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Mill Drying of Coal

By M. E. FITZE,[1] MILWAUKEE, WIS.

This paper develops the fact that the drying of coal in the mill while grinding, instead of in separate driers previous to mill operations, makes possible large savings in equipment, building, and operating costs, besides making possible a net gain in boiler-plant efficiency in the order of 0.5 per cent. This gain is due to improvement in air-heater performance, resulting from lesser gas flow through it as well as the reduction of approximately 6 per cent of the total gas to 150 F instead of the usual 350 F. Use of an auxiliary cyclone in the mill-vent circuit, with return of the coal separated therefrom to the mill, results in a coal loss from the system of only about 0.25 per cent.

IN THE burning of coal under power boilers one of the unavoidable losses involved is due to the moisture content of the coal as fed to the furnace. The loss is measured at the point where the flue gases leave the unit and consists of the heat of the liquid between the temperature of the incoming coal and the boiling point at atmospheric pressure, plus the latent heat of vaporization and the heat of superheat between the boiling temperature and the temperature of the outgoing gases. This is one of the minor losses in the burning of coal; nevertheless, it does attain appreciable proportions for the higher moisture coals, as shown in Fig. 1, which gives the percentage of loss for 12,000-Btu coal for various moistures and final flue-gas temperatures.

Besides the efficiency loss caused by the moisture content of the coal, moisture in the flue gas is a contributing factor to the accumulation of deposits in the lower temperature regions of air heaters and economizers, especially with the higher sulphur coals. These deposits seriously impair the heat-absorbing capacity of the surface and, in extreme cases, result in reduction of load-carrying capacity and sometimes in forced shutdown. Their removal is often difficult and expensive.

The extraction of moisture from coal for stoker firing has not been given much attention; in fact, some conditions of operation require "tempering" of the coal with water to improve combustion.

In order that over-all plant-efficiency gains, due to removal of moisture from coal may be fully realized, the moisture must be removed prior to its entrance to the furnace and be discarded to some point entirely beyond the heat-absorbing areas. Its removal must be accomplished with heat of the lowest temperature head available, preferably with otherwise waste gases.

It is evident that in the unit-fired system of pulverized-coal burning, hot-air injection into the mill is of value only in so far as it benefits mill operation and capacity. The moisture removed from the coal is injected directly into the furnace and constitutes the same loss as would be encountered in stoker firing.

The bin-and-feeder system requires removal of moisture (especially surface moisture) down to the lowest practicable value, in order that the powdered coal in handling, storing, and feeding will flow freely and uniformly at all times.

Early Drying Methods

Early developments in the art of pulverized-fuel firing by the bin-and-feeder system concentrated the drying efforts on the coal previous to pulverization. The first generally successful driers were of the cement-kiln type; inclined drums some 5 ft in diam × 40 ft long, rotating about 3 rpm. The coal was fed into the upper end, gradually working down and out at the lower end by the process of slow rotation. The coal was lifted by suitable "pick-up" plates on the inside of the drum and dropped through the hot gases which were admitted at the lower end and removed from the upper. Fines carried out by the gases were removed by a cyclone separator before the gases were discharged. Unless the drier was close enough to the boiler flue to make possible a supply of heat from that source it was necessary to equip the drier with a separately fired furnace of its own. To prevent overheating of the drier, large amounts of excess air were used which, of course, meant inefficient utilization of this portion of the coal (approximately 0.5 per cent) charged to the plant.

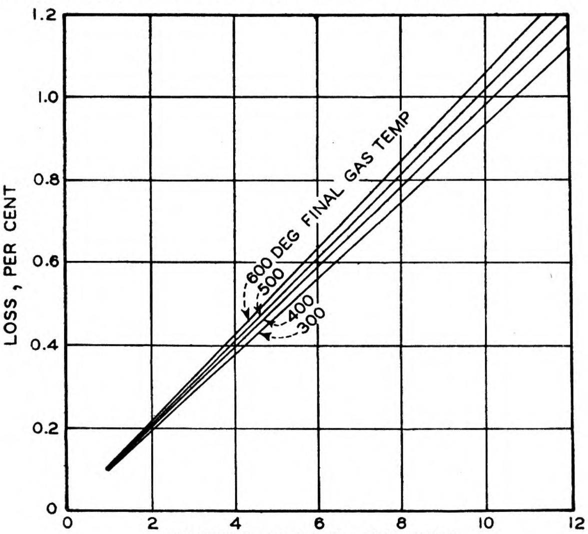

Fig. 1 Moisture Loss for Various Moisture Contents and Final Flue-Gas Temperatures
(12,000 Btu per lb of coal as fired.)

Later, driers were developed in which the coal was passed over steam-heated grids or plates with a stream of air passing counterflow to the coal. This made possible the utilization of bled steam from the turbines, but capacities were not very high for the space occupied and, where long steam lines were required, considerable installation expense was involved.

All these separate drying schemes were predicated on the use of a pulverizing plant separate from the rest of the boiler plant. This involved:

1 High building and equipment cost.
2 High maintenance cost.
3 High operating labor cost.

Mill-Drying Development

In order to overcome these three high-cost factors of separate

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Contributed by the Fuels Division and presented at the Joint Meeting, Birmingham, Ala., November 7–9, 1940, of the Coal Division of the American Institute of Mining and Metallurgical Engineers and the Fuels Division of The American Society of Mechanical Engineers.

coal drying and milling and, at the same time, to retain the inherent high efficiency and all-around operating flexibility of the bin-and-feeder system, the mill drying of coal with flue gas taken from the latter part of the working cycle was developed.

Fig. 2 shows diagrammatically the arrangement of mill-drying equipment installed in three plants of the Wisconsin Electric Power Company and now being installed in a fourth.

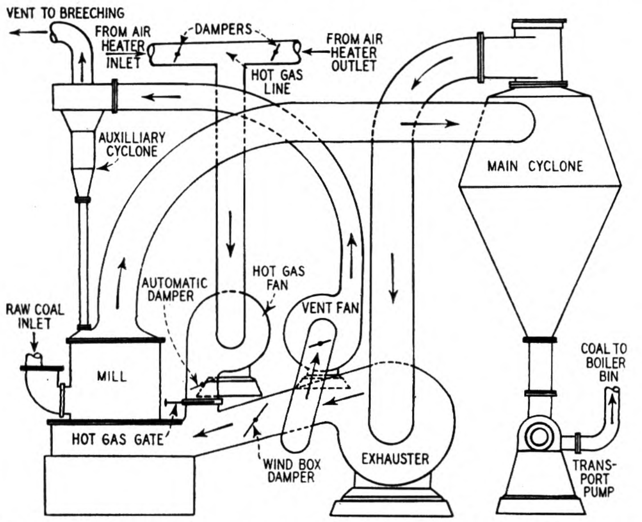

Fig. 2 Diagrammatic Arrangement of Mill-Drying Equipment Using Hot Flue Gas From Boiler Unit as Source of Heat

Flue gas is taken from the last pass of the boiler and the air-heater outlet, properly proportioned by suitable damper control to give the desired temperature, and injected by the hot-gas fan into the mill circuit at the entrance to the mill wind box. Gas temperatures about 550 F are used. In order to maintain pressure in the mill circuit at some constant value and to rid the circulating system of moisture removed from the coal, a vent fan withdraws from the circuit, at the discharge of the main mill fan and just prior to the injection of the hot gas, such volume of moisture-laden gases as is necessary. The vent fan discharges through a cyclone separator and thence to the flue for discharge with the boiler flue gas.

It will be noted that the mill exhauster is located on the clean side of the main cyclone, obviating the necessity of accelerating the entire mill output through the fan wheel. This results in a saving in fan power and blade erosion which is considerable. It also makes possible the use of a more efficient type of fan wheel. The power saving may amount to 2 or 3 kwhr per ton of coal.

The auxiliary collectors discharge their separated coal into the top of the mill; the principle of allowing considerable downward gas flow through their bases aids their efficiency materially. When separating coal of such fineness that 98 per cent passes a 325-mesh sieve, their efficiency averages upward of 80 per cent when 500 cfm is induced downward through the apex.

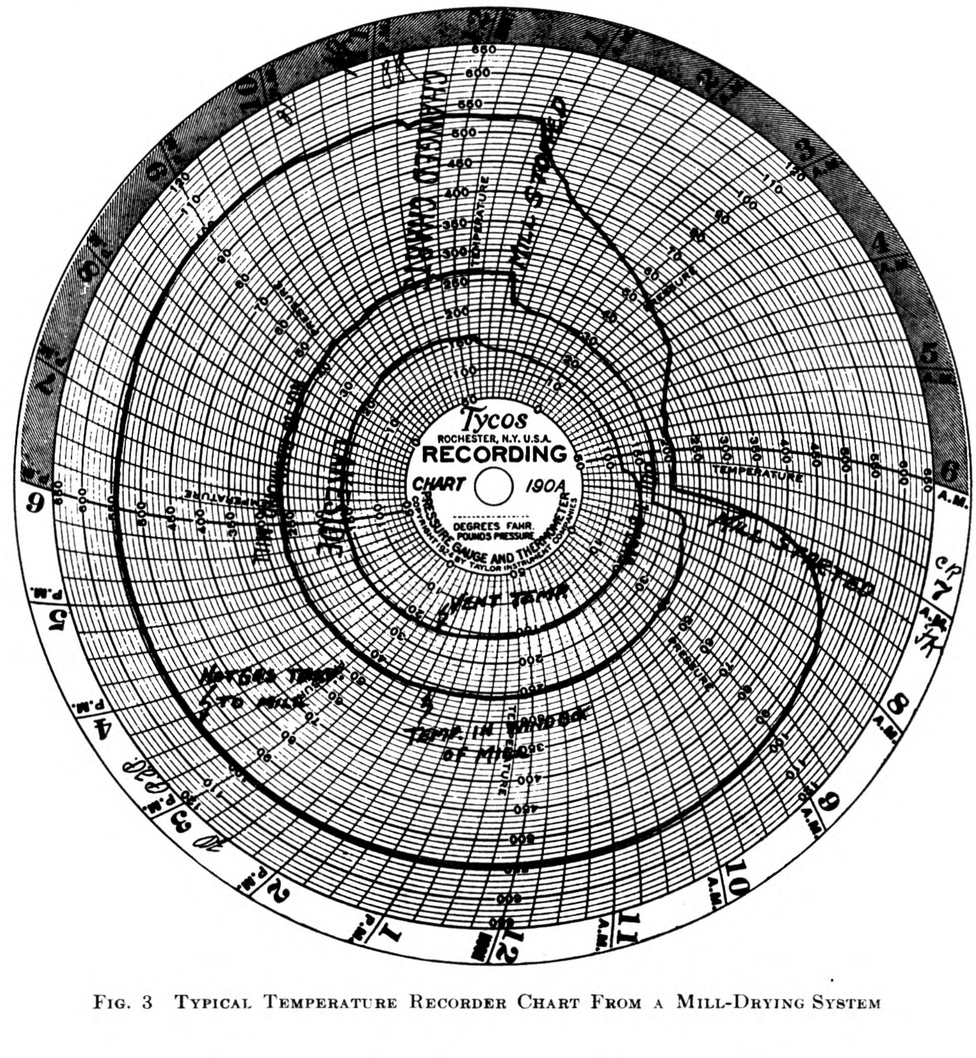

Fig. 3 Typical Temperature Recorder Chart From a Mill-Drying System

TABLE 1 TEST RESULTS OF A MILL-DRYING SYSTEM

General	
Date	1/8/30
Mill	20
Kind of coal	High fusion (Eastern)
Coal milled, tons	43
Hours run	3.58
Mill-motor current, amp	230
Pump-motor current, amp	23
Air Data	
Top of mill pressure, in. water	−6.9
Fan-inlet pressure, in. water	−17.4
Fan-discharge pressure, in. water	1.8
Pressure rise through fan, in. water	19.2
Wind-box pressure, in. water	−0.34
Wind-box temperature, F	210
CO_2 in hot gas, per cent	12.0
CO_2 in mill system, per cent	10.5
Air leakage, per cent	12.5
Air leakage (mostly through feeders), lb per hr	9440
Humidity of leakage air (approx), per cent	90
Moisture in leakage air, lb per hr	74
Return-air temperature (dry bulb), F	147
Return-air temperature (wet bulb), F	123
Saturation, per cent	50
Velocity head in return pipe (25 3/4 in. diam), in. water	1.62
Volume through mill, cfm	20100
Vent-fan-inlet pressure, in. water	−2
Vent-fan-discharge pressure, in. water	2
Pressure rise, in. water	4
Vent-fan speed, rpm	640
Velocity head in vent-fan-discharge pipe (23 3/8 in. diam), in. water	0.139
Vent, cfm	4700
Vent, lb per hr	17700
Moisture per lb of dry vent air, grains	590
Moisture removed in vent, lb per hr	1490
Hot-gas fan-inlet pressure, in. water	−3
Hot-gas fan-discharge pressure, in. water	0.28
Pressure rise through fan, in. water	3.28
Hot-gas fan speed, rpm	700
Velocity head in hot-gas line (29 3/4 in. diam), in. water	0.048
Hot-gas, temperature, F	470
Hot gas, cfm	5700
Hot gas, lb per hr	14700
Dew point of hot gas, F	100
Moisture per lb dry gas, grains	300
Moisture in hot gas, lb per hr	630
Dust Concentrations	
Vent-cyclone inlet (main cyclone outlet) lb per cu ft	0.0019
Vent-cyclone outlet, lb per cu ft	0.00023
Power Consumption	
Mill motor, kwhr	436
Total kwhr (mill, fan, pump, vent fan, and hot-gas fan)	816
Coal Data	
Proximate analysis as milled	
Moisture, per cent	5.2
Volatile matter, per cent	34.47
Fixed carbon, per cent	55.32
Ash, per cent	10.21
Total, per cent	100
Sulphur, per cent	1.80
Dry, Btu	13279
As received, Btu	12587
Temperature after milling, F	147
Temperature before milling, F	42
Temperature rise, F	105
Moisture before milling, per cent	5.2
Moisture after milling, per cent	2.1
Moisture removed, per cent	3.1
Fineness before milling	
Per cent through 1/4-in. mesh	61.6
Per cent through 1/2-in. mesh	88.5
Per cent through 3/4-in. mesh	93.9
Per cent through 1-in. mesh	100
Fineness after milling (Tyler Standard Screen Scale)	
Per cent through 200 mesh	60.9
Per cent through 100 mesh	81.8
Per cent through 48 mesh	97.16
Per cent through 28 mesh	99.75
Per cent through 20 mesh	99.95
Per cent through 10 mesh	100

Moisture Balance

	Lb per hr	Per cent
Moisture input with coal	1250	64
Moisture input with hot gas	630	32
Moisture input with room air leakage	74	4
Total moisture input	1954	100
Moisture remaining in coal	505	26
Moisture removed in vent gas	1490	76
Total moisture accounted for	1995	102
Moisture unaccounted for	−41	−2

When extracting flue gases for mill drying from adjacent to the air-heater gas inlet, flue-gas losses at the exit of the air heater are reduced, since less heat requires transfer and the ratio of air flow to gas flow is appreciably increased. In cases where no economizers are employed and the boiler-outlet-gas temperature is high, improvement in this ratio nets rapid gains in air-heater performance.

Further reduction in flue-gas losses results with mill-drying systems in the lowering of some 6 per cent of the flue gas to the vent temperature of 150 F instead of the 350 F at the outlet of the air heater. Thus, the cold moist coal is used as an "economizer," reducing flue-gas losses far below any present practice. Returning the vent to the furnace merely to reclaim the dust loss would not make these gains possible, for full gas flow through the air heater would result in the following:

1 No different air-heater performance from usual.

2 Vent gas which had already been cooled to 150 F would be reheated again to air-heater-outlet temperature.

3 Lowered moisture loss would not be realized, since the full moisture content of the coal would be injected into the furnace, just as with unit mill or stoker firing.

4 Air-heater deposits would be aggravated by the higher moisture gases passing through it.

Operation of Drying System

In operation, the system has proved highly successful. During the last 11 years 268,000 mill-hr have been amassed by eight mills aggregating some 55 mill-years of operation. Approximately 4,400,000 tons of both Eastern and Midwestern coal have been ground.

Maintaining an atmosphere of CO_2 in the mill circuit is a safety feature of major importance, for the operating record cited has been accomplished without any fires or explosions having occurred. With this inert gas in the system, safety codes allow the installation of mill-drying equipment within the boiler room without the use of fire walls for separation from other equipment.

In starting up the equipment, a CO_2 content in the system of upward of 12 per cent can be established in 1 min time by closing the wind-box damper shown in Fig. 2, opening the vent damper, hot-gas gate, and hot-gas automatic damper and allowing the high draft in the boiler outlet to pull gases backward through the system from the breeching.

In operation, the automatic hot-gas damper is held open pneumatically by suction in the top of the mill. Should the

TABLE 1 (*Continued*)

Dust Balance and Cyclone Efficiency

Lb per hr of coal in main cyclone outlet (vent-cyclone inlet)	2290
Lb per hr of coal in vent-cyclone inlet	540
Lb per hr of coal in vent-cyclone outlet	65
Main-cyclone efficiency, per cent	90.5
Vent-cyclone efficiency, per cent	88
Coal lost in vent, per cent	0.27

Heat Balance

(Datum is return-air temperature)

	Mill Btu per hr	Per cent
Heat input		
Hot gas, above return air	1.19	68.4
Electrical (mill and exhauster)	0.55	31.6
Total	1.74	100
Heat output		
Sensible heat in coal	0.63	36.2
Heat-up and evaporate moisture	0.88	50.6
Heat-leakage air	0.23	13.2
Total accounted for	1.74	100
Unaccounted for	0	0

Mill Performance

Tons milled per hr	12
Mill power, kwhr per ton	10.1
Total power, kwhr per ton	19

TABLE 2 PRINCIPAL INSTALLATION AND OPERATING DATA FOR THREE MILL-DRYING INSTALLATIONS OF WISCONSIN ELECTRIC POWER COMPANY

Plant	Lakeside	Port Washington	East Wells Street
Year installed	1929-1930	1935	1938
No. of units installed	4	2	2
Type of mill	Roller	Roller	Bowl
Rated capacity, tons per hr	15	15	12
Total hours run to date	212500	48000	7700
Operating Data (1939)			
Kind of coal	Midwestern	Eastern	Operating data not available
Tons milled	361900	181000	
Mill-hours	21620	10920	
Power consumed, kwhr	6433000	2730000	
Tons, milled per hr	16.7	16.6	
Kwhr consumed per ton	17.8	15.1	
Moisture			
Inlet, per cent	9.03	4.0	
Outlet, per cent	5.06	1.8	
Removed, per cent	3.97	2.2	
Fineness, per cent, through			
200 mesh	65.87	66.42	
100 mesh	85.56	89.10	
48 mesh	98.32	98.20	
28 mesh	99.81	99.83	
20 mesh	99.98	99.98	
10 mesh	100	100	

main exhauster fail, the loss of suction in the mill releases a holding latch and the damper is closed by a falling weight. This prevents injection of heat into the system at a time when it cannot be circulated.

Corrosion of the mill system was anticipated until experience with test specimens, placed in locations most favorable to corrosion, indicated that the flue-gas system would deteriorate equipment practically no sooner than an air-drying system. Thorough insulation of all piping and collectors apparently limits deterioration principally to erosion. No other maintenance problems of consequence have developed which are attributable to the use of flue gas.

Mill capacities have not been found different from those obtained using coal dried in separately fired driers. Practically the only troubles encountered have been in connection with feeding wet coal to the mills through supply hoppers and feeders.

Table 1 shows results of a test run on one of the mill-drying mills at Lakeside soon after the first installation was made, including data on moisture and heat balances and dust loss to the vent. The mill capacity found on this test is low and power consumption high, due to reduced mill speed for the trial and also coal that was unusually hard to grind. Outputs in the order of 15 to 18 tons per hr are more usual.

Later installations make use of a more efficient type of main cyclone, resulting in removal efficiencies of around 97 per cent instead of 90.5 per cent, so coal losses to the vent are correspondingly less.

Table 2 shows the principal installation and operating data for the mill-drying equipment installed to date by the Wisconsin Electric Power Company.

Fig. 3 shows a typical temperature recorder chart from a mill-drying installation.

Economy of System

An accounting of heat gains and losses due to the mill-drying system of coal preparation shows the following:

	Per cent
Improvement in air-heater performance; 6 per cent gas extracted (10 per cent less gas through heater = 30 F lower out temperature) (37.5 F lower flue-gas temperature = 1 per cent boiler efficiency)	0.48
Reduction of 6 per cent of flue gas from 350 F to 150 F	0.32
Total credit to mill drying	0.80
Coal lost in vent	0.25
Net credit due to mill drying	0.55

This tabulation shows that the drying is effected more cheaply than it can be done in the furnace or with air drying, for the system actually improves boiler-room efficiencies by 0.5 per cent. Operating separately fired or steam driers requires about an equivalent heat expenditure, which indicates that mill drying is one per cent more efficient than the older drying processes.

The economies mentioned are all thermal economies. In addition, mill drying must be credited with large savings in building, equipment, maintenance, and operation costs. The process has practically eliminated the fire and explosion hazards in the milling of coal.

Steam Generation in Steel Mills

By H. J. KERR,[1] NEW YORK, N. Y.

The author compares operating results of blast-furnace power plants for the years 1922, 1931, and 1940, at the same time indicating the changes which have occurred since 1931, when F. G. Cutler (1)[2] ably presented the developments for the preceding decade. In the course of the paper, an analysis is given of the factors delaying the use of high steam pressures in steel-mill practice, and the problems to be solved in raising steam temperatures. Numerous blast-furnace boiler installations, typical of the best modern practice, are briefly described and illustrated.

IN APRIL, 1931, the late F. G. Cutler, an outstanding steam engineer of the steel industry, presented, from this platform, a paper entitled, "Design Features and Operating Results of Fairfield Blast Furnace Power Plant" (1).[3] In Table 2 of that paper he gave a digest of the results that had been obtained, and compared them with those of 1922. In this comparison, Mr. Cutler showed the improvement obtained by: The more efficient combustion of blast-furnace gas, the use of electrostatic precipitators for the cleaning of gas, the increase in steam pressure from 150 to 340 pounds, the combination of pulverized coal and blast-furnace gas in the same boiler furnace, and the increase in capacity of boiler units from 70,000 to 175,000 pounds of steam per hour.

Nine years have elapsed since the presentation of Mr. Cutler's paper and, as many changes have taken place in steam generation during this period, it may be of interest to compare present-day conditions with those of that time. Table 1 shows such a comparison for 1922, 1931, and 1940:

The first two columns of Table 1 constitute additions to, and abbreviations of, Table 2 (1) as given by Mr. Cutler. In making up the third column, some liberty has been exercised, by taking known facts from central-station practice and modifying these values by consideration of steel-mill problems.

Changes in Steam-Generating Practice 1931–1940

Considering in turn the various items in the list, the following changes taking place from 1931 to 1940 will be noted:

1 Steam pressure increased from 340 to 900 psi, with steam temperature increasing from 650 to 850 F.

2 Boiler efficiency has not been materially increased, due largely to economics, but has been maintained, notwithstanding the higher steam temperatures of today, by the use of air heaters and economizers. Stove efficiency about the same.

3 Capacity of boiler units has been greatly increased. This development, in the steel industry, has been held back by the dirtiness factor of blast-furnace gas, since large units do not lead to flexibility of operations, unless they can be maintained in service over reasonably long periods of time. It will be noted that this limitation does not exist at Fairfield, since clean blast-furnace gas has been in use for some time. Other steel plants are making real improvements in this direction.

4 Kilowatthours per ton of product (surplus) increased, due largely to increased efficiency of high-pressure steam units.

In regard to furnaces, two yardsticks are given in Table 1. The first is the capacity factor of "Btu per cubic foot of furnace volume." In the past this expression has been greatly misused. It is, however, a definite measure of the size of furnace in use for the combustion of a specific quantity of fuel, and should not be discarded any more than it should be misused.

The table shows the reduction in this value during the transition from stokers to pulverized fuel, between 1922 and 1931, and then the increase in value as water cooling in furnace practice developed, as shown for 1940. These figures indicate that furnace sizes have not been determined by the combustion rates possible with pulverized coal.

Perhaps the values of Btu per cubic foot, with blast-furnace gas, furnish further evidence of this general statement, since these values are often as high as with pulverized coal, and have been for many years, notwithstanding the lower heat value of this fuel.

The second measuring stick for furnaces, "Btu available per square foot of equivalent cold surface," is one which the author discussed in 1932, at Pittsburgh (2). This factor, in general, denotes the amount of cooling received by the gases of combustion before entering the convection bank, which, in turn, determines the amount of trouble to be expected from slag. With coal-firing, it is to be noted that high values were used in the past with stokers operating at capacities which permitted burning the coal on the stoker grate. As the practice shifted to pulverized coal, and not considering unsatisfactory installations, this value seemed to level off at approximately 200,000 Btu per sq ft of equivalent cold surface for coal having an ash-fusion temperature of the order of 2200 F.

In the case of the blast-furnace gas, very much higher values

TABLE 1 STEEL-MILL POWER PERFORMANCE IN 1922, 1931, AND 1940

	1922	1931	1940
Steam pressure, psi	150	340	900
Steam temperature, F	450	650	850
Boiler efficiency, per cent	66	83	83
Stove efficiency, per cent	60	70	70
Dirt in blast-furnace gas, grains per cu ft	3	0.5	0.05
Size of boiler units, lb steam per hr	60000	175000	400000
Electric current per ton of product (surplus), kw	25.3	41	53
Boiler-furnace factors			
	Btu per cu ft of furnace volume		
Coal	35000	14000	40000
Blast-furnace gas	25000	20000	25000
	Btu available per sq ft of equivalent cold surface		
Coal	300000	207000	200000
Blast-furnace gas	350000	200000	150000

TABLE 2 SUPERHEATER ABSORPTION

Throttle pressure, psi	200	400	800	1200
Steam temperature required, F	570	695	835	935
Gas temperature drop across superheater; no by-pass, F	288	435	636	795
Heat absorbed by superheater, per cent	9.6	14.5	21.2	26.5
Gas temperature drop across superheater; with regulation for constant superheat to 0.5 load, F	...	...	960	1200

[1] Executive Assistant, The Babcock & Wilcox Co. Mem. A.S.M.E.

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

[3] Bibliography (1), Table 2, p. 17.

Contributed by the Fuels Division and presented at the Joint Meeting, Birmingham, Ala., November 7–9, 1940, of the Coal Division of the American Institute of Mining and Metallurgical Engineers and the Fuels Division 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.

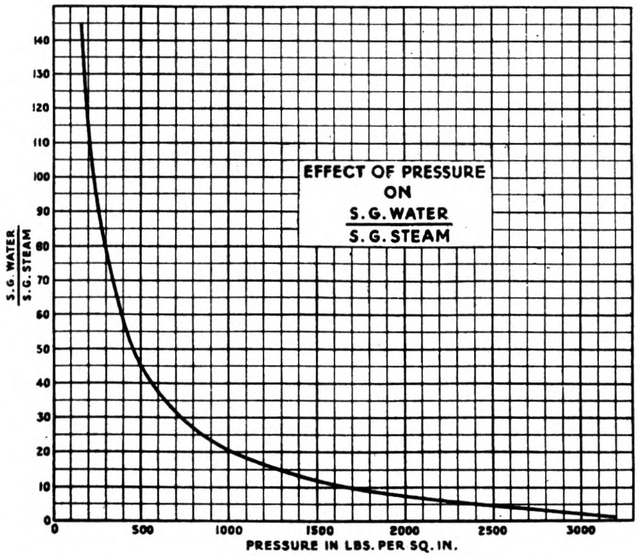

FIG. 1 EFFECT OF PRESSURE ON $\frac{\text{S. G. WATER}}{\text{S. G. STEAM}}$

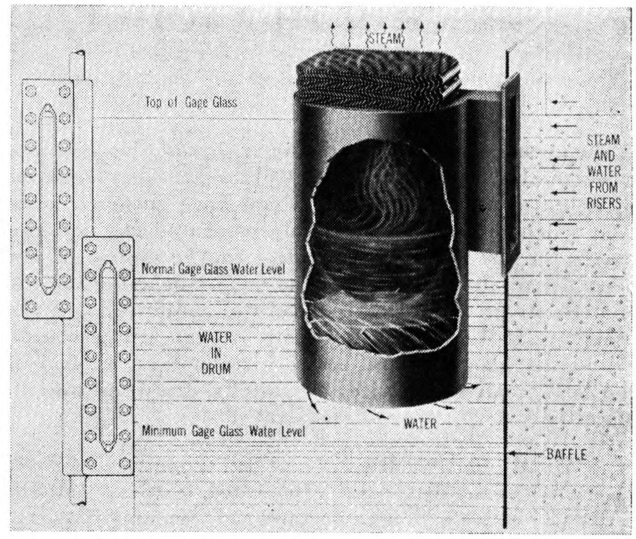

FIG. 3 CUTAWAY SECTION OF CYCLONE SEPARATOR IN BOILER DRUM

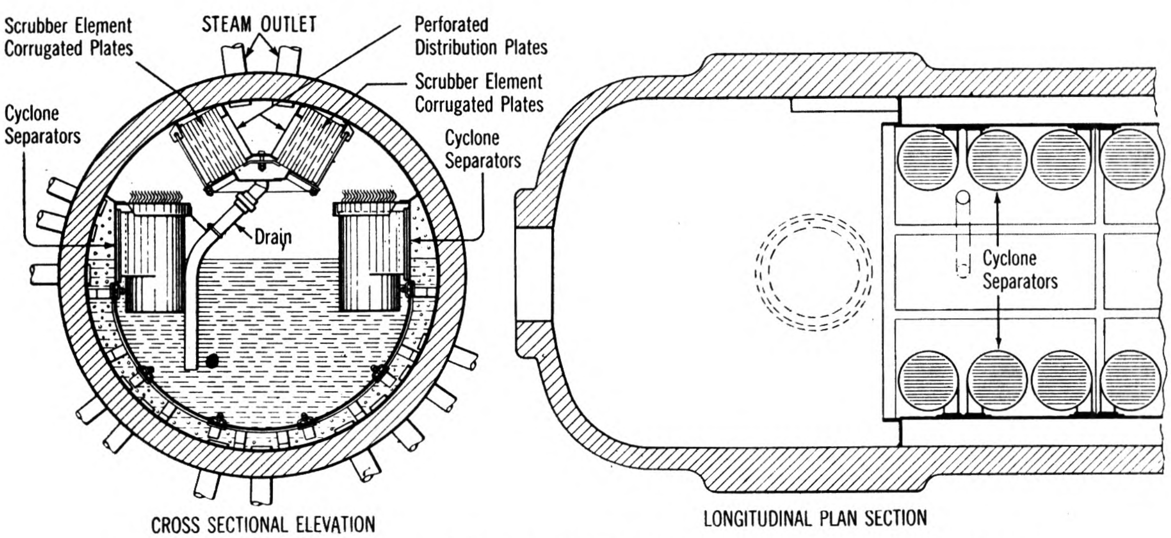

FIG. 2 TYPICAL INSTALLATION OF CYCLONE SEPARATORS IN LARGE CENTRAL-STATION BOILER DRUM

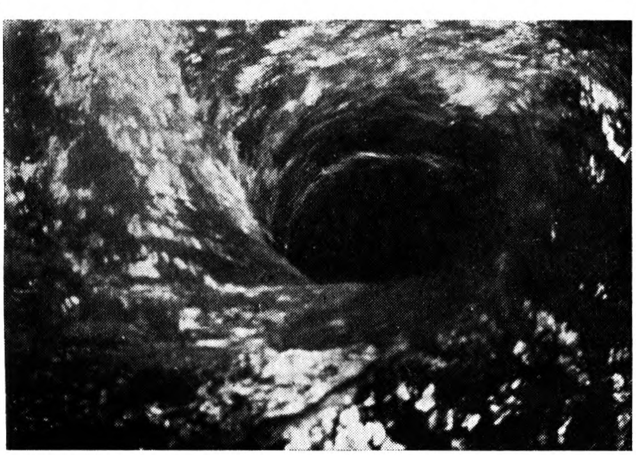

FIG. 4 VORTEX IN DOWNCOMER OF EXPERIMENTAL BOILER DRUM

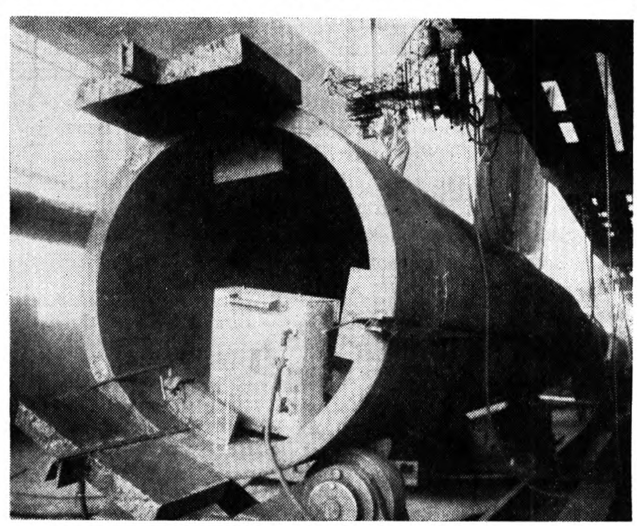

FIG. 5 WELDED DRUM FOR HIGH-PRESSURE BOILER IN COURSE OF CONSTRUCTION

of Btu per square foot of cold surface have been used in the past, as will be noted from Table 1. This, of course, was possible because of the low combustion temperatures obtained with this fuel. With later installations, utilizing blast-furnace gas, this value has decreased, solely because pulverized coal is being burned in the same furnace and is the determining factor in furnace design.

Within the last few years, E. G. Bailey has been giving considerable attention to this factor, and has referred to it in a recent paper (3).

Some discussion of the principal factors presented in Table 1 may be of interest:

A steam pressure of 1500 psi has now been in use for a sufficient period in central stations as to leave no doubt of its commercial success. One manufacturer has been using units at 1500 psi in process work, for years; a second manufacturing plant has in service 2200 psi pressure in connection with power and process, and another central station is now installing a very large unit for 2600 psi with steam reheat. Up to the present, the operating pressure in steel plants in this country has not exceeded 900 psi.

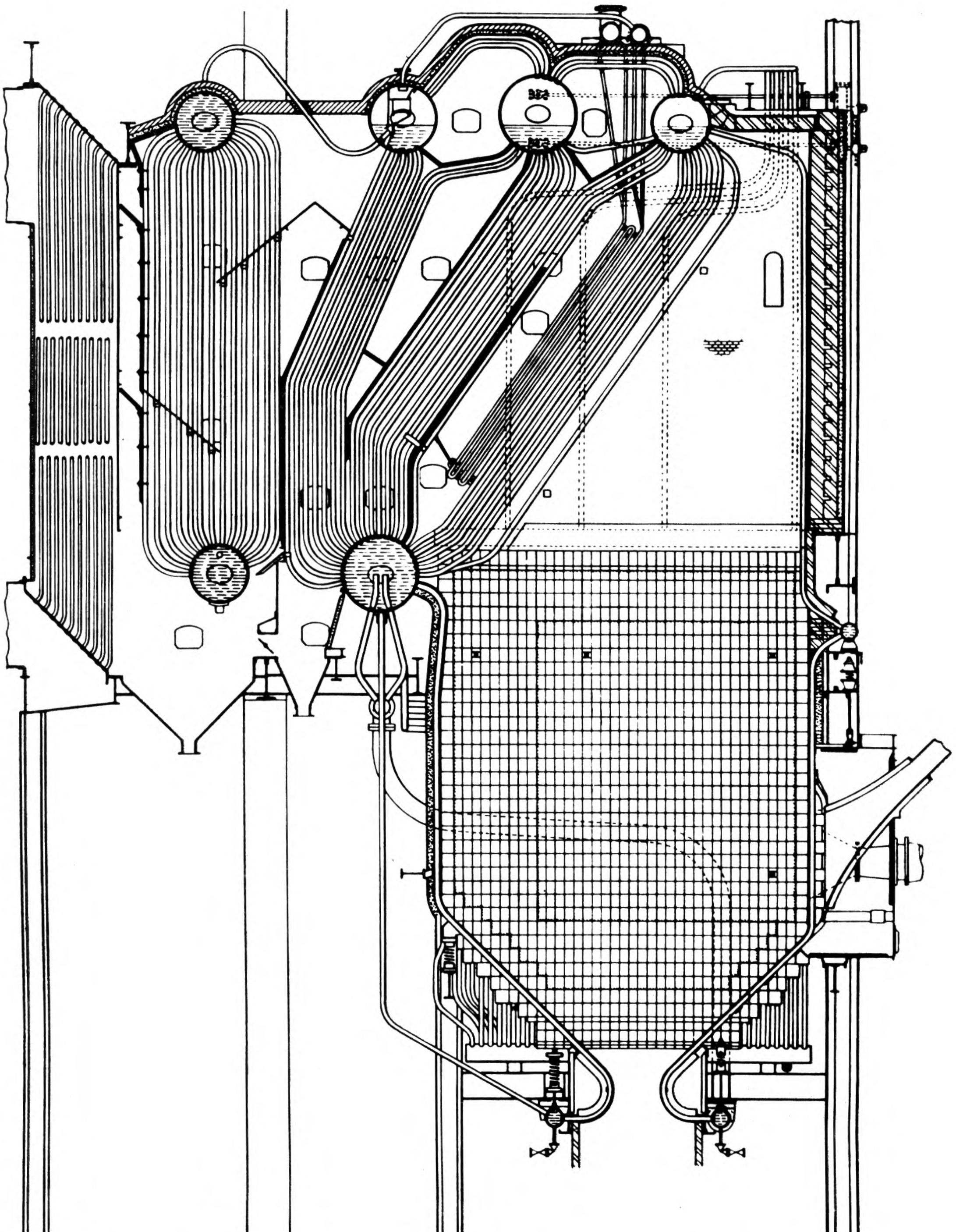

Fig. 6 Stirling Boiler Installed at Fairfield in 1932

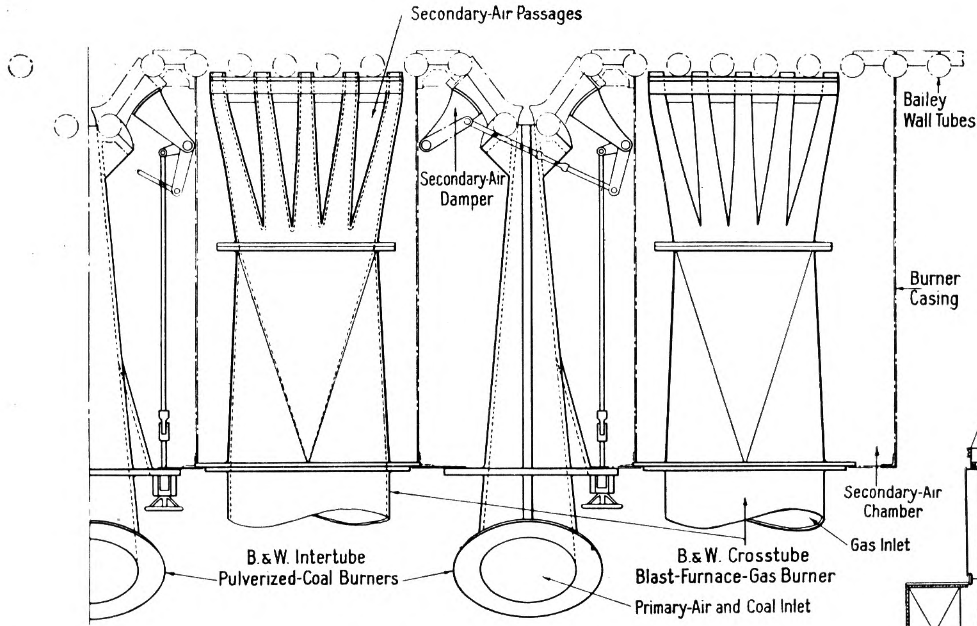

Fig. 7 (Left) Combination Burner for Pulverized Coal and Blast-Furnace Gas

Factors Delaying Use of High Pressures in Steel Mills

There are four factors which have caused some steel men to hesitate in using high-pressure steam:

The first is the matter of circulation. Many engineers believe there is a definite limit to the steam pressure permissible with natural circulation. Fortunately, the company, with which the author is connected, made a careful study of this problem about 10 years ago, which proved that natural circulation was entirely satisfactory with proper design for pressures up to at least 2600 psi.

This statement presupposes a definite separation of water and steam in the boiler drum, so that the water in the downtakes will be of normal density. As the steam pressure increases, Fig. 1, the steam and water densities approach each other, making separation of steam and water more difficult.

Not only is it necessary to remove the water from the steam but for proper circulation conditions, it is also necessary to remove practically all the steam from the water, so as to be able to maintain the desired circulation head in the downcomers. This has been satisfactorily accomplished by the present construction of cyclones followed by steam scrubbers, as shown in Figs. 2 and 3. In these cyclones, the force available for separating the steam and water may be 5 to 10 times the force of gravity. For more detailed information on this construction and results obtained, reference is made to a paper by M. D. Baker (4).

After the steam and water are separated in the drum, it yet remains for the designer to take care of such conditions as are shown by Fig. 4, which shows an experimental drum, looking down at the entrance to a downcomer. It will be noted that, if proper means are not taken to prevent its formation, a vortex of no mean proportions may result, in which case, the circulation head on a boiler unit may be tremendously decreased. It is perhaps not generally realized that, in a boiler unit furnishing 400,000 lb of steam per hr, there may be passing through the drum from 4,000,000 to 8,000,000 lb of water per hr.

The second factor which arises in connection with the use of high-pressure steam is that of feedwater. Central stations use condensate, whereas, steel plants generally use all make-up water. This is a very definite difference and one which requires every consideration. It may be stated that today feedwater can be quite satisfactorily treated for high-pressure steam operations, with the one question open, which chemists call by various names,

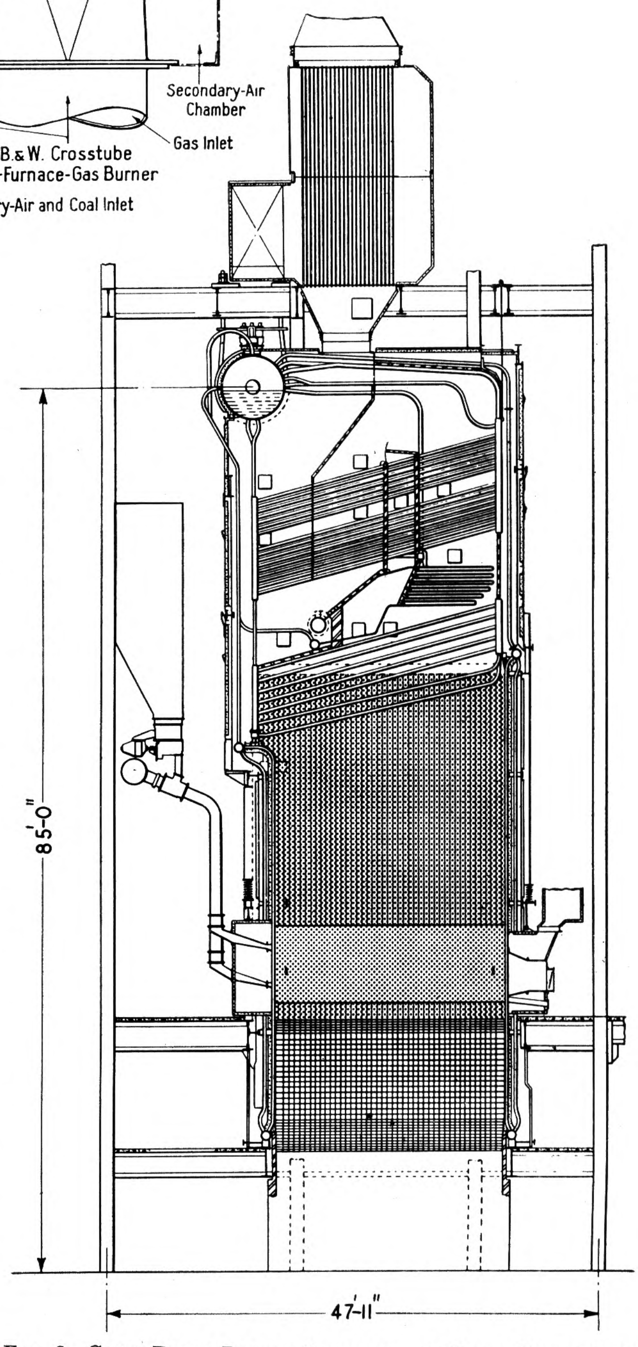

Fig. 8 Cross-Drum Boiler Installed at Eliza Furnace of Jones & Loughlin Steel Company

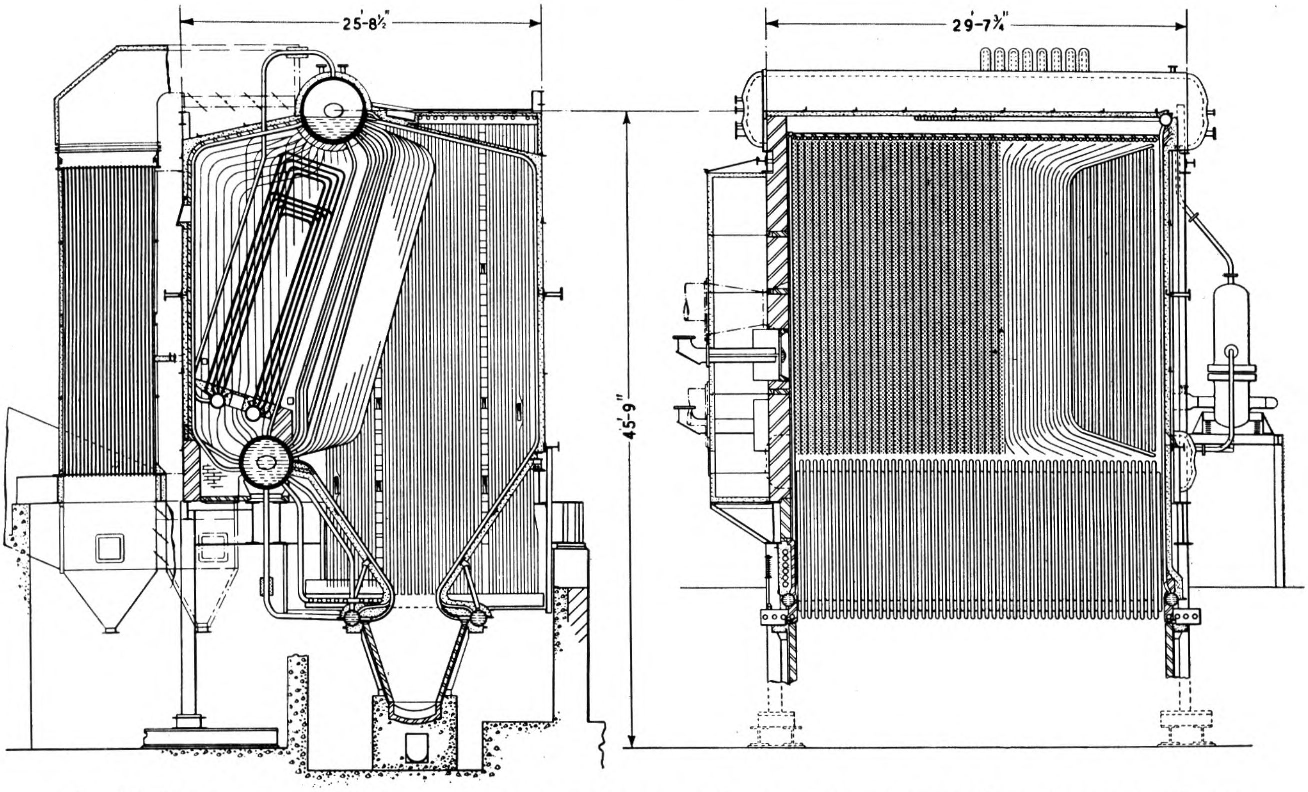

Fig. 10 Integral-Furnace Boiler for Firing With Pulverized Coal and Blast-Furnace Gas; Youngstown Sheet and Tube Company

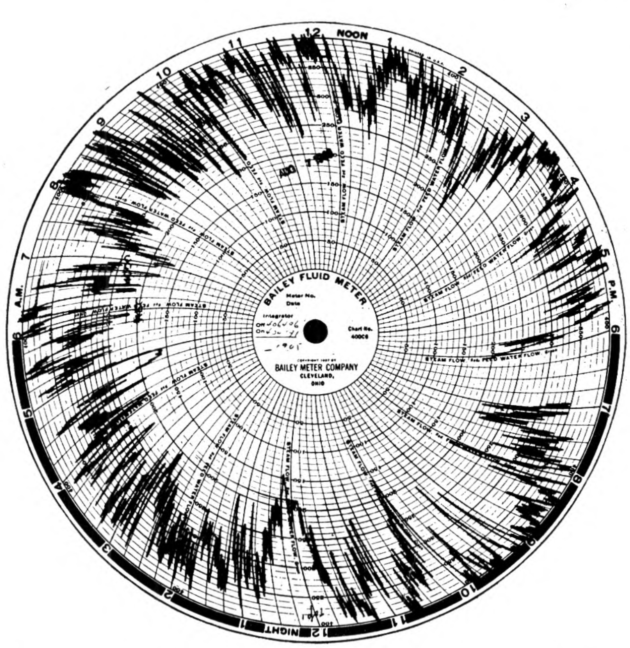

Fig. 9 Steam-Flow Chart From Boiler at Eliza Furnace

but which, fundamentally, is the problem of silica removal. Without going into details, it may be stated that the solubility of silica is reduced at higher temperatures, so that less concentration of silica is permissible at high pressures than at low. If, therefore, the water available has a natural silica content, scale may form at high pressures. Much work has been accomplished in silica removal, as evidenced by the results obtained at the Baton Rouge plant of Gulf States Utilities. These results were presented in a paper by M. C. Schwartz (5).

New and successful efforts are being made along other lines toward the elimination of silica. The Weirton Steel Company installation, operating at 850 psi pressure, using Ohio River water for feedwater, is evidence of successful operation at that pressure with practically 100 per cent make-up water. This installation is well described in a paper by H. G. Strassburger (6).

Priming, which might be accelerated by feedwater conditions, has been practically eliminated by the cyclones previously described. Where they are used, water at full density is maintained in the drum.

The third factor is one of mechanics. Fig. 5 shows a high-pressure welded drum in course of construction. There are no unsolved mechanical problems involved in meeting the requirements in equipment for using high-pressure steam. So long as steelmakers can make good steel of 70,000 psi tensile strength, boilermakers will make satisfactory boilers for any pressure which may be required. There is little difference, if any, in the reliability of the construction of a 1500-psi boiler and one built for 150 psi.

The fourth point which sometimes is cited in connection with high pressures is possible operating difficulties. In regard to this, the final paragraph of Mr. Cutler's paper (1) of 1931, is quoted:

"All of the operators were taken from other plants of the Tennessee Company, and they had very little if any experience with steam pressures over 150 psi, superheaters, turbines, condensers, pulverized coal, or automatic combustion control; however, the operating difficulties encountered have been but nominal."

Problems Encountered in Raising Steam Temperatures

The increase of steam temperature from 650 to 850 F in steel

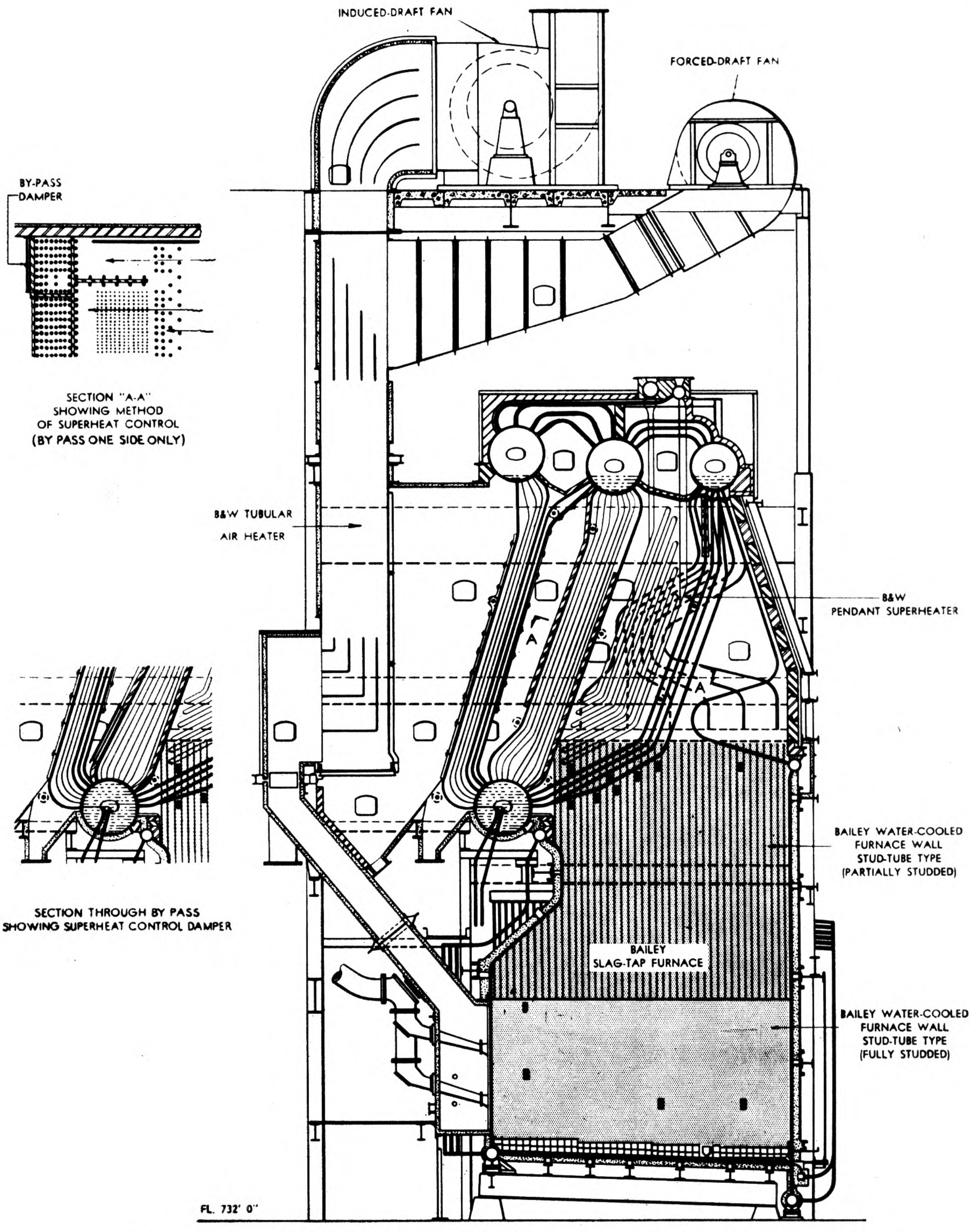

Fig. 11 Stirling Boiler With Slag-Tap Furnace; Weirton Steel Company

plants, and to 950 F in central stations, required the solution of three problems:

1 In selecting materials for superheated steam, carbon steels were unsatisfactory, due to oxidation and low creep strength. Partly as a result of the pioneer work done in the oil industry alloy metals have been developed, so that today eleven different chromium alloys are available for tubes, the chromium content ranging from 1 to 27 per cent. Therefore, materials are obtainable which are fully capable of meeting present-day requirements from the standpoint of both oxidation and strength.

2 As for tightness, the rolled joint has maintained its position with the increase in pressure, but has been found deficient with increase in temperature. Fortunately, welding was developed as required and, through its use, high-temperature construction gives less trouble from leakage than was common with constructions employed with lower temperatures. To permit the com-

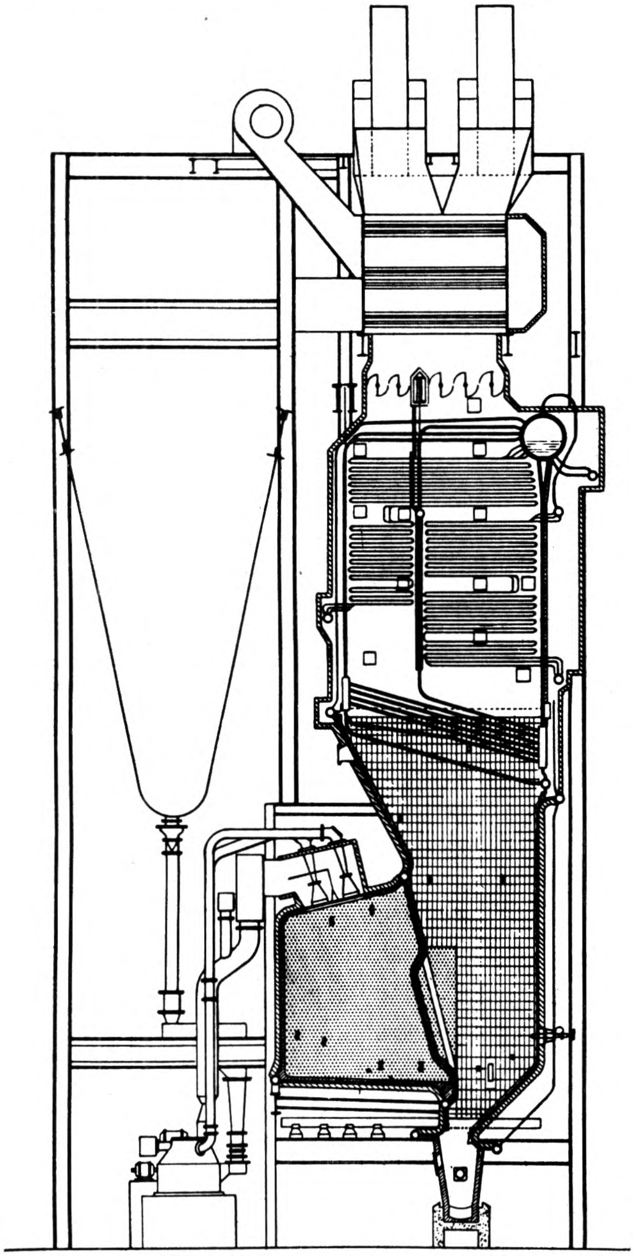

Fig. 12 High-Head Boilers in Service at Rivesville Station, Monongahela & West Penn Power Company

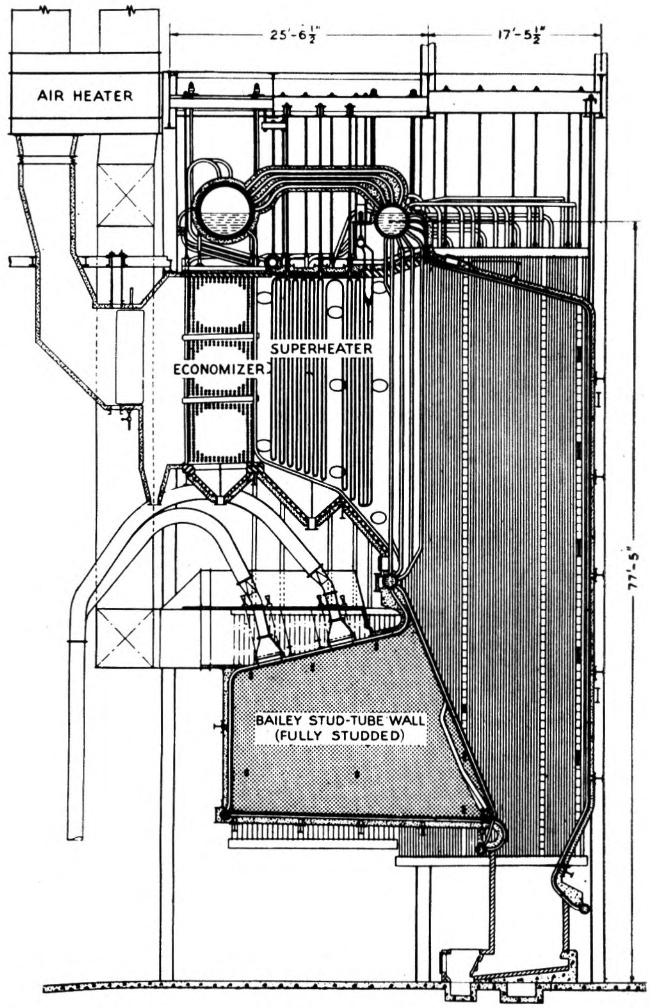

Fig. 13 Radiant Boiler in Service at Essex Generating Station, Public Service Electric & Gas Company, New Jersey

mercial application of welding to superheaters, material changes in their construction have been developed.

3 In conjunction with high pressure, Table 2, high steam temperature has changed the percentage of total heat required for superheat to such an extent as to have brought about new designs of boilers. The necessity of maintaining constant superheat over a considerable range in rating has accentuated this condition.

Present-Day Furnace Practice

With blast-furnace gas (particularly if clean and hot), few problems exist in present-day furnaces with properly designed burners, unless perhaps at low ratings with too cold a furnace. The reasons for this are that the flame temperature with this fuel is so low as to cause negligible slagging under any rates of combustion contemplated; and, since pulverized-fuel firing in the same furnace is required in steel mills, the limitations set by this latter fuel determine furnace design.

What, then, are the limitations of pulverized-fuel furnaces? As stated earlier, Btu per cubic foot is not the limitation, and for the reason that this expression means only the amount of fuel burned in a given space in a given time. The industry is still in the elementary class on this particular problem. This term has been used by many as an expression of limitation of furnace capacity; and, in some cases, this has been approximately correct, but only because other things happened to occur in proportion.

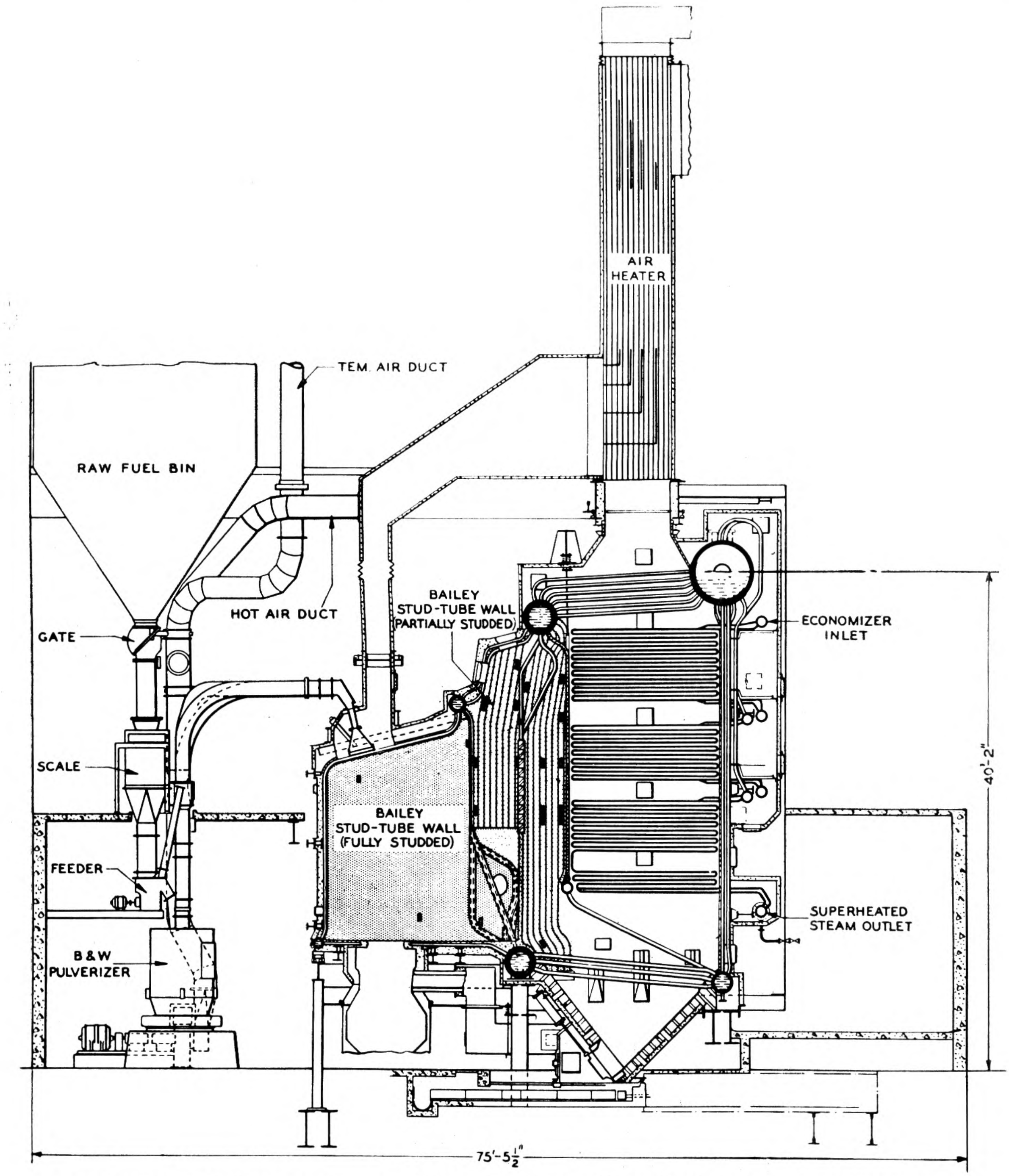

FIG. 14 OPEN-PASS BOILER IN SERVICE AT WEST END STATION, UNION GAS & ELECTRIC COMPANY, CINCINNATI

The real limitation of pulverized-fuel furnaces is slag—nothing else. Two methods are available to overcome this difficulty; (*a*) The elimination of the slag in the furnace gases or, (*b*) the reduction of the temperature of gases leaving the furnace.

