the hydrostatic test weighed

[3] Hydraulic Engineer, Newport News Shipbuilding and Dry Dock Company, Newport News, Va. Mem. A.S.M.E.

about 786 tons, including 375 tons of water. During the shop hydrostatic test of the first unit, the only one tested to date, deflection readings were taken in mils at 48 points. The casing joints were practically droptight. Leaks were found through the castings in only a few places.

A somewhat new detail of a rectangular-shaped water-sealing groove was employed for the round rubber cord at the joints. The width of the groove is slightly less than the diameter of the cord, creating a pinch which holds the cord in place without clamps while placing the adjoining part. This type of groove seems to be superior to the older type. A model test indicated that the flange could be backed off nearly $^1/_8$ in. before the cord would blow at 230-psi test pressure. This type, incidentally, would be equally effective for pressure in either direction.

The shop space available, as well as the shipping schedule, did not permit of making a complete shop assembly of all embedded and nonembedded parts at one time. However, separate assemblies were made so that every part was fitted to its mating part or parts and thus proper fitting in the field is assured.

The servomotors are carried by the walls of the pit liner. However, the pit liner is not designed to carry the servomotor reactions without the help of the concrete backing. Alternate bolts of the servomotor flange serve as foundation bolts and extend well back into the concrete.

It will be noted from Fig. 5 of the paper that a considerable radial clearance is allowed between the speed-ring vanes and the wicket gates and between the gates and the runner vanes. The former clearance is necessary in this case due to the irregular spacing of the speed-ring vanes previously mentioned, there being 14 vanes and 24 wicket gates. The latter clearance, between gates and runner with the gates fully open, is about $7^3/_8$ in. or 3.75 per cent of the runner diameter. Ample clearance at that point is, of course, required to reduce mutual interference of a hydraulic nature, particularly for turbines which must be operated over a wide range in head.

The runners are 197 in. nominal diam, the approximate limit for transportation in one piece, and each runner weighs about 125,000 lb. Such runners are given an accurate check for static balance by supporting them on a hardened spherical point at the axis, slightly above the center of gravity of the runner.

Provision is made for the possible admission of free air at three different points, Fig. 5 of the paper. First, an 8-in. valve is provided, cam-operated from the gate mechanism, to admit air at the lower gate openings through ports in the runner crown, just downstream from the runner vanes. With an initial normal tailwater level 7 ft above the center line of the runner, it is not expected that free air can be admitted through that valve. Later when the tailwater level drops to its minimum value, 6 ft below the center line of the runner, that valve will become effective. In the meantime use will be made of the 3-in. air line which terminates at the center of the runner cone, usually the point of lowest pressure when the water leaves the runner with a whirl at low- or high-gate openings, such as to create a vortex disturbance in the draft tube. The 3-in. air line takes free air from the turbine pit through a Maxim silencer, a check valve, and a gate valve. The system will be effective and take air only when required, that is, when a vortex exists at the tip of the runner cone. The amount of air may be partially controlled by throttling the gate valve. A third air line, 2 in. diam, is provided through the crown plate to the space above the runner, just outside of the intermediate seal, where a low pressure is expected to occur. Any of the three schemes of air admission may be used as the conditions of tailwater level or other operating conditions may dictate. Although the various conditions were studied during the running of the model tests, it is difficult to predetermine the exact adjustments needed for the field.

TABLE 1 COMPARISON OF BOULDER DAM AND GRAND COULEE TURBINES

Plant	Boulder	Grand Coulee
Rated unit, bhp	115000	150000
Rpm	180	120
Rated head, ft	480	330
Specific speed	27	33
Rated discharge, cfs	2400	4500
Inlet diameter of casing, ft	10	15
Number of casing sections	6	14
Weight of casing, lb	450000	582000
Maximum operating head, ft	590	355
Casing test pressure, psi	500	230
Diameter of wicket-gate circle, ft	17	19
Nominal diameter of runner, in	171	197
Height of wicket gates, ins	19	$34^3/_8$
Diameter top of draft tube, ft	11	14.33
Pit diameter, ft	22.5	24.0
Diameter of shaft, in	36	44
Length of shaft from center line of turbine to face of generator coupling, ft	26	43
Diameter of shaft coupling, in	$61^3/_4$	75
Size of main bearing, in	$36^1/_4 \times 29^1/_2$	$45 \times 41^1/$
Servomotor capacity, ft-lb	340000	400000
Operating ring torque, ft-lb	2100000	3180000
Total turbine thrust, lb	590000	925000

The complete shipping weight of one turbine is expected to be about 820 tons, including the draft-tube liner, the 15-ft expansion joint at the casing inlet, and the intermediate turbine shaft but not including the governor and governor piping.

Table 1 of this discussion gives a comparison of the principal data and dimensions of the Grand Coulee and Boulder Dam turbines.

It has given the writer unusual pleasure to be associated with the design and manufacture of these turbines which are of unprecedented size and horsepower capacity, being over 30 per cent more powerful than the most powerful hydraulic turbines now in operation. His company received most excellent cooperation from the engineers and inspection personnel of the Bureau of Reclamation.

Author's Closure

The discussion presented by Mr. Terry contains many interesting features not covered by the author and should add much to the value of the paper.

He is correct in his statement that the pit liner is not designed to take the full servomotor reaction but is aided by foundation bolts extending into the surrounding concrete.

He also properly points out that the 8-in. valve for admitting atmospheric air through ports in the runner crown may not become effective until the tailwater level drops to its minimum value and in the meantime use will be made of other available means of air admission.

Table 1 of the discussion gives an interesting comparison between the Boulder Dam and Grand Coulee turbines. The particular Boulder Dam turbines used in this comparison, however, were not designed to operate solely at 60 cycles, 180 rpm. The specifications issued by the Bureau of Reclamation for these turbines called for units which could operate satisfactorily at either 50 cycles, 150 rpm per minute, or 60 cycles, 180 rpm with no major change in the apparatus except the installation of runners suited to the particular speed. The author believes a better comparison with the Grand Coulee turbines may be obtained by the use of data pertaining to other Boulder Dam turbines which were designed for and are now operating at 60 cycles, 180 rpm.

Using these data for the same items as given in Mr. Terry's comparison, the following values for the Boulder Dam turbines will obtain:

Rated unit, bhp	115,000
Rpm	180
Rated head, ft	480
Specific speed	27
Rated discharge, cfs	2,340
Inlet diameter of casing, ft	10
Number of casing sections	5
Weight of casing, lb	381,000

Maximum operating head, ft	590
Casing test pressure, psi	500
Diameter of wicket-gate circle, ft	15.625
Nominal diameter of runner, in	$163^7/_8$
Height of wicket gates, in	21
Diameter top of draft tube, ft	10.83
Pit diameter, ft	20.75
Diameter of shaft, in	36
Length of shaft from center line of turbine to face of generator coupling, ft	26
Diameter of coupling between turbine and intermediate shafts, in	$61^3/_4$
Size of main bearing, in	$36^1/_2 \times 29$
Servomotor capacity, ft-lb	324,000
Operating ring torque, ft-lb	2,320,000
Total turbine thrust, lb	600,000

In conclusion the author wishes to thank Mr. Terry for his very interesting discussion. He also wishes to express his pleasure in being associated with him and his company in the design of the Grand Coulee turbines.

Economic Draft-Tube Proportions

By A. R. DAWSON,[1] TORONTO, CANADA

The purpose of this paper is to illustrate a new method whereby the most efficient and also the most economical proportions of a draft tube for a water-turbine installation may be found. A study is made of the excavation and concrete costs, as the ratios of draft-tube width to runner diameter and draft-tube depth to runner diameter increase. These costs are compared to the actual capitalized value of the horsepower saved due to the increased efficiency, as either ratio is increased. The comparison is made by means of computed curves and includes the ordinary range of power costs, as well as a range of load factors from 50 to 100 per cent, at which the unit might be expected to operate. The paper develops a method to overcome the seemingly prevalent practice of many turbine manufacturers of installing a certain definite size and shape of draft tube with a certain size runner without regard to the various factors which should enter into the selection.

THE tendency in the design of hydraulic-turbine draft tubes has been to produce draft tubes of highest possible efficiency. This tendency on the part of manufacturers to improve the performance of their product is entirely understandable and is appreciated by the purchasers. While manufacturers and users have undoubtedly given some thought to the economics of draft tubes, the literature dealing with this phase of the subject is but meager. It must be recognized that draft tubes can be designed and built with very high efficiency if no limitations on cost are imposed. On the other hand, there are many instances in which the conditions existing at a given power-plant site make it economically impossible to use draft tubes of the highest efficiency. This paper suggests one method of procedure by which engineers may estimate how far it is feasible and economical to go in the design of a draft tube for a given installation.

The method proposed herein is one which is well known to engineers and has been applied to many other problems, of which may be mentioned the selection of the proper diameter for hydraulic-power-plant penstocks. In essence, the method adds together the cost of construction and the capitalized value of maintenance and of the power lost through inefficient operation and other causes and, by selecting that design which shows the lowest total capitalized cost, indicates the most economical design.

In the case of draft tubes, there is no one design which is generally accepted by all manufacturers, nor is there one design of draft tube which is best for all types of hydraulic turbines. As a generalization, however, it may be stated that the longer the draft tube, the more efficient it is likely to be. In the usual design of power plants, in which the turbine is set with a vertical shaft, the draft tube may be thought of as being divided into two lengths: (1) The vertical section from the outlet of the turbine runner vertically downward to the elbow; (2) the horizontal portion of the draft tube from the elbow to the discharge end. It is not the author's intention to suggest that the data which will be used here, and which will serve to relate these two lengths of the draft-tube and machine efficiency are generally applicable, but they will serve to supply data from which an illustrative example can be worked out. The unit for which this study is made is shown in Fig. 1, which represents a proposed unit for a hydroelectric power plant operating under a head of 68 ft and generating 16,000 hp at 115.4 rpm. The diameter of the Francis-type runner is 12 ft and the specific speed is 74.8. The top of the draft tube is assumed to be 12 ft in diam.

From studies which have been made available, curves have been drawn to show the relationship of the turbine efficiency to the proportions of the draft tube. Two curves are available, the first, Fig. 2, showing the relation of the efficiency change to the ratio of the diameter of the turbine and the vertical distance from the bottom of the runner to the bottom of the tube, the latter distance being hereafter called the depth of the tube. The second curve, Fig. 3, shows the relations between the efficiency change, and the ratio of the diameter of the runner to the width of the draft tube at exit. For both curves, a dimensionless parameter was used and efficiency drop was chosen for plotting. It should be noted that the efficiency drop was used rather than actual efficiency, i.e., this is the loss of efficiency occasioned by too short or too narrow a draft tube, as compared with a deep or a wide one. These data have been obtained from the average of three different analyses, due to W. J. Rheingans on Francis-type runners, to F. Nagler on all types of runners, and to the N.E.L.A. Hydraulic Power Committee tests on straight draft tubes. The actual data were not directly available to the author, but were supplied through the courtesy of Mr. Nagler.

Effect of Variations in Draft-Tube Depths

The curve Fig. 2, giving the relation between the ratio $A = \frac{\text{Draft-tube depth}}{\text{Runner diameter}}$ and the loss in efficiency, is based on data from various sizes and shapes of turbines and draft tubes and has been assumed to be generally applicable to the unit being discussed. This curve shows that as A increases from 1 to 7, the loss in efficiency decreases from 4.65 to 0.05 per cent, i.e., there is an increase in turbine efficiency of 4.6 per cent. Applying this result to a 16,000-hp turbine operating at 100 per cent load factor, shows that the actual power lost by varying the depth of draft tube over this range changes from 744 hp to 8 hp, and in Table 1 is shown the power loss corresponding to intermediate values of A.

TABLE 1 POWER LOSS CORRESPONDING TO INTERMEDIATE VALUES OF A[a]

Ratio A	Depth of tube, ft	Efficiency loss Fig. 2, per cent	Horsepower loss 100 per cent load factor	Revenue deficiency at $10 per hp
1	12	4.65	744	$7440
2	24	2.32	371	3710
3	36	1.28	205	2050
4	48	0.78	125	1250
5	60	0.50	80	800
6	72	0.25	40	400
7	84	0.05	8	80

[a] Ratio $A = \frac{\text{Draft-tube depth}}{\text{Runner diameter}}$.

Based on a price of $10 per hp per annum, the revenue deficiency ranges from $7440 to $80, as is indicated in Table 1. If these sums are capitalized on a 5 per cent basis, they correspond to values ranging from $148,800 to $1600.

[1] Senior student in Mechanical Engineering, University of Toronto. Student Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

Although the efficiency increases as the draft tube is made deeper, it is obvious that for each installation there is a point at which extra cost of excavation and construction more than offsets the saving due to the decreased loss of horsepower. To find this point, an estimate of the costs of excavation and concreting must be made. The method of analysis will be illustrated by applying it to the plant shown in Fig. 1, which corresponds to $A = \frac{25}{12} = 2.08$. The depth of draft tube is shown as 25 ft, but the excavation should be carried lower, and it is estimated that 3 ft would be suitable.

In order to arrive at the actual cost, the volume of the excavation and of concrete to be placed should be determined with care, by means of drawings showing quite closely the finished dimensions of the work. However, in this case the method of procedure is being described and certain assumptions have been made so as to enable this to be done; the author does not suggest that these assumptions are particularly exact.

The horizontal length of tube is 37 ft from the center line of the turbine and it is estimated that the excavation would have to be carried 10 ft to the left, making 47 ft horizontal distance in all. The width of the tube is shown as 42 ft, made up of two 15-ft widths for the tube openings, a 5-ft division wall, and 3.5 ft of concrete between the tubes and the rock face. The author has followed the practice of certain designers in assuming that the rock must be blasted away in front of the tube, at an angle of 45 deg to the level of the top of the tube, this clearance being necessary to facilitate removal of the rock.

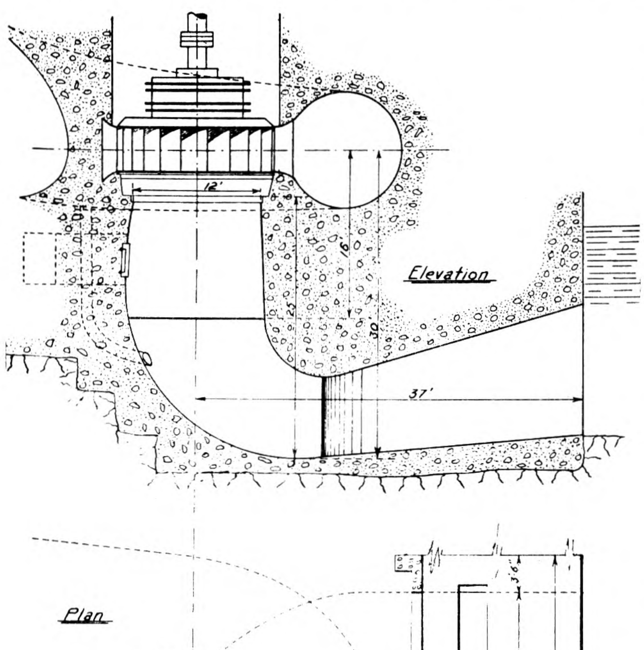

Fig. 1 Proposed Hydroelectric Unit for Which Economic Study Was Made

The total excavation would therefore be $(47 \times 42 \times 25) + (3 \times 47 \times 42) + (^1/_2 \times 25 \times 25 \times 42) = 68{,}397$ cu ft = 2533 cu yd. Rock excavation has been assumed to cost \$4 per cu yd,[2] considering average difficulties involved, including reasonable location of the plant, watering costs, and the cost of labor and transportation. Further, a calculation for this case indicates that the concrete placed is approximately 75 per cent of the volume of rock excavated, and the price of concrete in place has been taken at \$22 per cu yd,[2] including reinforcement, aggregates, transportation, labor, and form work.

For the value of $A = 2.08$, therefore, the cost of the tube is readily found, but it should be pointed out that design and engineering costs are not included. In the same way the construction costs for ratios A from 1 to 7 have been calculated and are set down in Table 2.

To compare the relative merits of the different depths of tube, the method of total capitalized cost is used to combine the cost of the tube and the revenue deficiency, because the sum of the effects of these two factors should be a minimum for the best tube. In calculating the total capitalized costs the following assumptions have been made:

1 That the useful life of the tube is 20 years and that it will be replaced at 20-year periods to perpetuity. This time may be thought too short, but experience shows that, for one reason or another, the tubes are replaced in from 20 to 30 years. Assuming a straight-line depreciation of 5 per cent, the capitalized value of the annual payments to a fund to provide replacement in 20 years would be equal to the first cost.

2 That annual repair and maintenance are 2 per cent of the original cost. The capitalized value of these annual payments to perpetuity at 5 per cent amounts to $^2/_5$ = 40 per cent of the construction cost.

3 That the capitalized value of the annual power loss, that is, the revenue deficiency, is 5 per cent to perpetuity, which would amount to 20 times the values in the last column of Table 1.

Based on the 16,000-hp turbine, taken as an illustration, running at 100 per cent load factor and with power selling at \$10 per hp per year, the values shown in Table 2 are obtained.

TABLE 2 CONSTRUCTION COSTS FOR RATIOS OF A^a FROM 1 TO 7

Ratio A	1	2	3	4	5	6	7
	\$1000 units						
Construction cost..	24.6	49.5	79.0	112.8	151.3	194.3	221.5
Depreciation fund..	24.6	49.5	79.0	112.8	151.3	194.3	221.5
Repair and maintenance fund.....	9.8	19.8	31.6	45.1	60.5	77.7	88.6
Capitalized value of power loss, Table 1...............	148.8	74.2	41.0	25.0	16.0	8.0	1.6
Total capitalized cost............	208.0	193.0	231.0	296.0	379.0	474.0	533.0

^{a} Ratio $A = \dfrac{\text{Draft-tube depth}}{\text{Runner diameter}}$.

Now, the selling price of power in Canada falls within the limits of \$10 to \$40 per hp per year, with very few exceptions, so

[2] The values of concrete and excavation costs are averages for existing plants of the Hydro-Electric Power Commission of Ontario, Canada.

(a) Chats Falls plant on Ottawa River, Ontario, for eight 28,000-hp turbines under 53-ft head, concrete \$15 per cu yd; excavation \$2.50 per cu yd.

(b) Ragged Rapids plant on Musquash River, Ontario, for two 5000-hp turbines under 38-ft head, concrete \$25 per cu yd; excavation \$6 per cu yd.

(c) Alexander plant on Nipigon River, Ontario, three 18,000-hp turbines under 60-ft head, concrete \$24 per cu yd; excavation \$4 per cu yd.

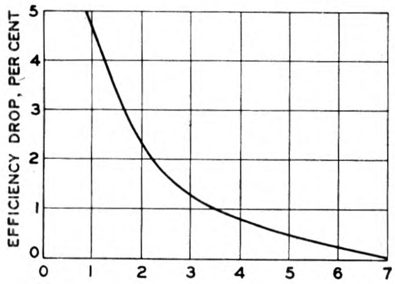

FIG. 2 CURVE SHOWING DECREASE IN EFFICIENCY DROP AS RATIO A INCREASES

$$\left(\text{Ratio } A = \frac{\text{Draft-tube depth}}{\text{Runner diameter}}.\right)$$

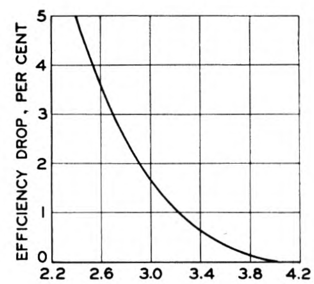

FIG. 3 CURVE SHOWING DECREASE IN EFFICIENCY DROP AS RATIO B INCREASES

$$\left(\text{Ratio } B = \frac{\text{Draft-tube width}}{\text{Runner diameter}}.\right)$$

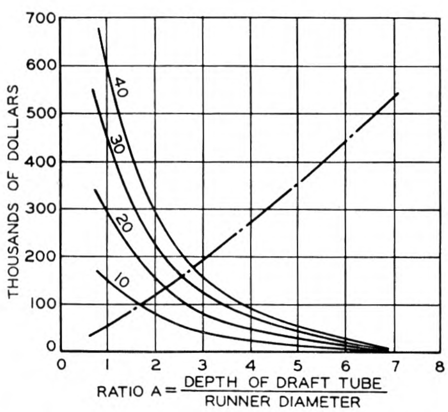

FIG. 4 SOLID LINES SHOW CAPITALIZED VALUE OF ANNUAL REVENUE DEFICIENCY AT 100 PER CENT LOAD FACTOR AND AT $10, $20, $30, AND $40 PER HP. DOTTED CURVE IS TOTAL FIRST COST PLUS CAPITALIZED VALUE OF DEPRECIATION AND MAINTENANCE

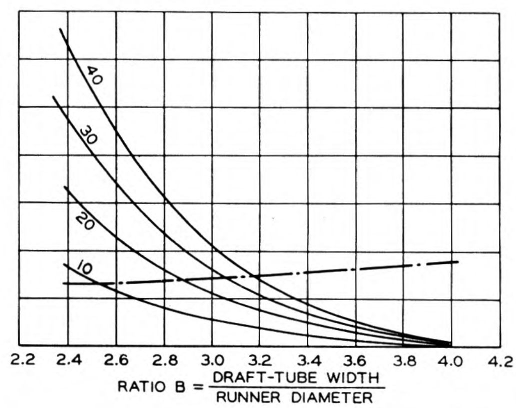

FIG. 6 SOLID LINES SHOW CAPITALIZED VALUE OF ANNUAL REVENUE DEFICIENCY AT 100 PER CENT LOAD FACTOR AND AT $10, $20, $30, AND $40 PER HP. DOTTED CURVE IS TOTAL FIRST COST PLUS CAPITALIZED VALUE OF DEPRECIATION AND MAINTENANCE

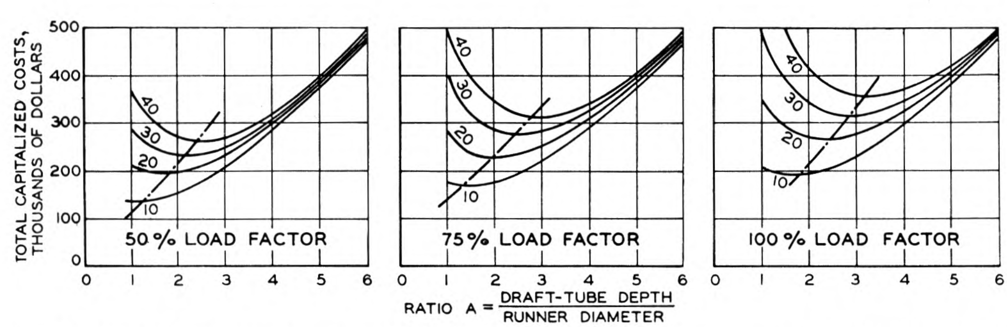

FIG. 5 TOTAL CAPITALIZED COSTS FOR THREE LOAD FACTORS AND FOR POWER AT $10, $20, $30, AND $40. DOTTED CURVES SHOW BEST TUBES

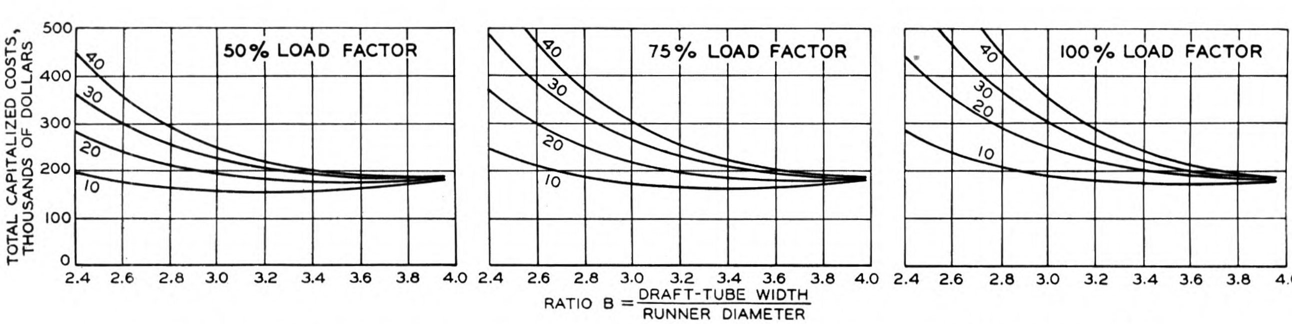

FIG. 7 TOTAL CAPITALIZED COSTS FOR THREE LOAD FACTORS AND FOR POWER AT $10, $20, $30, AND $40

that, by repeating the calculations for various power prices, a whole family of curves may be obtained in the same manner. For illustration purposes only, the values of \$10, \$20, \$30, and \$40 per hp per year have been dealt with in this paper. Also, since turbine units rarely run continuously at 100 per cent load factor, the entire procedure may be repeated for any desired load factor and, in this case, the calculations have been made for 100, 75, and 50 per cent load factors.

