Modern Boiler Furnaces

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The paper considers the fusing temperatures of various kinds of coal ash and slag, and their significance with respect to furnace temperatures, operation, and design.

A method of comparing furnace operating results is suggested consisting of (1) taking accurate gas temperatures throughout a furnace, (2) then calculating from them the rates of heat absorption in different zones, (3) calculating an "equivalent heat-receiving-surface temperature" for each zone, (4) taking samples of ash and slag from different portions of the furnace and determining the fusing-temperature range from such samples, (5) plotting these data as a "gas-ash temperature graph" against heat-absorbing surface, (6) comparing the data as shown on such graph with observed behavior of ash and slag in the furnace, and (7) comparing different furnaces, different coals, and methods of operation through the use of such graphs. Examples of the use of the method are given.

There is also included a comparison of methods for measuring furnace temperatures. Colored motion pictures illustrating some phenomena relating to slag in furnaces were shown at the time of presentation of the paper.

ASH AND SLAG from the burning of coal have long been recognized as important factors in boiler-furnace operation. Hand-fired boilers were often limited in output by clinkers in the fuel bed, with resulting excess burden on the firemen. Certain kinds of stokers were ill-adapted to handling clinkering coals, while others were developed to cope with the same coal much better. Forced-draft stokers extended the clinker trouble, formerly restricted largely to the fuel bed, to the furnace walls and boiler tube banks, where "bird nesting" of slag was a serious trouble. Pulverized-coal firing, as first applied to slightly modified furnaces, did the same, but to a greater degree, as a higher percentage of ash was in suspension.

Water-cooled furnaces have overcome the erosion of the walls, and aided in lowering the temperature of the gases before entering the boiler tube bank. The spacing of the boiler tubes has been increased to minimize the tendency of ash and slag to accumulate. Turbulent pulverized-coal burners, resulting in more complete combustion with shorter gas travels and low excess air, have also been developed.

It seems that as soon as there are solutions for one set of conditions, new problems of economic importance arise which can only be solved by further study of the factors which enter the problem.

Some of the new problems of today are found in the design of steam-generating equipment for higher capacities which will produce steam at high pressures and at temperatures of from 900 to 950 F under close control, and which can be erected within restricted building spaces available. In fact, the steam temperature must be under accurate control because of the requirements of the turbine and the approach toward safe limits of tube temperatures at the high pressures. Superheaters must absorb more than 25 per cent of the total heat produced in the furnace, and to do this with any reasonable amount of surface they must receive gases at fairly high and dependable temperatures. Furthermore, the economics of superheater size and construction calls for high steam velocities within the superheater elements, hence relatively small and closely spaced tubes for high gas-mass flow through the tube bank. The distribution of steam inside and of gas velocity and temperature outside to the different portions of the superheater should be quite uniform, to prevent localized overheating.

The greatest single difficulty to be overcome in accomplishing all of this is the ash in the coal being burned. Not so much ash as such, but ash converted into slag through high-temperature combustion; slag that is inevitable, even from the lowest possible combustion temperature when using many of the low-fusing-ash coals, which are available today. Ash constitutes the main problem; ash in the form of molten slag, plastic slag, sticky slag, sponge ash, and fly ash.

Dry ash removal from pulverized-coal-fired furnaces is satisfactorily accomplished with a wide range of ash-fusing temperatures, using turbulent burners, water-cooled furnaces, and rates of combustion from 25,000 to 35,000 Btu per cu ft per hr. Such units are suitable for steam outputs up to about 200,000 lb per hr, and for moderate steam temperatures. For larger units, the ash and slag limitations call for lower rates of liberation and/or higher fusing temperature of ash, and perhaps less turbulent burners. In many localities there is available only a limited amount of coal of high ash-fusion temperature and therefore large furnaces for dry ash removal are being used less extensively.

The nature of the ash in the coal is of greatest importance in designing a furnace for dry ash removal. The intense heat resulting from turbulent burners causes temperatures sufficiently high to fuse almost any ash locally within the active-combustion zone. For satisfactory operation with dry ash removal, the places where the ash discharges to the ashpit, and the gases to the tube bank, must be far enough away from the burner zone to permit the necessary cooling of the ash before leaving the furnace, or else it will adhere to the hopper or other cooling means at the bottom, and to the boiler tubes at the top. In any event, some ash will adhere to waterwalls of any known construction, around the middle of the furnace within the burner zone, as shown diagrammatically in Fig. 1.

If mechanical cleaning is applied to this area, and it is kept reasonably clean, the furnace may be too cold for stable ignition and efficient combustion, especially at lower rates. The rate of radiant-heat absorption may also be too high for the safety of the tubes, if the water conditions are not perfect. Experiences with gas- and oil-fired boilers bear out these points, and some protection of bare water-cooled walls is necessary to obtain complete combustion and protect the walls against tube losses.

With coal firing, the ash or slag accumulation in this zone is uncertain, and troublesome because of its uncertainty. With a combination of ash characteristics and rate of combustion which produce a sticky but not liquid ash formation on the walls, the thickness of sponge ash may increase and spread both up and down to greater areas, insulating the heat-absorptive surface so

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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.

that gas temperatures rise. Higher gas temperatures further accentuate the spreading and thickening until unduly high temperatures reach the boiler tube bank and the bottom of the ash discharge and cause slagging troubles in one or both of these locations. Thick accumulations of slag, sometimes quite bulky, may fall intermittently from the upper part of the furnace, causing damage to the furnace structure.

If the ash characteristics and temperature to which it is subjected are such that the slag is of a molten plastic nature,

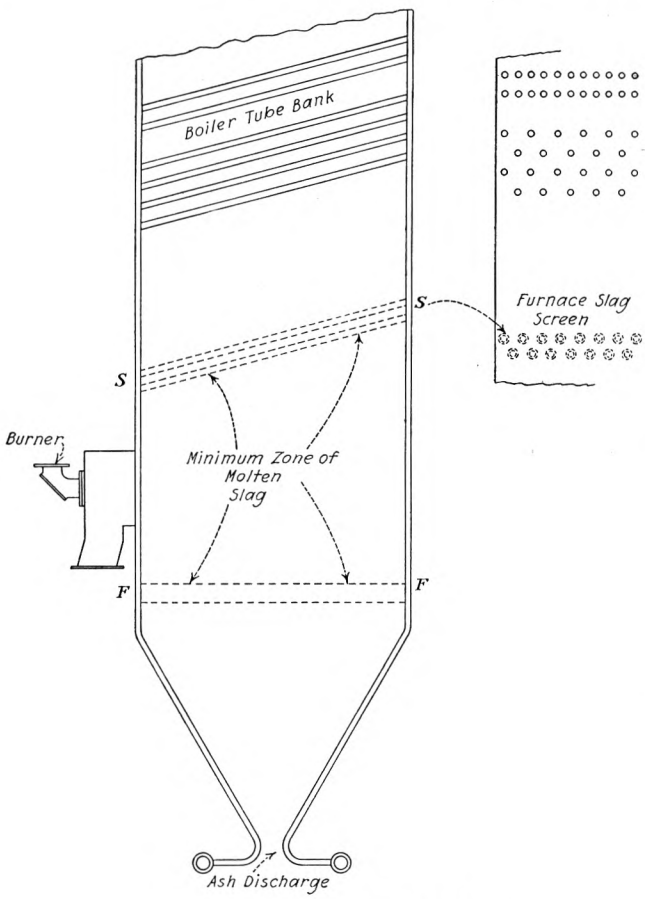


FIG. 1 DIAGRAMMATIC FURNACE

the covered area will also spread downward and upward, necessitating some cleaning to keep the coated area within bounds and also cleaning of the tube bank and ash discharge openings.

Another approach to the problem is to make the furnace of Fig. 1 with a floor at *FF*, which will accumulate molten slag, and to tap it out continuously or intermittently. This raises the furnace temperature in this zone so that the ash on the walls melts and thins down to an equilibrium thickness, with the net result that a higher rate of heat absorption per square foot of the lessened area takes place than existed in this zone before the floor was installed. The temperature in the slag zone is high enough to cause extending of the slag coating upward toward the boiler tube bank. This may call for cleaning of the walls in the upper part of the furnace, as the tube bank would otherwise have to be at an unreasonable height to prevent a gradual extension of slag, which would further insulate the wall and increase the gas temperature as the slag coating extended upward into the tube bank.

As radiant heat from the hot burner zone reaching the upper part of the furnace and boiler tube bank contributes to keeping the slag particles hotter than they would be if shielded, it is obvious that the placing of a screen *SS*, shown as consisting of water tubes coated with a slag-resisting refractory held on to the tubes by a suitable means, across the upper part of the furnace, making a two-stage furnace, serves a useful purpose. (1) The arch effect further increases the temperature of the combustion zone in the primary furnace, which increases the rate of heat absorption per square foot of remaining wall area and makes tapping of slag possible for higher-fusing-temperature ash, and for lower rates of combustion with a given ash. (2) The temperatures of ash and slag above the screen are reduced. (3) Any cleaning of the walls or boiler tube bank in the upper or secondary furnace is more effectively done in this cooler zone, and there is a more definite zone within which cleaning is possible. The screen cleans itself by the melting of the ash, which drips back into the primary furnace. (4) A larger percentage of the ash of the coal is collected than is recovered in a furnace for dry ash removal.

In actually building such furnaces the general scheme of Fig. 1 was carried out on the first installation at State Line Generating Co.[2] in 1929, except that the burners were changed to vertical firing, in order to utilize the furnace volume to better advantage and to collect a larger percentage of ash, especially the coarser particles.

There is full realization of the great importance of going into this problem thoroughly and obtaining sufficient data to make possible the most economical designs of boiler furnaces; designs that permit the operators to cope with the conditions of combustion and the removal of ash in the most satisfactory manner. Some of the results already obtained from these studies are so interesting and instructive that it seems best to present them now in a brief preliminary form.

TEST RESULTS

Table 1 gives, for comparison, certain data pertinent to the operating conditions and results from three different designs of boiler furnaces as shown in Figs. 2, 5, and 8, and plotted in Figs. 3, 6, 7, 9, and 10. As this work was undertaken primarily for gas-temperature studies and the comparison of different temperature-measuring devices rather than to check performance results and efficiency, complete test data were not taken. The heat input has been determined from the steam output, and the heat losses include heat returned in preheated air for combustion.

The nature and behavior of the ash and slag have also been observed during operation, samples have been taken of ash from the coal being burned, and samples of slag and the different kinds of ash from various parts of the furnace, slag screens, and tube banks. The fusing-temperature ranges of these samples have been determined with a view to learning whether or not the ash and slag samples, and their behavior at known temperatures in the Laboratory would add to the knowledge of actual temperatures in different zones of a furnace.

The gas temperatures were determined by a high-velocity thermocouple described by Dr. Mullikin,[3] and these readings were used to calculate the rate of heat absorption in the primary furnace, secondary furnace, and the boiler generating tubes. In the primary furnace the heat absorbed is the difference in sensible heat in the gases between the adiabatic temperature,

[2] Prime Movers Committee Report on Combustion, 1936, Publication D-7, Edison Electric Institute.

[3] "The Accurate Measurement of High Gas Temperatures," by H. F. Mullikin, *Power*, vol. 78, 1934, p. 565. This high-velocity thermocouple is inserted into the gas stream at any desired point in the furnace, and by an aspirator, gas is drawn across a shielded thermocouple at a sufficiently high velocity to bring the couple close to the true gas temperature, free from surrounding radiating influences.