While not accomplishing the first result completely, real progress has been made with slag-tap furnaces, from which 50 per cent of the ash in the coal may be removed in molten form.

The second method is to cool the gases to a temperature below the ash-fusion temperature before they leave the furnace. To do this requires a certain amount of cooling surface around the furnace and, hence, this result is measured by the expression, "Btu available per square foot of cooling surface." This sounds like a simple remedy, but it is perhaps not so simple. The boundary surface of a cube increases as the square of its dimension, while the volume increases as the cube; hence, with large-size units it becomes difficult to obtain sufficient surface without excessive volume. Again, if furnaces are made too cold, combustion rates decrease, and certain lighting and low-rating problems are encountered. Finally, with high superheat and high pressure, particularly with constant superheat maintained at low ratings, a definitely low limit of gas temperature exists, below which it is impractical commercially to obtain the superheat required.

In furnaces which have been installed by the author's company, progress has been made by combining the two principles discussed, as will be noted from the illustrations of various units, representing boilers of comparatively recent construction.

Blast-Furnace Boiler Installations

Fig. 6 shows a four-drum Stirling boiler, installed at Fairfield in 1932. This unit was designed for 300,000 lb of steam per hr with coal, and 200,000 lb per hr with blast-furnace gas. It is similar to the units described in Mr. Cutler's paper (1), but has two additional features; i.e., complete water cooling of the furnace, and a combination of coal and blast-furnace-gas burners, as shown in Fig. 7. This system has proved quite successful in combination firing. The capacities mentioned for both fuels have been exceeded.

Fig. 8 shows a unit installed at the same time as the new strip mill at the Eliza furnace of the Jones & Loughlin Steel Company, for a maximum capacity of 400,000 lb of steam per hr at 500 psi pressure. Fig. 8 is shown particularly for two reasons; i.e., it illustrates a gas by-pass around the superheater for the purpose of obtaining constant superheat at various ratings; and because this unit exemplifies what can be done to handle successfully the extremely variable load so often met with in steel-plant operations.

Fig. 9 is a steam chart from the unit, Fig. 8; variations in steam flow from 200,000 to 400,000 lb per hr will be noted.

This unit was installed in the same powerhouse as four existing boilers which were carrying as much of the swing in load as they could successfully handle. The new unit was required to accommodate the type of swings shown by the chart, Fig. 9. This was accomplished by installing, in the steam line of this boiler, a butterfly valve which opens wide as the load demand increases, thus obtaining the required steam flow which, in many cases, increases at too fast a rate to be obtained by change in fuel rate.

Fig. 10 is a sectional view of an integral furnace boiler unit in course of construction for the Youngstown Sheet & Tube Company, Chicago. Two such units have been in service for some time using pulverized fuel alone. This unit is to be fired with both pulverized coal and blast-furnace gas, at a maximum capacity of 170,000 lb of steam per hr and a steam pressure of 825 psi.

Fig. 11 shows a side view of the units at the Weirton Steel Company, which were described by J. H. Strassburger (6). This is a typical application of a slag-tap furnace to a four-drum Stirling boiler. These units, operating at 850 psi, 850 F steam temperature, and at a maximum capacity of 400,000 lb per hr, are showing highly reliable operating results. Another unit of the same type now being installed will combine both blast-furnace-gas and pulverized-coal firing.

Fig. 12 illustrates two units in service at the Rivesville Station of the Monongahela & West Penn Power Company. These units were designed for 350,000 lb of steam per hr at 1250 psi, and have operated at materially greater capacities. As shown by the illustration, the furnace is of the two-stage slag-tap type, designed for burning coal containing ash of 2100 F fusion temperature. The heat release at full load in the entire furnace is 40,000 Btu per cu ft. The heat release in Btu per sq ft of equivalent cold surface is approximately 150,000.

Fig. 13 shows a side elevation of the radiant-heat boilers operating at the Essex Station of the Public Service Company of New Jersey. They are designed for a pressure of 1475 psi and a steam temperature of 950 F, with a capacity of 600,000 lb of steam per hr. These units also have two-stage slag-tap furnaces. This design permits the combination of a high-temperature primary furnace, wherein are obtained high rates of combustion per cubic foot and a high value of heat available per square foot

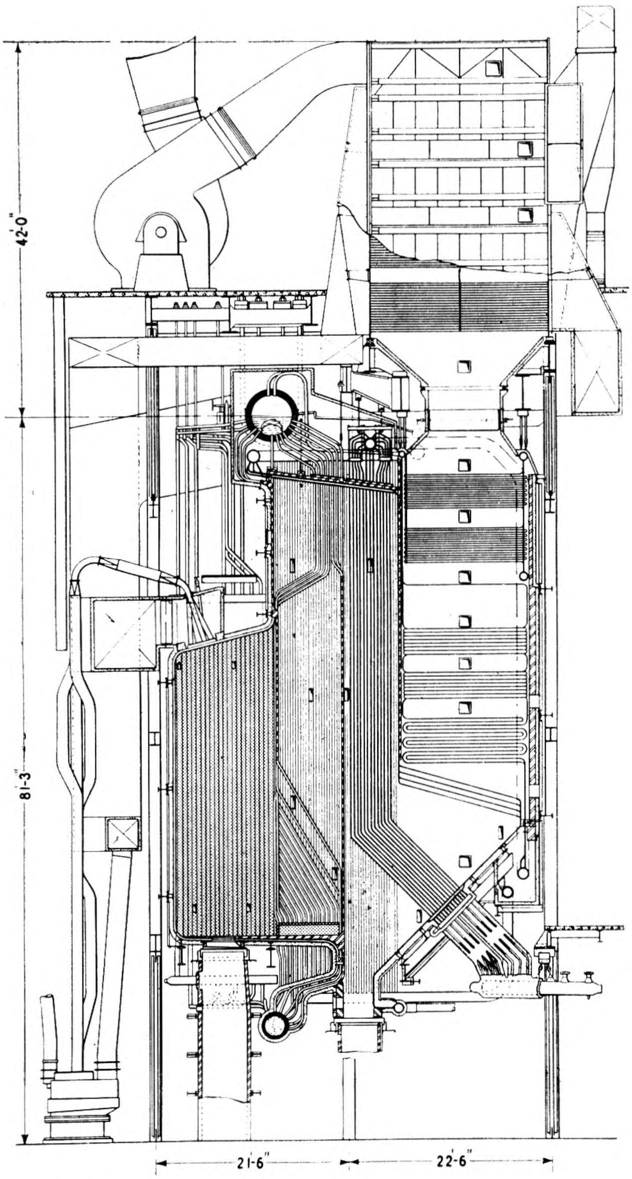

Fig. 15 Open-Pass Boiler to Operate at 2600 Psi Pressure; Twin Branch Station, American Gas & Electric Company

of surface, and a secondary furnace, furnishing a high cooling factor for the gases, which have already shed a considerable percentage of the ash in the fuel.

Fig. 14 is a side elevation of one of the units in operation at the West End Station of the Union Gas & Electric Company, Cincinnati. They were designed to fit into an existing building, and have a maximum steam production of 350,000 lb per hr at 1275 psi pressure and 910 F steam temperature. The primary furnace is of the same general type as that shown in previous illustrations and from which slag is tapped continuously through the center of the floor into a sluice tank. The release in the primary furnace is of the order of 82,000 Btu per cu ft.

To the rear of the primary furnace are two gas passages surrounded by partially studded tubes, providing for a gas travel of about 70 ft between the burners and the heating surface of the superheater and economizer. With this construction, a further step was made in the effort to obtain high furnace temperature for high combustion efficiency and the removal of a greater percentage of ash in the primary furnace. By the complete shielding of radiation from the primary furnace to the heating surface of the boiler proper, and by increasing the convection-transfer rate of the boundary tubes in the latter stages of the furnace, a lower temperature leaving the furnace is obtained. With this arrangement, the combustion rate up to the convection surface is approximately 40,000 Btu per cu ft. The Btu per sq ft of cold surface is 110,000.

Fig. 15 is a side elevation of the boiler now being installed at the Twin Branch Station of the American Gas & Electric Company, to operate at 2600 psi, 940 F steam temperature, and at a capacity of 550,000 lb of steam per hr. It is to be noted that the general form of this unit, which is for operation at the highest pressure for which natural-circulation boilers have been built, follows the same general design as the units which are shown in Fig. 14.

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Discussion

F. E. Norton.[4] In the following discussion it will be assumed that the steel mill includes blast furnaces and other sources of so-called waste fuels.

The paper illustrates eight boilers of which four are suitable for general steel-mill practice, utilizing coal or gas fuel, or a combination of fuels. The other four types are not, in general, suited to gas fuel or combined fuels.

In the paper, considerable emphasis is placed on the rate of heat release in boiler furnaces as it affects slag formation and consequent loss of efficiency. The author states: "The real limitation of pulverized-fuel furnaces is slag—nothing else." The writer agrees with this statement but would add that, for blast-furnace gas, the limitation is "mud" from the flue dust and ash in wet-washed gas.

The furnace gas may bring 0.03 to 0.05 grain of dirt per cu ft into the boiler furnace, but the effect on operation and efficiency depends upon whether or not the gas is wet or dry and upon the nature of the dust. The real criterion is not the Btu realized by the combustion of wet or dry gas, but lies in the effective use of the heat in making steam (i.e., dirty boilers versus clean boilers).

There is disagreement among operators of blast-furnace plants as to the relative merits of dry or wet gas in boilers. The disagreement may be due to the reaction of the blast-furnace department as to stove gas. Modern stoves require clean gas, and wet primary washing is the rule, while secondary wet electric precipitators are coming into general use for stove gas.

It is quite usual to wet-wash all the furnace gas and to divert a portion through precipitators for the stoves. This leaves the boilerhouse with wet gas, which quite often is very dirty and worse than the dry gas. The resulting effect on boiler steam production is very bad, especially if there are shots of dry dirty gas, as is often the case. The presence of any coal ash makes the condition yet worse. The mud deposits on boiler tubes and burners and soot blowers are incapable of removing it. Since the blast furnace, quite often, is given credit for the full coal equivalent of the gas, boiler troubles do not directly affect the furnace costs; indirectly, however, steam costs go up for blowing the furnace.

A further discussion of waste-gas fuel for boilers will bring out some essential features of boiler design and seems justified in view of the unsatisfactory performance of a number of plants using partially cleaned wet gas.

The use of dry dirty gas at stoves is not advisable but such gas may be used at the boilerhouse.

It is quite apparent that a satisfactory gas supply at stoves may not assure good results at boilers.

Of course, the obvious procedure would be to clean all the furnace gas with precipitators, at a very considerable investment cost. The dry gas would be cheaper. The alternative is to use dry dirty gas at boilers and wet-washed and precipitated gas at stoves.

An example of the use of dry dirty gas in two boilers, each rated at 165,000 lb of steam per hr, appears in a current article.[5] A third boiler is being installed at the Sydney (Nova Scotia) plant of the Dominion Steel and Coal Company. An efficiency of 80 per cent for the gas is reported. The performance of the first two boilers has been satisfactory. The writer's remarks are prompted by a comparison of these with similar boilers using wet-washed gas which are not at all satisfactory because of dirty gas.

The difficulties experienced with mud have several aspects. It is not likely that the operators will agree when experience at several plants is compared. For instance, the writer is familiar with a boiler plant in which dry fly ash from coke breeze is used to remove the deposit on tubes. The boiler output is satisfactory using wet gas.

Pulverized coal may or may not be effective in removing such deposits. If the coal is coarsely pulverized, and partly burned, the result may be a tar, flue-dust, coal-ash compound, which looks like slag and is difficult to remove from the boiler. Circumstances alter the effect.

The difficulty with wet gas is that the mud deposited from a rush of dirty gas will remain on the tubes for a long time. The gas may be perfectly clean for hours, but a few minutes of dirty gas will cut down boiler output for days. This aspect of the dirt problem leads to the statement that dry dirty gas is to be preferred to wet gas with more than say 0.05 grain of dirt per cu ft. This requires virtually perfect washing and precipitation; furnace slips or accidental gusts of dirty gas must be eliminated.

If dry dirty gas is to be used, it is essential that the boiler and furnace be arranged so that soot blowers can clean the tubes effectively and the dust can be removed from the furnace without loss of service or undue labor.

It is surprising that so few modern boiler plants are arranged to use dry gas. Boiler manufacturers should properly be interested in dust removal by mechanical means. The amount of money involved may be very large as illustrated by the following example which might hold for a plant with four blast furnaces.

Suppose a total of 1500 tons per day of coal equivalent is to be used in a boiler; say, 1000 tons by furnace gas and 500 tons by actual coal. The efficiency of the boilers with wet-washed dirty gas may fall to 60 per cent. With clean gas the efficiency may be 80 per cent. The amount of steam made by the clean gas would be $^1/_3$ greater than with the same amount of dirty gas.

[4] Special Engineer, Youngstown Sheet & Tube Company, Youngstown, Ohio

[5] "Hot-Blast Gas for Dominion Steel," by W. S. Wilson, *Power*, vol. 84, 1940 p. 733

The dirty boilers cannot give good results on coal and the efficiency of the entire plant would be lowered. The loss would all come on coal and would amount to 375 tons of coal per day for the same steam output.

With coal at $2.50 per ton, this amounts to $937.50 per day. Should the efficiency of dirty boilers be 65 per cent and the fuel consumption remain at 1500 tons per day, the loss would be 281 tons at $2.50 or say $702.50, as compared with clean boilers for the same amount of fuel. These figures are assumed from actual experience but are not taken from any actual records and do not represent actual operating data.

In such a plant, the loss of $700 per day amounts to $21,000 per month, as a result of using wet dirty gas. Per year, the loss may be $250,000. For this reason, the boiler design should contemplate the use of dry gas, unless capital is available for precipitators.

In view of such facts, it is difficult to understand why so few plants use dry dirty gas at boilers and even more so why plants use a mixture of dirty wet gas and dry gas. The boilers usually get the blame for poor efficiency and, in some cases, large investments have been made for economizers, air heaters, and other expensive devices to raise the efficiency, while the dirty gas wipes out all gain and adds the cost of primary washing.

In some cases the loss by poor efficiency may result in an increase of purchased power. If 1400 kw can be generated per ton of coal, which if purchased may cost 1 cent per kwhr, the equivalent coal cost would be $14 per ton. The actual figure may be as low as $7 per ton of coal, depending upon circumstances and upon power schedules.

It will be noted that boilers for dry dirty gas must be arranged for easy removal of dust. The proper arrangement of gas and coal burners is also of great importance and may have a bearing on the flue-dust problem. The air for combustion of furnace gas can be kept in contact with fuel until combustion is complete and 20 to 22 per cent of CO_2 may be reached in the burned gas.

On the other hand, the coal does not readily carry the air for combustion along with the flame. The products of coal combustion may be 14 to 18 per cent CO_2 and, if coal is discharged into an atmosphere of burned furnace gas, the combustion may be delayed or even stopped by the 22 per cent CO_2 atmosphere.

In this connection, Fig. 6 of the paper (Fairfield boiler) is of interest, as showing the coal burner pointed down. In this case the coal flame should separate from the gas flame. The boilers at Sydney, Nova Scotia, have similar coal burners.

The writer recalls an early application of pulverized coal to waste-gas boilers at another plant in Birmingham which gave great difficulty in starting the coal burners. These were placed in the roof of the furnace so that the flame was compelled to mix with burned furnace gas. The result was a very dirty furnace.

Fig. 10 of the paper shows parallel coal-and-gas firing with intertube nonturbulent gas burners. This boiler is not designed for normal use of coal. It would be well suited to the use of unwashed dry gas. The use of a desuperheater in the boiler circulation system is also to be noted.

The operation of waste-furnace-gas boilers apparently is a simple problem and casual inspection of such a plant would seem to confirm this: With perfectly clean gas and a steady load, corresponding to the amount of steam the gas can make, it is simple so long as the boilers are clean.

When a widely varying load must be carried with a constant supply of gas, it is evident that the fluctuations must be carried on coal or other fuel. The problem of regulation becomes serious if combined coal-and-gas firing is used. The cases the writer has in mind involve a swing of say 30 to 60 per cent of maximum capacity of the coal-fired boiler; the period of swing may be $1^1/_2$ min or less. It will be realized that rapid fuel and draft control must be provided to meet the load conditions. An airtight furnace is demanded, since a leaky furnace means excessive loss at the stack, unless the fans and dampers give balanced furnace pressure. For this reason the coal should be burned on boilers which are adapted to coal and widely swinging loads. The base load could be carried by gas in order to obtain full output from the boiler.

A comparison of designs in use leads to the conclusion that practice has not been crystallized into standards which might be expected in view of the large number of installations and the amount of money involved in fuel costs.

The $250,000 differential mentioned would seem to warrant careful consideration of the conditions at the boilers when the gas-supply system is being laid out. The requirements of the stoves should not be the sole factor in final design.

J. H. Strassburger.[6] Since July, 1936, The Weirton Steel Company has been operating two boilers, each having a capacity of 400,000 lb of steam per hr at 850 psi working pressure and 820 F total temperature. These boilers have been supplying steam to a 10,000-kw topping-turbine generator as well as steam through a reducing plant for general power and process use. Operating experience with this boiler equipment has been highly satisfactory. The actual handling of boilers at this pressure has been no more difficult for our personnel than the handling of boilers at lower pressures.

During the period mentioned, these boilers have been supplied with feedwater conditioned by the hot-process system, utilizing 100 per cent raw water to the softening plant. During the first 2 years of operation, these boilers had an availability of 89 per cent, during the next $1^1/_2$ years the availability averaged 91 per cent and, during the year 1940, these boilers will have had an availability of 96 per cent. During the last 4 years every shutdown on each of these boilers has been a scheduled one for the purpose of periodic inspection and cleanup.

We are now completing the installation of a third boiler and a second generator. The third boiler is for the purpose of replacing low-pressure equipment which has completed its useful life. From our experience with this installation, we would not hesitate to operate one generator on one boiler.

We have experienced no difficulty in the operation of the topping generator. The 10,000-kw machine is loaded so that month in and month out it produces approximately 11,000 kw for every hour of the month, with peaks up to 13,000 kw depending upon back-pressure conditions. The availability of this generator, based on an inspection every 2 years and on washing out periodically every 3 months, amounts to 98 per cent.

We believe that in the next few years an extended use of high-pressure boilers with companion generating equipment will be widely adopted by the steel industry, and should create large savings with no serious operating difficulties.

E. J. Kohn.[7] Mr. Kerr has presented an interesting paper which has a particular appeal to the writer because he sets forth the advantages of high-pressure steam as applied to a local plant.

Referring to the unit "kw per ton of product from surplus blast-furnace gas" as shown in Mr. Kerr's article, Table 1, the writer refers to Mr. Cutler's article,[8] which presents an over-all comparison typical of 1922 and 1930, including blast furnaces, blast-furnace stoves, boilers, and steam users. The net result is an increase for the 1930 period in the steam available for electric generation.

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[6] Combustion Engineer, The Weirton Steel Co., Weirton, W. Va.

[7] Chief, Bureau of Steam Engineering, Tennessee Coal, Iron and Railroad Company, Ensley, Ala. Mem. A.S.M.E.

[8] See reference 1 of Bibliography to paper.

Extending this comparison, Mr. Kerr justly emphasizes the advantages resulting from more efficient boiler design and operations, but fails to take into account the improvements incident to blast-furnace operation.

When one considers the improvements of blast-furnace practice, namely, ore conditioning, sintering, and possible air conditioning of blast, a lower coke rate per ton of iron would be in effect, which in turn would give a lower "heat in top gas." The net result would be less fuel per ton of iron available to the boilers which would materially lower the "kw per ton of product." Mr. Kerr's subject "Steam Generation in Steel Mills" would show up adversely if he considers improvements in blast-furnace practice and adheres to the unit "kw per ton of product from surplus gas."

For comparative results, conforming to the subject, Mr. Kerr should limit his comparison to the boilers and assume the heat to be the same in all cases. Mr. Kerr gives four reasons why steel men hesitate in the use of high-pressure steam: (1) circulation, (2) feedwater, (3) mechanics, (4) possible operating difficulties. As pointed out by Mr. Kerr, these problems have been overcome so we must look further for a possible reason for the hesitation.

Under feedwater, Mr. Kerr calls attention to the fact that central stations use condensate whereas steel plants generally use make-up water for boiler feed. From the latter one would infer that the steam was being used for mill operation. Relatively low-pressure steam, 200 lb and under, is required in the ordinary steel-plant operation, which of course, precludes the economical generation of steam at the relatively high-pressure range.

On the other hand if the steam is used for turboblower operation and electric-power generation, any refinement that could be economically applied to central stations could be applied in the same measure to steel plants. As evidence Mr. Kerr calls attention to the use of steam up to 900 lb in some steel plants. The entire question is one of economic balance between the cost of equipment installed as compared with the operating expense. Locally the relatively low cost of fuel would be largely a determining factor.

In brief the steel plants like the central stations will take the advantages to be derived from high steam pressure when it is economically possible to do so.

Author's Closure

The author appreciates the discussions, as they have added value to the paper.

Mr. Kohn has brought out the effect of improvements in blast-furnace operation and the cost of fuel on the general economic problem of steam generation in steel mills. It would be very difficult to cover the many variations in blast-furnace operation in a paper of this sort and, therefore, the author took as a base the blast-furnace data as furnished in Mr. Cutler's paper of 1932.

Undoubtedly the cost of fuel will appreciably affect the steam pressure and temperature conditions for most economical operation at any plant.

Mr. Strassburger has been good enough to give the actual experience which he has had over the last four years and which demonstrates the reliability of 850 lb pressure, 850 deg steam operation in steel-mill practice with 100 per cent make-up water.

Mr. Norton's discussion is a timely one as it brings out the problem which many of us have had to face when burning washed blast-furnace gas, namely, the tenacious scale formation on the boiler tubes. The steady increase in stack temperature when burning washed gas has made it necessary periodically to wash boilers on the outside as soot blowers will not remove this type of scale. Some experience indicates that periodic burning of pulverized coal in the same unit minimizes this trouble, as apparently the fine ash in the flue gas has sufficient abrasive effect to cut down this scale. With dirty gas the problem is the handling of large quantities of dirt in the unit and induced-draft fans. However, with dirty gas the stack temperature does not materially increase with reasonable use of soot blowers.

As indicated in the paper, the proper cleaning of blast-furnace gas results in long periods of operation of steam-generating units at maximum efficiency, and hence the desirability of such preparation of the gas where economically possible.

Coal Resources of Washington

By JOSEPH DANIELS,[1] SEATTLE, WASH.

Coal is the principal mineral fuel resource of Washington, overshadowed at the present time by the water-power developments at Bonneville and Grand Coulee. No petroleum is produced commercially, but there is a small production of natural gas. Wood, forest and mill, obtained largely as waste products of the lumber industry, is used extensively for heat, power, and industrial purposes. This paper discusses the location of coal regions, the characteristics of Washington coals, mining operations, transportation, cost of coal, and other factors, giving a comprehensive idea of the production and use of this fuel in the Pacific Northwest.

TO MANY engineers the State of Washington is best known for its water power. They have heard of its scenery, its timber, excellent harbors, fishing, and possibly something of its metallic- and industrial-mineral resources. The fuel resources are not so well known or understood. Facts concerning these resources, with particular reference to Washington coals, will, therefore, be presented in this paper.

Fuels Other Than Coal

In the State of Washington, no petroleum has been produced commercially, although much prospecting, mainly of the wildcat variety, has been conducted in various parts of the state. However, to date the results have been negative. A pipe line into Spokane from Montana and tanker transportation from California ports to tidewater points deliver the various fuel oils required to strategic distributing centers. California fuel-oil shipments into Washington averaged slightly over 9,000,000 bbl annually from 1930 to 1938.

A small production of natural gas, predominantly methane, of approximately 900-Btu value is obtained from a series of shallow wells in the Rattlesnake Hills field of Benton County, 18 miles northeast of Prosser. Distribution began in 1929; the gas is sold in seven cities in the Yakima Valley. The reported production in 1938 was 121,000,000 cu ft, of which 111,000,000 were sold. The natural gas has been mixed with butane-air gas before distribution. It does not appear that natural gas from outside sources can be economically piped and distributed through the state, although the matter has been discussed and some plans were once projected for lines from out-of-state fields. Brief mention should be made here that manufactured and liquefied petroleum gases are available to consumers in 29 cities other than those served by natural gas.

Forest wood and waste in the form of millwood, slabs, edgings, sawdust, and hogged fuel must be listed in any inventory of fuel resources. The exact volume or tonnage used is not known, but it plays an important part in supplying heat and power in domestic and industrial uses everywhere throughout the state. The availability and relatively low price make wood, particularly sawdust and hogged fuel, a very attractive fuel to many consumers, especially in western Washington.

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

Location of Coal Regions

Topographically the state can be partitioned into five divisions, represented by the Olympic Mountains or Coast Range on the west, the Puget Sound Basin, the Cascade Range, the Okanogan Highlands of northeastern Washington, and the Columbia Plateau area, making up the greater portion of the eastern and southeastern part of the state. The economically significant coal region lies along the western flank of the Cascades from the Canadian boundary on the north to the Columbia River on the south in Whatcom, Skagit, Snohomish, King, Pierce, Thurston, Lewis, and Cowlitz Counties, and on the eastern slope in a limited, although important, outlier in Kittitas County. This major coal region is served by four railroad systems, is close to water transportation, and is tributary to an excellent system of roads and highways which permits truck delivery to practically every part of the state. Most of the deposits lie within a radius of 100 miles of Seattle.

The United States Geological Survey, using certain measuring sticks, estimated the content of coal in 1917, and again in 1925, and reported a total of 63,877,000,000 tons of coal. Similar estimates were made for other portions of the country. For purposes of comparison with other districts on the Pacific Coast, a summary of the resources of Alaska, British Columbia, Oregon, and California is presented in Table 1.

TABLE 1 SUMMARY OF ESTIMATED RESOURCES OF COAL ON PACIFIC COAST

(Millions of tons)

Geographical division and rank of coal	Actual reserve	Probable reserve
Alaska[a]	Metric tons	
Anthracite and semianthracite	1931	.....
Semibituminous and bituminous	1369	.....
Subbituminous	3681	.....
Lignite	12612	.....
Total	19593	.....
British Columbia[a]	Metric tons	Metric tons
Anthracite and semianthracite	7	1343
Bituminous	23764	43925
Subbituminous or lignite	60	5136
		1800
Total	23831	52204
Washington[b]	Net tons	
Anthracite and semianthracite	23	.....
Bituminous	11412	.....
Subbituminous	52442	.....
Total	63877	.....
Oregon[b]	Net tons	
Bituminous	3000	.....
Subbituminous	7000	.....
Total	10000	.....
California[b]	Net tons	——
Bituminous	27	.....
Subbituminous	16	.....
Total	43	.....

[a] Refer to Bibliography (1).[2]
[b] Refer to Bibliography (2).

All of these estimates have been questioned because they were based on interpretations of broad geological structures, in some cases on inadequate data, and because they did not consider the limitations imposed by operating and economic factors which govern the workability of deposits. In a broad way, however,

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

since the same yardstick was used everywhere, the estimates give a relative or comparative picture of the magnitude of the resources. The figures appear to indicate that Washington originally contained the largest resource of tonnage in the Pacific Coast strip. Using actual reserve tonnages as a basis, Washington had 56.4 per cent of the total, British Columbia 21 per cent, Alaska 13.7 per cent, Oregon 8.8 per cent, and California 0.1 per cent. The significant position of Washington with respect to future supplies of solid fuel is indicated by this analysis. The figures quoted, however, cannot be used for estimating present commercial reserves or probable life of the various fields or districts within the major areas.

Coal Mining in Washington

Coal mining began in the state about 1850; shipments to adjacent territory were reported in 1860. No records of production are available before 1860, but since that date figures of production are well known. Production to the end of 1938 was 126,677,000 tons. Maximum output began shortly after 1900 and was maintained over a period of years until 1920. The production in 1910 was 3,911,899 tons; in 1915, it was 4,082,212 tons; the low point occurred in 1934, when only 1,382,991 tons were mined. Annual tonnages have been relatively small; the decline has been due mainly to slow growth of population and industry, increased efficiency of utilization, growing competition with liquid and gaseous fuel, and the development of hydroelectricity as a substitute for heat and power formerly supplied by coal.

Geologically, the coals of Washington are relatively young, that is, they are found in the Eocene formations of the Tertiary period. However, one important fact must be noted, i.e., dynamochemical forces were active during later parts of recent geologic time with the result that all ranks of coal from lignite to anthracite were produced in the state. A distinct relationship exists between the degree of metamorphism of the Eocene sedimentaries and the rank of coal; thus, for example, the higher-rank coals are found near the Cascade Range and, away from this area of deformation, the coals show complete gradations to lignites. This relationship of rank and structure has been an important factor in the development and operation of mines and in the preparation of coal for market.

The principal resource is subbituminous in rank; bituminous, both coking and free-burning, constitutes the next major proportion; lignitic and anthracitic coals a relatively small proportion of the total. The coking coals constitute an important asset in connection with their utilization for metallurgical uses. The greater part of the production, 77 per cent, has been bituminous coal; subbituminous is second with 23 per cent; lignitic coals have been insignificant; and anthracitic coals have not yet been produced in commercial quantities.

Mining conditions in the state are in general complex. Very few seams of gentle dip are mined, most operations are conducted on steep dips, mechanization is limited to a few operating properties, and hand methods predominate. Cost of mining consequently is high, and a large proportion of the output, 66 per cent in 1936, must be washed before shipment. Seventy mines were in operation in 1937, with a total output of 2,018,036 tons, valued at $6,402,968; the number of men employed was 2934. Days worked during the year were 202. Kittitas County was the largest producer followed by King, Whatcom, and Pierce.

It is not possible here to give many analyses or extended data on properties of coals. A few typical examples from each of the producing counties are submitted in Table 2. Accurate data giving analyses of face samples and delivered coals, softening temperatures of ash, agglutinating and slacking values, and other related data are contained in a Bureau of Mines report (3) and a supplement to that report. The engineer interested in the utilization of Washington coals will find these two publications timely and authentic.

Modes of Transportation

The coal produced in the north-central and in the western part of the state, reaches all parts of the state by four transcontinental railroad lines or their branches. Western Washington coal is trucked extensively in the Puget Sound and southwestern districts; a small tonnage of coal from the Roslyn field moves easterly by truck but the main movement is by rail; some coal is handled by tidewater transportation from Seattle, Tacoma, and Bellingham. The principal distribution area is within the state; there is a small interstate movement to Alaska, Idaho, and Oregon; and some export trade to British Columbia and Pacific Ocean points. The truck movement of coal amounted to 25.5 per cent of the tonnage produced in 1938. Storage of coal because of seasonal and transportation difficulties is unnecessary, and all the important population centers can easily be served directly from the mines.

Utilization of Washington Coals

Washington coals enter into every field of use and activity in raw, briquetted, pulverized, and carbonized forms. Household heating, steam generation, locomotive and steamship fuel, copper smelting, magnesite burning, and ceramic firing represent direct uses either in hand, mechanical-stoker, or pulverized-firing methods. Coal has been used to manufacture coke in bench, beehive, and by-product ovens, also in the various coal-gas, producer-gas, and water-gas processes. Briquettes are used in domestic and industrial firing and occasionally for orchard-heating purposes. The coal can be used as a reducing agent or as raw material in some chemical and metallurgical processes. Washington coke has been used in metallurgical plants, in foundries, in sugar refineries, in producer- and water-gas production, and in domestic heating. In short, some type of local product has been found suitable for coal utilization in its many fields and applications.

In 1938, the output was utilized as shown in Table 3. Although the demand for coal for some uses, namely, manufactured-gas plants, cement mills, smelters, miscellaneous industrial plants, and commercial and domestic heating, shows an increase over the last decade when expressed as a percentage of state

TABLE 2 TYPICAL ANALYSES OF TIPPLE SAMPLES OF WASHINGTON COALS

County and district	Designation and size,[a] in.	Preparation	Moisture, as received, per cent	Moisture-free basis: Volatile matter, per cent	Fixed carbon, per cent	Ash, per cent	Sulphur, per cent	Btu	Ash-softening temperature, F
Whatcom	Stoker, 5/8 sh − 5/16 sh	Washed	10.5	35.6	45.0	19.4	0.3	11010	2640
Whatcom	Buckwheat, − 5/16 sh	Washed	8.9	36.0	48.7	15.3	0.3	11600	2510
King									
McKay	Steam, − 1 sh	Washed	12.8	43.0	52.3	4.7	0.6	13310	2160
Cumberland	Steam, − 7/8 sh	Washed	5.4	36.9	45.0	18.1	0.6	12070	2910+
Renton	Buckwheat, − 1/2 sh	Washed	14.7	39.9	46.6	13.5	0.6	11790	..
Pierce	Steam, − 1 1/4 bar	Raw	3.2	35.9	52.0	12.1	1.0	13530	2280
Kittitas	Steam, − 1 5/8 rh	Washed and dried	4.9	38.4	46.9	14.7	0.3	12710	2370
Kittitas	Steam, − 3/4 rh	Washed and dried	3.4	38.4	48.4	12.8	0.4	12890	2600
Thurston	Steam, − 5 bar	Raw	19.8	40.4	47.0	12.6	0.7	11360	2420
Thurston	Stoker, 1 3/4 rh − 1/4 sh	Washed	22.4	42.8	46.9	10.3	0.6	11600	2390

[a] sh = square-hole screen; rh = round-hole screen; bar = bar screen.

TABLE 3 USES OF COAL, 1927, 1936–1938[a]

Use	Per cent 1927	1936	1937	1938
Colliery fuel	2	1	1	1
Railroad fuel, including shop and station coal	43	31	29	29
Public-utility central-heating plants	5	3	4	4
Public-utility steam-electric plants			1	0
Fuel-briquette plants	3	0	0	0
Beehive-coke ovens	2	0	0	0
By-product-coke ovens	3	3	1	0
Other manufactured-gas works, including boiler coal	2	2	3	3
Cement mills	6–8	11	11	12
Smelters	.....	2	2	3
All other industries	7–14	6	13	13
Commercial and domestic heating	15–23	37	33	32
Steamship bunkers	1–2	0	0	0
Foreign exports and shipments to Alaska	1–2	4	1	2
Shipments to adjacent states	.....		1	1
State production, net tons	2,635,062[b]	1,812,104[b]	2,018,036[b]	1,572,057[c]

[a] Refer to Bibliography (3).
[b] Production reported by Bureau of Mines.
[c] Production reported by State Inspector of Coal Mines.

production, the actual tonnages have decreased. The most drastic curtailments in demand have occurred in coal used by railroads, briquette plants, coke plants, and steamships. Railroads and commercial and domestic heating, however, continue to be the most important consumer units.

The production of fuel briquettes began in 1911, and reached a maximum in 1917, with an output of 109,177 tons. The total reported production 1914 to 1938 amounted to 1,069,302 tons. In addition to coal briquettes, petroleum-carbon briquettes, made at oil-gas plants in Portland and Seattle, are sold for domestic use, and they are also replacing coke and coal in some water-gas processes. "Packaged fuel" made from outside coals and from petroleum coke is being produced and marketed in Spokane.

Pulverized coal has been used at power plants, smelters, cement mills, and miscellaneous industrial establishments since 1918. The quantity used reached a maximum of 350,000 tons in 1937; in 1938, sixteen per cent of the output of the state was fired in this form. Table 4 shows the consumption in 12 plants

TABLE 4 POWDERED-COAL CONSUMPTION IN NET TONS BY TYPES OF USERS[a]

Year	Cement plants	Smelters	Utilities, principally steam plants	Miscellaneous	Total
1927	225000	42000	50000	2500	319500
1928	175000	42000	51500	2000	270590
1929	167736	45862	52807	2600	269005
1930	186408	44204	56203	3882	290697
1931	138643	34842	80984	10898	265367
1932	59409	26057	80004	10675	176145
1933	33438	22223	77943	8710	142314
1934	67882	27454	60350	10266	165952
1935	60956	43202	56827	8419	169404
1936	195913	46095	63451	9240	314699
1937	224562	41837	73069	9709	349177
1938	140752	45203	58131	9434	253520

[a] Refer to Bibliography (3).

using Washington coal. Slack and buckwheat sizes produced in Whatcom, King, and Kittitas Counties supply the greater portion of this market.

The degree to which coal mined outside of the state, wood, fuel oil, and hydroelectricity affected the distribution and use of fuels in 1936 is shown in the analysis given in Table 5.

Price Schedule for Coals

Minimum prices for coal have been proposed and recommended by the Bituminous Coal Commission for various market areas in Washington. The entire schedule is too complicated to include in this review; instead, a summary, Table 6, will be given of prices of principal industrial sizes for shipment into the Seattle market territory. It is necessary to remember that the coals from Kittitas and Pierce Counties are bituminous; Thurston, Lewis, and Cowlitz Counties are subbituminous of low rank; Whatcom County coals are bituminous; and the King County coals are divided into the subbituminous, low-ash coals of the McKay district, the subbituminous of the Renton district, and the bituminous of the Cumberland district. Market prices may be expected to exceed the recommended minimum code figures.

Coals for Coking Purposes

Washington contains the only important coking-coal resource along the Pacific Coast. California and Oregon possess none, so far as present evidence is available; western British Columbia has coking coals but not of commercial significance at the present time; Alaska contains high-grade resources which are practically undeveloped and remote from transportation. Eastern British Columbia produces coke which is mainly consumed in the "Inland Empire" region of Washington and Idaho. It appears certain that attention must be focused on Washington in any consideration of future supplies of proved resource of this coal.

The principal coking coals are found in the Wilkeson-Carbonado-Fairfax district of the Pierce County field. This area has supplied the major production of coking coals and beehive coke. Portions of the beds in the Roslyn field of Kittitas County possess fair coking properties; and some eastern King County coals have fair to good coking qualities which make them potential sources for mixing or blending purposes.

Coke has been produced since 1880, in beehive ovens, and one by-product plant in Seattle, primarily operated for gas, produced coke from 1914 to 1937. This coke, together with gas-house coke, has been used in a wide variety of applications. Its severest handicap has been a moderately high percentage of ash, offset to a degree by low sulphur content. Some investigation is under way to produce better coke by improved preparation of coals and by mixing suitable coals of low ash content. The production of beehive coke from 1880 to 1937 was 1,631,319 tons; production of by-product coke 1914 to 1937 was 709,429 tons; a grand total of 2,340,748 tons. No coke was produced in 1938 and 1939.

In this brief review an attempt has been made to show that Washington possesses large resources of coal of various ranks and grades suitable for the needs of modern industry. Many of these coals are not of as good quality as coals from eastern mines; geologic conditions of deposition were different, with the result that the beds contain higher ash and generally are more difficult to mine and to clean. However, the fact that they have been successfully used for the various needs and demands of consumers over nearly a century indicates the fact that they have a place to fill in the field of fuel utilization. Many investigations have been made of the washability, friability, slacking, and grindability properties, and of agglutinating and coking characteristics of these coals. These are outside the scope of this paper, but the reports are available to the engineer who may wish to secure detailed information; a few references are cited in the Bibliography. It appears that the matter of economics of use rather than availability and suitability of supply is the determining factor from the engineering standpoint.

Effect of Water-Power Developments on Use of Coal

The completion of gigantic programs of water-power development inevitably will curtail the demand for coal used primarily as sources of heat and power. Other outlets such as coke and gas manufacture, hydrogenation and low-temperature carbonization, chemical uses in metallurgical and electrothermal processes

TABLE 5 COMPETING FUELS AND HYDROELECTRIC POWER, 1936[a]

Fuel	Quantity, coal, net tons; oil, bbl; or electricity, kwhr	Coal or coal equivalent, net tons	Percentage of total power	Percentage of Washington coal output
Coal from other states	261,000	261,000	3.4	14.4
Coal from Canada	90,000	90,000	1.2	5.0
Total competing coal	351,000	351,000	4.6	19.4
Washington coal	1,812,000	1,812,000	24.0	100.0
Total coal	2,163,000	2,163,000	28.6	119.4
Wood	....	500,000[b]	6.6	27.6
Fuel oil	9,400,000	2,547,000[c]	33.7	140.6
Hydroelectric power	3,273,148,000	2,357,000[d]	31.1	130.1
		7,567,000	100.0	417.7

[a] Refer to Bibliography (3).
[b] Refer to Bibliography (4). No estimate of the amount of wood fuel used more recently than that of 1924 is available; consequently, that figure is used to indicate the approximate magnitude of competition from wood in 1936.
[c] A conversion factor of 0.271 ton of coal per bbl of oil was used; this factor is based on a carefully estimated average Btu of 11,400 per lb, as-received basis, for Washington coal, and an average Btu of 147,000 per gal for fuel oil.
[d] A conversion factor of 1.44 lb of coal per kwhr, the average for central stations in the United States in 1936, was used.

TABLE 6 RECOMMENDED PRICE RELATIONSHIP FOR WASHINGTON COALS VIA RAIL TRANSPORTATION INTO SEATTLE MARKET AREA

Size group, in.	Kittitas County	Pierce County	Thurston, Lewis, and Cowlitz Counties	Whatcom County	King County: McKay District	King County: Renton District	King County: Cumberland District
Mine run	4.00[a]	3.90	2.75	....	....	....	3.50
$3\frac{1}{2} \times 0$	3.75	3.90	2.50	....	....	....	....
2×0	3.15[b]	3.65	2.00[b]	....	3.75	....	3.00–3.25
$1\frac{1}{4} \times \frac{7}{8}$	....	....	....	3.25	....	....	....
$1\frac{1}{4} \times 0$	....	3.50	....	....	3.50	....	....
1×0	2.95[b]	....	1.50[b]	....	....	....	2.90–3.15
$\frac{7}{8} \times \frac{3}{8}$	....	....	....	3.00	....	3.25	....
$\frac{3}{8} \times 0$	2.75	2.85	1.00	1.50	2.85	1.50	1.75
$\frac{3}{32} \times 0$	....	....	....	....	1.60	....	....

[a] Prices listed shall be increased 25 cents per net ton for washed sizes.
[b] Prices listed shall be increased 10 cents per net ton for washed sizes.
NOTE: All prices are net ton, 2000 lb, f.o.b. transportation facilities of the mines. All size designations are for round-hole screens. When coal is subjected to any chemical, oil, or waxing process, an additional charge of not less than 10 cents per net ton shall be added.

may in the future offset part of the losses. In any event, the resource of Washington coal appears to be a valuable asset in the expected growth of the Pacific Northwest.

ACKNOWLEDGMENT

In the presentation of this paper, the author has made use of material prepared by him for publication in various earlier reports. Certain data and tables, dealing with production distribution and use in Technical Papers of the Bureau of Mines, have been quoted directly. Acknowledgment is here made of this assistance.

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Discussion

D. S. HANLEY.[3] The coal industry in the Pacific Northwest is fortunate in having a College of Mines at the University of Washington, with which is associated such men as the author and to whom the industry can go for advice and assistance. Professor Daniels has always been most cooperative in anything of a constructive nature affecting the coal-mining business in our state.

In my opinion, the coal industry of the State of Washington has failed to take the steps it should have taken years ago in the way of research and an energetic sales campaign not only to retain the business it enjoyed but to increase its markets. The inroads of fuel oil from California, commencing about the year 1911, have resulted in the loss of a substantial part of the coal market. Notwithstanding the fact of the large increase in population and in industrial activity, the total coal output of the state now is practically the same as it was 50 years ago. What was once one of the three or four largest industries in the state should not feel proud of its apathy in standing idly by and permitting its markets to be gradually taken away and absorbed by other fuels.

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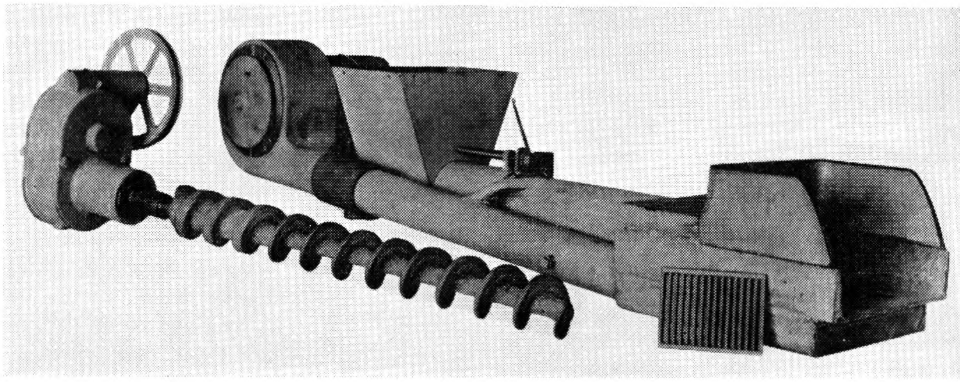

FIG. 1 PRINCIPAL PARTS OF OVERFEED STOKER, INCLUDING REDUCTION-GEAR BOX AND COAL-FEED SCREW; FAN HOUSING, ADJUSTABLE INTAKE DAMPER AND AIR DUCT; SLIDING FEED GATE, UNDERSIDE OF GRATE AND RETORT

Burning Characteristics of Washington Coals on Domestic Overfeed and Underfeed Stokers

By H. F. YANCEY,[1] K. A. JOHNSON,[2] AND J. B. CORDINER, JR.[3]

During the last 30 years, rapid advances have been made in the efficiency of fuel utilization in industry, but only during the last 10 years has comparable progress been realized in domestic heating. The most important advance in this field has been the development and wide application of the domestic coal stoker. The use of stokers for domestic heating originated in the Pacific Northwest, where two types, the underfeed and overfeed stokers are common. Elsewhere in the United States, the underfeed type predominates for domestic use. Studies[4] were made and reported upon in 1938 by the U. S. Bureau of Mines in cooperation with the College of Mines of the University of Washington on the subject of burning Washington coals on overfeed domestic stokers. Results were given of 25 combustion trials. Since then, the authors have conducted additional tests to include Washington and Oregon coals using both overfeed and underfeed stokers. This paper[5] describes the burning characteristics of caking and noncaking coals when fired on domestic stokers of both types installed successively in the same hot-water boiler. The results of burning trials with five coals selected from available data show the influence of variations in caking properties and ash-softening temperatures when used on both types of stokers.

THIS paper presents a comparison of the results of burning trials of coals, having different physical and chemical properties, on an overfeed and on an underfeed stoker. The coals were selected to demonstrate the influence of varying caking properties and different ash-softening temperatures on their performance with the two types of stokers. These two properties and size composition probably are the most important in affecting the behavior and suitability of coal for domestic-stoker burning. Caking is especially important in influencing the performance of coal on overfeed stokers, but is herewith considered for both types.

COAL USED

Five representative Washington coals, ranging in rank from subbituminous A to medium-volatile bituminous were fired. In caking properties, the coals ranged from nonagglomerating through weakly agglomerating to poor and good caking. In ash-softening temperature, they ranged from 2160 F to 2910 F. Ash contents ranged from 4.5 to 12.5 per cent.

Table 1 gives names and significant properties of coals tested. The description of caking or agglomerating properties is based upon the appearance of residue obtained by the standard method for the determination of volatile matter in the coal.[6]

The Harris and McKay coals were noncaking, but the other three coals, Roslyn, Roslyn-Cascade, and Wilkeson had caking strengths or agglutinating values of 700, 2300, and 10,240 g, respectively.

Comparative agglutinating values of coals from other states are as follows: Two samples from the Pocahontas No. 3 bed, West Virginia, had agglutinating values of 12,600 and 7130. One sample from the Pittsburgh bed in western Pennsylvania had an agglutinating value of 9810, while the Thick Freeport bed in the

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[4] "Burning of Various Coals Continuously and Intermittently on a Domestic Overfeed Stoker," by H. F. Yancey, K. A. Johnson, A. A. Lewis, and J. B. Cordiner, Jr., U. S. Bureau of Mines, Report of Investigations 3379, 1938.

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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.

[5] The work upon which this report was based was performed under a cooperative agreement between the Northwest Experiment Station, Bureau of Mines, United States Department of the Interior, and the University of Washington, Seattle, Wash. Published by permission of the Director, Bureau of Mines.

[6] "Agglomerating and Agglutinating Tests for Classifying Weakly Caking Coals," by R. E. Gilmore, G. P. Connell, and J. H. H. Nicolls, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 108, 1934, pp. 255–265.

TABLE 1 NAMES OF COALS TESTED AND THEIR PROPERTIES

Coal	Rank	Caking properties	Ash-softening temperature, F
Harris	Subbituminous A	Noncoherent	2910
McKay	Subbituminous A	Slightly coherent	2160
Roslyn 5	High-volatile A bituminous	Weak agglomerate	2340
Roslyn-Cascade 1	High-volatile A bituminous	Poor caking	2635
Wilkeson-Wingate	Medium-volatile bituminous	Good caking	2330

TABLE 2 ANALYSES OF COALS USED IN STOKER TRIALS, AS-FIRED BASIS[a]

Coal	Harris	McKay	Roslyn 5	Roslyn-Cascade 1	Wilkeson-Wingate
Size, in.	3/4–3/16	7/8–0	3/4–1/4	3/4–0	1 5/8–0
Analysis:					
Moisture, per cent	15.8	10.8	3.6	3.0	4.6
Volatile matter, per cent	32.9	38.2	38.0	37.0	24.7
Fixed carbon, per cent	38.8	46.5	46.5	48.7	58.8
Ash, per cent	12.5	4.5	11.9	11.3	11.9
Calorific value, Btu per lb	9850	11800	12390	12770	12930
Softening temperature of ash, F	2910	2160	2340	2635	2330
Agglutinating value, grams	0	0	700	2300	10240
Caking properties[b]	NAa	NAb	Aw	Cp	Cg

[a] Analyzed under supervision of H. M. Cooper, chemist, Pittsburgh Station, U. S. Bureau of Mines; ultimate analyses were made also but are omitted.
[b] NAa designates a nonagglomerate, noncoherent, volatile-matter residue; NAb, a nonagglomerate, slightly coherent residue that may be crushed by a 500-g weight; Aw, weak agglomerate; Cp, poor caking; Cg, good caking.

same general area showed a value of 6430. Samples of the Elkhorn bed in eastern Kentucky and the No. 6 bed in central Illinois had values of 4200 and 2120.

Table 2 shows the proximate analyses, calorific values, ash-softening temperatures, and agglutinating values,[7] or caking strengths of the five coals.

Stokers Used for Tests

Fig. 1 shows the principal parts of the overfeed stoker used in the work. The flat retort of this stoker gives a thin fuel bed typical of this method of burning. The available space on the retort is 9 in. wide × 8 in. long, giving a fuel-bed area of 0.5 sq ft. A sliding gate over the opening in the feed tube and the design of the screw allow the rate of feed to be varied between 10 and 25 lb of coal per hr.

Fig. 2 shows the underfeed stoker, installed in the hot-water furnace which was employed with both stokers, and the testing equipment. The retort of this stoker is of the usual pot type with an upper rim 11 in. in outside diam. The thickness of the fuel bed depends upon the type of coal being burned. A hydraulic mechanism allows the speed of the feed screw to be varied to feed between 11 and 27 lb per hr.

[7] "Test for Measuring the Agglutinating Power of Coal," by S. M. Marshall and B. M. Bird, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 88, 1930, pp. 340–383.

Furnace and Boiler

The boiler shown in Fig. 2, is of a size and type commonly used in residential heating by stoker or oil firing and comprises six vertical cast-iron sections. It has 52.6 sq ft of heating surface and a rated capacity of 565 sq ft of steam radiation or 905 sq ft of water radiation. The firebox is 23 in. wide, 20 1/2 in. long and 30 in. high from the top of the brick base to the crown sheet, giving a normal volume of 8.24 cu ft.

The distance from the grate of the overfeed stoker to the crown sheet was 26.5 in. The ashpit below the boiler in the brickwork was 4 in. deep and of the same cross-sectional area as the firebox. Ashes were removed through a door in the back section of the boiler. With the underfeed stoker, the distance from the top of the retort to the crown sheet was 22 in., but the floor of the refractory surrounding the retort was 2 in. below this, making the effective height to the crown sheet 24 in. The clinker was removed by means of tongs through the fire door.

Testing the Overfeed Stoker

The testing procedure for the overfeed stoker included a 2-hr adjustment period, an overnight hold-fire period, a 2-hr preheating period the following morning, and an 8-hr test period, during which the data were collected.

Immediately after the second 2-hr period, the stoker was stopped, the hopper and ashpit cleaned, a weighed quantity of coal was placed in the hopper, and the test was commenced. The useful heat recovered during the trial was measured by passing water through the boiler to two tanks set on dial-type platform scales, by means of which the heated water was weighed. The flow of water through the boiler was regulated to give a hot-water temperature of approximately 150 F at the outlet, while the temperature of the cold water fed to the boiler was about 50 F.

Continuously, throughout the test period, a sample of the flue gas was withdrawn from the smoke pipe at a constant rate, as determined by a flowmeter, and subsequently analyzed for carbon

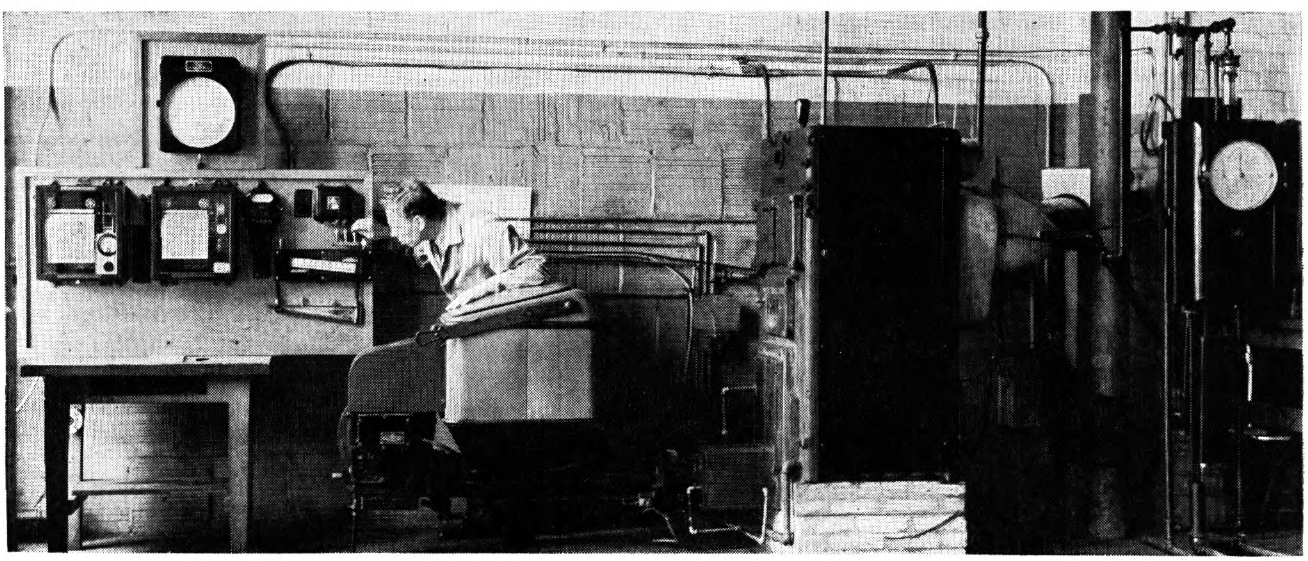

Fig. 2 Underfeed Stoker, Hot-Water Boiler, and Equipment Used in Coal-Burning Trials

dioxide, oxygen, hydrogen, carbon monoxide, methane, and unsaturated hydrocarbons. An absorption-type CO_2 recorder showed conditions during the test period.

The temperature of the flue gas was measured with a recording pyrometer, supplemented by the reading of an etched-stem mercury thermometer. The same method was used for determining the temperature of the hot water, except that a recording mercury thermometer was used instead of a thermocouple.

The condition of the fuel bed in the retort was noted from time to time through a Pyrex-glass observation window in the back door of the furnace. Comparative smoke readings were taken during the test at 15-sec intervals over a 30-min period, by comparing the density of the smoke with a Ringelmann chart.

At the end of the test, the soot was collected from the flue passages, smoke hood, and from the inside of the furnace by means of a vacuum cleaner.

Underfeed-Stoker Tests

The procedure for testing the underfeed stoker was similar to that for the overfeed stoker, with a few differences. Immediately after the second 2-hr period, the clinker was removed from the fuel bed and the thickness of the bed was recorded. The bed was observed periodically through the fire door. At the end of the test period, the clinker was removed. If necessary, enough loose ashes and coke or char were removed to give the same thickness of fuel bed as at the beginning of the test and appropriate adjustment was made.

Effect of Caking Properties

A comparison of the behavior of four of the five coals on both types of stokers, as indicated by the heat balance, is given in Table 3. The four coals tested were McKay, which is noncaking; Roslyn 5, weakly agglomerating with an agglutinating value of 700; Roslyn-Cascade 1, poor caking with an agglutinating value of 2300; and Wilkeson-Wingate, good caking with a value of 10,240. This is a range in caking properties from a coal which would produce only char on carbonization to one which would give a strong blast-furnace coke.

TABLE 3 PRINCIPAL RESULTS OF STOKER TRIALS OF CAKING COALS

Stoker	Overfeed			Underfeed		
Coal	McKay	Roslyn 5	Roslyn-Cascade 1	Roslyn 5	Roslyn-Cascade 1	Wilkeson-Wingate
Rate, lb per hr	13.3	13.4	14.2	15.5	16.7	17.0
Agglutinating value, g	0	700	2300	2020	3120	10240
Stack temperature, F	379	438	351	477	498	536
Carbon dioxide in flue gas, per cent	11.8	8.4	8.8	11.3	12.3	10.3
Heat balance:						
Efficiency, per cent	76.8	63.7	46.5	78.4	78.7	79.9
Losses, per cent						
Ashes	5.9	11.6	27.2	0.0	0.0	0.0
Soot	0.7	0.7	0.4	0.2	0.1	0.4
Dry flue gases	8.2	11.8	6.2	12.0	11.4	14.8
Moisture and hydrogen	5.5	5.0	4.2	5.4	5.2	4.4
Combustible in flue gases	0.0	0.5	6.5	0.0	0.4	0.0
Radiation and unaccounted for[a]	2.9	6.7	9.0	4.0	4.2	0.5
	100.0	100.0	100.0	100.0	100.0	100.0
Excess air, coal fired, per cent	48	77	27	58	45	70
Excess air, coal burned, per cent	59	102	79	58	45	71

[a] By difference.

Caking Coals on Overfeed Stoker

The most revealing item in the first part of Table 3, which deals with the overfeed burning of three coals that show increasing caking properties, is the efficiency. As caking increases, the efficiency decreases from 76.8 to 63.7 and to 46.5. At the same time, the loss of combustible in the refuse increases in the order, 5.9, 11.6, and 27.2, because the rate at which coal is supplied to the retort by the feed worm exceeds the burning rate; caking impedes the flow of air through the fuel bed.

All of the tests reported in this paper are continuous, that is, they do not simulate the off-and-on or intermittent operation of ordinary domestic-stoker operation. The latter type of burning also was used but is not reported in detail here. The ultimate effect, of course, is to lower the feed rate and allow more time for burning. The composition of the coal with respect to size is also an important factor in fuel-bed behavior, as is well known. For example, when the material finer than 4 mesh was removed from the 0 to $^3/_4$-in. Roslyn-Cascade coal (the one with the greater caking strength), it was burned continuously at about the same rate of feed with an efficiency of 57 per cent, compared to 46.5 per cent when unsized, and intermittently when sized with an efficiency of 65 per cent. This coal, however, would not be satisfactory for overfeed-stoker use and is not sold for that purpose.

Caking Coals on Underfeed Stoker

Table 3 also shows the results of continuous burning trials with the underfeed stoker on three caking coals, two of which, those of least caking strength, Roslyn 5 and Roslyn-Cascade, were burned on the overfeed. In contrast to the low efficiencies obtained with these two coals on the overfeed, they were burned with high efficiencies, 78.4 and 78.7 per cent, on the underfeed. The strongest caking coal, Wilkeson-Wingate, was burned with about equally high efficiency.

Differences in the caking properties of these coals affected conditions in the fuel beds. For example, as the agglutinating values of the coals increased, the resulting coke was less reactive, the formation of coke trees increased, and the fuel beds increased in thickness. Caking coals appeared to burn with thick fuel beds on the underfeed stoker, the thickness depending upon the natural accumulation of loose ashes, clinker, and coke. At the beginning of a test, just after a clinker was removed, the accumulation of loose ashes and coke gave a fuel bed that ranged in thickness from 3 to 7 in. above the top of the retort, according to whether the coal was weakly or strongly caking. As the test progressed, the thickness of the bed increased, owing to the formation of clinker. Just before the clinker was removed, either at the end of the test or at such time during the test as clinker removal became necessary, the thickness of the main part of the bed ranged from 7 to 9 in. The coke trees protruded occasionally as much as 9 in. above the main part of the bed.

Although large quantities of coke accumulated at various times during the burning trials, no difficulty was experienced in burning it, even with the strongest caking coal. The fuel bed went through irregular cycles of change. Starting with a low accumulation of coke, the coke built up in an irregular pile, sometimes with one or more coke trees extending almost to the crown sheet. This buildup of the fuel bed was accompanied by a period of slow combustion, during which the rate of combustion was slower than the rate of feed. The appearance of the fuel bed at this time usually was poor. As the cycle progressed, the coke trees fell and a period of rapid combustion reduced the bed to normal thickness.

Sherman and Kaiser, and Barnes,[8] all of the Battelle Memorial Institute, have presented a careful analysis of the formation and burning of coke in the underfeed fuel bed. Candee, working at Washington State College, designed a special retort with fuel-bed agitator to improve the burning of caking coals

[8] "Combustion of Bituminous Coal on the Small Underfeed Stoker," by R. A. Sherman and E. R. Kaiser, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 130, 1938, pp. 388–401. Also: "Fundamentals of Combustion in Small Underfeed Stokers," by C. A. Barnes, Bituminous Coal Research, Inc., Technical Report 4, 1938.

and to remove clinker or ash from the fuel bed automatically.[9] By better distribution of air to the fuel bed and breaking large pieces of coke to prevent blowholes, more satisfactory combustion was reported to have been obtained.

Burning of Noncaking Coals

Two coals, McKay and Harris, both of which are noncaking, were selected to show, first, the burning characteristics of noncaking coals on both overfeed and underfeed stokers and then to show the effect of a wide difference in ash-softening temperatures. Two different lots of each coal were used in the trials. The ash-softening temperatures of the two lots of McKay coal were 2340 and 2160 F. The variation in the softening temperature of the ash of the two lots of Harris coal was less, namely, 2910 and 2900.

Effect of Ash-Softening Temperatures

Overfeed Stoker. Table 4 shows the results of the tests of the two coals on both types of stokers. In considering the data for the overfeed tests, it should be remembered that the McKay coal is an unsized product, while the Harris coal is sized, as has been shown in Table 2. Despite this difference, nearly the same efficiencies, 76.4 and 75 per cent, were obtained. However, the loss of heat in combustible in the ashes amounted to 4.4 per cent with the unsized McKay coal and 2.1 per cent with the sized but higher-ash Harris coal.

Because of the differences in ash content and in ash-softening temperatures, the fuel beds of these two coals were different. Although the two coals were burned at nearly the same rate, the fuel bed with the McKay coal was only about 2 in. thick, as compared with a 5-in. bed for the Harris coal. The McKay coal, because unsized, gave an irregular bed, sometimes with large blowholes, whereas, the Harris fuel bed was uniform throughout the trial.

TABLE 4 PRINCIPAL RESULTS OF STOKER TRIALS OF NONCAKING COALS

Stoker	Overfeed		Underfeed	
Coal	McKay	Harris	McKay	Harris
Rate, lb per hr	15.9	14.8	16.9	15.6
Ash content, per cent	4.8	12.0	4.3	13.0
Ash-softening temperature, F	2340	2910	2160	2900
Stack temperature, F	471	422	475	425
Carbon dioxide in flue gas, per cent	13.5	10.1	12.2	9.3
Heat balance:				
Efficiency, per cent	76.4	75.0	75.0	73.6
Losses, per cent				
Ashes	4.4	2.1	0.0	0.0
Soot	0.4	0.6	0.4	0.3
Dry flue gas	9.1	10.9	11.0	11.8
Moisture	1.0	1.9	1.1	1.9
Hydrogen	4.4	4.8	4.8	4.3
Combustible in flue gases	2.0	0.0	0.5	5.4
Radiation and unaccounted for[a]	2.3	4.7	7.2	2.7
	100.0	100.0	100.0	100.0
Excess air, coal fired, per cent	24	70	44	79
Excess air, coal burned, per cent	30	75	45	80

[a] By difference.

Because of the wide difference in ash-softening temperature, the character of the clinker was different. McKay coal gave dense, well-fused clinkers; the Harris clinker was porous, friable, and fell off the end of the retort in small pieces. Sized McKay coal was also tested but, because the rate of feed was lower than in the other trials, detailed data are not given. The efficiency of this test (at 12.4 lb per hr), was 73.4 per cent and the loss of carbon in the ashes was 0.2 per cent. Conditions in the fuel bed were much more uniform than with unsized coal.