The curves, Fig. 4, show how the capitalized values of the annual horsepower losses or revenue deficiencies at varying horsepower values and 100 per cent load factor fall as the depth of draft tube increases. In the same figure, the total first cost and capitalized value of the depreciation and maintenance funds have been plotted on the dotted curve, which slopes in the opposite direction to the others. The curves in Fig. 5 show the sum of the items mentioned, giving the total capitalized costs for the four prices of power assumed and for 100, 75, and 50 per cent load factors.

Effect of Variation in Draft-Tube Width

This study is based on the results shown in Fig. 3, obtained from the same source as was used in Fig. 2, and shows the variation of efficiency drop with draft-tube width. From this curve, the results shown in Table 3 are obtained in a similar way to Table 1.

TABLE 3 POWER LOSS DUE TO VARIATION IN B^a VALUES

Ratio B	Draft-tube width, ft	Efficiency loss Fig. 3, per cent	Horsepower loss at 100 per cent load factor	Revenue deficiency at \$10 per hp
2.4	28.8	5.00	800	8000
2.6	31.2	3.50	560	5600
2.8	33.6	2.50	400	4000
3.0	36.0	1.70	272	2720
3.2	38.4	1.12	179	1780
3.4	40.8	0.670	107	1070
3.6	43.2	0.350	56	560
3.8	45.6	0.150	24	240
4.0	48.0	0.025	4	40

[a] Ratio $B = \dfrac{\text{Draft-tube width}}{\text{Runner diameter}}$.

In this phase of the calculation, the shape and dimensions of the sectional elevation at the top of Fig. 1 are assumed to be constant, the width of the tube alone changing. Proceeding in the same way as before, the rock excavation will be, for the case shown in plan in Fig. 1, i.e., $B = 2.5$: —$[(47 \times 28) + (^1/_2 \times 25 \times 25)] \times 42 = 68{,}397$ cu ft or 2533 cu yd.

Using the same excavation and concrete prices as before, the construction cost has been arrived at and tables similar to Table 2 made, and from these the results shown in Fig. 6 are obtained. Then in Fig. 7 the total capitalized costs for three load factors and for power at \$10, \$20, \$30, and \$40 have been plotted from which the most economical tube may be selected.

Conclusions

From the curves of Fig. 5, it is seen that the ratio $A = 1.8$, corresponding to 21.6 ft depth, would be the most economical if the unit is to be run at 100 per cent load factor most of the time and if the selling price of 1 hp is \$10 per year. However, if it runs at 100 per cent load factor and the value of power is \$40 per hp per year, then it would be more economical to use a depth of draft tube corresponding to a ratio A of 3.2 (i.e., depth $= 12 \times 3.2 = 38.4$ ft). Again, the ratio of 2.5 gives the most economical draft tube, if the unit is to be run at 75 per cent load factor and the value of 1 hp is \$30 per year, while for 50 per cent load factor, the ratio 2.5 corresponds to power at \$40 per hp per year.

Similarly, for this proposed unit, the ratio of draft-tube width to runner diameter is $B = 2.5$. Now, from Fig. 7, it is seen that the minimum points for the curves at \$10 per hp per year are at a ratio B of 3.6, 3.4, and 3.2 for 100, 75, and 50 per cent load factors, respectively, giving the corresponding optimum widths of 43.2 ft, 40.8 ft, and 38.4 ft. For values of power of \$20 per hp per year, or greater, the curves all approach a minimum at a ratio of $B = 4$ corresponding to a width of 48 ft. Thus, any width of draft tube greater than 48 ft for this turbine would not give additional efficiency or economy. Of course this is not the rule with any draft tube, since for other units with different horsepower ratings, first costs, or load factors, these curves might give definite minimum values such as were found previously with regard to the change of depth of draft tube. In this case, the most economic width could be fixed definitely.

There are many factors which affect the shape of these curves, both for variation in depth and variation in width but, in general, the curves take the same form, although the minimum points cover quite a wide range of ratios. For example, taking the unit studied in this paper as a basis, these facts become evident:

1 For an installation of greater rated power than 16,000 hp, the minimum points would move to the right, i.e., the more economic tubes would be both deeper and wider.

2 For higher selling prices of power, the minimum points would move to the right.

3 For higher interest rates than 5 per cent, the minimum points on the curves would move slightly to the left, but would not be changed greatly.

4 For higher costs of excavation and concrete, the minimum points would move to the left.

5 For lower load factors, the minimum points would move to the left.

6 For higher maintenance and repair values and for a shorter estimated life of tube, the minimum points would move to the left.

For a proposed installation, the rated horsepower is known, the costs of excavation and concrete can be closely approximated from boring tests and from the known location of the proposed unit or plant, and the selling price of power is usually well defined, as is also the existing rate of interest. Thus, the only factors which have to be assumed before being able to complete the foregoing calculations are the expected life of the tube, which in most cases is the same as the expected life of the entire plant, the expected annual repair and maintenance costs, and the expected load factor at which the unit will be operated.

It can be readily seen that these calculations and curves would be most valuable where (*a*) the limiting depth of the tube is fixed due to some irregularity of the rock formation from which excavation must be made for the draft tube, in which case the most economical width could be found; or (*b*) the width of the tube is fixed, as is more usual, due to limitations on the possible over-all length of the plant, in which case the most economical depth could be found.

Acknowledgments

Grateful acknowledgment is made to Mr. F. Nagler of the Canadian Allis-Chalmers, Limited, whose suggestions formed the basis for this paper, and also to Professor R. W. Angus, whose technical advice was greatly appreciated.

Discussion

F. Nagler.[3] This paper is commendable, in that it presents a type and completeness of analysis which the writer believes has not previously appeared in any of our technical publications.

The method of capitalizing efficiency losses seems to be slightly intricate, but Table 1 brings the results down to a basis that is definitely conclusive. Certainly, this method of arriving at a

[3] Chief Engineer, Canadian Allis-Chalmers, Limited, Toronto, Canada. Life Mem. A.S.M.E.

conclusion as to economical proportions of draft tube seems definitely more sound than the all too prevalent method of accepting blindly a proposal drawing that is usually an arbitrary picture based on precedent of the manufacturer. It may be admitted that a given draft tube, showing up well with a particular runner, may thereafter be especially fostered by the manufacturer and sold indiscriminately to the buying public. It is far from certain that such a tube will necessarily work well with even a different type of runner by the same manufacturer and even less so with a diverse design from another builder.

Naturally, the significant basis of this paper is found in the efficiency-variation curves, Figs. 2 and 3. It seems to the writer that the former should be definitely asymptotic, although Fig. 3 might not necessarily be so. Probably these curves are more open to criticism than anything else in the paper, as some such curves inherently form the basis of this kind of analysis. If the writer recalls correctly, the N.E.L.A. at one time sponsored or at least considered correlating the data from all manufacturers to arrive at some conclusion along this line. There seems to be no doubt whatsoever that there is a definite relationship between each of the three factors, tube height, tube length, and tube width, and efficiency loss. The losses must increase as the vertical height of a tube is decreased, all other things being equal. Whether the exact shape in this paper is correct or not is probably secondary to the rather definite method the author presents for arriving at economical proportions.

It might be in order to suggest that there is probably a complicated relationship between horizontal length and vertical height. This indicates the possibility of dealing with total length, which would be the combination of these two dimensions. The Ryburg-Schworstädt draft tube is a striking example of a tube which has a major portion of its diffusion section in its horizontal run. The writer doubts, however, that even such an extreme example detracts very definitely from the indication of the author's curve, Fig. 2, as it is extremely likely that its efficiency would be increased if its vertical run were longer. Some studies as to the combination of vertical and horizontal lengths seem to be in order.

Fig. 5 is of interest in showing graphically and rather strikingly the effect on draft-tube proportions that is exerted by value of power, and similarly illustrates the striking effect of load factor. This showing seems unique in indicating so concretely a definite difference in draft-tube proportions for changes in factors that usually enter not at all in fixing draft-tube proportions. It would be of interest, but probably very disappointing, if inquiry were made as to how many American plants had their draft-tube proportions influenced by such factors.

The author's conclusions are rather striking in their emphasis on the economical elasticity of draft tubes. The reason for conclusion No. 1, however, is not quite evident and further comment by the author on this feature would be of interest.

Arnold Pfau.[4] It may be said that the author's draft-tube investigations start with the assumption that the top of the draft tube is at a fixed elevation, so that costs due to depth of excavation are dependent only upon the selection of this depth.

The writer would point out that this top elevation, i.e., the setting of the turbine above the lowest tail-water level, depends upon and is materially affected by several factors, such as elevation above sea level and above all on the velocity of the water entering the draft tube. As is well known, the latter is related to the specific speed of a unit.

The selection of the type of turbine thus affects the depth costs materially. This is especially pronounced with the propeller-type turbine, having inherently very high entrance velocities of water into the draft tube and, thereby, requiring at the start a lower elevation of the top of the draft tube to minimize cavitation.

It seems that the use of Kaplan-type turbines for high heads has been somewhat overadvertised by our European competitors under Kaplan license. We find that an unbiased investigation as to economical costs of a Kaplan type and a Francis type discloses that, at certain heads and capacities, a Francis-type hydroelectric installation proves more economical as to over-all costs than a Kaplan type, for the following reasons:

1 For higher and higher heads, particularly exceeding 100 ft, the specific speed of a Kaplan type must be moderated, so that in fact its higher speed does not offer a material saving of generator costs over that of a Francis unit.

2 Likewise, the setting above (or better said, below) lowest tail-water level becomes very expensive due to excessive excavation.

3 The relatively higher percentage of overspeed above that of a Francis type involves a more costly generator.

4 The vertical thrust of the Kaplan type involves a more expensive thrust bearing.

5 For reasons of stability of speed, the inherent flywheel effect of a Kaplan-type hydroelectric unit must be materially greater than that of a Francis type. It is thus evident that all these factors will lead to a dividing line where the over-all costs point unmistakably to the selection of one or other type unit.

This paper as such is of value since it points out that, in the design of the draft tube, its costs must be weighed against the actual gain by reason of greater output due to higher efficiency. It is a case parallel with other design factors. For instance, a turbine runner with 0.25 per cent improvement in efficiency but with a greatly shortened life, due to cavitation, may prove less economical than a runner moderately less efficient but involving no outage and subsequent repair costs.

F. Schmidt.[5] The fundamental principles of the most economical draft tube must be applied to each individual case. One set of curves based on a given draft-tube design for a low-specific-speed Francis runner and prevailing construction conditions may not apply to a high-speed propeller-type runner under the same field conditions. Therefore, for each hydro project, a careful study of the draft tube must be made to determine the most economical design.

J. D. Scoville.[6] The principle that turbine efficiency increases with draft-tube depth and width is generally recognized. There are limitations, however, which make it difficult to use this fact in determining the economic proportions of draft tubes.

1 In a great many cases the draft-tube width is fixed by the scroll proportions and is not subject to an analysis such as the author's. The draft-tube depth is likewise controlled in some cases by the character of the material which must be excavated.

2 It is possible to offset deficiency in draft-tube depth by variations in design. It should be remembered that the tailrace excavation is to be considered as well as that for the draft tube. Fig. 8 of this discussion shows two draft tubes having identical area curves but different proportions. The width of both of them is the same. Draft tube B shows better efficiency than A, having more vertical height but a steeper upward slope of the horizontal leg. This might mean enough saving due to decreased tailrace excavation to offset the greater depth at the elbow. Deficiency in draft-tube depth can likewise be made up cheaply by

[4] Consulting Engineer, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis.

[5] Assistant Manager, Hydraulic Turbine Division, Allis-Chalmers Manufacturing Company, Milwaukee, Wis.

[6] Assistant Chief Engineer, S. Morgan Smith Company, York, Pa. Mem. A.S.M.E.

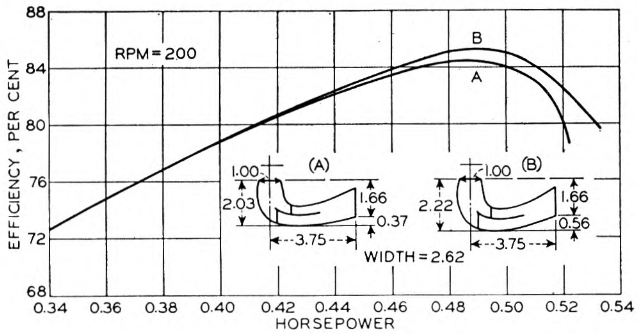

FIG. 8 TWO DRAFT TUBES HAVING IDENTICAL AREA CURVES BUT DIFFERENT PROPORTIONS

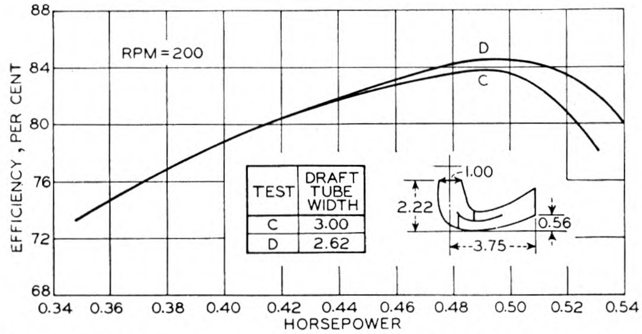

FIG. 9 EFFICIENCY CURVES WHERE SPLITTER IS USED IN HORIZONTAL LEG OF DRAFT TUBE

increased horizontal length, especially if this part of the tube is sloped upward.

3 The curve in Fig. 3 of the paper, showing increased efficiency with greater draft-tube width, might be correct for one manufacturer but is not right for all. By the use of a splitter in the horizontal leg of a draft tube, it is possible to offset the loss in efficiency due to a narrow tube. Fig. 9 of this discussion illustrates this graphically. In this case, it will be seen that the narrower of the two tubes shows the better efficiency. The area curves of these two draft tubes were identical.

The writer feels that the rigid application of economic principles to draft-tube design is too complicated to be feasible.

W. M. WHITE.[7] We are indeed indebted to the author for having presented this paper on the importance of draft-tube design as regards efficiency and financial return on hydroelectric developments.

[7] Manager and Chief Engineer, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

The writer does not agree with some of the conclusions which may be drawn from the data presented in the paper, but it is valuable to have this paper on record as a medium through which we may all offer contributions that will lead to a clarification of the all-important subject of depth of excavation for draft-tube design.

AUTHOR'S CLOSURE

In answer to Mr. Nagler's query, concerning the reason for conclusion No. 1: "For an installation of greater rated power than 16,000, the minimum points would move to the right;" it probably would have been clearer to have added, "provided all other factors affecting the curves remained constant." It is evident that, if the calculations had been made for a unit greater than 16,000-hp capacity, then the power loss, the revenue deficiency, and consequently, the capitalized value of the revenue deficiency would also be correspondingly greater. However, the total first cost and the capitalized value of the depreciation and maintenance would remain the same with the other factors remaining constant. Thus, when added together, the minimum points of the curves would move to the right of their former position showing that, with increased capacity, the more economic draft tubes would be both wider and deeper.

With regard to Mr. Scoville's limitation No. 1; the calculations would be of the greatest value where one of the dimensions of the tube was fixed due to some outside factor, such as certain required scroll proportions. In this case, the most economic depth could be found as discussed in the first part of the paper.

Limitation No. 2 is easily taken care of since the principles are flexible enough to allow the inclusion of various costs such as tailrace excavation which would not occur in every case. No doubt the actual calculations in practice would vary somewhat with each design but the method would remain basically the same.

As for Mr. Scoville's limitation No. 3, it must be pointed out that the theory put forth in this paper must be attempted only after the general type of draft tube has been decided upon. The shape of the efficiency-drop curves, as shown in Figs. 2 and 3 of the paper, is not changed materially over a wide range of tubes, varying from the straight-vertical type to the sharp-elbow type. However, these experimental data were obtained prior to 1927, at which time the use of a horizontal splitter was comparatively unknown on this continent. It is quite possible that a splitter might change the shape of these curves but this in no way detracts from the usefulness of the principles set forth in the paper. It is quite conceivable that another pair of average curves could be drawn for various shaped draft tubes containing splitter plates. In this case the most economic draft-tube proportions could be found as before, but based on the new experimental curves. This is just another example of one of the many fields which remain to be investigated in this study of economic draft-tube design.

Some Performance Characteristics of Deep-Well Turbine Pumps

By R. G. Folsom,[1] Berkeley, Calif.

Test data are presented on a comparable basis to indicate the relative performance of a series of deep-well turbine and propeller pumps. The range of performance characteristics obtainable with typical semiopen and closed impellers is shown and the losses introduced by axial adjustment of impeller-bowl positions are briefly discussed.

The centrifugal pump in all its many and varied forms has reached its high degree of perfection principally through trial-and-error methods of development. Although analytical analysis has been of value in the development of specific units, this process is restricted in application, due to the complicated flow conditions encountered in the centrifugal-pump impeller and case. The trial-and-error procedure does not insure the production of pumps of maximum possible efficiency but, from a commercial viewpoint, satisfactory results have been obtained through its application.

Since modern centrifugal pumps attain very high efficiencies, further research and development work will be of the most careful and painstaking type in order further to increase the performance. These investigations will include the study of isolated phenomena which control the flow through various portions of the pump. Such investigations are being pursued in many laboratories. Recent publications demonstrate that, in the past, various factors have been neglected which may have an appreciable effect on the measured performance of centrifugal pumps.

Detailed studies of various pump phenomena are being made at the pump testing laboratory of the University of California.[2] Certain results obtained from these investigations will be presented and briefly discussed in this paper. The work of this laboratory deals principally with vertical-shaft units, particularly the deep-well turbine and the propeller pump. Both of these types are examples of specialized centrifugal pumps, but the results obtained from the investigations will apply equally well to all centrifugal pumps. Some data were obtained on normal horizontal-shaft centrifugal pumps which are under investigation in the laboratory on a limited scale.

Comparison of Over-All Performance

The deep-well turbine pump has been developed to the stage where extremely high efficiencies are obtained for the design restriction of the diameter of the hole in which the unit is to be placed. Although pumps of this class appear to be essentially the same upon superficial inspection of the types of impeller flow passages, areas, and other features, the laboratory experiments demonstrate that the performances of units from various manufacturers vary over a wide range. In order to compare units of different design, shape, and size, use will be made of the term "specific speed," which is defined by the equation

$$n_{sq} = \frac{\text{rpm}\sqrt{\text{gpm}}}{H^{3/4}}$$

where

H = total head per stage developed by pump, expressed in feet of fluid being pumped.

The specific speed expresses the requirements for dynamically similar operation of all geometrically similar centrifugal pumps. In practice, the term "specific speed" has been found to be useful as an indication of the type of impeller required to meet certain pumping conditions. The specific speed always refers to the point of maximum efficiency of the pump-performance characteristic. The plotting of maximum efficiency versus the specific speed for many pumps throughout a wide range of specific speeds, and drawing the maximum envelope for these points will give a curve which indicates the maximum efficiency obtainable by a pump of a given specific speed. Such a curve was prepared by Hollander in 1937 for optimum efficiency of single-stage

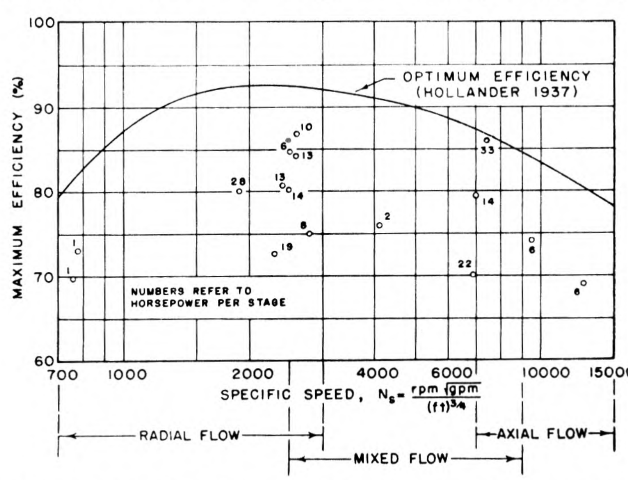

Fig. 1 Curve for Optimum Efficiency of Single-Stage Pumps; Points of Maximum Efficiency of Deep-Well Turbine Pumps

pumps and published by Daugherty,[3] and is reproduced in Fig. 1 of this paper. This particular curve applies to single-stage pumps of 12-in. nominal size. Due to hydraulic leakage loss and mechanical friction, larger pumps would be expected to show higher efficiencies and smaller pumps lower efficiencies.

The maximum efficiency of deep-well turbine units investigated by the pump testing laboratory is also plotted in Fig. 1. In order to indicate the relative size of the various units, the number alongside the plotted point refers to the horsepower per stage of the pump. It will be noted that the relative performance of commercial deep-well turbine pumps varies greatly; in fact, this limited series of tests indicates 15 points or more in efficiency as

[1] Assistant Professor of Mechanical Engineering University of California. Mem. A.S.M.E.

[2] "University of California Pump-Testing Laboratory," by R. G. Folsom, *Mechanical Engineering*, vol. 60, 1938, pp. 301–305.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

[3] "Hydraulics," by R. L. Daugherty, McGraw-Hill Book Company, Inc., New York, N. Y., 1937.

the difference between the best and poorest units. This spread indicates that the deep-well turbine is far from being a standardized product in which the performance of units from different manufacturers is very similar.