TABLE 1 DATA RELATING TO GAS-ASH TEMPERATURE GRAPHS

	DATA RELATING TO:		FIG. 2 AND 3.			FIG. 5 AND 6.			FIG. 5 AND 7.			FIG. 5 AND 9.				FIG. 8 AND 10.			
1	MAXIMUM CONTINUOUS STEAM OUTPUT; LB. PER HR.		600,000			450,000			315,000			300,000				300,000			
2	STEAM OUTPUT; LB. PER HR.		600,000			450,000			330,000			225,000				342,000			
3	STEAM PRESSURE; LB. PER SQ. IN		480			1,270			1,240			1,260				1,225			
4	STEAM TEMPERATURE; DEG. F.		850			935			900			923				910			
5	FEEDWATER TEMPERATURE; DEG. F.		315			330			380			350				350			
6	PROXIMATE COAL ANALYSIS AS FIRED	MOISTURE; PER CENT.	3.3			2.3			2.2			3.0				2.8			
7		VOLATILE; PER CENT.	30.0			33.9			36.5			35.8				36.0			
8		FIXED CARBON; PER CENT.	60.0			52.0			53.5			53.2				54.0			
9		ASH; PER CENT	6.7			11.8			7.8			8.0				7.2			
10		SULPHUR; PER CENT.	1.69			1.2			2.0			0.8				0.7			
11	HEATING VALUE AS FIRED; B.T.U. PER LB.		14,180			13,080			13,500			13,300				13,470			
12	ASH FROM SAMPLE OF COAL; INITIAL DEFORMATION; DEG. F.		2,200			2,335			2,030			2,180				2,350			
13	SOFTENING TEMPERATURE; DEG. F.		2,350			2,460			2,110			2,245				2,515			
14	FLUID TEMPERATURE; DEG. F.		2,610			2,740			2,410			2,860				2,680			
15	TOTAL COMBUSTION AIR AT BOILER OUTLET, PER CENT.		117			125			114 [4]			120 [5]				120 [5]			
16	TOTAL WEIGHT OF GASES; LB. PER HR.		770,000			590,000			407,000 [4]			282,500 [5]				438,000 [5]			
17	TOTAL HEAT INPUT ABOVE 80°F; MILLION B.T.U. PER HR. [1]		838			602			453			292				464			
18	HEAT LIBERATION, PRIMARY FURNACE; B.T.U. PER CU. FT. PER HR.		74,000			90,500			95,300			61,800				98,000			
19	TOTAL FURNACE; B.T.U. PER CU. FT. PER HR.		31,800			38,200			40,250			32,800				52,100			
20	TOTAL HEAT INPUT DIVIDED BY TOTAL PROJECTED SURFACE OF ALL FURNACE WALLS PLUS BOILER TUBE SURFACE BEFORE SUPERHEATER; B.T.U./SQ. FT./HR.		65,400			72,500			70,400			56,100				89,500			
			PRIMARY FURNACE	SECONDARY FURNACE	GENERATING TUBES	PRIMARY FURNACE	SECONDARY FURNACE	GENERATING TUBES	PRIMARY FURNACE	SECONDARY FURNACE	GENERATING TUBES	PRIMARY FURNACE	FIRST OPEN PASS	SECOND OPEN PASS	CAVITY BELOW SUPERHEATER	PRIMARY FURNACE	FIRST OPEN PASS	SECOND OPEN PASS	CAVITY BELOW SUPERHEATER
21	TEMPERATURE GAS ENTERING; DEG. F.		3,740 [3]	2,960	2,550	3530 [3]	2,900	2,600	3,830 [3]	2,840	2,500	3,580 [3]	2,920	2,510	2,070	3,650 [3]	2,950	2,670	2,410
22	TEMPERATURE GAS LEAVING; DEG. F.		3,100	2,550	2,120	3,010	2,600	2,300	2,940	2,500	2,120	2,920	2510	2,070	1,695	2,950	2,670	2,410	2,250
23	HEAT ABSORBED; MILLION B.T.U. PER HR.		185	117	102	114	68.5	55.5	120	51.9	45.7	64.5	34.8	37.6	14.7	103.5	40.0	36.4	21.5
24	PORTION OF TOTAL LIBERATION; PER CENT		22.2	14.0	12.2	18.9	11.4	9.2	26.7	11.5	10.1	22.1	11.9	12.9	5.0	22.2	8.6	7.8	4.6
25	HEAT ABSORBED; BTU PER SQ. FT. PER HR.		78,000	30,500	15,600	65,700	28,200	16,000	63,600	21,000	16,180	35,500	30,900	29,000	15,350	57,000	35,600	28,200	22,400
26	EQUIVALENT HEAT RECEIVING SURFACE TEMP. [2] DEG. F.		2,820	2,612	2,225	2,750	2,645	2,350	2,575	2,570	2,175	2,775	2,530	2,060	1,800	2,715	2,645	2,380	2,170

[1] Includes heat in preheated air, less heat required to evaporate moisture and less heat in unburned fuel.
[2] The equivalent temperature of heat-receiving surfaces within a zone is calculated according to the Stefan-Boltzmann law, using the mean of observed gas temperatures and the rate of heat absorption within the zone, assuming black-body radiation. This may not represent the true average temperature of the tube with its coating of mill scale, dust film, sponge ash, or slag, due to undetermined values of such factors as emissivity and the like, but it is suggested as a method of comparing results which undoubtedly will lead to further knowledge regarding heat absorption in boiler furnaces.
[3] Adiabatic.
[4] Secondary-furnace outlet.
[5] Bottom second open pass.

as given in Table 1, and the observed temperature of gases leaving the primary furnace. The latter temperature was taken as the average of gas temperatures just before and after passing through the slag screen between the primary and secondary furnaces. A Leeds & Northrup optical pyrometer was also used in the primary furnace only, as it checks the high-velocity thermocouple (HVT) closely near 3000 F. The rate of heat absorption in a zone divided by the total absorption surface of the zone gives the heat absorbed, Btu per sq ft per hr, for that zone. See Table 1, item 25.

Item 26, found in Table 1, is the "equivalent heat-receiving-surface temperature" and is shown on the gas-ash temperature graphs as a straight line for each furnace zone. It is calculated according to the Stefan-Boltzmann law, using the mean of observed gas temperatures and the rate of heat absorption within a zone, assuming black-body radiation with both the radiating and receiving emissivities as 1.0 and the angle factor as 1.0. For the heat absorbed per sq ft per hr in each zone, determined from gas temperatures in and out, there is a black-body equivalent temperature of heat-receiving surfaces, given in Fig. 4.

With the mean gas temperature in the cavity as the radiating body, and the rate of heat absorbed per sq ft per hr by the projected surface of the water-cooled walls and the tube banks, determined by the HVT temperature readings entering and leaving the cavity, it is simple to calculate the equivalent black-body-surface temperature of the receiving-wall boundary areas of the cavity. It is, of course, known that the emissivity is not constant and may be much less than 1.0. This temperature, which may not be a real temperature of any physical body, is then compared with the determined fusing-temperature range of the slag, and some useful conclusions can be drawn about ash deposits on walls and tubes, as shown later. The latter temperatures are determined from actual samples of slag taken from the walls, tap spout, and tube banks throughout the unit and tested in the laboratory according to the A.S.T.M. method.

It is noted in Table 1, items 21, 22, 25, and 26, that the primary-furnace adiabatic temperature in Fig. 3 is 3740 F, the gas temperature leaving is 3100 F, the rate of heat absorption is 78,000 Btu per sq ft per hr, and the equivalent heat-receiving-surface temperature is 2820 F. The slag on the walls, and that being tapped from the primary furnace is about 2600 F and observation showed the water-cooled walls to be covered with a relatively thin film of molten ash that flowed down to the floor continuously and was readily tapped intermittently. The ash-softening and initial-deformation temperatures have little significance in the primary furnace.

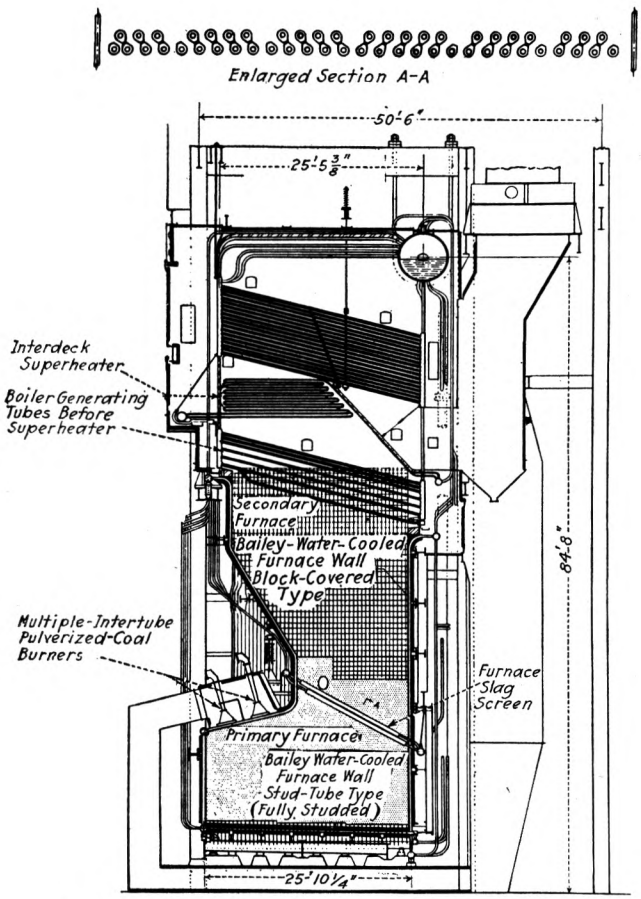


FIG. 2 BOILER UNIT WITH TWO-STAGE FURNACE
(See Fig. 3.)