Underfeed Stoker. Coal from the McKay bed is used widely for underfeed domestic burning, but the Harris coal is sold only for overfeed use. Data in Table 4 show that both coals burned with approximately the same efficiency on the underfeed and that the efficiency was nearly the same as with the overfeed stoker.

Combustion of the McKay coal on the underfeed stoker is characterized by a thin fuel bed. Because of the low ash-softening temperature, dense clinkers are formed close to the tuyère ring at the top of the retort. The accumulation of clinker, because of the low ash content of the coal and the high density of the clinker is slow. Nevertheless, this type of clinker, if allowed to accumulate too long, checks the flow of air supplied by the fan.

That a heat-balance statement, showing a reasonably high efficiency, fails to disclose the general suitability of a coal for stoker-burning, is exemplified by the data for the test of Harris coal on the underfeed stoker. Five underfeed trials were made on the Harris coal and the results of only the best of these is shown. Hence, the data do not truthfully represent the difficulty experienced in burning this coal on the underfeed stoker. To obtain these results, it was necessary to use considerable excess air and to carry a thick fuel bed. Because of the high ash-softening temperature, 2900 F, only 6 lb of clinker strong enough to be removed with tongs were formed. The remaining 10 lb were present in the fuel bed as loose ashes. Thus, the high ash-softentemperature, although advantageous with the overfeed stoker, renders this coal unsuitable for underfeed burning.

Summary and Conclusions

In general, it is concluded from the foregoing tests that the overfeed stoker is suitable for noncaking coals which have a wide range in ash content and ash-softening temperature. Even coals with very refractory ash fusing above 2900 F are suitable. No coals with an ash fusibility below 2100 F were tested. Because ashes are eliminated from the overfeed fuel bed as they are formed, large quantities of ash are easily discharged.

As is well known, caking coals are unsuitable for overfeed stokers. Weakly caking coals, even when burned in a sized condition, gave a low efficiency because excessive amounts of unburned coke were lost in the ashes. In contrast, weakly caking coals were burned with high efficiency on the underfeed stoker.

Two noncaking coals, one having a low ash content and low ash-softening temperature and the other having a medium ash content and high ash-softening temperature, proved satisfactory for overfeed burning, but the one with a medium ash content and high ash-softening temperature was unsatisfactory for underfeed burning, because the ash could not be fused to a clinker.

Both noncaking and caking coals may be burned satisfactorily on the underfeed stoker. Only one coal strong enough in caking properties to be used for manufacturing metallurgical coke was tested. This coal was burned in the underfeed stoker with a high efficiency, but some strongly caking and swelling coals are known to give difficulty. With respect to ash content and ash-softening temperature, the bituminous-type underfeed stoker ordinarily is limited to the burning of coals having low to medium ash contents and not too high ash-softening temperatures.

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[9] "The Development of a Domestic Stoker to Burn Washington Coals," by E. W. Candee, State College of Washington, Engineering Bulletin No. 56, 1938.

The Radiation of Furnace Gases

By H. C. HOTTEL[1] and R. B. EGBERT[2]

Data of various investigators on the emission and absorption of radiation by carbon dioxide and water vapor are reviewed critically. Recommendations of procedure in calculation of heat transmission by gas radiation are given, Fig. 3, with Equation [7] or [8] for carbon dioxide and Fig. 10 with Equation [17] for water vapor. Partial pressure P_w of water vapor, at a constant value of P_wL, affects gas radiation, but probably to a much smaller extent than has been reported in previous literature. A simplified procedure is presented to allow for the effect of the gas shape on radiant heat interchange.

THE last fifteen years have witnessed a recognition by engineers of the importance of infrared radiation from gases in affecting heat transfer, especially at high temperatures where radiation from such gases as carbon dioxide and water vapor may be several times the heat transmission due to convection. Within the last ten years the basis of estimation of such heat transfer has changed from one dependent on inadequate measurements of the infrared absorption spectrum at room temperature, combined with a series of simplifying assumptions, to one dependent on direct measurement of total radiation. Since such experiments are somewhat difficult and the range of variables to be covered is great, it is not surprising that no complete agreement exists at present as to the procedure engineers should adopt in the calculation of radiant heat transfer from flue gases. It is the object of this paper to examine critically the data obtained by various investigators, to present some new data, and to consider whether it is possible to resolve the conflict of existing recommendations.

General Formulation of Heat Interchange by Gas Radiation

There is both an experimental and a theoretical basis for expecting emission and absorption of radiation by gases to have importance in any high-temperature heat exchanger involving the presence of heteropolar gases, i.e., gases whose molecules are composed of atoms carrying charges. Since all gases except the elementary gases such as hydrogen, oxygen, nitrogen, and argon are to some extent heteropolar, only these elementary gases are free from infrared absorption and emission bands. Infrared gas radiation of importance in heat transfer is due to changes in the energy levels of the molecule owing to its rotation and interatomic vibration.

The net interchange of radiation between a gas and some other body, due to one radiating (and absorbing) component of the gas, can be represented by the following general formula

$$q/A]_{\text{net}} = (E_G - A_{G,S}) = f_1(L, P_G, P_T, C, T_G) - f_2(L, P_G, P_T, C, T_G, T_S) \quad \text{[1]}$$

where

$q/A]_{\text{net}}$ = net heat transfer (Btu)/(hr)(sq ft)
E_G = emission from gas to other body
$A_{G,S}$ = absorption, by gas, of radiation from other body
P_T = total pressure (constant at one atmosphere for present field of interest)
P_G = partial pressure of the radiating constituent of the gas, atm
C = composition of the remainder of the gas
L = length of radiant beam through gas mass, ft
T_G = gas temperature, deg R
T_S = temperature of other body (generally a surface) with which gas is exchanging heat, deg R

The effect on total emission of the composition C of the diluent is but incompletely known. Measuring the infrared absorption spectrum of carbon dioxide, Hertz (1)[3] found that hydrogen as a diluent gas in place of air increased the absorption by a small but definite amount, while the use of nitrogen or oxygen in place of air as diluent made no appreciable difference. Hence, at least in the case of carbon dioxide, variation in composition of the diluent nonradiating fraction of flue gas, which consists mainly of nitrogen with varying amounts of oxygen, will not affect the radiation from the gas, and can be ignored.

The heat-transfer relation now simplifies to

$$q/A]_{\text{net}} = f_1(L, P_G, T_G) - f_2(L, P_G, T_G, T_S) \quad \text{[2]}$$

Even this simplification leaves a large number of variables. Schack (4) and Hottel (3), in the absence of direct data, made two assumptions. The first was that the number of molecules in the path determined E_G or $A_{G,S}$, i.e., that the effect of P_G and L entered as a single variable, the product P_GL. The validity of this relation, known as Beer's law, had been examined in considerable detail by Hertz (1) in connection with carbon dioxide, and less completely by von Bahr (2). Hertz found that at a fixed total pressure the absorption depended upon P_GL and was substantially the same whether pure carbon dioxide and a short path length or a lower partial pressure and longer path length were used, so long as total pressure was held constant by a nonradiating gas such as air. As already stated, the use of hydrogen instead of air resulted in a small but definite increase in absorption, about 2 to 5 per cent, depending upon the absorption band studied. Von Bahr examined a wide range of gases and compared the absorption of monochromatic radiation by pure gas at one atmosphere in 3-cm path lengths with absorption by the same gas at 1/11 atm partial pressure and a 33-cm path length, the total pressure being maintained at one atmosphere by air or hydrogen. Generally, her measurements were restricted to a single spectral region near the peak of an absorption band. She found that, at constant total pressure and temperature, the absorption was dependent on the term P_GL alone for all bands investigated for carbon monoxide, carbon dioxide, methane, ethylene, acetylene, methyl ether, ethyl ether, and for the 3.0μ band for ammonia. For the 6.3μ band of ammonia, 1/11 atm and 33-cm path length produced only 90–95 per cent as much absorption as one atmosphere of ammonia in a 3-cm path length. Becker (5) found for HCl at the 3.3–3.5μ band a somewhat lower absorption when partial pressure is down and total pressure is maintained with air than for pure HCl at one atmosphere and a correspondingly shorter length, at constant

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

P_GL. It appears, therefore, from the work of Hertz, von Bahr, and Becker that at a constant total pressure of one atmosphere Beer's law is adequate for carbon dioxide but may lead to error, probably small, for certain other gases, notably ammonia. Water vapor, on which no data of this type are available, is chemically more similar to ammonia than to the other gases studied.

The second assumption was that, at a given P_GL, the absorption by a gas at any temperature T_G of radiation from a surface at T_S was equal to the emission from a gas at T_S (an assumption which is exact when $T_G = T_S$, by Kirchoff's law). That is, at constant P_GL and surface temperature the absorption by the gas, of radiation from the surface, is independent of gas temperature, even though increasing the gas temperature at constant P_GL decreases the number of radiating molecules. As will be shown later, this assumption is not always justifiable.

These two assumptions permitted a further simplification of Equation [1] or [2] to the form

$$q/A]_{\text{net}} = f_1(P_GL, T_G) - f_1(P_GL, T_S) \qquad [3]$$

that is, to a single function f_1 of two variables P_GL and T, evaluated successively at T_G and T_S.

Direct Measurements

The direct measurement of total gas radiation, made by sighting a total-radiation pyrometer through the gas onto some sort of background, is actually a measurement of net radiant heat interchange between the thermopile surface and what it sees, which latter may be gas or gas with a black-body background. Two methods of taking data are used. One employs a black body at room temperature as a background or makes use

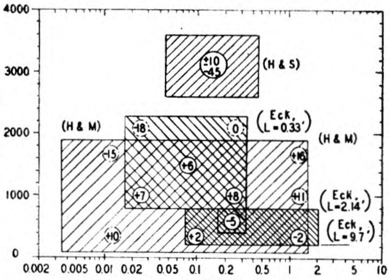

Fig. 1 Range of Variables Covered in Study of Carbon Dioxide Radiation

(Numbers in circles indicate percentage difference of different investigators, see text.)

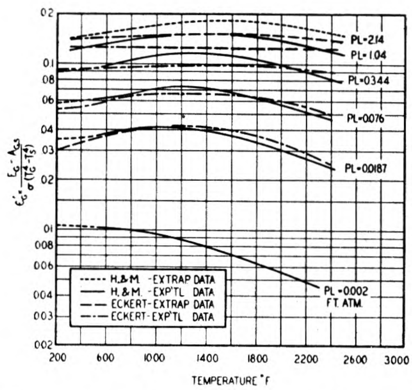

Fig. 2 Comparison of Data on Carbon Dioxide Emission

TABLE 1 SUMMARY OF EXPERIMENTAL MEASUREMENTS ON CARBON DIOXIDE

Author	Path length L, ft	Partial pressure P_c, atm	Gas temp t_G, F	Background temp t_S, F	Diluent gas	Knowledge of L	Uniformity of T_G	Stray radiation	Other comments
Hottel and Mangelsdorf, 1935	1.68	0.002, 0.005, 0.010, 0.020, 0.080, 0.167, 0.500, 1.000	70 to 1910	−297 and 70 ± 2 to 2480	Dry CO_2-free air	Concentration traverse made. Sharp gradient within isothermal zone; established L to 2%	Uniform to 2 F	1.3% of black-body radiation at gas temperature	Radiometer sighted horizontally through horizontal tubes with nozzles at ends and hot-dry air protection at ends to prevent absorption at edges of zone. Mirror in radiometer only
Hottel and Smith, 1935	0.64 to 1.312	0.241 to 0.369	2568 to 3778	70 ± 2	$N_2 + O_2$	No concentration traverse. Reliance upon boundary between premix flame and air	Very narrow angle required by radiometer so that beam passed through isothermal gas zone. Sharp temp gradients at end. Temp measured by sodium-D line reversal method	Not determined. Should be negligible, as no hot surfaces present	Narrow-angle radiometer sighted through long Meker flame from combustion of CO-air and CO-air-oxygen mixtures. Mirror in radiometer only. Radiation is due to products of combustion only
Eckert, 1937	2.14	0.0355, 0.062, 0.13, 0.25, 0.50, 1.00	212 to 752	75	N_2	No concentration traverse but a calculation of one made. Actually an error of 5 cm can be introduced under poor conditions according to experience of H.&M. with flow through similar nozzles	Good. Traverse given. No protecting air column at end except that due to partially preheated room air	Impossible to separate stray from mirror emission. Control experiment indicated total stray and emission from mirror was 1.8 to 2.6% of black body at gas temp	Apparatus involves two nozzles at bottom and mirror in top of vertical furnace. Plane mirror outside of furnace. Thermopile hence sees three mirrors in all. Originally, H_2 used as diluent but reacted with CO_2. To prevent heavier CO_2-N_2 mixture from mixing by gravity with air below furnace, high velocity through nozzles was used
	0.334	0.056, 0.122, 0.23, 0.49, 1.00	760 to 2300 (0.056–0.49); 390 to 2300 (1.00)	75	N_2	Same remarks as above	Same remarks as above	Stray = 1% of black body at gas temp	Apparatus similar in principle to H.&M. except that, instead of gas boundary formed by dry CO_2-free air at high temperature, partially heated unscrubbed atmospheric air is used
	9.7	0.46, 0.23	212	75	H_2	No concentration traverse; no nozzle system to obtain sharp boundary	Uniform in furnace, gradient occurring over 10 cm length at ends	Total stray + emission from mirror + reflection of background = 6% of black body at gas temperature	Vertical furnace with mirror in top in hot zone. Background was flat surface

of a mirror so that the radiometer is its own background. This method yields emission characteristics only when the gas temperature is high enough to make negligible the term representing absorption, by the gas, of radiation from a black body at room temperature. The second method requires black bodies at different temperatures, ranging from liquid air to temperatures exceeding 2000 F, and permits separation of emission and absorption characteristics of the gas. Until this type of measurement is made there is a question as to the validity of using the net interchange factors, calculated from data obtained by the first type of measurement, as a basis for heat-transfer calculations.

The first experimental measurements were those of Lent and Thomas (6) on products of combustion of blast-furnace gas flowing in a duct four feet in diameter. A radiometer was sighted through the gas stream onto a water-cooled black surface. Uncertain path length and presence of considerable stray radiation make the data reliable to probably no better than 20 per cent.

Schmidt (7) next published the results of the first comprehensive investigation of water-vapor-radiation interchange between pure steam at different path lengths and temperatures and a blackened surface at room temperature.

Hottel and Mangelsdorf (8) studied steam-air mixtures, carbon dioxide-air mixtures, as well as steam-carbon dioxide-air mixtures, and measured both emission and absorption by using hot and cold black-body backgrounds. They varied the partial pressure and temperature of the radiating gas and kept the path length constant.

Hottel and Smith (9) measured emission from products of combustion of carbon monoxide and carbon monoxide-hydrogen mixtures at temperatures up to 3780 F.

Eckert (10), like Schmidt, measured the net interchange between the gas and a surface at room temperature, studying mixtures of carbon dioxide and nitrogen, carbon dioxide and hydrogen, and steam and nitrogen. Eckert varied the partial pressure and the temperature and varied the path length by the use of three different furnaces.

Eberhardt (11) measured the radiation from the flue gases in a steel reheating furnace, by sighting a radiometer across the gas within the furnace, through two openings, one in each side of the furnace, onto a black body. The gas contained both carbon dioxide and water vapor and had a path length of 14 ft. The temperature and the partial pressures of the radiating gases were varied.

Brooks (12) measured the emission and absorption of atmospheric air containing both carbon dioxide and moisture, at room temperature, and varied the path length.

Carbon Dioxide

The extent of the experimental investigation of carbon dioxide is indicated by Table 1 and Figs. 1 and 2. As already mentioned, Hertz (1) and von Bahr (2) found that Beer's law was valid for the various monochromatic absorption bands of carbon dioxide, and Eckert's results on three furnaces of different lengths support the same conclusion. Hence the radiant heat interchange between carbon dioxide and a bounding surface in industrial high-temperature heat-exchange equipment, which is generally operated at a fixed total pressure of one atmosphere, is dependent upon three operating variables, namely, gas temperature, surface temperature, and the product of path length by partial pressure of carbon dioxide, P_cL.

Fig. 1 indicates the range covered by the three most comprehensive investigations, namely, those of Hottel and Mangelsdorf (8), Hottel and Smith (9), and Eckert (10). Gas temperature is plotted against P_cL. Each cross-hatched rectangle represents the range in which emissivity can be obtained by direct interpolation of the actual data points of a particular investigator. As can be seen from Table 1, Eckert used a black-body background at room temperature only. Hence his measurements were reported as a pseudoemissivity ϵ_G', which is a net interchange factor between the gas and a black body at room temperature defined as follows

$$\epsilon_G' = \frac{(E_G - A_{G,R})}{\sigma(T_G^4 - T_R^4)} \qquad [4]$$

where

E_G = gas emission
$A_{G,R}$ = absorption, by gas, of room temperature radiation
T_G = gas temperature
T_R = room temperature
σ = Stefan-Boltzmann constant

As the temperature of the gas increases ϵ_G' approaches the true gas emissivity $\epsilon_G = (E_G/\sigma T_G^4)$, and above 1000 F the two can for all practical purposes be considered equal. For comparison with Eckert, the results of Hottel and Mangelsdorf have been converted to ϵ_G' wherever that quantity differs from ϵ_G. The numbers in the small circles in Fig. 1 represent the percentage difference between the recommendations of the two investigators, i.e.

$$\text{Number} = \left(\frac{\epsilon_G',_{\text{H.\&M.}} - \epsilon_G',_{\text{Eckert}}}{\epsilon_G',_{\text{H.\&M}}}\right) 100$$

Since the data of Hottel and Smith on carbon monoxide flames (top rectangle of Fig. 1) were used by Hottel and Mangelsdorf in establishing the high-temperature extrapolation of the latters' final recommended curves and since the flame measurements were accurate to about ten per cent, the agreement of the H.&M. extrapolation is likewise within ±10 per cent of direct experimental data. This is indicated by the top number in the large circle. The lower number in the same circle is the percentage difference between Eckert's extrapolations and the H.&M. extrapolations. Fig. 2 likewise compares the two main groups of data; ϵ_G' is plotted as a function of temperature for several values of P_cL, with indication of whether each curve is based on data or extrapolations. An examination of both figures indicates that the recommendations of Eckert and of Hottel and Mangelsdorf agree to within 5 to 8 per cent in the range of temperature and P_cL where both investigators have experimental data. Outside this range at temperatures below 2000 F, discrepancies of 15 to 20 per cent exist. At high temperatures (about 3200 F) Eckert's recommendation is considerably higher.

Both Eckert and Hottel and Mangelsdorf took great care in measuring temperatures and kept errors caused by stray radiation down to less than 1.5 per cent of black-body radiation. The latter investigators determined the path length by a concentration traverse and used hot, dry, CO_2-free air as a windowless boundary to confine the radiating gas. Runs were made only when concentration gradients at the end were sharp and the temperature uniform through the center chamber. Although Eckert did not make such a concentration traverse, he used a system in which gas flow was probably somewhat steadier than that of Hottel and Mangelsdorf, and the concentration gradients at the boundary of his gas layer were probably satisfactory. Eckert allowed room air containing water vapor and carbon dioxide partially heated by the apparatus to form the boundary confining the radiating gas. As a result, concentration and temperature gradients occur simultaneously in the boundary layer, and the boundary gas contains some carbon dioxide and water vapor. This possibility of error may explain the discrepancies in the recommendations of the two investigators.

In addition to studies of emission, Hottel and Mangelsdorf

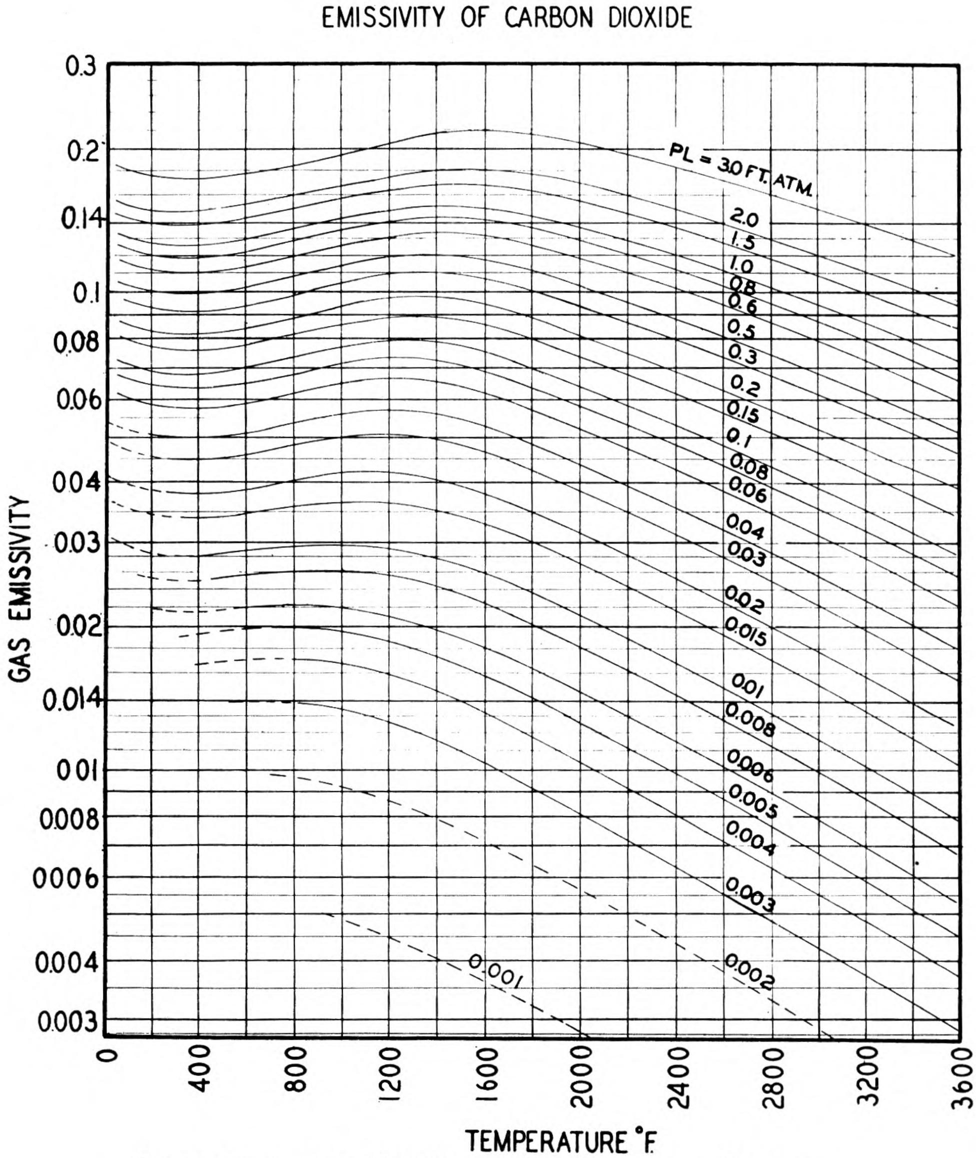

FIG. 3 RECOMMENDED WORKING PLOT OF CARBON DIOXIDE RADIATION

made measurements of absorption, by the carbon dioxide, of black-body radiation from backgrounds at different temperatures. They found that, on maintaining the path length and partial pressure of carbon dioxide and the temperature of the source of black-body radiation all constant and varying the gas temperature, the absorption of radiation by the gas increased with an increase in gas temperature. Tingewaldt's (13) measurements of the absorption of monochromatic infrared radiation by carbon dioxide at various temperatures have confirmed the aforementioned result.

Although this finding prevented the simplification of the heat-transfer relation from the form represented by Equation [2] to that represented by Equation [3], an empirical relation between gas absorption and gas emission was established; consequently a single plot of emissivity in terms of T and PL is sufficient to determine both the total emission of radiant energy and the absorption of black-body radiation by carbon dioxide. Absorption is given in terms of emission by the relation

$$\alpha_{G,S} = \left(\frac{T_G}{T_S}\right)^{0.65} \epsilon_{S,P_cLT_S/T_G} \quad \text{[5]}$$

where

$\alpha_{G,S}$ = absorptivity of gas at T_G for radiation from a black source at T_S

$\epsilon_{S,P_cLT_G/T_S}$ = gas emissivity at temperature T_S and at pressure-length product equal to P_cLT_G/T_S

A study of Table 1 and Figs. 1 and 2 does not permit an unequivocal decision as to which data to recommend for use. However, since the data of Hottel and Mangelsdorf are in good agreement with Eckert in the range covered by him and their extrapolation to high temperatures is in good agreement with Hottel and Smith's data, the H.&M. data have been used as a basis for constructing a working chart of emissivity versus temperature, for various values of P_cL. Such a chart appears as Fig. 3, to be used for evaluating terms in the expression for net radiant heat interchange between gas containing CO_2 and its bounding surfaces

$$q/A]_{CO_2} = \epsilon(E_G - A_{G,S})$$
$$= \epsilon\sigma(\epsilon_G T_G^4 - \alpha_{G,S} T_S^4) \quad \text{[6]}$$

$$= 0.1723\ \epsilon \left[\epsilon_{G,P_cL} \cdot \left(\frac{T_G}{100}\right)^4 - \left(\frac{T_G}{T_S}\right)^{0.65} \cdot \epsilon_{S,P_cLT_S/T_G} \cdot \left(\frac{T_S}{100}\right)^4 \right] \quad [7]$$

where

ϵ = emissivity (and absorptivity) of gray surface which bounds the gas.

Since the second or absorption term in the bracket is somewhat tedious to evaluate, it has been recommended (8) that when the gas temperature is high enough to make absorption much less important than emission (say, at least one third greater than surface temperature, absolute scale), the absorptivity $\alpha_{G,S}$ be assumed equal to gas emissivity at the surface temperature, E_{S,P_cL}. Then Equation [7] simplifies to

$$q/A]_{CO_2} = 0.1723\ \epsilon \left[\epsilon_{G,P_cL} \cdot \left(\frac{T_G}{100}\right)^4 - \epsilon_{S,P_cL} \cdot \left(\frac{T_S}{100}\right)^4 \right] \quad [8]$$

This simplified form covers most furnace problems. Eckert has presented an analysis purporting to show that for cases in which gas temperature exceeds surface temperature the maximum error introduced by the use of Equation [8] instead of [7] is about 4 per cent; but the analysis is incomplete. If, for purposes of simplifying the analysis of error, one assumes $\epsilon_{S,P_cL_1}/\epsilon_{S,P_cL_2} = (P_cL_1/P_cL_2)^a$ and $\epsilon_{S,P_cL}/\mathrm{E}_{G,P_cL} = (T_S/T_G)^b$, it may be readily shown that the error in use of Equation [8] instead of [7] is given by

$$\frac{q/A]_{\text{Eq. [8]}} - q/A]_{\text{Eq. [7]}}}{q/A]_{\text{Eq. [7]}}} = \frac{1 - (T_G/T_S)^{0.65-a}}{(T_G/T_S)^{4+b} - (T_G/T_S)^{0.65-a}} \quad [9]$$

Fig. 3 indicates that b varies from +0.3 at 1000 F and high

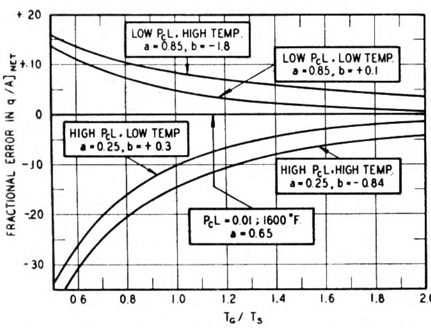

FIG. 4 FRACTION ERROR IN CALCULATION OF HEAT INTERCHANGE DUE TO CARBON DIOXIDE CAUSED BY USE OF APPROXIMATE EQUATION [8] INSTEAD OF [7]

P_cL's to —1.8 at 3000 F and low P_cL's; a varies from 0.85 at $P_cL = 0.005$ to 0.25 at $P_cL = 2.0$. Using Equation [9] and values of a and b limiting their range of importance, the percentage of error owing to use of Equation [8] has been calculated and is given in Fig. 4. It is apparent that caution should be employed in using the simplified form of equation.

Schack (14) has recently recommended an empirical equation for carbon-dioxide emission, obtained by fitting a simple power function of temperature and P_cL to the arithmetic mean of the Eckert and H.&M. data. Schack's equation is

$$E_G = 0.111\ \sqrt[3]{P_cL} \cdot \left(\frac{T}{100}\right)^{3.5} \text{ Btu per sq ft per hr} \ldots [10]$$

or

$$\epsilon_G = 0.644\ \sqrt[3]{P_cL} \Big/ \left(\frac{T}{100}\right)^{0.5} \ldots\ldots\ldots\ldots [11]$$

Inspection of Fig. 3 for which Equation [11] is proposed as a substitute makes it obvious that no such simple form of equation can be valid over a very wide range, since ϵ_G is proportional to the first power of P_cL at very low P_cL's and to a power of temperature varying from +0.3 at 1000 F and P_cL = 2 to —1.8 at 3000 F and P_cL = 0.01. However, in the temperature range 1000 to 2300 F and at values of P_cL between 0.02 ft atm and 1.0 ft atm Equation [10] gives values of emission which are within ten per cent of those calculated from the plot in Fig. 3. Outside this range the equation is greatly in error. For the absorption of black-body radiation by carbon dioxide Schack recommends the simplification already discussed—that $\alpha_{G,S}$ equals ϵ_G. His heat transfer equation then becomes

$$q/A]_{CO_2} = 0.111\ \sqrt[3]{P_cL} \left[\left(\frac{T_G}{100}\right)^{3.5} - \left(\frac{T_S}{100}\right)^{3.5} \right] \ldots [12]$$

This equation may lead to an error of ± ten per cent ± error given in Fig. 4, even in the range for which Schack recommended it. By factoring $(T_G - T_S)$ out of Equation [12], one may obtain an expression for an equivalent or pseudoheat-transfer coefficient.

This is

$$h_{CO_2\,\text{rad}} = 0.0039\ \sqrt[3]{P_cL}\ (T_{\text{ave}}/100)^{2.5}$$

in which T_{ave} is to an adequate approximation equal to the arithmetic average of gas and surface temperatures, degrees Rankine.

WATER VAPOR

The results on water vapor are less conclusive than those on carbon dioxide, and there is evidence that water-vapor emission at a fixed temperature and total pressure is not dependent solely upon P_wL, but rather on P_w and L separately. The chief investigations, summarized in Table 2, have been carried out in Danzig by Schmidt (7) and later by Eckert (10) under Schmidt's direction, and at The Massachusetts Institute of Technology, by Hottel and Mangelsdorf (9). Also at M.I.T. unpublished data by Eberhardt (11) and Brooks (12) have been obtained. Fig. 5 indicates the range of variables covered by the various investigators. Since Beer's law has not been verified for water vapor, L and P_w must appear as separate variables. The three independent variables P_w, L, and T produce a three-dimensional figure, isothermal planes through which are presented in Fig. 5.

Schmidt measured interchange between a thermopile at room temperature and its field of view, namely, a jet of pure steam issuing from a nozzle at a relatively high velocity (about 60 fps) and at temperatures between 250 and 1760 F with a flat blackened plate at room temperature behind the steam jet. The partial pressure was kept constant at one atmosphere, and the path length L was varied by the use of three different-sized nozzles and by the use of mirrors. No concentration traverse was made to determine boundary effects, but these must have been appreciable, considering the well-known injection effects of a jet discharging at high velocity into a relatively stagnant fluid. The longest path length of 0.596 ft was obtained by sighting the radiometer across a 6-cm jet onto a mirror, back across the jet to a second mirror, and finally across the jet a third time onto a blackened flat plate, with the result that the beam of radiation passes through six boundary layers. A temperature traverse across the jet indicated an irregular variation—as much as 140 F at a mean steam temperature of 1090 F. No control experiment to determine stray radiation is indicated. Since Schmidt used a background at room temperature only, he expressed his results as the pseudoemissivity ϵ_G' already defined. He calculated the

TABLE 2 SUMMARY OF EXPERIMENTAL MEASUREMENTS ON WATER VAPOR

Author	Path length L, ft	Partial pressure P_w, atm	Gas temp t_G, F	Back-ground temp t_S, F	Diluent gas	Knowledge of L	Uniformity of t_G	Stray radiation	Other comments
Schmidt, 1932	0.0325 0.0656 0.098 0.132 0.197 (0.394) (0.596)	1.0	250 to 1760	72 ± few deg (room temp)	None	No concentration traverse. Reliance on boundary of open jet. Two and 3 passes through jet to obtain L indicated in ( )	Poor. As much as 140 F irregular variation at higher temperatures (1000 F)	No control experiment indicated. Possibility of diffuse reflection of nozzle-wall radiation from flat-plate background	Radiometer sighted horizontally through vertical jet of steam discharging into cold air. Beam length doubled or trebled by use of mirrors, introducing edge-effects up to six times
Hottel and Mangelsdorf, 1935	1.68	0.005 0.010 0.020 0.040 0.080 0.167 0.500 1.00	70 to 1883	−297 and 70 ± 2 and 700 and 1840 and 2360	Dry CO_2-free air	Sharpness of gradient tested before each run by putting in CO_2 and analyzing. Runs only when gradient sharp. L good to 2%	Uniform to 2 F. Ends protected with hot, dry, CO_2-free air	1.3% of black-body radiation at gas temp	See comments in Table 1
Eckert, 1937	2.14	0.0305 0.061 0.122 0.260 0.534 1.00	250 to 750	75 (room temp)	N_2	See comments on Eckert's 2.14-ft furnace using CO_2, Table 1	Good. See Table 1	See Table 1, Eckert's 2.14-ft furnace. Possible error in method of applying correction	Apparatus has two nozzles, beam-entrance and beam-exit, at bottom of vertical gas furnace. Gold mirror inside furnace, plane mirror outside
	0.334	0.051 0.114 0.206 0.49 1.00	290 to 2300	Room temp	N_2	Same as above	Good. Same remarks as above	About 1% of black-body radiation at gas temp	See comments in Table 1 on Eckert's 0.334-ft furnace
	9.7	0.026 to 0.810	212	Room temp	N_2	See Table 1, Eckert's 9.7-ft furnace	Uniform in furnace. Gradient in ends occurring simultaneously with conc gradient. Varied from 70 to 212 F in 10 cm	See Table 1	Vertical furnace with mirror in top. No nozzles to produce sharp boundary. Background a flat surface. Possibility of fog formation at boundary from condensation of water vapor overflowing from apparatus

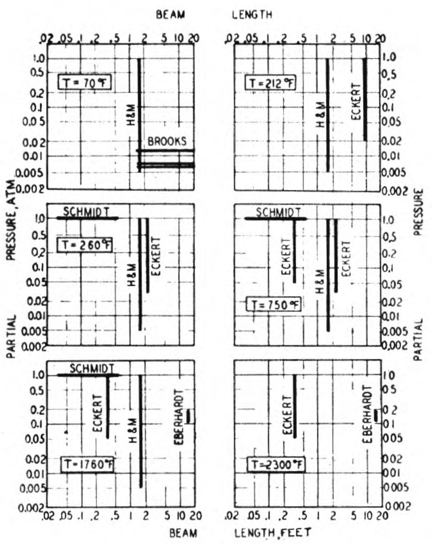

FIG. 5 RANGE OF VARIABLES COVERED IN STUDIES OF WATER-VAPOR RADIATION

emissivity of steam from monochromatic absorption measurements made by Hettner (15) on a pure steam column 3.5 ft long at 250 F, and used this value of emissivity to extrapolate his own results to high values of P_wL.

Hottel and Mangelsdorf measured the emission and absorption characteristics of steam-air mixtures, varying gas temperature, black-body temperature, and partial pressure and keeping the path length L constant at 1.68 ft. The temperature was uniform within 2 F over the entire path length. Before a run was made carbon dioxide was passed into the furnace to test the concentration gradient. When conditions of operation were such that the gradients were sharp, the system was thoroughly flushed free of carbon dioxide and water-vapor mixtures led into the furnace. A possibility of error arises here in that the conditions of operation which produced sharp concentration gradients with carbon dioxide might not have been maintained throughout the water-vapor runs. The stray radiation determined by a control experiment on dry carbon-dioxide-free air was about 1.5 per cent of black-body radiation, and was subtracted from all radiation measurements.

When the results of emission were compared with those of Schmidt, interesting discrepancies appeared. At small values of P_wL Schmidt's values of ϵ_G' are considerably larger than those calculated from Hottel and Mangelsdorf's measurements, while good agreement exists at high values of P_wL. These discrepancies indicated the possibility that P_wL was not a single independent variable but that increasing P_w produces a greater increase in emission than a corresponding increase in L. This possibility led Eckert under Schmidt's direction to make further investigations.

Eckert made measurements on three furnaces having different path lengths. One furnace was similar in design to that used by Hottel and Mangelsdorf, and confined gas having a path length of 0.334 ft. The gas temperature was varied between 290 and 2300 F, and the partial pressure of water vapor was varied between 0.051 and 1.0 atm, using nitrogen as a diluent gas. The stray radiation was small, about one per cent of black-body radiation, and the temperature was uniform throughout the gas column. No concentration traverse was made, and room air partially heated by the apparatus formed the windowless boundary between the radiating gas and the radiometer black-body system. Hence, possible sources of error include that of path length owing to possible uncertain boundary effect and that introduced by absorption and emission of radiation from the water vapor and carbon dioxide present in the air from the room.

A second vertical furnace contained a gold-plated mirror in its top in the heated zone. Radiation from a black body passed through one set of nozzles at the bottom of the furnace to the mirror, then back down through a second set of nozzles to a plane mirror, and thence into a radiometer. The path length was 2.14 ft, and the gas temperature was varied between 250 and 750 F. The emission from the hot gold mirror plus any stray radiation that was present was determined by a control experiment with dry nitrogen in the furnace, and was used to correct the measurements. Air from the room, as before, formed the windowless gas boundary. A concentration traverse at this boundary was not made, though one was calculated.

A third furnace also contained a gold mirror and confined gas with a path length of 9.7 ft. This furnace was heated by condensing steam to 212 F and the gas, preheated to the same temperature, entered the top of the furnace in back of the mirror and overflowed out of the bottom, no nozzle system to get a sharp gas boundary being used. A water-cooled flat surface formed the background for the radiometer. The radiometer used in all of Eckert's work appears to be of good design, and was flushed with dry nitrogen gas. Eckert calculated all his results as the pseudoemissivity ϵ_G'.

In Fig. 6 the measurements of the different investigators are plotted as a function of P_wL for three different temperatures. Schmidt's results are higher than any of the others and the discrepancies increase as P_wL is decreased, but the data are in better agreement at high temperatures than at low. Eckert's results on his shortest path length agree with Schmidt's when pure steam is used but decrease more rapidly than Schmidt's with decreasing P_wL and are somewhat higher than the values calculated from the data of Hottel and Mangelsdorf. Eckert's values calculated from the data on his 2.14-ft furnace lie considerably below both his own values from the 0.334-ft furnace and those of Hottel and Mangelsdorf. The values calculated from the data on Eckert's largest furnace, $L = 9.7$ ft (taken at one temperature only), are lower than all other results. After he had examined his own data Eckert concluded that Beer's law does not hold for water vapor. Working at 212 F where he had data at a common value of P_wL and T from three furnace lengths plus Schmidt's data at $P_w = 1$, Eckert concluded that the four different measured emissivities could be expressed as a single function, the product (emissivity at the value of P_wL in question and at $P_w = 1$ atm) $\times$ (power function of P_w), i.e.,

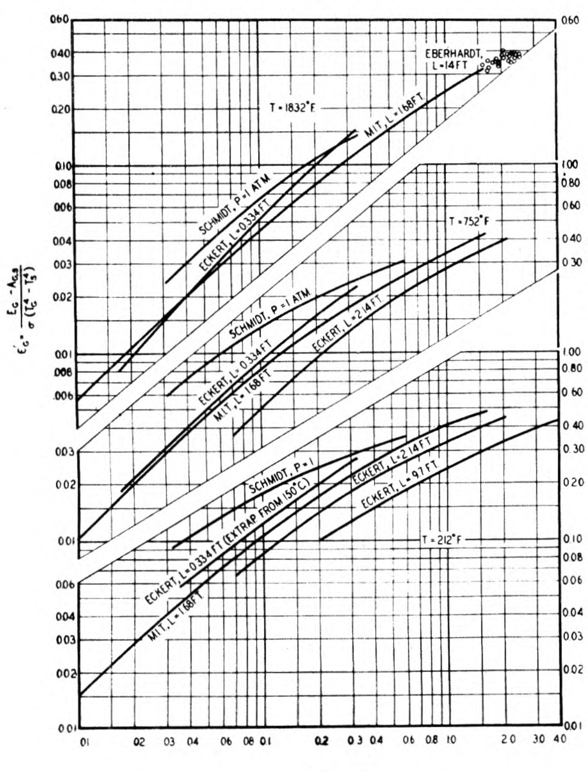

FIG. 6 COMPARISON OF DATA ON WATER-VAPOR EMISSION

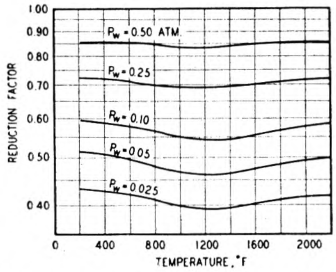

FIG. 7 ECKERT'S PROPOSED ALLOWANCE FOR EFFECT OF WATER-VAPOR PRESSURE ON EMISSION FROM WATER VAPOR AT CONSTANT P_wL

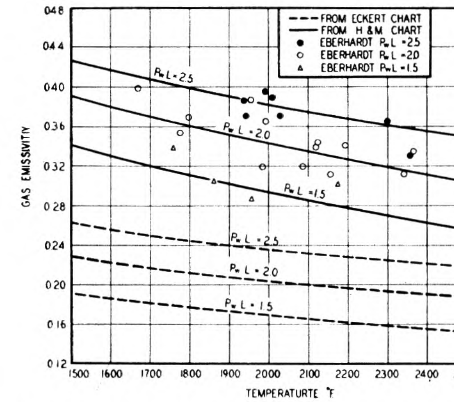

FIG. 8 COMPARISON OF DATA OF EBERHARDT ON WATER-VAPOR EMISSION WITH RECOMMENDED CURVES OF H.&M. AND OF ECKERT

$$\epsilon_G' = \epsilon_G']_{\text{at } P_w = 1} \cdot (P_w)^x \ldots\ldots\ldots\ldots\ldots [13]$$

where the power x is a function of temperature only. At temperatures usually attained in industrial furnaces Eckert had data at one path length only, $L = 0.334$ ft, and hence could not verify this equation at high temperatures. However, he assumed that it was valid and used Schmidt's data in conjunction with his own to determine the factor $(P_w)^x$ at all temperatures. He presented this factor graphically as a function of temperature; his graph is reprinted here as Fig. 7.

Eberhardt (11) made gas-emission measurements from a steel reheating furnace in an industrial plant. He sighted across the furnace through square openings 15 in. in diameter with a radiometer which was carefully designed for minimum stray radiation and which required an object at 20 ft equal to the diameter of its mirror to fill the field of view. Eberhardt determined the temperature and concentration of carbon dioxide at various points along the line of sight and found that both were uniform for a distance of 14 ft, with sharp gradients in temperature and concentration occurring simultaneously at the edges. The carbon dioxide concentration was determined by Orsat analysis while the water-vapor concentration was calculated from the results of careful analyses of the fuel gas. The total radiation of the gas owing to both carbon dioxide and water vapor was measured with the radiometer. From the known path length of 14 ft and measured partial pressure of carbon dioxide the contribution of that constituent was calculated from Fig. 3 and subtracted from the total measured radiation. The remainder was radiation owing to water vapor alone.[4] The temperature varied between 1670 and 2370 F, while P_wL varied between 1.5 and 2.5 ft-atm. The results were converted to show the relation between ϵ_G and temperature at three fixed

[4] A small correction for superimposed radiation (see later discussion) was applied.

values of P_wL, by assuming that over the small range of P_wL involved the slope of the ϵ_G — P_wL relation was the same as in the H.&M. data. These converted results appear as data points in Fig. 8, along with the recommended curves of Hottel and Mangelsdorf (solid lines) and Eckert (dashed lines) corresponding to the same values of P_wL of 1.5, 2.0, 2.5 ft-atm.

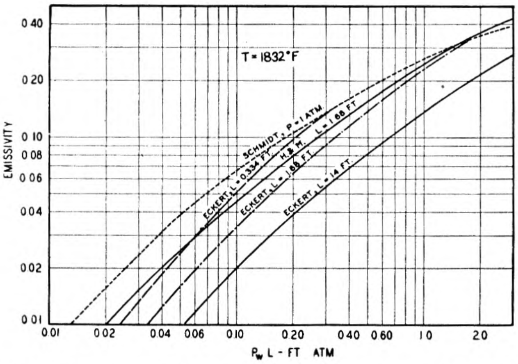

Fig. 9 Comparison of Recommended Curves for Water-Vapor Emission at 1000 C (1832 F)

It is apparent that Eberhardt's data are in excellent agreement with the solid lines, the maximum deviation of ±10 per cent appearing to be random and due to indeterminate experimental errors. It is also apparent that the data are considerably higher than the curves recommended by Eckert, the calculation of which curves involves a correction downward owing to the low P_w even though P_wL is high.

Eberhardt's data were also corrected to a common temperature of 1832 F and plotted as points in Fig. 6. An extrapolation of the curve of Hottel and Mangelsdorf would pass through these points even though Eberhardt's data were obtained on a path length nine times as great and at correspondingly lower P_w's.

The emissivity of water vapor as a function of P_wL at 1832 F (1000 C) for three path lengths, L = 0.334 ft, L = 1.68 ft, and L = 14 ft, according to the recommendations of Schmidt, Hottel and Mangelsdorf, and Eckert is plotted in Fig. 9. Since both Schmidt and Hottel and Mangelsdorf assumed the validity of Beer's law, one curve suffices for all three path lengths. Eckert's recommendation yields three different curves when his correction for the effect of partial pressure is applied.[5] For the longest path length, L = 14 ft, values of P_w for the small values of P_wL were out of the range covered by Eckert's plot of correction factors. The values for the correction factor at 1832 F, according to Eckert's plot, can be calculated from the relation: factor = $(P_w)^{0.242}$. It is apparent that at the long path length of 14 ft Eckert's recommendation is quite low. This strongly suggests that his correction for pressure is excessive, at least for small partial pressures or high temperatures. Eckert himself suggests that the deviations from Beer's law which he found are due to association of water-vapor molecules. An increase in the partial pressure increases the association, and if associated water vapor has a higher emissivity than the unassociated state, then increasing the partial pressure at constant P_wL should increase the emissivity. At very low partial pressures or at high temperatures the association of water vapor becomes negligible. Therefore, if association is the cause of the deviation from Beer's law the law should be valid at very low partial pressures, rarely encountered in industrial practice, and at high temperatures such as encountered in most industrial furnaces. The agreement between Hottel and Mangelsdorf and Eberhardt as well as the fairly good agreement between Eckert's measurements at his highest temperatures on the 0.334-ft furnace and those of Hottel and Mangelsdorf indicate that this is the case.

Some measurements of the emission and absorption of radiation from atmospheric air containing moisture at room temperatures have been made by Brooks (12). A sensitive radiometer was sighted through laboratory air upon either of two black bodies, one filled with liquid air and the other with hot water. The distance between the black body and the radiometer was varied from 1.5 to 20 ft. Calibration of the radiometer was accomplished by permitting the radiometer to view the black body directly. Brooks made an attempt to correct for errors owing to inability to eliminate air with moisture and CO_2 between the radiometer and black body in the calibration of the radiometer, and for errors owing to boundary effects at the black bodies and radiometer. The boundary effect was particularly bad for the liquid-air-cooled black body as indicated by a visible fog which issued forth from it. Brooks' assumptions regarding the nature of these boundary effects were not sufficiently accurate to eliminate errors. About 15 per cent of the radiation was due to carbon dioxide in the air. When the radiation owing to carbon dioxide is subtracted from the measurements the remainder, radiation from water vapor, is about 40 per cent higher than calculations based on an extrapolation to room temperature of the H.&M. data at a P_L of 0.01 ft-atm, where the path length L was the same for both investigators. At Brooks' longest path length (20 ft) his measurements are in agreement with the H.&M. extrapolation. Brooks' measurements, while subject to considerable error, confirm Eckert's conclusion that at room temperature an increase in path length at constant partial pressure increases water-vapor emission less than does corresponding increase in partial pressure at constant path length. They also indicate that both the Eckert and the H.&M. extrapolations of emission measurements to room temperatures are possibly low. These discrepancies are of no consequence in furnace calculations.

Margaret Fishenden (16) in 1936 published the results of some radiation measurements on the products of combustion of city gas. Measurements on gas varying from 400 to 1600 F, at P_wL = 0.205 and P_cL = 0.075, yielded results from 5 to 21 per cent higher than calculations based on the H.&M. data and from 9 to 29 per cent higher than the Eckert data. The discrepancy in each case increased with temperature. Owing to stray radiation, uncertain path length, and temperature gradients along the line of sight of her radiometer Miss Fishenden's measurements are subject to considerable error. Recently she made measurement (17), yet to be published, on the absorption of radiation from hot black bodies by low-temperature steam-air mixtures, measurements which confirmed the inadequacy of Beer's law at low temperatures.

All these results point to the conclusion that at low temperatures Beer's law is not valid, and that some kind of correction factor must be used in conjunction with a single family of curves involving the three variables, emissivity, temperature, and P_wL. However, Eckert's proposed correction factor seems to be excessive at low temperatures as well as invalid at high temperatures. At very high values of P_wL, theory indicates that gas emissivity approaches unity and hence is independent of P_w at constant P_wL. Likewise if association is the cause of the experimental deviations from Beer's law then gas emissivity is independent of P_w at constant P_wL at very low values of P_w. Eckert's correction factor, a function of P_w and independent of P_wL, is such that unit emissivity cannot be attained at infinite P_wL unless P_w = 1, and it does not become constant when P_w is very small. One concludes that the correction factor is therefore theoretically unsound and is in error at low values of P_w and high values of P_wL.

[5] It is to be remembered that at this temperature Eckert has data at only one path length, namely, at L = 0.334 ft.

A more logical type of correction factor to be used in conjunction with the charts already published would be one which corrected P_wL instead of emissivity. One such form of correction might be

$$P_wL]_{P_w=1} = P_wL]_{\text{actual}}\,(1 - K + K \cdot P_w) \ldots\ldots [14]$$

This form of correction conforms to the conditions that emissivity be unity and independent of P_wL at infinite P_wL and that the correction become independent of P_w at very small values of P_w. Existing data are not sufficient to determine the factor K in the foregoing equations as a function of temperature. Eckert's data at 750 F and $L = 0.334$ ft (his most reliable furnace) and that of Hottel and Mangelsdorf can be brought together by Equation [14] with $K = 0.35$. This would give

$$\left(\begin{array}{l}P_wL \text{ for use with} \\ \text{charts based on} \\ P_w = 1\text{, such as} \\ \text{Schmidt's}\end{array}\right) = (P_wL)\,]_{\text{actual}} \cdot (0.65 + 0.35\,P_w) \,.\,. [15]$$

and

$$\left(\begin{array}{l}P_wL \text{ for use with} \\ \text{chart based on} \\ L = 1.68\text{—that of} \\ \text{H.\&M. (Fig. 10)}\end{array}\right)$$

$$= 1.56 \left\{\sqrt{1 + 1.97\,(P_wL)\,]_{\text{actual}}\,(0.65 + 0.35\,P_w)} - 1\right\} \,.\,. [16]$$

Since many other functions besides (14) exist which conform to the limits imposed by theory, equations such as [15] and [16] are certain to be replaced by better recommendations when adequate data become availale. Such an experimental study is now under way at M.I.T.

With the extensive and somewhat conflicting data on water-vapor emission in mind, particularly the Eberhardt data taken in the range of P_w and T encountered in furnace practice, the authors make tentative recommendations for calculations: For temperatures above 1200 F use the chart, Fig. 10, based on the H.&M. data, without any correction for deviations from Beer's law. For lower temperatures a correction should probably be used, and [16] is recommended temporarily. In the vicinity of 212 F Eckert's correction may be used, though we feel it overcorrects.

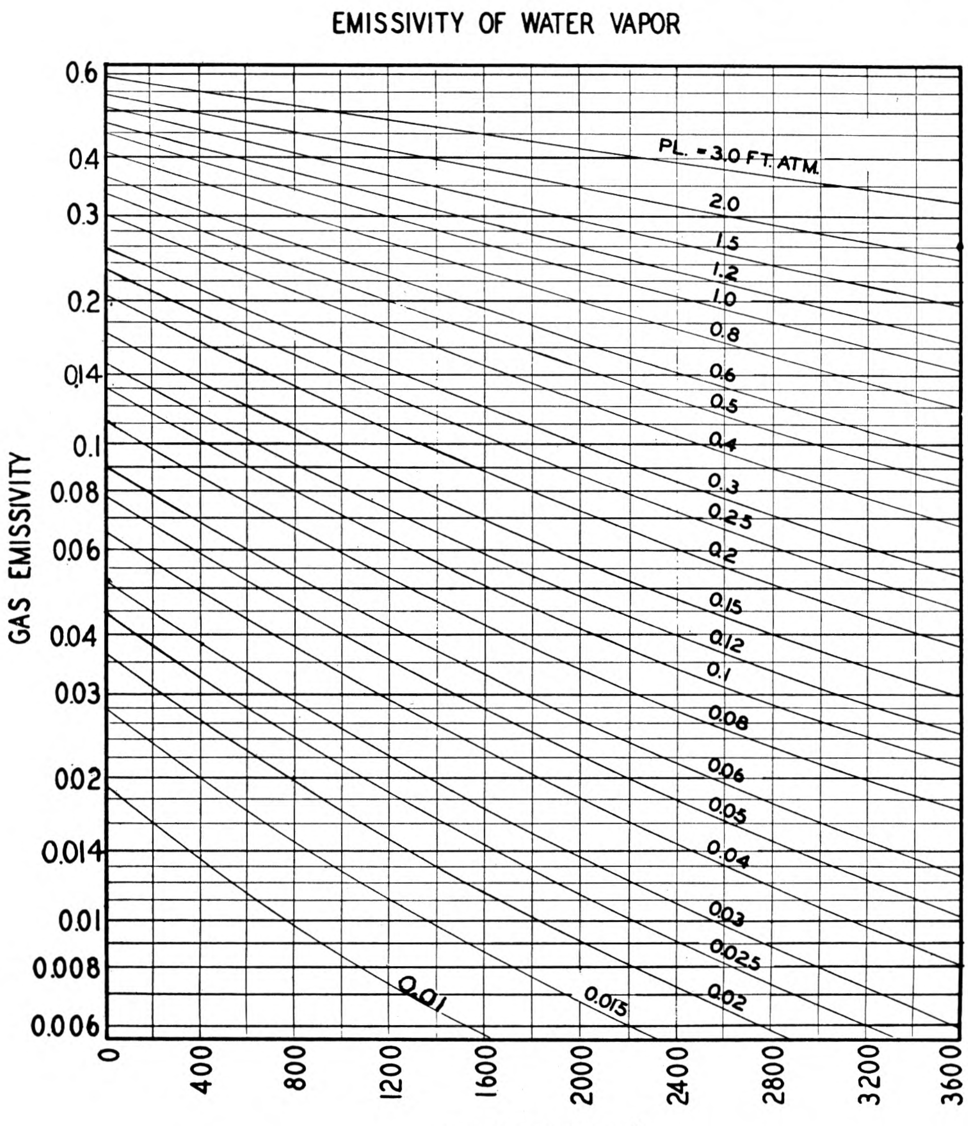

Fig. 10 Recommended Working Plot of Water-Vapor Radiation

The only measurements on the total absorption of black-body radiation by water vapor are those of Hottel and Mangelsdorf. They found that absorption by a gas at constant P_wL of radiation from a black body at a given temperature T_S is independent of gas temperature. Hence the absorption is equal to the emission the gas would exhibit if at the temperature T_S of the body. The

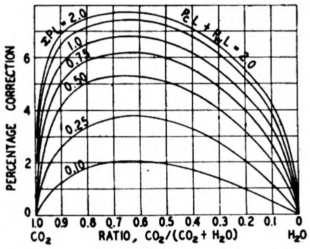

Fig. 11 Correction Due to Superimposed Radiation From Carbon Dioxide and Water Vapor

equation for radiant heat interchange between gas containing water vapor and its bounding surfaces is then of the form of the simplified carbon dioxide equation

$$q/A]_{H_2O} = 0.1723\left[\epsilon_{G,P_wL}\left(\frac{T_G}{100}\right)^4 - \epsilon_{S,P_wL}\cdot\left(\frac{T_S}{100}\right)^4\right]..[17]$$

where ϵ_G and ϵ_S are read from Fig. 10 at t_G and t_S and at a common P_wL.

Schack (14), using the arithmetic mean of the H.&M. data and those of Eckert, has developed equations for water-vapor emission similar to those already presented for carbon dioxide. He assumes that the emission increases with partial pressure at constant P_wL according to Eckert's recommendation. His equation is

$$q/A]_{H_2O} = 1.08\ P_w^{0.8}L^{0.6}\left[\left(\frac{T_G}{100}\right)^3 - \left(\frac{T_w}{100}\right)^3\right] \text{Btu per sq ft per hr} \quad ....[18]$$

Between 900 and 2400 F and $P_wL = 0.02$ to $P_wL = 1.0$ ft-atm this equation gives values of emission that are from 15 per cent lower to 15 per cent higher than those calculated from Fig. 10 when the path length is 1.68 ft. At other path lengths larger deviations may be expected due to the probably excessive correction for effect of P_w. The equation is recommended only where errors of 20 per cent can be tolerated. As with carbon dioxide, Equation [18] may be converted to yield a pseudo-heat-transfer coefficient by factoring out $(T_G - T_w)$. It is

$$h]_{H_2O\ rad} = 0.0324\ P_w^{0.8}L^{0.6}T_{ave.}^2 \ ....[19]$$

Mixtures of Carbon Dioxide and Water Vapor

The total emission from a gas containing both carbon dioxide and water vapor is less than the sum of the emissions due to the carbon dioxide and to water vapor, each evaluated as though the other were not present, because the emission and absorption bands of the two gases overlap. The magnitude of the difference between actual emission and the evaluated sum is a function of PL, temperature, and relative concentrations of the two gases. This difference is designated by the symbol K defined and by the following equation

$$E_{c+w} = E_c + E_w - K$$

Hottel and Mangelsdorf made the only extensive measurements of emission and absorption of radiation by mixtures of carbon dioxide, water vapor, and air. However, the accuracy of the determination of K was low since it involved differences of quantities of similar magnitude.

Eckert (10) made calculations of this difference K from the monochromatic absorption data. These calculations are in fair agreement with the measurements of Hottel and Mangelsdorf. This correction term varies from zero at small PL's to a maximum of 10 per cent of the total radiation at the highest PL's for which emission measurements have been made. Hence it is not necessary to know K accurately when probable errors in the existing recommendations for gas emission are considered. Fig. 11 presents the difference factor expressed as the percentage K' by which $(E_c + E_w)$ must be reduced to give E_{c+w}, plotted as a function of the ratio $\dfrac{P_c}{P_c + P_w}$ for several values of $P_cL + P_wL$. Since K' varies with temperature and the chart is presented for use at all temperatures, K' may be 50 per cent in error but this introduces at most a 4 per cent error in the calculation of the total radiant heat transmission from gases containing both carbon dioxide and water vapor.

Effect of Gas Shape

The use of a definite value of P_GL in calculating interchange between a gas mass and its bounding surface presupposes a gas shape for which path length L is constant in all directions through the gas. The only shape for which that limitation is applicable is a hemisphere of gas radiating to a spot on the center of its base. For actual gas shapes a suitable mean value of L must be obtained, the radius of an "equivalent" hemisphere. This problem has been presented in some detail (3), and for various gas shapes of industrial importance mean values of L have been given. Table 3, column 2, gives references for the various shapes studied. Hottel and Port (21) have shown that at very low values of P_GL where E_G (or ϵ_G) approaches proportionality

TABLE 3 BEAM LENGTHS FOR GAS RADIATION

Shape	Bibliographic references	Characterizing dimension, D	Factor by which D is multiplied to obtain mean beam length L: When $P_GL = 0$	For average values of P_GL
Sphere	(18), (3)	diam	2/3	0.60
Infinite cylinder, radiating to walls	(18), (3)	diam	1	0.90
Rt. circ. infinite cylinder, rad to spot on center of base	(10)	diam	...	0.90
Rt. circ. cylinder, ht = diam; rad to whole surface	.....	diam	2/3	0.60
Same; rad to spot on center of base	(10)	diam	...	0.77
Infinite cylinder of half-circular cross section; rad to spot on center of flat side	(10)	radius	...	1.26
Space between inf. parallel planes	(3), (20)	separating distance	2	1.8
Cube	(3)	edge	2/3	0.60
1 × 2 × 6 rectangular parallelopiped, radiating to		shortest edge		
2 × 6 face	(3), (21)		1.18	1.06
1 × 6 face	(21)		1.24	
1 × 2 face	(21)		1.18	
all faces	(21)		1.20	
Space outside infinite bank of tubes with centers on equilateral triangles; tube diam = clearance	(3), (10)	clearance	3.4	2.8
Same, except tube diam = one half clearance	(3), (10)	clearance	4.45	3.8
Same, except tube centers on squares, tube diam = clearance	(10)	clearance	4.1	3.5

to P_GL, the value of L for *any* gas shape radiating to its bounding walls approaches as a limit the simple expression, four times the mean hydraulic radius of the shape, i.e., four times the gas volume divided by the area of the bounding walls. For the range of P_GL encountered in practice, the mean path length L is always less. A study of rectangular parallelopipeds of widely varying dimension ratios led to the conclusion that a satisfactory approximation consists in taking 85 per cent of the limiting value, four times mean hydraulic radius. This simple rule works quite well for other gas shapes, as borne out by a comparison of the last two columns of Table 3, giving values of L for various gas shapes for $P_GL = 0$ and for P_GL in the industrially important range. Since in the latter range E_G (or ϵ_G) varies as about the 0.3 power of P_GL, a 10 per cent error in choice of L produces only a 3 per cent error in the calculation of heat transmission.

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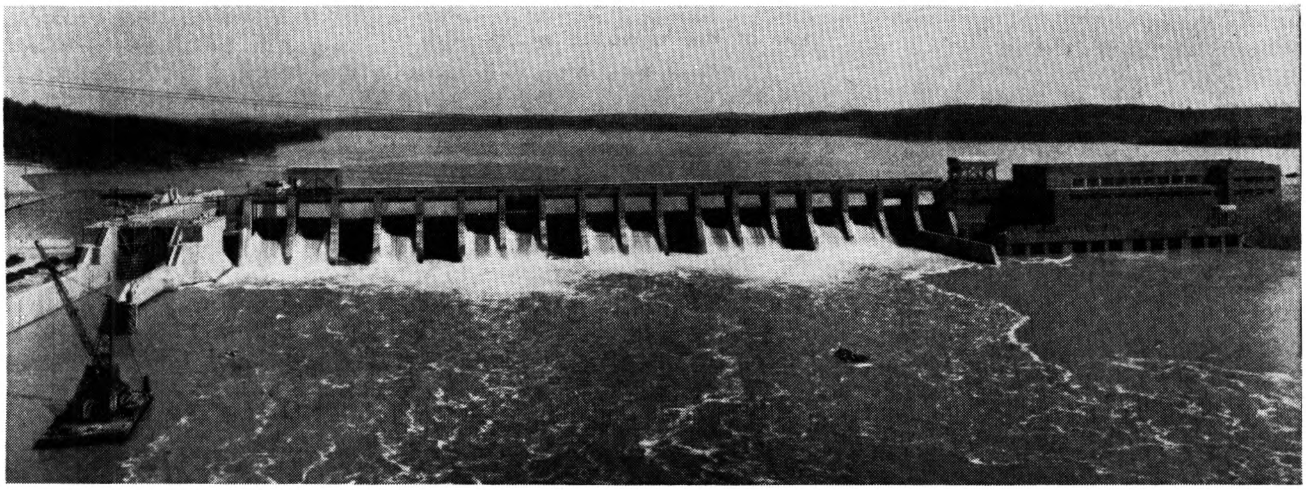

FIG. 1 CHICKAMAUGA DEVELOPMENT—A TYPICAL MULTIPURPOSE PROJECT

Kaplan Turbine Installations of the Tennessee Valley Authority

By G. R. RICH[1] AND J. F. ROBERTS,[2] KNOXVILLE, TENN.

Eight Kaplan turbines recently installed in the Pickwick Landing, Guntersville, and Chickamauga plants of the Tennessee Valley Authority have a total capacity of 300,000 hp at rated head, and in physical size are among the largest constructed. The paper reviews the function of Kaplan turbine plants in the Tennessee River development as a whole, the determination of turbine requirements, the power-station arrangement, and noteworthy features of turbine design, construction, and erection.

GENERAL DESCRIPTION

THE development of the Tennessee River, in accordance with the terms of the Tennessee Valley Authority Act, comprises a series of multipurpose projects for the provision of a channel for 9-ft navigation in the river from Paducah, Ky., to Knoxville, Tenn., the control of destructive floodwaters in the Tennessee and Mississippi River basins, and the generation of hydroelectric power.