The efficiencies of deep-well turbine pumps are generally lower than those of normal horizontal-shaft units, as all hydraulic losses between pump entrance and immediately beyond the discharge elbow and mechanical losses of the drive shaft are charged against the pump. The laboratory tests were made on pumps with riser columns about 10 ft long, thus having small losses as compared with the usual field installation. A slight increase in the efficiency percentage can be realized as the pumps are multistaged to four or five stages. No consideration is given to this feature in this paper.

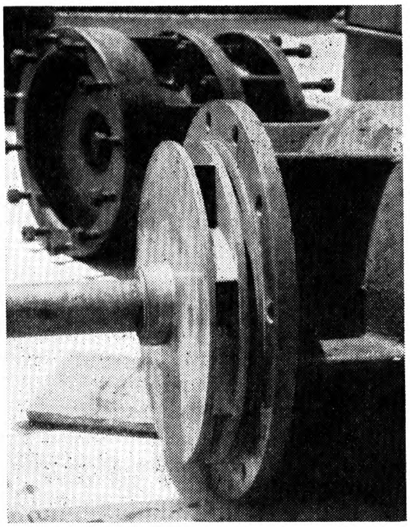

Fig. 2 Impeller for Radial-Flow Deep-Well Turbine Pump

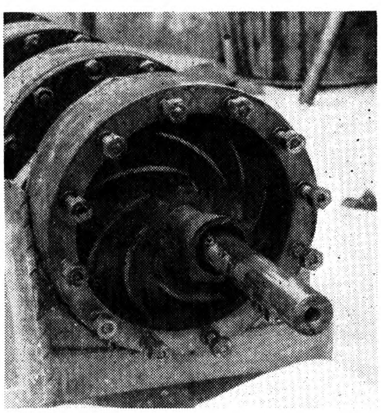

Fig. 3 Type of Semiopen Impeller for Mixed-Flow Pump Unit

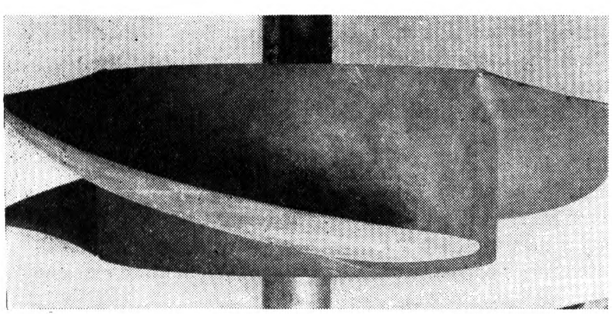

Fig. 4 Axial-Flow Unit or Propeller Pump

Deep-Well Turbine Pump Losses

There are many ways of classifying centrifugal pumps but, on the basis of the impeller shape and design, there are three principal types, namely, radial-flow, mixed-flow, and axial-flow units. The three characteristic types are illustrated, respectively, in Figs. 2, 3, and 4. These different types of impellers are produced with a variety of shapes and construction features. The radial- and mixed-flow impellers are used extensively in deep-well turbine pumps. The two types of construction usually adopted are the closed impeller, as illustrated by the radial-flow unit Fig. 2, and the semiopen impeller, as illustrated by the mixed-flow unit, Fig. 3. The axial-flow unit Fig. 4, which is sometimes referred to as a propeller pump, may have blades of widely varying shapes and areas. The principal impeller features other than

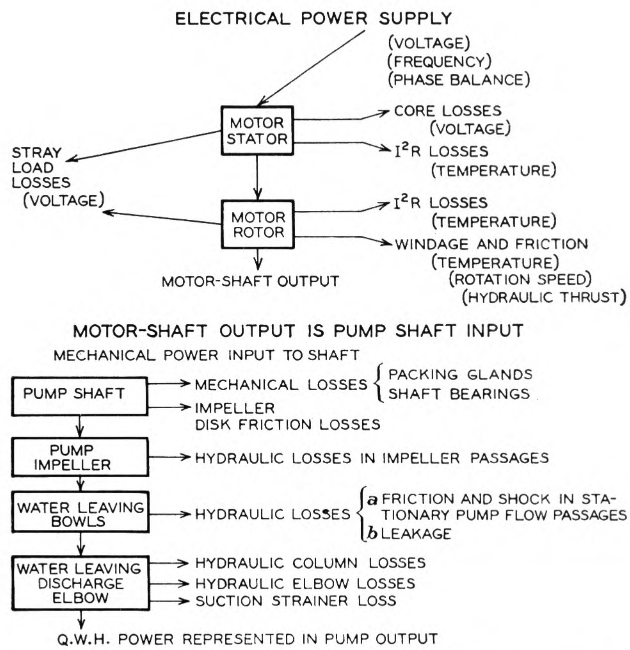

Fig. 5 Energy Flow Sheet and Losses for Typical Deep-Well Turbine Pump Installation

(Values in parentheses indicate variables controlling magnitude of losses.)

blade shape and passages are the methods of attaching the impeller to the rotating shaft, and the method of sealing the suction intake from portions of the pump under the approximate impeller discharge pressure. The methods of attaching the impeller to the shaft will not be discussed in this paper, as this is a purely mechanical problem and in no way affects the hydraulic performance of the unit so long as the relative position of impeller and shaft is fixed. Sealing methods vary widely and have a large influence on the hydraulic performance of the pump.

A centrifugal pump consists essentially of an impeller rotating in a housing with sealing glands on a rotating shaft to prevent external leakage into or from the pump, and with seals inside the pump between the high- and low-pressure areas. In general, the deep-well turbine pump has these features of the normal centrifugal pump but, in addition, it is located at the lower end of a flow column which surrounds the drive shaft, powered at the surface. In order to meet these conditions, special arrangements of impeller and bowl flow passages, guide and thrust bearings, must be made. The chart, Fig. 5, indicates energy flow and losses for a typical deep-well turbine unit.

Axial Adjustment

The relative position of the impellers with respect to the pump bowls will be adjusted by the shaft nut at the motor coupling. For gear-head or other drive, provision is made for vertical adjustment of the shaft. This adjustment may be of primary importance in many types of deep-well turbines since it controls the leakage quantity which is a short-circuit loss in the pump, and is a direct loss in the performance of the unit as a whole.

The radial clearance between the wearing ring and the bowl is the principal control of the leakage quantity of a normal closed impeller. For this type, axial adjustment of the impellers has little or no effect on the pump performance. Some closed impellers are made without the usual "skirt" or wearing ring, but include a sealing surface which depends upon axial adjustment to control the leakage. The semiopen impeller depends upon a close fit between the blades and bowl to reduce leakage to a minimum.

In so far as possible, the results of all experiments have been expressed in dimensionless form. For example, the discharge rate

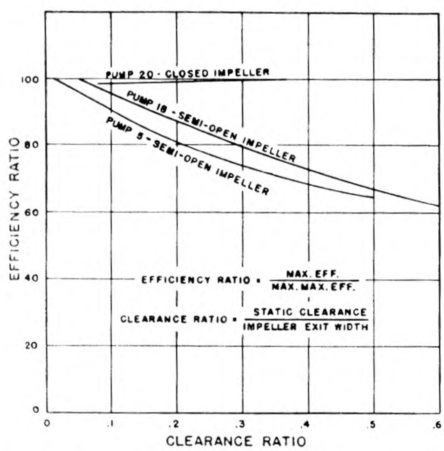

Fig. 6 Typical Clearance and Efficiency Results of Semiopen and Closed Impellers

at any point of operation is expressed as a ratio of actual discharge at that point with the discharge corresponding to the point of maximum efficiency. In a similar manner, clearances of impellers (the clearance being the axial distance between the lower portion of the impeller and the bowl) are expressed as a ratio of clearance between the bottom of the impeller and bowl to width of impeller flow passage at the outlet. Fig. 6 shows typical results of clearance and efficiency of semiopen and closed impellers. This graph demonstrates the relatively small change in performance of a normal closed impeller as compared with that of a semiopen impeller. Closed impellers with special seals for vertical-shaft adjustments have characteristics similar to all semiopen impellers. The work of this laboratory has shown that, when a normal closed impeller is adjusted with reasonable care to the middle of the adjustment range, the variations in performance for different adjustments are less than the error in the usual performance tests.

Fig. 7 illustrates the typical performance curves for semiopen impellers with various clearance ratios. Summing up the characteristics of deep-well turbine pumps, as affected by variations in axial adjustment, we find (1) the head, discharge, hydraulic thrust, power, and efficiency of normal deep-well pumps with normal closed impellers are not measurably affected by axial clearance within the possible range of axial adjustment; (2) at

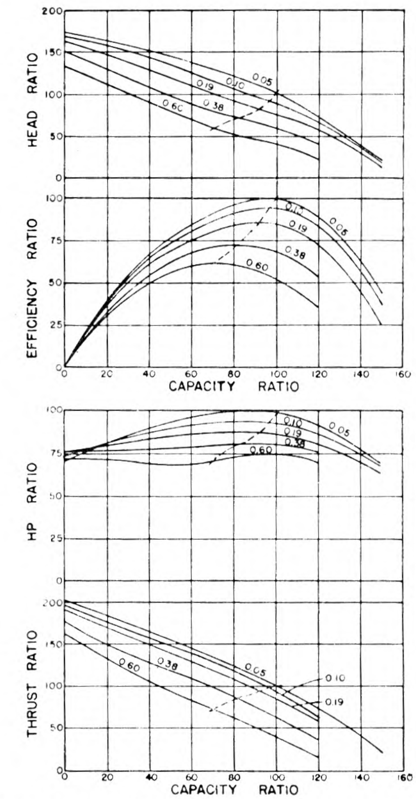

Fig. 7 Performance Curves for Semiopen Impellers With Various Clearance Ratios

(Figures on curves refer to clearance ratio; dotted lines represent conditions at maximum efficiency.)

constant head, the efficiency, power input, hydraulic thrust, and discharge of deep-well pumps with semiopen impellers decrease with increase of axial clearance; (3) closed impellers, equipped with sealing devices subject to axial adjustment, exhibit performance characteristics similar to those of semiopen impellers.

The foregoing discussion is restricted to new pumps and in no way considers the application of specific designs to conditions where wear, due to abrasive particles in the water, is of importance. These data are presented to call attention to the fact that correct adjustment of some types of design is necessary to obtain and to maintain the highest possible efficiency. These conclusions apply equally well to horizontal-shaft or other centrifugal pumps having impellers similar to those described.

Conclusions

The performance of similar commercial deep-well turbine pumps varies over a considerable range. Further tests may indicate a greater range of divergence.

A clear understanding and a careful study of losses will be necessary for a further improvement in performance of present high-efficiency units.

Initial and maintained high efficiency of centrifugal pumps requires careful axial adjustment for certain types of impeller construction. No data are presented regarding the change in performance with wear, a factor which may be of primary importance in some installations.

Acknowledgment

The author wishes to acknowledge assistance furnished by the personnel of Works Progress Administration Official Project No. 65-1-08-113, under the sponsorship of the department of mechanical engineering, University of California.

Discussion

J. W. Daily.[4] It has been observed in the hydraulic machinery laboratory of the California Institute of Technology that some pump designs of the horizontal-shaft, medium-specific-speed type show definite improvements in head and efficiency when subjected to increased submergence. For a few pumps, this improvement persists even after inlet heads of 50 ft and above are reached. The important conclusion from such results is that many pumps designed for and installed with relatively large submergence margins must actually operate under cavitating conditions. Consequently, in tests made for research purposes, adequate representation of pump performance cannot be made without reference to the submergence.

The author describes tests of deep-well turbine pumps and compares them on the basis of over-all performance. No information is given as to the method of testing, but an earlier article by the author[5] indicates that variable-submergence tests are possible. However, the absence of any consideration of inlet pressures here leads to the conclusion that the tests are made at practically constant submergence. Referring to the plotted points of the author's Fig. 1, it is noted that there is a lack of system to the variation in maximum efficiency (for a particular specific-speed range). This allows some speculation as to existence of a behavior in deep-well pumps similar to that observed in the radial-flow, horizontal-shaft type. It would be particularly interesting to learn whether or not the performance of units with semiopen impellers of medium and high specific speeds would improve with increased pressure on the suction side. Submergence is likely to be important not only when operating with normal clearances, but also when the clearances are large. The writer recognizes that, practically, the physical setting dictates the submergence but, as the author has indicated, improvement in design dictates painstaking investigation for optimum results. It is felt that a quantity of such information on a variety of designs should be useful to the designer.

J. M. Hait.[6] In my opinion, this paper, concerning the results of tests on deep-well turbine pumps, represents the most accurate and unbiased report on relative performances which has yet been published. The general results agree very well with similar tests which we have made with entirely different laboratory equipment. There are a few comments the writer would like to make concerning the presentation of these test results:

In Fig. 1 of the paper, the maximum efficiencies of several types of deep-well turbines have been plotted against specific speeds. An optimum-efficiency curve is shown, which, in effect, indicates the degree of development of the pump tested. As far as we can determine, the pump used for the optimum-efficiency curve was a 12-in. horizontal centrifugal pump. The question arises as to whether it is the best basis for comparison, in determining degree of development of relatively small deep-well turbine pumps, since the radial space for conversion of velocity head is definitely restricted in the case of the latter, while the discharge from the impeller of the horizontal-type pump is at the plane of the impeller rather than normal to it.

The number of stages of the various units which were tested and plotted is not noted, but it is stated in the paper that a slight increase in efficiency may be realized by multistaging to 4 or 5 stages. It has been our experience that the additional efficiency by multiple staging varies with the class of pumps, that is, whether they are straight centrifugal or mixed flow and that, in the average case, pumps with 8 stages or above perform with an efficiency approximately 6 points higher than a single-stage pump of the same type. Thus, it seems that the number of stages tested is an important item when efficiencies are to be compared.

In showing the clearance between the semiopen impellers and the bowl, a clearance ratio is used. The writer feels that it would also be of value if the reader were to know the absolute clearances involved, in which he is dealing, and as to whether these clearances are those only obtainable by expert operators in a laboratory or whether they can be easily obtained by an average operator with field settings of the ordinary range of magnitude between 200 and 400 ft. The change in length due to the torque imposed on the shaft in combination with the hydraulic thrust, and the upward reaction of the water being turned as it enters the impeller create a change in the shaft length, complicating the field adjustment materially. Accordingly, the writer feels it to be of importance, in considering this study, to know the order of magnitude of the clearances involved.

In the summary of the effects of axial adjustments on the performance, it is stated (3) "closed impellers, equipped with sealing devices subject to axial adjustment, exhibit performance characteristics similar to those of semiopen impellers." It should be noted that this statement is not true when the closed impeller has both a close radial-skirt clearance and a seal at the bottom of the skirt with the customary axial adjustability.

M. Mull.[7] This discussion will be made from a commercial standpoint, pertaining to the issues discussed in this paper, and their influence upon the recommendation and application of different types of deep-well turbine and vertical wet-pit pumps as manufactured by our company. The writer's comments are based on his observations and experiences with this type of pumping equipment throughout the Northwest during the last 15 years.

The deep-well turbine can truly be called a western product, as it had its inception on the Pacific Coast. For this reason its development can easily be studied from many of the older installations, to determine the result of the various ideas on design and construction. It is true that many design and development methods have been employed to obtain the highly efficient deep-well turbine and other vertical modifications offered on the market today. As stated by the author, "the trial-and-error procedure does not insure the production of pumps of maximum possible efficiency," and it may be years before these actual results are accomplished from a truly commercial standpoint. Accuracy and long life in pumping equipment are achieved, particularly by eliminating so-called special features, using only conservative and time-proved designs.

It is not possible today, with modern manufacturing methods, to develop new designs under the regular schedule of production, because standard current modifications are used in order to speed up production and fill orders on definite delivery schedules. Development work is done on certain-size pumps which are found to be lacking in performance with competitive units of the same characteristics. Competitive bidding on the larger government and municipal specifications is responsible for keeping definite records of performance on the various sizes offered. Many of the individual points of design are constantly under investigation and upon the results are based new development programs from year to year. This development schedule is studied and a definite program laid out for each year, in order to keep abreast of competition.

Today it is possible to offer vertical close-coupled wet-pit-type

[4] Manager, Hydraulic Machinery Laboratory, California Institute of Technology, Pasadena, Calif. Jun. A.S.M.E.

[5] "University of California Pump-Testing Laboratory," by R. G. Folsom, *Mechanical Engineering*, vol. 60, 1938, pp. 301–305.

[6] Chief Engineer, Peerless Pump Division, Food Machinery Corporation, Los Angeles, Calif. Mem. A.S.M.E.

[7] Sales Engineer, Fairbanks, Morse & Company, Seattle, Wash.

pumps for application in the high-head field, with over-all efficiencies exceeding those of horizontal split-case-type centrifugal pumps within their range. This is even more evident where it is necessary to use multistage horizontal units. Fewer hydraulic losses are experienced in the completed installation using the vertical deep-well turbine pump, as compared with the use of horizontal multistage or series-connected pumps with all of the appurtenance piping.

The term "specific speed," which is applied in a nontechnical manner in the commercial field, is a watchword in the selection of pumps which will give satisfactory performance for the operating conditions specified. Propeller, axial-flow, mixed-flow and deep-well turbine pumps, as well as all other centrifugal pumps, have a specific-speed range in which they are designed to operate with quiet smooth performance and without cavitation, and at a minimum of submergence. It is always desirable to select pumps, particularly in the low-head field, which come well within their specific-speed range. However, there are times, in the application of such pumps, when it is necessary to sacrifice the most efficient pump, which is within the correct limits of specific speed for that one modification, because of the high first cost of the equipment. This is found to be true on extremely low-head jobs, where the best performance will often be found at extremely low revolutions, thus causing high cost of the electric motor. (Specific speed must not be confused with pump rpm.)

The author has expressed in an enlightening and graphic manner, the losses to be accounted for in the deep-well turbine pump. The Hydraulic Institute has also made these points very clear, and through this source standardization along this line has been effected to a large degree.

The observations, outlined in the paper, on the subject of "axial adjustment," are important, as they pertain to actual results from operation of deep-well turbine pumps. It has been found, with this type of pumping equipment, that enclosed impellers with a long "skirt" and arranged for a vertical adjustment of ample proportions, will give better efficiency in the hands of the average operator, than will other types of semiopen impellers.

The conclusion, offered in this connection, is not only in accordance with those found in the field, but substantiates the fundamental reason for this type of design. Where highly efficient well-built deep-well turbines are selected for applications requiring continuous service, the enclosed-type impeller should be specified. The construction throughout the bowl assembly should be of a high order of precision, with bronze bearings and renewable wearing rings both in the bowl casting and in the impeller skirt.

For deep-well turbines, which are required to change capacity throughout a small range, as is often specified in cases where the capacity of a well may be less at one season of the year than at another, the semiopen impeller is desirable. With this adjustment, the vane of the impeller is lifted from the seat in the bowl, thus allowing a greater clearance as described by the author. However, this effect to reduce the capacity is always a costly one as it decreased the efficiency of the pump.

The vertical deep-well pump, employing good standard forms of design which have been proved in service for the last decade, has now been developed to a point where it can compete in efficiency and cost with the horizontal centrifugal pump. It has found a very definite application for certain types of pumping requirements. The deep-well turbine should be carefully applied. When a pumping problem is under consideration, a definite analysis should be made to aid in arriving at an unbiased choice between the horizontal centrifugal pump and the vertical deep-well turbine. Many manufacturers of only one type of pumping equipment will almost invariably recommend and try to sell their products regardless of the correct application involved, which practice should be carefully guarded against by all engineers.

Author's Closure

The problem of the correct magnitude of submergence to obtain satisfactory centrifugal-pump performance is always present but becomes particularly troublesome for large-specific-speed units. Mr. Daily has called attention to this important variable, the discussion of which was omitted from the paper. Of equal importance are the fluid velocities, in magnitude and direction, at the impeller inlet. For the tests reported at the large specific speeds, values larger than 5000, the submergence varied from about 3 to 4 ft, and the other units had submergences from about 5 to 12 ft. All tests were conducted with the pump installed approximately in the middle of the laboratory 8 × 8-ft test pit.

Submergence has much less influence on multistage pumps than single-stage units as the first stage is the only one of the multistage construction that is affected to an appreciable extent by the pressures and velocities at inlet. Thus with the normal multistage deep-well turbine pump, the small change in performance at increases in inlet head becomes insignificant. With single-stage large-specific-speed units, the submergence is one of the limiting factors restricting application. On the basis of unreported tests, increases in efficiency of less than one per cent would be expected for these units with relatively large increases in inlet heads.

Results have not indicated appreciably different behavior with respect to inlet heads for semiopen impellers with different clearances. All semiopen impeller pumps tested were multistage and thus the phenomena would not be apparent, as it occurs in the first stage only.

Mr. Hait is correct in his conclusion that the optimum efficiency applies to 12-in. horizontal pumps. This curve was selected to indicate degree of perfection as it has been previously published, and sufficient data have not been made available to the author by deep-well turbine manufacturers to allow him to develop a generalized curve representing all companies. Some such curve should be used when making a comparison as the expected maximum efficiency of any type of centrifugal pump varies with the size and specific speed.

Mr. Hait indicates increases in efficiency through multistaging somewhat in excess of usual values and that appreciable increases occur for a larger number of stages. This increase in efficiency is a function of the pump design, size, and relative distribution of losses. Thus, if bowls only are considered, a different value of efficiency increase will exist than if the complete pump with riser column discharge elbow and suction piece are included. The number of stages for the units tested are the following:

Specific speed	Stages	Specific speed	Stages
760	6	2600	5
780	6	2600	4
1900	2	2800	2
2300	1	4100	10
2400	2	6900	1
2500	2	7200	1
2500	7	7400	1
		9500	1
		12700	1

Axial adjustment of deep-well turbine pumps is considered in some detail in a recent publication.[8]

One series of tests with an impeller having radial-skirt clear-

[8] "The Axial Adjustment of Deep-Well Turbine Pumps," by Morrough P. O'Brien and Richard G. Folsom, University of California Press, Publications in Engineering, vol. 4, no. 2, 1940, pp. 19–26.

ance and axial adjustability showed performance changes with axial adjustment similar to the semiopen impeller characteristics except that the magnitude of the performance change was much reduced.

Mr. Mull's remarks from the field standpoint are gladly received, although the author would like to point out that some of the conclusions are open to considerable argument. It is almost impossible to specify any one type of construction as a standard because of the wide variety of conditions to which these pumps are applied.

Centrifugal-Pump Performance as Affected by Design Features

By R. T. KNAPP,[1] PASADENA, CALIF.

This paper presents some of the results of a study of Grand Coulee pumping-plant characteristics. The research program was conducted for the Bureau of Reclamation by the California Institute of Technology in its hydraulic machinery laboratory (1)[2] and has been in progress since January, 1938. While the principal object was to determine the operating features for pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is believed by the author that investigations of a somewhat similar nature offer the most reliable means for securing the characteristics desired in hydraulic units, both pump and turbine, for practically any given set of conditions.

Need for Information

The need for a thorough study of the Grand Coulee pumping plant arises basically from the tremendous size of the units proposed, i.e., 1600 cfs capacity with a motor of approximately 65,000 hp for each pump. Full knowledge of the pump characteristics is required, due to the great range of operating head, from 295 to 367 ft, which is caused by the variation of the inlet head, from + 80 ft to + 5 ft. Also, the probable operating cycle makes it desirable to have as high a capacity as possible when operating against the high head. In addition to the matter of the proper relationship between the capacity and head over the operating range, the following items were considered to be important for satisfactory pump operation:

(*a*) Freedom from cavitation over the entire operating range
(*b*) Low radial forces due to hydraulic unbalance.
(*c*) Freedom from unstable regions within the operating range.
(*d*) Constant-speed operation.
(*e*) Satisfactory transient performance which will permit simple shutdown procedure.
(*f*) Suitable characteristics when operating as a turbine to provide the possibility of utilizing units for peak-load power development.