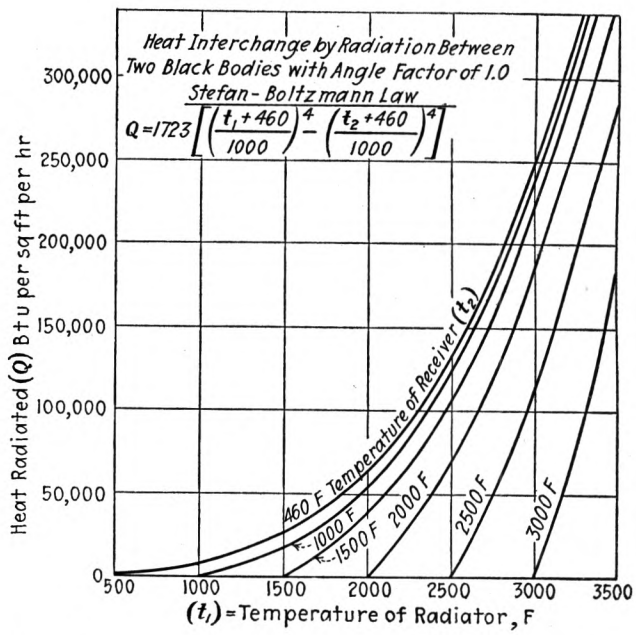
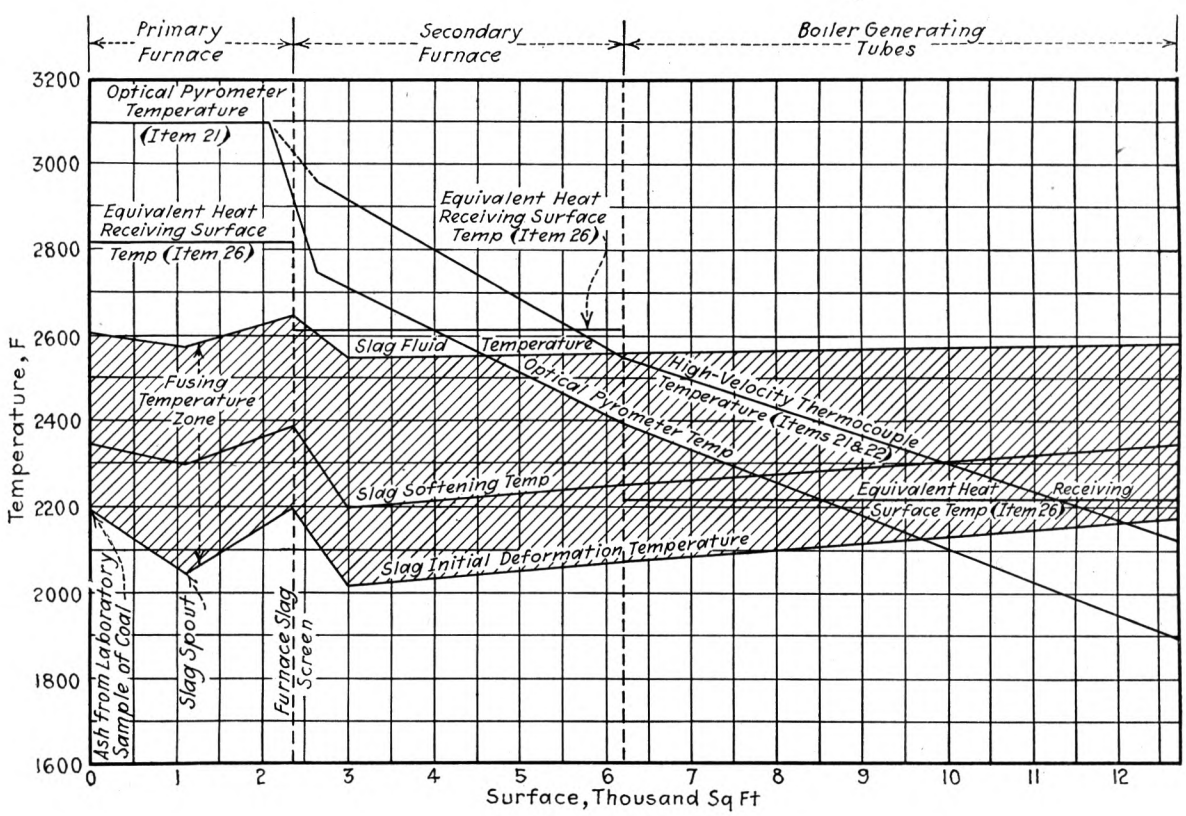


FIG. 4 HEAT INTERCHANGE BY RADIATION BETWEEN TWO BLACK BODIES WITH ANGLE FACTOR OF 1.0

FIG. 3 (BELOW) GAS-ASH TEMPERATURE GRAPH FOR UNIT SHOWN IN FIG. 2



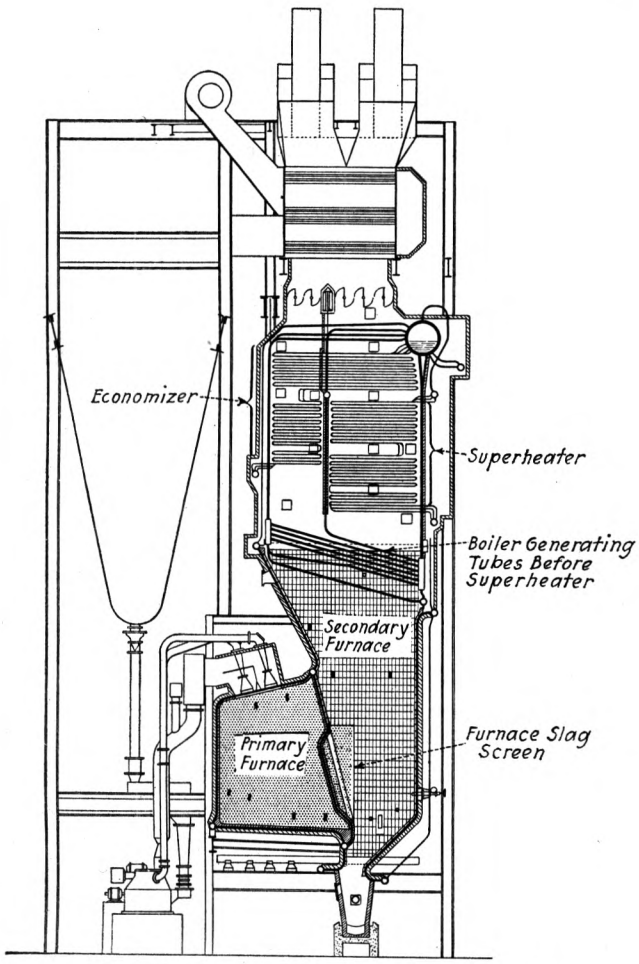


FIG. 5 BOILER UNIT WITH TWO-STAGE FURNACE
(See Fig. 6.)

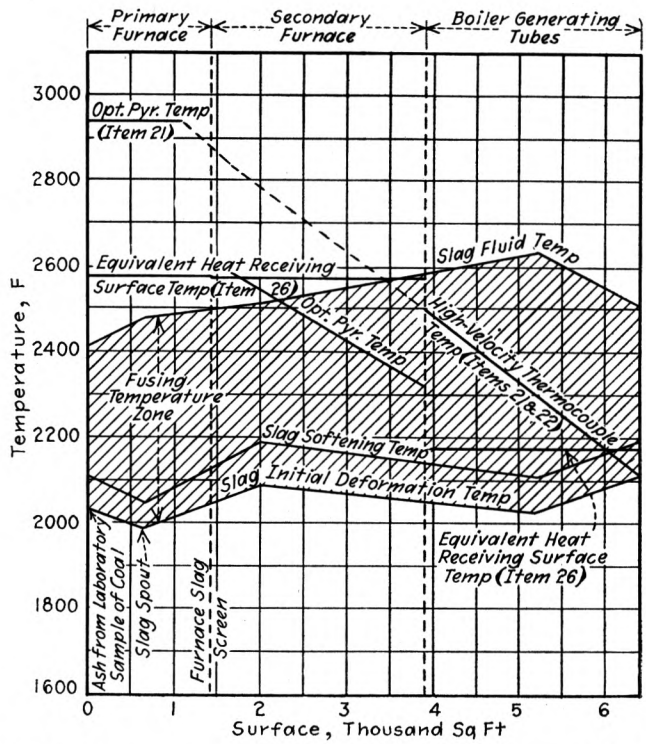
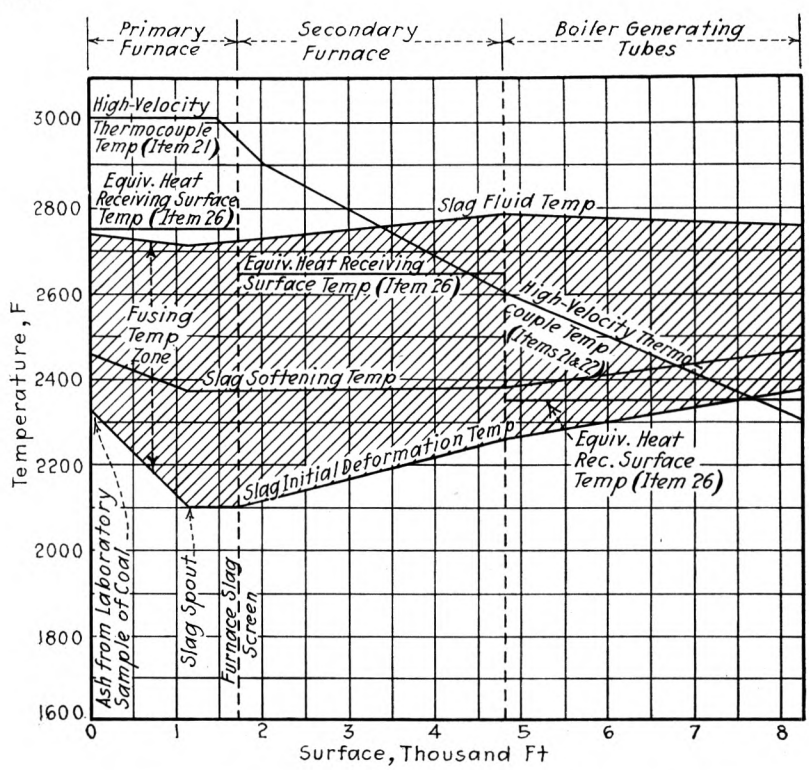


FIG. 7 GAS-ASH TEMPERATURE GRAPH FOR UNIT SHOWN IN FIG. 5

FIG. 6 (RIGHT) GAS-ASH TEMPERATURE GRAPH FOR UNIT SIMILAR TO FIG. 5



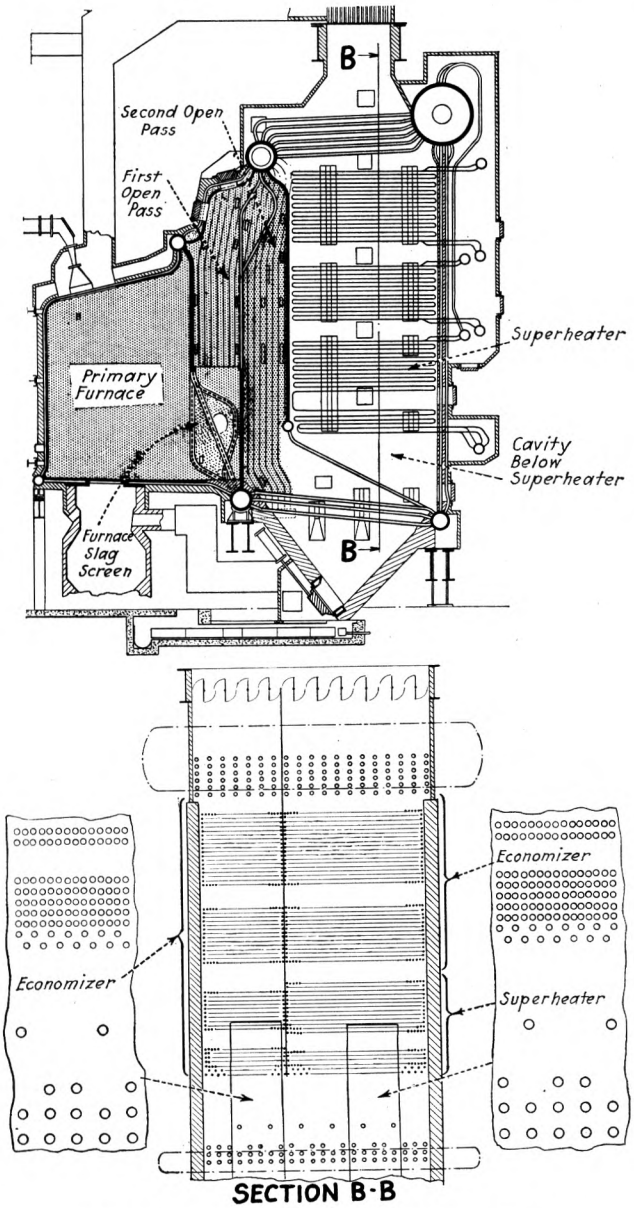


FIG. 8 BOILER UNIT WITH OPEN-PASS FURNACE
(See Figs. 9 and 10.)