The prescribed navigation improvement is accomplished by means of locks and a continuous succession of pools, the minimum levels of which are governed by the drawdown which will afford a minimum but adequate navigation channel at the next project upstream, while the surcharge levels for flood control were fixed with reference to the resultant damage to cities, railroads, highways, and land. To augment the flow of the Tennessee River during the dry season and for the retention of headwater floods, storage projects are also provided at strategic locations on certain of the tributary streams.

The cycle of operation of the reservoir system is dictated by flood-control considerations during the winter and spring months, and by navigation and power requirements during the summer and autumn. This mode of operation is feasible because large floods on the Tennessee River, as at Chattanooga and below, occur only between the middle of December and April and are due primarily to the transit of typical seasonal storms along the river basin, starting at the western end and moving in a direction from southwest to northeast.

Accordingly the basic program is to deplete the storage reservoirs on the tributaries to low level by December 15, and to retain water in these reservoirs at safe rates in the interval between December 15 and April 15 and, subsequently, at higher rates governed by stream flow and use. On the other hand, the main river reservoirs are operated during the flood season with particular reference to immediate needs at Cairo, Ill., and the lower Mississippi River, and are depleted in the fall to a level consistent with good navigation; surplus water from the river reservoirs being discharged between flood crests on the Mississippi.

The major physical features of the various component projects are summarized for convenient reference in Fig. 2 and Table 1. With the completion of Kentucky, Watts Bar, and Coulter Shoals, the development will afford 650 miles of high-grade waterway, 9,000,000 acre-ft of flood storage (sufficient to reduce flood crests on the Mississippi 2 ft between Cairo and the Arkansas River), and 1,800,000 kw ultimate hydroelectric capacity.

DETERMINATION OF PLANT CAPACITY

In project planning, the ultimate capacity to be provided at each individual plant should be determined in advance, so that power-station intakes and draft-tube foundations may be provided during the initial construction to accommodate the ultimate number of units required.

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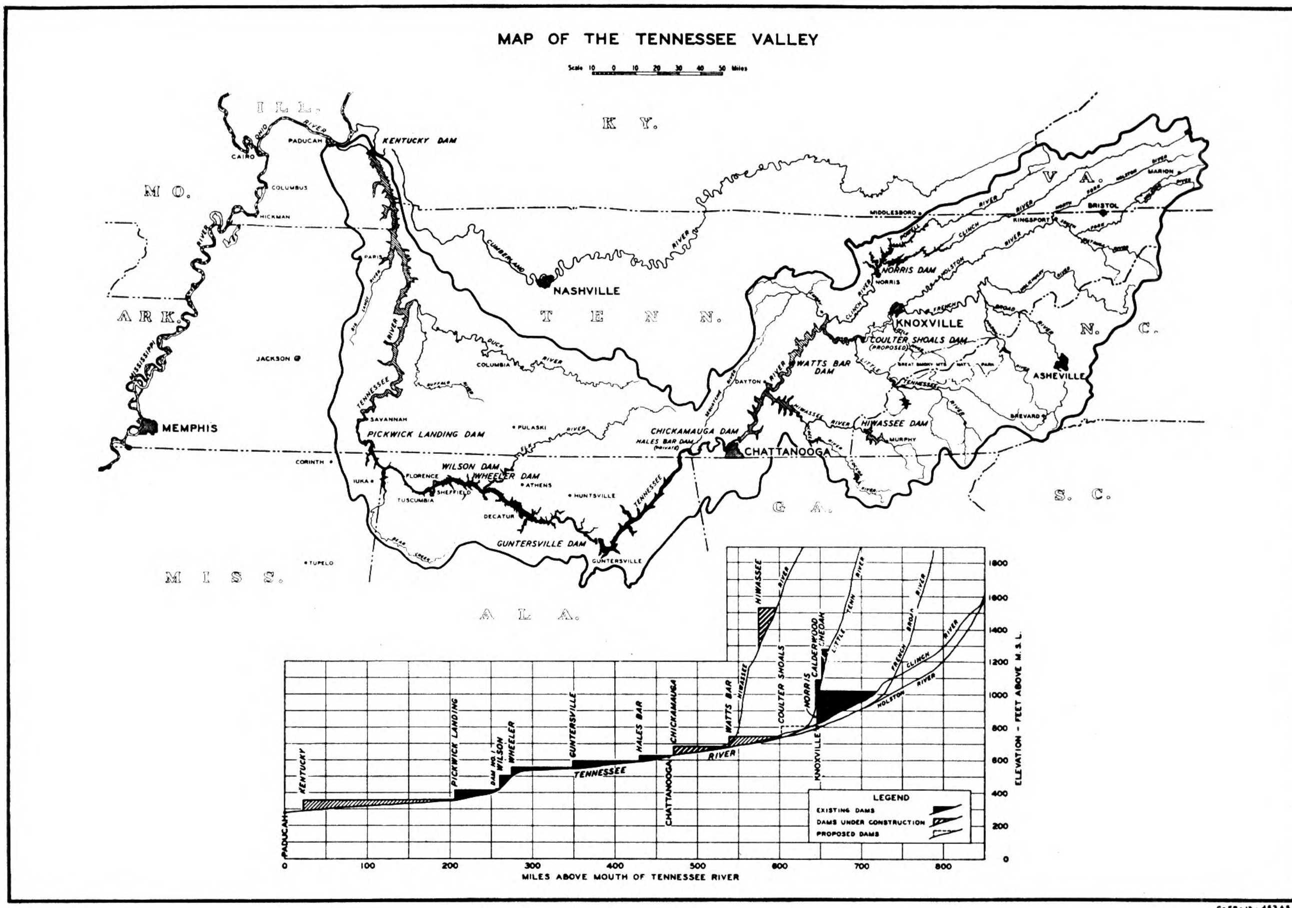

FIG. 2 MAP OF THE TENNESSEE VALLEY

With the regulated flow at each plant established by operation of the reservoir system in the combined interest of flood control, navigation, and power generation, the determination of ultimate plant capacity for any particular project must be approached from the standpoint of the power system as a whole. The method employed has been to determine the ultimate primary or continuous energy output of the entire system of plants, based upon the regulated flow at each site, and to establish the corre-

TABLE 1 PROJECT FEATURES

	Navigation		Reservoir						Power					
Project	Size of Lock Chamber (feet)	Maximum Lift of Lock (feet)	Area at Top of Gates (acres)	Volume at Top of Gates (acre-feet)	Controlled Flood Storage (acre-feet)	Length of Spillway (feet)	Spillway Capacity (second-feet)	Backwater Length (miles)	Rated Head (feet)	Head for Best Efficiency (feet)	Present[4] Plant Capacity (kw)	Ultimate[4] Plant Capacity (kw)	Effective Capacity During High Flow - 1926-1927 Flood (kw)	Type of Turbines
Kentucky[1]	110x600	73	256,000	6,100,000	4,570,000	960	1,100,000	184.4	48	51	--	160,000	74,000	Kaplan
Pickwick Landing	110x600	63	46,800	1,091,000	418,000	880	910,000	52.7	43	56	72,000	216,000	36,000	Kaplan
Wilson	60x300[2]	90	16,200	600,000	--	2,212	629,000	15.5	95 & 92	95 & 92	184,000	444,000	419,000	Francis
Wheeler	60x360	53	68,300	1,150,000	429,000	2,400	687,000	74.1	48	48	64,800	259,200	245,000	Propeller
Guntersville	60x360	45	70,700	1,019,000	282,000	720	625,000	82.1	36	37	72,900	97,200	69,000	Kaplan
Hales Bar	60x267	37	5,800	126,000	--	1,200	--	39.8	35	35	50,500	50,500	20,000	Francis
Chickamauga	60x360	58	37,200	655,000	353,000	720	600,000	59.0	36	48	81,000	108,000	67,000	Kaplan
Watts Bar[1]	60x360	70	41,600	1,132,000	370,000	800	550,000	72.4	52	57	90,000	150,000	150,000	Kaplan
Coulter Shoals[1]	60x360	70	13,900	336,000	130,000	800	500,000	50.0	65	70	--	96,000	64,000	Kaplan
Norris	--	--	40,160	2,567,000	2,020,000	300	54,000	72.0	165	180	100,800	100,800	100,000	Francis
Hiwassee	--	--	6,280	438,000	365,000	224	130,000	22.0	190	200	57,600	115,200	115,000	Francis
Blue Ridge[3]	--	--	3,290	197,500	183,000	110	55,000	10.0	147	--	20,000	20,000	20,000	Francis
Ocoee No. 1[3]	--	--	1,380	76,700	25,900	362	--	7.5	110	--	18,000	18,000	18,000	Francis
Ocoee No. 2[3]	--	--	--	--	--	--	--	--	250	--	18,800	28,200	18,000	Francis
Great Falls[3]	--	--	2,290	55,100	49,900	450	150,000	--	142	--	29,400	29,400	29,000	Francis

1 Under construction.
2 Two lock chambers.
3 Acquired by purchase of completed projects; dependable flood storage not fully determined.
4 Generating capacities are based upon actual performance which exceeds guaranteed performance. Table 2 gives guaranteed capacities.

TABLE 2 TURBINE DESIGN DATA

	Guntersville	Chickamauga	Pickwick Landing
Maximum headwater elevation - feet	605	701	430
Normal headwater elevation - feet	594	682	413
Minimum headwater elevation - feet	590	673	408
Normal tailwater elevation - feet	555	635	356
Minimum tailwater elevation - feet	550	628	356
Maximum tailwater elevation - feet	600	697.5	422
Maximum head - feet (net)	42	52	60
Minimum head - feet (net)	5	3.5	5.5
Head for best efficiency and speed - feet	37	48	56
Rated head - feet (net)	36	36	43
Rated horsepower	34,000	36,000	48,000
Maximum horsepower	39,000 at 39 ft	42,000 at 40 ft	55,000 at 47 ft
Generator continuous rating - 60° C, 0.90 power factor - kva	27,000	30,000	40,000
Generator capacity - 60° C, continuous at 0.90 power factor - kw	24,300	27,000	36,000
Rated speed - rpm	69.2	75	81.8
Specific speed at rating - rpm	145	161	163
Head for runaway speed - feet	42	52	60
Runaway speed - rpm	189	218	200
Turbine manufacturer	S. Morgan Smith	Baldwin Southwark	Allis-Chalmers
*Generator manufacturer	General Electric	Allis-Chalmers	Westinghouse
Number of units, present	3	3	2
Number of units, ultimate	4	4	6
Value of sigma at rating	1.12	1.32	0.90
Diameter of runner at throat - inches	265	264	292
Number of blades	5	5	6
Blade adjustment	Automatic oil pressure	Automatic oil pressure	Automatic oil pressure
Rated discharge - cfs at rating	9,500 at 36 ft	10,200 at 36 ft	11,200 at 43 ft
Peripheral coefficiency at rating	1.66	1.79	1.98
Peripheral speed, turbine, runaway fpm	13,100	15,070	14,900
Peripheral speed, turbine, normal fpm	4,810	5,180	6,180
Discharge, coefficient of gates as orifice = C in $CA\sqrt{2gh}$ at rating from model test	.404	.430	.471
Elevation centerline of distributor - feet	558	632	358.583
Elevation centerline of runner - feet	549.875	623.04	349.50
Spacing center to center of units - feet	78'0"	80'0"	80'0"
Weight of rotating element - turbine and generator - pounds	973,000	936,000	917,000
Head for maximum hydraulic thrust - feet	42	52	60
Maximum hydraulic thrust - pounds	914,000	1,052,000	1,500,000
Total load on thrust bearing	1,887,000	1,988,000	2,416,000
Type of thrust bearing	Kingsbury	Kingsbury	Kingsbury
Capacity of thrust bearing - pounds	2,000,000	2,075,000	2,700,000
Type of generator setting	Umbrella	Umbrella	Umbrella
WR^2 of turbine and generator (lb-ft^2)	81,200,000	81,700,000	80,500,000
Type of scroll case	Concrete	Concrete	Concrete
Governor manufacturer	Woodward	Woodward	Allis-Chalmers
Gate servomotor:			
Capacity - ft-lb (oil pressure 300 p.p.s.i.)	368,000	517,000	772,000
Operating pressure - p.p.s.i.	250-300	250-300	250-300
Minimum time to close gates - seconds	8	8	8
Speed droop adjustment	5%	5%	5%
Blade servomotor:			
Capacity - ft-lb (oil pressure 300 p.p.s.i.)	610,000	635,000	887,000
Minimum time to open or close blades - seconds	10	10	10
Maximum time to open or close blades - seconds	40	40	40
Type of draft tube	Elbow	Elbow	Elbow
Splitter	Yes	No	Yes
Velocity through intake trashracks, gross area, fps at rated discharge	4.1	4.2	5.0
Velocity at draft-tube exit - fps at rated discharge	7.1	7.4	7.2
Scroll case:			
Clear width between main piers - feet	66'0"	66'0"	69'0"
Thickness main piers - feet	12'0"	14'0"	11'0"
Number intermediate piers	2	2	2
Thickness intermediate piers	6'6"	6'6"	6'6"
Offset centerline of turbine from centerline of scroll case	5'0"	3'0"	5'3"
Height of intake openings - feet	43'0"	46'0"	40'0"
Draft tube:			
Clear width between main piers	66'0"	66'0"	69'0"
Thickness main piers	12'0"	14'0"	11'0"
Number intermediate piers	2	2	2
Thickness intermediate piers	6'0"	6'0"	6'6"
Elevation lowest point of tube	497	572	294
Horizontal length of draft tube	85'0"	85'0"	85'0"
Height of draft-tube openings - feet	24.54	25.54	23.0

*All generators 3-phase, 60-cycle, 13,800-volt

Note: Generating capacities listed in this table are guaranteed capacities. The actual capacities obtained in operation are given in table 1.

sponding ultimate peak capacity required for the entire system, on the basis of an assumed annual load factor of 60 per cent, together with an allowance of 15 per cent for machine outages and an additional allowance to compensate for the loss of capacity from decreased head at the river plants during extreme floods. Additional capacity at 100 per cent load factor is also provided to carry such high-grade secondary energy as may be available 75 to 80 per cent of the time.

The minimum ultimate capacity assigned to any particular plant must be at least large enough to utilize the entire regulated flow at the site at a constant rate of demand or, in other words, 100 per cent load factor. The additional system peak capacity required for variable load demand is apportioned among the various projects largely in inverse ratio to the unit incremental capacity cost. The general effect of this method of apportionment is to provide peaking capacity at the plants having the higher heads, although consideration must be given to possible pondage limitations and to the reduction of head during extreme floods.

Selection of Turbines

In the integrated power system, the Pickwick Landing, Guntersville, and Chickamauga plants will be operated at high capacity factor during periods of ample flow in the main river, and the tributary plants at Norris and Hiwassee will, as a general rule, be operated intermittently. Conversely, during periods of low flow in the main river, the tributary plants will operate at high capacity factor, while main river plants like Pickwick Landing, Guntersville, and Chickamauga will be assigned to service at relatively low capacity factor. As shown in Table 2, the generating units for these plants will be required to operate over a great range of load and head conditions; and, since the maximum head in all cases is less than 60 ft, movable-blade propeller turbines of the Kaplan type, with their characteristic high efficiency over a wide gate range, are ideally suitable.

In the case of Chickamauga, which is typical of the run-of-river plants, the estimated continuous power available from the regulated flow is about 50,000 kw, and the ultimate installation to meet system-capacity requirements about 100,000 kw. In view of the immediate power-market conditions and present and future operating characteristics, it was decided that an ultimate installation of four 25,000-kw units, with three units installed initially, would afford the required degree of flexibility. In field operation the turbines exceed the guaranteed ratings so that the four units will have an actual capacity of 108,000 kw.

Referring again to Table 2 and Fig. 3, the basic requirement of the turbine purchase specification is a machine capable of 36,000 hp at a head of 36 ft, or during flood periods, with best efficiency and speed selected for 48 ft head, which obtains during the major part of the period of normal operation. With reference to cavitation, the bidder was required to state guaranteed horsepower outputs for the schedule of headwater and tailwater elevations given in Table 3.

At an early stage of the project design and well in advance of inviting bids, the manufacturers were given complete information pertaining to head and tailwater elevations and other governing physical features, and were requested to comment on pre-

TABLE 3 CAVITATION AND EFFICIENCY TESTS AT BALDWIN LOCOMOTIVE WORKS OF CHICKAMAUGA TURBINE

(Runner, 5-blade, 264 in. 75 rpm; 11-in. model runner for cavitation tests; 16-in. model runner for efficiency tests)

Tailwater Elevation	Headwater Elevation	Head Feet	Guaranteed Horsepower	Allowable Horsepower from Cavitation Tests	Allowable Horsepower from Model Efficiency Tests
630	674	44	42,000	44,100	49,500
630	678	48	42,000	45,900	56,300
630	682	52	42,000	48,100	63,300
634	674	40	40,750	41,500	42,600
634	678	44	42,000	46,500	49,500
634	682	48	42,000	49,900	56,300
638	672	34	33,200	35,100	35,500
638	677	39	40,000	42,000	41,300
638	682	44	42,000	48,000	49,500

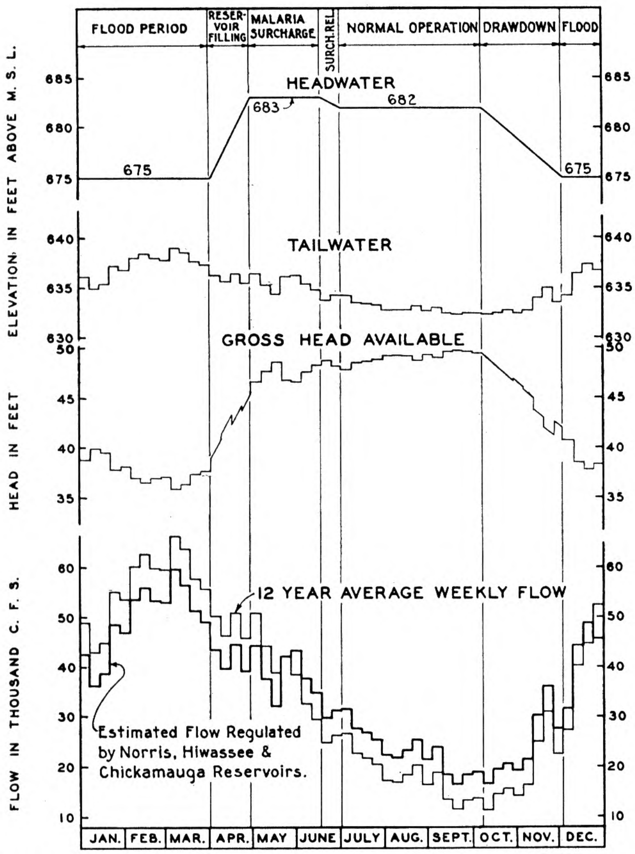

FIG. 3 RESERVOIR OPERATION DIAGRAM OF CHICKAMAUGA PROJECT

liminary turbine specifications. By thus working in close cooperation with the turbine manufacturers, it was found possible definitely to specify the speed of operation, the elevation of the center line of the distributor, the unit spacing, and the depth of draft tube, so as to afford a fixed common basis for bidding, and yet allow each bidder reasonable latitude for employing his own characteristic design.

Model Testing

To avoid the difficulty of measuring large prototype discharges and to insure an ample margin of safety against excessive cavitation, acceptance of the turbines with reference to efficiency and cavitation was based upon laboratory tests of homologous model runners complete with homologous scroll cases and draft tubes. Acceptance with respect to capacity was based upon actual prototype performance.

The Moody formula was employed to compute the efficiency of the prototype turbines from the model-test results

$$\mathrm{Eff}_2 = 100 - (100 - \mathrm{Eff}_1)\frac{(D_1)^{1/4}}{(D_2)}\frac{(H_1)^{0.01}}{(H_2)}$$

D_2 = diameter of prototype runner
D_1 = diameter of model runner
H_2 = head on prototype turbine
H_1 = head on model turbine
Eff_2 = efficiency of prototype turbine
Eff_1 = efficiency of model turbine

The correction for head difference is comparatively unimportant and has been omitted from specifications for turbines for plants now under construction. The correction for diameter is of governing importance and, in the case of the 16-in. model runners for the 264-in. Chickamauga turbines, converts a model efficiency of 87 per cent to a prototype efficiency of 93.6 per cent. The accepted method of employing the formula is to compute the correction for the point of maximum efficiency of the model and to add this single percentage to all the model-test results.

Table 3 is abstracted from the cavitation-test results of the Chickamauga model tests conducted at the laboratory of the Baldwin Locomotive Works. It will be noted that cavitation is a limiting factor at the higher heads for which the tailwater is relatively low, but that, at the lower heads and higher tailwater elevations, the capacity is limited solely by the ability of the runner to deliver power.

Power-Station Arrangement

Fig. 4 shows the general cross section of the Guntersville power station, which is typical for all three Kaplan plants. Because of low entrance velocities and care taken to inhibit the formation of eddies, the corresponding head losses are relatively small; and the intake structure is comparatively short and simple, designed essentially to give proper direction to the flow filaments entering the scroll case. On the other hand, the velocity of exit from the runner is relatively high; a considerable proportion of the total available energy still remains in the discharge leaving the wheel; and for proper efficiency a long draft tube is necessary to regain this energy. Under the hydraulic requirements mentioned, structural economy is readily obtained by designing the intake and draft-tube substructure as a monolith and utilizing their combined mass to sustain the hydrostatic load.

Particular attention has been given to the economic elevation of turbine runner. Minimum turbine and generator costs are obtained by setting the turbine runner so far below minimum tailwater that the wheel diameter is no longer limited by cavitation requirements, but is established solely by the maximum power that the turbine runner is capable of delivering. However, except for the case in which solid rock is covered by an exceptional depth of overburden, minimum structural costs result from keeping the runner as high as possible with respect to tailwater, so as to reduce the volume of rock excavation, the hydrostatic load on the structure, and the yardage of concrete required for stability. Since the size of waterways is a function solely of the discharge necessary to produce the rated power at the rated head, variations in wheel diameter to meet the cavitation requirements, corresponding to various assumed runner elevations, need cause no change in unit spacing, but merely affect the diameter of the upper portion of the draft tube in the vicinity of the runner. In case the manufacturer does not have model-test results available for the desired ratio of diameter of throat ring to size of draft-tube water passages, advance model tests at the purchaser's expense may be warranted. Another consideration is that the larger wheels set at higher elevation have in-

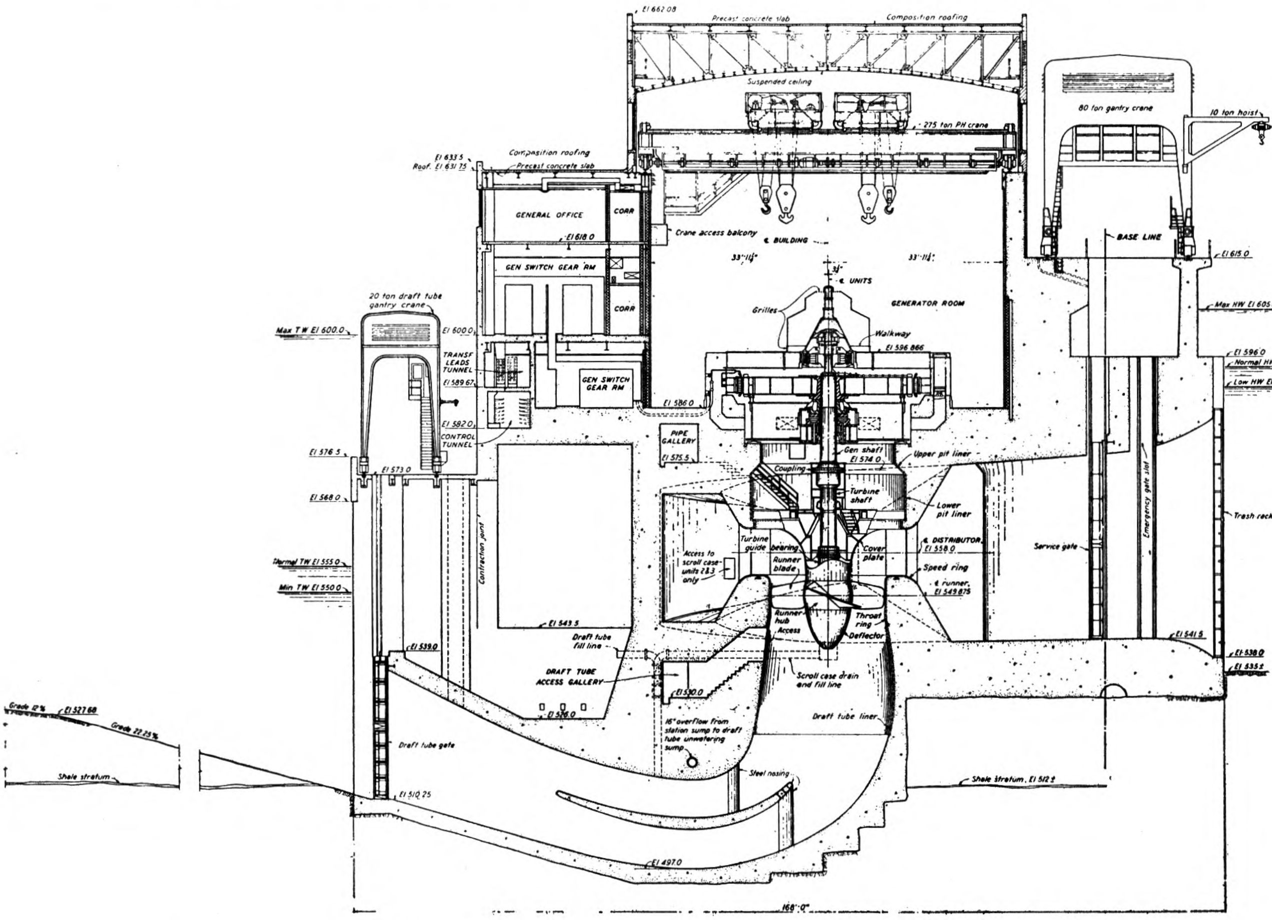

FIG. 4 CROSS SECTION OF GUNTERSVILLE POWER STATION
(Turbine capacity: 34,000 hp, 36 ft head, 69.2 rpm. Generator capacity: 27,000 kva, 13,8000 volts, 3 phase, 60 cycles.)

creased capacity at reduced head during flood conditions. Obviously, no sweeping generalization can be made, and the economic elevation of runner setting must be studied in close cooperation with the manufacturers for each particular plant so as to yield the highest ratio of net annual return to annual fixed charges.

The adopted disposition of mechanical and electrical auxiliaries is believed to utilize the limited space immediately adjacent to the generating units to best advantage so as to reduce the size of service bay to a minimum. The electrical bay is located downstream from the generator room where a foundation is provided by the long piers of the draft tube. For maximum convenience in operation, and to give the shortest and simplest arrangement of high-pressure oil piping to the Kaplan head and the wicket-gate servomotors, the governor-actuator cabinets of the duplex type are located on the main floor of the generator room between the companion pair of generating units. With the governor actuators on the main floor, space remains in the main draft-tube piers, at the elevation of the inspection tunnel where there would otherwise be excess concrete, for a compact arrangement of draft-tube unwatering pumps and operating valves. This location of unwatering pumps in the main piers between units not only eliminates space which would otherwise be required in the service bay, but also materially reduces the length of suction lines and friction-head loss in unwatering the draft tubes.

Because of the characteristic location of Kaplan runners from 10 to 12 ft below average low tailwater, and the necessity of providing ready facilities for inspection and maintenance, the draft-tube unwatering system, including draft-tube stop logs and a gantry crane, represents in itself a sizable investment. The unwatering pumps for each plant have an aggregate rated capacity of 10,000 gpm at rated head and an actual capacity of about 16,000 gpm under average conditions. The pumps are the deep-well type with low-level runners, so as not to require priming, and are capable of unwatering the draft tube completely in from 1 to 2 hr.

TURBINE DESIGN AND CONSTRUCTION

In accordance with general practice, turbine contracts of the Tennessee Valley Authority stipulate that the manufacturer shall prepare and be responsible for the design, including the determination of scroll case and draft-tube waterways, in conformity with the governing physical conditions and general requirements enumerated in the purchase specification. The authors will describe briefly, from the standpoint of the purchaser's engineers, certain of the design elements covered in the specification, with the request that the manufacturers' engineers amplify the treatment of design and construction features in the subsequent discussion.

Welded Construction. In view of the increasingly successful use of welded-plate construction in the heavy-machinery industry, the Authority's specifications afforded bidders the option of employing either cast-steel or welded-plate construction for the speed ring, head cover, lower guide-vane ring, discharge ring, gate-shifting ring, and wicket gates. Plate construction for the draft-tube liner and the upper pit liner is, of course, standard practice. The specifications required that the design of welded joints and connections and the fabrication of welded-steel parts conform to the Boiler Construction Code of The American Society of Mechanical Engineers, Section VIII, for Unfired Pressure Vessels. It was also stipulated that the welding conform to paragraph U-69 of the same code, and be stress-relieved, excepting the draft-tube liner, upper pit liner, and minor details, welding for which was specified to conform to paragraph U-70. The contractor for the Guntersville turbines, the S. Morgan Smith Company, employed welded construction very extensively and with very satisfactory results. Fig. 5 shows the welded speed ring of the Guntersville turbine.

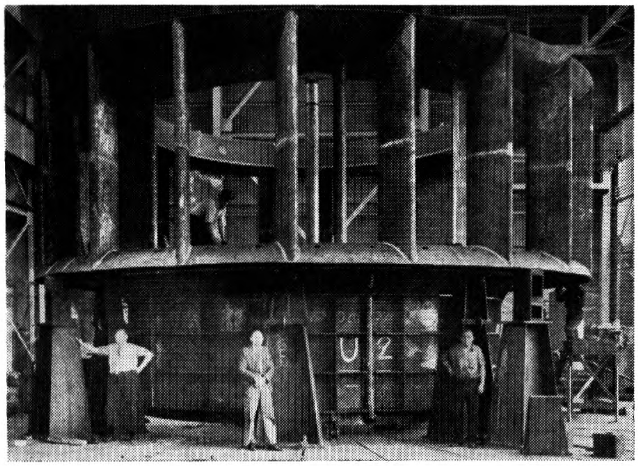

FIG. 5 WELDED SPEED RING; GUNTERSVILLE PROJECT

Turbine Shaft. Fig. 6, showing the cross section of the Chickamauga turbines, is fairly typical of all three plants. Subsequent to the design of the Pickwick Landing machines, improved facilities at the steel-forging plants made it practicable to specify that the upper end of the turbine shaft be enlarged in forging so as to form the operating cylinder for adjustment of the runner blades, and flanged at the upper end for coupling with the generator shaft. The lower flange of the generator shaft forms the upper cover of the blade servomotor cylinder. This particular feature permitted the elimination of two large steel castings for the servomotor, which were formerly interposed between the turbine and generator shafts, and two corresponding intermediate flanged bolted joints, leaving only a single flanged bolted coupling to be gasketed tight against possible oil leakage. The simplified detail has proved entirely satisfactory and much superior during shop assembly and field erection and alignment. The bottom connection of the oil-pressure-supply lines, extending inside the generator shaft from the Kaplan head above the generator down to the blade-shifting servomotor, is made by screwed, rather than by flanged fittings, which were formerly used for the purpose and required that the coupling between the turbine and generator shafts be opened 10 or 12 in. to permit making the connection. The screwed fitting permits insertion or removal of the oil-supply pipes from above without dismantling the coupling.

Turbine Guide Bearings. The main turbine guide bearings are the water-lubricated adjustable type with shoes of lignum vitae or a molded plastic material, such as bakelite or Insurok, while the corresponding shaft sleeves are a special corrosion-resistant steel. Water-lubricated bearings were selected in preference to oil-lubricated bearings for three principal reasons: (1) Optimum location of the bearing for its primary function of support, immediately adjacent to the turbine runner with no stuffing box and oil chamber interposed between the runner and bearing; (2) most accessible and convenient location of stuffing box above the guide bearing, where adjustments can be made while the unit is in operation; and (3) elimination of the hazard of burning out the bearing due either to the leakage of water past the stuffing box or flooding of the turbine pit during the rather frequent condition of high tailwater.

In general, the problem of providing adequate lateral support is much more acute for propeller runners than for Francis wheels, owing principally to the relatively greater diameter and weight due to lower heads, and to the inherently greater hydraulic instability under low-gate conditions. It is characteristic of Francis wheels that the runner is located in elevation at about the center line of the distributor; and, consequently, the oil-lubricated type of bearings for such machines may be located above the head cover in a relatively free-draining position without involving an excessive distance between the bearing and the runner to be supported. On the other hand, because of the characteristic design of propeller-type turbines, in which the runner is placed 6 or 8 ft below the center line of the distributor, the oil-lubricated type of bearing, if adopted, must be placed in a relatively small, conical chamber down below the head cover in order to keep the distance between the bearing and the runner to be supported within acceptable limits. In the event of a defective stuffing box, even a low rate of leakage might soon fill the surrounding space and cause the bearing to burn out.

Runner-Hub Lubrication. Correct lubrication of the mechanism within the Kaplan runner hub, which is continuously submerged under a head of from 10 to 60 ft, is an important element of design. The runner-blade trunnions, operating in bronze bushings under bearing pressures of between 2000 and 3000 psi, must be provided with an unfailing film of the proper lubricating oil, uncontaminated by grit and free from any appreciable amount of water. In addition, the hub oil has a second essential function as a protective coating and inhibitor of corrosion for the many steel surfaces of the internal mechanism.

By means of a revolving oil chamber, located on the main shaft just below the generator coupling, a continuous oil pressure of from 8 to 10 psi is maintained, sufficient to insure a steady supply to the trunnion journals. The revolving reservoir conserves an appreciable volume of oil displaced during each downward movement of the main operating shaft into the runner hub.

The runner-blade trunnions are sealed by means of a chevron type of packing, retained in the packing space under pressure between two stainless-steel rings. The arrangement is adequate to prevent either the infiltration of water and grit along the blade trunnions during high tailwater conditions or the loss of oil from inside the hub during operating conditions of high vacuum adjacent to the blades. Leakage of oil from the hub is restricted on the average to about $^1/_4$ gal per day, and the design is so rugged that a tight seal is assured for several years of operation. The revolving reservoir is equipped with a sight-gage glass, by means of which, when the unit is stopped, the quantity of oil in the runner hub can be immediately determined and make-up oil added, if necessary.

Simple and effective means are provided for draining any accumulation of leakage water which would, if neglected, endanger the safety of the trunnion bearings and cause corrosion of the operating mechanism. Because it is heavier than the lubricating oil, any leakage of water along the trunnions will collect in the bottom of the hub when the unit is stopped. In the case of the Guntersville design, which is representative in principle for all three manufacturers, drainage of water from the bottom of the hub, under the static head of oil in the annular reservoir, is accomplished by drilling out the center of the blade-operating shaft so as to form a pipe communicating with the bottom of the hub and terminating at the upper end in a test connection, readily accessible at the exterior surface of the turbine main shaft above the stuffing box

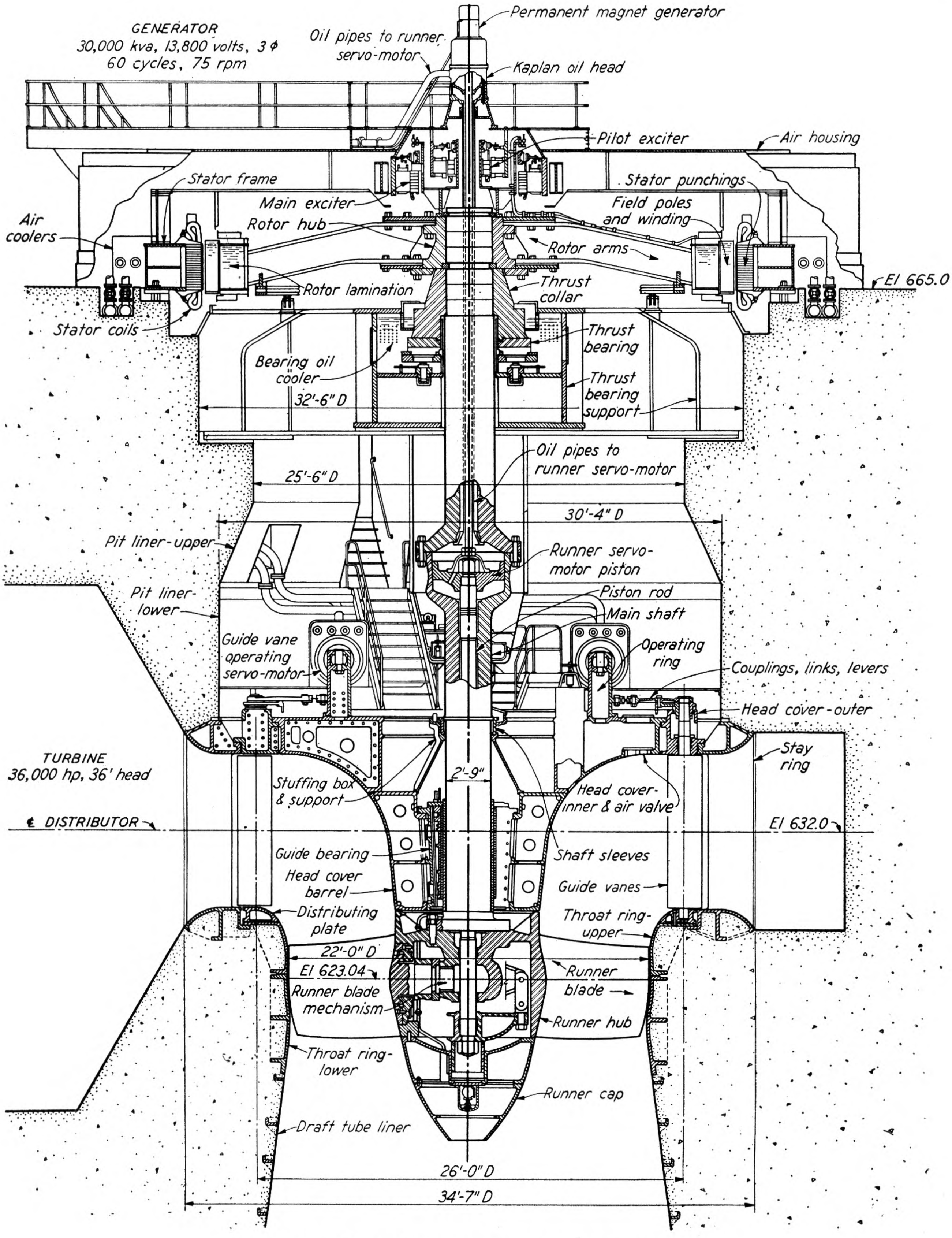

Fig. 6 Cross Section of Chickamauga Turbine

Alignment of Turbine and Generator Shafts. In conformity with the purchaser's specifications, turbine and generator shafts, complete with thrust-bearing collars and generator rotor hubs, are fitted, coupled, and given a rotating-alignment check at the manufacturer's plant prior to shipment to insure that the combined shafts are straight and that the face of the thrust collar lies in a plane perpendicular to the combined-shaft axis.

Before operation of the unit, a field alignment check of the entire rotating assembly of the turbine and generator is made to demonstrate that the axis of rotation of the combined shafts does not depart from the vertical more than 0.003 in. at any point, also that the maximum diameter of circle described by any point on the shaft in rotating about that axis is not more than 0.01 in. This so-called rotation check is made with all turbine and generator guide bearings backed out, and with the entire rotating assembly simply suspended from the thrust bearing. The rotating assembly is turned, by means of a block-and-tackle arrangement attached to the rotor spokes, through four intervals of 90 deg each and back to the original position. At each position, micrometer tram measurements are made between the turbine and generator shafts and four fixed suspended plumb wires. Any corrections necessary to plumb the shafts with respect to the vertical can readily be made either by the adjusting studs under the Kingsbury bearing shoes or by the insertion of adjusting plates under the thrust-bearing supporting beams. If the throw of the shafts exceeds 0.01 in. on the diameter at any point, indicating that the thrust collar is excessively out of perpendicular to the shaft axis, the requisite field adjustments are more difficult; and, consequently, every check should be made at the factory to insure that this feature is acceptable before shipment.

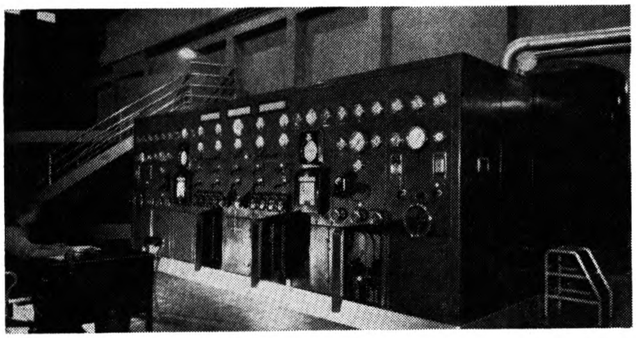

FIG. 7 GOVERNOR-ACTUATOR CABINET; PICKWICK LANDING PROJECT

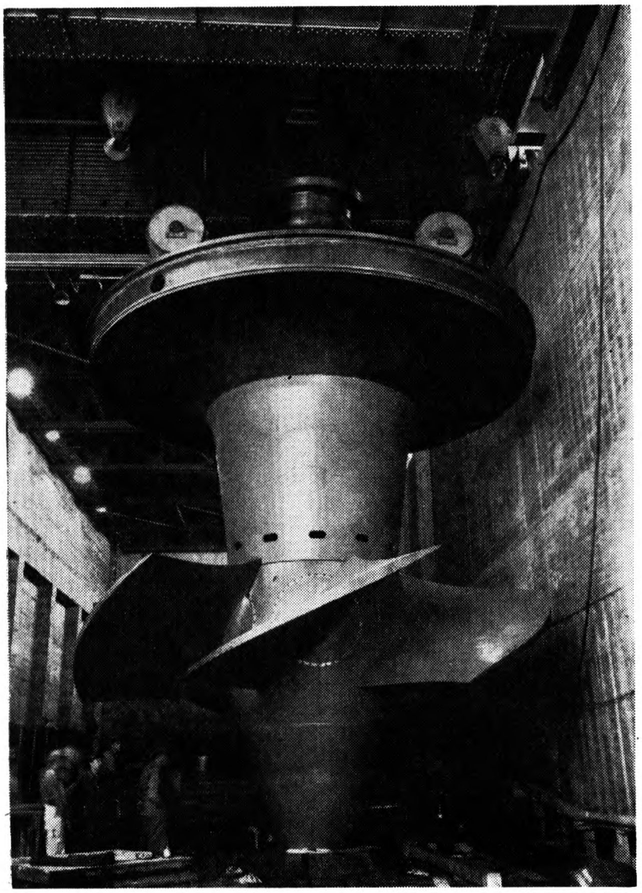

FIG. 8 HEAD COVER AND RUNNER ASSEMBLY; PICKWICK LANDING

Governors. The governors are of the cabinet actuator type with the entire mechanism, including the governor-flyball elements, relay valves, motor-driven oil pumps, Kaplan valves, and Kaplan controls, enclosed in a compact cabinet, the base of which forms the sump tank. The governor controls and the requisite gages and temperature and pressure indicators are mounted upon the front face of the cabinet.

In the Guntersville and Chickamauga plants, where three units have been installed initially, duplex actuators are provided in a single combined cabinet located between the units. Owing to the necessity of providing clearance at the temporary end wall of the power station, individual cabinet actuators are provided for the third units and planned for the fourth units. At the Pickwick Landing plant two units were installed initially and provided with a duplex actuator.

The governors are equipped with motor-driven flyballs, gear-type motor-driven oil pumps, cable-type restoring mechanisms, and welded-steel oil piping with flanged connections only where necessary for ease in assembling and dismantling. The operating oil pressure is between 275 and 300 psi. The oil piping and restoring cables to the wicket-gate servomotor in the turbine pit are located in floor trenches entering the cabinet from below. The piping and restoring cables to the Kaplan head above the generator are located beneath the generator walkways and enter the actuator cabinet from above.

CONCLUSION

Experience at the Pickwick Landing, Guntersville, and Chickamauga plants indicates that Kaplan turbines in the range of physical size from 22 to 24 ft may be satisfactorily designed, constructed, and erected to meet the exacting demands of variable head and load service. Engineers are frequently confronted with the question of whether still larger units might be economical. There are many obstacles to increased size, such as the difficulty of obtaining large steel castings for runner hubs and blades without a prohibitive percentage of rejections, and the difficulty of designing large complicated concrete substructures to withstand the attendant indeterminate shrinkage stresses without consuming a prohibitive interval of time for dissipation of setting heat during construction operations. It also appears that, although the practical limit of refinement in machine-shop tolerances for such large heavy machinery has already been reached, still further refinement would be necessary in the case of larger machines to obtain, during field erection, the degree of accuracy in turbine and generator-shaft alignment required for the satisfactory operation of such large heavy rotating masses. In the opinion of the authors, the Pickwick Landing, Guntersville, and Chickamauga machines approach the maximum practicable size of Kaplan turbines which may be fabricated and erected with present facilities.

Discussion

J. M. Mousson.[3] The authors state that efficiency tests on the prototype units are not contemplated to avoid the difficulty of measuring large unit discharges and that the step-up, based on laboratory tests, is being used as a criterion for the acceptability of the units. It would appear, however, that the reliance on the Moody step-up formula exclusively is not a guarantee for proper prototype performance because: (1) It was developed based on experience with Francis runners. (2) It rests exclusively on one manufacturer's laboratory conditions and may not be applicable unconditionally to those of others. (3) Even under favorable conditions field experience thus far gained with propeller-type units of the fixed-blade or adjustable type indicates that no further increase in step-up may be expected above 150 in. runner diam. Some engineers even contend that with these types of turbines a gradual decrease in step-up could be expected above the maximum occurring at about 150 in. diam.

It is believed that the complete negation of prototype testing is a serious handicap to progress in the art, particularly serious when this viewpoint is held by engineers associated with such a vast enterprise as the T.V.A. No one will contend that the period of development of the propeller-type turbine is over and, therefore, continuous and substantial efforts are yet to be made and are imperative to achieve efficiencies closer to the ideal.

In addition, it must be emphasized that the over-all costs per horsepower installed are an important factor. Certain expensive features adopted by one or the other turbine manufacturer remain yet to be justified from an economic point of view. However, no justification can be obtained without prototype testing. This was ably pointed out some time ago by one of the authors' associates[4] to the effect that the economics of draft-tube splitters should be carefully investigated and that there was a real opportunity as well as a necessity to do so with the completion of Wheeler and Pickwick Landing Dams. This opportunity is now even more striking with Guntersville and Chickamauga in operation, both plants having Kaplan turbines of nearly identical dimensions but only the draft tubes of the former development are provided with draft-tube splitters.

A recent feature of Kaplan turbines, referred to in the paper, is the adoption of cables for the restoring mechanisms instead of the rigid rods previously used. It would be interesting to know whether or not the prestressed cables stretch appreciably in service and, if so, what the permanent elongating characteristics are with relation to time.

Based on past experience, the correct cam design controlling the gate-blade relation cannot be obtained through model testing and some form of index testing is required on the prototype units. It would be of value to know what types of index method are being employed on the various installations of the T.V.A.

F. Nagler.[5] The authors present comprehensively significant engineering data which cannot help but answer questions occurring to many engineers interested in hydroelectric power.

It would be of interest if the authors would comment a little further on Table 3. It is not evident whether the tests for cavitation were made at the heads listed in the third column, or whether the cavitation coefficients were arrived at by testing the models at some other heads and simply applying the results so obtained. It is noted that the authors state the head difference, in connection with the application of the Moody formula, is comparatively unimportant, from which it might be assumed that the head differences for the cavitation tests may closely approximate those to be experienced in the powerhouse.

The writer would like to ask whether some of the hydraulic-thrust figures shown in Table 2 have been checked by field measurements. The writer has quite a collection of such data, obtained by calibrating the deflection of the bridge by means of the fairly well-known weight of rotor, shaft, and runner, and then using this calibrated bridge to determine the additional hydraulic thrust. If such figures are available for any one of the three plants shown in Table 2, they would be of considerable interest.

The care taken by the engineers of the T.V.A. in prechecking the turbine and generator shafts for alignment is particularly noteworthy. This seems to be an increasingly desirable practice. Did the authors find that shop tests showed comparable straightness and truth with the rotational check made in the field?

It would also be of interest to have further comment on, or comparison of, the cable-type restoring mechanisms with the older torsional-shaft types. They seem to possess such advantages in cheapness and flexibility of installation that any operating disadvantages observed should be brought out.

R. E. B. Sharp.[6] The optional use of either cast or welded-plate steel as the material for the speed or stay ring, as well as for the other parts mentioned by the authors, is a sensible provision in turbine specifications. The turbine manufacturer is thus permitted to take advantage of his preferred design and shop practice and in some cases to improve deliveries by the use of that material most readily available. The use of plate steel is preferable for those parts of the water passages subject to cavitation, such as the throat or discharge ring, due to its greater resistance to this action. Runner blades of steel plate, welded to a skeleton frame, would undoubtedly resist the cavitating action better than cast-steel blades, but the matter of strength and cost would be formidable problems.

As brought out by the authors, the use of a water-lubricated bearing results in a minimum amount of overhang of the runner below the bearing. This is a desirable feature as affecting the critical speed of the shaft. With both runner and generator rotor overhung beyond the only two guide bearings provided, it is essential that these calculated deflections be a minimum.[7]

The actual amount of hydraulic thrust on the turbine runner can be readily determined in the field, by application of the principle that the deflection of the beams supporting the thrust bearing is proportional to the load. The deflection due to the known total weight of the revolving parts is measured with the turbine shut down, and again when in operation. The relation between the hydraulic thrust so measured, the runner-blade pitch at various radii, and the power developed can be used as a check on the effective-flow distribution through the runner.

The intermittent turning of the runner blades under load through only a small percentage of a total revolution, with frequent periods without movement, prevents the maintenance of an effective oil film at the loaded portions of the blade bearings, with the result that a high coefficient of friction must be overcome by the blade servomotor, even though the runner hub is filled with oil. This coefficient is undoubtedly lower than it would be were the load on the blades not of a very live nature with some vibration to permit to some extent the seepage of an oil film into the desired locations.

[3] Hydraulic Engineer, Safe Harbor Water Power Corporation, Baltimore, Md. Mem. A.S.M.E.

[4] "Economic Aspects of Energy Generation," a Symposium, Trans. A.S.C.E., vol. 104, 1939, pp. 942–1008; discussion by R. M. Riegel, pp. 1014–1015.

[5] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

[6] Chief Engineer, I. P. Morris Dept., Baldwin Southwark Divsion, The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

[7] "Lateral Vibration of Shafts," by L. F. Moody, *Product Engineering*, vol. 6, Feb., 1935, pp. 57–60; March, pp. 98–100; April, pp. 142–143.

The ideal method of lubrication with bushed bearings, such as are used, would be to have a continuously operating high-pressure oil pump which would provide an oil film at the desired points. The mechanical complications of such an arrangement with the possibility of derangement and consequent loss of such an oil film have so far militated against the use of such a feature.

The use of antifriction bearings in the hub, of the roller or ball type, would greatly reduce the friction load. The lack of adequate bearings of this type of stainless steel, such as would be necessary to take care of possible water entrance into the hub, has been a deterrent in the adoption of this type of bearing. The nature of the live load on the blades, in conjunction with only a partial revolution with long periods at one position, and a consequent Brinelling action of the balls or rollers on the races, is an undesirable feature. While Terry has adopted with success this type of bearing in his design of automatic adjustable-blade runners, the writer is of the opinion that additional successful service experience is necessary to justify this type of bearing, particularly in view of the major operation required in the renewal of such bearings, involving the removal of the runner from the unit. On the other hand, in spite of the high coefficient of friction with bronze-bushed bearings of the type which have become standard in Kaplan runners, no bearing renewals have yet become necessary to the writer's knowledge although, in some instances, difficulty has been encountered due to the bushing turning with the blade shank in the runner hub. To prevent this action from occurring, it is good practice, by the use of dry ice, to shrink the bushing into the hub and retaining rings with resulting actual knitting of the bushing into the surrounding cast steel, which much more effectively prevents relative motion than any press fit. The writer believes that the use of molded plastic material for the bushings in the runner hubs, with some reduced friction coefficient, has possibilities, as this material in certain instances has been found superior to bronze in its ability to withstand high bearing loads.

Authors' Closure

In connection with Mr. Mousson's discussion, it should be emphasized that the remarks of the authors with reference to model testing were purposely limited to commercial acceptance tests for Kaplan turbines. The results of research activities, including liberal use of prototype testing to explore debatable features of economic design, will be made available at some future date.

The authors are under no delusion that the Moody formula is an instrument of extreme precision, but simply take the position that, in the present state of the art of current-meter gaging and the attendant possibility for interminable controversy, advance model testing furnishes a practical workable device having general commercial acceptability as a basis for contract. In this connection it is pertinent that manufacturers are not yet prepared to guarantee Kaplan-turbine efficiencies in excess of the order of 89 per cent. Whether or not the step-up relation holds within narrow limits is not material, because the tests exceed the guarantees by a comparatively wide margin; for instance, model tests on a 16-inch runner show an efficiency of 87 per cent, which steps up to $93^1/_2$ per cent for a 22-ft prototype, exceeding the guaranteed efficiency of $89^1/_2$ per cent by a 4 per cent margin. Conceding Mr. Mousson's statement that the Moody relation is not dependable for prototype diameters exceeding 150 in. (although the authors know of no concrete evidence to support such a contention), it is reassuring to note that a 38-in. prototype of the 16-in. model would have a Moody efficiency of $89^1/_2$ per cent, and a 150-in. prototype would show $92^1/_2$ per cent, which exceeds the guarantee by 3 per cent. The point is that, under the present range of contract efficiencies which are available to the purchaser of Kaplan wheels, absolute refinement in stepping up efficiency acceptance tests is of merely academic interest.

TABLE 4 HYDRAULIC THRUST DETERMINATIONS

	Wheeler	Guntersville	Chickamauga
Dead load: rotor, shaft, and runner, lb...	803,000	973,000	936,000
Deflection, generator bridge, dead load, in..	0.018	0.031	0.028
Maximum deflection when operating, in...	0.042	0.053	0.053
Hydraulic thrust, computed from deflection, lb............................	1,070,000	687,000	836,000
Hydraulic thrust, estimated by manufacturer, lb............................	1,190,000	914,000	1,052,000
Gross head during tests, ft...............	46.7	42.6	49
Maximum head assumed for thrust estimate, ft............................	52	42	52
Gate opening for maximum thrust, per cent............................	100	60	40
Blade tilt for maximum thrust, per cent....	Fixed	55	5

Mr. Mousson asks what amount of stretch has been found in the cable-type restoring mechanisms for Kaplan turbines. Over the first year of operation the permanent elongation or stretch appears to be in the neighborhood of $^1/_{16}$ in., which is insignificant in a 40-ft length with a movement of 24 to 36 in. Readjustment is very simple and requires about two minutes. There is some elasticity to these cables, and tests indicate that they may fail to show gate movements of 0.2 of 1 per cent and less.

To determine the proper shape of the cam controlling the gate-blade relation, Winter-Kennedy-type taps are used, readings being taken with two or even three sets of taps giving different coefficients, at five to seven different blade tilts and at least six or seven different gate openings, resulting in 30 to 36 test points at each head. From these data, by plotting curves of $KW/D^{1/2}$, the optimum blade angle for each gate opening can be readily established, independent sets of Winter-Kennedy points being used to confirm this determination. While different sets of taps sometimes show slightly different characteristics, the corresponding determinations of the proper blade angle usually coincide within very narrow limits.

Referring to Mr. Nagler's comments, the cavitation tests shown in Table 3 of the paper were made at the new cavitation laboratory of the Baldwin Southwark Corporation, with heads varying from 15 to 27 ft, averaging about 20 ft. The cavitation coefficients so obtained were applied without correction to the prototype conditions. The model efficiency tests were conducted under even lower heads, between 2 and 4 ft, and the corrections for head in the Moody formula were neglected in stepping up these efficiencies. The horsepower of the model was computed directly, no correction being made for increased efficiency.

The hydraulic-thrust figures contained in Table 4 have been obtained by measuring generator bridge deflections with dead load and with hydraulic thrust.

Shop checks on the individual and the combined shafts have been reliable, especially for checking coupling alignment. However, with generator construction such as used at Guntersville and Chickamauga, where the generator thrust collar is a separate piece, shrunk and keyed to the generator shaft, several cases have occurred which indicate that these collars shift under load, sometimes throwing the rotating parts out of true position. The writers now specify integrally forged thrust collars as the best assurance against such type of misalignment.

Mr. Sharp's suggestion of the possibility of using some form of molded plastic material for the runner hub bushings is interesting, especially if it would permit water lubrication. Maintaining oiltight seals around the runner-blade trunnions, with provision for refilling with oil, and draining out any infiltrating water add considerably to the cost and complicates the design. With molded plastic bearings and either bronze or stainless sleeves provided on the trunnions, the hub could remain full of water, thus reducing the cost and possibly eliminating the use of oil, particularly in the smaller sizes of units.

The authors wish to thank those who have discussed the paper for their valued comment.

A Study of the Development of Skill During Performance of a Factory Operation

By RALPH M. BARNES,[1] IOWA CITY, IOWA, AND J. S. PERKINS,[2] CHICAGO, ILL.

While in general the many studies into the nature of skill have been concerned with the total time required to accomplish a given task and the influence on time values of varying conditions pertaining to a specific operation, this paper is primarily devoted to a time study of the elements entering into the performance of an industrial task. The investigation constitutes a pioneer effort to study the effect of practice on a typical factory operation, conducted under laboratory conditions.

The work was undertaken jointly by the Western Electric Company and the University of Iowa. The equipment was made, the tests were run, and the data were compiled in the industrial engineering laboratory at the University of Iowa. The statistical work, tabulation of results, and other activities incidental to the preparation of the final report were handled in the offices of the Western Electric Company.

Introduction

Of CURRENT and outstanding interest in the industrial world of today are those problems related to the teaching, the acquisition, and the measurement of skill. The importance of the subject is reflected in the many studies which educational institutions, industrial organizations, and psychological laboratories have made relative to the nature of skill.

All of these investigations are concerned with the time required to accomplish a specific task and with the manner in which varying conditions can affect such time values. However, practically all investigations to date have been concerned with the total or cycle time required for performance of a specific task, and have given only limited attention to the time required for performing the various therbligs[3] of which any task consists. The present investigation has taken this further step, and has been directed primarily at study of the change in therblig time values resulting from practice in performing an industrial task.

The various aspects of the study on which information was sought are as follows:

1 To study the effect of practice on a typical factory operation carried on under laboratory conditions.

2 To study the learning curves of the various elements of the operation as they were performed by each of the different subjects.

3 To study the consistency between subjects in learning the same element.

4 To study the effects of "speeding" and "soldiering."

5 To study dispersion and its relation to the average performance time.

6 To study the effects of fumbling on the normal learning curve.

7 To study the several ways in which a transport load and pre-position element was performed.

8 To examine the effectiveness of several rating techniques on data of known quality.

9 Finally, to study the effect of practice on the relation of eye movements to hand motion.[4]

Transition From Purpose to Method

In order to study the effects of learning on the elements of an operation or industrial motion cycle, it was regarded as most feasible to set up under laboratory conditions the operation of feeding parts to a punch press. This motion pattern was selected primarily because:

1 It is an extremely common motion pattern in industry, being very similar to punch-press work, as well as to other operations such as feeding parts to tapping machines.

2 It is short and thereby offers the opportunity for a high degree of learning in comparatively little time.

3 It is a complex operation requiring coordinated action of both hands, the eyes, and one foot (for pressing a pedal). This complexity offers opportunity to study the acquisition of skill over a variety of elements.

The part being fed into the punch press was a relay spring, Fig. 1.

The operation was performed by grasping a relay spring from the supply tray by the left hand. The left hand then turns the spring so that it may be grasped properly by a pair of tweezers. The tweezers are held in the right hand and, accordingly, then the part is passed from the left hand to the tweezers in the right hand. The right hand locates the part in the die, releases hold of the part, and returns for another part. As the right hand reaches for the next part, a pedal is pressed which ejects the part at the back of the die.

Fig. 2 shows one cycle of the operation. Frames 3 to 6 show the left hand reaching for a selected relay spring. Then the attention shifts to the die to locate the part held by the right hand, frames 6 to 12. During this time (frames 6 to 12), the left hand is pre-positioning the next part for the tweezers. Now, reverting to the part held in the tweezers; when this part is located in the die the tweezers in the right hand reach for the part held by the left hand. As the right hand is reaching, the pedal is pressed, ejecting the part from the die. The attention at this time is directed in transferring the part from the left hand to the tweezers in the right hand, frames 13 through to frame 2.

Fig. 3 is a schematic diagram showing the paths of the hands, and the points of fixations of the eyes as laid out over the workplace.

Laboratory Equipment

Equipment Used in Performing Motion Cycle. The equipment

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[2] Industrial Engineer, Western Electric Company.

[3] Work has been arbitrarily divided into eighteen common elements called therbligs. For further information, refer to "Motion and Time Study," by Ralph M. Barnes, second edition, John Wiley and Sons, Inc., New York, N. Y., 1940, chap. 6.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[4] Because of the limitations of space not all of these nine aspects of the study are presented in this paper. For full details, refer to University of Iowa Studies in Engineering, Bulletin 22, University of Iowa, Department of Publications, Iowa City, Iowa, 1940.

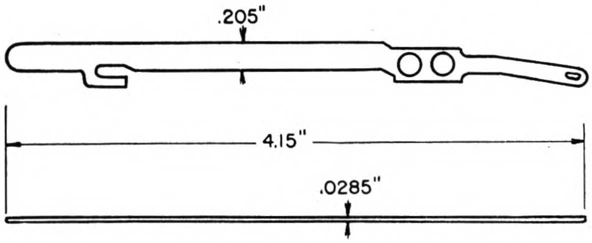

Fig. 1 Relay Spring

was so made and arranged as to impose on the subjects the same demands and limitations surrounding the motion pattern of punch-press work in an industrial shop. The part Fig. 1, used in the laboratory study, was in the same condition as when used in the shop. It was irregular in shape but all in one plane.

The laboratory equipment does not "process" the part. The part is merely located in the die, as though to be formed, and then ejected without being formed. A total of 250 parts were thus used over and over again during the run of the experiment.

A pair of aluminum punch-press tweezers, $^3/_{32}$ in. thick and $7^1/_2$ in. long, exactly the same as used in the shop for this operation, were used in grasping the part from the left hand and then placing the part in the die. In order to record the opening and closing of the tweezers, a 6-v circuit was completed at the instant the tweezers closed. This circuit, working through an electric relay, operated one of the pencils on the kymograph. By making a series connection to both points of the tweezers, the kymograph pencil Fig. 4 would jog laterally on the tape when the tweezers closed; when the tweezers released the part opening the circuit, the pencil would jog back to its original position.

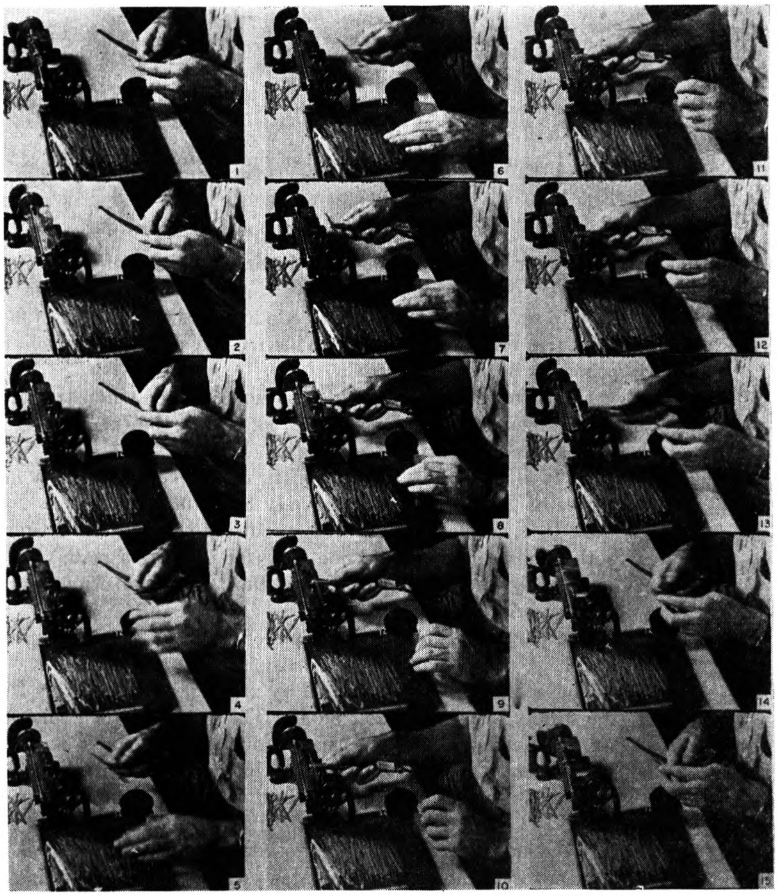

Fig. 2 One Cycle of the Operation

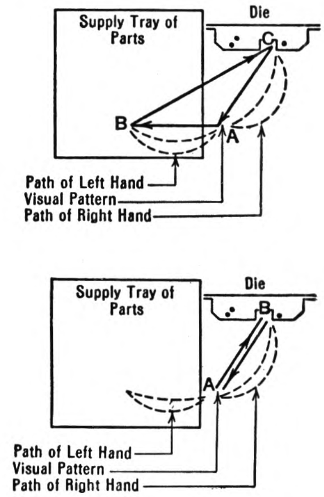

Fig. 3 Comparison of Visual Patterns

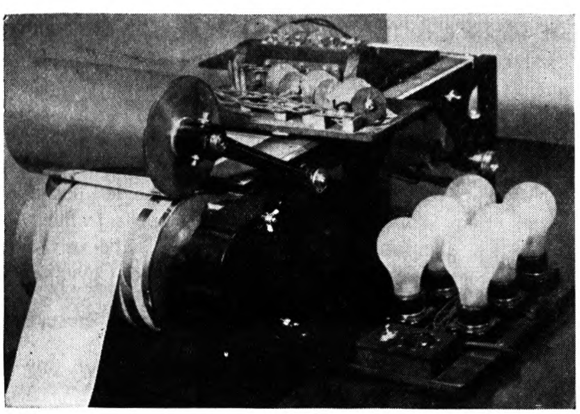

Fig. 4 Electrically Operated Kymograph Measures and Records Time

The die used in the study was made in the University of Iowa, mechanical-engineering laboratory, and was similar to common shop dies. Three, bullet-nosed pilot pins were used to locate the part in the die. The die operated by a cord attached to the foot pedal in such a way that, when the subject pressed the foot pedal, as a shop operator would to trip a press, the die rotated 180 deg, ejecting the part with a loud thump as the die hit the stop.