Furthermore, to obtain the lowest-cost unit, including the motor, it was necessary to determine the maximum permissible operating speed for which units could be obtained that could also satisfy the foregoing requirements.

Model and Prototype Pumps

The pumps contemplated for installation at Grand Coulee are unprecedented in size and power requirement. They are to be installed vertically and will be of the single-stage single-suction type. Each unit is expected to have a capacity of about 1600 cfs. The normal head against which it is to deliver is about 295 ft. Approximately 65,000 hp will be required. It is estimated that the pump for this duty will have a discharge nozzle of 8 to 10 ft diam, an impeller of from 12 to 17 ft diam, with an eye dimension of from 6 to 10 ft. The width of the impeller at discharge will be in the neighborhood of 20 to 36 in. The speed range is from 150 to 200 rpm or possibly slightly higher.

Although the present study is not a "model" study, but rather an investigation of the possible characteristics of the machines, the units tested in the laboratory may be thought of as models in order to visualize a size comparison. On this basis the model ratio would range from $12^1/_2$ to 15. The studies were all made at or near the full prototype head. The capacities varied from 7 to 10 cfs at the operating point. The horsepower requirements fell within the range of 290 to 400. All of the units had maximum efficiencies in the vicinity of 90 per cent. The discharge-nozzle diameters were 8 in. The impellers varied from $12^1/_4$ to $14^1/_2$ in. diam, with eyes of from 6 to 8 in. and with discharge widths of from $1^1/_2$ to $2^1/_2$ in. Testing speeds fell between 2100 and 2600 rpm.

It will thus be realized that these test pumps are comparatively large machines, therefore, accurate passages and vane angles may be expected. Furthermore, the large size and high efficiency of these units permit drawing direct conclusions concerning the performance of prototypes. To reduce the number of variables, several cases were designed to operate with the same impeller, thus making it possible to ascertain clearly the characteristic-performance differences between such case types as single-volute, double-volute, and fixed-vane-diffusor constructions.

Presentation of Data

In order to make the results from the different units directly comparable, the characteristic curves have been plotted on a percentage basis. The normal operating head at Grand Coulee is 295 ft. This has been taken as 100 per cent. The capacity at this head is therefore designated as 100 per cent. The maximum efficiency of each unit has been used as the reference value for that unit, and has been plotted as 100 per cent. It should be noted that the maximum-efficiency point will not coincide necessarily with the 100 per cent capacity and head point. Whenever plotted, torques and horsepowers have had, as a 100 per cent reference, the corresponding values at 100 per cent capacity and head. For example, since the prototype-head range is from 295 to 367 ft, this system gives an operating-head range of from 100 to 125 per cent.

Comparison of Normal Operating Characteristics

Capacity-Head and Efficiency Characteristics. During the course of this program, several series of experiments were made in which a single impeller was tested in two or three different cases. In order to establish a basis for the discussion of the results, a brief résumé of the respective functions of the impeller and the case of a centrifugal pump seems desirable.

The impeller adds energy to the fluid flowing through it. At the discharge from the impeller, this added energy is in two forms: (*a*) an increase in pressure, and (*b*) an increase in velocity. The case has two functions: (*a*) to collect the fluid as it discharges

[1] Associate Professor of Hydraulic Engineering, California Institute of Technology. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., September 3–6, 1940, of The American Society of Mechanical Engineers.

around the impeller periphery, and (*b*) to transform a large part of the velocity into pressure with as little loss of energy as possible.

If an impeller could be tested alone under such conditions that, for all rates of flow, the discharge would be uniform around the periphery, its basic operating characteristics would be determined. A perfect case would be one which would have no losses over the entire operating range; therefore the combination of the impeller, operating in such a case, would have the identical performance characteristics which were obtained from the impeller operating alone. Since no real case is without losses and, furthermore, since no real case is equally efficient over the entire operating range, the performance of the unit as a whole is always lower than that of the impeller alone. The deviation will be least in the zone in which the case characteristics match the impeller

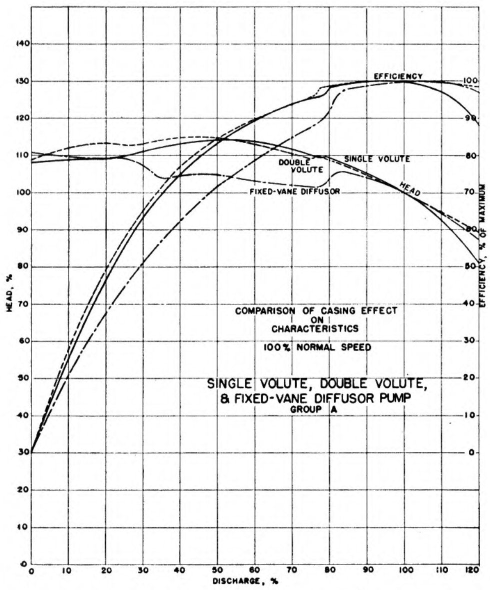

Fig. 1 Comparison of Casing Effect on Pump Characteristics; Group A, 100 Per Cent Normal Speed

characteristics to best advantage, and will increase on both sides of this zone. For a good pump, the case must match the impeller within the high-efficiency zone of the latter.

The case will affect the over-all performance of the pump in two ways: (*a*) through energy losses in the case itself, and (*b*) in additional energy losses induced in the impeller. Fundamentally, the case can affect the impeller performance only in one way, i.e., by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. If the discharge is not uniform from all parts of the impeller periphery, it follows that there must be pulsating flow in the impeller passages, nonuniform entrance conditions at the eye, and presumably increased losses, both in the impeller and in the case. In this simplified picture, the secondary effects of the impeller shrouds, the circulation existing between them, and the casing walls and leakage losses to the suction sides are disregarded.

From this discussion, it will be realized that a comparison of the characteristics of different units, made up of various types of cases operating with the same impeller, resolves itself into a comparison of the relative matching of these cases to the impeller and of casing losses, both intrinsic and induced in the impeller. Fig. 1 shows such a comparison for a series of units designated as group A. The first unit was designed as a single-volute pump to operate at a prototype speed of 150 rpm. The double-volute case was then constructed, using the same design methods. It was anticipated that, if everything worked out satisfactorily, the performance of the double-volute pump would be the same as that of the single-volute unit. The fixed-vane-diffusor case was designed around the same impeller.

If the curves for the single- and double-volute cases are compared, a striking difference is observed in the high-capacity region. The head curve for the double-volute case does not fall off as rapidly as that for the single-volute pump, and the efficiency also remains higher. The same is true to a lesser extent in the low-capacity region. However, the maximum efficiency is about the same. Since these maximum efficiencies are high, both cases are very satisfactory in the region of the design point, but the double-volute case apparently matches the impeller characteristics better in the low- and high-capacity regions. It must be remembered that, because of the two passages, the double-volute case has a lower effective hydraulic radius and, therefore, a higher skin-friction loss. For this reason, the wide region of high efficiency is all the more surprising.

The fixed-vane-diffusor case, operating with the same impeller, shows the same high maximum efficiency observed in the other two cases. However, the characteristic curves are quite different in shape. The maximum-efficiency point comes at a somewhat higher capacity for the diffusor case, and this maximum efficiency is not sustained over as wide a region. This is reflected in the head-capacity curve. It will be noticed on both sides of the design point that the diffusor-case head curve lies

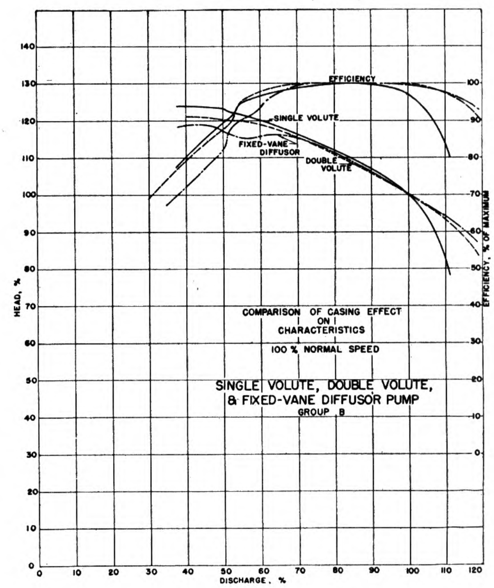

Fig. 2 Comparison of Casing Effect on Pump Characteristics; Group B, 100 Per Cent Normal Speed

under that of the double-volute curve. In the low-capacity region, i.e., from zero up to 75 per cent, the efficiency of the diffusor is markedly lower than that of the other two cases. This is probably the result of the discrepancy between the angle of the fixed guide vanes and that of the flow leaving the impeller under these conditions.

Fig. 2 shows the same comparison for an entirely different set of cases, working with another impeller. This series of units, group B, was designed for a prototype speed of 180 rpm in comparison with the 150-rpm speed of group A. Since the head and capacity are fixed, this 20 per cent increase in operating speed results in a 20 per cent increase in the specific speed as well, which corresponds approximately to a 16 per cent decrease in the diameters of the impeller and the base circle of the case.

The relative performances of the single- and double-volute cases are practically the same as those observed for group A, except that the single-volute pump shows its peak efficiency between 80 and 90 per cent of design discharge. This indicates that the case is too small for the specified conditions. The result is that, at the normal operating point, the efficiency is only about 97 per cent of the maximum. This accounts for the fact that its head-capacity curve is apparently above those for the double-volute

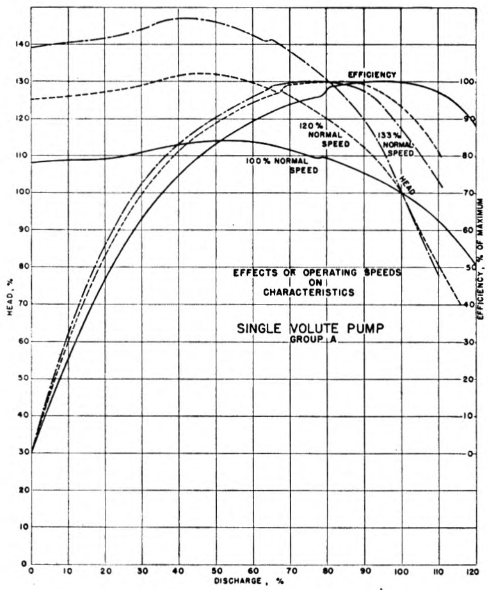

FIG. 3 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; SINGLE-VOLUTE PUMP

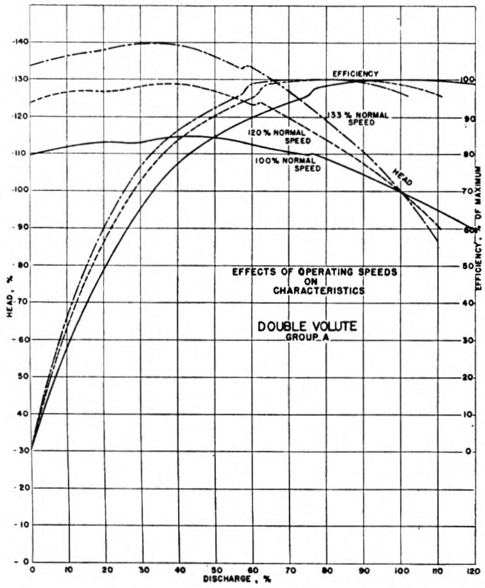

FIG. 4 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; DOUBLE-VOLUTE PUMP

type and fixed-vane-diffusor case, i.e., the steepness is obtained by sacrificing efficiency. The double-volute case again shows a surprisingly wide range of high-efficiency operation but, in this series, the diffusor case nearly duplicates its performance. However, the sharp drop in efficiency for the low-capacity region is again observed to be a diffusor-case characteristic.

Choice of Operating Speed. A given pump is designed to operate at a definite specific speed. In general, test results show that the unit has its maximum efficiency at this condition. However, if the performance characteristics show a reasonably broad zone of high efficiency, it may be possible to secure a better agreement between the pump characteristics and the field requirements if a different operating speed is chosen. The effect of the choice of operating speed may be observed in Figs. 3, 4, and 5. Fig. 3 shows the performance of the single-volute unit of group A operating at speeds of 100, 120, and 133 per cent of the design value. Fig. 4 presents the corresponding performance of the double-volute case, and Fig. 5 that of the fixed-vane pump.

All three units show the same trend, i.e., a marked steepening of the head-capacity characteristics with increase in operating speed. A closer examination of the three sets of curves shows that there are apparently two causes for this increase in steepness, (*a*) an increase due to the normal increase in the steepness

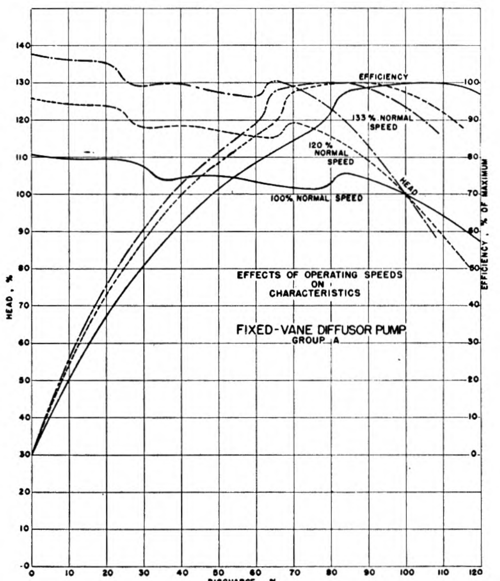

FIG. 5 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; FIXED-VANE-DIFFUSOR PUMP

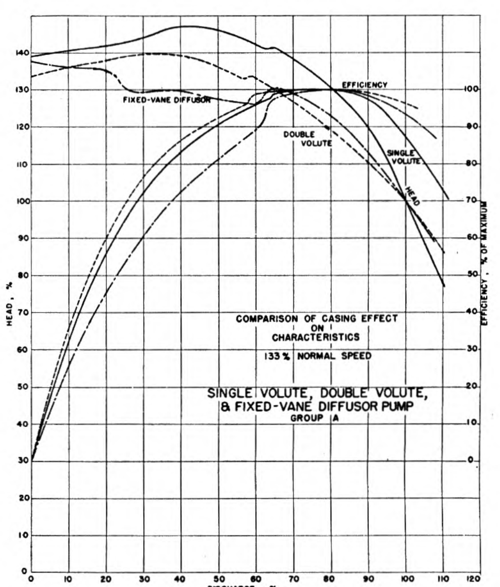

FIG. 6 COMPARISON OF CASING EFFECT ON PUMP CHARACTERISTICS; GROUP A, 133 PER CENT NORMAL SPEED

of the impeller characteristic, as the capacity is increased, and (*b*) an increase in steepness due to a decrease in the efficiency of the case. Fig. 4 illustrates the effect of the former. It will be noted that, in the region of from 80 to 100 per cent discharge, the efficiency is high for all speeds, in fact, the lowest value is 96.5 per cent of the maximum. Thus, the change in steepness for this machine must be due largely to the shape of the impeller characteristics. The fixed-vane pump, Fig. 5, shows a larger variation in steepness, but the efficiency drops to 92 per cent in the same capacity range. Likewise, the single-volute pump, Fig. 3, shows an even greater variation in steepness, but the efficiency goes down to about 87 per cent of the maximum.

The difference between these three cases, when operated at the higher speed, is shown very clearly in Fig. 6. Here, the performance characteristics for the same three units, presented in Fig. 1, are plotted for a speed of 133 per cent of normal. At the design speed of Fig. 1, the head-capacity characteristic of each of the three cases shows about the same slope in the vicinity of the operating point. At the 33 per cent overspeed, however, the difference in steepness is quite marked.

From these comparisons, it would seem that, in view of the factors so far considered, the steepness of the head-capacity characteristics can be varied appreciably by choosing the speed at which the pump is to operate. If the choice is limited to speeds within the high-efficiency range, slight loss accompanies the variation. The double-volute case offers the widest possibilities within these limits because of its broad zone of high-efficiency performance. If steeper characteristics than those corresponding to the basic impeller performance are desired, they can be obtained only through sacrifice of efficiency. It should be remembered, however, that in this investigation no attempt has been made to explore fully the possibility of varying the impeller characteristics themselves.

Minor Operating Features

Hydraulic Balance and Radial Thrust. In the section, "Comparison of Normal Operating Characteristics," it was stated that the case can affect the impeller performance only by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. Since this is an important feature for pump operation, it was thought desirable to make some experimental determinations of the pressure variation in the volute for the different types of cases. Consequently, piezometer connections were installed in the various cases—they were at constant radius. The piezometers for each case were spaced around a circle the diameter of which was slightly greater than the impeller and they covered the full 360 deg. Thus, the readings from them give a good picture of the pressure distribution around the impeller discharge.

Figs. 7, 8, and 9 show these measurements for the three cases of group B. The ordinates of all three curves are the static pressure at the piezometer connections, expressed in a percentage of the normal head produced by the pump. If the measurements for the single-volute pump, Fig. 7, are studied, it will be seen that the pressure distribution is reasonably uniform in the vicinity of the normal capacity; in fact, the most uniform distributions of those shown seem to be for the 93 per cent capacity. A glance at Fig. 1, shows that this is about the point of maximum efficiency. For higher and lower capacities, the pressure distribution is far from uniform and must affect the impeller discharge appreciably.

Fig. 8 shows that, for the double-volute pump, conditions are quite similar except that, of course, there are two pressure cycles in the 360 deg of the case. It will be noted here, however, that the range of pressure variation is considerably lower than in the corresponding single-volute case, although the basic design factors are similar.

Fig. 9 shows that the fixed-vane-diffusor pump has an even lower range of pressure variation. It should be remembered that these pressures are taken at a diameter corresponding to that of the impeller, i.e., at the inner side of the guide vanes. It will be seen that the pressure distribution is still nonsymmetrical. This is presumably due to the effect of the single volute on the outside of the guide vanes proper.

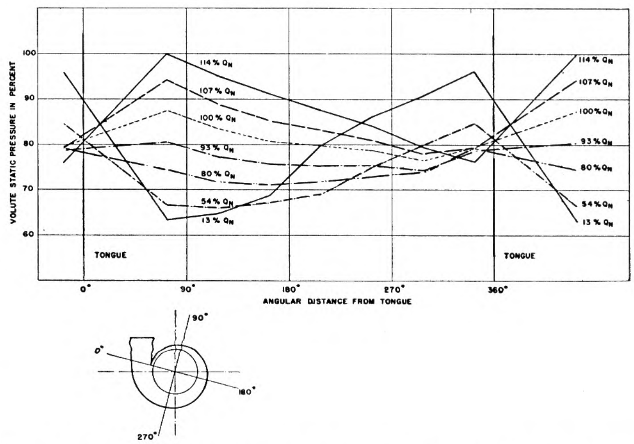

Fig. 7 Static-Pressure Distribution; Single-Volute Pump, Group B

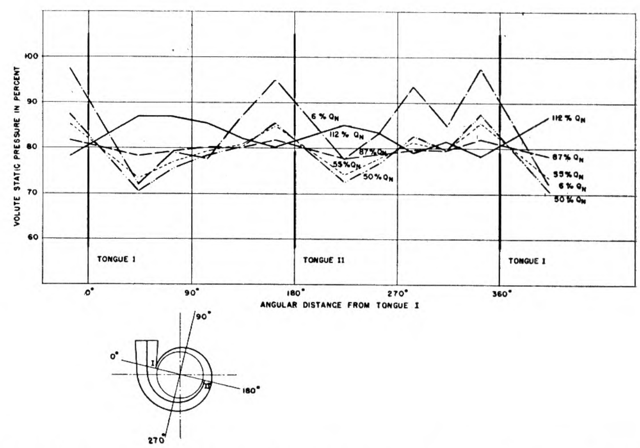

Fig. 8 Static-Pressure Distribution; Double-Volute Pump, Group B

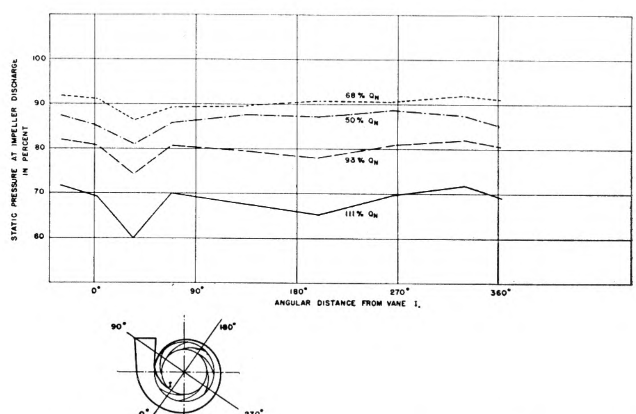

Fig. 9 Static-Pressure Distribution; Fixed-Vane-Diffusor Pump, Group B

Although this pressure variation must have a very marked effect upon the hydraulic performance of the unit, from an operating point of view, there is an even more direct result. A nonuniform pressure distribution such as, for example, the one shown in Fig. 7, for 114 per cent Q_n, indicates that there is a resultant radial thrust upon the impeller. This force must be taken care of in the mechanical design of bearings, case, and shaft, and may well be the controlling factor in the choice of shaft diameter and other important details. Failure to recognize this factor may result in mechanical contact of the wearing-ring surfaces and rapid deterioration of the equipment.

If the pressure-distribution diagrams for the double-volute pump, shown in Fig. 8, are integrated over the 360 deg, it will be found that the resultant radial force is small, since the effect of each of the two volutes nearly cancels the other. This is apparently true for all capacities and represents a distinct advantage of this type of construction. The resultant radial thrust upon the impeller of the diffusor pump, Fig. 9, is much lower than for the single-volute, but is somewhat higher than that of the double-volute. However, it should present no serious design problem, since it is not large.

It should be noted again that the radial unbalance of the fixed-vane-diffusor pump is due to the same cause that produced it in the single-volute pumps, i.e., the presence of the single volute itself. The main reason that the variation in pressure distribution and the resultant thrust are so much lower with the diffusor pump is that the flow is discharged into the volute at a much lower velocity than it is in the case of a single volute. If a fixed-vane case were designed, in which the vanes were used only as stay bolts and not as diffusors, high resultant radial forces should be expected. The importance of the investigation of these radial forces is illustrated by the fact that, for a good single-volute prototype, the unbalanced thrust is of the order of 50 tons. This would make illusory the feature of bearing-load elimination, commonly attributed to the vertical design.