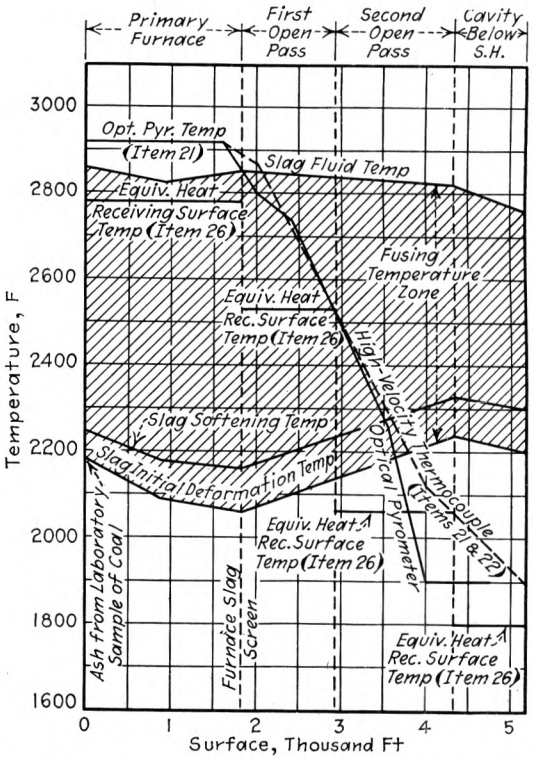


FIG. 9 GAS-ASH TEMPERATURE GRAPH FOR UNIT SHOWN IN FIG. 8, AT MEDIUM RATING

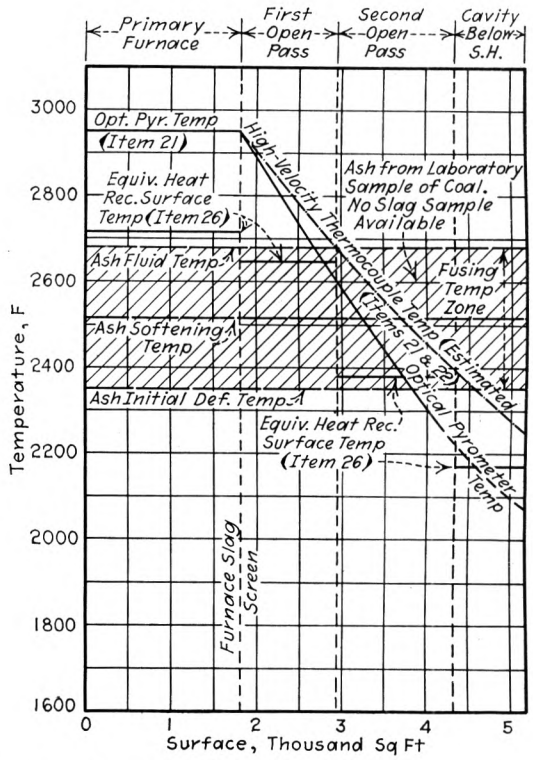


FIG. 10 GAS-ASH TEMPERATURE GRAPH FOR UNIT SHOWN IN FIG. 8, AT HIGH RATING

In the secondary furnace similar calculations show an equivalent heat-receiving-surface temperature, item 26, of 2612 F and a slag fluid temperature of 2550 F. As a check in the actual furnace, the slag forms a sheet covering over the water-cooled wall and a slow flowing process takes place, maintaining an equilibrium condition without cleaning. The slag falls off in sheets from $\frac{1}{2}$ in. to 1 in. thick when the boiler is removed from the line or sometimes when the output is reduced. The gas enters the boiler generating-tube bank at 2550 F, slightly below the fluid temperature, and leaves at 2120 F, slightly below the initial deformation temperature of 2170 F for the slag collected from that point.

In other words, the temperatures throughout the boiler tube bank are within the sticky range of the ash which collects there, and mechanical cleaning by means of telescopic deslaggers and retractable jets using steam-water mix are needed and are used regularly to keep this bank clear for gas flow and to keep it absorbing enough heat to pass gases of the correct temperature to

the superheater. There is no slag problem in the upper tubes of the boiler bank nor in the superheater, which checks with the temperatures and the slag characteristics.

This method of study is somewhat analogous to using Seger cones in ceramic work as a measure of furnace temperatures as related to the results desired in burning ware. The ash and slag supply a basis of comparison of effective results and enable performance results to be studied with greater assurance as to their accuracy and applicability to other designs.

The equivalent heat-receiving-surface temperature of the boiler generating tubes of 2225 F looks high, but it is to be remembered that it is a calculated temperature. However, these tubes are never thoroughly cleaned, for even the best of cleaning leaves a residual coating of ash which may be several hundred degrees above the tube temperature. An increase in ash or slag coating may raise the new surface temperature surprisingly high. While the equivalent black-body-surface temperature is not real and an emissivity of less than unity will lower it, the actual surface temperatures cannot be much lower and be in line with the behavior of the ash as observed at the various locations.

These equivalent black-body-surface temperatures will of course not be uniform throughout the secondary furnace and tube bank, but it is thought best for the time being to calculate them on the simple unity basis and plot them as horizontal lines until further studies can be completed. For the present it is thought this equivalent surface temperature will be useful in comparing results between different conditions such as methods of cleaning, effect of different kinds of coal, methods of firing, adjustment of excess air, and so on.

It should be noted that these data in Fig. 3 from the furnace of Fig. 2 are obtained when operating steadily at 600,000 lb of steam per hr and, for comparison with other graphs in this paper, the rate of liberation is 74,000 Btu per cu ft per hr in the primary furnace alone and 31,800 in the primary and secondary furnace combined.

The total input to the furnace is 833,000,000 Btu per hr, or 65,400 Btu per hr available per sq ft of total heat-receiving surface in the entire furnace and boiler generating-tube bank. The actual absorption in the primary furnace is 78,000 Btu per sq ft of wall surface per hr, while in the secondary furnace it is 30,500, and in the tube bank 15,600 Btu per sq ft of tube surface per hr.

The gas-ash temperature graph, Fig. 6, was obtained from a furnace, as shown in Fig. 5, operating at its full continuous-load condition. The ash is somewhat variable and the averages are shown in Fig. 6. The slag from this ash has a relatively high fluid temperature which, coupled with a tendency to form a sticky adhering formation at lower temperatures than indicated by its initial-deformation temperature, makes a troublesome cleaning and operating problem.

None of the rates of heat absorption in Figs. 5 and 6 are as high as the corresponding rates obtained from the furnace of Figs. 2 and 3, even though the rates of heat liberation are about 20 per cent higher and the heat input per square foot of total furnace-wall and boiler-tube surfaces (item 20) is about 10 per cent higher. This result is now believed to be largely due to the higher fluid temperature of the slag, which is about 2750 F as compared with 2550 F for Fig. 3. The nature of the slag structure, especially on the boiler tube banks, is also significant, as the same means for deslagging are much less effective in Figs. 5 and 6.

Another more direct comparison is between Figs. 6 and 7, where both have the identical cross section of furnace, but the latter is burning a different coal with lower-fusing ash than Fig. 6. The furnace of Fig. 7 is narrower in almost direct proportion to its output at full load, at which rate the data were obtained.

The observed behavior of the slag throughout the furnace and boiler of Fig. 5 with the different coals of Figs. 6 and 7 is consistent with the equivalent heat-receiving-surface temperatures.

The boiler of Fig. 8 has a rated continuous output of 300,000 lb of steam per hr. The test of Fig. 9 was run at 225,000 lb of steam per hr, or 75 per cent of its full-load conditions, with one kind of coal, and the test of Fig. 10 at 342,000 lb of steam per hr, or 14 per cent above full load with a different coal. Unfortunately, slag samples were not obtained from the furnace during the latter run, but the nature of the ash with low iron content would presumably produce a slag with little change in its fusing range from that of the laboratory determination from the coal ash, which is plotted.

The operation of this type of furnace, with two open passes and a small cavity between the last pass and the superheater, with no boiler tube bank whatever, has been very satisfactory.

The slag is easily tapped continuously from the primary furnace with ash of 2800 F fluid temperature and at ratings below 70 per cent of normal. The slag in the first open pass flows slowly back to the primary furnace, and this pass needs no cleaning or attention, which is also true of the primary furnace.

The second open pass accumulates some slag, but is more or less self-cleaning due to the down flow of gases and the gravity action on the accumulation which, with the coal being used, is near its initial-deformation temperature.

Some cleaning of slag from the screen tubes at the top of the first open pass is required. This is being done with a retractable-nozzle deslagger.

Single-jet retractable blowers are also used occasionally at the entrance to the superheater and economizer by-pass, to clear the slight amount of sponge ash which accumulates there.

The behavior of ash within a furnace is of such importance that a few comments regarding it should be kept in mind while discussing the subjects of temperature and rates of heat absorption.

In burning pulverized coal from modern direct-fired systems with turbulent burners in hot furnaces and the proper excess air, all but a small residue of the coarser coal is burned completely in the furnace and only residual ash particles remain. A microscopic examination of such fly ash discloses tiny fused black, brown, gray, white, and transparent spheres, together with crystals of unfused ash.

There are also some carbon or coke particles from the small residue of coarser coal present in the fly ash. A microscopic study of these is even more interesting and significant in connection with slag formation. Such a piece of coke has on its surface a number of beads of ash, many of which are fused into multicolored spheres and are the residue of ash in the coke which has already been burned from the surface of the particle. They are held to the coke by the adherence to small particles of ash which are only partly exposed at the surface of the coke. Many of these spheres and particles of carbon-free ash are continually falling free from the coke, hence the explanation for fly ash being so much smaller in size than if each particle of coal retained all of its ash as a single particle. The most significant result of this particle of coke with its nodules of ash is that its continued burning keeps its temperature above that of the carrying gas and the ash, being kept hotter, is more likely to stick to wall or tubes, holding the carbon where it burns still more rapidly due to the scrubbing action of the gases. As the combustion is completed, the residual ash is largely retained at a relatively higher temperature and is added to the accumulation of slag.

This is one explanation of the generally accepted fact that coarse coal and improper combustion are conducive to slag accumulations. The slag in boiler tube banks from fuel beds is largely carried up in the form of burning particles of coke rather than of free ash itself.