The pedal also operated a Veeder counter, used primarily to keep a count of the cycles of practice during the practice periods.

Equipment Used for Measuring and Recording Time. The kymograph, shown in Fig. 4, is a device for measuring and recording time with great accuracy. This machine was built in the industrial-engineering laboratory at the University of Iowa. Narrow paper tape is drawn across the kymograph table and under solenoid-operated pencils by means of two rollers driven at uniform speed by a synchronous motor.

Opening and closing electric circuits by means of photoelectric cells and switches causes the pencils to move laterally, making jogs in the lines drawn on the moving strip of paper, Fig. 5. The tape travels at a uniform velocity of 1914 in. per min.

Four pencils were used for recording time in this study. The work of the left hand was divided into three elements, the work of the right hand into three elements, and the work of the foot into two elements. Fig. 5 is a reproduction of the record made by the solenoid-operated pencils for one cycle of the operation.

Camera and Projecting Equipment. Motion pictures were taken by the Eastman Special and Bell and Howell 70-E cameras. Color pictures were taken of one subject for demonstration purposes and of another for frame-by-frame analysis.

The pictures were taken at 1000 frames per min with the camera always started on a full-wound spring. These cameras were checked, showing that an error of not more than 2 per cent occurred from full-wound spring to the automatic stopping point.

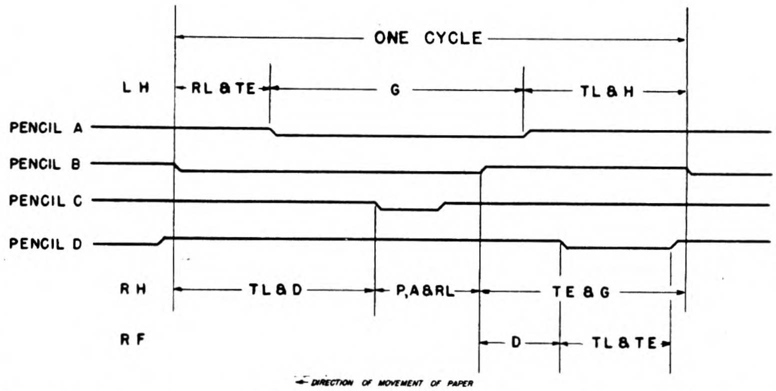

FIG. 5 REPRODUCTION OF ONE CYCLE OF RECORD MADE BY SOLENOID-OPERATED PENCILS ON KYMOGRAPH

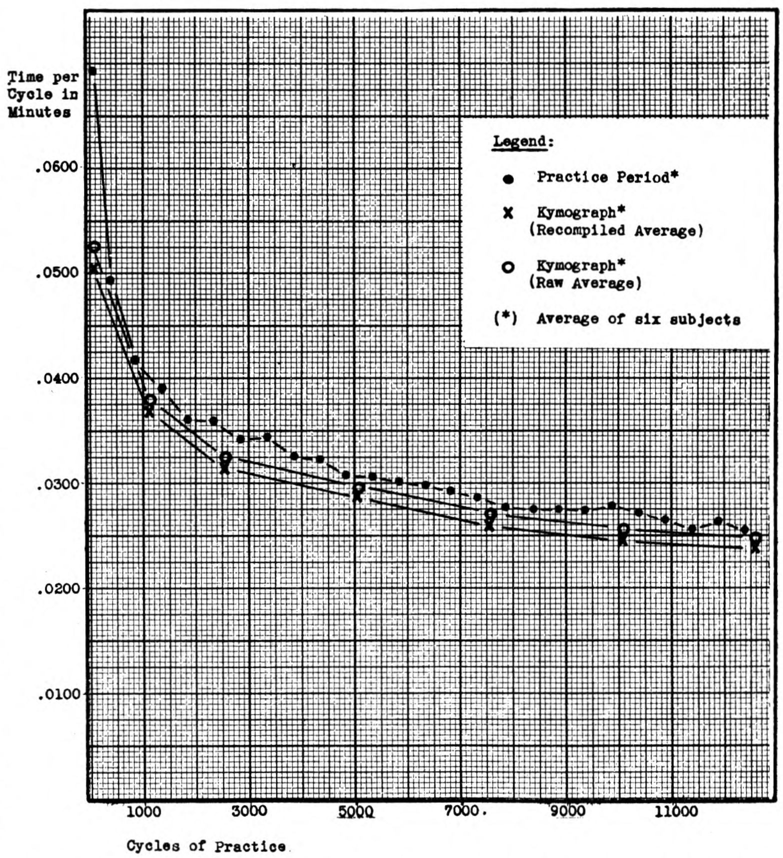

FIG. 6 LEARNING CURVES; COMPARING KYMOGRAPH DATA WITH PRACTICE-PERIOD DATA

SCHEDULE OF PRACTICE AND SAMPLING PERIODS

It is generally recognized that short and frequent periods of instruction and practice produce greater skill in shorter time than longer and infrequent periods. This consideration was regarded as important in arranging the schedule which was for 500 cycles a day, 5 days a week, Monday through Friday inclusive for the 5-week run.

The practice periods were divided into two parts of 250 cycles each, the time being taken on each of the two runs. On days scheduled for timing by the kymograph and camera, the first run as a practice run was for 250 cycles, the second run was for 200 cycles, and the third run was for 50 cycles. When obtaining a sample of 50 cycles by means of the kymograph it was necessary to have the subject actually perform about 75 cycles. The unrecorded practice amounts to not more than 2 per cent. It is, however, about the same for each subject and for the purpose of this experiment believed to be of minor importance.

Definitions. "Practice" normally refers to the period of work in which the subject is diligently applying himself to learning the motion cycle, as described.

"Sampling" refers to the use of the kymograph as a means of timing elements of cycle. It also includes taking moving pictures both for analysis in terms of the therblig times and as a means of sampling for demonstration purposes.

THE SUBJECTS

The six subjects who performed the work in these experiments were all students at the University of Iowa and undertook this work voluntarily in the interest of furthering a promising experiment. A consideration of the physical make-up of these students leads to the conclusion that there is no significant difference between them and employees encountered as applicants for industrial work of this sort.

INTERPRETATION OF RESULTS

Elimination of Abnormal Observations. In the raw data, it was noted that there were occasional observations far removed in value from the rest of the group. Such values have little effect on the average but have considerable effect on the standard deviation. To avoid the effect of these abnormal values, a refining process was applied to the data to eliminate each value more than two standard deviations removed from the arithmetic mean. The procedure was to:

1 Compute the arithmetic mean.

2 Compute the standard deviation.

3 Eliminate from data all values more than two standard deviations distant from the arithmetic mean.

4 Recompute the arithmetic mean; this figure is referred to as the "recompiled average."

5 Recompute the standard deviation of the refined data; this figure is referred to as the "recompiled standard deviation."

The average of the recompiled cycle times is also shown in Fig. 6, and demonstrates that the extreme values were of the greater magnitude. The difference between the raw data and the recompiled averages ranges between 2.8 and 3.7 per cent.

Analysis of Therblig Combinations. Definitions for therblig combinations are given in Table 1. The data collected in this experiment permit study of the manner in which the time required to perform individual therbligs or combinations of therbligs varies as skill is acquired. This study permits a consideration of the manner in which the over-all learning curve is built up from the learning curves of the elements within the cycle.

Some of the learning curves take the general form of total cycle-time curves encountered in the literature. In other cases, the final proficiency attained is but slightly different from the pace set at the very beginning of the study. Finally, in those therbligs which are not necessary to the attainment of the best cycle time, it is found that there is no real consistency by subjects and that the tendency, if any, is to retain the starting time and perhaps starting method rather than to improve the time for such unimportant therbligs.

For some therbligs there is a comparatively wide range of dispersion among the subjects and for others there is but little variation in the final time required for performance. The therbligs will now be considered individually. Cycle-time learning curves are given in Fig. 7.

TABLE 1 DEFINITION OF THERBLIG COMBINATIONS: FOR KYMOGRAPH STUDY

Member	Therbligs	Definition	Beginning of measurement	End of measurement
Left hand (LH)	RL&TE	Acts of relinquishing control and reaching for next part	From instant tweezers are closed on part	Until hand interrupts beam of light in reaching for a part on supply platform
	G	Act of obtaining control of a part	From instant hand interrupts beam of light	Until hand has complete control of part, allowing beam to activate photocell as hand is drawn from supply table
	TL,PP&H	Act of turning part ready to be grasped, carrying it into transfer area and holding it while tweezers close on it	From instant hand leaves beam of light	Until tweezers are closed on part
Right hand (RH)	TL&D	Motion carrying part toward die	From instant tweezers are closed	Until part and tweezers interrupt beam of light located over and at front edge of die
	P,A&RL	The acts of directing visually part to nest; assembling part in nest, and opening tweezers	From instant beam of light over nest is broken	Until tweezers open
	TE&G	Acts of moving tweezers for next part and gaining control of part	From instant tweezers open	Until tweezers are closed on next part
Right foot (RF)	PD	A delay in action of pedal movement	From instant tweezers open	Until die begins to discharge part[a]
	TL&TE	Acts of pressing pedal down and withdrawing foot	From instant mercury switch completes circuit	Until mercury switch opens circuit

[a] The mercury switch is activated by the pedal which ejects the part and also rotates a fast-moving cam, thereby tipping the mercury switch. This delay also occurs in the reverse process when opening the circuit, thus effecting a compensating factor.

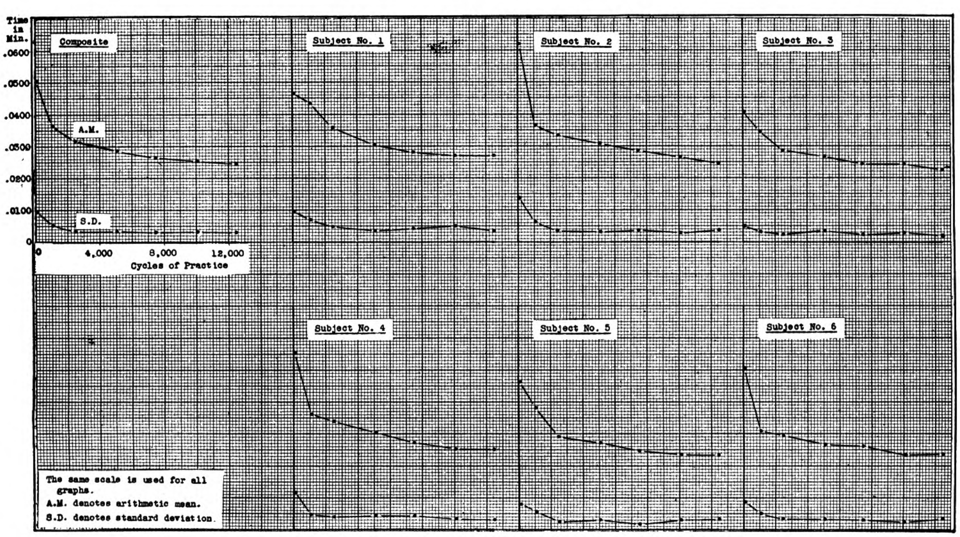

FIG. 7 CYCLE-TIME LEARNING CURVES; COMPOSITE AND INDIVIDUAL GRAPHS; ARITHMETIC MEAN AND STANDARD DEVIATION

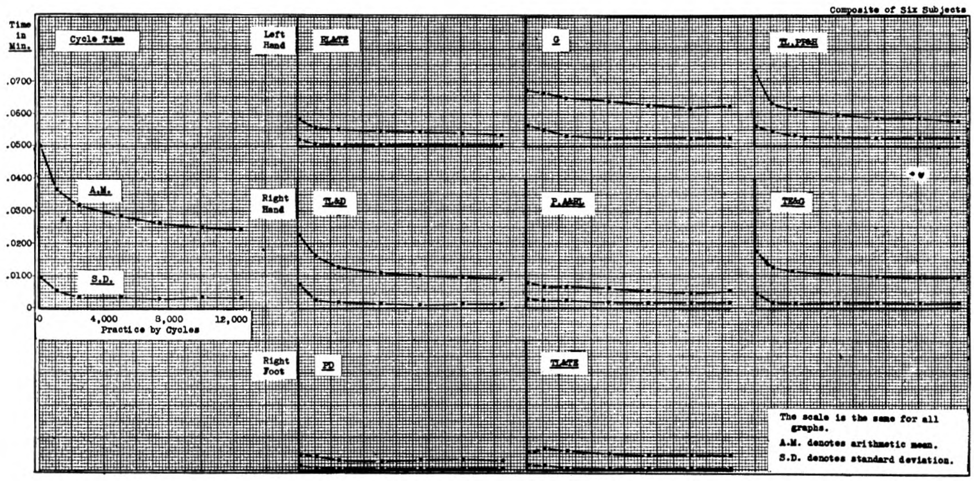

Fig. 8 Learning Curves of Cycle Times and Elements; Composite of Six Subjects; Arithmetic Mean and Standard Deviation

Left-Hand (LH) Operations

Release Load and Transport Empty (RL&TE). This is the element of the left-hand releasing the part previously grasped by the tweezers and then reaching to the supply tray for the next part. The trend of the average time from first sample to last sample shows that this element is one in which practice has considerable effect on the performance time. The time was reduced to 45 per cent of the starting time.

This is an interesting observation, especially in consideration of what the subject was doing. It is during this element that he

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	59.82	33.78	44.54	52.23	45.82	36.53	45.45

looks at the supply of parts, selects a particular part, and then considers how he will transport load and pre-position the part before he grasps it. There are four ways of pre-positioning a part, depending upon which of the four possible ways the part is fortuitously reposing in the supply tray. It seems that the poor initial time of some of the subjects may be the result of indecision during the process of determining how to pre-position.

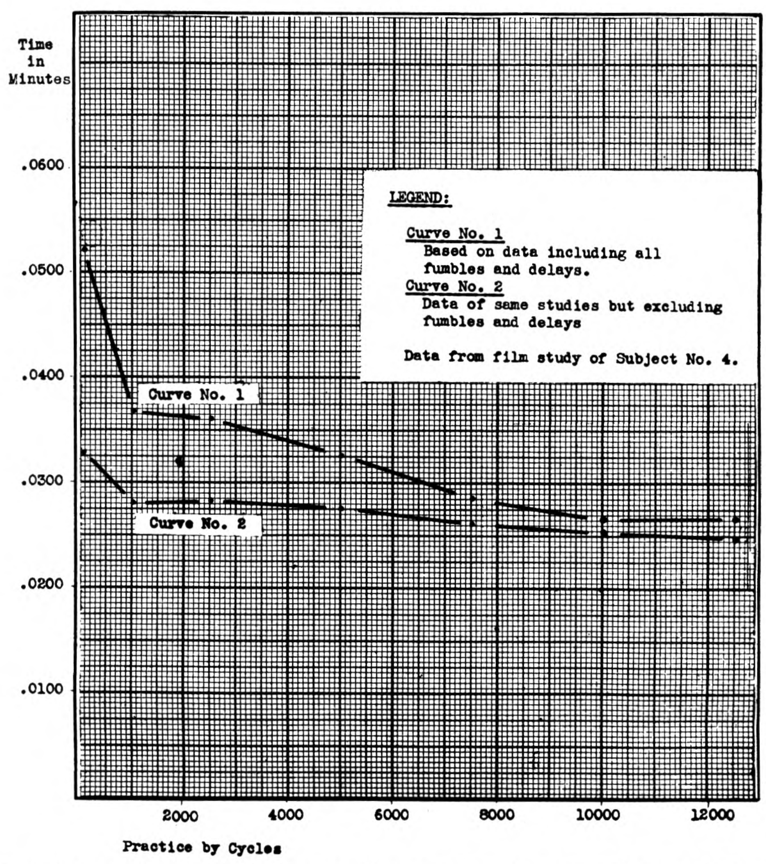

Fig. 9 Curves Showing Effects of Fumbles and Delays on Learning Curve

As soon as the subject completes this selecting part of the task, he looks to the die to direct the right hand, so he then inmediately has another mental problem. Since these mental problems are solved in chronological order, mental agility is an important factor in the TE therblig time. The decision time is probably the predominant part of the element time at the outset, but reduces to become a minor influence in the later stages.

A study of the standard deviation shows a reduction in the early stages and a rise toward the final stages in the study. The reason for this is not readily evident but, undoubtedly, the phenomenon carries some significance because it is contrary to normal expectancy. The subjects generally show the same trend both in change in the average time and in the standard deviation.

Grasp (G). Because of the nature of the timing apparatus, the grasp includes a little of the transport motion. Other than this, the subject's visual attention at this stage of the cycle is in directing the right hand to place the part in the die.

Three of the subjects required more time at the end of the study for this element than they did at the beginning.[5] The general trend, however, does show a slight reduction in time. The average of the individual scores shows that the final stage amounted to 65.65 per cent of the time for the initial performance.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	106.93	68.86	66.60	54.26	54.01	43.21	65.65

In studying the trends of the standard deviation of the learning curve there is a common trend for the final performance to have a lesser deviation than the first performance.

Transport Load, Pre-Position and Hold (TL,PP&H). This element is the one in which the part is carried by the left hand to the tweezers; it includes turning the part ready to grasp and is held in place until the tweezers close.

There is a strong tendency rapidly to decrease the performance time. In considering the "last sample to first sample ratio" the reduction is to 38.12 per cent of the starting time.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	35.54	19.15	48.81	35.53	27.11	62.57	38.12

A study of the graphs Fig. 8, indicates a rather strong tendency to conform to the normally conceived learning curve.

Right-Hand (RH) Operations

Transport Load and Delay (TL&D). This is the element of the cycle in which the right hand carries the part held by the tweezers toward the die. It does not include placing the part in the die.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	43.04	33.49	50.45	37.88	43.72	38.79	41.23

The table shows a fairly consistent reduction in time and the graph shows the curves to be quite normal and without peaks or flutter, averaging 41.23 per cent of the starting time.

It is during this phase of the cycle that the attention of the eyes is directed toward the selection of a part, and also in directing the left hand to the part. Also, during this element the eyes shift to the die and thus the attention is shifted to the direction of the right hand. As the part held by the tweezers penetrates the beam of light in front of the die, the element is terminated.

Considering what is going on during the element, the mental task of selecting, the directing of a hand, the shifting of the eyes to the die and the shifting of the attention to a different problem, the consistency of the progress is surprising, as is also the consistency of the standard-deviation learning curve. It is also noted that the coefficient of variation is particularly low for this element.

Furthermore, this particular element is one in which, possibly, a visual pattern change might be manifested. After a few thousand cycles of practice, subjects would now and then use only two fixations to a cycle while at other times they use three fixations per cycle, according to the original instructions.

Position, Assemble, and Release Load (P,A&RL). This element is for the short motion in locating the part in the die.

The impression from first observation of the data is that this series is a most erratic group of learning curves. Two of the subjects, after several hundred cycles of practice, took more time than for the initial sample. Five of the subjects took more time in the last sample than they did in the previous sample. In addition, there were other peaks and mounds. However, the final study shows that, on the average, the subjects were performing in 66.7 per cent of their starting time.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	67.03	52.69	52.81	90.54	74.91	62.18	66.69

In the section of therblig findings based on the film data, it was found that the RL was performed very consistently, consequently if the RL were deducted, the remaining P and A must be the erratic elements.

In reviewing what is taking place, it should be noted that it is this phase of the cycle in which the part is placed over the pilot pins. And although the pilot pins facilitate the positioning, in many cases the subjects do not locate the part exactly right in the first locating movement, and small adjustment movements are necessary. If the subject tries to perform this motion too fast, he probably will misplace the part and waste time in recovering it. Less over-all time would be required if the subject worked more slowly but exactly.

The standard-deviation learning curve is likewise erratic for each subject and also leaves much to be explained. The coefficient of variation for this therblig exceeds that of any others.

Transport Empty and Grasp (TE&G). This element follows the release of the tweezers at the die. The TE&G includes the reaching of the right hand and tweezers to the transfer area, and the closing of the tweezers on the part.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	83.62	43.94	60.61	47.32	46.86	49.42	55.3

Right-Foot (RF) Operations

Pedal Delay (PD). This element is defined as the delay between the opening of the tweezers and the closing of the mercury switch actuated by the foot pedal.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent..............	68.43	99.84	62.52	14.34	60.20	74.23	63.26

The inconsistency of data in the learning trend for each subject as well as the inconsistency between subjects shows that these curves involve a phenomenon distinctly different from that previously encountered.

The subjects were instructed to press the pedal as soon as the part in the die was released. However, since a fast follow-up in the foot action after the release does not necessarily reduce the cycle

[5] All subjects to a minor extent performed slight variations in the cycle at the outset of the study. However, practice had the effect of making all subjects perform more like the instructed method.

time, the subjects all floundered around, trying faster and slower times as they pleased. More than 50 per cent of the time the foot was not busy, so it really made little difference how fast or slow the subjects coordinated the hand RL with the pedal TL. Accordingly, each subject took whatever time suited him.

It is interesting to note that only one subject early in the study seemed to find a rhythm which she tended to maintain throughout the study.

Transport Loaded and Transport Empty (TL&TE). In somewhat the same respect, the TL&TE is considered. This element is the time to press and release the pedal.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent......	118.07	121.38	68.91	118.68	67.70	42.02	89.46

There is no consistency within the individual studies, except in the case of one subject who tended to perform in about the same time throughout the study; and there is no consistency between subjects. This element is analyzed as one which is not important in cycle-time performance, consequently, the subjects perform it just as they wish to.

Fumbles and Delays

In analyzing the motion-picture film of one of the subjects, fumbles or delays were noted in varying frequency among the several therbligs. Relating the number of fumbles and delays to the total number of cycles shows clearly which are the troublesome elements.

Any hand movement not performed in accordance with instructions has, for the purpose of this study, been considered a fumble. Any retardation not necessitated by the methods as prescribed has been considered a delay. For example, when the left hand holds the part for the tweezers to grasp, the therblig is an instructed hold; however, were there a fumble in bringing the part to the grasping area, causing the hand to wait for the part, then the delay occurring in the grasp therblig of the right hand would be included as a delay. In this way a fumble or delay by one hand can result in a delay for the other hand. Table 2 shows in which therbligs and to what extent the trouble occurs by percentage, at the various stages of acquisition of skill.

TABLE 2 PERCENTAGE OF THERBLIGS IN WHICH FUMBLES OR DELAYS OCCURRED

After practicing, cycles	Left hand RL	Left hand TE	Left hand G	Left hand TL & PP	Left hand H	Right hand H	Right hand Tl, P&A	Right hand RL	Right hand TE	Right hand G
50	0.0	87.5	25.0	50.0	0.0	0.0	50.0	6.3	56.3	0.0
1050	0.0	6.7	40.0	93.3	0.0	0.0	73.3	0.0	0.0	6.7
2550	0.0	0.0	76.7	56.7	0.0	0.0	70.0	0.0	0.0	23.3
5050	0.0	0.0	26.7	54.8	0.0	0.0	74.2	0.0	0.0	29.0
7550	0.0	0.0	15.4	11.1	0.0	0.0	11.1	0.0	0.0	18.5
10050	0.0	0.0	15.1	5.9	2.9	0.0	5.9	0.0	0.0	8.8
12550	0.0	0.0	23.1	19.2	0.0	0.0	15.4	0.0	7.7	3.9

Effects of Fumbles and Delays on a Learning Curve

Fumbles and delays, which occur due to the lack of experience, have much to do with the form of a learning curve. The film analysis permits the graphing of two learning curves as follows:

1 One showing the normal curve, i.e., including all fumbles and delays.

2 The other showing a learning curve which excludes fumbles and delays, Fig. 9.

By excluding these obvious fumbles and delays, the learning curve is surprisingly flat. At the outset, the cycle time, excluding fumbles and delays, amounts to 63 per cent of the total time and, at the finish of the study, the data excluding fumbles and delays amounts to 93 per cent.

There follows a mathematical analysis of the differences of the two learning curves:

Cycle time at outset................	0.0521 Min
Cycle time at finish................	0.0272 Min
Improvement.......................	0.0249 Min

The improved performance can be traced to the following overlapping causes:

Reduction in fumbles and delays............	0.0168 Min
Faster performance.......................	0.0081 Min
Total................................	0.0249 Min

This analysis indicates that of the improvement obtained, about two thirds may be assigned to the elimination of obvious fumbles and delays and only about one third may be assigned to faster movements and better coordination. This proportion was somewhat surprising and may require reconsideration of some of the concepts of learning. It may develop that the phenomenon known as rhythm, coordination, deftness and smoothness, as distinguished from awkwardness, may consist not of a general status but of the absence of specific and definable fumbles and delays.

Change in Visual Pattern

During the early part of the learning period, it was observed that occasionally the subjects did not perform the regular visual sequence. While normally the subjects were required to perform according to the instructions, this departure from the standard method was regarded as having special significance, and the subjects were allowed to develop this departure in a natural manner.

According to the instructions, three visual fixations were to be performed in conjunction with the hand movements of each cycle. The first fixation was to occur as the left hand reached to grasp the part from the supply tray; the eyes then were focused on the supply tray during the selection of a part. The second fixation occurred as the right hand placed the part in the die, and at this time the eyes were focused on the die. The third fixation occurred as the tweezers in the right hand grasped the part being held by the left hand; the eyes then were focused on the part, Fig. 3. After several thousand cycles of practice, the tendency of the subject was sometimes to use three fixations and sometimes to use two, as distinguished from the exclusive use of either two or three. This mixture may in reality have been an intermediate stage of shifting from the use of three fixations to two fixations.

It is natural to question whether the subjects should have been instructed to use a two-fixation pattern. In response to this question, there is well-founded opinion that to do this would have caused considerable trouble. In the early training, looking in sequence toward the three places was essential.

When the subjects used the two-fixation pattern, the paths of the hand movements were the same; the only obvious difference was the points of fixations. The eyes would first fixate on the part during the transfer from the left hand to the tweezers in the right hand. The second fixation would occur when the eyes were focused on the die when placing the part in the die, as illustrated in Fig. 3.

Comparison of Time for Two- and Three-Fixation Methods

The film study provides quantitative data showing what proportion of cycles was being performed by the two-fixation method, and also a comparison of the average time values for each of the two methods during the transition.

	After practicing							
Cycles	5050		7550		10,050		12,550[a]	
No. of fixations	3	2	3	2	3	2	3	2
Percentage of sample	90.0	10.0	81.0	19.0	56.3	43.7	60.0	40.0
Time (avg), min	0.0323	0.040	0.0281	0.038	0.0279	0.0299	0.0363	0.028

[a] Because of scanty data taken at the 12,550 period, the 10,050 stage is generally referred to because it is statistically more significant.

In summary, it is observed after 10,050[a] cycles of practice:

1 That 44 per cent of the sample consists of the two-fixation method.

2 That a trend of the average time per cycle for each of the two methods indicates that, if the study had been carried on longer, that the two-fixation method would have occurred still more frequently.

3 That the two-fixation method would have taken less time than the three-fixation method, if the study had been carried on further.

This is regarded as a rather original finding and is important because:

1 The subjects (with one exception) were not aware that this change was taking place. They believed that they were following the instructions (and they actually were so far as their hands were concerned) but somehow they were not conscious of the visual pattern change that was taking place until toward the end of the study.

2 Instructions by industrial engineers seldom include directions for the visual pattern. The fact that it appears natural for an operator to change from one method to another without showing apparent trouble such as an irregularity in the cycle-time learning curve is a valuable finding. It is recognized that changes of hand movements alone will cause various effects on the learning curve, but the data do not seem to manifest any of the effects that the introduction of a change in method would be expected to show. Also, it is of practical importance to the industrial engineer to know that he can instruct a worker at the beginning to perform three fixations, and mention that ultimately only two fixations should be included. This may effect a faster learning period.

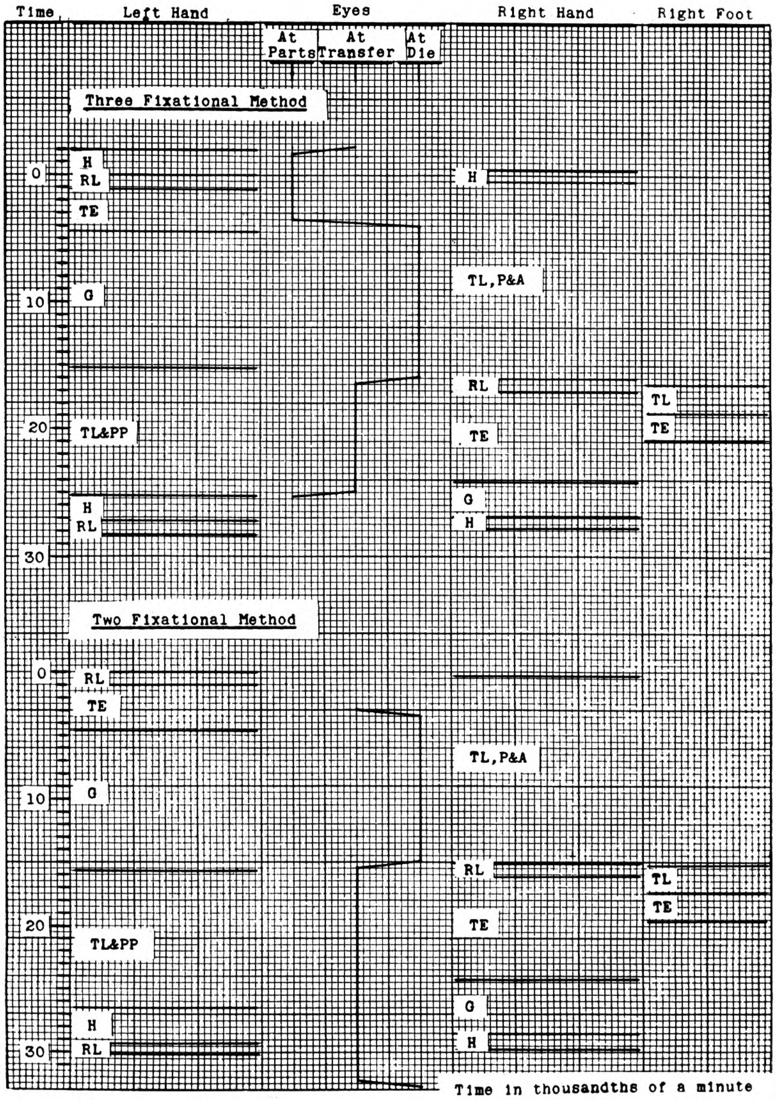

FIG. 10 SIMULTANEOUS-MOTION CHART, SHOWING EYE MOVEMENTS

TABLE 3 PERCENTAGE CHANGE FROM FINAL NORMAL RUN TO SPEED RUN

	Left hand			Right hand				Right foot	
Subject no.	RL&TE	G	Tl, PP&H	TL&D	P,A& RL	TE&G	Cycle time	PD	TL&TE
1	− 3.33	− 7.90	− 3.18	−17.82	− 7.96	+ 0.65	−7.49	+ 3.33	−11.54
2	+ 0.07	−19.42	+39.01	− 9.55	− 6.15	+13.72	+0.29	− 1.83	−11.18
3	− 2.46	− 3.45	+13.15	− 4.30	+ 7.29	+ 6.75	+0.85	− 3.35	+12.66
4	− 8.98	− 5.24	+ 0.38	−18.40	+ 7.11	+ 4.05	−5.41	+37.35	− 4.89
5	−13.08	+ 2.07	−10.70	−19.79	+20.01	− 0.69	−4.18	−33.83	+ 0.89
6	+ 5.51	+ 9.55	−16.66	− 8.75	+10.89	+ 7.99	−1.87	− 5.48	+16.27
Weighted avg, per cent change.......	− 3.95	− 4.63	+ 2.82	−13.18	+ 5.15	+ 5.21	−3.12	− 7.08	− 1.80

NOTE: A negative sign indicates that the speed run was faster than the normal run, and a positive figure indicates that the "speed run" was actually slower.

TABLE 4 PERCENTAGE CHANGE FROM NORMAL SAMPLE TO SOLDIERING SAMPLE

	Left hand			Right hand				Right foot	
Subject no.	RL&TE	G	TL, PP &H	TL&D	P,A& RL	TE&G	Cycle time	PD	TL& TE
1	+22.75	+18.27	+118.63	+ 43.91	+152.32	+23.74	+59.05	+ 13.08	+34.01
2	+24.90	+ 8.23	+ 90.59	+ 18.44	+ 29.41	+35.20	+31.42	+ 1.62	−12.35
3	+25.36	+47.31	+ 46.82	+ 62.24	+ 16.19	+42.82	+42.80	+ 55.52	+44.61
4	+39.27	+30.58	+160.09	+180.34	− 18.86	+41.17	+76.24	+3349.39	− 7.22
5	+12.71	+23.27	+ 33.97	+ 30.59	+ 1.83	+31.90	+20.90	+ 36.79	+41.29
6	+ 7.83	+ 6.20	+ 10.70	+ 14.70	− 2.05	+19.51	+10.96	+ 23.19	+38.51
Weighted avg, per cent change......	+21.84	+21.94	+ 74.75	+ 56.86	+ 27.23	+32.15	+40.99	+ 195.13	+25.48

NOTE: A positive sign indicates more time during soldiering than during last normal run.

Study of Relation of Eye Movements to Hand Movements

The film data permit making a simultaneous-motion chart, Fig. 10, of the events as they occurred after practicing 10,050 cycles. It shows the relation of one hand movement to the other, the relative duration of the visual fixations, and the instant the eyes started to shift. It also includes the action of the leg movement.

The two-fixation cycles are segregated from the three-fixation cycles, and this offers additional information regarding the manner in which the work was performed.

It was noted that in all therbligs requiring eye fixation, the eyes left the scene of action before the action was completed, Fig. 10. In making the transfer of the part from the left hand to the tweezers, the eyes moved away before the tweezers closed on the part. Likewise, when the left hand went to grasp a part from the supply tray, the eye moved away before the left hand had completed the grasp. Finally in locating the part in the die, the eyes moved away before the tweezers released the part.

From the viewpoint of the industrial engineer, the practical result is that the eye precedes the associated TE movement of the hand. It is from this viewpoint that quantitative measurements were made.

These findings should be considered in the preparation of a synthetic motion cycle. A careful analysis of eye movements and eye fixations should be considered. One could not say that the eyes always leave so many thousandths of a minute ahead of a TE or a TL, because sometimes it may be necessary to check visually the part in the die after the hand is away from the die. However, a careful analysis of the visual work in conjunction with the hand work would make possible a reasonably accurate forecast of the coordination to be found between these members in a motion pattern.

Speed Run

At the end of the last normal run each subject performed 15 cycles as fast as he could. This is called the speed run. It was explained to each subject that data were wanted which would reflect his most skillful performance at maximum effort. To be sure that fatigue would not affect the results, only 15 cycles of data were to be taken after the subject had gained "momentum" by performing a few cycles. Table 3 has been prepared to show the percentage change from the final normal run to the speed run.

The following conclusions were drawn from Table 3:

1 The subjects averaged 3 per cent faster performance for the speed run.

2 Performance during the regular kymograph periods, although without specific directions for speed performance, was evidently at slightly slower speed than top speed.

3 It appears that the subjects all could perform the TL&D element faster at will, but it was at the expense of the other two elements for that hand.

4 The six subjects do not seem to follow a common trend in the changes caused by the speed run.

Soldiering

If each subject, after acquiring a known degree of skill, decided to restrict his output, what would happen to the elemental time values? In order to explore this question the subjects agreed that they would hold back from top performance but only to such an extent that the tendency to hold back would not be apparent. The amount that each held back was a problem each individually solved. Table 4 was prepared to show the percentage change from the final normal sample to the soldiering sample.

The following conclusions are drawn from these data:

1 The extent to which the subjects increased performance time and yet considered it undetectable varies widely ranging from 11 to 76 per cent, averaging 41 per cent.

2 The averages of all elements are affected to some extent, the subjects averaging an increase of more than 20 per cent for each element.

3 The element showing the greatest increase in time was the TL,PP&H. This element provided a very plausible excuse in performing slowly, because, as one observed the workers, he could see that the act involved thinking of how to turn the part was causing the worker in this element to make slow movements.

4 There appears to be no common pattern or trend in soldiering.

Conclusions

Ordinarily a study of the conclusions reached in an investigation of this type will give to the reader a fair understanding of the problems encountered in the study. However, because of the pioneering nature of this study and because of the recondite nature of the findings, a study of the conclusions alone appears in this case to be inadequate, unless supplemented with some associated study of the details. For these reasons also, a special effort has been made to include, in the body of the University of Iowa Bulletin,[4] much of the original data and all essential details.

Summarized Findings of the Investigation

The learning curves for the various therbligs show marked differences in characteristics. Some of them fall quite rapidly and others quite slowly. Clearly then the learning curve for the total cycle will be determined by the proportions in which the task contains rapidly learned and slowly learned therbligs. In the case of therbligs having no bearing on the cycle time, there is no definite tendency to reduce the therblig time as practice continues.

The elimination of noticeable fumbles and delays was in this study a greater factor in improving the cycle time than was the increase in speed alone. Here again, some therbligs are more susceptible to fumbles and delays than others, and the cycle-time performance will be influenced by the kind of therbligs which go to make up the cycle.

The data are inconclusive as to whether the learning curve for any one therblig will be alike or unlike for all subjects. The data are also inconclusive as to whether any one subject will adopt a predictable pattern for handling the various therbligs.

It was found that the visual pattern used by the subjects changed during the course of the study, and for the most part this change took place unknown to the subjects themselves. It was also found that the eyes can and do leave the scene of action before the action is completed.

An effort by the subjects to slow up their pace as far as possible without the change being detected resulted in changes in time for all therbligs, but not uniformly. This same lack of uniformity was evident during an effort by the subjects to "race."

Discussion

W. R. Mullee.[6] Many of us in the field of time-and-motion study have wondered how long it takes an operator to learn. We have been unable to ascertain the facts because we could not standardize the conditions as well as in the laboratory. This paper clearly indicates that time studies should not be taken on an operator before he has practiced the operation perhaps 1000 cycles.

In some plants where standards are set from time studies rather than from data, it can easily be seen that the resulting time standard will be too high. A time study which is taken at the beginning of the training period will be about $1^1/_2$ times as high as that obtained after 1000 cycles, and almost twice as high as that taken after, say, 10,000 cycles. This throws a great burden on the time-study investigator in attempting to rate such a wide variation in performance. Many of us feel that time studies cannot be rated accurately for performances which vary more than 20 per cent from normal for any given work element. The authors' experience seems clearly to indicate the desirability of not taking studies until the operator has become sufficiently skilled to eliminate at least most of the fumbling, etc.

It would be very interesting if another experiment were carried out to determine rapidity of learning when both hands are used simultaneously. From observations made in the plant, it seems evident that a longer period is required under such conditions.

Another interesting variation would be to determine the effect on learning if the number of therbligs to be learned were increased, i.e., if the cycle were made longer by introducing additional work elements. Also, to compare the rate of improvement under such conditions with that shown in this paper.

Summing up the discussion, the paper seems to prove the desirability of establishing time standards from data as compared with individual time studies; and indicates that jobs should be planned to avoid, as much as possible, conditions which might cause fumbles. Two thirds of the improvement noted was in the elimination of fumbling and one third was due to faster movements and better coordination.

[6] Supervisor, Standards Department, American Hard Rubber Company, Butler, N. J. Mem. A.S.M.E.

A New Steam Engine and Boiler

By S. L. G. Knox,[1] Englewood, N. J., and J. I. Yellott,[2] Chicago, Ill.

The paper presents descriptions of a new type of steam engine and a forced-circulation boiler which were designed by Mr. Knox and tested by Professor Yellott. The engine has a maximum capacity of 90 hp and is a high-speed compound reversible unit with uniflow exhaust. The description relates particularly to the unusual features of the inlet valves. The boiler employs a new principle for obtaining forced circulation. An impeller placed longitudinally in a lower drum exerts a pumping action which results in a uniform and rapid circulation through a number of riser tubes which comprise the walls of the combustion chamber. The paper is presented as a progress report. It is the authors' purpose by this means to present new and improved ways of accomplishing familiar ends, rather than to give an exhaustive discussion of the development of the particular units involved.

THE current interest in high-pressure prime movers and forced-circulation steam generators gives timeliness to the following brief description of a compact steam plant which embodies both of these features. Originally designed to provide an efficient, powerful, and directly reversible unit for mobile purposes, the plant utilizes a high-speed compound uniflow engine, for which highly superheated steam is supplied at pressures up to 700 psi by a unique forced-circulation boiler. In its present state, the plant is noncondensing, but the engine may be discharged into a condenser and utilize to good advantage the additional energy thus made available.

The first part of the paper will be devoted to a description of the unique features of the engine. Results are also presented of a test run on the engine at the Stevens Institute of Technology.

The second section of the paper will describe the steam generator, which combines the merits of a multitubular water-level boiler with the advantages of forced circulation. Test results are also presented.

The Knox Engine

The prime mover of this plant is a high-speed compound uniflow engine of the steam-reverse type.[3] It is designed to operate at 1000 rpm, taking steam at 700 psi 750 F, and exhausting either to atmosphere or to a condenser. In brief, the engine is double-expanding, with one high-pressure and two low-pressure cylinders. The inlet valves are of the piston type, with certain unique variations which will be described later. These valves are actuated by an auxiliary crankshaft, which is driven by gears from the main crankshaft. Exhaust from all cylinders is accomplished by uniflow ports, with auxiliary exhaust provided by the piston valves.

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[3] A steam-reverse engine is one in which the direction of rotation is reversed by changing the direction of flow of the steam. This is accomplished by using one set of ports in the valves as steam-admission ports in one direction, and as exhaust valves in the other direction.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

The outstanding feature of the engine is the valve gear, by which a number of useful functions are performed without introducing more moving parts than the minimum number required for a simple nonreversing engine with the same number of cylinders. Most important of these functions are the provision of expansion and the automatic increase of torque at starting and low speeds, without additional mechanism. The result of the special features of the engine is a steam consumption of 14.6 lb per bhp-hr (noncondensing), and an engine efficiency of 53.3 per cent, obtained under test when the engine was loaded to 70 hp or about 75 per cent of its capacity.

The relative size and general appearance of the unit are shown in Fig. 1. The general plan of construction is shown in Fig. 2,

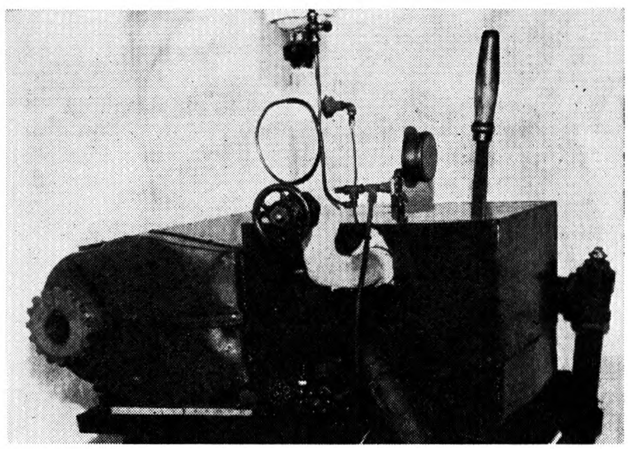

Fig. 1 The Knox Engine

which gives the principal details of construction of the engine, but omits the passages by which the steam makes its way from the valves to the cylinders. The single high-pressure cylinder, $3^1/_4 \times 4^1/_2$ in., is located between the two low-pressure cylinders, $4^1/_2 \times 4^1/_2$ in., as shown in the cross section of the cylinder block, Fig. 3.

Steam is introduced through a control valve which serves both as a throttle and as a steam-distributing valve. The direction of rotation of the engine is changed from forward to reverse simply by pushing the control lever to its forward or reverse position.

Sequence of Valve Events

When running forward, the steam passes from the control valve, designated as V in Figs. 2 and 3, to the inner edges of the high-pressure valve, shown as a in Fig. 3. This valve, as well as the other two valves b and c, is a piston valve. These differ from the usual piston valves in two important respects: First, the valve has a rolling motion as well as a reciprocating motion; thus the path of any point on the valve is an ellipse. The second unique feature is the serrated edges, which can be seen in Fig. 4. The combination of this unusual motion with the shape of the valve edges gives rise to the characteristics which will be discussed.

The high-pressure cylinder discharges into a receiver space, which is composed of the unused volume within the engine casting. The discharge takes place through exhaust ports in the middle of the cylinder which are uncovered by the piston in the conventional uniflow manner. Auxiliary exhaust, necessary to keep compression down to a reasonable amount, is provided by the piston valves mentioned previously.

As is explained later, each valve is connected to two cylinders; the outer edges to one cylinder, the inner edges to another. If the direction of rotation of the engine is such that the outer

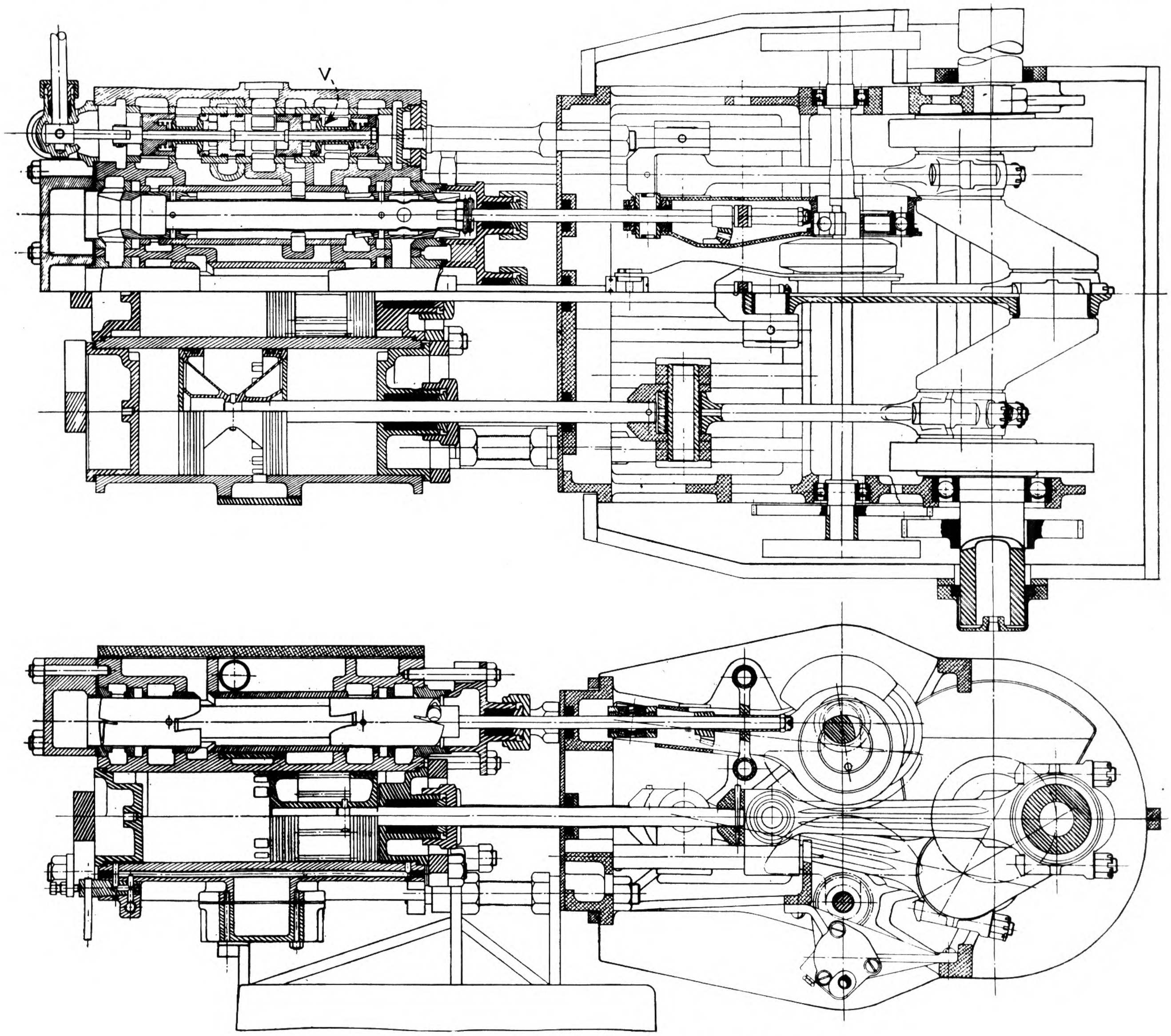

Fig. 2 Plan and Cross Section of the Engine

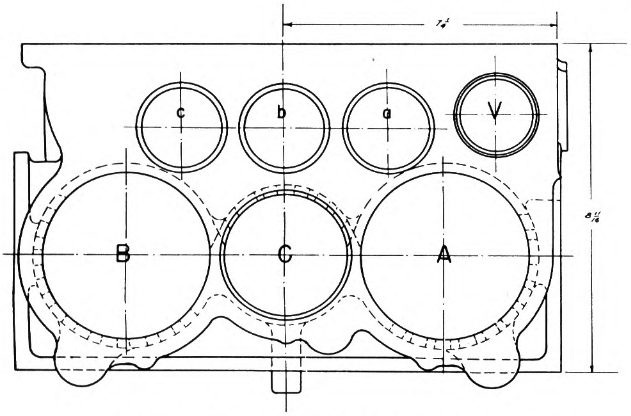

Fig. 3 Cross Section of Cylinder Block

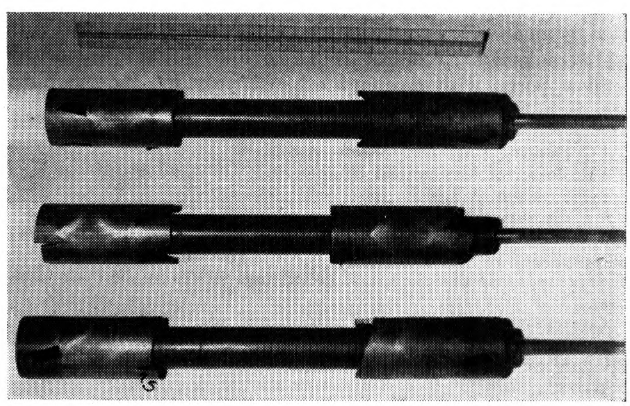

Fig. 4 Piston Valves

edges of a valve admit steam to a cylinder, the inner edges of the same valve will open during the exhaust stroke of the other cylinder, thus providing auxiliary exhaust for the second cylinder without any additional parts being required.

Running forward, steam from the receiver is directed by the outer edges of valve *b* to low-pressure cylinder *A* (Fig. 3) and by the inner edges of valve *c* to the low-pressure cylinder *B*. These two cylinders exhaust through central ports similar to those in the high-pressure cylinder, the steam being carried through a manifold to the exhaust line. Auxiliary exhaust is provided for cylinder *B* by the inner edges of valve *b* and for cylinder *C* by the outer edges of valve *c*.

The unique feature of this engine, as compared with other steam-reverse engines, lies in the fact that each valve supplies a particular cylinder when the engine is running forward, but a different cylinder when the engine is reversed. When the control valve is thrown to the reverse position, steam is no longer admitted to the inner edges of valve *a*, but rather to the outer edges of valve *c*. In the same manner, steam passes from the receiver to the outer edges of valve *a* and the inner edges of valve *b*, whence it passes to cylinders *A* and *B*, respectively. The opposite edges of the valves again act as auxiliary relief valves. It is by this means that a cutoff of 30 to 40 per cent (depending upon the slope of the serrated edges) can be obtained in all the cylinders of this engine. Earlier steam-reverse engines necessarily had 100 per cent cutoff and consequently their steam rate was several times as high as that of the present engine.

The usefulness of the serrated edges in combination with the unusual rolling motion, lies in the fact that the effective cutoff can be lengthened, thus increasing the torque of the engine. The rolling motion of the valve is imparted by a simple arrangement which combines a segmental bevel pinion on the valve stem with a segment of bevel gear on the eccentric strap. The vertical component of the motion of the strap, caused by the rotation of the eccentric, causes the valve stem to turn first in one direction and then in the other. Both stem and strap have the same reciprocal motion, and so the gear segments are always in mesh.

The serrated forward admission edges of the valves, shown in Fig. 4, exemplify one of the advantages which may be obtained as a result of the combined reciprocating-and-oscillating movement of the valves. In an engine where high power-to-weight ratio is desired, as was the case in the present engine, this construction permits lengthening the admission period for a given angle of advance of the eccentrics. This angle is limited to a rather narrow range, to obtain reversibility without serious reduction of efficiency. Because of the serrations, the valve, rolling over as it reaches its dead center, remains open for a longer period than would otherwise be possible with the proper angle of lead.

In engines where the power-to-weight ratio is not a prime consideration, these serrations can be omitted and the admission edges provided with piston rings, thus giving a shorter effective cutoff with resultant increase in total expansion ratio.

At *S*, Fig. 4, is shown a small slot or notch running longitudinally. The purpose of this slot is to lengthen the cutoff beyond that which would be possible with only the reciprocating motion of the valve, when the eccentrics are set at an angle of advance which gives an economical point of cutoff. If this slot were exposed to steam when the valve approaches the point of admission, steam would be admitted to the cylinder so far in advance of dead center that the engine would not operate. However, due to the rolling motion of the valve, the slot is covered by a bridge as it approaches the point of admission, and is rolled out into the steam space only after dead center has been passed. It remains exposed to steam until nearly the end of the stroke, however. Its width ordinarily would be from 2 to 3 per cent of the circumference of the valve. Thus, when the engine is starting or running at low speed, it is supplied with steam at approximately full pressure during practically the entire stroke. As it speeds up, the amount of steam which gets through per stroke becomes less important. At maximum speed the effect on efficiency is negligible. Thus, it is possible to obtain the effect of a long cutoff, and the resulting increased torque at starting and low speeds, with a fixed economic eccentric position. This eccentric position would normally give the same short cutoff at all speeds, except for this slot and the rolling motion of the valve. In this way is obtained the same effect of long cutoff and maximum torque at low speeds, with short cutoff and high efficiency at high speeds, which in other engines is only obtainable with a number of extra moving parts, requiring hand or governor manipulation.

Engine balance, particularly important because of the high operating speed, is accomplished by two sets of counterbalances. One set is mounted on the eccentric shaft which rotates in one direction in a plane above the cylinders. The other set is mounted on the oil-pump shaft which rotates in the opposite direction in a plane below the cylinders. The system is so effective that there is virtually no vibration when the engine is running either forward or in reverse.

Lubrication of the bearings, cranks, and eccentrics is accomplished by splash and by forced circulation from a small vane pump, through holes drilled in the crankshaft, to the surface of each crankpin. Lubrication of the valve and pistons is accomplished by a multiplunger lubricator which sends cylinder oil into the main line and to each end of the three valves.

The over-all dimensions of the engine are shown in Fig. 1. The weight of the engine is 450 lb and the maximum power of the engine is 90 hp, giving a weight of 5 lb per bhp.

The Engine Tests

A number of trial tests were made with the engine during its development period. The tests followed standard practice. Pressures and temperatures were measured with calibrated gages and thermometers. Condensate was weighed to determine the steam consumption of the engine. Back pressure was atmospheric in all tests.

The load on the engine during these tests was supplied by a water brake which was constructed especially for this purpose. The design established by Professor E. P. Culver was followed, and the brake operated in a very satisfactory manner. This type of brake was particularly desirable because it operated equally well in both directions of rotation.

The limitation on the output of the engine was imposed by the boiler which at the time of the engine tests was unable to deliver more than 1000 lb of steam per hr with smokeless combustion. Later the combustion was improved until the boiler developed more than 1500 lb per hr without smoke.

The results given inTable 1 are typical of those obtained during the tests of the engine.

TABLE 1 TYPICAL ENGINE TEST RESULTS

Length of run, min	30
Initial pressure, psi abs	595
Initial temperature, F	750
Back pressure, psi abs	15.2
Rotative speed, rpm	1020
Brake output, hp	69
Steam used, lb	503
Steam rate, lb per bhp-hr	14.6
Ideal steam rate, lb per bhp-hr	7.79
Engine efficiency, per cent	53.3
Engine heat rate, Btu per hp-hr	17500
Engine thermal efficiency, per cent	14.55

The present engine was the first of its type and was designed especially with a view to light weight per horsepower-hour, with the sacrifice of the maximum efficiency otherwise obtainable. In view of this, of its small size and the fact that no changes have been made in the design as the result of experience, although various improvements have been indicated, it is reasonable to assume that, with large engines, running condensing, steam consumption much below the 14.6 lb per bhp-hr obtained with the present engine should be realized.

If, as may reasonably be expected, a water rate of 10 lb or better per bhp-hr should be attained, the fuel consumed per horsepower-hour, while considerably greater than that of a Diesel engine, would cost less because of the much lower price per gallon of fuel usable in the furnace of a boiler, as compared with that required in a Diesel.

The Knox Boiler

This boiler, which was developed to supply steam to the Knox engine, had to be compact, relatively light, and capable of pro-

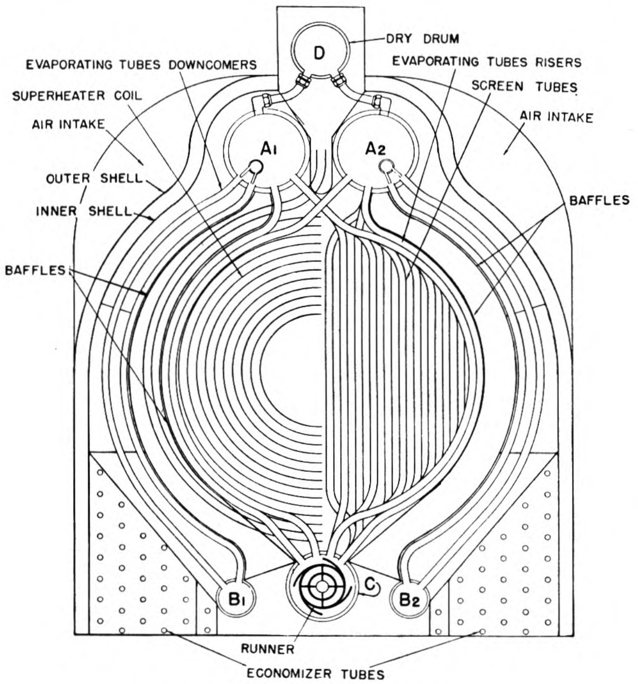

Fig. 5 Cross Section of Boiler

ducing 1500 lb per hr of steam at 700 psi 750 F with smokeless combustion and high efficiency. The necessity for forced circulation was immediately realized, but the difficulty of controlling the once-through or coil boiler made it desirable to retain the stability and reliability of the conventional multitubular drum boiler.

These requirements were met by using a new method of forced circulation, whereby a simple impeller in the middle lower drum, Fig. 5, causes the water to flow up through the riser tubes which surround the combustion chamber.

The feedwater is pumped into the two steam drums A_1 and A_2, after passing through the economizers which will be discussed later. Leading downward from these two drums are the downcomers, which are located in the third pass of the present boiler, and which connect to the two smaller lower drums B_1 and B_2. The center drum C at the bottom of the boiler is connected to the outer drums by headers at both ends, and within this center drum is located a hollow impeller. Figs. 5 and 6 show these points quite clearly. This impeller is driven by a shaft which emerges from the header through a simple stuffing box. When the impeller is rotated, the pumping action creates a pressure which forces the water in drum C to pass upward through the risers which form the walls of the combustion chamber. The suction caused by the displacement of this water causes the water in B_1 and B_2 to flow into the middle drum, and thus a vigorous forced circulation is established. This circulation is quite uniform along the length of the drum, as is shown in Fig. 7. This illustration is from a photograph taken when the boiler was not under pressure, but when the impeller was being operated. Glass plates pressed against the end of the drum prevented the water from spilling out of the boiler. Rough measurements indicate that the velocity is about 4 fps under the circumstances prevailing when the photograph, Fig. 7, was taken.

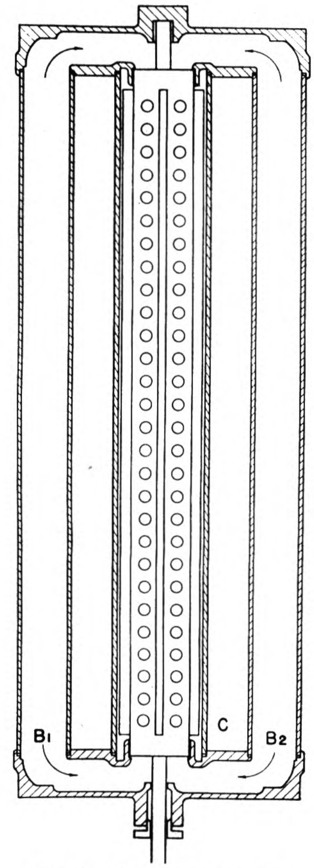

Fig. 6 Cross Section of Lower Drums

Since this pumping action is assisted by the natural circulation, it is obviously yet more powerful when the boiler is generating steam. In fact, the pump may be considered as a booster, when applied to a boiler having natural circulation, adding to the velocity which natural circulation will produce. It will be realized that this internally assisted circulation differs in several important respects from other well-known forced-circulation principles. By using the lower drum as the casing of the pump, the expense of a separate high-pressure pump is eliminated and, of far greater importance, uniform circulation is obtained. Balancing of the individual water circuits is automatically accomplished, since the added pressure comes from the centrifugal action of the impeller, which is uniform along the length of the drum. There is no tendency for the first tubes to rob the remaining risers, as long as the runner is adequate in size.