Instability. Figs. 1 and 2 show that there are discontinuities in the head-capacity curves for all six cases. Such discontinuities appear to be characteristic of centrifugal-pump performance and are practically always found whenever tests of sufficient accuracy and detail are made. These discontinuities apparently are the result of a change in the flow from one regimen to another. For different design conditions, it seems that this change in flow can be localized either in the impeller or in the case. In addition, if the change is large enough in the impeller, it may also produce a significant change in the flow in the case. These flow discontinuities produce unstable ranges in the pump performance and, therefore, good practice indicates that the operating zone should not approach them too closely. For example, in the present study, one criterion tentatively proposed is that the maximum operating head should be at least 10 ft (3.5 per cent) below the break in the curve, as it is approached from the high-capacity side. This appears to be a quite satisfactory margin of safety for units having a reasonably small change in head at the discontinuity point, but may be somewhat inadequate for pumps having discontinuities as large as that shown by the fixed-vane diffusor of group A. For such pumps, it would seem advisable to restrict the maximum operating head to 1 or 2 per cent lower than the lowest value at the discontinuity region.

It is interesting to consider that significant information can be obtained by comparing the discontinuity regions, as shown by the capacity-head curves, with the torque or horsepower curves for the same conditions. If the flow regimen changes within the impeller passages, there will be a corresponding difference in the amount of angular momentum imparted to the fluid and this, in turn, will be apparent on the torque and horsepower curves. Thus, it may be concluded that, if a discontinuity in the head-capacity curve is reflected in the torque curve, the change in flow at least originates in the impeller. Conversely, if a discontinuity in the head-capacity curve is not accompanied by a similar break in the torque or horsepower curves, the change in the flow probably is localized in the casing. Unfortunately, space does not permit the plotting of the torque curves in Figs. 1 to 6, inclusive.

Cavitation Limits

Basic Limit of Eye Design. For each given design of an impeller eye, there is a relationship between capacity and inlet head which defines the beginning of cavitation. This basic limit, of course, assumes that, for all capacities, the flow has a normal velocity profile at the pump inlet; that the flow into the eye is circumferentially uniform; and that there are no tangential-velocity components present before the eye is entered. The difference between the basic characteristics of various eye designs for the same specific speeds will depend upon the abilities of the designers to keep their static pressures up and to eliminate local high-velocity regions in the vicinity of the passage entrances.

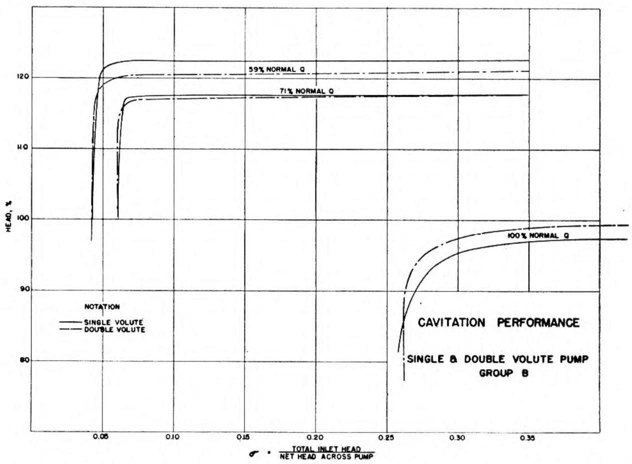

Fig. 10 Cavitation Performance; Single- and Double-Volute Pump, Group B

For a given impeller, however, this basic eye characteristic can be considered as the ideal limit for good performance. In actual operation, it can be modified either by the entrance conditions in the inlet piping approaching the pump, or by the reaction of the case on the inlet flow. The effect of the inlet piping is, of course, an installation problem, and will not be considered here, but the effect of the case is a question of basic pump design.

Effect of Case on Basic Limits. The effect of the case on the impeller characteristics has been discussed previously in the sections, "Comparison of Normal Operating Characteristics" and "Hydraulic Balance." It was seen that, in both high- and low-capacity regions, the case could produce a nonuniform pressure distribution around the impeller discharge. This must result in a pulsating flow in the impeller passages. Cavitation performance under these conditions must differ from that of steady flow. Previous studies at the laboratory (2) have shown that, under some conditions such as quite low capacity, the pressure unbalance on the impeller may be great enough to cause backflow from the case to the eye. Recent investigations also indicate that, in the same low-capacity region, the inlet tips of the impeller vanes may induce a radial-pressure difference sufficient to distort the flow further. It is difficult to separate these two phenomena, but together they seem to explain the "prerotation" which has been observed at times in pump inlets.

In an attempt to ascertain the effect of the various cases on

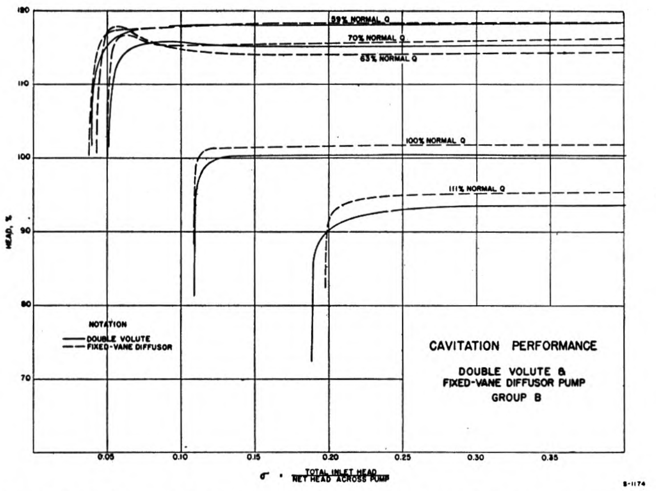

Fig. 11 Cavitation Performance; Double-Volute Case and Fixed-Vane-Diffusor Pump, Group B

the cavitation characteristics of the unit, Figs. 10 and 11 have been prepared. In both figures, the cavitation parameter σ has been plotted against the pump head for a series of constant capacities during which the inlet head was continuously lowered until cavitation was fully developed. Fig. 10 shows the comparative performance of the single- and double-volute cases of group B. It will be noted that the differences are slight, so slight in fact that little significance can be placed upon them. It is unfortunate that no runs are available at very low capacities, since this is the region in which the pressure distribution around the impeller differs widely for the two cases. Fig. 11 compares the double-volute case and the fixed-vane diffusor. These units are also from group B, but the results are not directly comparable to those of Fig. 10, because slightly different impellers were used in the two series of tests. Here, it will be noted that for one capacity the fixed-vane diffusor has a cavitation performance quite different from that shown by all other curves. The head rises rapidly, as σ decreases from 0.12 to 0.06. Since no such behavior is observed for either the single- or double-volute cases, it must be assumed that the fixed-vane-diffusor case is responsible for the difference.

The following logical explanation has been suggested by D. P. Barnes of the Bureau of Reclamation. The capacity at which this deviate behavior occurs is in the region for which the diffusor-vane angles must differ from the calculated discharge angle of the impeller. If cavitation starts in the impeller, it may quite possibly produce a change in the angle at which the flow leaves the impeller. If this angle more nearly coincides with that of the diffusor vanes, then the diffusion should be more effective and, therefore, the pump head should rise. Thus, it is possible that this rising head line on the σ diagram may be an indication of the beginning of cavitation, and hence marks a poorer rather than a better pump performance.

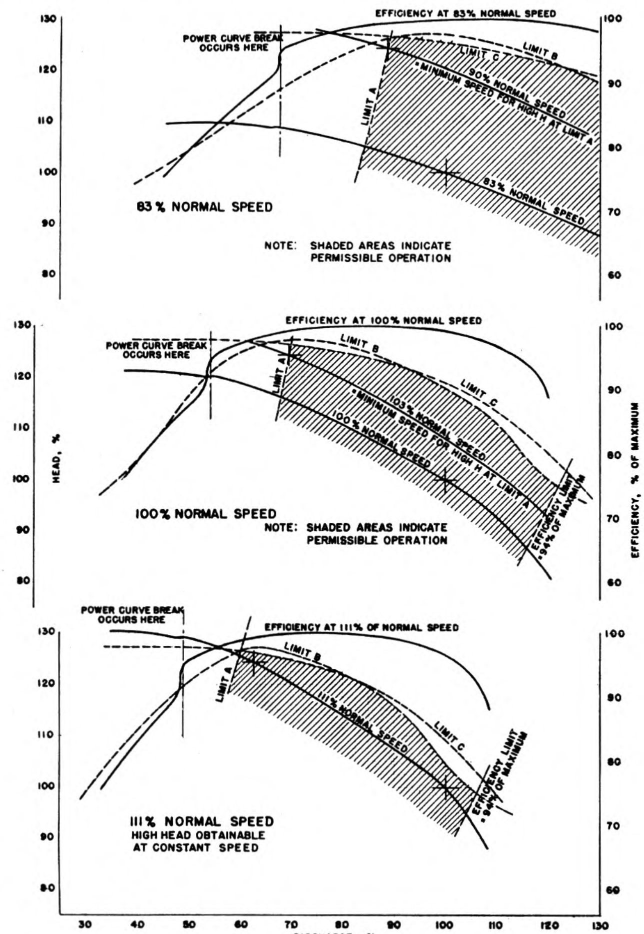

Fig. 12 Diagram Showing Limitations Upon Allowable Operating Regions of Characteristics of Double-Volute Pump, Group B

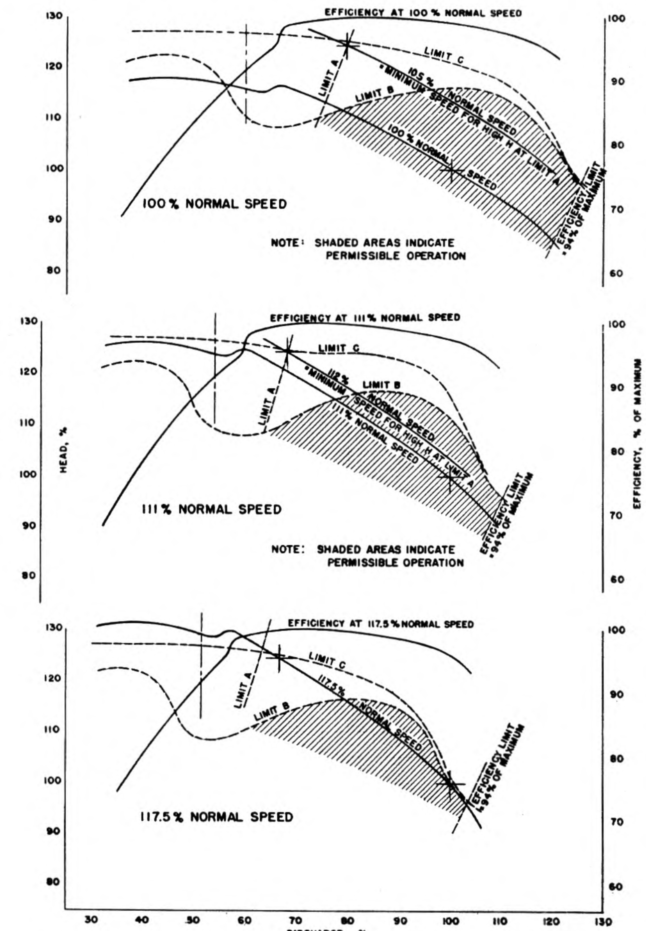

Fig. 13 Diagram Showing Limitations Upon Allowable Operating Regions of Characteristics of Fixed-Vane-Diffusor Pump, Group B

Selection of Operating Region. In an actual pump installation, the physical requirements impose many limitations upon the allowable operating regions of the pump characteristics. For example, the avoidance of discontinuity points has already been discussed. Freedom from cavitation is likewise necessary and this, in turn, is affected by the variation of inlet pressure due to change in reservoir level, etc. Fig. 12 presents a graphical diagram of these limitations as applied to the double-volute case of group B for three possible operating speeds. The limitations imposed are those obtained from a preliminary study of the Grand Coulee conditions. Limit A locates the permissible approach to the discontinuity or instability region. Limit B bounds the region for freedom from cavitation, as determined by the point at which there is a 0.5 per cent head drop on the σ curve (Fig. 11). Limit C bounds another cavitation parameter which is somewhat more complicated but which may be more satisfactory for certain units. The maximum and minimum operating heads are shown by the large crosses. The high-capacity boundary of the zone of permissible operation is arbitrarily defined by the condition that the efficiency has dropped to 94 per cent of the maximum value. The zone of permissible operation is indicated by the cross-hatched area and, within this zone, all of the criteria are met. This very useful type of presentation has been developed by D. P. Barnes.

It will be noted that, when the unit is operated at a speed of 83 per cent of the design value, only the low-head high-capacity portion of the required operating region can be covered. To obtain the high head required, an increase in speed to 90 per cent of the design value is necessary but, if this speed variation is permissible, the entire operating region can be covered satisfactorily. Conditions at the design speed are somewhat similar except that, to meet the high head condition, a speed increase to only something over 103 per cent is required. Operation at a speed of 111 per cent, however, permits the entire operating range to be obtained within the zone of permissible operation at constant speed.

Fig. 13 shows a similar diagram for the fixed-vane-diffusor pump of group B. Here, however, it is seen that, over a range of from 100 to 118 per cent of design speed, it is impossible to find any combination of constant- or variable-speed operation which will cover the desired range and yet meet the limitations imposed. It will be noted that in this unit the most serious deviation from limitations is from the "limit B" cavitation parameter.

Turbine Operation for Steady and Transient Conditions

One of the characteristic features of a pump installation is that transient conditions are quite commonly encountered under which the pump is called upon to operate as a turbine. Thus, for example, if the pump is operating normally and power should fail, unless there is a check valve in the line, the unit will slow down, reverse, and come up to runaway speed as a turbine, thus passing through the region of pump operation, a region of complete energy dissipation, and through the entire zone of turbine operation. In the design of large pump installations it is, therefore, very important for the plant designer to know the characteristics of the machines over the entire range of operating possibilities, in order that adequate provision may be made for maximum shaft torques, pressure surges, centrifugal forces, etc.

Complete Characteristic Diagrams. One of the first investigations of this complete range of pump operation was made by Kittredge and Thoma (3). It is convenient to present this information on a single diagram (4). Figs. 14 and 15 are two such diagrams for the single- and double-volute pumps, respectively, of group B. It will be noted that families of constant-head, constant-torque, and constant-efficiency lines are plotted against coordinates of capacity and speed. The performance of the unit at any constant speed is given by the intersection of these families of contours with a vertical line passing through the speed chosen.

Turbine Runaway Speed. The runaway speed of the unit, when operating as a turbine, is given by the intersection of the zero-torque line in the turbine region with the head curve corresponding to the pressure across the pump for that particular condition. For short pipe lines of ample proportions, this head is nearly the same as the pumping head since, under these conditions, the friction losses would be quite small. If the runaway speed exceeds the operating speed by a sufficient margin, it may be the controlling factor in the structural design of the impeller. Since the absolute value of this runaway speed is constant for a given unit operating under a given head, its value relative to the operating speed is determined by the choice of the latter. This can easily be seen by referring to Fig. 14. Consider that the normal operating head is represented by the 100 per cent head line. For the Grand Coulee installation, the maximum possible head which can cause turbine operation is about 120 per cent. The 120 per cent head line intersects the zero-torque line in the turbine zone at a negative speed of about 135 per cent. With a runaway speed of 35 per cent above that of normal operation, the impeller stresses may become quite serious. However, if it were decided that more suitable characteristics could be obtained by operating as a pump at 120 per cent of the design speed, then the runaway speed would exceed that of normal operation by about 12 per cent.

Turbine Operation for Possible Peak-Load Power Development. The Grand Coulee pumping plant of course is only a part of the total Grand Coulee project. A major function of the latter is power development. One of the problems always confronting a power project is the provision of sufficient capacity to meet peak-load demands. Therefore, the possibility has been suggested of using the pumping plant as a peak-load power supply by allowing the water to flow back from the upper reservoir, thus operating the pumps as turbines and the synchronous motors as generators. It will be noted in both Figs. 14 and 15 that these units have zones of very high efficiency in the turbine region, practically identical with the maximum efficiency obtained as pumps. Since the power must be supplied at constant frequency, it is necessary that the speed of turbine operation be the same as that of the pump. It is, of course, desirable to get as much power as possible from the turbines. However, the zone of turbine operation is determined by the selection of the pump operating speed.

For example, if in Fig. 15, the pump is considered to operate at 100 per cent speed, the torque and therefore the horsepower available in the turbine region will be 75 per cent of the corresponding values for the pump. For the high-head condition, i.e., for 120 per cent head, the turbine output will go up to about 110 per cent of the normal pump input at 100 per cent head. If, however, a normal operating speed of 111 per cent is selected for the pump, as was shown to be desirable in Fig. 12, conditions are quite different. Now, it will be observed that the normal torque input to the pump is 130 per cent for the low-head condition and about 120 per cent for the high-head condition, whereas, the corresponding turbine operation shows a torque of only about 35 per cent for the low-head condition and about 85 per cent for maximum-head. These values must be corrected to the new reference of 130 per cent, which was the input torque to the pump under normal head conditions. On this basis, the turbine output varies from 27 to 60 per cent of the power input to the pump at normal operating head. This output would appear to be so small as to be of doubtful value for a peak-load power supply. The trend, indicated by these examples, appears to be general, i.e., for a given design, if the operating point as a pump is located at a relatively low capacity, the operating speed will be low, the

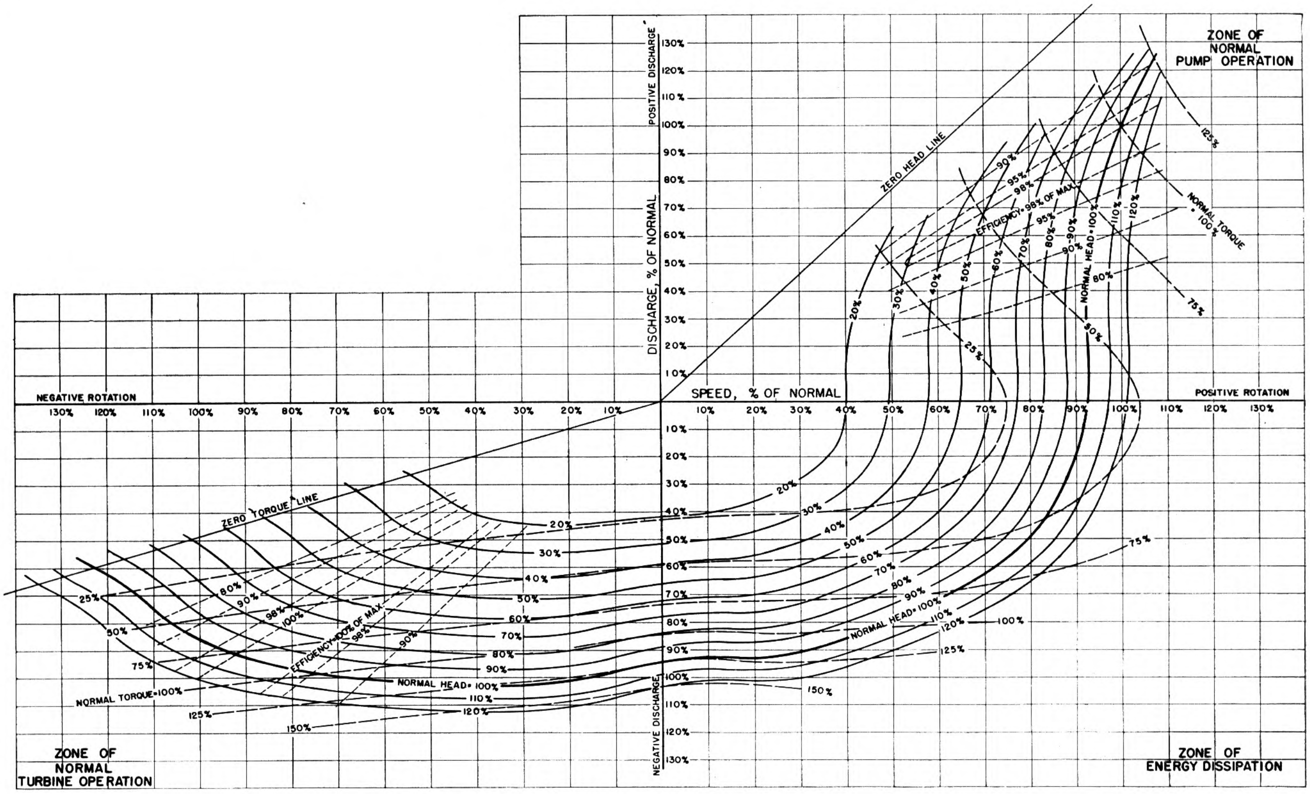

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FIG. 14 COMPLETE CHARACTERISTIC DIAGRAM, SINGLE-VOLUTE PUMP, GROUP B

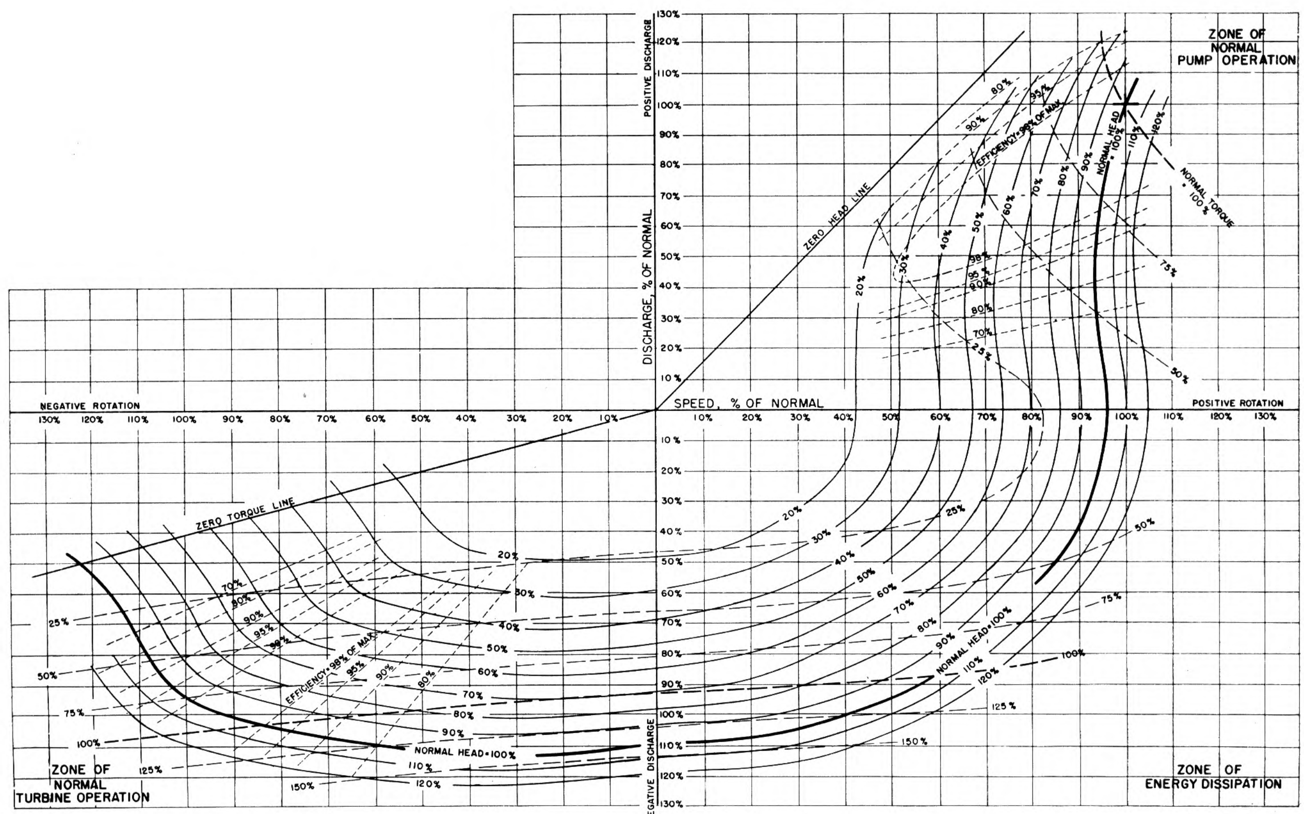

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FIG. 15 COMPLETE CHARACTERISTIC DIAGRAM, DOUBLE-VOLUTE PUMP, GROUP B turbine capacity will be high and the runaway speed will be high; whereas, if the operating point is chosen at a relatively high capacity and speed, the turbine capacity and the runaway speed will both be comparatively low. Thus, one more factor is added to the complicated set of requirements involved in the choice of the proper unit for the given installation.