While it is well recognized that ash in coal is a complex sub-

stance and individual constituents of it vary in fusing temperature from as low as 1800 F to as high as 4000 F, it is treated as a mixture; in the laboratory, ash residue from a slowly burned out coal sample is formed into a cone-shaped briquet and its fusing-temperature range is determined in a muffle furnace and reported as (1) initial-deformation temperature, (2) softening temperature, and (3) fluid temperature. The initial deformation is that at which the cone starts to distort and is really indicative of when the lowest-fusing-temperature particles begin to soften and cement the higher-fusing particles. The softening temperature is that at which the cone has softened enough to turn the apex downward, or to bend the axis about 90 deg from its initial vertical position. This particular temperature has been referred to in the past as the fusing temperature and has been generally used as the important temperature in measuring the behavior of the ash and comparing the relative values of coal. When coal is to be burned in fuel beds this temperature is of value for comparisons, but it is now believed to be of little significance in large pulverized-coal-fired furnaces. The fluid or liquid temperature is important in connection with slag-tap furnaces and for a knowledge of the behavior of ash in water-cooled furnaces operating at temperatures above the fluid or liquid temperature of the ash, as most of them do at their normal and maximum ratings. In this paper "fusing temperature" is used as an inclusive generic term covering all three of the specific temperatures within the band just described.

Within a furnace or boiler unit, the ash collecting at any point may not be a true average of all that which is in the coal. Dense, hard-grinding particles, such as sulphide from pyrites, are more likely to be retained on the floor of a furnace. High-fusing-temperature particles are more likely to pass on through the entire furnace and boiler unit as fly ash, as they themselves are not sticky but they may be retained to some degree by adhering to sticky ash already retained on the surfaces. The particles which are molten have a double chance of being retained within the furnace, due to their sticking to cold surfaces, and surely to hot sticky surfaces.

Comparison of Methods for Measuring Furnace Temperatures

(*a*) The high-velocity thermocouple previously mentioned is the most accurate practical method now known for measuring gas temperatures.

(*b*) Certain forms of optical pyrometers (disappearing-filament type) are reasonably accurate for the highest-temperature zones of furnaces, where uniform temperatures prevail.

(*c*) Bare thermocouples have not been found particularly useful for gas-temperature measurement.

(*d*) Radiation pyrometers and other forms of elements sensitive to heat and light are being carefully studied, as it is desirable to know more than merely gas temperatures.

(*e*) It is desirable to know the temperature of slag particles floating in the gas stream, especially as they approach tube banks. Such slag particles are undoubtedly hotter than the carrying gases leaving a burner, if still within view of the hotter zones of the furnace; and cooler than the carrying gases when approaching cooler surfaces, especially when shielded from the radiant heat of the hotter zones. Perhaps a bare thermocouple of the proper size more nearly approaches the true temperature of the slag particles until some better method can be devised.

(*f*) It is desirable to know the temperature of floating particles of coke, especially just as they strike an arresting surface, also as they continue to burn when so arrested. No work has yet been done in this direction.

(*g*) A most important factor that should be known is the heat reflected and radiated from the wall surface. If this can be determined, it may give a more direct method than we now have of measuring the rate of heat absorption at the wall.

Summary

1 The fusing-temperature determinations on ash from a sample of coal, as now burned out in the laboratory according to A.S.T.M. standard methods, are not accurately indicative of the corresponding temperatures of the slag as formed in furnaces from the same coal. Many other data beside those shown here indicate much greater deviations. Usually the slag accumulated in the hotter zones near the burners and on the floor, or in the slag-tap section, is lower in all points than that determined from the coal sample by the standard method. There is a general tendency for the slag and ash collected beyond the hot zone to be higher in all significant temperatures.

2 Some ash or slag adherence takes place on all furnace walls, boiler, superheater, and other surfaces. So long as it adheres at all it tends to become cumulative. Dry nonsticky ash may maintain a close equilibrium by falling off due to gravity or being carried on by the normal gas velocity.

Sponge ash or even vitrified slag may fall off in pieces intermittently by its own weight, especially if the rate of combustion is changed from time to time, and the surface to which it adheres is not holding it too tenaciously. Continuous high-rating operation is usually conducive to continued accumulation of sponge ash or plastic ash in the zones where the ash is within the sticky-temperature range. This zone in the furnace should be as small as possible. It may be desirable to clean it by hand lancing or some deslagging means if the need for uniform conditions warrants.

The parts of the furnace where the temperature is maintained above the fluid temperature of the ash which deposits there, is kept at a more uniform condition than any other part because of the continued melting and running away, leaving a base of plastic slag. The lower the fluid temperature of the slag the greater the heat absorption in a given furnace with a given rate of heat input, and also the lower the gas temperature leaving such zone.

3 Ash and slag having a wide spread between the initial-deformation point and the fluid temperature are more likely to be troublesome than those with a smaller temperature range and a sharper melting point. With a wider spread like that in Fig. 6 from 2100 to 2730 F the fly ash must pass through a wide temperature range, and therefore a large surface exposure, before it is cooled from the liquid phase where it is easily handled, to the dry nonsticky stage. The coal of Fig. 9 covers even a wider spread from 2060 to 2850 F. In some furnace arrangements this might be extremely difficult to handle, although in the furnace of Fig. 8 no difficulty has been encountered.

4 A valuable method of comparing furnace-operating results has been discovered consisting of (1) taking accurate gas temperatures throughout a furnace, (2) calculating from them the rates of heat absorption in different zones, (3) calculating an equivalent heat-receiving-surface temperature for each zone, (4) taking samples of ash and slag from different portions of the furnace and determining the fusing-temperature range from such samples, (5) plotting these data as a gas-ash temperature graph against heat-absorbing surface, (6) comparing the data as shown on such graph with observed behavior of ash and slag in the furnace, and (7) comparing different furnaces, different coals, and methods of operation through the use of such graphs.

5 The equilibrium conditions with molten-slag-covered surfaces is a subject deserving further study. The data presented here were obtained under what were thought to be equilibrium conditions, brought about by long periods of operation at uniform load. However, only further continued observations will determine when equilibrium has actually been obtained.

Others who may follow this line of study should keep this in mind. For instance, if a unit that has been operating a long time at high rating is suddenly reduced to low rating, the reduced furnace temperature may cause a complete stoppage of flowing slag over the furnace walls and floor. Continued accumulation of ash will increase the thickness of slag until the furnace temperature is increased by the insulating effect, and fluid conditions are again restored. The ash accumulates more slowly at the lower rates of combustion. When the rate of heat input is again increased there will be a period of higher temperature until the surplus slag film is melted away.

6 The studies presented herein, indicate that the nature of ash in the coal being burned, has such an important bearing upon the operating results from furnaces, that proper selection of coal must be made for many, if not all, furnaces. It is believed that further studies along the lines herein set forth will enable furnaces to be designed which will permit wider latitude in the selection of fuel.

7 The superheater bank, usually consisting of relatively closely spaced small tubes, must be kept clean. This means that the fly ash should be below its sticky temperature, yet the gases should be as hot as possible for high superheat. The velocity and temperature of gases should be as uniform as possible to all parts of the entering area of the superheater for the protection of the tubes. Some cleaning device is needed, as a gradual accumulation of sponge ash or dust is inevitable. It should be kept uniformly clean, because if certain areas accumulate sponge ash and direct the flow of gases to limited portions of the superheater the higher gas-mass flow may quickly overheat such portions of the superheater tubes.

8 Colored moving pictures of furnace interiors have been helpful in studying ash and slag formation on heat-absorbing surfaces, with a view to obtaining better methods of removal and preventing the formation. By taking such pictures at high speed and from points close to the furnace doors, many phenomena can be seen in much more detail than from personal observation. Time and speed measurements can also be taken.

9 In conclusion, it should be remembered that this paper includes data from only four installations. The results were so illuminating that it was deemed to be to the benefit of professional progress to publish the data and method of comparison now, so that others active in this work can utilize what has been reported. Criticism and suggestions will be appreciated as an aid in developing and applying this method in studies which will surely be pursued further in the future.

Acknowledgment

Acknowledgment and appreciation are given to R. Shellenberger, Dr. H. F. Mullikin, and E. D. Grimison, associates of the author, for their diligent and constructive assistance in this work; without their help and cooperation the data here presented would not have been available.

Discussion

J. M. Drabelle.[4] This paper is particularly valuable to operating men who have in their charge large boilers and furnaces operating at high ratings.

For the last year the Iowa Electric Light and Power Company has been experimenting with the motion-picture camera and Kodachrome film for obtaining photographic records for use in studying stoker and furnace operation. Experience seems to indicate that colored motion-picture film is a valuable tool for obtaining a clear, concise, and readily examined picture of furnace operation. It should be borne in mind that, being in color, relative color values are not destroyed as they are when a furnace is observed through conventional blue-glass screens.

The equipment which we have been using is an Eastman Ciné Kodak equipped with an F-1.9 lens. To protect the camera from the intense heat of the furnace Aklo glass made by the Corning Glass Works, Corning, N. Y., has formed the shield through which the photographs have been taken. This glass has the peculiar property of transmitting light but not transmitting heat waves. Two screens of Aklo glass are used in series held loosely in a suitable frame. The lens of the camera is placed immediately back of the second sheet of glass. Up to the present time, no attempt has been made to use the conventional exposure meter. The diaphragm of the camera is stopped wide open. The resulting photographs have shown slag phenomena and burning phenomena very clearly, particularly on underfeed and chain-grate stokers. Formation of clinkers as well as the stratification in burning gas streams from the fuel bed are also depicted. To those having furnace trouble, this is a possible means of studying conditions within the furnace. Once the trouble is determined, a suitable remedy may be applied.

H. D. Harkins[5] and A. C. Flinn.[6] One of the most interesting points brought out in the paper is the conclusion that, for the same heat release, the transfer coefficient would be higher with coals having low-fusion-temperature ash due to the fact that the low-fusion ash forms a thinner, more fluid slag which offers less resistance to the flow of heat to the tubes. From this it would follow that low-fusion-ash coals would require a smaller amount of surface in the primary furnace or that, given similar conditions, a higher transfer rate would obtain in a slag-tap furnace, where the temperature must be such as to keep the ash fluid, than in a dry-bottom furnace. This indicates the possibility of building smaller and, consequently, cheaper furnaces for what have been considered the more difficult (low-fusion-ash) coals. Heretofore larger and more costly furnaces have been built for them.

The comparison between the furnaces, Fig. 2 and Fig. 5, shows about 19 per cent greater transfer rate in the primary furnace with the coal of lower ash-fusion temperature. However, it does not appear to have been proved conclusively that ash fusion is the direct cause or that other effects may not have contributed to the difference in heating rates.

For instance, Sherman[7] at Battelle Memorial Institute, in working on pulverized-coal flames from four types of bituminous coal, found variations in the emissivity coefficients and also in the initial flame temperatures. The emissivity coefficients varied as much as 30 per cent. Due to the fact that the coals having the highest emissivity also had the highest initial temperature, the heat radiated varied even more than this amount.

Since these data on radiation from the flames of several coals indicated a difference greater than that ascribed here to the difference in slag fluidity, the question arises as to whether emissivity or other similar effects may not have accounted for some of the difference in transfer rates?

E. A. Hitchcock.[8] It seems only a short time ago—about

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[6] Fuel Engineer, E. I. du Pont de Nemours & Co., Wilmington, Del.

[7] "Burning Characteristics of Pulverized Coals and the Radiation From Their Flames," by R. A. Sherman, Trans. A.S.M.E., vol. 56, 1934, paper FSP-56-6, pp. 401–410.