General Disposition of the Boiler Surface

The general arrangement of the heating surface is shown in

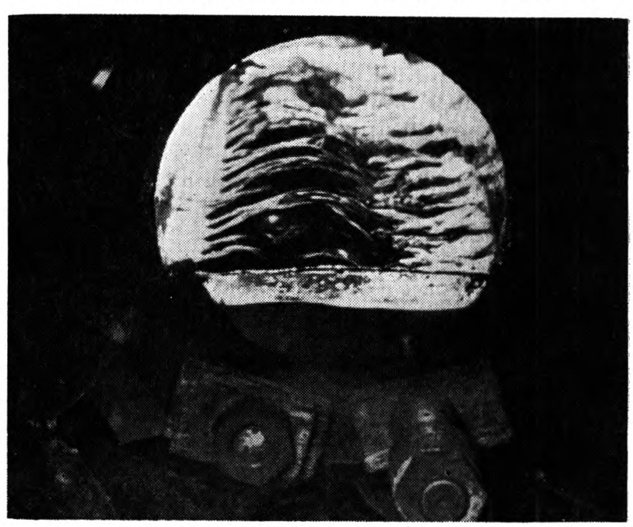

Fig. 7 Flow From Risers Into Upper Drum

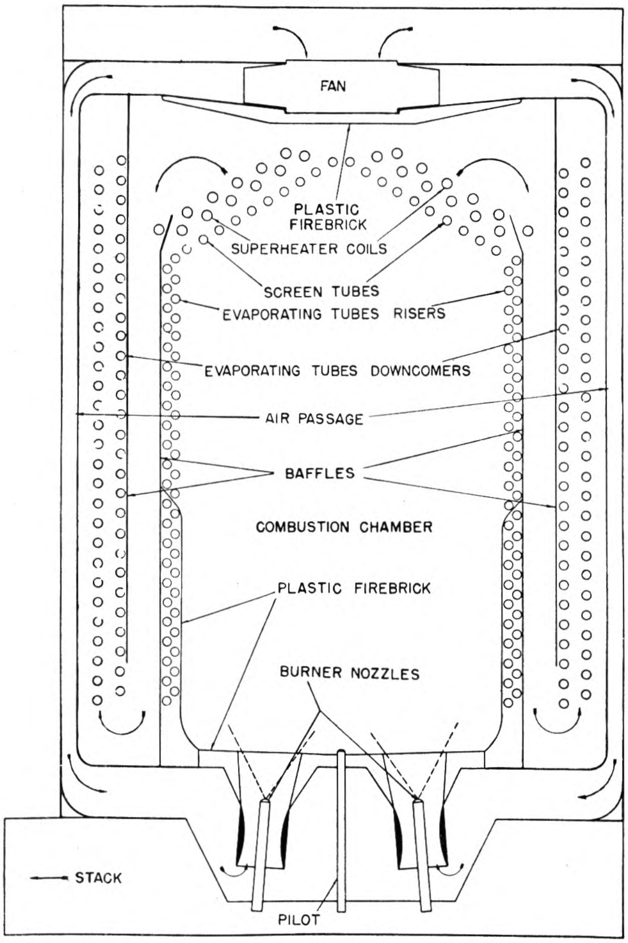

Fig. 8 Section Through Center of Boiler

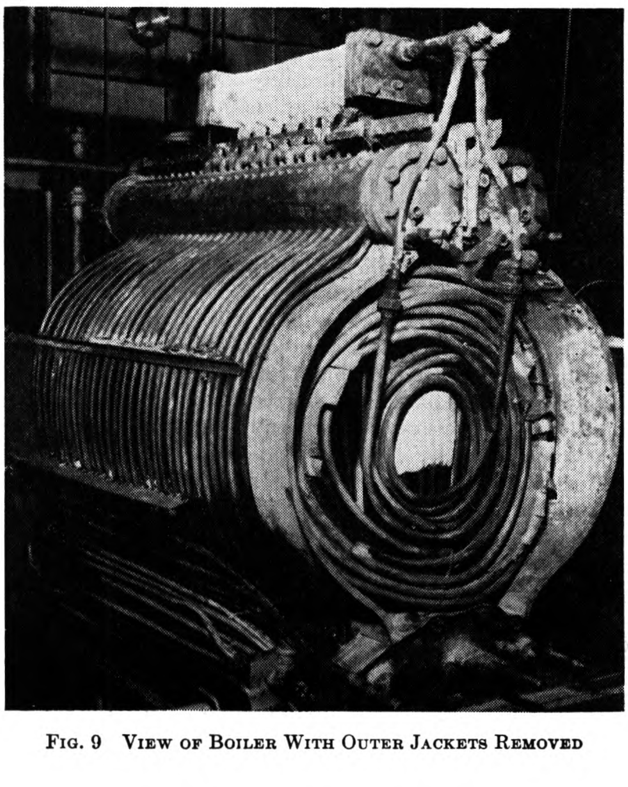

Fig. 9 View of Boiler With Outer Jackets Removed

Figs. 5, 8, and 9. The risers, through which the water is forced upward to the drum by the impeller, consist of four rows of stainless-steel tubes, each being $^1/_2$-in. outside diam × 0.43-in. inside diam, and approximately 3 ft long. The tubes are staggered, the distance between center lines being $1^3/_{16}$ in. The combustion space is thus entirely waterwalled, although it was found necessary to put refractory over the first few rows of tubes near the burner to provide radiant heat for vaporizing oil, Fig. 10, thus obtaining smokeless combustion and improved efficiency.

The downcomers are located in the third pass of the boiler. The heat transmission is much less intense in these tubes because they are not exposed to the radiation from the flame.

In order to supply dry steam to the superheater, a dry drum D is provided above A_1 and A_2 and is connected to them by a number of small tubes. This dry drum is only necessary because of the small diameter of the upper drums in this particular boiler. The boiler performed quite well for a long time without this drum but, under intensive operation, slugs of water would occasionally go through the superheater. In another boiler of this type having a single upper drum of a diameter suitably proportioned to the capacity of the boiler, no dry drum would be needed and a single upper drum would take the place of the three drums at the top of the present boiler. Perforated metal screens are also placed above the water line in A_1 and A_2 to prevent priming, or carry-over of large slugs of water.

The superheater is composed of a spiral coil of tubing and is located at the back of the combustion space, facing the burner, but separated from the combustion space by a screen of riser tubes. It was originally intended to install a second superheater around the burner, but this was found to be unnecessary. Control of the temperature of the superheated steam was accomplished by injecting feedwater into the inlet of the superheater. Steam temperatures above 800 F were obtained, although the temperature was usually kept around 750 F. The location of the superheater can be seen in Figs. 8 and 9.

Three economizers are used, with the result that the flue-gas temperature leaving the last economizer is always below 400 F. The first two economizers are located in the last pass, on each side of the unit, in space which is available because the general shape of the boiler proper is cylindrical. One of these economizers can be seen in Fig. 9 while Fig. 5 also shows them. The third economizer consists of coiled tubing, located in the duct leading to the stack.

Air Supply and Flue-Gas Disposal

Since the Knox boiler is intended to operate without the benefit of a stack to produce a natural draft, it is provided with a unique forced-draft system. The entire boiler is covered with an airtight metal casing, which is in turn partly enclosed within a removable aluminum outer jacket, Fig. 5. Between this outer shell and the jacket on either side of the boiler is a space through which the incoming air is drawn by the forced-draft fan. The fan consists of a cast-aluminum impeller, running on ball bearings, and driven by a V-belt and pulleys from the motor which also drives the water-circulating impeller. Major variations in fan capacity are accomplished by changing the driving pulleys and minor variations are made by restricting the air intake opening.

The air, which is drawn in through the fan after having passed over the outer shell of the boiler, is then forced through the space between the inner and outer shells, and is preheated to about 350 F before it enters the combustion chamber. The air pressure at the burner inlet is about 2 in. of water, under the usual operating conditions. As a result of this unusual arrangement of the air supply, the jacket around the boiler is at almost the same temperature as the surrounding air and, consequently, the losses due to radiation and convection are negligible.

The course of the flue gases can be seen in Fig. 8. After passing through the coiled superheater at the rear of the combustion chamber, the gases divide and turn back through the second pass, in which it was originally intended to locate more superheater tubes. Experience showed that the one coil was enough and, consequently, in future boilers this empty second pass would be omitted and the size and weight of the boiler thus reduced. This would also reduce the pressure required to force the gases of combustion through the boiler. Upon leaving the second pass, the gases go back through the third pass in which the downcomers are located. The gases then move down, as shown in Fig. 5, into the economizer space, and again come to the front of the boiler, where

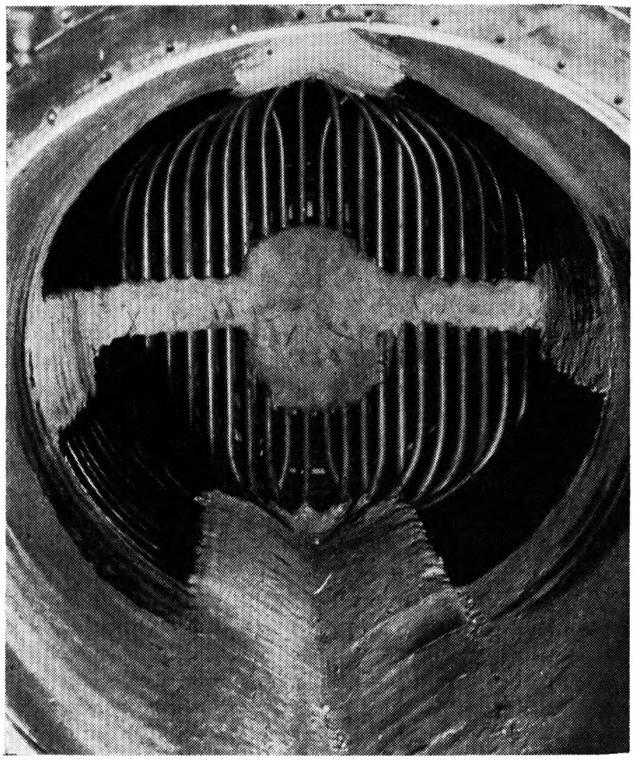

Fig. 10 Inside the Combustion Chamber

they enter the duct which leads to the last economizer and finally to the stack.

The usefulness of the last economizer is demonstrated by the test results, which show that the flue-gas temperature entering the duct was about 590 F at full load, while the gas temperature at the last economizer outlet was about 380 F.

Oil-Burning Equipment

The most serious problem encountered in the development of the boiler was the smokeless and efficient combustion of fuel oil at extremely high rates in a very small chamber. The combustion equipment which was in use when the tests were run consisted of four main burners, arranged around a small pilot burner, as shown in Fig. 8. The individual burners were located centrally in venturi-shaped openings through which the combustion air entered. The burners were standard 80-deg tips of the type used on domestic units, and the maximum capacity of the boiler could be varied by changing the size of the burners.

The boiler was started by electrical ignition of the pilot burner, which was located in the center of the refractory disk. After the main burners were ignited, the electrical ignition was shut off. The oil supply to the burners came directly from a small motor driven gear pump which kept the pressure at about 160 psi gage. The oil supply passed through a solenoid valve which was so connected into the control system that it could be shut off by the various safety devices which will be described.

The combustion chamber was partially lined with plastic refractory, as shown in Fig. 10, in order to give sufficient incandescent radiation surface to vaporize the oil. The amount of this refractory lining was determined by experience, and it was kept to a minimum so that the surface of the risers would be available for absorbing the radiant heat of the flame.

The furnace volume was about 5.67 cu ft, and this proved to be adequate during the official test for the release of 1,970,000 Btu per hr. The average heat release was thus about 350,000 Btu per cu ft per hr. Later the rate of smokeless combustion was increased to well over 400,000 Btu per cu ft per hr.

Controls and Safety Appliances

The problem of control in the boiler is greatly simplified because of the presence of the large upper and lower drums, which give a relatively large water capacity. Should the water level in the drums rise or fall as much as 2 in. above or below the usual line, no harm would be done. This is equivalent to more than 2 min output, about 15 times as long as it takes the controls to act, and many times as long as would be available for the much faster controls required for continuous-tube boilers. This feature of the boiler is particularly important, because it means that the fuel and water supplies do not have to be instantaneously adjusted.

Except for the safety valve, all controls are electrically operated. The safety valve is a double, spring-loaded poppet valve, which is connected to the two upper drums by pipes of ample size. The safety valve had no occasion to function during the tests.

The pressure control consisted of a spring-loaded bellows, equipped with contacts which were connected in series with the solenoid valve in the oil line. When the pressure rose above the set value, the contacts opened, the oil flow was shut off, the four main burners went out, and the pressure immediately fell. Since there is little refractory in the boiler to provide heat storage, there is consequently very little lag in the response of the pressure controller. The boiler can operate safely at pressures up to 700 psi, but during the test it was run at 500 psi gage.

The high- and low-water controls are of the thermal-expanding type, in which a brass tube alters its length in response to changes in the water level in the drum. Electrical contacts are opened when the water level falls too low, and the fuel supply is thus shut off. When the water level becomes too high, another set of contacts opens and these close a solenoid valve in the suction side of the feedwater line.

Manual control of steam temperature was in use during the tests. This control was accomplished by injecting water into the inlet of the superheater, the amount of injected water being regulated by a needle valve. The occasional wide swings of temperature which are to be seen in Fig. 11 are due to the fact that the water-level control affects the superheat control by altering the injection-water pressure. Subsequently, the superheat was controlled automatically.

It should be noted that a water-level glass is provided, the connections being taken from the right top drum A_2, in Fig. 5. This is an unusual feature for a forced-circulation boiler, since most of that type have such wide variations in water level that a glass would be impracticable.

During the test, the boiler was supplied with water by a multicylinder motor-driven reciprocating pump. The pump was kept running at constant speed, and the water delivery was regulated by the solenoid valve in the suction line. The water went from the pump through two separate circuits, each passing through two economizers, and then into one of the two upper drums.

Tabulation of Boiler Data and Report of Tests

The general over-all dimensions of the boiler proper are shown in Fig. 9. The height from the base of the boiler to the top of the cover over the dry drum is 38 in., the over-all length is 60 in., and the width is 36 in. The floor space occupied by the boiler is thus 15 sq ft. The secondary economizer is outside of the boiler and occupies about 3 sq ft if placed horizontally, or about 1 sq ft if placed vertically. If the space under the boiler now occupied by the fan motor is included, the over-all height is 57 in. A motor for driving the fan and runner could be placed within the present rectangular space 36 in. wide × 38 in. high, at the end of the boiler, or elsewhere if minimum height is required for the installation. If the unused second pass were eliminated, the boiler would be only 30 in. wide. Table 2 Gives general boiler data.

TABLE 2 GENERAL BOILER DATA

Heating surface		
Primary economizer, sq ft	26	
Secondary economizer, sq ft	32	
Total surface in economizers, sq ft		58
Downcomer surface, sq ft	43	
Riser and screen, sq ft	42	
Total evaporating surface, sq ft		85
Superheater surface, sq ft		11
Total heating surface in boiler, sq ft		154
Furnace volume		
Combustion chamber, cu ft	5.67	
Total volume within outer shell, cu ft	40.3	
Weight data		
Total weight of boiler (including refractory), lb	1235	
Weight of water in boiler, hot, lb	85	

Combustion data

Fuel burned—Essoheat medium, No. 2 domestic fuel oil; specific gravity 0.85 at 78 F; heating value 19,400 Btu per lb; 14.38 lb of air required per lb of oil
Air supply—forced draft, preheated to 360 F, 2-in. water pressure at burner intakes
Burners, number and type—four 80-deg tip, 3 gph capacity
Oil pressure (during test) 150 to 160 psi gage

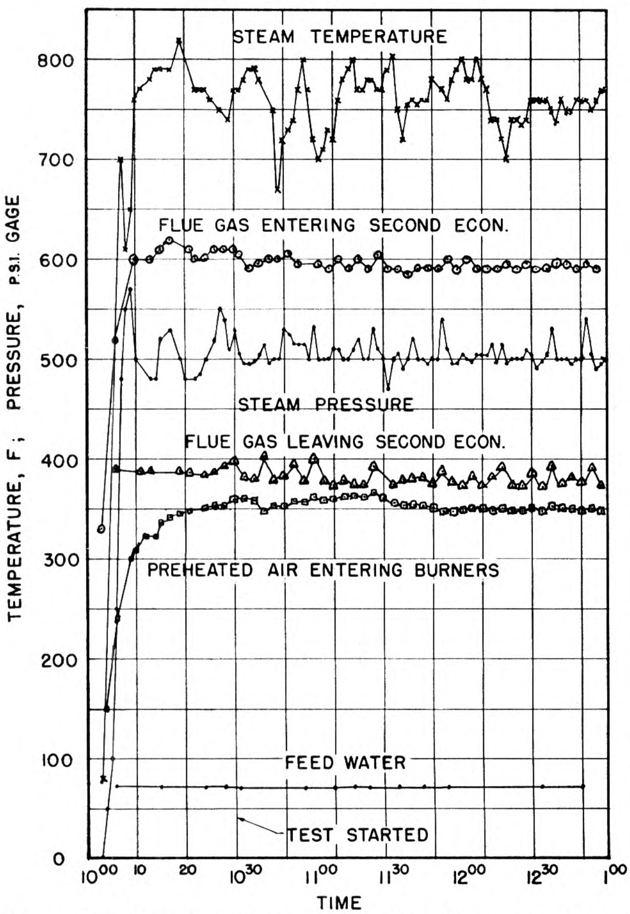

Fig. 11 Air, Flue-Gas, Steam, and Oil Data; Full-Load Test of Knox Boiler

Tests were run on the boiler on June 13, 1939, to determine the steam capacity, efficiency, and operating characteristics of the boiler at maximum smokeless output with four burners, each rated at 3 gph. In brief, the tests were conducted as nearly as possible in accordance with the A.S.M.E. Test Code for Oil-Fired Steam Generators, although the duration of the tests was less than that specified in the code. Table 3 shows the results.

TABLE 3 FULL-CAPACITY TEST

Test no.	1
Date of test	6/13/39
Duration, hr	2 1/2
Steam generated, lb	2966
Rate of generation, lb per hr	1186
Oil burned, lb	254
Evaporation, lb steam per lb oil	11.7
Pressure, average, psi abs	515
Temperature of steam, average, F	755
Enthalpy of steam, Btu per lb	1387
Temperature of feedwater, F	71
Heat added per lb, Btu	1348
Heating value of oil, Btu per lb	19400
Efficiency, per cent	81.3
Average CO_2 in flue gas, per cent	12
Air temperature, F	80
Flue-gas temperature, F	380
Loss to flue gas (approx), per cent	13.6
Loss to incomplete combustion, unburned carbon, etc. (by difference), per cent	5.1

The data for the tests are given in Fig. 11. It will be seen that the boiler was started from a cold condition at 10:03 a.m. The steam pressure began to rise almost at once and reached 500 psi within 4 min. The steam temperature rose with equal rapidity and reached 700 F within 4 min. The beginning of the test was deferred until 10:30 a.m. when equilibrium had been reached and all readings were steady.

The steam pressure was maintained at about 500 psi gage and the temperature at nearly 750 F. The variations in the steam temperature are due in part to difficulties with the water-level control, which altered the pressure of the water which was injected to control the steam temperature.

The air and gas temperatures varied but little during the test. The air entering the burners remained at 350 F, the flue gas entering the economizer at 590 F, and the flue gas leaving the second economizer remained at approximately 380 F. The feedwater temperature was constant at 71 F. The percentage of carbon dioxide in the flue gas ranged between 11 and 12.5 per cent.

The averaged results have been presented in Table 3. Some allocation of the losses can be made from the percentage of CO_2 and the flue-gas temperature. With an average CO_2 percentage of 12, the percentage of excess air was about 27 and the weight of dry flue gas produced per pound of oil burned was about 18 lb; 1.13 lb of water were produced by burning the hydrogen in the fuel. With an average flue-gas temperature of 380 F, losses were

	Per cent
Loss to water vapor	6.93
Unavoidable loss to dry gas	5.25
Avoidable loss to excess air	1.44
Total loss to flue gases	13.62
Percentage absorbed by water and steam (efficiency)	81.3
Unaccounted for (difference)	5.08

There was no smoke whatsoever during this test, and the boiler functioned smoothly. The test was terminated at 1:00 p.m. because it was felt that sufficient data had been obtained to give a reliable picture of the boiler performance.

The unit was quiet in operation, the two outer jackets serving to muffle the noise of the fan and the flame. There were no difficulties with the pressure control, nor any puffs when the burners went back into action after being shut off for a time. Ignition was positive and immediate.

Measurements for carbon monoxide in the flue gas were not

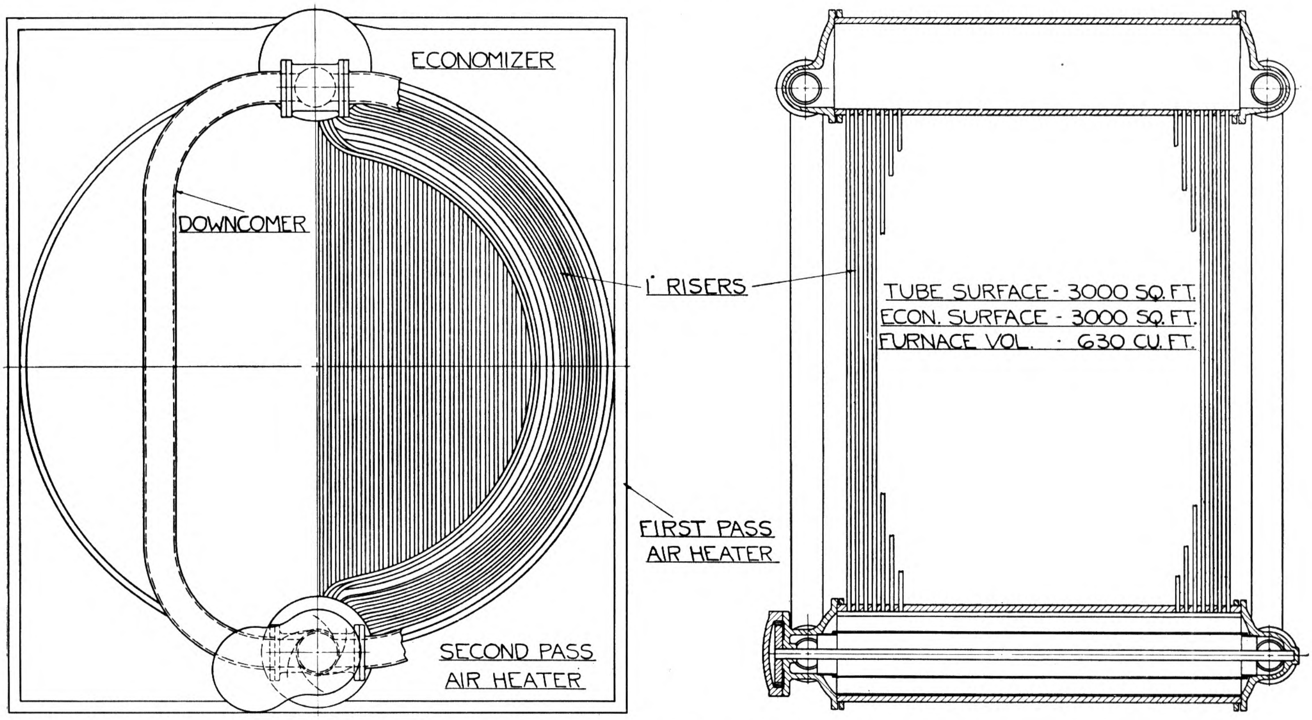

Fig. 12 Front and Side Views of Proposed Two-Drum Boiler

made during the test, since none had been found during the many earlier tests of the boiler at which the authors were present. In view of the large excess-air percentage, it is unlikely that any appreciable carbon monoxide existed.

There were no deposits of carbon in the combustion chamber, as Fig. 10 indicates. The refractory lining and the screen tubes were entirely free of carbon, and only a thin layer of soot adhered to the riser tubes. Furnace temperature was not measured but, judging from the color of the flame and from the heat release, it must have been very high.

No trouble was experienced with the packing on the impeller shaft, despite the fact that the pressure was between 500 and 600 psi, and the water must have been nearly saturated. Both fan and impeller were driven by the same 1.5-hp a-c motor.

After completion of the reported test, further development and tests of the boiler were conducted. During these tests, slightly over 1500 lb of water per hr were evaporated with smokeless combustion. While the boiler weighed 1235 lb, 300 lb of this were due to portions of the boiler installed for sound protection only, bringing the net weight of the boiler down to 935 lb. If correction is made for the amount of steam which would have been generated with the same fuel consumption, and at the same temperature, had the water been condensed and recirculated to the boiler at 200 F, it will be found that the weight of the boiler per pound of steam evaporated reduces to 0.56 lb.

The extremely high heat release, 430,000 Btu per hr per cu ft of combustion space, without harmful effects upon the tubes, refractory, or the general structure of the boiler, testifies to the effectiveness of the unique circulatory system used in the boiler.

The boiler as described was built with the special view of minimum over-all dimensions for a given output. Two water-level drums were used to obtain a maximum area of steam-release surface without the increase of height and weight which would have resulted had a single drum of equal steam-releasing surface been used. Where these limitations which controlled the design of this particular boiler are not essential, the boiler may be built with two drums instead of six, the downcomers consisting of large

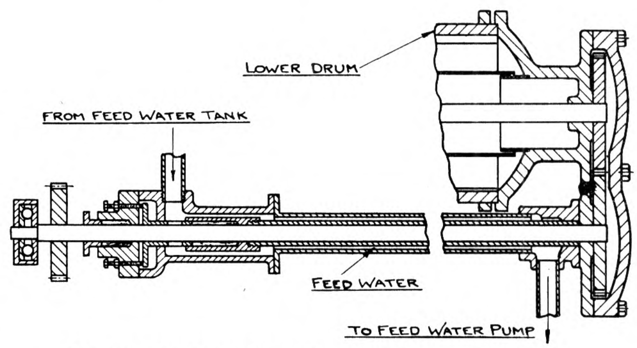

Fig. 13 Proposed Method of Driving the Impeller

tubes connecting the upper and lower drums at their ends, all the small tubes being risers. Such a construction is shown in Fig. 12.

Fig. 13 shows an improved method of driving the runner. The runner shaft carries a gear which connects with another below the driving shaft, running beneath the boiler and throughout its full length. This driving shaft is surrounded by a pipe carrying the feedwater. Thus, the packing is not subjected to high temperature, and could, in fact, be omitted, as the amount of water which would leak back along the driving shaft would be insignificant. Since it would go directly into the feedwater, there would be no loss of heat.

Conclusion

This paper has been devoted to a presentation in some detail of the unique features of an engine and a boiler, each of which is thought by the authors to embody certain advances over current practice. The test results are representative of the performance of these units.

Acknowledgment

The authors wish to express their appreciation of the assistance rendered by Kenneth Comey, of the Knox Engineering Company, and by Aaron Levine, of Stevens Institute of Technology.

Discussion

F. O. Ellenwood.[4] This paper is interesting and stimulating because it contains considerable information about a new engine and boiler, concerning which future progress reports will be welcome. The present performance data suggest interesting possibilities and it is sincerely hoped that future developments may place both the new engine and the new boiler on a successful production basis.

The test data show an engine efficiency of about 53 per cent, which is an excellent performance for an engine of this size. In this connection, will the authors indicate where the chief losses occur in this engine? In other words, what portion of the available energy is not utilized by reason of cylinder condensation, mechanical losses, thermal losses from the outside of the cylinder, and fluid-friction losses?

The new system of forced circulation in the boiler seems to be excellent and it is hoped that periods of long runs will still find it functioning properly. The improved method of driving the impeller, as indicated in Fig. 13 of the paper, seems to the writer to be a step in the right direction.

For a steam generator of this capacity to give an efficiency of 81 per cent, as has been shown by the test, means that great care has been taken in its design, and the writer desires to congratulate those who have been responsible for the development of this unit.

[4] Professor of Heat-Power Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

W. F. Ryan.[5] It is regrettable that the authors did not include in the published data the power consumption of the auxiliaries during the tests of this very interesting steam-generating unit.

In view of the relatively high draft loss and the power input to the circulating pump, it would ordinarily be assumed that the auxiliary-power consumption would be an excessive proportion of the limited output of the combined engine and boiler unit. If this is not the case it would naturally increase the value of the paper to indicate that fact by actual test results.

Authors' Closure

In reply to Professor Ellenwood's comments it is probable that a large portion of the losses in the engine are due to fluid friction. The unusual valve arrangement of this engine is possible only because of the somewhat involved paths through which the steam is led to the appropriate cylinder, and it is necessary to sacrifice some efficiency in order to attain the ends which were sought in the design.

In reply to Mr. Ryan, the power consumed by the a-c motor which drove both the fan and the impeller was about 0.75 hp. During a typical run, the current to the motor was 9 amp, at 220 v. The boiler feed pump which was used during the tests was driven by a large d-c motor for speed control, and the power which it required was far greater than that which would have been needed by a more efficient combination. In any event, the feed-pump power with this boiler should not be different from that with any other boiler of similar pressure and steam generation.

[5] Mechanical Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

A Study of Circulation in High-Pressure Boilers and Water-Cooled Furnaces

By JOHN VAN BRUNT,[1] NEW YORK, N. Y.

This paper is devoted to a discussion of the elements of boiler and furnace design dealing with the problem of obtaining adequate circulation for the higher pressures now demanded by current practice. After explaining the action within a simple evaporating circuit, the author indicates the procedure in analyzing waterwall circuits and calculating the circulation in boiler tubes.

Nomenclature

THE following nomenclature is used in the paper:

D_s = density of saturated water, lb per cu ft
D_d = density of downcomer mixture, lb per cu ft
D_r = density of riser mixture, lb per cu ft
d = tube diameter, ft
f = dimensionless friction factor
g = acceleration due to gravity, fps per sec
H = total heat absorbed by tube surfaces of furnace and boiler, Btu per hr
L = height from water level in drum to bottom of the circuit, ft
L_1 = distance from furnace floor, per cent of the furnace height
l = length of tube, ft
l_1, l_2 = see Q_1 and Q_2
Q = heat absorbed in distance L_1 as per cent of total heat absorbed by furnace
Q_1 = heat absorption by lowest portion l_1, of furnace-wall tube
Q_2 = heat absorption by greater portion l_2, of furnace-wall tube including l_1
S = summation of all losses due to flow velocity
V = linear velocity, fps
V_1 = linear velocity at entrance, fps
V_2 = linear velocity at exit, fps

Boilers in General

High-pressure steam-generating units of capacities and types installed and operated during the last few years are, with few exceptions, fired by pulverized coal in more or less completely water-cooled furnaces.

The problem of designers of such units is to provide a furnace of correct design to burn the specified fuel satisfactorily and, in addition to the furnace walls, such convection surface as is necessary to generate the required amount of steam. Superheating surface, economizer and air-heater surface must be provided and proportioned to give the desired over-all operating characteristics and efficiency.

It is not the purpose of this paper to go into the details of such proportioning of surfaces, but rather to bring out the elements of the circulating system and to show that the design discussed is both simple and adequate.

[1] Vice-President in Charge of Engineering, Combustion 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.

Natural circulation in a boiler may be defined as the movement of water, and steam and water, through the boiler tubes in conformity with the available head or difference in density of the circulating fluid in the downcomer and riser circuits.

Analysis of Simple Circuit

The simplest evaporating circuit is a U-tube Fig. 1(*a*), of uniform diameter and without restrictions in either the riser or downcomer leg. Such a circuit will, when heated on the riser leg, produce the maximum circulation, that is, it will pass through the circuit a maximum weight of water and steam for given conditions of tube diameter and head.

Any departure from this simple circuit which introduces resistance to flow, such as headers or junction boxes at *X*, or reduced area of tubes *Y* Fig. 1(*b*), will reduce the weight of water circulated. Heating the downcomer Fig. 1(*d*), or reducing the height, Fig. 1(*c*), will decrease the available head and, therefore, the flow.

Consider for example a simple U-tube circuit as in Fig. 1(*a*), having 30 ft of vertical heated length with the downcomer leg not heated, and passing water at the saturation temperature. Assume that the mean density of the mixture in this riser is 0.8 that of saturated water in the downcomer; then 24 ft of water in the downcomer will balance 30 ft of the mixture. Since there is 30 ft of solid water in the downcomer the static head is 6 ft. This 6 ft of head is available to overcome all friction and other resistances in the circuit. In the circuit considered, there are five losses; namely, entrance resistance, friction in the downcomer and in the riser, loss due to accelerating the mixture in the riser, and the exit loss from the riser. If these losses, expressed in feet head of saturated water, total 6 ft, the circuit is in equilibrium with 0.8 density in the riser circuit. If at an assumed velocity the resistances total but 4 ft, more water will flow through the circuit, thereby increasing the density in the riser and reducing the static head, and increasing the velocity and magnitude of the losses until equilibrium is established. Should the resistances at an assumed velocity total 8 ft, less water will flow through the circuit, thereby decreasing the density in the risers, increasing the static head, and reducing the velocity and resistances until equilibrium is reached.

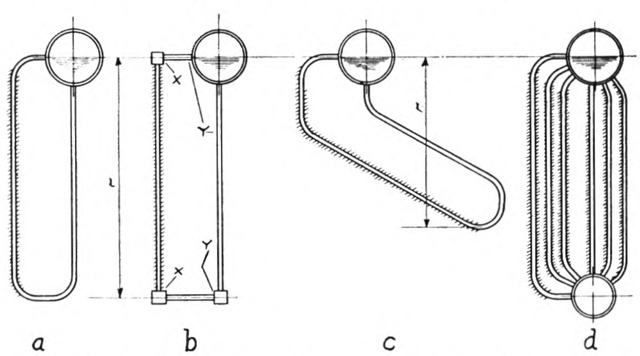

Fig. 1 Evaporating Circuits of Various Forms

If, in the same circuit, we assume that the downcomer is heated and that the steam is generated in the downcomer in an amount sufficient to reduce the mean density of the downcomer mixture to 0.9 that of saturated water, this mixture will enter the riser in which additional steam is generated to give a mean density of 0.7 of saturated water. The equivalent static head is $(0.9 \times 30) - (0.7 \times 30) = 6$ ft, which as already noted is available to overcome resistances.

From the foregoing rough illustration, it will be seen that circulation will be stabilized at whatever density and velocity exists when all losses equal the static head; i.e., head in downcomer equals head in riser plus all losses expressed in head.

The term "dryness fraction" is the percentage of steam by weight in the mixture. "Top dryness" is the percentage by weight of steam at the top of the tube or at the point where heat input ceases.

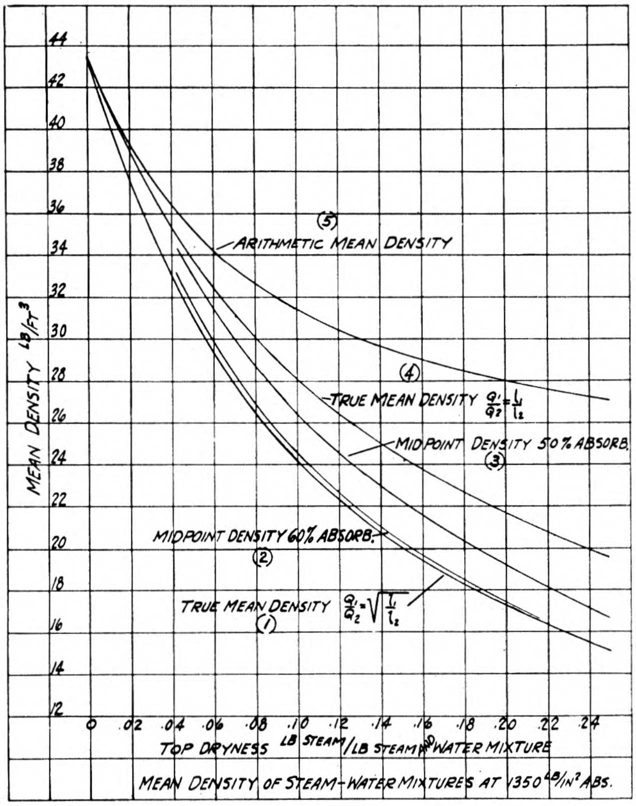

FIG. 2 MEAN DENSITY OF STEAM-WATER MIXTURES AT 1350 PSI ABS

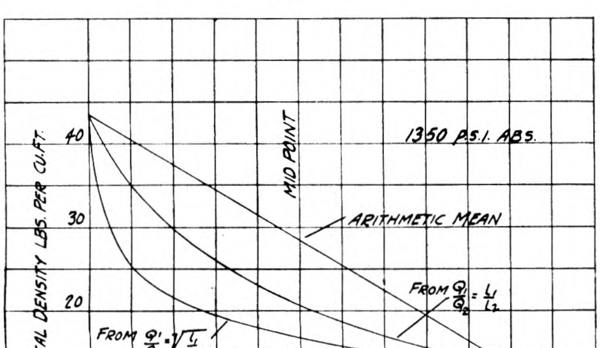

FIG. 3 CHANGES IN DENSITY ALONG TUBE

Inasmuch as the static head available to overcome all resistances depends upon the density of fluid in the risers and downcomers, any analysis of circulation must begin with the determination of the density. This must be based on the correct rate of heat absorption by the exposed surface in the circuit being analyzed.

Density Versus Heat Absorption

Fig. 2, curve 1, gives the true mean density in a heated riser for heat absorption based on $\frac{Q_1}{Q_2} = \sqrt{\frac{l_1}{l_2}}$, or expressed in the percentage absorbed in any portion of the tube, Q (per cent) $= 10\sqrt{L_1}$.[2] Curve 2 shows the mid-point density based on 60 per cent of the heat absorbed in the bottom half of the tube. Curve 3 shows the mid-point density for 50 per cent heat absorbed up to mid-point. Curve 4 is the true mean density for uniform absorption, and curve 5 is the arithmetic-mean density. All of these curves are based on saturated water entering the bottom of the tube.

Fig. 3 illustrates the changes in density along the tube, the lowest curve is based on $\frac{Q_1}{Q_2} = \sqrt{\frac{l_1}{l_2}}$. The next curve corresponds to a heat input of $\frac{Q_1}{Q_2} = \frac{l_1}{l_2}$ and the top curve is the arithmetic mean of entrance and exit densities. The curves in Fig. 3 are plotted for saturated water at the bottom and a top-dryness fraction of 0.18, corresponding to a top density of 13.3 lb per cu ft, and a relative top density of 0.305 that of water at saturation temperature.

To visualize further the effect which these methods of calculating density have on the available static head, the available head for 100 ft of height of mixture when balanced against 100 ft of saturated water is shown in Fig. 4.

In analyzing the circulation in a furnace-wall circuit, one must start with the total heat absorbed at maximum load and, as may be seen from the curves in Figs. 2, 3, and 4, select the absorption

[2] "Factors Affecting Metal Temperatures of Furnace and Boiler Tubes," by W. S. Patterson, *Combustion*, August, 1940, pp. 24–29.

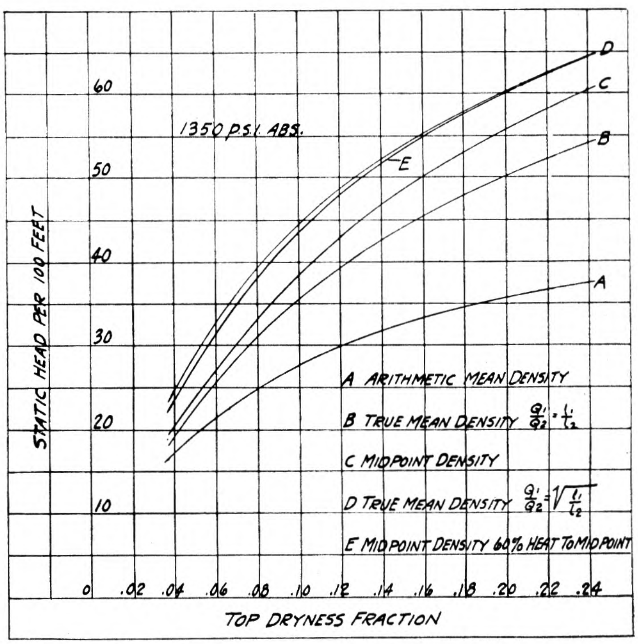

FIG. 4 EFFECT OF DENSITY-CALCULATING METHODS ON AVAILABLE STATIC HEAD

rates along the tube which will give the correct density. Density calculated for conditions of curve 1, Fig. 2, will give the maximum available head. However, it is quite probable that ash or slag accumulations on the lower part of furnace walls will at times lower the absorption rate in the lower part of the furnace. Design must provide for these unfavorable operating conditions, such as producing minimum available head. These conditions may result in uniform absorption for each increment of tube length and, under some unusual furnace condition, perhaps localized on a few tubes, it is possible that the absorption in the upper half of the tube may exceed that in the lower half. Such a condition will result in a higher mean density than that shown by curve 4, Fig. 2.

Calculations for Furnace Tubes of Typical Unit

For purposes of analysis of waterwall circuits, a unit of 650,000 lb per hr capacity at 1350 psi pressure and 925 F total steam temperature is selected. Fig. 5 is a diagram of such a unit.

From Fig. 5, it will be noted that the rear waterwall risers are connected through the lower drum to the fourth and fifth rows of tubes. An alternate connection to the top drum is indicated by the dotted line joining the rear waterwall with the front row of boiler tubes. Either arrangement is satisfactory, the selection depending upon the spacing of front boiler tubes, which in turn is fixed by the number of superheater elements.

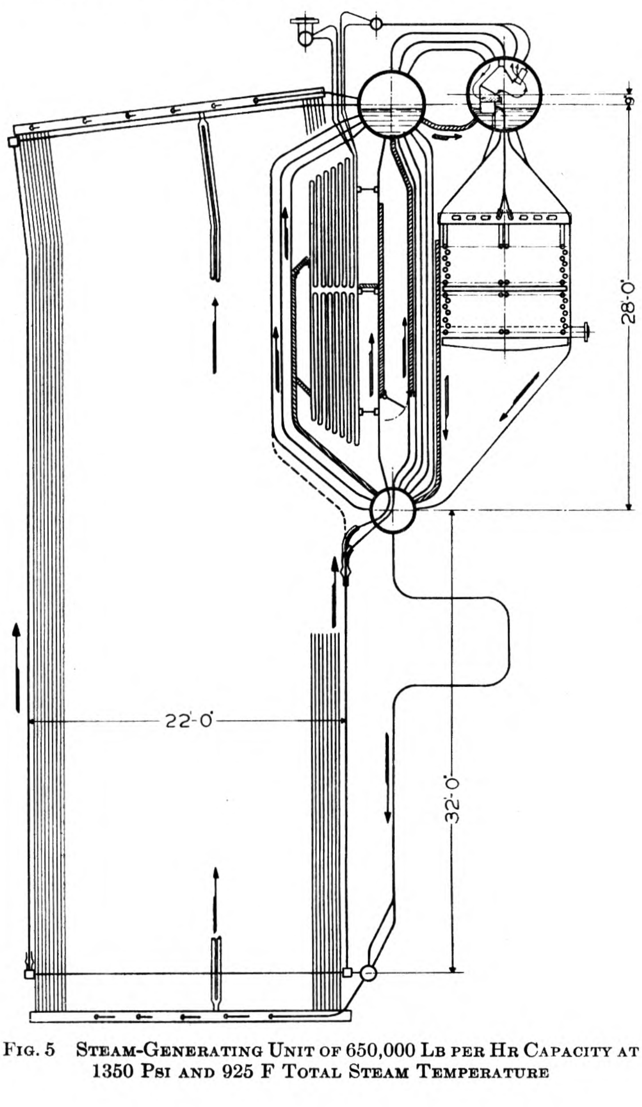

Fig. 5 Steam-Generating Unit of 650,000 Lb per Hr Capacity at 1350 Psi and 925 F Total Steam Temperature

Steam generated in the boiler downcomers returns to the upper drum through some of the tubes in the fifth row.

The feedwater temperature leaving the economizer is assumed to be 470 F. This feedwater will be raised to saturation temperature in a steam washer; to do this, 170,000 lb of steam per hr will be condensed in the washer. All of the water entering into circulation beyond the drum containing the washer will be at saturation temperature and, therefore, all heat absorbed by evaporating tube surface will produce steam. $H \div 584 =$ lb of steam (where H is the total heat absorbed by the unit and 584 is the heat of evaporation), and the total steam to be generated will be 650,000 + 170,000 = 820,000 lb per hr.

The furnace is 30 ft wide and the boiler 35 tubes wide. There are three rows of tubes ahead of the superheater and eight rows behind it, of which the fourth and fifth rows are risers, the sixth, seventh, eighth, and ninth downcomers, and the tenth and eleventh downcomers from the rear or offtake drum. Also, there are 35 horizontal water circulators between the two upper drums.

The furnace is so proportioned that 730,000 lb of steam per hr will be generated in the walls, 50,000 lb in the front three rows of boiler tubes in addition to that absorbed in those tubes by direct radiation from the furnace and, in the remaining eight rows 40,000 lb, of which approximately 25,000 lb will be evaporated in the four rows of downcomers, Nos. 6, 7, 8, and 9 and 12,000 lb in risers Nos. 4 and 5.

The wall-tube units are two 3-in. bifurcated tubes, each pair of 3-in. tubes being forged into a short $3^1/_4$-in. end at top and bottom. The bottom of the furnace is made up of 70 fin tubes 3 in. diam, the front and back walls of 110 tubes 3 in. diam or 55 pairs of 3-in. bifurcated units, and the side walls of 80 tubes 3 in. diam or 40 bifurcated units. The four furnace walls are supplied by 72 downcomers $3^1/_2$ in. diam between the lower drum and the round distributing header.

From the distributing header 70 nipples 3 in. diam lead to the rear waterwall header and 20 tubes $3^1/_2$ in. diam to each of the lower side-wall headers. The upper front-wall header is connected to the front drum by 70 fin tubes 3 in. diam and the upper side-wall headers by 30 tubes 3 in. diam.

Using an average absorption rate of 70,000 Btu per sq ft projected area of furnace wall per hr, the evaporation in each 60 ft of 3-in. front and side-wall tube will be $\dfrac{0.25 \times 60 \times 70{,}000}{584} = 1800$ lb of steam per hr. Using this figure, the steam generated in the furnace walls will be as follows:

Front wall, 110 × 1800 = 198,000 lb; rear wall 110 × 1800 = 198,000 lb; side walls 80 × 2 × 1800 = 288,000 lb. The roof of the furnace with an estimated absorption rate of 40,000 Btu per sq ft per hr will evaporate 45,500 lb per hr. This makes a total evaporation of 729,500 lb per hr. The remaining 90,000 lb of steam will be generated by convection in the eleven rows of boiler tubes as previously described.

An analysis will be made of the circulation in the front waterwall, side wall, and front row of boiler tubes only. These will serve as examples of the method which can be applied to any circuit in the boiler.

Briefly stated, the problem is to determine the equilibrium point at which the available static head is equal to the sum of all losses in head due to velocity

$$\left(\frac{D_d}{D_s} - \frac{D_r}{D_s}\right) \times L = S$$

in which

D_s = density of water at saturated temperature, lb per cu ft
D_d = density of mixture in downcomer, lb per cu ft

D_r = density of mixture in riser, lb per cu ft
L = length (height) from water level in drum to bottom of circuit, ft
S = sum of all resistances

To determine the total amount of water entering the bottom of all evaporating tubes divide the total weight of steam generated by the assumed top-dryness fraction. From the weight of water thus found subtract the weight of steam to find the amount of water discharged into the front drum. Similarly, to find the amount of water entering the furnace-wall tubes, divide the weight of steam generated in the walls by the assumed top-dryness fraction.

From the weight of water discharged by the front drum, subtract the amount flowing through the horizontal water circulators to the rear drum to obtain the weight flowing through the boiler downcomers.

From these flows, the velocities of water in the boiler and waterwall downcomers is obtained for any top dryness. The water to the furnace walls is assumed to be distributed to the four waterwalls in proportion to the calculated requirements.

These flows and velocities for front and side-wall circuits are given in Table 1, as well as other necessary velocities, also mean densities, top densities, and available head. Mean densities are taken from curve 4, Fig. 2.

TABLE 1 FLOWS AND VELOCITIES, ETC., FOR FRONT AND SIDE-WALL CIRCUITS

Top-dryness fraction (assumed)	0.135	0.15	0.165	0.18
Steam generated, lb per hr	820000	820000	820000	820000
Water to evaporating surface, lb per hr	6080000	5460000	4970000	4550000
Steam to rear drum, lb per hr	820000	820000	820000	820000
Water discharged to front drum, lb per hr	5260000	4640000	4150000	3730000
Water flow to rear drum, lb per hr	1330000	1330000	1330000	1330000
Water to boiler downcomers, lb per hr	3930000	3310000	2820000	2400000
Water to waterwall downcomers, lb per hr	5400000	4875000	4450000	4050000
Velocity in boiler downcomers, fps	5.95	5.05	4.35	3.65
Velocity in waterwall downcomers, fps	11.6	10.5	9.55	8.75
Velocity in 18-in. connecting nipples, fps	8.9	8	7.3	6.7
Velocity in floor tubes, fps	4.45	4	3.65	3.35
Velocity in side-wall downcomers, fps	8.1	7.35	6.65	6.1
Velocity in $3^1/_4$-in. bifurcated ends, fps	4.72	4.25	3.83	3.54
Velocity in 3-in. tubes at bottom, fps	2.83	2.55	2.32	2.12
Mean density in 3-in. wall tubes, lb per cu ft	25.25	24.37	23.5	22.7
Velocity at point of mean density, fps	4.87	4.55	4.28	4.15
Density at top of 3-in. wall tubes, lb per cu ft	17.03	15.05	14.1	13.3
Velocity at top of 3-in. wall tubes, fps	7.7	7.4	7.15	6.9
Velocity entering roof tubes, fps	12.1	11.7	11.5	10.85
Density leaving roof tubes, lb per cu ft	14.3	13.3	12.45	11.8
Velocity leaving roof tubes, fps	13.5	13.1	12.7	12.4
Available head, assuming saturated water in downcomer, ft	25.2	26.3	27.5	28.7
Velocity in side-wall risers, fps	15.4	14.8	14.3	13.8

From these velocities and densities, the various entrance, exit, friction, and acceleration losses are determined. Considering a circuit from the front drum through downcomers and up through the front wall, there are twenty-one distinct losses, including entrance, exit, and friction in boiler and waterwall downcomers, in 18-in. nipples, in floor tubes, in furnace-wall risers, and in roof tubes; in addition there are acceleration losses in boiler downcomers, furnace-wall risers, and in roof tubes. Entrance resistance cannot exceed $\frac{1}{2}\frac{V^2}{2g} \times$ relative density. Exit resistance cannot exceed $\frac{V^2}{2g}$ times relative density. For friction loss the Fanning equation is used. Friction loss $\frac{4f\,l\,V^2}{d \times 2g} \times$ relative density. Relative density is evaluated at the point where velocity is calculated.

V = velocity at point of mean density, fps
g = 32.2 fpsps
f = 0.006
l = length of tube, ft
d = diameter, ft

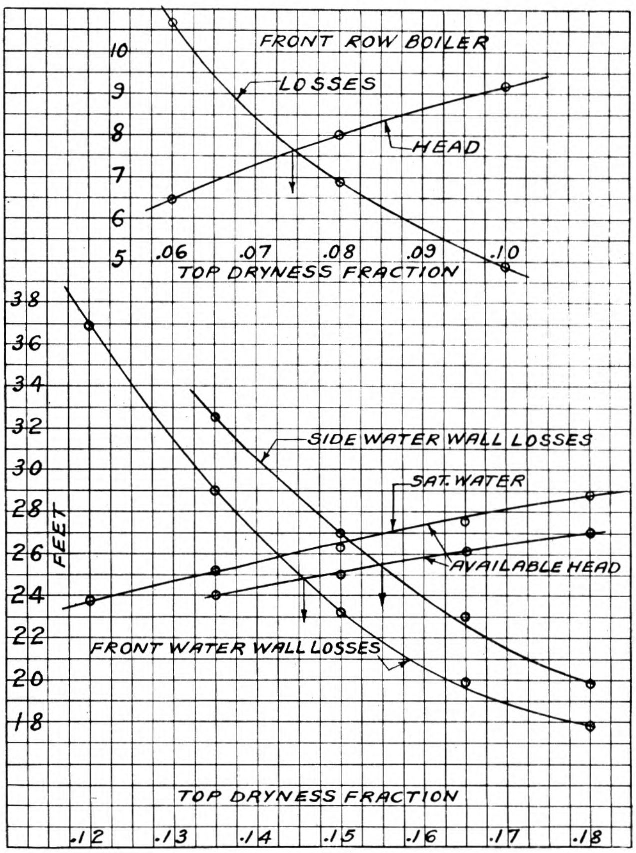

FIG. 6. LOSS IN HEAD FOR VARIOUS TOP-DRYNESS FRACTIONS

TABLE 2 RESISTANCES DUE TO VELOCITY

	Entrance	Exit	Friction	Acceleration	Total front	Total side
Boiler downcomers, ft	0.147	0.294	0.97	..	1.41	....
Waterwall downcomers, ft	0.707	1.415	7.85	..	9.92	11.33
18-In. nipples, ft	0.415	0.830	0.167	..	1.41	....
Floor tubes, ft	0.104	0.208	0.228	..	0.544	....
60-In. wall tubes, ft	0.228	0.730	1.15	0.35	2.458	....
Roof tubes, ft	0.665	0.72	2.04	0.44	3.86	....
Side-wall downcomers, ft	0.345	0.69	2.7	..	....	3.735
60-Ft side-wall tubes, ft	0.228	0.730	1.15	0.35	....	2.458
Side-wall risers, ft	0.955	0.955	3.6	..	....	5.5
Total front-wall circuit, ft					19.602	
Total side-wall circuit, ft						23.023

The acceleration loss is $\frac{V_1}{32.2}(V_2 - V_1)$, where V_1 = velocity at entrance and V_2 = velocity at exit.

There may be some differences of opinion as to the correct or most logical method of figuring friction losses in heated tubes. In this paper, the friction factor of 0.006 is used, although 0.005 is generally accepted. It can be seen, however, that even if the friction losses in the riser tubes are doubled, the top-dryness fraction of both side and front walls will still be less than 0.17.

As an example, an analysis of flow through the front and side-wall circuits with a top-dryness fraction of 0.165 will be taken. For this condition, the velocities, head, densities, etc., will be as given in column 3, Table 1.

Because of the change in area, where the $3^1/_4$-in. bifurcated end joins the wall tubes and the change in direction and turbulence, the entrance resistance in the bottom and top wall header will be figured as $\frac{V^2}{2g}$ instead of $\frac{1}{2}\frac{V^2}{2g}$

From the flow and velocities in column 3, Table 1, the resistances in the various elements of the circuits in feet of water at saturation temperature are calculated and listed in Table 2.

Similar calculations for top-dryness fractions of 0.135, 0.15, and 0.18 may then be plotted, as in Fig. 6. At the intersection of the available head and resistance curves, the circuit is in equilibrium and the corresponding top-dryness fractions that exist may be read from the curve.

Calculations for Boiler Tubes

The same method is applied to the circulation in the front bank of boiler tubes. Taking one of the front rows of boiler tubes, the evaporation is first determined by adding to the heat input by radiation, the convection transfer, and dividing by the latent heat. The evaporation per tube is approximately 2400 lb per hr. Again, as with the furnace tubes, this weight is divided by the assumed top-dryness fraction to find the weight of water entering the bottom of the tube.

As 25,000 lb of steam is generated in the boiler downcomers, the mean density in the downcomers is determined from the amount of steam generated and the flow through downcomers corresponding to the assumed top-dryness fraction. Velocities through the entire circuit and other necessary data are given in Table 3.

At 0.18 top-dryness fraction 2,400,000 lb of water flow through the boiler downcomers. The mean density in the downcomers is approximately the same as the mid-point density, thus 2,400,000 —12,500 = 2,387,500 lb of water per hr. The volume of steam and water = 58,875 cu ft; density = 2,400,000 ÷ 58,875 = 41 lb per cu ft. Relative density = 41 + 43.5 = 0.94. For other dryness fraction the densities are 41.2, 41.5, and 41.8, and the relative densities 0.945, 0.955, and 0.962.

The tubes in the front row of the boiler are 3 in. diam and approximately 34 ft long. There are only four resistances; namely, entrance, exit, friction, and acceleration. From inspection of the losses calculated for the waterwall circuits, it is obvious that with top-dryness fractions of 0.15 or more the available head will be greatly in excess of that required. To plot the equilibrium curves for the circuit, assumptions of 0.10, 0.08, and 0.06 top-dryness fractions will be used. No attempt will be made to correct the flow of water through the boiler downcomers to correspond to these assumed dryness fractions as the change in relative density would be small. The essential data are given in Table 3.

The available heads and losses are plotted at the top of Fig 6 and the head and loss curves drawn through the points. The intersection will be found at 0.075 top-dryness fraction.

The upper available head curve Fig. 6, is based on saturated water in the downcomers. Correcting this curve for downcomer densities of 0.94, 0.945, 0.955, and 0.962, respectively, at 0.135, 0.15, 0.165, and 0.18 top dryness, the head will be as in the lower curve. The top-dryness fractions at the equilibrium points are 0.145 for the front wall and 0.155 for the side wall.

TABLE 3 FLOW DATA, FRONT ROW OF BOILER TUBES

Top-dryness fraction (assumed)	0.06	0.08	0.10
Lb of steam per hr per tube	2400	2400	2400
Amount of water entering bottom of tube, lb	40000	30000	24000
Volume of water, cfs	0.256	0.192	0.1535
Velocity entering, fps	8.45	6.4	5.12
Mean density from curve 4, Fig. 2, lb per cu ft	32.4	30.0	28.0
Velocity at mean density point, fps	11.35	9.3	7.96
Velocity at exit, fps	15.0	12.9	11.7
Density at exit, lb per cu ft	24.5	21.8	19.0
Head in downcomer, saturated water, ft	27.3	27.2	27.1
Head of saturated water in riser, ft	20.85	19.2	17.9
Available head, saturated water, ft	6.45	8.0	9.2
Entrance loss, saturated water, ft	0.55	0.32	0.203
Exit loss, saturated water, ft	2.18	1.30	0.930
Friction loss, saturated water, ft	6 20	3.95	2.650
Acceleration loss, saturated water, ft	1.71	1.30	1.040
Total losses, ft	10.64	6.87	4.823

It is possible to equalize the flow velocity in side and front walls either by increasing the resistance in the front-wall circuit, or decreasing the resistance in the side-wall circuit.

Conclusions

It appears from the foregoing analyses that the circulation in high-pressure units of the type illustrated is adequate for the maximum load conditions and with a comfortable margin for substantial overloads.

Similar analyses of furnace-wall tubes $2^1/_2$ in. outside diam show equally adequate circulation.

Should it be necessary to reduce the losses below the figures submitted herein, downcomer areas should be increased.

If the height of boiler furnace is decreased, the available head and friction losses in the downcomers and risers also decrease. All inlet and outlet losses and friction losses in the horizontal members are unchanged; therefore, there is a lower limit of height below which the circulation would become progressively less until the top-dryness fraction approaches unity, under which conditions the evaporating tubes would fail from overheating. It is obvious, therefore, that for high-capacity units of low height, natural circulation may become inadequate.

For those who are interested, a more complete analysis of circulation flow, reference is made to a recent paper,[3] by W. Yorath Lewis and Struan A. Robertson.

Discussion

E. G. Bailey.[4] The author has shown the application of the well-known fundamental laws of boiler circulation to a large high-pressure steam-generating unit of recent design. This undoubtedly will give many engineers a clearer picture of the problems involved in calculating circulation than they have had from the simpler illustrations available in the engineering literature.

While the author has made clear his assumptions and his general procedure in carrying out calculations, the writer believes it should be brought out even more clearly that assumptions should be correct, if the final calculations are to be in keeping with the actual results from the unit in operation.

It should be further emphasized that experimental results from actual tests should be used to confirm or modify some of the more important assumed factors. The author amplifies the different methods of calculating mean density of steam-

[3] "Circulation of Water and Steam in Water Tube Boilers and the Rational Simplification of Boiler Design," by W. Yorath Lewis and Struan A. Robertson, Proceedings of The Institution of Mechanical Engineers, London, England, vol. 143, June, 1940, pp. 147–175.

[4] Vice-President, The Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

water mixtures, which is the fountainhead of boiler circulation. From actual field data, it is believed that the effective circulating head is likely to be less than any of the assumptions he has made, and certainly less than the "true mean density" used in the final calculations. This difference is believed to be partly due to the relative velocity of the steam and the water in the mixture. The mean density of any given circuit depends a great deal upon the relative heat absorption in different parts of that circuit. That of itself is a problem which requires considerably more data from field tests, because there are a great many diverse conditions in different types of units, different types of firing, and different ash and slag conditions.

The author has introduced an expression "dryness fraction" which will be new to many engineers. It is in reality "per cent of steam by weight" or the reciprocal of one often used, which is the "mixture-to-steam ratio." Another term "pounds of water per pound of steam" is often used. Still another term "per cent of steam by volume" has much greater significance. It might be well for some consideration to be given to standardizing terms, so that we will all talk the same language.

Units of the type described in the text, having both boiler and waterwalls supplied from a common source, must be designed so that the proper circulation will obtain in each circuit when all circuits are in equilibrium. This cannot be assured without solving all circuits simultaneously.

In his conclusions, the author states that losses are reduced by increasing downcomer areas. The fact should also be mentioned that, in some types of circuits, this can be brought about to better advantage by increasing the riser areas.

The fourth paragraph of the conclusions seems somewhat confusing because the friction losses in the horizontal members are actually reduced when the total circulating flow is reduced by the lower height of the heat-absorbing circuits, although the resistance of the horizontal members may be unchanged.

Fred Dornbrook.[5] In his conclusions the author indicates that circulation is adequate "until the top-dryness fraction approaches unity." He probably does not wish to convey the impression that high-pressure high-capacity designs can employ, for instance, 99 per cent steam and 1 per cent water in the risers under peak conditions.

It is submitted that, for the high reliability requirements of modern high-pressure high-capacity boiler units, circulation requires more involved study than simply a calculation of average circulation velocities and dryness factors. Perhaps the author will tell more of his experience in regard to limiting velocities, dryness fractions, and heat-transfer rates.

M. H. Kuhner.[6] The author's treatment of the theory of available force to produce natural circulation for higher-pressure steam-generating units is a valuable addition to boiler-design information. Such analyses are necessary to prevent the operating difficulties of higher-pressure installations, resulting from insufficient circulation, as reported for some installations placed in service during the last few years.[7]

The author may wish to amplify his conclusions by mentioning that entrance and discharge losses of tubes connected to boiler drums and waterwall headers are difficult to determine. These losses depend largely upon the density of the fluid and direction of flow in reference to entrance and discharge openings and the possible turbulence existing in drums and headers. From the standpoint of practical design, it may be suggested that waterwall headers be made large in internal areas and that the water supply to bottom headers and the offtakes from the upper headers be liberal in area and uniformly distributed over the length of these headers. This for the purpose of obtaining minimum disturbance of flow in the vicinity of entrance and discharge of the steaming tubes.

In applying the author's theories to practice, it should further be pointed out that the distribution of heat to individual groups of steaming tubes, such as the group of side waterwall tubes or rear waterwall tubes, is not uniform for each single tube. Those tubes placed closest to the source of heat, such as the tubes in the middle of the side waterwalls or those directly opposite the burners in the rear wall of the furnace, may contain a steam-water mixture of considerably lower mean density than other tubes of the same group placed in the corners of the furnace, or which may be otherwise shielded from direct radiation. The mean top-dryness fraction of the individual group of tubes may show to be safe while at the same time the top-dryness fraction in a few of the tubes may be unity, so that these tubes are overheating. The same problem applies to the steaming tubes of the boiler. It is known that those tubes placed over the center of the furnace are subjected to higher radiation and gas temperatures than the tubes near the side walls.

It would be interesting to apply the author's theoretical investigation of circulation to one or more of the installations discussed in E. P. Partridge and R. F. Hall's paper.[7]

The further conclusion, drawn from the author's paper and applied to practice, shows the importance of proportioning the flow areas in circulatory systems of waterwalls and boilers of higher-pressure installations as liberally as practical and with a minimum of obstructions or changes in cross-sectional areas, so that the losses of flow in tubes, headers, and drums be kept at a minimum.

Another important factor influencing the available force producing natural circulation is the density of the fluid in the downcomer tubes. The buoyancy of steam formed in downcomers tends to oppose downflow of water. Therefore, downtake tubes must not be exposed to high gas temperatures.

It would appear, that, if a given boiler unit should be operated at various pressures, a particular dryness fraction would be realized for each particular pressure. A definition by the author of the force governing this relation would be interesting.

E. F. Leib.[8] If the heat absorption in parallel circuits is not equal for each circuit, the possibility exists that the flow rate through certain tubes varies widely from the flow through the others; under certain conditions, the flow can reverse and a riser may operate as a downcomer. This is due to the circumstance that various flow rates may result in the same pressure difference. This condition is referred to as instability. In the following, two methods are outlined to examine whether, for a given case, the same pressure difference which is maintained by the flow rate through the bulk of the tubes may also correspond to other flow rates.[9] Uniform heat input over the entire tube length is assumed.

Forced-Flow Tubes

This system has no recirculation and no economizer. The tubes are very long. Therefore, pressure losses other than due to friction can be neglected. The direction of flow is assured, but the rate may be undetermined. The feedwater enters

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[6] Chief Mechanical Engineer, Riley Stoker Corporation, Worcester, Mass. Mem. A.S.M.E.

[7] "Attack on Steel in High-Capacity Boilers as a Result of Overheating Due to Steam Blanketing," by E. P. Partridge and R. F. Hall, Trans. A.S.M.E., vol. 61, 1939, pp. 597–621.

[8] Combustion Engineering Company, Inc., New York, N. Y.

[9] "Unstabilität der Strömung bei natürlichem und Zwangsumlauf," by M. Ledinegg, *Die Wärme*, 1938, pp. 891–898.

cold, is heated to saturation temperature in the lower section of the tube, and is completely evaporated in the upper section. The relative length of both sections depends upon the specific flow, i.e., the flow rate divided by the heat input. Since the friction loss is less in the section containing the liquid than in the section containing the steam-water mixture, it is possible that the pressure loss decreases while the saturation point moves upward due to an increasing flow rate. The relation between flow and friction is obtained as follows:

The force of friction is assumed proportional to the kinetic energy of the flowing fluid and to the area of friction between fluid and tube, hence

$$dF = f \times \frac{\rho w^2}{2} \times 2\pi R dl$$

where f = friction factor, ρ = density of fluid, w = flow velocity, $R = \frac{D}{2}$ = tube radius, l = tube length.

Then, the pressure due to this force is obtained by division by the cross area

$$dP = \frac{dF}{\pi R^2} = \frac{f}{R}\,\rho w^2 dl \quad [1]$$

Substituting the specific volume $v = \frac{1}{g\rho}$ and introducing the flow rate

$$G = \pi R^2 \frac{w}{v} \text{ lb per sec}$$

Equation [1] can be written

$$dP = \frac{fv}{g\pi^2 R^5}\, G^2 dl \quad [2]$$

From this equation the pressure drop can be calculated for both tube sections. Over the entire tube length L the heat Q is supplied in unit time. If the heat necessary to bring the water to saturation temperature is H Btu per lb, then the length of the lower tube section L' is

$$L' = L\frac{GH}{Q}$$

The specific volume of the fluid nearly equals the volume of saturated water v'. Then, the pressure drop in the lower section is

$$\Delta P' = \frac{fv'L}{g\pi^2 R^5}\,\frac{H}{Q}\,G^3 \quad [3]$$

The volume of the steam-water mixture in the upper tube section is

$$v = x(v'' - v') + v' \quad [4]$$

where v'' = specific volume of saturated steam.