Transient Behavior. The transient behavior of a pump is a function not only of the pump characteristics, but also of the pipe-line characteristics and other hydraulic and inertia features of the entire installation. The prediction of transient behavior has been briefly discussed in one of the previous references (4). Figs. 16 and 17 show typical transient characteristics for the double-volute pump of group B. These were computed by the

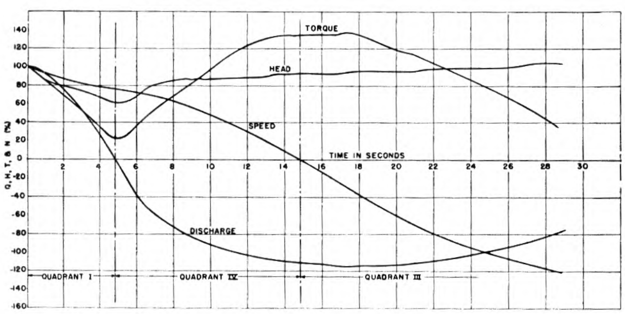

FIG. 16 TRANSIENT CHARACTERISTICS OF DOUBLE-VOLUTE PUMP, GROUP B

(Calculation for power failure when operating at 100 per cent speed and normal-head conditions.)

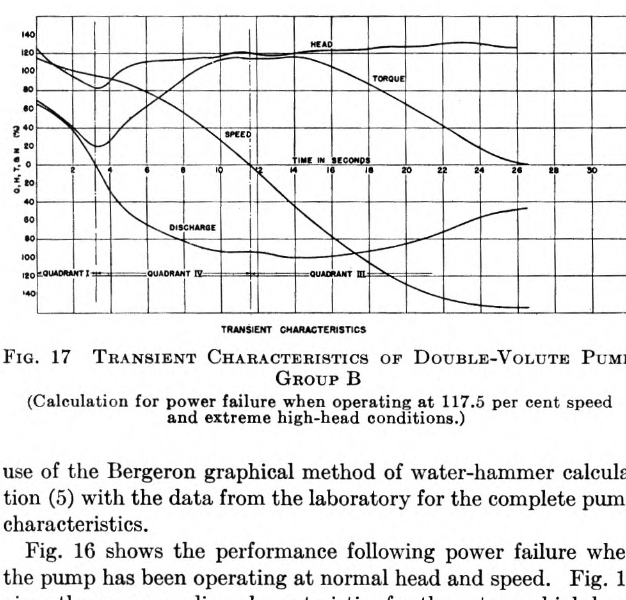

FIG. 17 TRANSIENT CHARACTERISTICS OF DOUBLE-VOLUTE PUMP, GROUP B

(Calculation for power failure when operating at 117.5 per cent speed and extreme high-head conditions.)

use of the Bergeron graphical method of water-hammer calculation (5) with the data from the laboratory for the complete pump characteristics.

Fig. 16 shows the performance following power failure when the pump has been operating at normal head and speed. Fig. 17 gives the corresponding characteristics for the extreme high-head condition, with the pump operating at 118 per cent of design speed at the time of power failure. For both conditions, it will be noted that the head fluctuations are quite moderate and present no problem. On the other hand, it is somewhat startling to imagine a 65,000-hp unit changing from a normal pump, operating at full speed in one direction, to a turbine operating at runaway speed in the other direction in an elapsed time of only 26 to 28 sec. The torque curves show that the maximum shaft stresses increase to 40 per cent above the normal operating value. The runaway speeds correspond closely to those already discussed.

Fig. 16 indicates that the unit operation remains in quadrant IV for as much as 10 sec. This is a region of complete energy dissipation, since energy is being poured into the machine through the deceleration of the rotating mass while, at the same time, energy is being given up in the machine by the fluid flowing through it. Little is known about cavitation conditions in this region, aside from the fact that they are apparently quite serious. It is felt that quadrant IV operation offers a fruitful field for further investigation.

SUMMARY OF RESULTS

Limitation of Program. Before summarizing the results, it should be re-emphasized that, although this investigation has shed some light on a few of the factors involved in the selection of the type and design of pump to meet particular needs of a given installation, the amount of information is still very meager. Many possibilities of casing design remain to be explored. Cavitation limits are yet too empirical in character, and the possibilities of obtaining more desirable performance for a given installation through changes in the impeller design are barely touched.

Operating Characteristics and Speed. The over-all performance of a pump, using a given impeller, is greatly affected by the case design. For a given type of case, the characteristics may be varied considerably by the choice of the point at which the case "fits" the impeller. Of the three types of cases studied, the double-volute type appears to give the widest high-efficiency range.

A well-designed impeller has a fairly wide range of speeds over which it will operate satisfactorily when delivering against a given head. A proper choice of case "fit" therefore will result in a unit having the desired operating speed. For a given combination of impeller and case, the head-capacity characteristics can be "steepened" by choosing the operating point at a relatively high capacity and speed. If a head-capacity steepness greater than that of the basic impeller performance is desired, it can be obtained only by the sacrifice of efficiency, i.e., by pushing the operating point to a capacity out beyond the zone of maximum efficiency. This is equivalent to using a casing too small for the desired capacity.

Hydraulic Balance and Radial Thrust. Within the zone of maximum efficiency, the fit of the case to the impeller is usually satisfactory enough to produce a relatively uniform pressure distribution around the impeller discharger. Therefore, operation in this zone is accompanied by little or no radial thrust. Operation at higher or lower capacities distorts this uniformity and results in radial thrust. The resultant force on the impeller and shaft is highest for the single-volute case. The fixed-vane-diffusor construction greatly reduces the magnitude of the force and it is eliminated by a well-designed double-volute casing.

Instability. Discontinuities in the head-capacity characteristic seem to be an inherent feature of centrifugal pumps, or at least of high-efficiency ones. These discontinuities probably are due to changes in the flow regimen, either in the impeller or case. They often limit the extent of the satisfactory operating range. The closeness with which they may be approached is presumably a function of the magnitude of the discontinuity.

Cavitation. Cavitation is an impeller phenomenon and is relatively insensitive to casing design. However, severe unbalance of the pressure distribution around the impeller discharge may change the cavitation conditions. Cavitation usually produces a change in the head-capacity characteristic. In general, the head is lowered, but under some circumstances it seems that it may be first increased. Cavitation forms one of the major limitations in determining the zone of satisfactory operation. If, in order to obtain other desirable characteristics, the operation point for a given impeller is chosen some distance away from the design point, it may be necessary to modify the eye design to secure satisfactory cavitation elimination. As yet, no satisfactory quantitative determination of the inception or degree of cavitation has been developed.

Turbine Operation. In general, a centrifugal pump can be operated very satisfactorily as a turbine and, over a limited range, with an efficiency equal to the best performance as a pump. In special cases, it may be feasible to utilize this possibility to supply a peak-load power demand by reversing the flow and operating the pump as a turbine and the motor as a generator. If this is to be done, careful consideration must be given to the design of the unit, since the selection of the pump operating point determines the turbine performance as well. The conditions for securing the optimum pump characteristics, turbine operation, and low runaway speed are usually not compatible, and therefore the relative value or the different elements of the performance must be evaluated carefully.

Conclusion

Although this study was designed to answer specific questions covering the selection of operating features for the pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is anticipated that, in the future, there will be more and more demand for hydraulic units, both pump and turbine, the characteristics of which are particularly adapted to the installation requirements, and it is felt that studies of the kind herein reported offer the most reliable means of securing the desired result.

Acknowledgments

During the period of the investigations, the Bureau of Reclamation, through the chief engineer and members of his technical staff, has kept in intimate contact with the work and has contributed much to its progress. Especial acknowledgment is due to Mr. D. P. Barnes, resident representative, of the Bureau, who has taken an active part in the experimental investigations and their analyses.

The program has been carried out under the immediate direction of Prof. Th. von Kármán, Prof. R. L. Daugherty, and the author. The technical staff has been in charge of Mr. J. W. Daily. The results reported are the joint product of the entire staff and should be so considered.

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Development of a Major Principle in Pulverized-Coal Firing

By FRED L. DORNBROOK,[1] MILWAUKEE, WIS.

Positive avoidance of the plastic-ash phase at all surfaces bounding the furnace has required emphasis upon heat-absorption performance of furnace surfaces. Evolution of the present-day furnace, which is practically self-cleaning, reliable, of high efficiency, and can operate continuously month after month under variable conditions of coal, load, and attendance, is treated in the paper from its inception in Milwaukee during 1918 to the present new installations of the author's company.

THE following 20-year-old conclusions, concerning the "slagging" problem and furnace heat absorption, constitute a basic principle in pulverized-coal firing, and also furnish the theme for this paper:

"If the refuse is to be removed easily, the bottom of the furnace must be kept below the temperature at which ash becomes sticky."

"A better way of keeping the furnace temperature . . . . slightly below that of the running slag . . . . is to expose a large amount of boiler heating surface to radiation from the furnace."

These statements were made in a bulletin[2] which covered an investigation of powdered coal as fuel at the Oneida Street Station in Milwaukee (now called East Wells Street Station), where experimental work had been carried on for some months.

A major problem in burning pulverized coal concerns the incombustible portion of the coal, namely, the ash. If the ash could be removed from the coal before firing, an enormous simplification of the combustion process would occur. Almost every element in the boiler plant would be affected favorably. Thus a minor constituent of coal requires the major attention of the designer and operator.

The Furnace Temperature Scale

Fig. 2 was prepared to show graphically the general conclusion that the ash problem dictates that over 50 per cent of the total heat absorption must occur in many furnaces of modern design if ash-slagging problems are to be avoided.

In an all-brick furnace, having no heat-absorbing surface, a theoretical temperature of 3700 F would be reached. This assumes 16 per cent CO_2 (15 per cent excess air) and 600 F preheated air. This temperature is 1400 F above the 2300 F ash-softening temperature of average Pennsylvania coal.

The flue gas or products of combustion must be cooled from 3700 F to 800 F by the water and steam surfaces before entering the air heater. This is a total of 2900 F of cooling and, since the specific heat of the gases varies only slightly over the temperature range, the furnace must absorb $\frac{1400\text{ F}}{2900\text{ F}}$ or 48 per cent of the total net absorption to lower the furnace exit temperature below the ash-softening temperature. Actually over 50 per cent should be

Fig. 1 Historic Pulverized-Fuel Boilers

(Tests of boiler No. 5, in foreground, were subject of Bureau of Mines Bulletin.[2])

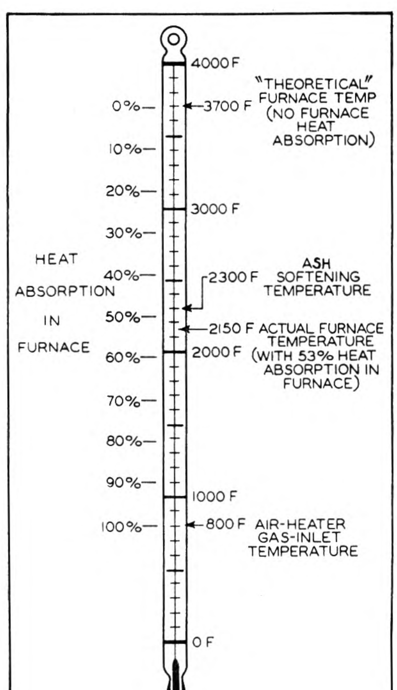

Fig. 2 Furnace-Temperature Scale

(Showing that over one half of heat absorption of modern boiler units must occur in furnace if ash-softening temperatures are not to be exceeded at boiler entrance.)

[1] Chief Engineer of Power Plants, Wisconsin Electric Power Company. Mem. A.S.M.E.

[2] "An Investigation of Powdered Coal as Fuel for Power-Plant Boilers," by H. Kreisinger, John Blizard, C. E. Augustine, and B. J. Cross, U. S. Bureau of Mines, Bulletin No. 223, 1923.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

TABLE 1 TRENDS OF FURNACE HEAT ABSORPTION IN MILWAUKEE BOILER UNITS

(Showing reduction of furnace exit temperatures, in spite of larger sizes, by greater furnace heat absorption)

Item		Oneida St.		Lakeside					
		Without water screen	With water screen	Boiler room No. 1	Boiler room No. 2	Boiler room No. 3, boiler No. 19	Port Washington, boiler No. 1	East Wells St., boiler No. 16	Commerce St., boiler No. 25
1	First operated, year	1918	1920	1920	1923	1928	1935	1938	1941 (scheduled)
2	Steam output (design max) lb per hr	18000	23000	80000	131200	300000	690000	225000	375000
3	Heat input, million Btu per hr	24.1	31.2	110.0	183.5	430	890	326	470
4	CO_2 at boiler outlet, per cent	14	15	13	15.5	14.5	14.25	14.8	15
5	Efficiency, over-all, per cent	79.7	78.8	87.5	88.8	85.7	86.0	85.3	86.1
	Heat-absorbing surface, sq ft:								
6	Boiler, convection surface	4680	4680	13057	17650	23640	44087	13500	24000
7	Furnace, radiant surface[a]	101	171.0	618	1324	5338	9748	4609	6391
8	Economizer, convection surface	0	0	7603	14256	0	0	0	0
9	Air heater, convection surface	0	0	0	0	23640	121000	52300	66500
10	Furnace volume, cu ft	1608	1463	6450	11650	29780	61300	19951	30500
11	Btu released per hr per sq ft of furnace surface[a]	238000	182000	178000	138000	80500	91500	70500	73500
12	Btu released per hr per cu ft of furnace volume	15000	21300	17000	15700	17800	14500	16300	15400
13	Average ash-fushion temperature, F	2300	2300	2150	2150	2150	2250	2250	...
14	Average furnace temperature (calculated maximum output), F	2350	2350	2100	2100	2000	2100	1950	2000

[a] Includes projected surface of boiler tubes and superheater surfaces corrected to equivalent water surfaces.

provided to have some margin to cope with operating conditions and contingencies.

This example indicates that low excess air and high temperature of air preheating require greater heat absorption in the furnace, if ash problems are to be avoided. Air preheaters are being used in many plants in place of economizers since higher steam pressures are causing more extensive use of extracted steam to heat feedwater. The heat absorbed in the air heater will raise the furnace temperature unless the designer has provided for increasing heat absorption in the furnace.

Larger steaming capacities of boilers naturally have placed emphasis on the ash problem because the furnace heat-absorption area does not increase as rapidly as does the furnace volume when boiler outputs are increased. Doubling of furnace dimensions normally increases furnace volume 8 times but available water- or steam-wall area is increased only 4 times. Thus larger boiler units tend to have hotter furnaces unless the volumetric rate of combustion is decreased proportionately.

Milwaukee Experiences

Table 1 summarizes Milwaukee experience in this regard, starting with the small Oneida Street units, including the large Port Washington boiler unit, and ending with the Commerce Street installation now under construction. It shows:

1 The large increase in size of boiler units. Port Washington output is 30 times that of Oneida Street units. (See item 2 of Table 1.)

2 Retention of low furnace temperatures to a point below ash-softening temperature. (See item 14.)

3 Decrease of excess air, increase of air preheat, more furnace cooling, and uniformity of Btu per cu ft per hr heat release.

4 Decrease of Btu release per hr per sq ft of furnace heat-absorbing surface to about $^1/_3$ of early practice.

Comparison of items 13 and 14, Table 1, will show that, at maximum rated output, the furnace exit temperatures of the Lakeside units are about 50 F below ash-fusion temperatures. Since these units serve topping turbines and normally operate at 83 per cent of maximum rated output, it can be said that extensive experience with these four large units urges that furnace exit temperatures be limited to 200 F below ash-softening temperature, for normal operation.

Ever since Lakeside fuel changed in 1934 from Pennsylvania and West Virginia coal to southern Illinois coal, experience with this matter of maintaining a margin against slagging of boiler inlet tubes has been accumulated. Excess air required slight increases, cleaning methods were studied and, in general, operation had to be more carefully conducted in order to sustain load-carrying reliability. The more recent installations in subsequent stations have been provided with a greater margin before troublesome slagging can occur.

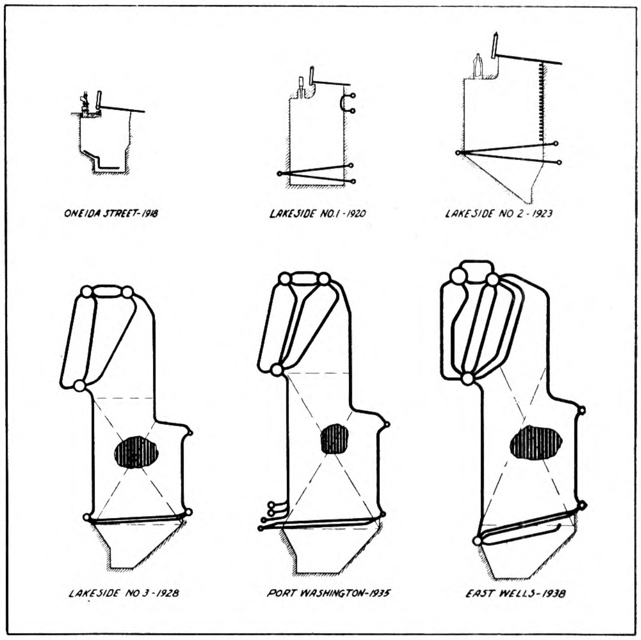

Fig. 3 Evolution of Furnace Cooling of Milwaukee Boiler Units

(Showing relative size, extent of brickwork, and "cold" surfaces [in black], and illustrating definite trend toward greater furnace heat absorption.)

The "furnace exit temperature" at Lakeside and in subsequent stations is actually about 200 F above the boiler inlet temperatures and is measured by high-velocity thermocouples. Calculations are made for this furnace exit temperature at the location where the gases enter the triangular portion above the arch level Fig. 3, in order to correlate the calculated heat-transfer rates with an extensive experience of radiant superheater transfer rates. When it is borne in mind that gases are reduced about 200 F in this triangular section, one realizes from Lakeside experience that boiler-inlet-gas temperatures 200 to 300 F below ash-softening temperatures afford scant margin against first-pass clogging.

History of Milwaukee Installations

The original experiments at Oneida Street were conducted on a 4680-sq ft water-tube boiler in 1918. The management of The

Milwaukee Electric Railway and Light Company (now Wisconsin Electric Power Company) encouraged the application of pulverized coal to this boiler. They considered it a good business risk to place the boiler at the disposal of the operating crew for experimental purposes feeling that, even under these conditions, the boiler would be available when needed. Almost endless experiments and tests were made before a furnace of the proper design was developed; in fact, one furnace was rebuilt five times before obtaining a satisfactory shape and volume.

The water screen played an important role in arriving at an acceptable furnace performance in this original pulverized-coal unit. To quote the Bureau of Mines Bulletin:[2] "The first 16 tests showed the impossibility of running the furnace with low excess air without the ash fusing at the bottom of the furnace. Consequently experiments were made with a water coil or screen to cool the bottom of the furnace. This coil, by partly screening the bottom from the radiation of the flame and by absorbing radiation from the bottom, kept the temperature of the bottom below that of the fusion point of the ash. Screen No. 5 in Fig. 13 was most effective in preventing troublesome slag at high rates of combustion and with low excess air."

Lakeside Station

At Lakeside, where the second application of dry-ash furnaces was made, the slag problem was largely overcome by the installation of so-called water screens in all furnaces. These screens were connected into the circulating system of the boiler by means of vertical risers. It was in this installation, too, that a radical departure from the convection type of superheater was attempted through the introduction of the then new radiant-heat-absorbing superheater. In 1923, John Anderson, in a technical report[3] on the subject stated: "In view of the keen interest manifested toward the furnace problem, the advent of this type of superheater on a commercial scale happens at an opportune time. It seems to be the consensus of opinion of many of the foremost power-plant engineers that the ultimate solution of the present furnace problems will be in steam- or water-cooled furnace walls. The installation of the water screen and the radiant-heat superheater seems to confirm the existence of a tendency toward steam- and water-cooled furnace walls."

It is natural that the developments in the first two of Lakeside's boiler rooms should become part of the design for its third boiler room. It is here that Lakeside's four 1300-lb boilers are located. Dry-ash furnaces with radiant superheating surfaces were again the order. Each of the four high-pressure boilers serves a topping turbine of 7700-kw capacity, all of which exhaust at constant pressure to the plant's 300-lb header after the steam passes through reheaters, located within the individual boilers.

The temperature of steam exhausting from the topping turbine increases with decreasing load, thereby requiring a reheater of convection characteristics to obtain a uniform outlet temperature over the operating load range. As a result, radiant reheating is not required. Radiant superheating surfaces, however, are used in the side walls of these furnaces with waterwall surfaces in front and rear.

Port Washington Station

In the Port Washington boiler, which is the largest on the Milwaukee system, the opportunity existed for using a radiant

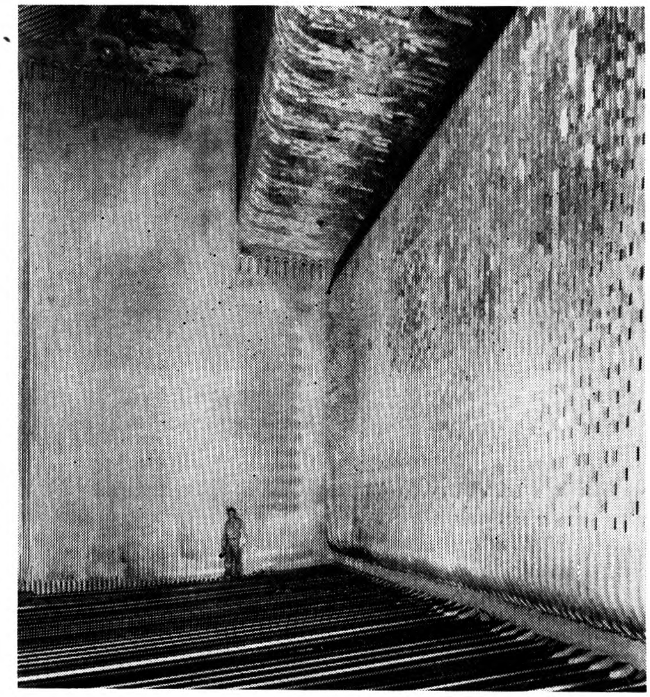

Fig. 4 Cleanliness in Spite of Large Output
(With an output of 690,000 lb per hr, the Port Washington unit does not cause slagging troubles because of adequate heat-absorbing area. The condition of the furnace without cleaning, after a 6-months' continuous run at normal output of 465,000 lb per hr, is shown herewith.)

reheater surface advantageously. This resulted from a combination of uniform throttle steam temperature over a wide load range and compound-turbine operation. The upward steam-temperature trend of the radiant surface at lower loads fits well with the lowered outlet steam temperature from the high-pressure section of the turbine, thereby permitting the maintenance of a uniform reheater outlet temperature over the same wide load range. Incidentally, this inherent control of reheated-steam temperature has aided materially in maintaining high thermal efficiency in the plant month after month. The placing of the steam-cooled surface in the furnace wall has not only aided in retaining the desired low furnace temperature but has also resulted in a gain to the plant in thermal efficiency.