[8] Dean Emeritus, College of Engineering, The Ohio State University, Columbus, O. Mem. A.S.M.E.

thirty years—that I was with a group of engineering students in a new pumping station of the city of Cleveland. How we admired that pumping equipment, which was the "last word." And the boiler-room equipment! I can still see the highly polished brass feed lines and those shining gages. In one respect we could hardly believe our eyes for those gages stood at the unbelievable pressure of 180 lb per sq in. My conservatism relative to future developments led me to exclaim: "That is the limit!" What an unwise statement! It certainly cured me of making prophecies applying to the fields of engineering and science.

Nearly twenty years ago I was dining with Benjamin G. Lamme, who was for many years chief engineer of the Westinghouse Electric & Manufacturing Company. At that time the largest turbogenerators built were about 50,000 hp. I felt the limit had been reached but ventured to ask Mr. Lamme what in his opinion would be the limit in size of such a machine. His immediate reply was: "The limit is the customer's pocketbook." I little thought at that time I would see a verification of Lamme's statement, but I did—a 225,000-hp machine going through the Westinghouse shops and later in operation in the Richmond Street Station of the Philadelphia Company. Now I assume, the end is not yet.

The most startling advancement, as I see it, has been on the steam-generating side. This progress has not been easy sailing by any means. One very important phase is now being presented by the author. When he became acquainted in a practical way with the burning of bituminous coals, boilers having steam capacities of 7500 lb per hr at 150 lb per sq in. pressure were about the upper limit. Where are we now? One million pounds of steam per hour at 1500 lb per sq in. pressure or more and 330 F superheat. An increase in capacity of over 130 times the equipment of thirty-five or forty years ago, to say nothing of the increases in pressure and temperature.

It was my privilege, about thirty-eight years ago, to be associated with the author on various kinds of fuel and combustion tests. For about four years we "bled and died" together, he occupying the important position of "go-between" from coal pile to boiler furnace, usually of the Dutch-oven type, while I kept the records and fulfilled sundry other little jobs which required very little brawn and, I guess, a small amount of mentality.

To generate the steam by the shovel method was no bed of roses. It was no punching of buttons and watching of recording and indicating gages. The slicing and cleaning of fires, and the pulling of clinkers was a man-size job; at times an asbestos suit would have been acceptable. The progress that has been made in this particular direction is as great as from the labor of the galley slaves to the duties of the boiler-room crew of an oil-burning ocean liner.

It was, I am sure, this early experience with many kinds of coal, producing ash and slag of various characteristics, which convinced the author that here was one of the most difficult and trying problems in the use of fuel. Many were the times, after a few hours running, that the grate bars became so clogged with fused ash that a shutdown was necessary! The importance of considering this slagging factor in testing coals became so apparent that all clinker above a certain size was sorted from the total refuse and so recorded. My records show that this factor was first determined by us in April, 1899. This early experience was, I believe, the beginning of the author's interest in ash and slag, considered as among the most deterrent factors in the economical generation of steam. As a result, there soon followed determinations of the fusing temperature of ash. The importance of this factor is recognized today.

It was in November, 1911, that the author read a paper before the Ohio Society of Mechanical, Electrical, and Steam Engineers. His statements and conclusions in that paper[9] were based upon the results of twenty-eight complete heat-balance boiler trials with fuels, having ash-fusing temperatures running from 2210 to 2700 F, and also upon a valuable tabulation of thirty-two fusing temperatures of fuel ash, ranging from that of anthracite at 3150 F to bituminous at 2140 F. This tabulation included also the percentage of ash, and sulphur, and in several cases the iron oxide in the ash.

My purpose in calling attention to this early investigation work is to bring about a consciousness that slag was a recognized problem nearly forty years ago. It was controlled partly by brute force and awkwardness. It was the "imp" in the coal pile. Most of us have had experience enough to know that Mother Nature seldom surrenders her treasures without some tough work somewhere by someone. That coal imp is still with us, although our methods of handling him have changed greatly.

We are, I am sure, pleased that the author has come forward with suggested methods for comparing furnace operation. Support for his recommendations is the results obtained upon three types of furnace settings. In these investigations a most unusual and important feature was the application of colored moving pictures in studying furnace activity. They were obtained with considerable hazard to the observer. It is to be expected that, as a result of these investigations and those to follow, there will be advancements in furnace designs and reduced operation costs. However, I believe a marked increase in heat efficiency, even with the best practice of today is not to be expected, for it is not there to get.

Ten days ago I was at the Glenlyn plant of the Appalachian Power Company and there learned from M. P. Lawrence, superintendent, that their latest installation, a powdered-coal furnace with wet bottom, had a record of nine months of 24-hr continuous operation, with Pocahontas coal, generating approximately 350,000 lb of steam per hr and giving an efficiency of between 85 and 86 per cent. I understand a later installation at the company's Logan plant, having a capacity of 1,000,000 lb per hr at 1425 lb per sq in. pressure, dry bottom, is giving about the same efficiency with a bituminous coal of much higher ash content than the Pocahontas.

The experience of M. P. Lawrence with powdered coal at the Glenlyn plant has been so satisfactory that he agrees completely with those who believe the use of powdered fuel in large generating units is destined to become general. It is conceded, however, that at present the ash characteristics are important deciding factors.

For example in the Glenlyn plant is a chain-grate equipment 24 ft wide by 26 ft long, upon which bone coal with from 15 to 25 per cent ash works most satisfactorily. That coal condition would probably make combustion by the powdered-coal road difficult and impractical. As I see it, from my forty-five year point of view, the ideal combustion of coal is not reached yet; there are a few factors still to be licked. The author of this paper has handed to us, on asbestos gloves, some hot nuts to crack.

P. B. METCALF.[10] Too many statments have been made in the past by various people that pulverized-coal firing is a panacea for troubles encountered with any and all types of coal, in other methods of firing. It is gratifying to have the author's frank statement: "The studies presented herein, indicate that the

[9] "Fusing Temperature of Coal Ash and Its Relation to Rate of Combustion," by E. G. Bailey, *Journal* of the Ohio Society of Mechanical, Electrical, and Steam Engineers, November, 1911, pp. 53–67.

[10] Steam Engineer, New England Power Service Company, Boston, Mass.

nature of ash in the coal being burned, has such an important bearing upon the operating results from furnaces, that proper selection of coal must be made for many, if not all, furnaces."

The statement is made early in the paper: "If mechanical cleaning is applied to this area, and it is kept reasonably clean, the furnace may be too cold for stable ignition and efficient combustion, especially at lower rates." This conclusion appears to be at variance with other authorities on the subject, notably with a statement made by Henry Kreisinger,[11] who states: "Beyond the temperature required for quick ignition of the fuel, nothing is gained by going to higher temperature. In fact, there are experimental indications that high temperatures retard the combustion."

In view of this seeming conflict of conclusions on the subject, it would be helpful if the author could give further information toward clarifying the issue.

The explanation of why coarse coal and improper combustion are conducive to slag accumulations is most interesting and further emphasizes the need for fine pulverization with an absolute minimum of coarse particles.

J. F. MUIR.[12] This paper, illustrated by motion pictures has the distinction, within the writer's experience, of being the first presentation of its kind dealing with the effects and significance of ash transformations as they occur in the boiler furnace.

The following comments are made as a member of the A.S.M.E. and not as an employee of any company.

One of the many points of importance brought out by the author and one that deserves repetition and emphasis is the fact that, in the design of the high-capacity boiler, it is not sufficient, as has been done heretofore, to take a single temperature value for the change in ash from and to its solid state in the combustion process. It is pointed out that, in setting up design fundamentals, it is necessary to give consideration to and make allowances for first, the initial-deformation temperature; second, the softening or plastic temperature, which has been commonly referred to as the fusion temperature; third, the fluid temperature which is the one that must be maintained in the slag-tap bottom. The author includes all of these three specific temperatures under the category of "fusion temperature," which classification incidentally might be referred to as the "fusion-temperature range," since it involves more than one temperature. While the application of this fusion classification in combination with the gas-ash-temperature graph to the design of the combustion system is quite complicated, nevertheless, these are definite advances in the development of design fundamentals having practical possibilities.

Operators of high-capacity units have recognized the significance of these three temperature classifications and have learned that combustion temperature, when properly concentrated to give complete combustion in the furnace, will, of course, liquefy any kind of ash. To avoid slagging of boiler, superheater, or economizer convection surfaces the gas-entering temperatures should, if anything, be below the softening or middle-temperature classification. Furthermore, to meet the operator's idea of slag prevention, gas flow should be uniform in all sections of the assembly and the velocity of the gas, entering the convection sections, should be relatively low to get away from adhesive impact of slag against tubes. The operator has also learned that for rapid and complete combustion, fineness of pulverization and high turbulence over the entire combustion zone are essentials.

It has been mentioned that slag in the furnace is inevitable, but it might be added that slagging is intensified to a great extent in units designed for high capacity, high-steam temperatures and high rates of heat liberation. Operating experience with units fired with pulverized coal during the last two years indicates that the greatest barrier to holding the boiler in service for extended periods is the deposit and accumulation of slag or ash while in a semiliquid or a plastic state in the convection sections, particularly those having narrow gas passages between rows of tubes.

The elimination of these troubles and to meet the more or less ideal conditions which have just been described might call for an appreciable increase in tube surfaces over that provided in many installations. The first cost would also be higher; but if an increase in the availability of the unit can be accomplished, the higher first cost might readily be warranted and approved. High availability, it should be emphasized, is the operating factor that is necessary to establish economic justification for the installation and for the money invested in the project.

It may be pertinent to mention that in some of the recent units the time of travel through the furnace, from the burner to the boiler-tube-bank inlet, is about 1 sec. During that extremely short interval the volatile and carbon in the coal are consumed while the ash liberated has been transformed from solid to liquid and back again to a solid, all of which process seems almost incredible in the space of time available. Under these conditions, it is not surprising that more than 50 per cent of the ash in various stages of plasticity passes through the assembly, requiring higher-cost collector equipment to prevent undue release of ash to the atmosphere.

One of the most serious troubles in the combustion system of the high-pressure and high-temperature units and the one, which possibly causes the most interruptions to service, is the stopping up of the gas passages in the convection-type superheater with slag and ash. Steam blowers in many cases have failed to dislodge accumulations, while hand lancing in many cases necessitates shutting down the boiler. Because of the inaccessibility of the superheater passages, the manual process is a slow, awkward, and expensive operation.

To superheat steam above 900 F, with superheaters of the convection type, at the entrance to the superheater section, requires a gas temperature close to 2000 F. With an ash-fusion-temperature range of, say, 1900 to 2300 F, it is next to impossible to cool the ash down to a more or less dry state until the ash particles have advanced a considerable distance into the superheater gas passages. Now, if the gas passages between the superheater tubes were large and the gas velocity relatively low or, say, in the order of 50 ft per sec, it is possible that the superheater would remain free from slag choking for long periods. However, superheaters have not been made with wide spacing; in fact, passages between tubes are actually less than 1 in. and gas velocities up to 100 ft per sec are quite common. These conditions may be satisfactory for ash of high-fusion temperature, but with ash fusion in the 1900 to 2300 F range and, particularly for gas carrying in suspension over 50 per cent of the ash content in the fuel, slag stoppages are quite likely to occur. It should be pointed out that these conditions refer particularly to boilers of the inclined straight-tube type with counterflow convection superheaters.