For uniform heat absorption, the ratio of the dryness x at the point l to the dryness x_0 at the end equals the ratio of the pertaining lengths of the upper tube section

$$\frac{x}{x_0} = \frac{l - L'}{L - L'}, \quad [5]$$

All heat, added in the upper section, serves to supply the heat of evaporation r. Then, the total amount of steam generated is given by

$$Grx_0 = Q\,\frac{L - L'}{L} \quad [6]$$

From Equations [5] and [6], the local dryness fraction is obtained as

$$x = \frac{Q}{Gr}\,\frac{l - L'}{L} \quad [7]$$

and the local volume of the steam-water mixture from Equation [4] is

$$v = \frac{Q}{Gr}\,\frac{l - L'}{L}\,(v'' - v') + v' \quad [8]$$

If this value is substituted into Equation [2], integration of this equation between the limits L' and L then gives the pressure drop in the upper section as

$$\Delta P'' = \frac{fLv'G^2}{2g\pi^2R^5}\left(1 - \frac{GH}{Q}\right)\left[\frac{v'' - v'}{v'}\left(\frac{Q}{Gr} - \frac{H}{r}\right) + 2\right] \quad [9]$$

and the pressure drop through the entire tube is

$$\Delta P = \Delta P' + \Delta P'' = \frac{fLv'Q^2}{2g\pi^2R^5} \times \left[\frac{BH^2}{r}\left(\frac{G}{Q}\right)^3 - 2\left(\frac{BH}{r} - 1\right)\left(\frac{G}{Q}\right)^2 + \frac{B}{r}\frac{G}{Q}\right] \quad [10]$$

where $B = \frac{v'' - v'}{v'}$.

From this relation the pressure drop, for constant values H, can be represented by curves of the type shown in Fig. 7 of this

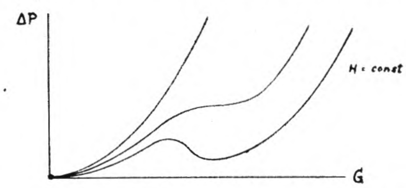


FIG. 7 CURVES REPRESENTING PRESSURE DROP FOR CONSTANT VALUES OF H

discussion. For small values H, there is a monotonous rise of pressure drop with the flow rate. At a certain higher value H, the curve shows a horizontal point of inflection, which indicates the transition from the stable region to the unstable region. For still higher values H, the curve has a maximum and a minimum; between these, the pressure drop decreases while the flow increases. In this region, three different values of the flow rate result in the same pressure drop; any one of these flow rates may exist in such a tube, when the pressure drop is given by the operating conditions of the other tubes. The location of the maximum, minimum, and point of inflection depends only on the specific-flow rate, $\frac{G}{Q}$ (lb per Btu). The value H, above which the region of instability begins, is that for which the first and second derivatives of Equation [10] are zero. It is found by differentiation that

$$H = (4 + \sqrt{12})\,\frac{r}{B} = 7.464\,\frac{r}{B} \quad [11]$$

while the pertaining value of the specific flow is

$$\frac{G}{Q} = \frac{\sqrt{12} - 3}{6} \times \frac{B}{r} = 0.07735\,\frac{B}{r} \quad [12]$$

Further improvement may be applied to these calculations by assigning to the friction factor (which is here assumed as constant) a certain function of the Reynolds number and thus handling f as dependent upon G also.[10]

It must be determined whether the range of instability can be reached under usual operating conditions. Since complete evaporation of the water was assumed, the flow rate which satisfies the instability condition must not be higher than the amount of water which can be evaporated by the heat quantity Q, namely

$$\frac{G}{Q} = \frac{1}{H + r} \qquad [13]$$

The curves for $\frac{G}{Q}$ from Equations [12] and [13] are plotted in Fig. 8 of this discussion, against the saturation pressure. It

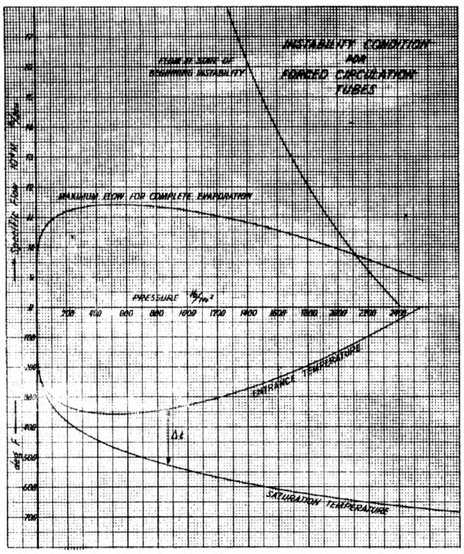


Fig. 8 Instability Condition for Forced-Circulation Tubes

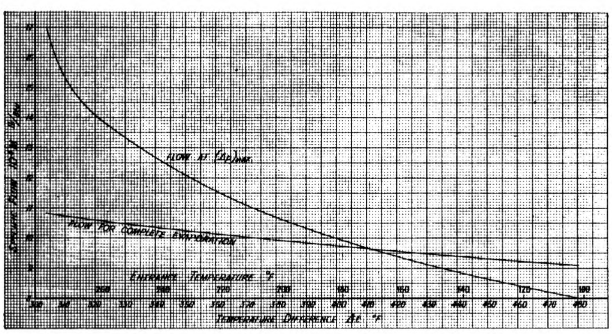


Fig. 9 Instability in Forced-Circulation Tubes at 1350 Psi

follows from the diagram that only for pressures higher than 2120 psi the range of beginning instability is reached. In the lower part of Fig. 8 are plotted the entrance temperatures of the water belonging to the values H, as calculated from Equation [11] for the same pressures. It is shown that, for those pressures where the range of instability could be reached, the entrance temperatures are below 120 F, a condition which is quite unlikely to be encountered. Since, for still higher values H the maximum of the ΔP curves moves to smaller values G, there is still a possibility that the range of instability may be reached. This check is made in Fig. 9 of this discussion for 1350 psi pressure. The specific flow $\frac{G}{Q}$, when ΔP has a maximum, has been calculated for various values H by differentiation of Equation [10] and plotted against the corresponding entrance temperatures, together with the curve of the amount of water which can be evaporated according to Equation [13]. It follows from Fig. 9 that, only for entrance temperatures below 170 F, the range of instability can be reached at this pressure. It has thus been shown to be quite unlikely in forced-flow tubes that the range of instability may be reached under normal operating conditions.

Natural-Circulation Tubes

The problem has a different aspect in a natural-circulation system. Only a fraction of the circulating water is being evaporated. The entering water has been heated in an economizer to a temperature approaching saturation. Therefore, the saturation point can be assumed to be at the tube entrance, and a steam-water mixture of varying dryness fills the entire tube. The direction of the flow is not assured and reversion of flow may occur, but only one value of the flow rate is possible as long as the fluid flows upward. The pressure difference between the lower and upper header consists now of the static head, the entrance loss, the acceleration loss, and the friction loss. We consider a circuit through a system of waterwall (vertical) tubes. For upward flow, the three losses diminish the pressure with the height; they act in the same direction as the static head, and the pressure drop due to the losses must be added to that due to the head. For downward flow, the losses act in the opposite direction to that of the static head, and the pressure drop due to them must be subtracted from that due to the head. The pressure difference due to the static head is

$$\Delta P_s = \int_0^L \frac{dl}{v} \qquad [14]$$

where v has to be substituted from Equation [4]. In this case ($L' = 0$) the ratio of the local dryness to the top dryness is

$$\frac{x}{x_0} = \frac{l}{L}$$

and the amount of steam generated is

$$Gx_0 = \frac{Q}{r} \qquad [15]$$

Then the dryness at the length l is

$$x = \frac{Q}{Gr}\frac{l}{L}$$

If this value is substituted into Equation [14], integration gives the static head as

$$\Delta P_s = \frac{GLr}{BQv'} \ln\left(1 + \frac{BQ}{Gr}\right) \qquad [16]$$

The entrance loss is

$$\Delta P_e = \frac{v'G^2}{2g\pi^2R^4} \qquad [17]$$

The acceleration loss is

$$\Delta P_a = \frac{x_0(v'' - v')}{g}\left(\frac{G}{\pi R^2}\right)^2$$

or with the use of Equation [15]

[10] "Stabilität der Strömungsverteilung in Heizflächen mit Zwangsdurchlauf," by A. Kleinhans, *Archiv, für Wärmewirtschaft*, 1939, pp. 135–138.

$$\Delta P_a = \frac{v'BQG}{g\pi^2R^4r} \qquad [18]$$

The friction loss is obtained from Equation [9] if we take $H = 0$

$$\Delta P_f = \frac{fLv'G^2}{2g\pi^2R^5}\left(\frac{BQ}{Gr} + 2\right) \qquad [19]$$

The total pressure difference is

$$\Delta P = \Delta P_s \pm (\Delta P_e + \Delta P_a + \Delta P_f)$$

where the plus sign is for riser tubes and the minus sign for downcomer tubes. From Equations [16], [17], [18], and [19] follows the total pressure difference

$$\Delta P = \frac{LrG}{BQv'} \ln\left(1 + \frac{BQ}{Gr}\right) \pm \frac{v'}{g\pi^2R^4}\left[\left(\frac{1}{2} + f\frac{L}{R}\right)G^2 + \left(1 + \frac{1}{2}f\frac{L}{R}\right)\frac{BQ}{r}G\right] \qquad [20]$$

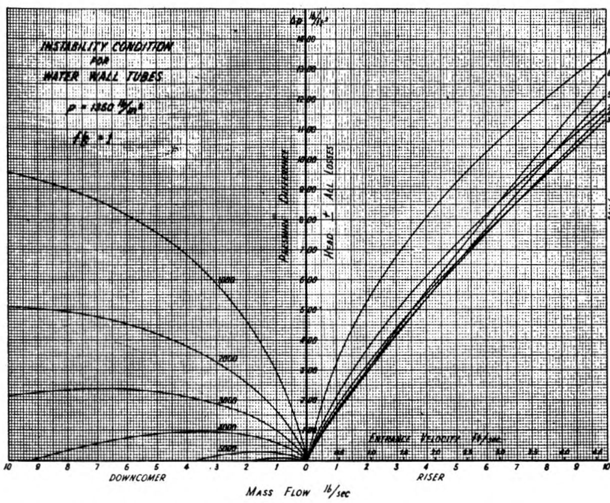


FIG. 10 INSTABILITY CONDITION FOR WATERWALL TUBES
$\left(p = 1350 \text{ psi}; f\frac{L}{D} = 1.\right)$

This relation between pressure difference, flow rate G, and heat absorption Q is shown in Fig. 10 of this discussion for the following example:

The pressure is 1350 psi, tube length 50 ft, tube diameter 3 in., friction factor $f = 0.005$. The steadily rising curves on the right-hand side refer to riser tubes, while the curves on the left-hand side which all go through a maximum refer to downcomer tubes. To each heating rate belongs one curve, consisting of two branches. When the mass flow in a riser is large enough, such that a horizontal line traced through the pertaining pressure difference does not intersect with the corresponding branch for the downcomer, then the flow is stable. There will be one flow rate at each heating rate, where the horizontal line through the pressure difference just touches the curve branch for the downcomer. This flow rate then represents the limit between the stable and unstable region for the given heating rate. For lower mass-flow rates, the horizontal line will intersect twice with the corresponding branch for downward directed flow; each of the three flow conditions may then exist in the tube, and the flow is unstable. Therefore, waterwalls must be laid out for such flow rates as assure stability of flow at the desired heat-absorption rate.

AUTHOR'S CLOSURE

Mr. Bailey is right in stating that the assumptions made in this paper should be correct. It would be better to state that all assumptions should be as nearly correct as it is possible to make them under the known conditions. It should also be understood that such assumptions cannot and need not be exact.

The desirability of actual tests to confirm circulation calculations cannot be questioned. In the absence of such tests, the operating records of a large number of similar high-pressure boilers may serve as a substitute.

The criterion of adequate circulation is the complete protection of the tubes. Just what is necessary to obtain such complete protection is not positively known. Each case is a problem in itself. In general, the degree of protection is dependent upon the percentage of water by volume, the velocity of the mixture, the inside diameter of the tube, and the position of the tube, i.e., whether it is vertical or at some angle from the vertical.

The mean-density determinations must start with the total heat input to the circuit and also the distribution of this input along the tube. If heat inputs are as assumed in Figs. 1 and 4 of the paper, the densities will be as calculated for these conditions, unless the steam moves faster than the water, in which case the mean density will, of course, be higher than calculated. However, for tubes of $2^1/_2$ in. inside diam and less, it is believed that there is little difference between the steam and water velocities, except for a relatively short distance from the bottom end of the tube.

If the arithmetic-mean density is used in calculating head and the same top-dryness fractions are used, in the assumed case the front and side walls will be in equilibrium at top-dryness fractions of 0.17 and 0.18, corresponding to approximately 75 per cent steam by volume.

Designers recognize the fact that absorption in different parts of a circuit, or in different parallel circuits, will vary due to conditions of firing, burner location, slag accumulation, etc. Here, the designer must rely on judgment based on his experience.

The term "dryness fraction" was used by Lewis and Robertson in the paper[3] referred to in the author's paper. The term seems to be fully explanatory and certainly less clumsy than "per cent of steam by weight." The reciprocal of top dryness, that is, the weight of water entering the circuit, divided by the weight of steam generated is the circulation ratio, which the author believes is also fully explanatory. The term "dryness fraction" can be used for any amount of steam by weight at any part of the circuit for which this information is desired or known, while the top-dryness fraction is the percentage of steam by weight at the point where the heat input ceases.

If different circuits are nearly identical, they may be solved simultaneously, otherwise they are solved separately and afterward corrected for any deficiency. In the paper, for example, the side wall and front wall are in equilibrium at different top-dryness fractions. Although not necessary, these circuits can be brought into equilibrium with the same top-dryness fraction by decreasing the resistance in the side-wall circuit.

Losses may be decreased by increasing either riser or downcomer areas as Mr. Bailey suggests. In the assumed boiler, there are as many risers as can be conveniently used, therefore, the best method for reducing the side-wall losses is by decreasing the downcomer area.

There might readily be some confusion in following the conclusion that decreasing the height of the furnace progressively decreases the circulation. However, if the total heat input remains the same, the available head will decrease and the top-dryness fraction will increase. With the increase in dryness, resistance will also decrease, but not as rapidly as the loss in head.

Mr. Dornbrook has brought up a point which is well worth discussion. It is the author's opinion that the safe "dryness fraction" at the top of a steam-generating tube depends upon a number of conditions, e.g., tube diameter, velocity, position of tube, whether horizontal or inclined, pressure, and heat input at the top end of the tube. It is believed that at any pressure a mixture of 75 to 80 per cent steam by volume will provide ample protection to tubes at the top end, provided the velocity of the mixture is sufficient to cause turbulent flow.

It is hoped that, as experience with large high-pressure units becomes more extensive, data on limiting velocities, dryness fraction, and transfer rates will become available.

With reference to Mr. Kuhner's comments on entrance and discharge losses in headers, these losses, expressed in head of fluid flowing, may be assumed as correct, provided the flow of fluid is across the header and not lengthwise. If the flow through the header is axial, naturally, headers should be of ample cross section to insure low axial velocity. In the paper, entrance losses were figured at twice the maximum theoretical figure of $^1/_2$ $V^2/2g$ to compensate for possible turbulence losses in headers.

If top-dryness fractions are figured for the maximum heat input, tubes having less than maximum heat absorption will have a lower top-dryness fraction. If, however, adjacent tubes absorb heat in greatly differing amounts, circulation may become unstable unless each tube connects directly with the drum without the interpositioning of an intermediate header.

The influence of heated downcomer tubes in a boiler bank is pointed out in the paper and, in such units as described, the lowered density due to the small amount of heat absorbed in the downcomer bank is not sufficient to impair the circulation. A downcomer bank, following a large furnace and a high-temperature superheater, absorbs but a small fraction of the total absorbed by the boiler and furnace.

As to the effect of change in pressure on the top-dryness fraction, it is obvious that, due to the increase in specific volume of steam, the calculated velocity of the mixture will increase at lower pressure for the same top dryness. The increased resistance will then increase the top-dryness fraction. Whether this decrease in percentage of water will necessitate a lowered rating will depend upon the liberality of design. For the relation between top-dryness fraction and relative volumes of steam and water refer to a previous paper[11] by the author.

As to Mr. Kuhner's suggestion that this method be applied to a study of the circulation of the boilers described in the paper[7] by Messrs. Hall and Partridge, it is the author's opinion that circuits such as shown in Fig. 3 of that paper cannot be analyzed by any known method.

In circuits such as that given in Fig. 14, showing the furnace at Rivesville, and Fig. 27, the Logan Station, also from that paper, analysis discloses entirely adequate circulation for the protection of the upper ends of the circuits. In such tubes, the velocity of the water entering is low, probably between $1^1/_2$ and $3^1/_2$ fps. At these velocities steam will segregate along the top of the tube and will remain so segregated until the velocity becomes high enough to cause turbulent flow. Such steam segregation or blanketing is the cause of the corrosion encountered in these installations.

Mr. Leib calls attention to the study of stability conditions of flow. He has improved the method developed in Germany by eliminating some simplifications and introducing the correct terms instead. It certainly is desirable to enable the designer to check flow conditions in a given tube system by means of an exact method. It must, however, be borne in mind that the method devised requires the knowledge of the amount of heat absorbed by the individual tubes in a parallel circuit. In general, this amount of heat can only be estimated. Therefore, the success of the method hinges entirely upon the accuracy of this estimation. As the knowledge of temperature distribution in furnaces progresses, this accuracy will undoubtedly improve, and, at the present state of our knowledge, a check of the stability of circulation in all cases where there is any doubt of stability is very desirable.

11 "Design of High Capacity Boilers," by J. Van Brunt, Trans. A.S.M.E., vol. 60, 1938, Fig. 10, p. 488.

Recent Developments of the Pease-Anthony Gas Scrubber

By R. V. KLEINSCHMIDT[1] AND A. W. ANTHONY, JR.,[2] CAMBRIDGE, MASS.

This paper brings up to date recent developments in an improved cyclonic-spray scrubber. Theory of design is discussed briefly, but more attention is devoted to actual installations and their performances, troubles, and possibilities. A table of test data is given covering units from 200 to 50,000 cfm. Scrubbing of boiler flue gases for fly-ash removal is covered with particular emphasis on possibilities for improved efficiencies and lower costs; possible benefits and economies through recovery and sale of SO_2 are indicated. Two installations for treatment of tar fog are described. Application to the cleaning and cooling of blast-furnace gas is reported in a preliminary way. A modification, retaining the principles of fine atomization but with the gas and liquid in countercurrent contact, is described, and compared favorably with bubble-cap columns and packed towers. Miscellaneous special applications are listed briefly.

IT IS four years since M. D. Engle (3)[3] reported to the A.S.M.E. on the first full plant installation of Pease-Anthony cyclonic-spray scrubbers. Since that time, development has proceeded actively so that, although these scrubbers are still in regular operation, they must, in the light of present standards of efficiency and economy, be regarded as obsolete. It would not be difficult to modernize this installation, for the physical changes involved are minor, involving only nozzles and baffling, as will appear later.

Theory of Scrubbing

The cyclonic-spray scrubber consists of a cylindrical chamber with a tangential gas inlet of suitable cross section near one end and a central gas exit at the other end, Fig. 1. A suitable spray of finely atomized particles of the scrubbing fluid is formed near the axis of the cylinder in the region directly above the inlet. Rotation of the gas in the chamber, due to the tangential entrance of the gas at a controlled velocity, causes the spray particles to travel outward through the gas to the walls of the chamber. The radial motion of the water particles across the gas stream causes them to collide with the dust particles and carry them to the walls from which they are washed down and discharged from the scrubber.

At the time these flue-gas scrubbers were designed, we had only hazy notions of the theoretical basis of design. The patent[4] indicated the necessity for using finely atomized spray, but it was believed that this was mainly because of the larger frontal area and better distribution of water particles. This view did not account for the rapid decrease in scrubbing efficiency, which is noted in the case of fine dusts, fumes, and smoke in all spray types of gas washers. Various reasons have been assigned for this phenomenon, among which may be mentioned inability to wet the dust particles, adsorbed gas layers, electric charges, and surface coating of the water droplets by the large number of fine dust particles which each water drop must remove. In most cases these appear to be decidedly minor effects, the major effect being simply derived by considering the aerodynamics of the collision of two particles surrounded by a gas. If the two particles are of comparable size the gas has little effect on the

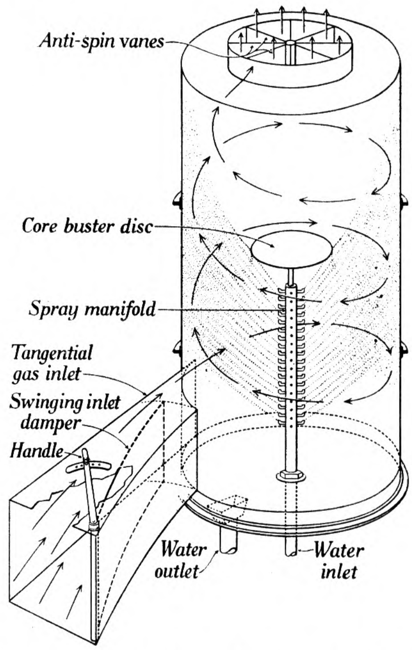


Fig. 1 Schematic View Showing Elements of Cyclonic-Spray Scrubber

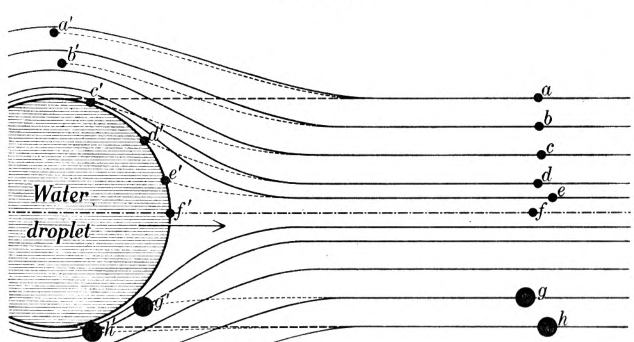


Fig. 2 Collision of Moving Water Droplet With Dust Particles of Various Sizes

[1] Physicist, Arthur D. Little, Inc., Consulting Engineer, Pease-Anthony Equipment Company. Mem. A.S.M.E.
[2] President, Pease-Anthony Equipment Company.
[3] Numbers in parentheses refer to the Bibliography at the end of the paper.
[4] "Method of Washing Gases," by F. F. Pease, U. S. Patent No. 1,992,762, Feb. 26, 1935.

Contributed by the Process Industries Division, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

collision but, if the particle which is in motion relative to the gas is large, it will be surrounded by a streamline pattern in the gas which will carry small air-borne particles around it without collision, Fig. 2. This effect has been demonstrated by H. G. Houghton (1)[4] in his work on sampling fog droplets in the air. Houghton has found that the sampling efficiency of a flat plate held in an air stream falls off rapidly for fog particles of a diameter smaller than about 0.001 of the width of the exposed plate. In the case of water particles hitting gas-borne dust particles, the effective diameter ratio is of the order of 200 to 1. This permits a direct quantitative computation of the effectiveness of any type of spray on any dust, fog, or fume of known particle size.

As to the type of spray, we have obtained with Houghton's cooperation considerable information on the numbers and sizes of spray particles formed by certain types and sizes of nozzles at various pressures, Table 1 (table for one nozzle). As to the sizes and properties of dusts, a great deal has been written in a general way, but it is difficult to obtain adequate data for design purposes on most of the finer dusts and fumes, Tables 2 and 3. In fact, in the case of extremely fine fumes and fogs, it may be simpler to determine particle sizes from operation of a pilot plant than to measure them directly, especially in the case of liquids or tarry fogs which agglomerate readily. Once the characteristics of the dust are known, it is a simple matter to apply the collision theory to determine the diameter and height of scrubber chamber, the number and size of nozzles, pressure of water, and pressure drop of gas. Several of these factors may be selected to meet the particular conditions, the others being then determined by the design theory. The details of this design method have been given by Kleinschmidt (2), but a résumé will be given here.

TABLE 1 SIZE DISTRIBUTION OF WATER DROPLETS FROM CENTRIFUGAL NOZZLE, 0.188 IN. AT 65 PSI, 60 F[a]

1	2	3	4	5	6
Diameter of drop[1] microns	Number of drops measured	Number per cc. of spray[2]	% of volume	Area per cc. of spray sq. cm.	% of area
25	878	97,250	0.072	1.9	1.48
50	460	51,000	.334	4.0	3.12
100	190	21,000	1.11	6 6	5.15
150	89	9,850	1.75	6.9	5.38
200	53	5,870	2.46	7.4	5.78
250	33	3,650	3.00	7.2	5.61
300	22	2,440	3.46	6.9	5.38
350	16	1,770	3.97	6.8	5.30
400	13	1,440	4.83	7.2	5.61
450	11	1,220	5.84	7.8	6.09
500	10	1,107	7.25	8.7	6.79
550	8	886	7.74	8.4	6.55
600	7	776	8.79	8.8	6.87
650	6	664	9.59	8.8	6.87
700	5	554	9.68	8.5	6.64
750	4	443	9.84	7.8	6.08
800	3	332	8.94	6.7	5.22
850	2	221	7.13	5.0	3.90
900	1	110	4.22	2.8	2.18
950	0	0	0	0	0
1200	0	0	0	0	0
	1,811	200,583	100.00	128.2	100.00

[a] Courtesy of American Institute of Chemical Engineers.

The collision theory assumes that dust removal is due solely to mechanical collision of moving water droplets with dust particles which happen to be in their path. It is also assumed, as previously explained, that only water particles less than 200 times the diameter of the dust particles are effective. It is assumed that the effective length of path of the water particles through the gas is the radius of the scrubber. These assumptions lead to conservative estimates of scrubbing efficiency in most cases. Taking into account the fact that the probability of hitting a dust particle becomes less as the concentration of dust in the gases decreases, we obtain as an expression for the efficiency of scrubbing:

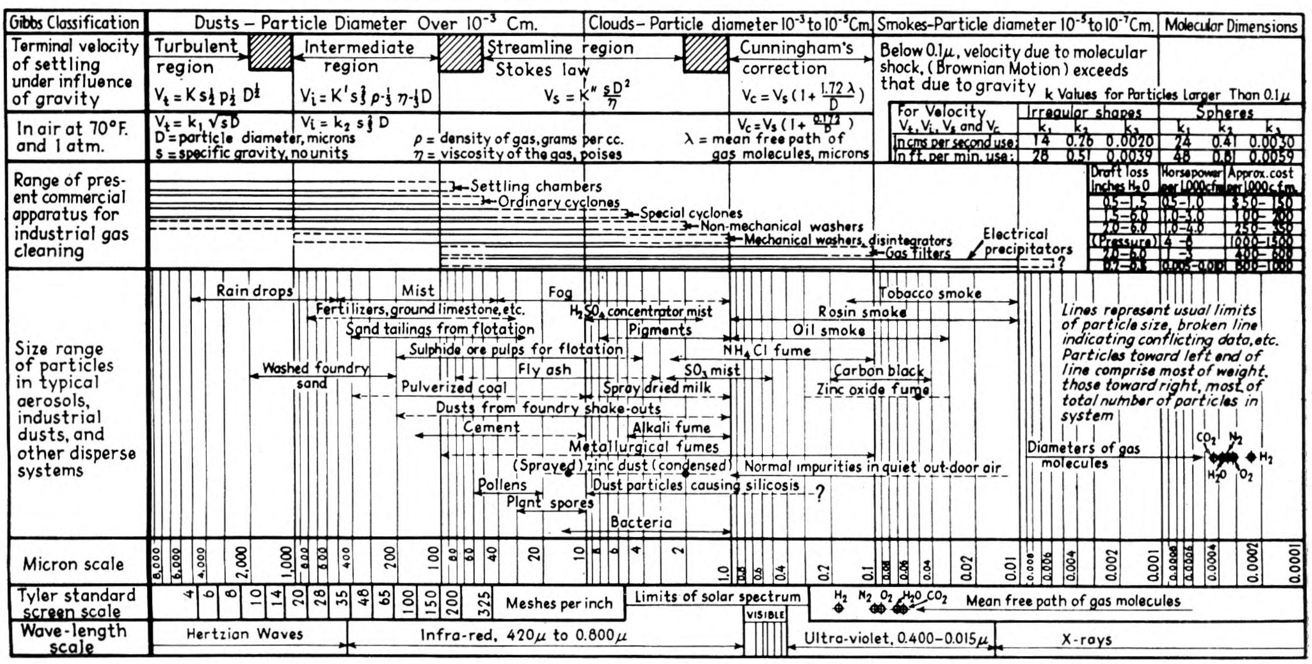
$$\text{Efficiency} = 1 - e^{(3DW/4dG)}$$

where D = diameter of scrubber, in.
d = diameter of water particles, in.
W = effective volume of water sprayed, cfm
G = volume of gas scrubbed, cfm

Values computed from this equation are given in Table 4.

Since the fineness of atomization which will be effective depends upon the size of the dust particles, it is necessary, in ap-

TABLE 2 SIZES AND PROPERTIES OF DUST PARTICLES AND FUMES[a]



[a] Courtesy of *Chemical and Metallurgical Engineering*.

plying this formula, to work out the efficiency of removal for each size range of the dust to be caught. The height of the scrubber and the inlet velocity required are computed from the centrifugal force necessary to drive the finest spray particles across the gas stream while the gas is within the scrubber.

It is interesting to note that entrainment of spray particles in the exit gases may occur at low gas loadings, since the centrifugal force decreases as the square of the gas inlet velocity while the axial velocity decreases only in proportion to the inlet velocity. Also, the gases rotate more or less in the manner of a solid cylinder, so that the centrifugal force at the axis is zero. Two recent improvements now incorporated in scrubber designs take care of these conditions. For low gas loadings, a swinging damper in the inlet permits control of that area with variation of gas flow, so as to maintain the gas inlet velocity at a suitable value. The height of the scrubber can be materially decreased by putting a circular plate baffle just above the spray manifold to act as a core buster to force all gas and spray away from the axis out into regions of higher centrifugal forces. See Fig. 1

Pilot Plants

It has already been noted that pilot plants are frequently desirable when handling new types of dusts or unusual conditions. Several recent installations have been made for this purpose, and it is important, in considering the results obtained, to bear in mind their nature and significance.

The purpose of a pilot plant is primarily to obtain certain specific engineering information, which cannot be obtained in the laboratory, for use in design of a larger plant. It is not intended to demonstrate the efficiency either of a unit of its own size or of a larger unit. In fact, it is important that it should not be designed for high efficiency since, if overdesigned, it becomes more difficult to obtain useful data. For this reason efficiencies of 50 to 80 per cent are better than higher ones. Another fact to be remembered is that, for a given efficiency, the ratio of spray to gas scrubbed is, inversely, as the diameter of the scrubber, so that a pilot-plant scrubber of 3 ft diam would require 5 times as much water per 1000 cu ft of gas as would a 15-ft unit of the same efficiency. At the same time, since the gas must accelerate the spray droplets introduced, excessive amounts of water reduce the rotation of the gas unduly and necessitate high pressure drop in the pilot-plant unit. For these reasons and others of similar nature, it is not possible to demonstrate either the efficiency or the economy of a large installation in a pilot plant. It is possible, however, to obtain data from which efficiency and economy may be computed with considerable accuracy.

Engineering Problems

The scrubber shell proper presents no serious problems. It may be made of steel and lined with acidproof tile if corrosive conditions are severe, or it may be a self-supporting bonded-tile structure. Large-size glazed sewer pipe has been suggested for small scrubber shells, Fig. 3. In any case, corrosion and erosion must be carefully considered. Gas flows of from 1000 to 60,000 cfm (60 F basis) are easily handled in single shells, at axial velocities of 100 to 500 fpm.

Nozzles present the most serious problems in materials. For fine atomization, large numbers of relatively small nozzles are required, and high pressures are desirable. Available materials and cost limit the pressures used to approximately 200 psi even with clean water; and waters containing silt or recirculated dust limit the pressure to around 60 psi. Even at these pressures, mechanical strength, and corrosion and erosion resistance limit the available materials. Some of the so-called "lava" materials have been useful in resisting acid corrosion; they are cheap, abrasion-resistant and resistant to most acids, but must be protected from mechanical and hydraulic shock. Brass is good only for clean water at low pressures. Hard rubber has been used to resist dilute fluorine compounds at low pressure. A new type of nozzle now under development gives promise of eliminating most of the erosion troubles experienced with spin-chamber

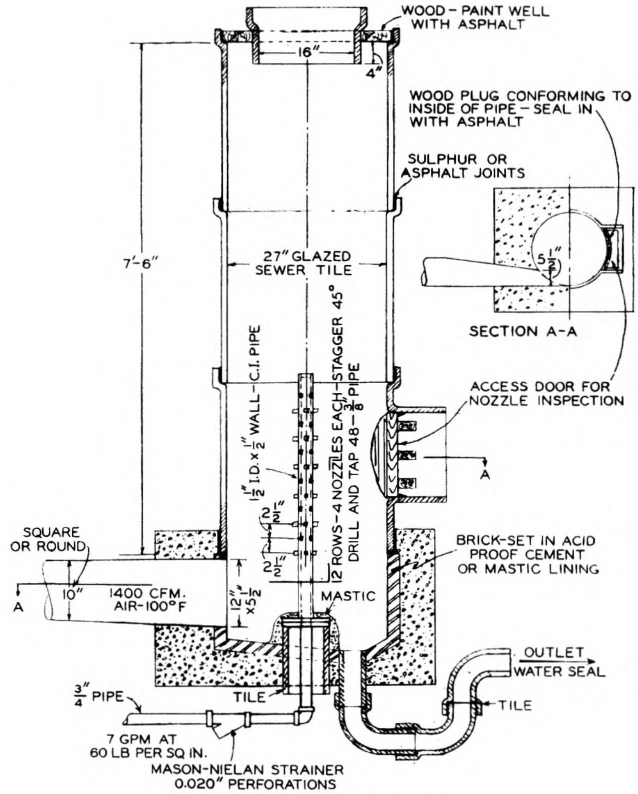


Fig. 3 Acid-Fume Scrubber

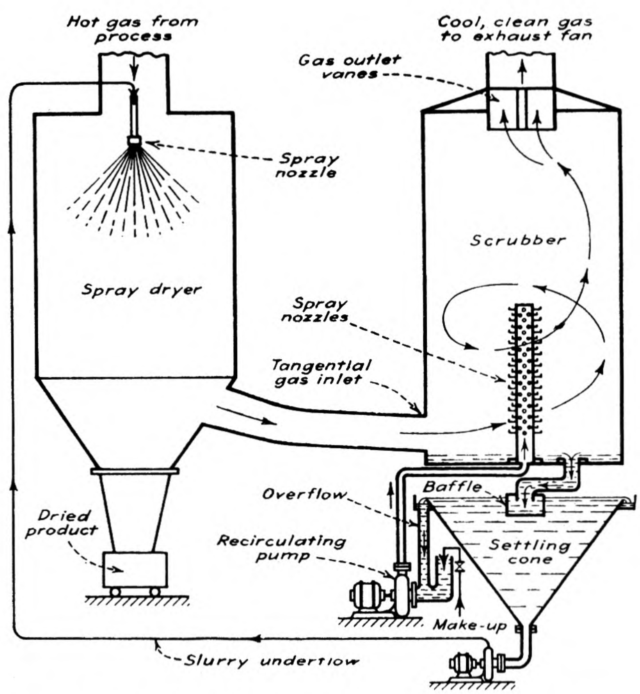


Fig. 4 Diagram of System for Recovery of a Dry Dusty Product

pressure nozzles. Every problem, however, requires a special study of the nozzle, and pilot-plant experience will often be valuable, until a vast background of experience has been accumulated.

Auxiliary Equipment

Handling Solids. Most scrubbing jobs require not only getting the solids into the liquid, but also getting them out of the liquid, either for recovery of the solids or for recirculation of the liquid. The solids may have value, or may be a costly nuisance. Local conditions usually dictate which of the numerous methods available should be selected. Fly ash from boilers has been settled out in Callow-type inverted cones, and the thickened underflow run to ash tanks for further settling. Tar from a certain tar fog formed a sticky mass which would not flow; as there was plenty of fresh water available, the effluent water drops its tar in an outdoor sump which is cleaned by a clam-shell-bucket crane several times a year, and the water then goes to waste. Soluble, valuable chemicals can be concentrated by recirculation in many cases, with a bleed-off going to processes already in operation. When hot gases containing chemicals are to be cleaned, the scrubber may be tied in with a spray drier, as shown in Fig. 4, the scrubber recovering dust from the drying operation while acting as a concentrator of the dilute solution.

Pumps. Centrifugal pumps with rubber-lined casings and rubber-coated impellers have given satisfactory service for 60 psi 115 F, weakly acid water carrying a small amount of fly ash in suspension. Numerous alloys were tried, without success. The cost and corrosion problems in high-pressure pumps limit the use of high nozzle pressures for corrosive conditions.

Piping. Rubber-lined pipe, with different degrees of hardness, has been useful. Tile pipe is frequently cheaper but must be supported. Lead pipe or lead-lined steel pipe and monel pipe have been used for certain severe conditions.

Although much of the development work on auxiliary equipment has been related to corrosion resistance, this problem is no more severe than with other types of wet scrubbers. The simplicity of the cyclonic-spray scrubber facilitates the attainment of resistance to corrosion and has therefore directed our attention to problems which no other equipment could handle.

Commercial Developments—Flue Gases

Flue-gas scrubbing is one of the most promising fields for this equipment, but widespread application has been held up pending full development of SO_2 recovery and sale, as a logical attack on the corrosion problem.

The only major installation of cyclonic scrubbers has been described by Engle (3). Two boilers were equipped in 1930, and an additional boiler was equipped in 1931. Engle covered the subject comprehensively, so only three points of universal interest will be considered here: efficiency, capital costs, and operating costs.

1 Efficiencies (on weight basis) of the installation were reported by Engle, and substantially confirmed by one of the authors. Little has been done since 1932, to improve efficiencies; in fact, owing to lack of fundamental knowledge, many things have been done tending in the opposite direction. In any event, by the use of more nozzles, better suited to the requirements, high efficiencies for these scrubbers can be maintained and yet keep within present costs of power for draft losses and for pumping.

Two scrubbers per boiler were installed in 1930, each rated at 72,500 cfm at 515 F. At the time of the Engle tests, these units had about 90 centrifugal-type nozzles of 0.1875 in. orifice, operating at about 55 psi, and showed full-load efficiencies of 80 to 82 per cent. Disregarding requirements for saturation of the gases, 180 efficient nozzles of 0.063-in. orifice, at the same pressure, would pass about 25 per cent of the amount of water, but the efficiency would rise to about 88 per cent. If 270 similar nozzles were used, with 37.5 per cent of the water, the efficiency would be about 93 per cent.

An improved type of nozzle will be ready for service in the near future, which will yield the same quantity of fine atomization with few of the coarse ineffective droplets, so that about 50 per cent of the gallonage of water need be pumped. Of course, to scrub with fine atomization, the gases must first be saturated but, assuming saturation of the gas, the following efficiencies can be realized commercially:

Nozzles Size, in.	Nozzles No.	Gpm, per cent	Removal efficiency, per cent
0.1875	90	100	80 to 82
0.063	180	25	87 to 88
0.063	270	38	92 to 93
0.063	New type	19	93 to 95
0.063	New type	38	96 to 98

Power requirements for pumping would decrease proportionately. While the foregoing figures may seem to be overoptimistic, similar methods in design indicated 98 per cent efficiencies on the scrubbing of blast-furnace gas, with realization of better than 99 per cent, as will be shown.

If entrainment in the scrubbed gases be eliminated by the addition of disks and swinging inlet dampers, less hot air need be added before the induced-draft fans, and a small increase in efficiency of the station can be realized. The swinging inlet dampers should be adjusted to maintain substantially constant inlet velocities at all gas loads and temperatures; draft loss will be the same at all loads, involving slight increases in fan power only at the lower loads.

2 Regarding the item of first cost, several years ago, when a prominent manufacturer was licensed to make and sell the scrubber, complete installations were quoted at prices which worked out at 25 cents to 35 cents per rated cfm (hot) for boilers of 100,000 lb per hr and larger. Corrosion was to be met by use of nonmetallic materials in contact with the gases and solutions.

If, however, corrosion can be eliminated so that plain, unprotected steel can be used, the cost of the scrubber, pump, piping, and nozzles can be cut at least 50 per cent. Apparently, the best chemical method is the use of an alkaline washing solution; however, the scrubber is such an excellent gas-liquid contactor that the alkaline solution will soon become acid, unless continuously replenished or regenerated to maintain the alkalinity. In England, lime or chalk slurries are continuously replenished, with a minimum of corrosion, but a considerable mass of fly ash and gypsum must be disposed of at considerable expense. H. F. Johnstone at the University of Illinois has studied extensively the regeneration of alkaline solutions and very recently he has reported important conclusions (4). Although he reports the absence of corrosion in his pilot plant, nevertheless he recommends some protection of steel surfaces, but not to the extent described by Engle. Acidproof units installed now for solids removal will, of course, be able to absorb SO_2 simultaneously, if and when the public convenience and necessity require.

3 Maintenance of the scrubber proper has been low, but in the case of the protective coatings of the induced-draft fan and uptakes, it has been high because no entirely satisfactory coating has yet been developed. On the other hand, erosion of induced-draft-fan rotors has been negligible; life expectancies of rotors under normal acid conditions are now certainly 10 years, and possibly equal to those of forced-draft-fan rotors. Future costs should be lower, because of advances in the art of synthetic resinous coatings, which are now available having substantially zero

moisture absorption; these are needed for the interiors of the induced-draft-fan casings, the uptakes and breechings. Heretofore, all the coatings tried have absorbed moisture and transmitted it to the metal beneath, with resultant corrosion, so that the coatings have required annual replacement. Such coatings probably will not be needed when alkaline scrubbing solutions are used; the indications are that plain steel can then be used without protection, or with a coating which normally should last for several years.

Entrainment at low gas loadings has increased the corrosion troubles in the past. Entrainment is unnecessary, and can be completely eliminated, as previously pointed out.

Miscellaneous—Flue Gases

Disposal of Fly Ash. Building blocks utilizing considerable fly ash are reported from Detroit (5). The cement industry reports beneficial results from the admixture of certain percentages of fly ash (6). The mixed-fertilizer industry uses it as a diluent in some cases (5). Its disposal after removal from the flue gas is still a problem, although the firing of pulverized coal in wet-bottom furnaces apparently retains more of the ash, and proportionately less goes up the stacks.

Mist Plume. Since the gases at the top of the stack contain considerable water vapor, during cold weather this becomes visible as a white plume or cloud, having a good deal of the appearance of steam although not quite as dense. The action of this plume has been studied under all weather conditions, and in general it behaves as would any other body of heated gas. It rises when all stack gases rise, and it strings out horizontally or even downward when wind conditions compel. In cool weather, as the gas emerges from a large stack, only the water vapor in the outer shell of gas, in immediate contact with the cool atmosphere, condenses, but turbulence and diffusion soon cause further condensation in the interior of the plume. A little later, evaporation of the fog droplets on a large scale sets in, and soon the white plume is gone. On days of high humidity, as would be expected, the plume is more persistent, and it has been observed stringing out as far as $^1/_2$ mile.

Acid Rain. Fears have been expressed that an acid rain would fall on those beneath. We have been at some pains to find evidences of acid attack chargeable to the scrubber, but have found none. Qualified chemical engineers, who have studied this question of acid effects from the effluent scrubbed gases, have concluded that the over-all results are beneficial as to SO_3, which is largely scrubbed out by the wash water and ultimately run to waste, and with but slight effect as to SO_2, some of which probably is absorbed by the fog droplets as formed at the top of the stack, but is soon liberated when the fog evaporates.

Sulphur Dioxide Recovery

Some of the benefits of SO_2 recovery as a by-product have already been suggested. When flue gases are thoroughly scrubbed, the SO_2 present is a problem from either the standpoint of corrosion or of disposal of the recoverable by-product. However, if the gases are dry-cleaned, there is no problem at least to the boiler-plant operator, since the SO_2 is turned loose to the prevailing winds. Papers have been published to prove that SO_2 harms neither plants nor human beings, but nothing has yet appeared showing beneficial effects on life, or on steel, concrete, paint, etc. In short, SO_2 is on the defensive, and is generally considered as a nuisance which must be tolerated for the present, at least, in this country.

The ideal solution of the combustion problem would be to attain stack discharges containing neither solids nor acid gases, but consisting only of nitrogen and oxygen, with CO_2 and water vapor. Probably this ideal is not possible of achievement, but 90 to 95 per cent elimination is not only possible, but may be accomplished at no great increase in over-all cost, if and when due credit is allowed for the advantages, which include:

(*a*) Credit for sale of SO_2.

(*b*) Virtually complete elimination of need for maintenance of protection against corrosion.

(*c*) Minimum boiler outage chargeable to maintenance of scrubbers.

(*d*) Smaller induced-draft fans and less power to drive them, because gases are handled at lower temperatures.

(*e*) No erosion of induced-draft-fan wheels.

(*f*) Improved public relations, if the public is advised of the new methods being installed.

There are of course disadvantages:

(*a*) The price for SO_2 is not fixed and definite, and it will fluctuate with supply and demand, but it probably will not depart far from the price of the contained sulphur. The quantities available may be enormous, relative to present supplies. New outlets must be developed, and fortunately at least one is in sight, i.e., the new plastics containing nearly 50 per cent sulphur dioxide (4).

(*b*) Power-plant operators must know more chemistry.

(*c*) Duplication where solids eliminators are already installed.

(*d*) Fixed charges on increased plant.

If public interest is justified in requiring substantial elimination of cinders and fly ash from power-plant stacks, there is even more justification for requiring the elimination of SO_2, because probably the complete job can be done ultimately at small increase in annual cost of effective solids removal. If, however, SO_2 recovery is expected to show a profit after meeting annual costs not only on its own operations but also on those for solids removal as well, then the outlook for general acceptance is not favorable.

Tar-Fog Scrubbing

Fig. 5 shows a commercial cyclonic scrubber 8 ft diam $\times$ 20 ft high operating to remove tar fog from 28,000 cfm of the vent gases from ovens in which a pitch binder is decomposed. The resulting tar forms a viscous mass which adheres to the surfaces of ducts and fans, and which had caused serious trouble by clogging all other types of cleaning equipment, both wet and dry. This scrubber has been functioning satisfactorily for more than a year. Tar-fog droplets, as collected on a glass slide, are all below 10 mu and 89 per cent below 2 mu. Tests on this scrubber confirm the design theory as to effect of increasing water pressure. At 40 psi water pressure, the efficiency was 65 per cent and this was increased to 80 per cent by raising the pressure to 110 psi. This performance is entirely satisfactory to the users in that maintenance on the fan has been practically eliminated whereas, previously, the fan could not be kept in balance for more than a few days at a time. The water from this installation is not recirculated but is run to waste.

An important but secondary function of this scrubber is its action as a flame stop. The gases laden with tar fog are normally at moderately high temperatures, but occasionally fire breaks out in the tar deposited in the ducts, with possible damage to the induced-draft fan. The scrubber functioned as expected through one such fire, although all paint was blistered from the inlet duct.

The scrubber shell of this installation is of unprotected steel and its life has been rather short; it has been patched up and is still in operation but is scheduled to be replaced in the near future. Lava nozzles of smaller size are now being used but reports on efficiency with these nozzles have not been received.

Fig. 6 shows a pilot-plant scrubber installed to treat a tar fog

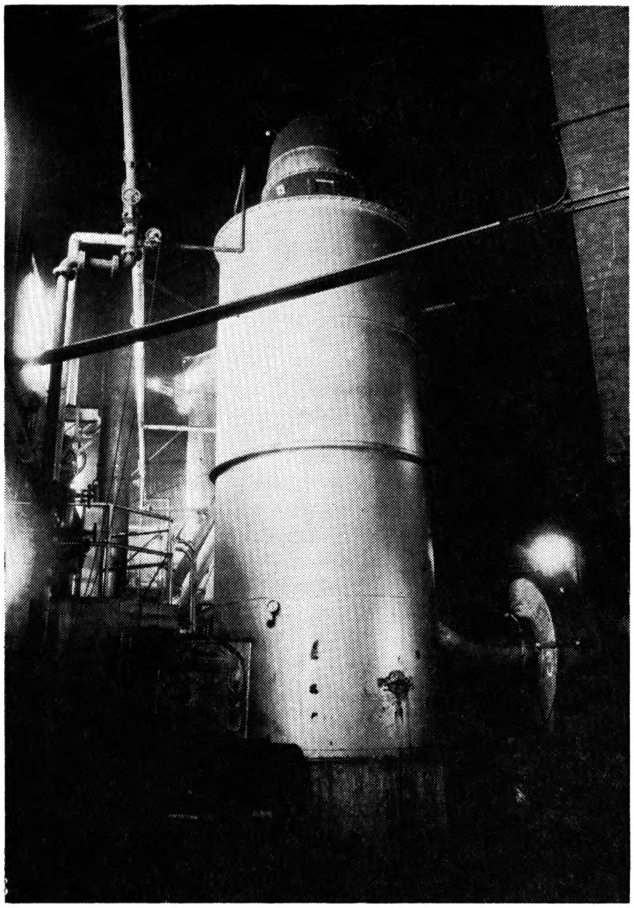


Fig. 5 Scrubber Handling 28,000 Cfm of Gas Containing Tar Fog

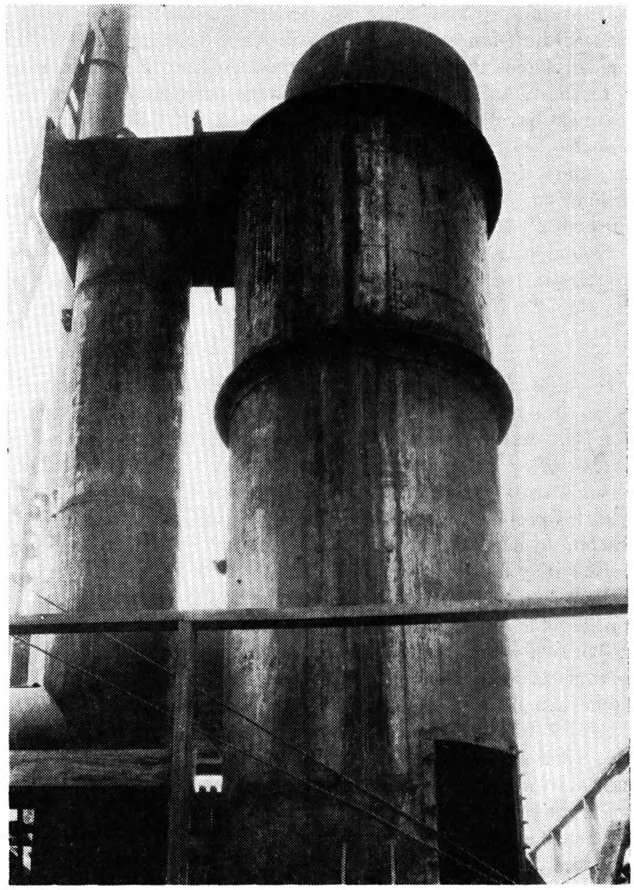


Fig. 6 General View of Tar-Fog Scrubber With Pressure-Regain Type of Outlet

from a sintering operation, the individual tar droplets ranging from 2 mu down to below 0.25 mu, the limit of resolvability by an ordinary microscope. From theoretical considerations, there is strong evidence that size of these tar droplets goes down to 0.1 mu and below. Efficiencies reported by this unit are lower than anticipated, chiefly because of the small size of the individual tar droplets. The most intensive scrubbing which has been provided, namely, 12 to 15 gal per 1000 cu ft through 335 nozzles (Fig. 7) of 0.037 in. diam at pressures as high as 320 psi made little effect on the appearance of the effluent cloud of tar fog, even though 50 to 75 per cent by weight of the material had been removed. Calculations in the light of the collision theory indicate that the individual tar droplets are present in overwhelming numbers. Of the 15 trillion tar-fog droplets in each cubic foot of dirty gas, about 3000 droplets were removed by each of the three billion water droplets formed by the nozzles per cubic foot of the gas. Doubling of this already intensive scrubbing should raise the efficiency to 85 or 90 per cent, which might begin to be perceptible to the naked eye.

Blast-Furnace Gas

The conditioning of blast-furnace gas has been considered one of the most difficult of gas-cleaning operations. Outlining the problem, approximately 25 per cent of the heat value of the coke charged into the top of a blast furnace comes out with the gases from the top as CO, in about 25 per cent concentration, or 90 to 105 Btu per cu ft. The heat quantities are enormous, and the gas is used in stoves for preheating the blast air, in large gas engines for blowing and power generation, for miscellaneous heating purposes; any residuum is burned under boilers for steam production. The requirements for cleanliness vary widely. Frequently raw gas is burned under boilers; however, Harmon (7) has recently pointed out the savings to be made by cleaning to 0.05 grain per cu ft. Gas for stoves should be cleaned to 0.02 grain per cu ft, and for engines to 0.015 grain per cu ft and better (8). Dehumidification also is necessary to obtain higher combustion temperatures and to minimize condensation and freezing in the gas mains in the winter.

To provide this cleaning and cooling, most blast furnaces are now equipped with three mechanisms handling the dirty gas in series, as follows:

1 Dry dust catchers, i.e., large dry cyclones, removing the brickbats and coarse dust.

2 Primary washers, of which there are numerous types, most of them operating to clean the gas down to 0.25 grain per cu ft. Large quantities of water are used in order to cool and dehumidify the gas while cleaning, 20 to 30 gal or more per 1000 cu ft of gas, to absorb the heat content of the dirty gas.

3 Secondary cleaners, which include rotary disintegrators and electrostatic precipitators. Preliminary data indicate that the cyclonic scrubber will perform simultaneously the duties of mechanisms Nos. 2 and 3 in one unit, with low power costs both for draft loss and for pumping, as well as much lower capital costs.

The operator of a number of blast furnaces, learning of the performance of this type scrubber for fly-ash removal from boiler flue gases with small quantities of water, recognized its possibilities for his purposes where much larger water quantities were

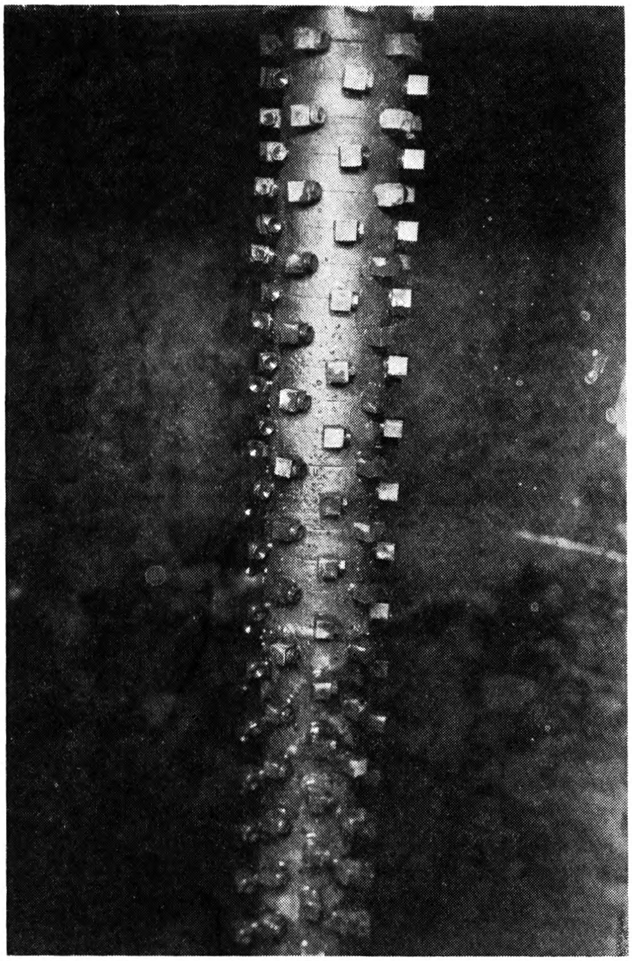


Fig. 7 Close-Up of Spray Manifold Showing Some of the 335 Nozzles With Orifice 0.037 In.

required. A full-scale preliminary tryout was arranged, involving the conversion of an existing primary washer (one of three in parallel) to the cyclonic type by changing the gas inlet from radial to tangential, and by installing an axial spray manifold of more than 500 brass nozzles. The gas piping, designed to handle one third of the gas from the furnace, was not changed as it was expected that this piping could handle all the gas during the short periods of test. Unfortunately, the draft loss through the overloaded gas piping was so high that it was not possible to obtain the complete test data originally planned for. Two test points were obtained and these showed efficiencies (weight basis) of 99.5 per cent. During normal operation, when dirty gas loadings run 2.5 to 3 grains per cu ft, the clean gas contained 0.012 grain per cu ft. During a period of severe slipping in the furnace, when dust loadings run 15 to 25 grains per cu ft, the clean-gas test showed 0.068 grain per cu ft. In addition, the operators considered the gas to be exceedingly dry, and were pleased with the degree of cooling obtained, in spite of the fact that only 17 gal of water per 1000 cu ft were used. The pressure for production of pig iron has rendered impossible the making of desired changes in the gas piping. This development has also been delayed with the problem of nozzle materials capable of withstanding erosion from the gritty water; the new nozzle referred to previously is on test, and gives promise of being satisfactory.

Chemical Applications

For solids, recovery primarily the scrubber has interesting possibilities in connection with the recovery of soda fume, along with considerable quantities of SO_2 and SO_3, from the discharge of the furnaces in which is burned the black liquor from paper mills cooking kraft, or sulphate pulp. The soda represents a recoverable value of considerable proportions. The particles, however, are very fine and not readily recovered. Pilot-plant tests indicate that the fume is almost entirely below 2 mu with a majority of the material in the neighborhood of 0.2 mu. These tests also indicate that the material can be caught in a carefully designed scrubber. The solution should be recirculated to provide a high concentration of soda in the effluent liquor, so that it can be economically treated for the recovery of the soda. The

TABLE 3 CHARACTERISTICS OF FINE DUSTS

(Dusts assumed to be spheres of density = 2)

Diameter, mu	Settling rate, fpm	No. particles per grain (millions)
20	4.8	15.6
10	1.2	125
5	0.30	1,000
2	0.048	15,600
1	0.012	125,000
0.5	0.003	1,000,000
0.2	0.0004	15,600,000
0.1	0.00007	125,000,000

TABLE 4 EFFECT OF QUANTITY OF SPRAY ON DUST REMOVAL

Number of times gas volume is swept by spray	Efficiency of dust removal per cent
1.0	63.2
1.5	77.7
2.0	86.5
3.0	95.0
4.0	98.2
5.0	99.34
6.0	99.76
7.0	99.91
8.0	99.97

TABLE 5 GAS-SCRUBBER TESTS

Rated Capacity c.f.m. Saturated	Gas and Dust	Conditions		Dimensions		Draft Loss in H_2O	Eff. %	Spray					Corrosive Conditions	Construction Materials			
		gr./c.f.	Size	Dia. ft.	Height ft.			Gal. per Mcf.	Nozzle No.	Orifice Dia.	Press. psi.	Recirculate		Shell Material	Shell Life	Nozzles Material	Nozzles Life
200	Chemical fume	17.0	0.5-3.5	1.0	6.0	9.	94	10	24	.037	60-360	Yes	Alkaline	Steel	?	Brass	Hours
"	Tar fog	2.-3.	0.1-2.0	"	"	"	91	10-30	"	"	75-360	No	SO_2, HF	"	?	"	"
750	Laboratory unit	Miscellaneous		1.5	5.0	1.5-2.3	-	3-12	max. 36	.046	40-600	No	-	"	Indef.	"	-
2,000	Soap dust	Var.	large	3.0	6.0	1.0	99+	1-5	rotary		-	Yes	No	"	"	"	Indef.
2,000	Chemical fume	2.25	0.2-2.0	3.0	30.0	4.0	65	6	48	.046	400	No	SO_2, etc.	"	"	Monel	?
5,000	Tar fog	2.0-3.0	0.1-2.0	4.0	12.0	5.0-12.0	50-75	5-15	335	.037	100-320	Yes/No	SO_2, HF	Lead lined	"	Monel/Brass	Days
15,000	Soap dust	Var.	large	7.0	12.0	2.5	99+	0.25-1.5	rotary		-	Yes	No	Steel	"	Brass	Indef.
25,000	Tar fog	.018-.027	0.1-2.0	8.0	20.0	3.0	65-80	2.8-4.6	164	.096	45-118	No	SO_2, HF	"	8 mo.	"	6 mo.
45,000	Boiler fly ash (one unit)	.25-2.5	2.0-5.0	10.3	20.0	0.8-1.5	82-95	2-3	80	.188	50-55	Yes	SO_2, etc.	Acid brick	10 yrs.	Lava	2 yrs.
50,000	Blast furnace	2.50	-	12.0	60.0	9.0	99+	17	534	.140	75-80	No	No	Steel	Indef.	Lava	Indef.

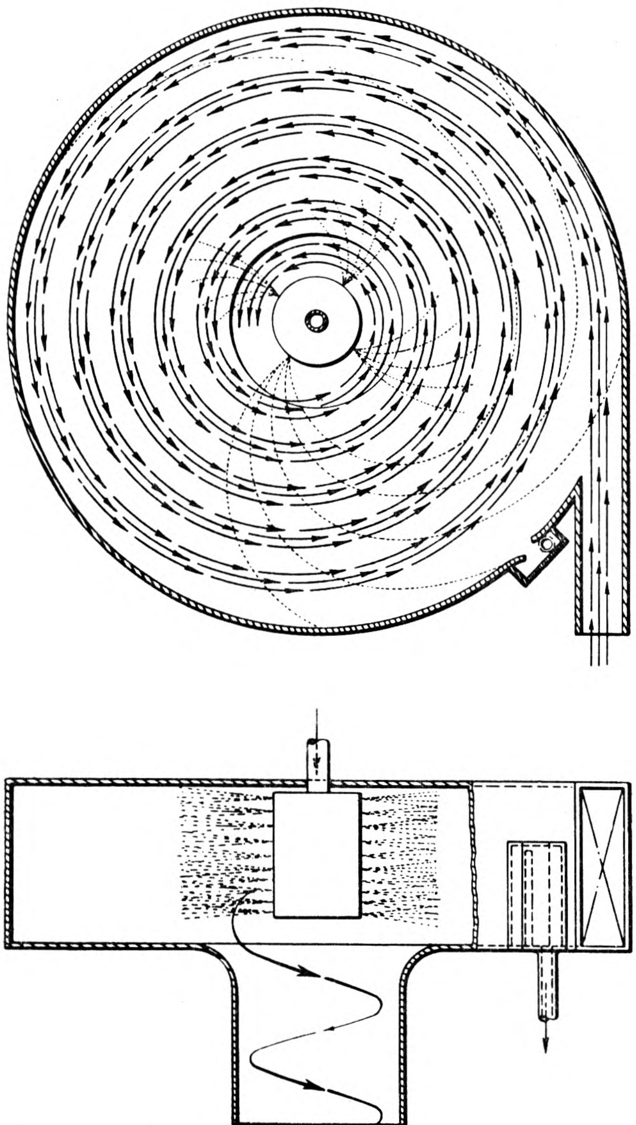


FIG. 8 VARIATION OF CYCLONIC-SPRAY-SCRUBBER PRINCIPLE, GIVING COUNTERCURRENT CONTACT BETWEEN GAS AND LIQUID

solution is strongly acid due to SO_2 and SO_3 in the gases, and the scrubber materials should be similar to those used in flue-gas scrubbing. Such an application offers an attractive economic picture as well as reduction of atmospheric pollution, both by solids and by "paper-mill odor."

The cross-current scrubber is able to provide such intimate gas-liquid contact that it is being considered for many chemical applications, among them gas absorption, removal of dilute acid gases from industrial process gases, elimination of odor nuisances from sewage and garbage-treatment plants, recovery of solvents, recovery of natural gasoline, vacuum fractionation of lubricating oils and of synthetic liquids of high molecular weights. It may be used as an evaporator or concentrator, as with a spray drier, Fig. 4.

Data on the efficiency of this scrubber for the absorption of soluble gases, i.e., SO_2 from flue gases, obtained in a large-scale installation, have been reported by Johnstone and Kleinschmidt (9). In this case 96 to 99 per cent of the SO_2 in the gases was absorbed in the scrubber under conditions which gave approximately 85 to 90 per cent dust removal. Thus the absorption efficiency is much higher than the efficiency of dust removal. This accords with theoretical predictions, since diffusion of the gas molecules increases the effective range of action of liquid particles.