As steam pressures increase, reheating becomes more necessary to the economical operation of the steam cycle. This influence, together with the trend toward higher steam temperatures directs attention toward radiant steam surfaces, because they become increasingly more important as superheating and reheating mediums. The low-temperature furnace with its greater wall areas permits installation of radiant superheating and reheating surfaces and thus points the way toward further progress in the art of steam-power generation.

Fuel Versatility and Reliability

The ability to change from one coal to another to permit economies in the purchase of coal when price differentials change has long been a goal sought by boiler-plant owners. A versatile furnace which is not affected by a change in the constituents of coal, such as occurs when changing from a so-called high-grade coal to a low-grade coal, is of inestimable value. At Lakeside, for instance, the coal situation changed decidedly and rather suddenly during the depression. Eastern coal had been the most economical, but when midwestern operators began producing coal in large quantities and offering it at attractive prices, the economic picture changed. Tests in Lakeside's furnaces indicated

[3] "The Use of Pulverized Coal Under Central Station Boilers," by John Anderson. Paper presented before the Technical League of the Employes Mutual Benefit Association, The Milwaukee Electric Railway & Light Company, Milwaukee, Wis., February, 1920. Because of its historic value in the art of combustion, this paper has been reprinted by the Combustion Engineering Company, Inc., New York, N. Y.

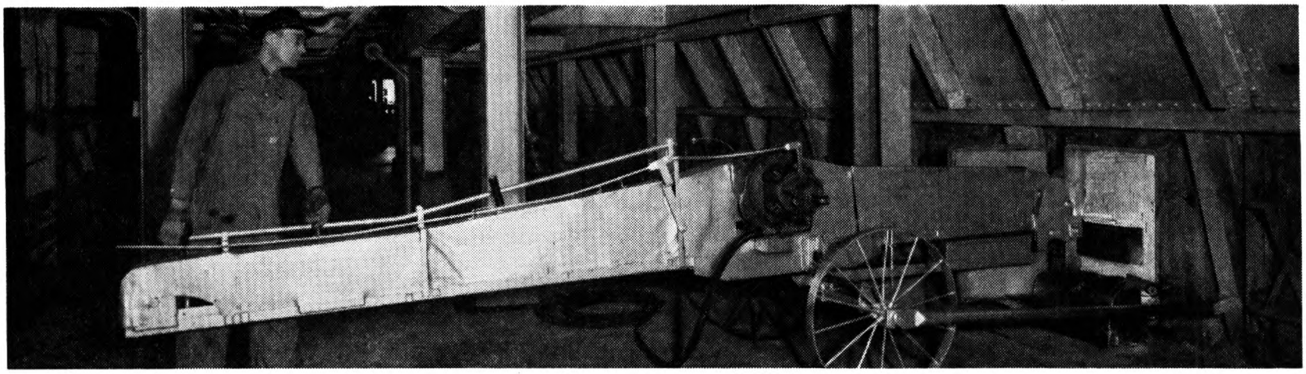

Fig. 5 Reliable Ash Removal

(A method that does not require any particular boiler-loading schedule nor reduce boiler-unit availability is shown. See Fig. 6 for outline of this mechanical hoe and furnace hopper bottom.)

that the new coal could be burned without difficulty and without undue added costs. On the basis of the tests, the change was made in 1934. Since that time, Lakeside has made appreciable annual savings, even after allowing for increased costs incurred by the poorer quality of the coal (ash disposal, etc.). This is tangible evidence of the value of a versatile furnace.

Reliability is an important factor in producing low over-all costs and its value cannot be overemphasized. The four high-pressure boilers at Lakeside have an availability record of 94.2 per cent over a 9-year period in spite of an average of 15 stops and starts per month. The Port Washington boiler has operated at an availability of 93.9 per cent over a 4.1-year period, Table 2.

TABLE 2 AVAILABILITIES OF FIVE 1300-LB BOILER UNITS FOR SEVERAL YEARS

(A total of 40 boiler years' experience at an average availability of 94 per cent attests to the point that moderate furnace temperatures are conducive to high reliability)

Year	Lakeside, average of 4 boilers, per cent	Port Washington, 1 boiler, per cent
1931	92.0	...
1932	93.8	...
1933	97.6	...
1934	94.0	...
1935	95.3	100[a]
1936	93.0	91.1
1937	95.0	93.0
1938	93.0	97.5
1939	93.9	93.4
Average (weighted)	94.2	93.9

[a] Started November 22, 1935.

These reliability records have been obtained on boilers having moderate furnace temperatures.

Efficiency Aspects

Consistently holding to optimum efficiency conditions for the many different operating situations experienced in typical plant routine is possible with the type of furnace under discussion. Change of coal, unusually high or low load, disability of a portion of auxiliaries which affect the furnace, or trouble with regulating superheat temperature, and several other situations that can cause expenditure of extra fuel with less versatile furnaces, affect the dry-ash furnace comparatively little. With favorable burners and proper air admission, combustion at low ratings is stable and efficient.

While it may be possible to operate a hotter furnace during many of the conditions mentioned without sacrifice of economy, provided special attention is given, it is usually necessary to compromise for the sake of reliability. When a boiler unit is needed definitely for daily peak loads, its operators will not take a chance of rendering it unavailable for the peak. A positive margin against clogging the first pass of boiler tubes, for instance,

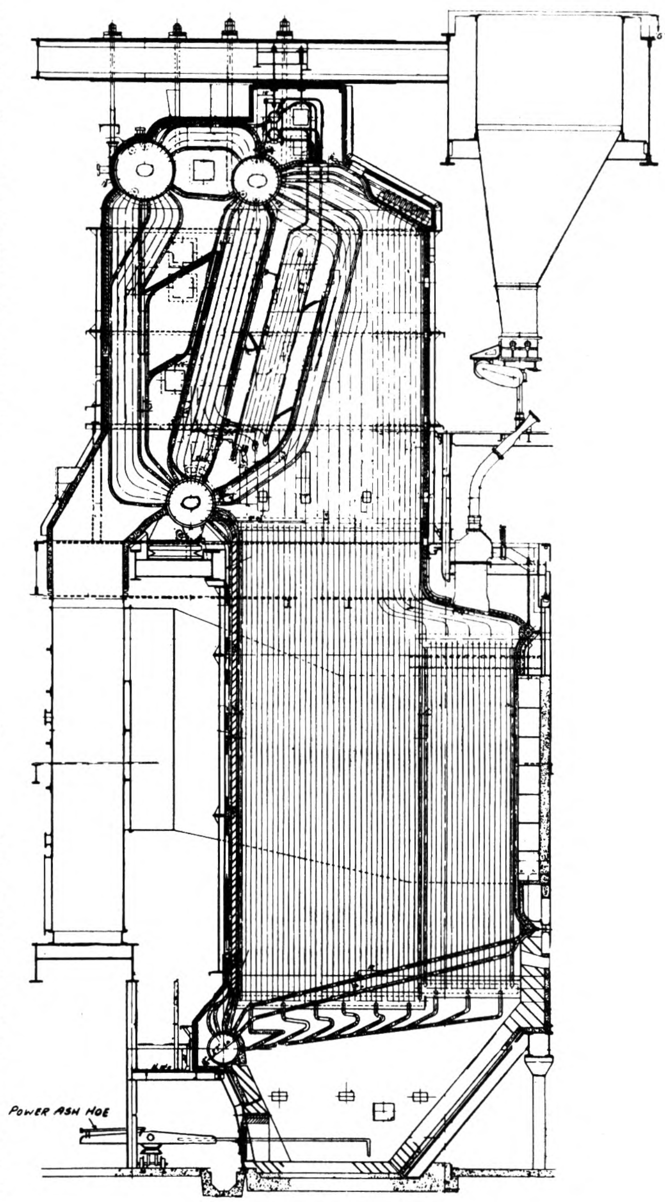

Fig. 6 A Recent Design

(This East Wells Street, 650-psi, 225,000-lb per hr boiler unit employs more liberal furnace cooling than any of its predecessors.)

must be provided continuously. Many boiler troubles are cumulative and, once they start, are difficult to arrest unless load can be reduced.

It is not what a boiler unit can do under special test, but what its operators actually get from it over a period of several years that is the criterion of boiler-unit efficiency. From available data, there appears substantiation of the premise that dry-ash furnaces are conducive to high sustained efficiency.

Long-Time Maintenance Aspects

It appears probable that furnace-maintenance costs increase in proportion to furnace temperatures, even when brickwork deterioration is disregarded. It has been found by experience that long-time service in furnace duty will cause maintenance which would not be anticipated even upon careful inspection after a year of operation.

When steel absorbs heat at the rate of 50,000 Btu per sq ft per hr, there is a 200-F temperature difference per in. of thickness perpendicular to the plane of absorption. This causes stresses of considerable magnitude. When this occurs, at least in the case of integral extended surfaces, minute cracks may start in a year or two, and then tenaciously travel without limit, even into the inside of the tube itself. Steel at high rates of heat transfer behaves like masonry at low temperatures; short-time service is no guarantee against long-time freedom from destructive cracking.

If several furnace tubes fail on different occasions from the same trouble, then all similarly affected tubes, even those deteriorated to a much lesser degree, must be replaced. Less drastic maintenance was possible in the days of many small boiler units serving one turbine, but present-day practice requires that boiler units be kept at peak reliability at all times. Cases are known where forced outage of one boiler caused prompt outage of other boilers operating in parallel because the assumption of the high load caused similar trouble. Operators, who have experienced the distress of troubles that become cumulative, thereafter try hard to avoid their cause.

Dry-Bottom Versus Dry-Ash Furnaces

There is a difference between a dry-bottom furnace and a dry-ash furnace. Dry-bottom furnaces have experienced serious falling of enormous accumulations of dense slag from upper parts of the furnace onto screen tubes or floor tubes. The dry-ash furnace permits of no destructive nor outage-provoking accumulations at any point, not even on brickwork. Attachment of ash is so fragile that the light honeycomb deposits fall when reaching a few pounds weight.

Superheat-Temperature Control

Variations in ash deposits will present a problem of steam-temperature control in boiler units of the future, where heat absorbed by superheating surfaces may exceed 50 per cent of the total heat absorbed by the entire unit. The dry-ash furnace is not entirely free from ash deposit but the variations in ash deposit will not cause a severe problem in control of superheat temperature.

Economics of Furnaces

Wet-ash furnaces are chosen frequently because of minimum installation costs. Space limitations often require the use of small furnaces; in fact, many have been installed between the same building columns used for older units of less output.

Boiler-unit reliability has a decided bearing upon total investment costs and operating costs. Savings due to furnace size are lost if the design lacks reliability and requires more spare boiler units. It is felt that too much emphasis cannot be placed upon reliability.

Conclusions

This paper shows how the present-day furnace in Milwaukee has been evolved from the original pulverized-fuel furnace in 1918, and traces the development of a major principle in pulverized-fuel firing.

In order to prevent ash from sticking to the inside surfaces of any furnace, these surfaces must be designed to cool properly the ash below its plastic state.

Experiences and data are presented indicating that a practical solution of the so-called "slagging problem" consists of designing boiler furnaces for such appreciable heat absorption that gases are approximately 200 F below ash-softening temperatures when entering the first rows of boiler tubes.

Discussion

J. M. Drabelle.[4] The Milwaukee Electric Railway and Light Company and the late John Anderson are two names which will always be inseparably linked together in the commercial development and use of pulverized-coal firing in large power boilers in this country.

The fundamental principles established by that company and by Mr. Anderson are as applicable today as when they were established. Making use of such principles, the topping unit at the Cedar Rapids Power Station of the Iowa Electric Light and Power Company is of interest.

The boiler has a nominal maximum rating of 300,000 lb of steam per hr at a temperature of 750 F and a pressure of 750 psi gage. The coal fired is from the Illinois strip-mine fields and has a heating value of 10,100 Btu, moisture content 18.3 per cent, an ash content of 10 per cent, and a sulphur content of 2.9 per cent, the ash having a fusion point of approximately 1900 to 2000 F.

The boiler unit and waterwalls were furnished by the Springfield Boiler Company, Springfield, Ill. The furnace is unique in that it is divided into two sections separated by a vertical waterwall in the center. This wall is of the open type and is made up of two banks of tubes, one bank each per furnace section. The total water surface of the sidewalls of the furnace based on projected area is 4315 sq ft. The total water surface facing the fire, including the first row of boiler tubes and the V-bottom ashpit tubes, is 4810 sq ft. The total cubic volume of the two furnaces is 15,385 cu ft. Including the V-bottom ashpit section, the total is 16,125 cu ft.

The principal dimensions of each section of the furnace are width 12 ft 8 in.; depth 19 ft 6 in.; height, from upper section of V-bottom ash section to first row of steam-generating tubes of the boiler, 31 ft 9 in.

The performance of this furnace with pulverized coal has thoroughly proved the accuracy of the author's conclusions, i.e., heat-absorbing surface is so arranged as to avoid hot-gas-flow sections entering the tube bank of the boiler with consequent fouling, bridging, and other difficulties typical of some furnaces.

Each section of the furnace at the nominal rating of the boiler is fired by two pulverized-coal burners handling 9.2 tons of coal per hr per furnace section. There have been no troublesome deposits of any kind; such deposits as have appeared on the sidewall tube bank are of the light, fragile, honeycomb type as described by the author. There has been absolutely no objectionable smoke or other troubles due to this comparatively cold furnace.

J. B. Johnson.[5] From the viewpoint of the inspecting engi-

[4] Consulting Engineer, Iowa Electric Light and Power Company, Cedar Rapids, Ia.

[5] Engineer, The Travelers, Milwaukee, Wis.

neer, it can be mentioned that there is another factor affecting the economy of operation of boiler units of the type described by the author. Observation over a period of years has developed the conclusion that a distinct reduction in time of outage, either scheduled or forced, is obtained by dry-ash-furnace operation. The limiting factor for cooling speed under fan control has been found to consist almost entirely of an hourly temperature drop that is not so great as to be unfavorable to the tube joints in the drum.

Access to the interior of the furnace and boiler passes being appreciably hastened by this factor, it is evident that the availability percentage is less affected by the outages for inspection and maintenance, whether scheduled or otherwise.

There is still another item on the favorable side from the standpoint of the inspecting engineer. At the most, there is a light "whisker ash" which is easily removed, and the bare surfaces are exposed for easy and rapid examination. This again means a saving in outage time and, not only can the inspections be made more rapidly, but their quality is enhanced.

In these days when the color of the metal surfaces has a distinctive message for the inspection engineer, a rapid method of securing a clean surface is indeed appreciated. In the case of the dry-ash furnace, all that is required to secure a truly bare surface is a light water wash, atomized with compressed air.

Summing up, it can be stated that this type of furnace has inherent advantages favoring the availability percentage, due to a decrease in time for cooling the furnace, resulting in quicker attainment of temperatures at which inspection and maintenance operations may be conducted with less time and labor being required for making ready for these operations.

Author's Closure

Use of a dividing wall in the furnace reported by Mr. Drabelle, in order to cool it sufficiently for trouble-free operation with unusually poor coal, is certainly of interest and of significance.

Mr. Johnson's appropriate remarks suggest that better inspection possible with the dry-ash furnace assists in realization of higher reliability.

Steam Locomotives—Notes on Ages and Proportions, With Suggestions for Improvements

By J. L. Ryan,[1] Springfield, Mo.

Locomotives are often built and maintained in kind for their service life, renewal parts being made according to their original design, whereas at but little if any additional cost, they might be renewed to modern design and proportions. In this paper the author discusses the high percentage of the total number of steam locomotives having road assignment which do not have modern proportions and which will be continued in service for many years. This situation leads to the suggestion that increased capacity and economy may be built into them at slight additional expense by following the practice of making required maintenance renewals according to modern proportions, with particular emphasis on adequate steam space, increased gas area through the boiler, a high degree of superheat, and valve events for speed and capacity.

The demand for faster service, longer runs, and high mileage on the railroads has left almost all of them with many locomotives on their hands which are not adapted to meet such requirements. In other words, the horsepower demand cannot be met. Many of these locomotives may be improved for faster and more sustained service by making changes which will not incur a great deal of expense.

From month to month one may see, in publications concerned with railway transportation, articles giving the proportions and design features of locomotives that are being delivered to some railway. If the reader is not something of a student of the motive-power field, he may come to the conclusion that the railways are being well stocked with new locomotives. This, however, is far from being true. In fact, time passes so rapidly, making obsolete locomotives which we are inclined to consider as modern, that those of us concerned with motive-power problems may well be startled by the actual conditions when making compilations of the locomotives handling our transportation services, their ages, proportions, and construction.

Degree of Obsolescence of Nation's Motive Power

In attempting to approximate the extent to which our road-service steam motive power may be considered modern, the author uses as an example the locomotives of his employer, which is considered an average-size railway. Out of an ownership of 610 locomotives, 425 or 70 per cent are assigned to road service. In view of the speeding up of freight and passenger schedules, the horsepower rating of locomotives is a better yardstick to apply than the rated tractive effort which is so frequently used. Thus, using Cole's values for cylinder-horsepower rating for locomotives built prior to 1920, and the railway company's test results for those built in 1920 and later, the 425 locomotives having road assignment have a rating of 1,096,100 hp, an average of 2579 hp each. Table 1 indicates the periods in which certain of these locomotives have been built.

TABLE 1 PERIOD OF BUILDING LOCOMOTIVES OF THE ST. LOUIS-SAN FRANCISCO RAILWAY

	Horsepower rating	Percentage of total
1935 and later	132700	12.0
1930 and later	220700	20.0
1923 and later	543200	49.5
1919 (U.S.R.A.) and later	623500	56.7

Of the locomotives producing 220,700 hp built or rebuilt in 1930 and later, only 132,700 hp, or 12 per cent of the total considered, fully meet the transportation department's operating requirements and have the desired proportions for economy of operation and maintenance. There are 31 locomotives included in this 12 per cent, having an average rating of 4300 hp. Numerically, these locomotives are 7.3 per cent of the total having road-service assignment.

Now considering the railroads as a whole, we find that, in 1939, reports[2] were filed for 45,965 steam locomotives. Should the road-service ratio of 70 per cent be applied to the 45,965 steam locomotives reported in order to arrive at the approximate number having road assignment, we would have a total of 32,175.

The record[3] of purchases of steam locomotives for service in the United States, 1934 to 1939, inclusive, is given in Table 2.

TABLE 2 LOCOMOTIVE PURCHASES; 1934-1939

Construction orders placed	Service: Road	Service: Yard
1934	63	9
1935	17	11
1936	349	84
1937	149	27
1938	33	2
1939	88	2
Total	699	135

Numerically the 699 steam locomotives listed in Table 2, purchased for road service, constitute only 2.2 per cent of the 32,175 steam locomotives considered as having assignment to this service. These new locomotives have approximately double the rated horsepower capacity of the average of the total and accumulate mileage at rates 2 to 3 times that of the average. On this basis, they should account for 10 to 15 per cent of the transportation movement. This leaves 85 to 90 per cent of the movement being handled by locomotives built prior to 1934. A number of the freight locomotives, built in the period 1928 to 1931, were proportioned to meet present operating requirements; the majority, however, while having good boiler proportions and good steam

[1] Mechanical Engineer, St. Louis-San Francisco Railway Company.

Contributed by the Railroad Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

[2] Twenty-Eighth Annual Report of the Chief Inspector, Bureau of Locomotive Inspection, Interstate Commerce Commission, U. S. Department of Commerce, Washington, D. C., 1939.

[3] "Locomotive Purchases by American Railroads," Annual Statistical issues of *Railway Age*, vol. 98, 1935, p. 155; vol. 100, 1936, p. 71; vol. 102, 1937, p. 70; vol. 104, 1938, p. 76; vol. 106, 1939, p. 79; January 6, 1940, p. 78.

distribution, continued with wheel diameters which are a handicap today.

Returning to the figures on the locomotive ownership of the author's employer, it will be observed that, of the 1,096,100 rated horsepower, representing the total capacity of the 425 locomotives assigned to road service, locomotives having 12 per cent of the rated total are considered modern, while locomotives accounting for 44.7 per cent of the rated total were built commencing with the U.S.R.A. period and from then on to the time when those having modern operating proportions were constructed. With the groups of locomotives in mind which will fall within the period of construction of the 44.7 per cent mentioned, it is suggested that studies similar to the following be undertaken with the object of making maintenance replacements as nearly according to modern proportions as possible in preference to the "as-built" proportions. Regardless of our opinions with respect to the economical retirement age of equipment, these locomotives will, in all probability, be continued in service for many years.

In some instances at no additional cost, and in many instances at a nominal additional cost, distinct improvements may be effected in the capacity and economy of locomotives by the reproportioning of parts which are subject to renewal from time to time through the routine of maintenance.

Boiler Proportions Which Should Be Examined

The boiler is an excellent starting point when reviewing the design and proportions of a locomotive for possible improvement.

Many of the boilers designed in the days of drag service have inadequate steam space for the high steam-release rate obtained under present operating conditions. When a new firebox is applied, this condition can be readily corrected. The lowering of the crown sheet 3 in. will increase the volume of the steam space 20 to 25 per cent. This is frequently sufficient to transform a poor water-carrying boiler into a good performer. When this is effected, the results are:

(*a*) Better performance on line of road;
(*b*) Higher superheat temperature;
(*c*) Reduction in maintenance of valves, pistons, and superheater units.

Locomotives, designed for operation on heavy-grade lines and having permanent reassignment where only light-grade lines are encountered, should be checked for the lowest reading of the water glass relative to the highest point of the crown sheet and for the visible length of the water glass used. A gain of 15 to 20 per cent in steam space is at times possible by a slight lowering of the water glass and reduction in its visible length, maintaining the same degree of safety in operation on the light-grade line as prevailed on the heavy-grade line for which the locomotives were built.

The gas area through the barrel of the boiler is one of the all-important details which should be checked in order to provide the maximum attainable. In the design of locomotives constructed in the period 1919 to 1930, some railroads incorporated practices in the spacing of tubes which today are recognized as not being consistent with spacing that may be followed with good results, water treatment and welded flues effecting this permissible change.

A case in point was the building some years ago of 50 type-2-8-2 locomotives by a certain railroad, following in detail the boiler dimensions of the U.S.R.A. 2-8-2 B, except for the layout of the tube sheets. The latter type had 45 flues $5^1/_2$ in. in diam, and 247 tubes $2^1/_4$ in. in diam. The 50 locomotives of the 2-8-2 type mentioned have 45 flues $5^1/_2$ in. in diam and 219 tubes $2^1/_4$ in. in diam.