Why have close spacing and high gas velocity been adopted? As the author points out, the superheater for these high steam temperatures must absorb more than 25 per cent of the total heat produced in the furnace, which is an extraordinary job for any kind of a superheater. With this great amount of heat absorption, the unit must necessarily be compact to fit into restricted

[11] "Combustion of Pulverized Coal," by Henry Kreisinger, Trans. A.S.M.E., vol. 60, May, 1938, paper FSP-60-8, p. 289.

[12] Power Engineer, American Water Works and Electric Company, New York, N. Y. Mem. A.S.M.E.

accommodations. To hold the first cost at an acceptable and competitive point, small tubes and close spacings are desirable to obtain maximum rates of heat transfer.

With possible frequency of interruptions in service, caused by stoppages in superheaters of the closely spaced type, purchasers would, of course, be interested in developments which will minimize these interruptions to the greatest possible extent. With this end in view, the first consideration might be to avoid, if possible, this excessive concentration of closely spaced tubes and the second to lower the temperatures of the gas and the amount of ash in suspension entering the superheater surfaces.

If the superheater could be divided into two parts, transferring the final-temperature section to panels in the furnace where the superheating would be done by radiant heat, the remaining convection element would then operate at lower gas temperatures and would be called upon to absorb, say, about 15 per cent instead of 25 per cent of total heat produced in the furnace. Why not substitute sections of steam-cooled walls for water-cooled walls in the furnace, as has already been done in a few installations? It might be necessary to locate these steam-cooled walls in the upper section of the furnace to avoid exposing surfaces of this type to the extreme temperatures of the combustion zone.

Without doubt a combination superheater of this kind would be a great deal more expensive than a straight convection unit. Whether the results, as regards increased availability of the boiler, would be realized and whether the purchaser would feel justified in paying a much higher price is problematical. No doubt these things would take a long time to settle. Be that as it may, the fact remains that the convection-type superheater, in its present arrangement and design, restricts the availability of the modern boiler fired with pulverized coal. Operators can well afford to pay handsomely for an improvement in these operating conditions and they look hopefully to the future for an appreciable reduction in the loss of output and the high costs incident to removal of slag from superheater passages. Meantime the redesign and improvements in the arrangement of tubes in the convection elements of the boiler by the manufacturers may in the near future produce higher availability and an economical solution of slagging in the gas passages of these elements.

Availability and the production of steam-generating units for high pressures and temperatures that will deliver specified rated capacities without interruption for periods of 6 or 12 months or longer must be the objective. The central-station company cannot afford to add spare boiler capacity as was the usual practice ten years ago. There is no serious thought that slagging conditions, as they occur in the high-pressure and high-temperature boilers, will be instrumental in forcing a return to the almost forgotten spare-boiler installation. The results of the author's investigation and research constitute a definite advance in the development of units to cope properly with the slagging problem and should contribute greatly to improvements in boiler design and construction, in order to meet the operators' expectations as regards availability.

E. B. Powell.[13] This paper records an important step in the pathway toward scientific design of the steam-generating unit and one which brings attainment very markedly nearer. The research of Armacost, Gordon, Hottel, Mullikin, Schmidt, Uchida and Tanabe, and Wohlenberg, to mention some of the most recent investigators, was necessary in developing a sound theoretical basis for correlation and interpretation of data and in improving tools and technique. The work which the author describes represents, in the writer's opinion, the most comprehensive factual analysis of the practical aspects of heat transfer in the combustion and immediately related zones of the modern steam-generating unit that has so far appeared. Yet, as the author himself points out, this is but an initial step along a very promising pathway. I am sure we all hope that he may continue as his paper suggests. It would be of much assistance to those others who follow if he would supplement his paper by a specific statement of the methods which he employed in gas-temperature measurement. Precision in such measurement requires complete freedom from interference of surroundings and is particularly important when different surface arrangements are under analysis. The steam-power industry is again indebted to the author for his pioneering work.

[13] Consulting Engineer, Stone & Webster Engineering Coporation, Boston, Mass. Mem. A.S.M.E.

H. P. Reid.[14] The writer is particularly interested in the methods which were used by the author to measure gas temperatures at various locations in the boiler.

The difficulty in obtaining gas temperatures that are representative and usable in studying boiler and furnace operations is fully appreciated. For several years the writer has made use of the high-velocity thermocouple for research and test purposes and would seriously consider application of such thermocouples for continuous records, if a stable and satisfactory design could be obtained.

Table 2 contains representative data illustrating errors that may occur by the use of the protected thermocouple. These data were obtained from recent tests on the performance of a rotary cement kiln and the waste-heat boiler attached to it. Temperatures of the gases leaving the kiln were taken by both protected or shielded thermocouples and also by high-velocity thermocouples. The figures given are average for a number of readings taken during the test period.

TABLE 2 DATA ILLUSTRATING POSSIBLE ERRORS FROM USING PROTECTED THERMOCOUPLE

	Gas temperatures F leaving kiln			Boiler efficiencies, per cent, calculated from	
Test no.	By protected thermocouple	By high-velocity thermocouple	Temperature difference F	Protected thermocouple readings	High-velocity thermocouple readings
273	1465	1590	125	67	62
274	1540	1862	322	73	59

In the tests, Table 2, the waste-heat boilers were operating considerably below rating. This explains the low boiler efficiencies. It is believed they are representative for this particular type of waste-heat boiler installation.

Bernhard Schroeder.[15] The results of tests made on three different installations are of much value but insufficient as a basis for definite conclusions. No doubt more extensive investigations will be undertaken by many operating companies. In this way the problem can be much more rapidly solved for various kinds of coal.

Knowing how difficult it is to obtain reliable data on which to base conclusions, one is inclined to wonder what is the degree of accuracy of the data presented in the paper? For instance, when powdered fuel is blown into a furnace, the various-sized particles of fuel immediately start to distill off the volatile gases. If sufficient oxygen is present, the gases will be burned; if not, they will not be completely consumed. With the amount of excess air ordinarily present in a boiler furnace, such as given in the paper, namely, 14 to 25 per cent, it is not difficult to consider the furnace being filled with gases of almost any composition, together with solid particles of carbon. All of these can only be

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[15] President, Sargent & Lundy, Inc., Chicago, Ill. Mem. A.S.M.E.

burned when coming into contact with the required oxygen. It would appear quite possible for the burning gases to pass through the primary and perhaps through the secondary furnace before finding the necessary oxygen to complete their combustion.

On looking over the author's calculations, it is evident that the combustion is assumed to be completed before the gases leave the primary furnace. If they are not, the calculations are all affected to some extent. Therefore, it is necessary that the completeness of the combustion should be established. In future work complete analyses of the gases should be made to determine this point.

The gases leaving the boiler may show 15 to 20 per cent excess air, but in actual boiler operation the proportions of excess air, as between the various burners, may be quite different. The distribution of coal in the burner itself often leaves much to be desired and, unless the turbulence is very great, a satisfactory fuel and air mixture is not obtained until the gases have entered the boiler tubes or even the superheater.

Until there is some radical change in the art of burning coal, slag in some form will always be present. If it can be determined where the slag shall be deposited, by suitably proportioning the heating surfaces and the gas temperatures, then adequate means can be provided for its removal with a minimum of trouble and expense.

V. J. Skoglund.[16] The data given in this paper are useful as a guide in the analysis of heat transfer by radiation in furnace cavities. It would be ideal if point-by-point heat-transfer rates (Btu per hr per sq ft) and temperatures could be calculated from fuel and air operating conditions for any type of furnace cavity, using only the fundamental laws of thermodynamics, heat transfer, and fluid mechanics. To do this is as yet impossible but, starting with simplifying assumptions based on experimental data, rational analysis may some day make a useful contribution to the solution of practical furnace-radiation problems. At present the furnace-radiation analyst is handicapped by the lack of reliable data on conditions within furnace cavities.

An unsolved analytical problem of practical value is the calculation of point-by-point heat-transfer rates (Btu per hr per sq ft) for specified temperatures and emissivities of gases and suspended particles which vary from point to point within a cavity. The following additional data, taken simultaneously with gas temperatures and the corresponding over-all furnace conditions, are necessary before undertaking the development of a solution for the foregoing problem:

1 Intensity of emission or total emissivity of the mass of gases and suspended particles within the cavity for all possible directions at different points on its surface; there is at present no practical way of measuring this and the first step would be the development of such a method.

2 Measured wall temperatures over all of the surface; in the ash and slag problem as discussed by the author, measured wall temperatures are needed to establish the significance of those deduced from fusion properties of ash and slag samples.

3 Approximate thickness and mean conductivity of the slag on absorbing surfaces; the thickness may still vary with time after a long period at a uniform load.

The author makes the statement, "(*g*) A most important factor that should be known is the heat reflected and radiated from the wall surface." Although a rational solution of the above unsolved analytical problem might be laborious, it would be very valuable inasmuch as it would include the determination of that factor as well as surface temperatures.

The author's method of calculating an "equivalent heat-receiving surface temperature" is very simple but, even though the Stefan-Boltzmann radiation law is used, the method should not be considered rational since the use of an average gas temperature and an emissivity of 1.00 are not consistent with the actual mechanism of radiation. The method is of value only if results can be correlated for a large number of different furnaces and coals. This may require the use of empirical values of emissivity differing in different zones instead of the value 1.00.

The paper includes the statement, "(*e*) It is desirable to know the temperature of slag particles floating in the gas stream, especially as they approach tube banks." Attention is called to the mathematical results of Wohlenberg and Wise,[17] which are a qualitative indication of Δt, the difference between gas and particle temperatures, for points not near cold surfaces. Although their analysis is for very much idealized furnace conditions, after studying their paper, the writer believes the statement in the summary applies to average values in actual furnaces, "Hence for any reasonable length of gas travel in the cavity the ash-particle temperature is substantially that of the gases except for regions close to cold surface." Where flow conditions are unsteady, as is usually the case in actual furnaces, instantaneous values of Δt might be considerable for particles accelerated relative to the gas stream in regions where the temperature gradient in the gas is high.

V. V. Soldatoff.[18] Although the author states that consideration of the efficiency of steam-generating units is not the object of the paper, it is of great interest to the writer to bring out this point when speaking of boiler furnaces. For the combustion of pulverized coal must be provided (*a*) time, (*b*) temperature, and (*c*) excess of air. These last two items are well taken care of in primary furnaces but not the item of time. One way to extend the time, in the writer's opinion, would be to place a furnace slag screen of full capacity in the proper place, so that the sticky particles of ash could be retained there, fused with the fluxes provided. The mass could slide down to the bottom of the furnace.

As long as one function of the primary furnace is the smelting of coal ash, more should be known about it. Besides the chemical analysis, its mineralogical structures should be known. For example, uncombined iron ore in the presence of coal or coke will supply oxygen for burning off the carbon. This would be carried to the upper parts of the boiler, spreading a deposit of dust there. This would not happen if the amount of flux necessary to fuse the ash could be blown in with the pulverized coal and thus complete the slag formation on the furnace slag screen. Bearing in mind that the initial temperature of gases is too high for direct application, the fluxing of ash will be justified.

The author points out that the temperatures of gases entering the generating tubes of the boilers, Figs. 2, 5, 7, and the second pass of the boiler Fig. 8, vary from 2500 to 2670 F. This proves that convection heat emission can be applied to the boilers if the vital portion of tubes be protected by a permanent covering, e.g., "porcelain enamel," and forced circulation of water applied to them. Porcelain enamel, which is a thin layer of refractory, would allow placing the boiler generating tubes immediately after the slag-screen tubes, giving a narrow space for possible dust collection. This would produce an insulating effect, but only for two or three rows of tubes.