Countercurrent Arrangement

Fig. 8 shows an important variation of the cyclonic-spray-scrubber principle, which yields true countercurrent contact between gas and liquid, with several obvious advantages. The scrubber shell is disk-shaped, large in diameter in relation to its height. The gas enters the full height of the periphery by means of a narrow slot, and pursues an inward spiral path, leaving axially at the center. The spray droplets are introduced substantially along the axis, and are taken up by the spinning gas body and pursue spiral paths across to the periphery. This arrangement has been tried out in a laboratory size 12 in. diam $\times$ 3 in. high, which passed 60 to 95 cfm and yielded contact equivalent to from 2 to 4 effective plates. This type of unit would be effective in absorption of ammonia in water, of H_2S in weak alkaline solution from which it is to be recovered by heating, of benzol vapors in wash oil, and numerous similar applications.

It is expected that, as more experience is gained, it will be possible for many applications to exceed with these units the performance of bubble-cap columns and packed towers with much lower space, weight, and costs.

Conclusions

The cyclonic-spray scrubber described has several advantages of importance; it is simple and versatile, has low draft loss, and has nothing to clog except spray nozzles. Efficiency is readily adjustable by change of pressure on the atomizing nozzles, and it can be very high for solids down to about 0.5 mu in size. Entrainment can be reduced to substantially zero, of importance in many chemical processes. This scrubber is especially effective for the economical cleaning of large volumes of gases, such as blast-furnace and boiler-flue gases. It can act as an absorber or gas-liquid contactor with simultaneous removal of dusts, as for instance SO_2 and fly ash from boiler flue gases.

Acknowledgments

So many different individuals and organizations have had a part in the development of this scrubber that a long list would be required to name them. Since, however, in many cases, identifications have not been possible, specific acknowledgments are omitted.

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Discussion

H. F. Johnstone.[5] The authors' reference to the stack-gas investigation being conducted at the University of Illinois warrants discussion of two points mentioned in the paper (*a*) the economics of sulphur-dioxide removal and, (*b*) the application of cyclone-spray scrubbers as gas absorbers.

On several occasions, the writer has published statements regarding the cost of removing sulphur dioxide from dilute waste gases. The most recent of these was the result of a detailed study of what appears, at least from the chemical and mechanical standpoint, to be the best method of accomplishing this purpose.[6] Viewed from any position, it is recognized that the application of any process of sulphur-dioxide removal to large quantities of stack gases requires a large and costly installation. The item of greatest uncertainty and perhaps of greatest cost is the disposal of the recovered material, whether it be as a waste product or as a chemical raw material to which some value can be attached. In any case, much more information is required before it can be stated that simultaneous dust removal and sulphur-dioxide recovery is feasible and that definite savings can be made in the cost of the scrubber installation by removing the corrosive conditions encountered in the circulation of the acid solution.

The paper refers also to the tests made by Dr. Kleinschmidt and the writer on the absorption efficiency of a large dust-recovery unit in which an alkaline solution was circulated for the purpose of the tests. The high efficiency obtained in this case, compared with the known low absorption efficiencies of simple spray scrubbers, can be explained easily on theoretical grounds. For certain purposes, therefore, especially when saturation of the solvent is not desired, the wet cyclone would seem to fulfill the needs of many chemical-engineering absorption problems. Here again, it is unfortunte that more information is not available from other installations. It is particularly desirable to know the effect of the dimensions of the scrubber, of the location and size of the entrance duct, and of the number and type of nozzles on the absorption efficiency. Knowledge of the nature of the flow in the vortex and especially the tendency for the droplets to coalesce would be valuable. While the authors did not mention the latter point, it is obvious that it must be of great importance in dust removal also. Simple calculations will show that, unless the spray from the nozzles is quite uniform, the probability of coalescence of the small drops with larger drops is extremely great at radii above 3 or 4 ft. Consequently, the statement in regard to the relative water requirements of scrubbers of different sizes appears to be subject to some limitation.

[5] Professor of Chemical Engineering, University of Illinois, Urbana, Ill.

[6] "Recovery of Sulfur Dioxide From Waste Gases," by H. F. Johnstone and A. D. Singh, University of Illinois, Engineering Experiment Station, Bulletin No. 324, 1940. Abstract, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1037–1049.

Authors' Closure

Professor Johnstone has brought out several points which are of interest in connection with gas scrubbing but which the authors felt unable to treat adequately in the allotted time. As to the commercial recovery of sulphur dioxide, it seems that Dr. Johnstone's position is one which arises from his academic point of view. It is unquestionably true that we do not at the present time know all that we would like to know about the economic and industrial problems confronting the recovery of sulphur dioxide. At the same time it is also probable that such complete knowledge never can be and never has been acquired with respect to any commercial process. It is felt that Dr. Johnstone's publications on the subject indicate an adequate basis for a careful commercial study of a large installation which, like all first installations of radically new processes, must be regarded as experimental. The cost figures, which in Dr. Johnstone's opinion do not appear too favorable, include such unknown factors as the percentage of solution lost in carry-over in the gases and other similar losses from the cycle. Such items can only be definitely determined in large-scale operation, and if they become important, as they appear in Dr. Johnstone's figures, they can usually be reduced by proper design or operating procedure. From the authors' own experience, it is concluded that the loss of alkaline solution in the scrubbing step would be practically negligible.

As to the factors affecting the efficiency of absorption, their experience has been that it is a very simple matter to obtain such high percentages of absorption with the present type of scrubber that it is not necessary to go to any elaborate determination of the minimum required equipment except in special cases which might arise. Coalescence of the small drops into larger drops undoubtedly occurs to a certain extent since a spray which scrubs out small dust particles should also scrub out water droplets. Practically, coalescence appears to have little effect on the performance of scrubbers as is indicated by the fact that the actual performance follows rather closely the computed efficiencies.

In view of the many uncertainties involved in applying both the theory and the experimental results previously obtained on other installations to new and different problems, it is believed that further development of the scrubber will be along the lines of engineering development and experience in numerous applications, rather than in any extensive laboratory study of the well-known factors involved.

Relationship of Viscosity to Rate of Shear

By L. J. BRADFORD[1] and F. J. VILLFORTH, JR.,[2] STATE COLLEGE, PA.

This paper reports tests designed to check the validity of the assumption that lubricating oils belong to the class of a Newtonian fluid, which is defined as one in which the force required to shear it is directly proportional to the rate of shear. It is on this assumption, which in recent years has been questioned, that equations for the behavior of bearings have been based. Experimental evidence is produced in support of Petroff's equation which states that the torque required to rotate a journal, concentric with its bearing, is directly proportional to the product of the absolute viscosity and the rate of shear of the fluid separating the journal and the bearing. The agreement of the results with those predicted by the Petroff equation upon the assumption of the independence of viscosity from the rate of shear holds for all of the oils investigated.

A NEWTONIAN fluid is defined as one in which the force required to shear it is directly proportional to the rate of shear. Lubricating oils are generally supposed to belong to this class of fluid, and the equations applying to the behavior of bearings are based on this assumption.

Within the last few years the validity of this assumption has been questioned, and certain experimental evidence has been produced to show that the resistance to shear varies with the rate of shear. In other words, the viscosity of an oil is a function of the rate of shear as well as of temperature and pressure. The claim has been advanced that shearing of the laminas, which may be considered as forming the film separating two moving plates, causes the molecules making up the film to orient themselves. This orientation is said to cause a reduction in the resistance to motion of each lamina with respect to its neighbor. This is another way of saying that molecular orientation caused by flow results in a decrease in the viscosity of the fluid. The claim is also made that the higher the rate of shear, the greater the resulting orientation and, consequently, the greater the decrease in the viscosity of the fluid sheared.

Since the entire treatment of bearings operating in the fluid-film region has been built up on the supposed independence of viscosity from the rate of shear, a modification of the hydrodynamic theory of lubrication would be necessary if such independence really does not exist.

Apparatus Used

Two pieces of apparatus were used in this work. The first consisted of a steel ring suspended within a rotating steel bowl by means of a flexible steel rod. A fixed clearance was maintained between the ring and the bowl. Oil was introduced into the bowl and thrown by centrifugal force into the clearance space through which it passed. The assembly of this piece of apparatus is shown in Fig. 1. Details of the bowl and ring together with the more important dimensions are shown in Figs. 2 and 3. It will be seen that the mean diameter of the bowl was 5.2495 in. and that of the ring was 5.2429 in., giving a diametral clearance of 0.0066 in. when the ring and the bowl were at the same temperature. This clearance changed if the ring and bowl had different temperatures. It was also affected by the expansion of the bowl due to centrifugal force, and corrections for these changes had to be made to suit the conditions obtaining at the time of the observations.

The bowl was a loose fit on the driving spindle, which permitted it to center itself at running speeds.

The rod suspending the ring was 0.125 in. in diam and 19.5 in. long. It possessed but slight lateral stiffness and was therefore able to center itself with respect to the bowl when the latter was running. The attachment at the upper end was constructed so as to permit the ring to be approximately centered in the bowl while the latter was at rest. Thus, only a small amount of bending of the suspension rod was required to obtain the necessary degree of centering.

The temperature of the ring was determined by means of thermocouples placed in its wall, as shown in Fig. 2.

The temperature of the bowl was more difficult to determine, and two methods were tried. In the first, a hole was drilled in the edge of the bowl, as shown at A in Fig. 3. This hole was filled with oil and plugged. The apparatus was operated at a desired speed until conditions became constant and then stopped. The plug was removed and a thermocouple junction was placed in the oil. Several readings were taken, and the times at which the observations were made were noted. A plot was then made of temperature against time, with the time the bowl stopped taken as zero. The temperature of the bowl at the instant it

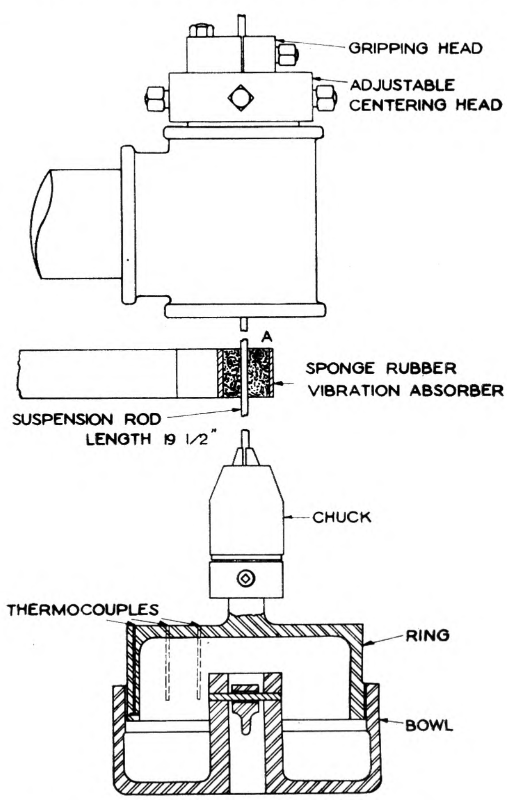

Fig. 1 Assembly View of Test Apparatus

[1] Professor of Machine Design, The Pennsylvania State College.
[2] Research Assistant, The Pennsylvania State College.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

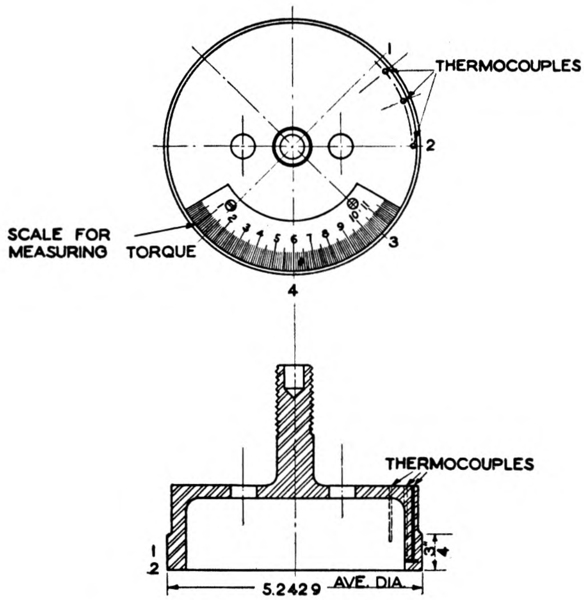

Fig. 2 Details of Ring

Radial position	Axial positions 1	2
1	5.2430	5.2429
2	5.2429	5.2429
3	5.2429	5.2429
4	5.2428	5.2429

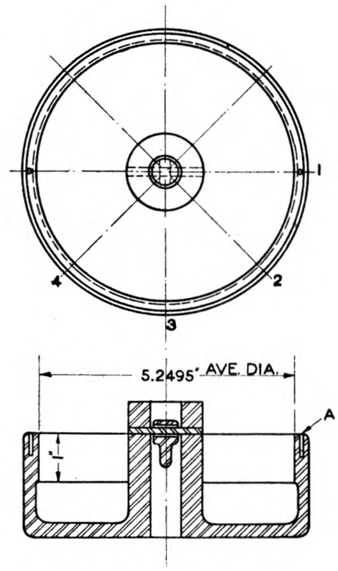

Fig. 3 Details of Bowl

Radial position	Axial positions 1	2
1	5.2494	5.2497
2	5.2494	5.2494
3	5.2495	5.2494
4	5.2494	5.2497

stopped was determined by extrapolating the curve to zero time.

This method was not found to be practicable because only 10 or 12 sec were required for the bowl to reach the temperature of the ring. This was usually too short a time to permit obtaining a sufficient number of readings to determine accurately the form of the time-temperature curve.

In the second method, a thermocouple junction was pressed lightly against the bowl and the indicated temperature observed. Since the rubbing of the junction against the bowl produced

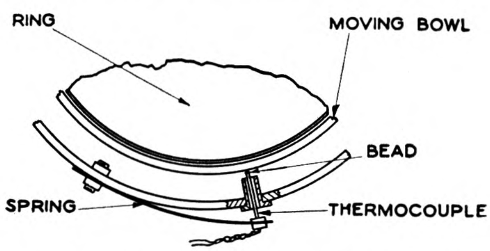

Fig. 4 Detail of Thermocouple on Bowl

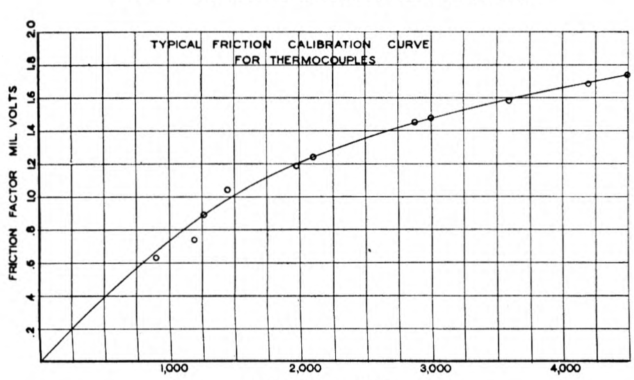

Fig. 5 Typical Friction Calibration Curve for Thermocouples

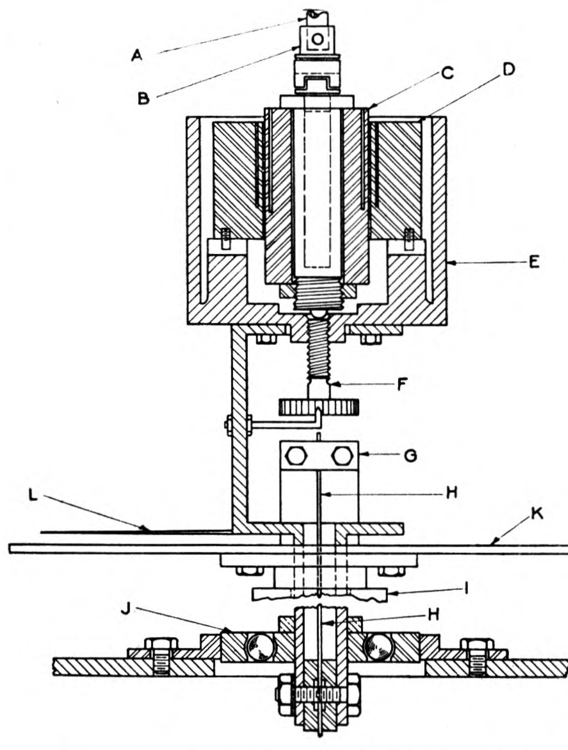

Fig. 6 Arrangement of Tapered-Plug Viscosimeter

A—Drive from drill press
B—Oldham coupling
C—Rotating plug
D—Stationary ring
E—Oil cup
F—Adjusting screw
G—Stationary grip
H—Torsion rod
I—Upper steady bearing
J—Lower steady bearing
K—Drill-press table
L—Pointer indicating angle of twist

friction which resulted in temperature, precautions had to be taken to keep this low and uniform, and also to determine its amount. The thermocouples were mounted as shown in Fig. 4, and were pressed against the bowl by means of a light flat spring. They were so arranged as to be out of contact with the bowl except when readings were desired. This was done to prevent wear on both bowl and junction. Just enough pressure was used to keep the junction in contact with the bowl at all times.

In order to determine the amount of temperature rise due to friction the bowl was brought to room temperature. A run was made with the ring removed and the temperature indication at various speeds noted. A plot was then made of temperature against speed. Fig. 5 is typical of these plots. One of these calibration runs was made before each test run to determine the torque-speed relationship. The temperature rise due to friction, taken from this curve, was then subtracted from the temperature indicated by the thermocouple during the torque-speed run. The difference was taken to be the temperature of the outer surface of the bowl.

A rough check was made to determine the drop in temperature between the inner and outer surfaces of the bowl. As this indicated a drop of only about 1 deg, no attempt was made to correct for it.

Preliminary runs showed that severe lateral vibration of the rod suspending the ring occurred at each of several critical speeds. It was overcome by enclosing the rod in a pad of sponge rubber which damped out the vibrations but was sufficiently yielding to permit the ring to center itself without bending the rod unduly. This damping pad is shown at A in Fig. 1.

The second piece of apparatus consisted of the well-known tapered-plug viscosimeter, built from descriptions given by Albert Kingsbury.[3] Only slight departures were made from his specifications. The most important were (1) the introduction of an Oldham coupling in the drive in order to permit the plug to center itself with respect to the bearing, and (2) the introduction of a certain amount of freedom in the mounting of the bearing within the oil-container cup, in order to make possible the centering of the bearing over the suspension rod. Fig. 6 shows the

[3] "Heat Effects in Lubricating Film," by Albert Kingsbury, *Mechanical Engineering*, vol. 55, Nov., 1933, pp. 685–688.

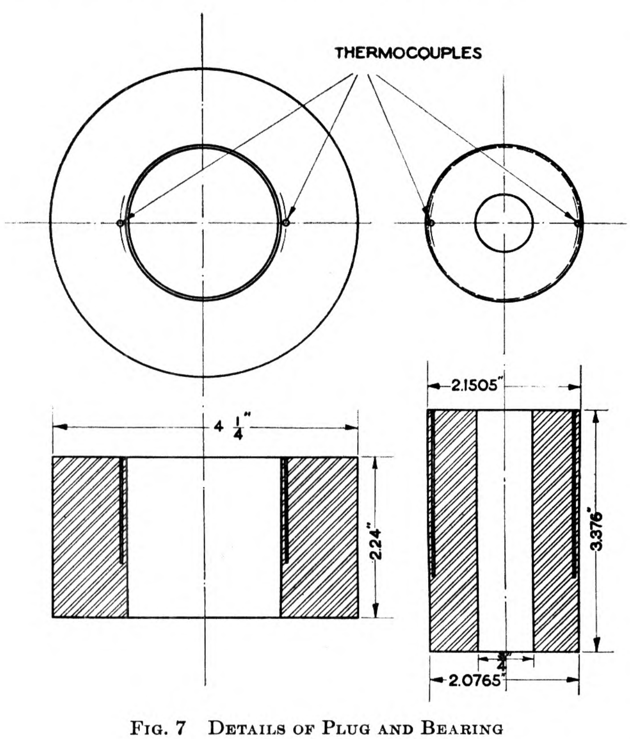

FIG. 7 DETAILS OF PLUG AND BEARING

arrangement. The dimensions of the plug and bearing are shown in Fig. 7.

Two thermocouple wells were drilled into the bearing, as shown, as close to the bearing surface as possible. The thickness of metal separating the bearing surface from the thermocouple well was about $^1/_{32}$ in. Two wells were similarly placed in the plug. These also are shown in Fig. 7.

Test Procedure

When using the rotating-bowl apparatus, the ring was first centered within the bowl while the latter was at rest. Oil, heated to about 180 F, was then supplied to the bowl by a gravity feed, and the bowl was brought up to the desired speed. The oil supply was then adjusted so as to be just sufficient to make good the quantity thrown out through the clearance space. The speed was held constant until the thermocouples indicated that constant temperature conditions had been reached. This required about 30 min. The angle of twist of the suspension rod and the temperature of each thermocouple were then read.

When using the tapered-plug viscosimeter, the oil was placed in the cup holding the bearing and raised to a level slightly above the top of the plug, care being taken to see that the clearance space between the plug and the bearing contained no air. The supporting micrometer screw at the bottom of the cup was then adjusted to give the clearance at which it was desired to operate. The plug was then rotated at constant speed until constant torque and temperature conditions were reached. The angle of twist of the suspension rod and the temperatures of the thermocouples in the bearing were observed, after which the plug was stopped and the temperatures of the thermocouples in it were noted.

The mean temperature of the oil film was taken as the mean between the temperatures of the ring and the bowl in the case of the first apparatus and the mean between those of the bearing and plug in the case of the tapered-plug viscosimeter.

The tapered-plug apparatus was used for rates of shear between 1330 and 239,000 reciprocal sec. The ring-and-bowl apparatus was used for rates of shear between 50,000 and 320,000 reciprocal sec.

Five kinds of oil were used. These are listed together with their viscosities in Table 1. It will be noted that this list includes mineral oils from several geographical fields and also one vegetable oil, namely, olive oil.

TABLE 1 VISCOSITIES OF OILS USED

Oil	Absolute viscosity —centipoises— 130 F	210 F
Havoline 10 W	14.8	4.15
Olive	20.4	7.05
Quaker State 10 W	15.2	4.33
Sunoco 10 W	10.6	3.13
Texaco 10 W	15.1	4.14

Results of Tests

Petroff's equation states that the torque required to rotate a journal which is concentric with its bearing is directly proportional to the product of the absolute viscosity and the rate of shear of the fluid separating the journal and bearing. The plot of torque against rate of shear should, therefore, be a stright line passing through the origin, if the viscosity is constant. Otherwise, it will be a curve of some sort, depending upon the manner in which the viscosity varies. This relationship was used as a convenient means of detecting any variation in viscosity which might be caused by variations in the rate of shear.

The values of torque, observed on each of the machines described, were corrected for temperature and original viscosity differences to one arbitrarily chosen value. The resulting values were then plotted against the corresponding values of the rate of

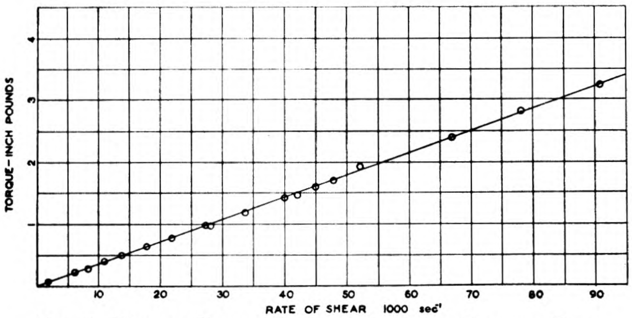

FIG. 8 TORQUE VERSUS RATE OF SHEAR ON TAPERED PLUG
(Oil used: Havoline 10 W.)

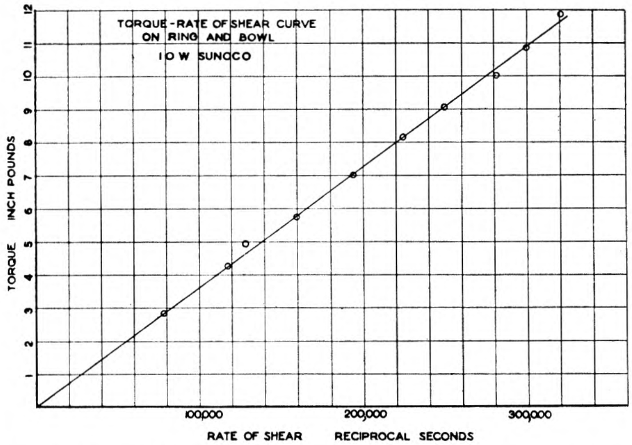

FIG. 9 TORQUE VERSUS RATE-OF-SHEAR CURVE ON RING AND BOWL
(Oil used: Sunoco 10 W.)

shear. Any variation in viscosity, other than that due to temperature, would cause a departure of these points from the line representing Petroff's equation. Fig. 10 shows the results obtained. There is no consistent deviation from the Petroff line

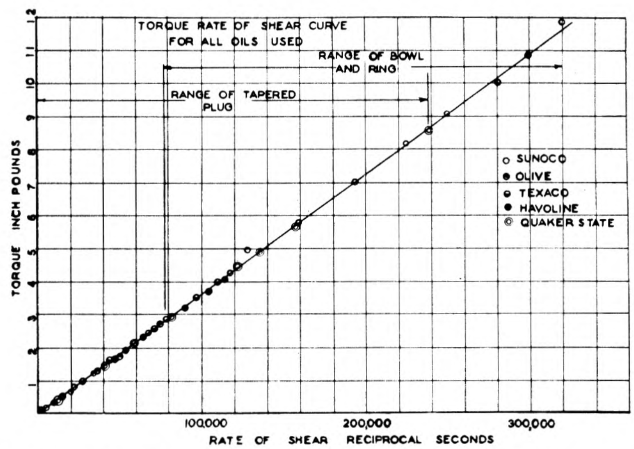

FIG. 10 TORQUE VERSUS RATE-OF-SHEAR CURVE FOR ALL OILS USED

at any rate of shear throughout the range investigated. Furthermore, there is no departure of any single observation to an extent greater than can be attributed to experimental errors normally present in work of this kind. This is true of the observations made with both machines, and the results obtained from each are in complete accord in the portion of the field covered jointly. The agreement of the results with those predicted by the Petroff equation, upon the assumption of the independence of viscosity from the rate of shear, holds for all of the oils investigated.

From these results it would appear that, if there is a change in viscosity due to molecular orientation, it occurs in a portion of the field not investigated by the authors. The most likely portion is the region of very low rates of shear and close proximity of the boundary surfaces. This portion should command our next attention.

Discussion

M. D. HERSEY.[4] Can the authors supply information as to the clearances used with the tapered plug, and the range of film viscosities or film temperatures covered in their investigation?

A distinction may be made between Newton's law and Petroff's in that the former requires only proportionality, while the latter involves the calculation of a constant in terms of clearance and length. Should we consider that both laws have been verified and, if so, within what estimated limits of accuracy?

Kingsbury's investigation[3] indicated an approximately parabolic distribution of temperature over the cross section of the film. The temperature drop from the middle of the film to either metal surface in the ring and bowl may be calculated[5] on the assumption of radial conduction under steady-state conditions. For a viscosity of 4.5 cp (Table 1 of the paper) and conductivity 0.016 lb per sec deg F at 210 F, this temperature difference approximates 5.6 F at the maximum rate of shear, 320,000 reciprocal sec. Are such effects negligible within the limits of accuracy of the present work?

AUTHORS' CLOSURE

The tests reported in the paper were undertaken to determine whether the viscosity of the oils investigated varied with the rate of shear. Petroff's equation was made use of because it offered a ready means of detecting variations of viscosity with rate of shear. The authors accepted the statement, "Thus it appears for any given bearing, the friction torque is proportional to the viscosity of the lubricant and the speed," which appears on page 26 of Mr. Hersey's book "Theory of Lubrication."[5] Their claim is that the results secured support the conclusion that viscosity does not vary within the limits of the investigation.

It is felt that since the exact thermal conductivity of the oils treated was not available, calculation of the effect of temperature gradient across the thickness of the film would not add to the accuracy of the results.

[4] Research Director, Morgan Construction Company, Worcester, Mass. Fellow A.S.M.E.

[5] "Theory of Lubrication," by M. D. Hersey, John Wiley and Sons, Inc., New York, N. Y., 1936, pp. 116–117.

Effect of Temperature on Coiled Steel Springs Under Various Loadings

By F. P. ZIMMERLI,[1] DETROIT, MICH.

This paper consists of a presentation of the results of tests conducted in the laboratories of the author's company on the effect of various stress-temperature combinations on steel springs. These results are in the form of charts which, because of the immense number of tests involved, are believed to be accurate. For the carbon steels tested, it is shown that there is a temperature-stress equilibrium point about 400 F. Below this point there is a definite temperature-stress equilibrium. Above this point it is simply a time-temperature curve since, eventually, the springs will fail. The paper draws particular attention to the value of various strain-relief heatings after coiling.

Tests on springs hardened and tempered after coiling, as compared with those made of pretempered wire, give no evidence that the former method is to be preferred. This is contrary to the general understanding in the industry.

The paper shows that for each type of material there is an optimum Rockwell hardness for best heat resistance. Both S.A.E. 6150 and 9260 steels have greater resistance to load losses due to heating than carbon steels. They in turn are exceeded by 18-8 stainless and high-speed steels. A difference in time between 10 days and 3 days to reach true temperature-stress equilibrium exists between the various steels.

EVERY mechanical device, somewhere within itself or in its production, calls for the use of springs. During the last few years, in particular with the demand for greater operating speeds, the temperatures at which these springs function have continually increased. Data regarding the temperature-stress relationship of springs have as yet no general publication, and appreciation among engineers of the results which may be attained has lagged.

To remedy the condition, the laboratories of the author's company about 10 years ago commenced the task of testing all available spring materials in this respect. It soon became apparent that the amount of work involved would be immense and that too long a time would be required for completion of the original program.

To the end of speeding up the work, the International Nickel Company agreed to investigate its products, such as monel, inconel, Z nickel, and the like, in this regard. Preliminary tests on copper alloys, such as brass and bronze, proved that these materials were useless above 225 F and they were eliminated, leaving the scope of the project to cover coiled steel springs.

Before tests could be conducted for the purpose of obtaining the necessary design information, it was essential to know: (*a*) How long a time was necessary for a loaded spring to be held at a temperature in order to reach equilibrium;[2] (*b*) the effect of the degree of flexibility in the spring; (*c*) whether a spring should be pressed solid before or after the heating operation which removes coiling strains; and (*d*) whether or not the process of wire manufacture influenced the results to any great extent, i.e., would wire, built to the same specification by competing manufacturers, act the same?

Preliminary Tests

For these preliminary tests springs were coiled from pretempered material, ground, and heated to 800 F. They were then pressed solid with a load 100 lb in excess of their carrying capacity. The length, outside diameter, load, and wire size of each spring were noted. The springs in sets of 10 were placed over bolts which pushed steel collars against them and thus compressed the spring to predetermined lengths, checked with micrometers. The springs were then exposed to the desired temperature for different periods of time. They were then removed and checked again for load carried. The load testing was carefully performed, the springs being checked on the scales to within 0.001 in. in height, using a 0.001 gage, so that no error due to scale travel was possible.

Stresses were calculated before and after heating, using the Wahl formula

$$S = \frac{8PD}{\pi d^3}\left(\frac{4c-1}{4c-4} + \frac{0.615}{c}\right)$$

where $c = D/d$
P = load, lb
D = mean diameter, in.
d = wire diameter, in.
S = stress in outer fiber, psi

All stresses are those due to loads at room temperature and are not the stress on the wire at oven temperature. If it is desired, a correction could be made for this by obtaining the modulus at the various temperatures employed. This value was reported[3] by W. P. Wood, G. D. Wilson, and the author in 1930.

If G = modulus at room temperature, G_1 at oven temperature, S the stress at room temperature, then S_1, stress under oven conditions, has the relation

$$\frac{S}{S_1} = \frac{G}{G_1} \quad \text{or} \quad S_1 = \frac{SG_1}{G}$$

This can be proved from the formula of spring deflection

$$f = \frac{8PD^3N}{Gd^4}$$

where N = number of coils, and the other symbols are as previously stated.

The springs were not corrected for bolt expansion because the

[1] Chief Engineer, Barnes-Gibson-Raymond Division, Associated Spring Corporation. Mem. A.S.M.E.

Contributed by the Special Research Committee on Mechanical Springs, and presented at the Annual Meeting, New York, N. Y., December 2–6, 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] By "equilibrium" is meant a stable condition such that further exposure to a given temperature will not cause any additional loss in load carried by the spring.

[3] "The Effect of Temperature Upon the Torsional Modulus of Spring Materials," by W. P. Wood, G. D. Wilson, and F. P. Zimmerli, Proceedings of the A.S.T.M., vol. 30, 1930, part 2, pp. 351–360.

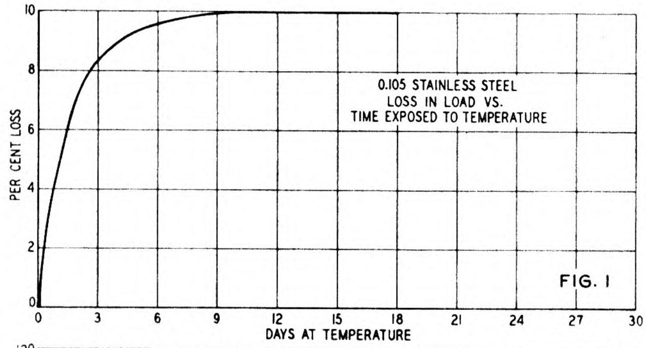

FIG. 1

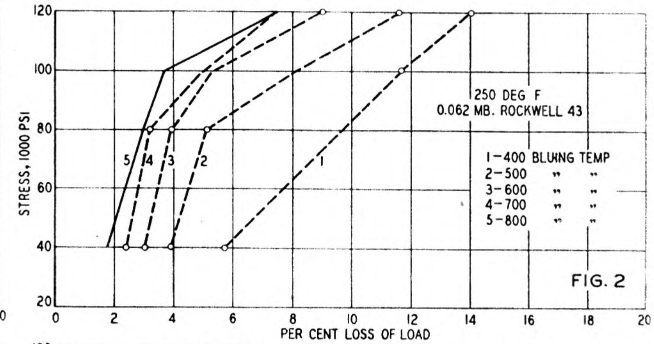

FIG. 2

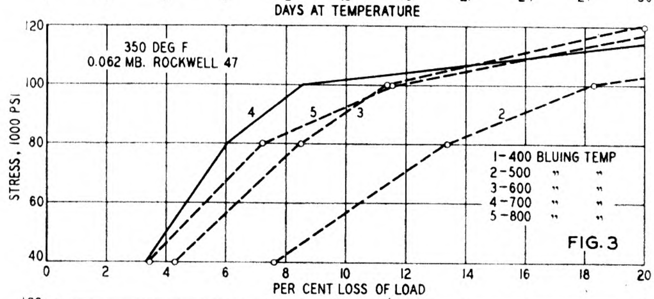

FIG. 3

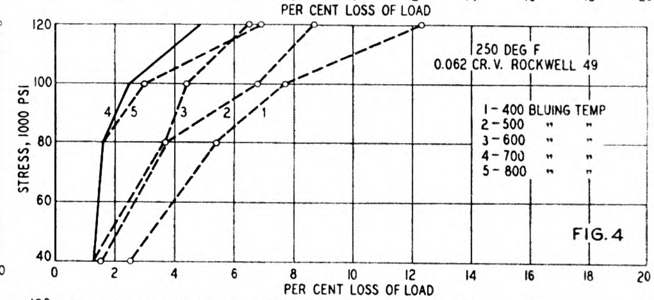

FIG. 4

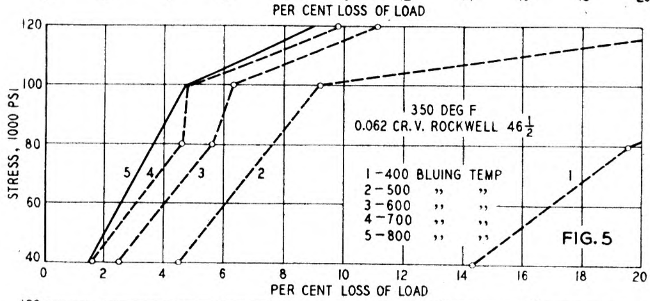

FIG. 5

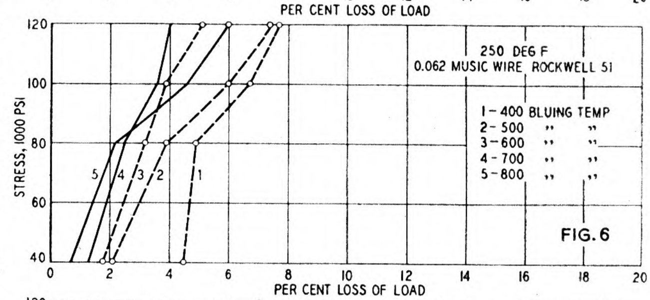

FIG. 6

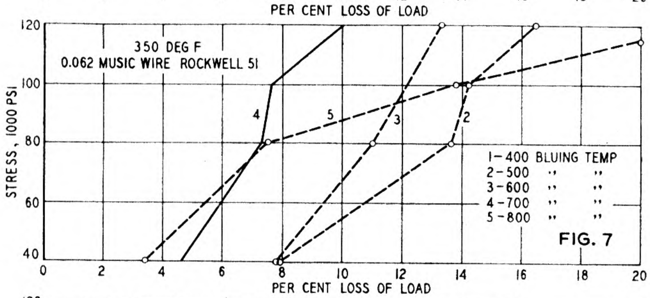

FIG. 7

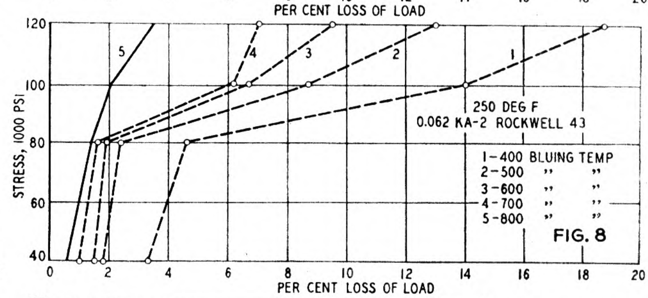

FIG. 8

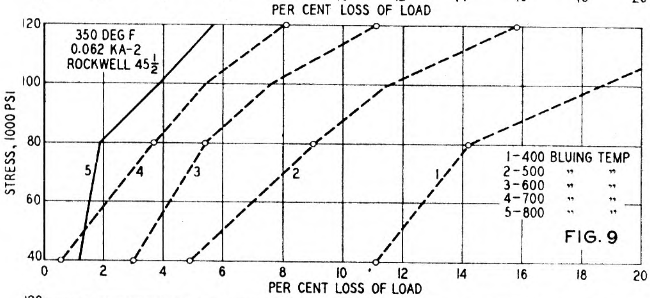

FIG. 9

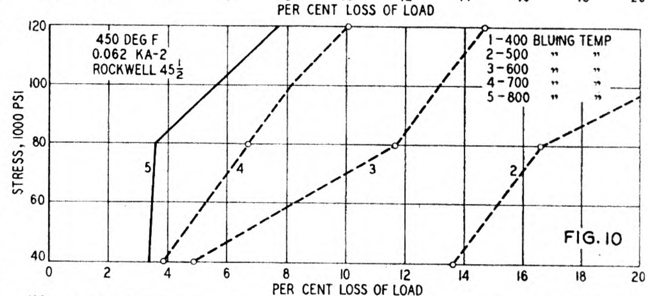

FIG. 10

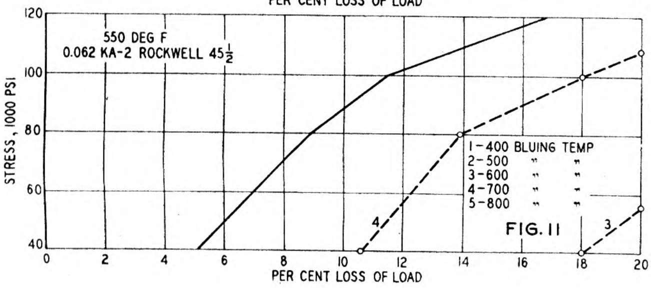

FIG. 11

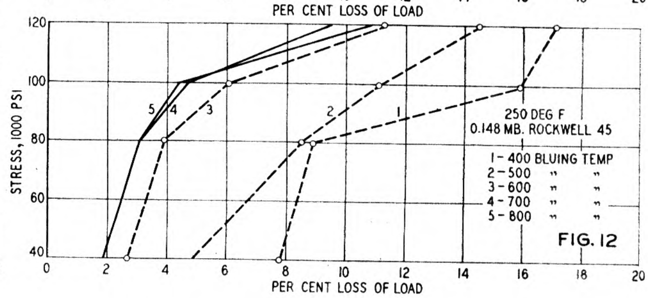

FIG. 12

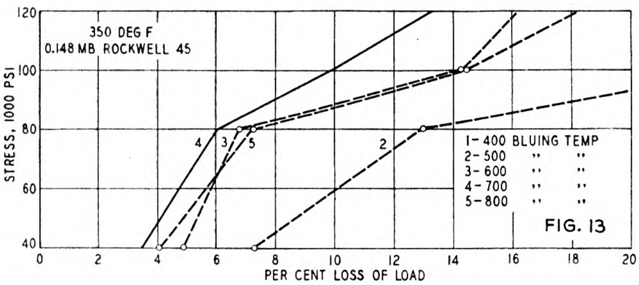

FIG. 13

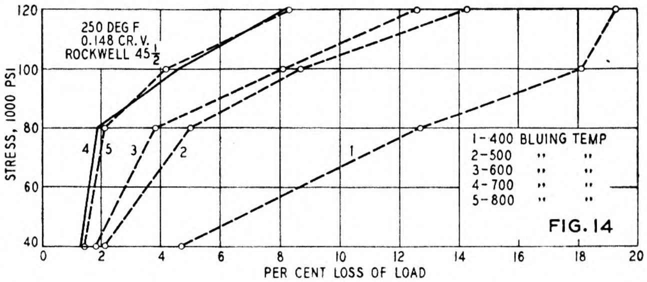

FIG. 14

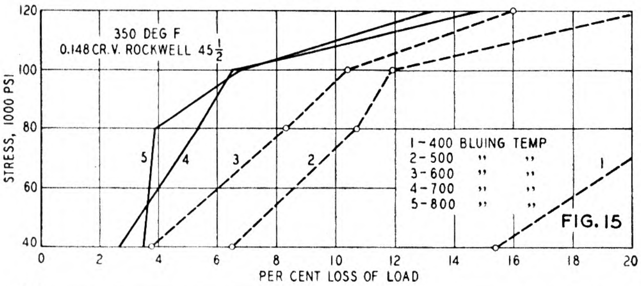

FIG. 15

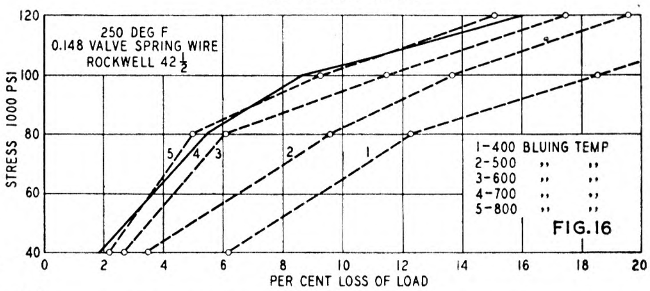

FIG. 16

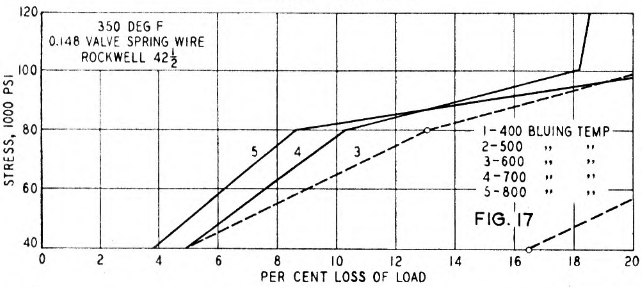

FIG. 17

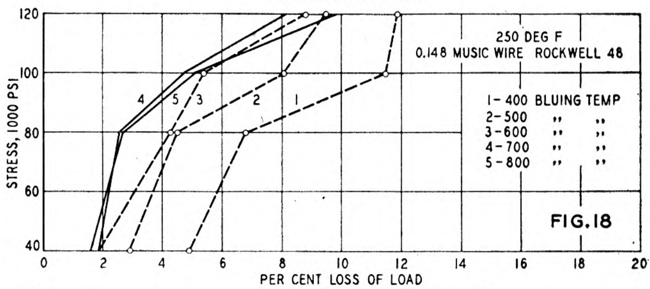

FIG. 18

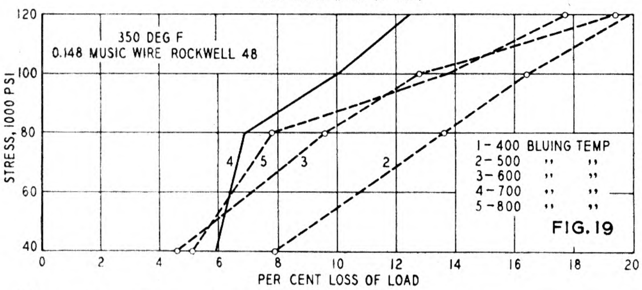

FIG. 19

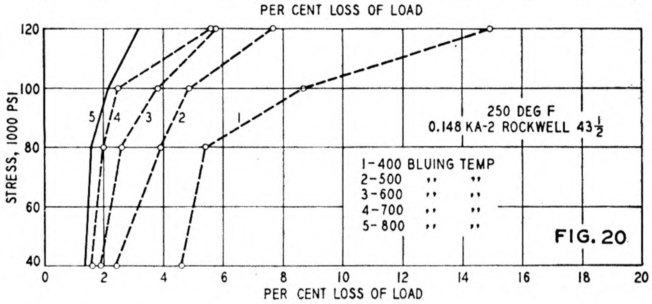

FIG. 20

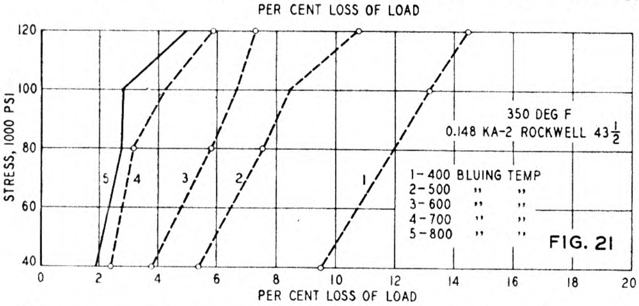

FIG. 21

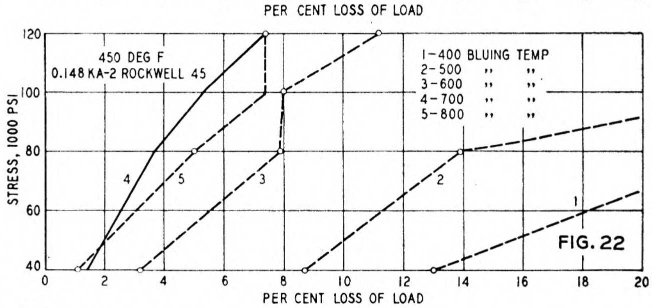

FIG. 22

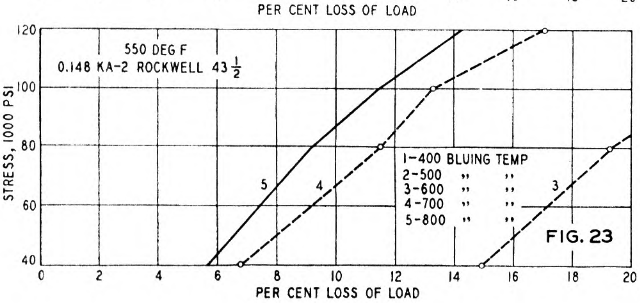

FIG. 23

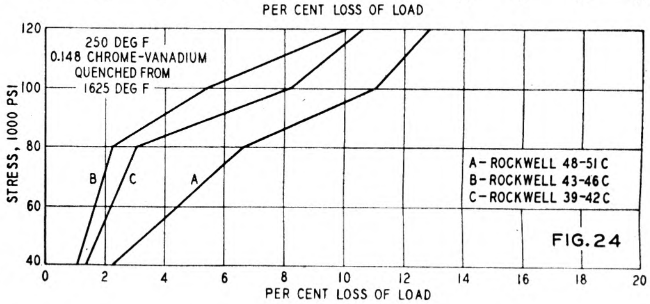

FIG. 24

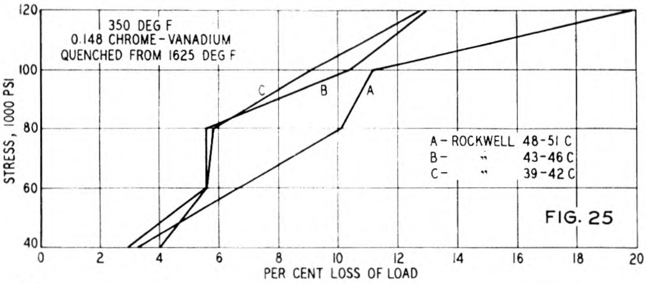

FIG. 25

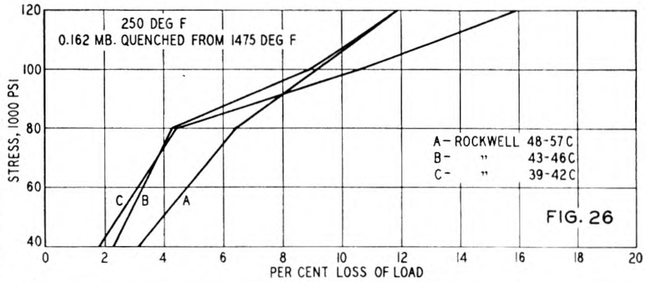

FIG. 26

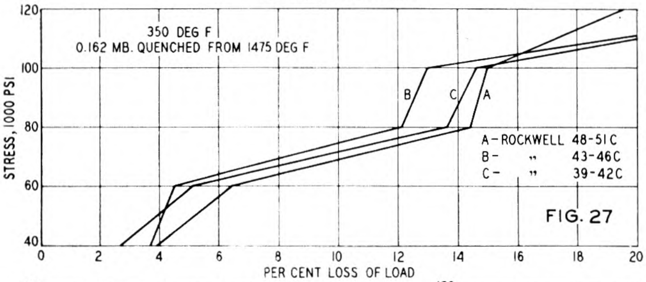

FIG. 27

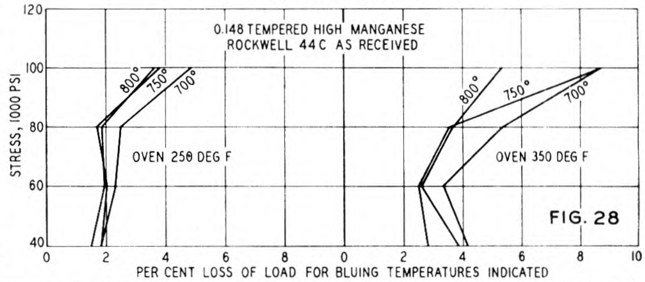

FIG. 28

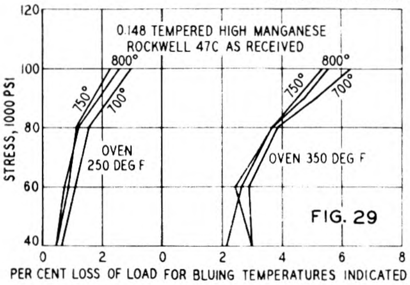

FIG. 29

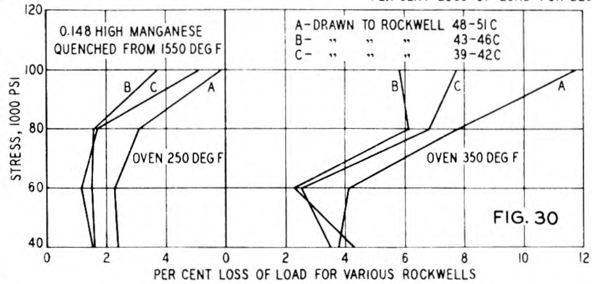

FIG. 30

CURVE	DRAWING TEMP	ROCKWELL "C"	OVEN TEMP
A	1150° F	57-59	600° F
B	1250	52-52.5	600
C	1150	57-59	700
D	1250	52-52.5	700

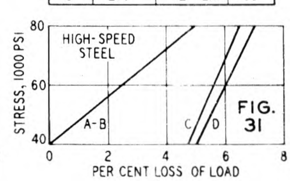

FIG. 31

wire in the spring actually expanded slightly to cover this error.

Lots of ten springs were set at a given stress and temperature and run for successive periods until no further loss of load was noted. Then another combination of temperature and stress was similarly tested until the entire field of possibility had been covered. These results indicated that the usual spring steels should be divided into two major groups, i.e. (1) straight carbon steel, low-alloy steels, and (2) high-alloy steels.

The first group consisted of S.A.E. 1065, X1065, 1080, 1095, 6150, and 9260. The second group was composed of 18-8 stainless steel and the usual 18-4 high-speed steel. Group 1 definitely reached equilibrium within 72 hr at heat, regardless of stress or temperature, provided the latter was less than 400 F. Tests up to 216 hr gave the same results as 72 hr, while 60 hr were extremely close, losses being but small fractions of 1 per cent of the total. Above 400 F, there was no time at which equilibrium was reached until the coils of the spring, when released, were still closed against each other. This is quite similar to the behavior of steel in tension at elevated temperatures, where it has been noted that, up to the equicohesive temperature, creep is observed for a definite time. It is postulated that the metal becomes strain-hardened at that time and resists further deformation.

The two samples in the second group were tested in a similar manner and equilibrium was reached not in 72 hr but in 10 days. A curve is appended which gives results of 18-8 stainless steel subject to 250 F at 120,000 psi for a period up to 18 days. This material will not reach equilibrium if the temperature greatly exceeds 550 F, but the set is slow so that for some uses it is satisfactory at slightly higher temperatures. High-speed steel is similar, except that higher temperatures are possible. The later discussion of commercial production tests will amplify these statements.

The next point to be considered was the rate of deflection of the spring itself. To test this, twenty springs were made from the same bundle of music wire with the same outside diameter, in the manner previously outlined. Ten had $5^1/_4$ coils and ten had $10^1/_4$ coils. When stressed at 40,000 psi, the following result was obtained after 72 hr:

$10^1/_4$ coils........................	loss 2	per cent
$5^1/_4$ coils........................	loss 1.6	per cent

A further test at 100,000 psi was made with the following results:

$10^1/_4$ coils........................	loss 9.3	per cent
$5^1/_4$ coils........................	loss 10.4	per cent

It was concluded that the degree of flexibility of the spring was not a factor governing loss of load due to temperature.

Two hundred springs were divided into two lots of 100 each. These springs were made of oil-tempered wire. One lot was pressed solid before the bluing or strain-relieving draw of 750 F, and the other lot was pressed after the heating. The springs were subject to various stresses at temperatures up to 400 F. Results showed that springs pressed after heating were more uniform and slightly better. Therefore, this procedure was adopted as standard.

The final preliminary tests were conducted on springs made from wire manufactured by four different mills. Slight differences were noted, the products of some mills showing a greater loss than others. However, in wires other than the hard-drawn quality this difference was not excessive. Hard-drawn wire varied greatly. Evidently, each manufacturer patents his wire differently (if at all). The results were so confused that it was decided not to run the production tests on this cheap type of material.

Procedure for Production Tests

The procedure in running the remaining tests, the results of which are shown in the accompanying curves, was as follows: A sufficient number of springs of each of the materials to be tested

TABLE 1 CHEMICAL COMPOSITION OF MATERIALS USED IN TESTS

Material	C	Mn	P	S	Cr	V	Ni
0.148 Music wire.	0.91	0.31	0.018	0.022	....	....	....
0.148 M B.......	0.66	0.76	0.020	0.036	....	....	....
0.148 Cr V.......	0.52	0.75	0.007	0.020	0.87	0.18	....
0.148 Swedish valve spring...	0.65	0.56	0.021	0.019	....	....	....
0.148 KA-2......	0.24	0.42	.....	.....	18.2	....	9.21
0.062 Music wire.	0.91	0.31	0.024	0.018	....	....	....
0.062 M B.......	0.59	0.75	0.020	0.025	....	....	....
0.062 Cr V......	0.50	0.73	0.009	0.018	0.97	0.18	....
0.062 KA-2......	0.12	0.41	.....	.....	19.2	....	9.14
0.148 High Mn (tempered)....	0.70	1.34	0.022	0.024	....	....	....
0.148 Cr V (annealed)....	0.54	0.69	0.011	0.026	0.89	0.17	....
0.162 M B (annealed)....	0.62	0.73	0.018	0.055	....	....	....
0.148 High Mn (annealed)....	0.70	1.34	0.022	0.024	....	....	....
0.152 High-speed steel..........	0.76	0.31	(tungsten 18.03)		3.83	1.10	....

TABLE 2 ROCKWELL C HARDNESS OF MATERIALS USED IN TESTS AFTER HEATING TO

Material	400 F	500 F	600 F	700 F	800 F
0.148 Music wire..........	52	51	52	48	45.5
0.148 M B..............	45	46	47.5	45	42
0.148 Cr V..............	47	46	46	45.5	44.5
0.148 Swedish valve spring	43	43	43	42.5	43
0.148 KA-2..............	44.5	45.5	45	45	43.5
0.105 Music wire..........	52.5	53	51	51	48
0.062 Music wire..........	52	52	52.5	51	47.5
0.062 M B..............	49	50	50	47	43
0.062 Cr V..............	49	49	49.5	49	46.5
0.062 KA-2..............	45	45	45	46	45.5

Material	As received	700 F	750 F	800 F
0.148 High Mn...........	46	46	45	44
0.148 High Mn...........	44	44	44	43
0.152 High-speed steel....		(See data sheet)		

were obtained, about 120,000 pieces in all being required. These springs were heated to the various strain-relieving temperatures, indicated on the charts, for 30 min at heat, in an L & N Homo furnace. Ten springs were used for each stress-temperature test and the average loss plotted on the curves. All tests, except on stainless steel and high-speed steel were run 72 hr at heat. These high-alloy steels were given 10 days to be sure an equilibrium condition was obtained. All springs were pressed solid after heating for stress-relief. This was done with a load 100 lb greater than that necessary to close the coils. It was hoped that this procedure would aid in establishing uniformity of tests.

The springs were tested on accurately checked scales using a 0.001-in. gage, in order to obtain the desired load and height on each spring for a given stress. The springs were placed in a constant-temperature electric oven, being held on special bolts, with square seating collars, to the desired height and load. Upon removal from the oven, the springs were air-cooled and retested for load at room temperature to the nearest 0.001 in. and the results expressed as percentage loss of load corresponding to the given stress were plotted.

In working with these various materials, the data given are to the highest possible commercial application of the product. Thus, some curves indicate temperatures only as high as 350 F. Tests at 400 F on these same steels demonstrated that the steels were erratic, hence, not of commercial importance. Therefore, no data sufficiently accurate for any design problems could be presented. At 450 F, these selfsame steels simply collapsed when given sufficient time.

The effect of wire size in the limits tested is not great. Curves are presented in the interests of economy on only two sizes, i.e., $^1/_{16}$ and 0.148 in. Data are not available on any size larger than $^3/_{16}$ in. In general, the tendency is toward greater load loss with larger sizes at the same stress. The curves are as plotted from points taken at a minimum of four different stress figures. To reduce possible error, each of these points is the average value of ten springs.

Even with all these precautions it will be noted that the data give results which at some temperatures cause the curves of various bluing temperatures to cross each other. These points have been checked in some cases and the same result obtained. At present no good explanation has been developed for this experimental fact.

The materials used in these tests are listed in Table 1 and the hardness figures in Table 2.

Conclusions

In the author's opinion, a study of the work done justifies the following conclusions:

1 The usual spring steels are reliable when stressed 80,000 psi or less up to temperatures of 350 F. Between 350 F and 400 F and, at stresses up to 120,000 psi, the same continuity of results is lacking, but with proper forethought some commercial success might be expected.

2 The use of ordinary spring steels over 400 F is not possible.

3 Steels, hardened and tempered after coiling into springs, at the same hardness value, have no advantage over springs made of pretempered wire properly blued, under the conditions investigated.

4 Stainless steel of the 18-8 type resists temperature and stress better than other spring steels, except perhaps high-speed steel.

5 A middle hardness range in quenched-and-drawn springs is preferable to either high or low ranges.

6 An optimum temperature to heat springs after coiling for heat resistance is the highest one which will not render the hardness or other physical properties of the material objectionable.

7 The present Swedish valve-spring wire stands heat very poorly, in fact, is less satisfactory than many other steels.

8 Both high-manganese and silicomanganese steels equal the chrome-vanadium steel tested and may have commercial advantages.

Acknowledgments

The author wishes to acknowledge the assistance of the entire laboratory force in preparing this work; in particular A. C. Stenhouse, now with Vauxhall in England, G. D. Wilson, and Glen Brookes.

It can be readily appreciated that this is the work of many when it is realized that over 250,000 separate spring weighings were made in addition to other testing work.

It is to be hoped that other investigators will carry on the work into the field of larger wire sizes. The possibility of different heats and heat-treatments of high-speed steel and stainless steel should be completed. In particular, the investigation of the newer steels now in use and which we have not tested, such as the high-chromium series and chromium-molybdenum types, should be started if complete data are desirable.

Discussion

R. C. Zeidler.[4] As a user of great quantities of springs, the company with which the writer is connected has experienced its share of problems. In some small measure the experience gained during the last year and a half may contribute to the general knowledge of the subject.

Our larger springs are used as clutch-pressure springs. While they are subject to high temperatures, and design limitations impose high stresses, they are under only static loads. Consequently, we are not interested in fatigue life but only in springs which show a minimum of load loss under hard clutch-operating conditions. For practical and economical reasons, all of our

[4] Assistant Engineering Manager, Long Manufacturing Division, Borg-Warner Corporation, Detroit, Mich.

pressure springs are of S.A.E. 1065 oil-tempered or similar steels, covered by the author's MB specification. The following discussion pertains to this class of steel.

Our method of testing is similar to that used by the author, except that the springs are pressed twice to 300 lb, which is $1^1/_2$ to 2 times the normal load of the springs, before final selection for load is made. This is done to offset variations in load caused by failure of the vendor to remove most of the set and by poor packing and rough handling in shipping. Thus, springs supplied by various vendors are all placed on an equal test basis. Any further set therefore is due to the heat alone.

The data given in this discussion are for springs with stresses ranging from 75,000 to 106,000 psi, corrected according to Wahl formula and Rockwell hardness 43–46 C scale. Through a series of tests it was found that, after 15 hr in the oven at 350 F, the springs had lost approximately 90 per cent of the total load they would lose if allowed to remain in the oven until the equilibrium point was reached. This was found to be sufficient for all practical purposes so that 15 hr has been adopted as our standard, since it permits a ready overnight check. Preliminary tests showed that these springs are not practical above 400 F, with the stress above 75,000 psi. Some springs tested showed 8 to 10 times more loss between 400 and 500 F than between 300 and 400 F, but only 1.5 times more between 300 and 400 F than between 200 and 300 F.

In ordinary clutch service, these springs probably would never reach a temperature of 400 F but, in order to check this point, a number of clutches returned from actual field service were examined. Some of these appeared to have had normal usage, while others evidently had been abused. Other clutches of known qualities were given severe road tests. From the average load losses of the springs, the probable temperatures reached were found to be in the 200 to 300 F range. As a consequence, it was decided that, for an accelerated test, 350 F would be adopted as a standard. At this temperature, springs still react somewhat along theoretical lines.

Together with numerous other tests, twenty-five different types of pressure springs, as received from various vendors, were all given a 20-min treatment at 750 to 775 F and then pressed twice under 300 lb. They were then carefully selected for load, clamped between test plates at their respective working heights, and placed in the oven at 350 F for 15 hr.

Of the twenty-five springs tested, seventeen came within the stress range of 75,000 to 96,000 psi. The maximum load loss on the springs of this group was $5^1/_2$ per cent. The other eight springs, ranging from 96,000 to 106,000 psi, showed a gradual increase to 9.5 per cent. This does not mean that all of the springs fell exactly on a straight line between these points but that an approximate average was obtained. We found several unexplainable discrepancies, such as the author mentioned in his paper.

As the result of the foregoing tests, the design drawings for these springs now call for a maximum allowable load loss due to heat. Routine checks are made daily on six springs from each production shipment. These are placed in the oven the day they are received and allowed to remain for 15 hr overnight, thus making them available for production the following day if they are satisfactory.

While we feel that only a beginning has been made from our findings up to the present time, we believe that an ordinary grade of oil-tempered wire, in a spring of reasonable design, given the proper manufacturing attention, will provide a clutch spring which will be satisfactory for all practical purposes. Further, we believe that still greater improvement can be made without the use of more expensive steels.

Author's Closure

Mr. Zeidler's remarks are very interesting and timely. The results, assuming his mean stress to be 85,000 lb per sq in. for the first 17 springs, check our results within 1 per cent and are lower due to the time interval he used. On the 8 springs whose mean stress is 100,000 lb per sq in., curve 13 gives a 10 per cent loss in load. This is within $^1/_2$ per cent of Mr. Zeidler's figures. We consider this an excellent check on the utility and accuracy of our work. We are extremely glad that Mr. Zeidler has come forward with this information.