A kindred condition can also be found in the proportioning of some boilers having combustion chambers with the water space around the chambers greater than is now required for good practice. The area of the back tube sheet is generally the limiting factor in the tube application to these locomotives. A reduction of the water space around the combustion chamber when applying a new firebox could be capitalized upon through the application of additional boiler tubes.

Boiler-Tube-Sheet Layout and Superheat

With the results at hand on the improved cylinder performance of modern and semimodern locomotives, a high percentage of which is attributable to steam-chest temperatures of 700 to 750 F, when we apply new tube sheets in the course of maintenance, the reproportioning of the tube layout to provide high steam-chest temperatures offers an excellent opportunity for increased capacity and economy.

The tube-sheet layout of the U.S.R.A. locomotives is proportioned so that the $5^1/_2$-inch flues will have a flue-gas area of 45 to 46 per cent of the total gas area through the boiler. Such a proportion with the type-A superheater gives a steam temperature approximately 100 F below that desired in today's operation.

Table 3 shows the tube-and-flue application with resulting proportions for the U.S.R.A. 2-8-2 B locomotive as built, as well as a number of possible applications without requiring any change in the crown height or water space around the combustion chamber. The order of application to attain increased capacity, as well as for fuel economy, would be as follows:

1 Type-E superheater.
2 A 6 × 9 layout of $5^1/_2$-in. flues with the application of HA superheater units or their equivalent.
3 A 6 × 10 layout of $5^1/_2$-in. flues, with the top corner flues omitted; application of 58 type-A superheater units.

Increasing the capacity of the superheater effects a material gain in addition to that of reducing the steam rate per unit of work, since the increased number of units reduces the pressure drop, which at a high work rate is equivalent to a substantial increase in the boiler pressure.

TABLE 3 POSSIBLE TUBE AND FLUE APPLICATIONS ON U.S.R.A. 2-8-2 B TYPE LOCOMOTIVE WITHOUT CHANGE IN DIMENSIONS OF BACK TUBE SHEET

Superheater type	A	HA or equivalent	A	E
Superheater flue layout	5 × 9 (As built)	6 × 9	6 × 10	......
Distance over tube sheets, ft	19	19	19	18
Number of $5^1/_2$-in. flues	45	54	58	201–$3^1/_2$
Number of $2^1/_4$-in. tubes	247	217	196	62–$2^1/_4$
Vertical pitch of $5^1/_2$-in. flues, in	$6^1/_2$	$6^1/_2$	$6^1/_2$	4.22 (Mean pitch)
Pitch of $2^1/_4$-in. tubes, in	3	$2^{15}/_{16}$	$2^{15}/_{16}$	3
Heating surface of flues, sq ft	1226	1471	1580	3298
Heating surface of tubes, sq ft	2752	2418	2184	697
Total heating surface, tubes and flues, sq ft	3978	3889	3764	3995
Superheater heating surface, sq ft	993	1742	1280	1920
Steam area through superheater, sq in	51.3	61.6	66.1	71.92
Net gas area through boiler, sq in	1414	1447	1438	1427
Net gas area through $5^1/_2$-in. flues, per cent	45.1	52.9	57.2	......
Approximate temperature range of steam in branch pipe, deg F at high work rate	630–640	HA 710–730	700–720	710–730
Maximum evaporation, tubes and flues (Cole's values)	37984	37125	36175	39320
Maximum evaporation, including firebox heating surface (Cole's values)	54809	54010	53060	56205

TABLE 4 EXAMPLES OF REPROPORTIONED TUBE-SHEET LAYOUT BY ST. LOUIS-SAN FRANCISCO RAILWAY

	Example No. 1			Example No. 2	
	As built	Reproportioned	Reproportioned	As built	Reproportioned
Superheater type	A	A	HA	A	A
Superheater flue layout	5 × 9	7 × 9	6 × 9	5 × 8	6 × 8
Combustion chamber—with or none	With	With	With	None	None
Back tube sheet altered	..	No	No	..	Crown sheet lowered 3 in.
Distance over tube sheets, ft-in.	22-0	22-0	20-0	21-0	20-11
Number of $5\frac{1}{2}$-in. flues	45	63	54	38	48
Number of $2\frac{1}{4}$-in. tubes	251	211	242	225	176
Vertical pitch of $5\frac{1}{2}$-in. flues, in.	$6\frac{3}{4}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$
Pitch of $2\frac{1}{4}$-in. tubes, in.	$3\frac{1}{8}$	$3\frac{1}{16}$	$3\frac{1}{16}$	$3\frac{1}{8}$	$3\frac{1}{16}$
Heating surface of flues, sq ft	1420	1988	1548	1144	1440
Heating surface of tubes, sq ft	3240	2724	2839	2772	2160
Total heating surface, tubes and flues, sq ft	4660	4712	4387	3916	3600
Superheater heating surface, sq ft	1233	1726	1834	978	1235
Steam area through superheater, sq in.	51.3	71.8	61.6	43.3	54.7
Net gas area through boiler, sq in.	1426	1556	1500	1245	1233
Net gas area through $5\frac{1}{2}$-in. flues, per cent	44.7	57.4	49.3	43.3	55.2
Temperature range of steam in branch pipe, deg F, at high work rate	590–620	690–710	[a]690–720	...	680–700
Maximum evaporation, tubes and flues (Cole's values)	41610	42220	41370	36010	33170

[a] Combustion chamber lengthened and siphon applied. Firebox heating surface increased 18.3 per cent.

TABLE 5 EXAMPLES OF CHANGE IN VALVE EVENTS AS MADE BY THE ST. LOUIS-SAN FRANCISCO RAILWAY TO MEET ALTERED OPERATING REQUIREMENTS; EXAMPLES OF RECENT CONSTRUCTION

	Example No. 1		Example No. 2		Example No. 3 recent construction		
	Original	As altered	Original	As altered			
Locomotive type	4-6-2	4-6-2	4-6-2	4-6-4	2-8-2	4-8-2	4-8-2
Class of service	Passenger	Light fast passenger	Heavy passenger	Conversion, heavy fast passenger	Freight	Freight	Freight
Boiler pressure, psi	200	200	210	225	235	250	210
Cylinders, diameter and stroke, in.	24 × 28	24 × 28	26 × 28	26 × 28	27 × 32	27 × 30	29 × 32
Drivers, diameter, in.	69	73	74	74	64	70	70
Valves, diameter, in.	13	13	13	13	14	14	15
Maximum travel, in.	$7\frac{1}{2}$	$7\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{2}$	$7\frac{3}{4}$	$8\frac{1}{4}$
Steam lap, in.	$1\frac{1}{4}$	$1\frac{1}{2}$	1	$1\frac{5}{8}$	$1\frac{15}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$
Lead, in.	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$
Exhaust clearance, in.	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	0	0	0
Maximum cutoff, per cent	...	82	...	79	77	77	76

Table 4 contains examples of reproportioning the superheater application on two classes of locomotives by the author's employer.

Effect of Valve Events on Locomotive Operation

As important as the proportioning of the boiler and the superheater, are the valve events upon the operation of a locomotive. Classes should be checked having in mind today's assignment. A 4-6-2 type built to handle the heavy trains of another period, with valves having 1 to $1\frac{1}{8}$-in. steam lap, should not be assigned to light, high-speed trains without altering the valves and valve gear to provide events to suit. Locomotives of the 2-8-2 type, designed in the days of drag service, may be found operating on near passenger schedules and with practically the original restricted steam ports and valve events. This necessarily results in loss of power and fuel.

Classes, which are receiving the application of new cylinders, should have the diameter of the valve, area of the exhaust channels, and the steam ports carefully examined. They should be proportioned to meet today's requirements. Only a few locomotives need be involved to justify the cost of a new cylinder pattern, should it be required, in order to obtain the desired proportions.

Considering the fact that locomotives in freight service are rated today on their power output at piston speeds of 1200 to 1400 fpm, instead of on their initial tractive effort, the responsibility devolves upon the mechanical engineers at least to point out the potential power increases which may be effected through moderate changes. At the time of heavy shopping, a valve gear, providing drag-service events, can be replaced with a gear providing modern events, often at slight cost over that which would be involved in maintaining the original in kind.

Table 5, examples Nos. 1 and 2 are instances of altering the valve gears to meet changed assignments and operating conditions. In both cases the originals were for passenger service with running speeds of 55 to 60 mph. The alterations were made to provide valve events to accommodate an economical cruising speed of 70 to 75 mph, with occasional top speeds of 80 to 85 mph.

Should one review a table showing the steam lap, lead, exhaust clearance, valve diameter, etc., for the various locomotives recently built, the question could well be asked what proportions and valve events should be provided to meet today's operating requirements most satisfactorily? The locomotives which we are considering are those built from 1919 to 1930, the majority having working pressures within the range of 200 to 250 psi.

The problem is to provide the highest possible mean effective pressure at piston speeds of 1200 fpm and higher. L. H. Fry's recent review[4] of the reproportioning of locomotives by the Paris-Orleans Railway may be read to advantage by those having to do with steam-locomotive proportions; also by those having to do with the maintenance. In the latter case, the review should be studied in order that a better understanding will exist when a slight increase in maintenance is assumed in order to effect a substantial increase in the work-rate capacity.

Indicator cards, Figs. 1 to 4, inclusive, are shown as an example of the increase in mean effective pressure which may be effected through the adoption of a long steam lap. They were taken from a 2-8-2-type locomotive having 45 type-A superheater units, 27 × 32-in. cylinder, 14-in-diam valve, $8\frac{3}{4}$-in. maximum travel. The valve setting for cards, Figs. 1 and 3, was as follows: $1\frac{1}{4}$-in. steam lap, $\frac{3}{16}$-in. lead, 0-in. exhaust clearance, $1\frac{5}{8}$-in-width steam port. The valve setting for cards, Figs. 2 and 4, was as follows: $2\frac{1}{2}$-in. steam lap, $\frac{3}{16}$-in. lead, $\frac{1}{16}$-in. exhaust lap, $2\frac{3}{16}$-in-width steam ports.

Example No. 3 Table 5 shows the steam lap, lead, exhaust clearance, valve diameter, maximum cutoff, etc., which the author's company uses as the most practical for fast heavy freight service. With a $1\frac{15}{16}$-in. steam lap and valve travel to provide 75 to 77 per cent maximum cutoff, auxiliary starting ports are not required. Cards shown in Figs. 5 and 6 were

[4] "The Locomotive in France," by L. H. Fry, *Railway Mechanical Engineer*, vol. 112, 1938, p. 473; vol. 113, 1939, p. 1; vol. 113, 1939, p. 345.

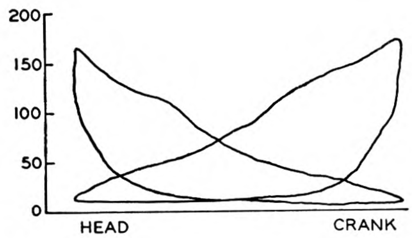

Fig. 1

Piston speed	953 fpm
Cutoff	36.5 per cent
Boiler pressure	199 psi
Mep, head end	53.8 psi
Mep, crank end	66.7 psi
Horsepower, r.s.	1004
Total engine horsepower	2008

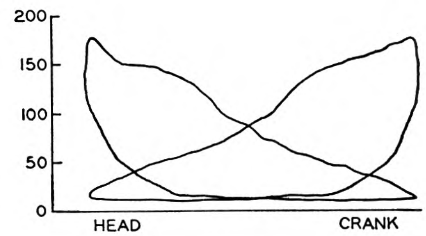

Fig. 2

Piston speed	1041 fpm
Cutoff	35.8 per cent
Boiler pressure	195 psi
Mep, head end	69.2 psi
Mep, crank end	76.1 psi
Horsepower, r.s.	1330
Total engine horsepower	2660

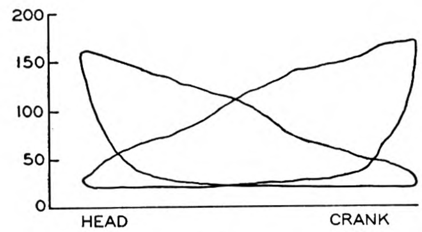

Fig. 3

Piston speed	991 fpm
Cutoff	53 per cent
Boiler pressure	198 psi
Mep, head end	67.7 psi
Mep, crank end	76.8 psi
Horsepower, r.s.	1257
Total engine horsepower	2514

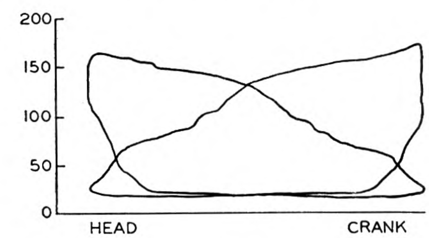

Fig. 4

Piston speed	964 fpm
Cutoff	51 per cent
Boiler pressure	193 psi
Mep, head end	86.6 psi
Mep, crank end	93.8 psi
Horsepower, r.s.	1531
Total engine horsepower	3062

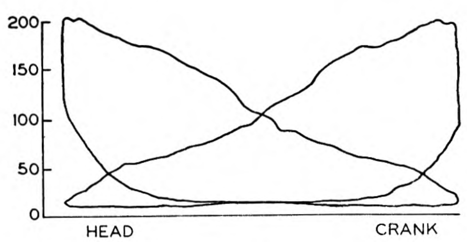

Fig. 5

Piston speed	1041 fpm
Cutoff	38 per cent
Boiler pressure	237 psi
Mep, head end	91.3 psi
Mep, crank end	92.3 psi
Horsepower, r.s.	1629
Total engine horsepower	3258

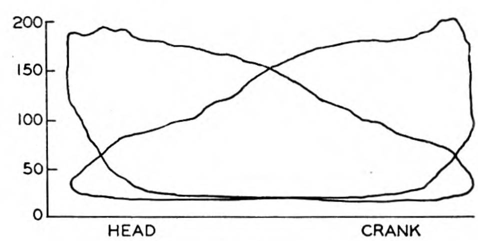

Fig. 6

Piston speed	1016 fpm
Cutoff	52 per cent
Boiler pressure	237 psi
Mep, head end	111.5 psi
Mep, crank end	115.0 psi
Horsepower, r.s.	1960
Total engine horsepower	3920

taken from the 2-8-2-type locomotive listed in Table 5, having 14-in-diam valves with $1^{15}/_{16}$ in. steam lap. These cards were taken when the locomotive was operating with a piston speed of approximately 1000 fpm. It is not difficult to visualize the shrinkage in the mean effective pressure which would result from either a reduction in the diameter of the valve or in the steam lap. With today's piston speeds of 1200 to 1600 fpm, it is doubly important that the inflow and outflow of steam be as unrestricted as is practicable. The use of valves of a diameter which may be considered large need not incur excessive weight. Lightweight built-up valves using gas pipe with a wall thickness of $^3/_{16}$ in. and steel castings having $^1/_4$ in. section have been standard practice with the author's company for 7 years.

Boiler Pressure

In the light of the excellent on-line-of-road operation which has been obtained from locomotives of recent construction, with working pressures within the higher pressure range, one might be inclined to the thought that the 200 to 225 psi working pressure, to which many of the locomotives built in the 1919 to 1929 period are limited, presents an extreme handicap to one attempting to provide increased economy and power with which to meet today's operating requirements.

From the economy viewpoint, some encouragement may be obtained from a review of the design of the recently built high-pressure locomotives. The adoption of the higher working pressures without modification of design to provide for increased ratio of expansion does not admit of the increase in thermal efficiency of the engine which is ordinarily considered a result of the use of the higher pressure. The decrease in the differential between the two is particularly true where a heavy work rate with reduced ratio of expansion is involved, the locomotives in both pressure ranges working at approximately the same cutoff or with the same ratio of expansion.

High pressure is forced at times where high piston thrust is required, and the cylinder diameter must be limited to keep within clearance limits. There is no denying that high pressure gives an engine a "smartness" of response; however, from the standpoint of capacity and for operation within the present operating requirements of high-speed freight service, much may be accomplished with working pressures of 200 to 225 psi. An example of this is on one of the divisions of the author's company, where the two groups of 4-8-2-type locomotives listed in example No. 3 of Table 5 are in a pool. The steaming capacity of the locomotives in the two groups is approximately the same. The 250-lb locomotives have 54 type HA superheater units with 61.6 sq in. of steam area through them. The size of the valves, events, etc., are given in Table 5.

When it was decided to condition the group of locomotives having 210 psi to work in a pool with the 250-lb locomotives, the $5^1/_2$-in. flues were increased from 45 to 63, with a resulting increase in steam area through the superheater units from 51.3 to 71.8 sq in.; the dry pipe and branch pipes were increased to suit; 29-in. cylinders having 13-in-diam valves were replaced with 29-in. cylinders having 15-in-diam valves; valve gears providing $6^1/_2$ in. maximum travel were replaced with gears providing $8^1/_4$ in. maximum travel; valves having 1-in. steam lap were replaced with valves having $1^{15}/_{16}$-in. steam lap. There is some difference in the response of the two groups, so far as the enginemen are concerned, but none, so far as the dispatchers are concerned, both handling the same tonnage on the same schedules.

Steaming Capacity—Value of Feedwater Heating

When treating the subject of providing the maximum possible capacity in existing steam locomotives, the steaming capacity which may be added to the boiler at high work rates by the application of feedwater-heating equipment, utilizing exhaust steam, should be analyzed and a distinction made between the percentage of return on the investment and the percentage increase in power; also that a net 10 per cent increase in boiler capacity is 12 to 13 per cent at the drawbar.

Conclusion

The groups of locomotives that were built in the years 1919 to 1930 offer, in general, a fertile field for a substantial addition to the work-rate capacity of our locomotives through the adoption of a policy of providing proportions to give a high degree of superheat, low pressure drop from boiler to steam chest, and valve events to conform with present-day operating requirements, these changes building up the mean effective pressure at the higher work rates and without increasing the maximum stress in frames, driving axles, crankpins, rods, etc.

These suggestions are not advanced with the intention of detracting from the thought and effort which ordinarily are put into the development of new locomotives but merely to take advantage of the opportunity that is offered daily in our shops to build greater capacity into locomotives which are having renewals made as maintenance routine and involving only a nominal amount of re-engineering.

Discussion

C. T. Ripley.[5] This paper suggests the need for improvement in valves and valve gears, in order to get better performance while a locomotive is working at high capacity. The writer is in agreement with the author's conclusions as to the possibilities of the poppet-type valve. There appears to be no question but that better results can be secured with this type of valve, provided the operating mechanism is satisfactory. Some years ago experiments were made with the Caprotti design on American locomotives. These showed a number of desirable features, but difficulties developed in making the parts rugged enough to withstand service in large locomotives. This design did, however, show very satisfactory drifting characteristics.

This matter of drifting has not been referred to, although it is a most important one. Practically all American locomotives have been built without any suitable device to make them drift satisfactorily, unless considerable steam is used in the cylinders. There have been numerous so-called drifting valves, but these have been inadequate to meet the requirements. On downgrades it is necessary to have the throttle open considerably beyond the pilot-valve stage, in order to get proper lubrication and avoid pounding. This results in a wastage of fuel and the necessity for increased braking, which also causes a loss in brake shoes and wheels. In some of the western mountain territory, locomotives drift as much as 28 per cent of the total time; for example, from Albuquerque to Gallup, a distance of 127 miles, they drift about 27 per cent of the time. From Gallup to Albuquerque, 127 miles, about 28 per cent of the time. From Seligman to Needles, 150 miles, they drift 67 per cent of the time. It is appreciated that these are extreme conditions and apply only to mountainous territory. However, there is a very considerable amount of drifting even on the more level eastern railroads. There is a distinct need for improvement in devices to permit of more economical and satisfactory drifting.

One western railroad made a study of this matter and designed large by-pass valves, connecting the ends of the valve chambers, similar to the practice on German locomotives. These connections must be about 9 or 10 in. in diam, in order to meet the requirements. They are automatic in operation, as the pressure in the steam chest closes the by-pass valve. When the pressure is not present, coiled springs act upon the valve to open it. There is also a steam connection to the cab, which can be used in emergency in case the valves are stuck open, but it is seldom necessary to use it. Operation of these valves has been very satisfactory and there is no difficulty from pounding or in maintaining lubrication in the cylinders. They have reduced the amount of braking necessary on the trains and have resulted in marked fuel savings. Recent tests have shown a saving of 7 per cent in fuel, while operating over the territory between Barstow, Calif., and Albuquerque, New Mexico, a distance of 748 miles. It may be noted that the saving in braking, through use of proper types of drifting valves, increases in the case of locomotives and trains equipped with roller bearings with their lessened friction.

The use of the poppet valve will probably make it unnecessary to have any special devices applied for improving drifting characteristics. In the case of old engines which are to be rebuilt, as suggested by the author, and in designing new engines with piston valves, some consideration should be given to this matter.

Arthur Williams.[6] Those familiar with the locomotives of the St. Louis-San Francisco Railway referred to in this paper will agree that the author and the mechanical department of that railway have produced excellent results. It is the writer's impression that this is due to the careful consideration given to all details of locomotive and cylinder design. In discussing the various items, the paper starts with the possibility of lowering the crown sheet to increase the volume of the steam space, and finishes with the valve gear and valve settings. Between the two points, the gas area through the boiler, steam area through the boiler, steam area through the superheater units and steam pipes, degree of superheat, and pressure drop of the steam are all considered.

In general, it is possible to improve a locomotive with a small superheater by increasing the number of superheater units and flues. This will increase the superheater heating surface and

[5] Chief Engineer, Technical Board of The Wrought Steel Wheel Industry, Chicago, Ill. Mem. A.S.M.E.

[6] Manager, Production Engineering Division, The Superheater Company, East Chicago, Ind. Mem. A.S.M.E.

superheat, but care must be taken that the evaporative heating surface is not decreased too much with a corresponding decrease in boiler efficiency. In Table 4 of the paper an increase in the heating surface of the tubes and flues was obtained with the larger superheater, but this was done with a decrease in the flue spacing. It is not always possible to do this. The type-E superheater is designed to give the maximum superheater and evaporative heating surface so that a high superheat is obtained at the same time as a low smokebox temperature and high boiler efficiency.

Author's Closure

The author believes that Mr. Ripley's discussion of the poppet-type valve refers either to the paper[7] presented by Charles F. Krauss, "Notes on the Trends in Reciprocating Valve Mechanisms Employing Piston Valves" or to the paper[7] presented by A. G. Hoppe, "Notes on Valve and Valve Motion Designed for Modern High-Speed Passenger Steam Locomotives." The paper presented by the author did not touch on the subject of the poppet-type valve.

Mr. Williams' discussion points out that in general it is possible to improve a locomotive with a small superheater by increasing number of superheater units and flues, but that when increasing the superheating surface care must be taken that the evaporative heating surface is not decreased too much with a corresponding decrease in boiler efficiency. The author agrees with Mr. Williams, but adds that the increased degree of superheat obtained through an increase in the number of superheater units effects an improvement in cylinder performance throughout the entire range of work rates from low to high. The decreased evaporating capacity is not vital except at the higher work rates and then is generally more than offset by the decreased steam rate resulting from the higher degree of superheat. This is particularly true of oil-burning locomotives.

[7] Abstracts of both of these papers were published in *Mechanical Engineering*, vol. 63, February, 1941, pp. 140–144.