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[17] "The Distribution of Energy in the Pulverized-Coal Furnace," by W. J. Wohlenberg and D. E. Wise, Trans. A.S.M.E., vol. 60, 1938, paper FSP-60-19, pp. 531–547.

[18] Consulting Metallurgical Engineer, New York, N. Y.

Convection heating of water-tube boilers is not new. The same principle applies in Steamotive and Velox boilers but without any protection of their water tubes. This method should produce rapid steam generation with considerable economy of fuel and a decrease in the structural cost of the power station.

C. H. SPARKS.[19] The author's researches stress an important point; they illustrate conclusively that space requirements, in the larger units at any rate, cannot be met by drastic alterations to the system of generating the steam. The problem is essentially a furnace problem and the fact remains that the furnace will have to be designed upon principles which remain unaltered irrespective of whether natural or forced circulation is employed.

The author implies that dry-bottom furnaces are suitable for units up to 200000 lb evaporation per hr, but that for units larger than this slag-tap furnaces should be used. The interpretation to be placed on this statement is that the slag-tap furnace cannot be built as economically as the hopper-bottom furnace for evaporations of 200000 lb and under. In practice however, boiler units having hopper-bottom furnaces have been successfully built for evaporations as high as 450000 lb per hr. The heat release at the maximum continuous rating of such boiler units seldom exceeds 25000 Btu per cu ft per hr. Rates of heat release much in excess of this figure are not generally acceptable for the reason that the gases and entrained dust reach the boiler heating surface at a temperature above fusion point. In slag-tap furnaces in order to insure that the ash can be readily tapped, a high rate of heat release is maintained in the primary furnace and this normally results in a boiler approximately two thirds as wide as that which would be accommodated over an equivalent hopper-bottom furnace. The slag-tap furnace can be justified if it can be shown that, at a given load, a lower gas temperature entering the tube bank results, than that which can be attained when burning the same fuel in a hopper-bottom furnace designed for the same evaporation. Severe problems have been encountered in keeping boiler units clean and although deslagging has been successfully accomplished with some of the later designs of water sprays the problem of deslagging is indeed formidable.

The author seems to be of the opinion that the trouble of slagging is accentuated in a high-pressure boiler plant. There may be a number of reasons for this, one of which is that to justify the expense of high pressure, a high continuous load is usually carried, which gives the slag slight chance of breaking away naturally. Another reason may be that the steam temperature is high. This makes it necessary to reduce the amount of boiler-heating surface in front of the superheater so that the slag eventually forms on the small-diameter tubes of the close-pitched superheater surface. It is not improbable that engineers in the near future will endeavor to exploit the savings to be made by the application of high steam temperatures to low and moderate pressures. When this time arrives, somewhat similar slagging conditions on boilers operating at lower pressures may have to be faced.

E. H. TENNEY.[20] Looking back over the varied experiences of the last fifteen years, the ash and slag factor alone has in one installation or another determined such major elements of boiler performance as (*a*) the upper limit of output, (*b*) continuity of service, (*c*) furnace efficiency as limited by excess air, (*d*) the practical and economic value of a specific fuel, and (*e*) the level of operating costs. This paper is an important contribution to the practical analysis of this problem and as such is of interest to designers and operators.

The author suggests that dry-ash furnaces of large size may call for less turbulent burners. If by this is meant high-velocity burners, intended to maintain furnace turbulence, the writer agrees. A truly turbulent burner is one designed to create an intimate mixture of coal with the air needed for combustion at or near the burner. Such treatment has proved essential for the successful use of low-fusion-ash Mid-West coals where high furnace capacities are to be developed. It results in a more rapid heat liberation within a smaller furnace volume. This takes place with a clearer flame than is the case with the streamline burner. The increased time element contributes to the cooling of the ash to a dry condition. The clear flame permits unimpeded radiation to the furnace walls and results in a lower furnace temperature than is the case with "clouded" or so-called "insulated" burning. When furnace temperatures are permitted to reach 2800 F, cooling the ash to 2000 F is difficult. Recourse must be had, as the author points out, to the questionable use of water sprays on high-temperature boiler surfaces for slag removal in the boiler passages.

As he has suggested, it is important to explore the furnace to determine temperatures at various stages of combustion to make certain that the ash is cooled sufficiently before striking boiler surfaces or superheaters. Several years ago the writer made a series of observations on furnace temperatures, comparing turbulent and streamline firing. It was demonstrated that for low-fusing-temperature ash coal, proper mixing of the coal and air led to more rapid combustion and a relatively low-temperature furnace. This eliminated the necessity for creating high temperatures in one part of the furnace (for ignition or melting the ash) to be followed by the troublesome necessity of cooling the gases before reaching the boiler proper.

W. J. WOHLENBERG.[21] The author is to be congratulated on having presented effective visual evidence, by means of motion pictures, concerning some of the phenomena which occur in the pulverized-coal-fired furnace cavity. It was noted, among other things, that, under most circumstances, by far the larger percentage of fuel particles ignited and burned out very soon after entering the cavity. Such evidence confirms the results of calculation based on theoretical analysis of these conditions which are found in reference.[17] Thus it demonstrates, among other things, that the radiant mean positions of burning particles is not far removed from the point at which the fuel particles enter the cavity, when most of the air supplied for combustion enters with the fuel.

The author's method of arriving at surface temperatures is interesting. Investigation of this indicates that the results for the secondary furnace and boiler generating tube zones are not affected greatly by such variations of emissivity of the contents of the cavity which are likely to occur. Thus variation of actual emissivity values, from the assumed value of 1.0 to, say 0.5, changes the results for surface temperatures in these zones by only about 200 F.

However, for the temperature conditions which exist in the primary zone the results of calculation are extremely sensitive to emissivity values. A change here from the assumed value of unity to 0.5 changes the calculated results 2000 to 2500 F. It is essential therefore in this case that accurate data concerning the emissivity be available. The mere assumption that it is close to unity does not appear to be sufficient to justify any conclusions concerning surface temperature values arrived at on the basis of the assumption.

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As shown in reference,[17] ash particles in suspension contributed very considerably to the radiating power, i.e., to the emissivity of the contents of the cavity. This is true even on the basis of a single ash particle for each coal particle. The author implies

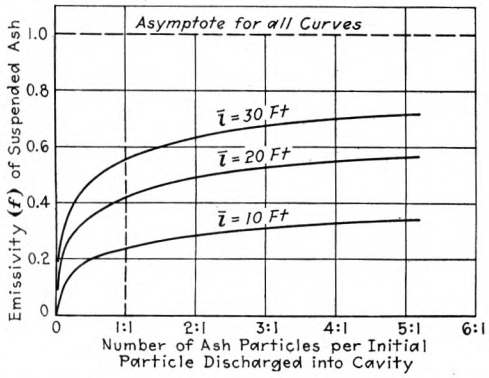


FIG. 11 INFLUENCE OF RATIO OF NUMBER OF ASH PARTICLES, FORMED PER INITIAL PARTICLE FIRED, ON EMISSIVITY OF CLOUD IN SO FAR AS PARTICLES ALONE ARE CONCERNED
(Ash particle surface A per lb fuel fired is 85 sq ft for abscissa ratio = 1:1, gas temperature T_g = 2500 F.A.)

that, on the average, there may be several ash particles per coal particle. If this is the case then radiation due to the ash particles is even more important than the results of reference[17] indicate in this respect. The author apparently based such implications on an examination of coarse particles. But by far the larger number of particles in a pulverized-coal sample are very small; in fact, so small that it is easily conceivable that many of them contain no ash, while others may be entirely ash.

In order to indicate the possible magnitude of effects of the above nature, curves shown in Fig. 11 were plotted. These are based on an application of Fig. 11, of reference.[17] It is assumed that each ash particle is an opaque sphere with a radiationally black surface. The radiant mean thickness of the cloud is $\bar{l}$. Differences of ordinate values, corresponding to differences in abscissa values, thus indicate the maximum amounts by which the emissivity of the cloud may be affected, because of variation in the number of ash particles formed per particle in the initial sample. The real differences would be less than those shown because, as the abscissa ratios increase, the ash particles become so small as to become more and more transparent to radiation. It is thus obvious that, until more is known about this state of affairs, the assumed ratio 1:1 used as a basis in reference[17] is probably safer than any other assumption which might be made in this respect.

Under (*e*) the author makes statements concerning the relation between temperatures of slag particles and other parts of the cavity. In this connection the reader is again referred to certain results in reference.[17] There it is conclusively shown that, thermal conduction between gases and ash particles is so powerful, the average temperature of the ash-particle surface must be very close to the gas temperature. This is true even when each fuel particle yields but a single ash particle. If, as the author implies, there are considerably more ash than fuel particles the difference between the temperatures of gases and ash-particle surface would be even less. The point is this: Thermal conduction between gases and particles has a far greater influence on the temperature of the particle surface than has radiation, if the particles are very small. The smaller the particles the greater is the relative influence of thermal conduction. Hence, the author's statements under (*e*), while probably quite true for the large particles, have less and less weight as the particles decrease in size. For the latter, the temperature will vary with that of the gases. If this varies due to local conditions, then the temperature of the small particle will follow substantially the variations of the gas temperature.

AUTHOR'S CLOSURE

Since the preparation of this paper a great many additional tests have been made, some on the same boilers and with the same kind of coal, others on additional boilers with different kinds of coal, sometimes varying the rating, the excess air, and other conditions. These later data definitely confirm the value of this method of measuring gas temperatures and using for comparison the calculated equivalent heat-receiving-surface temperature. They have shown that the rates of heat absorption in the primary furnaces of the slag-tap type vary closely with the adiabatic temperature and the actual slag-fluid temperature according to the Stefan-Boltzmann law.

Actual fluid temperatures of slag formed in a furnace may differ from those determined from the ash in the coal by the usual A.S.T.M. method due to segregation within the furnace and also due to the varying degrees of reducing and oxidizing influence on the iron in the ash in different parts of the furnace.

High rates of combustion seem to be beneficial from the standpoint of completeness of combustion, high rate of heat absorption, and large recovery of ash collected and tapped from a furnace.

The foregoing comments answer many of the questions raised in the discussion. Messrs. Harkins and Flinn properly bring out the importance of the radiating temperature as well as the receiving-surface temperature. Coal with low total-moisture content burned with high-temperature air and little excess air produces high furnace temperatures and correspondingly high rates of heat absorption. The temperature of the slag on the heat-receiving surface is an additional but independent factor.

Mr. Muir's comments are appropriate, and it is toward a more economical design of all parts of the furnace, boiler, and superheater that this method of testing is being actively continued.

The remarks of Messrs. Sparks and Tenney relate to the features of dry-ash removal. One point should not be overlooked, viz.: As furnaces become larger the relation between furnace volume and heat-absorbing surface changes materially. The Btu available per sq ft of heat-absorbing surface must be considered more than the Btu per cu ft of furnace volume. Long flame and delayed combustion usually give a lower peak temperature than does concentrated turbulent combustion, but as Mr. Tenney says, less slag trouble may result from properly arranged turbulent burners. This point should be further investigated with special reference to excess air and the fusing-temperature range of the ash as actually deposited, in comparison with that accredited to the ash in the coal by the A.S.T.M. method.

Professor Wohlenberg's discussion is much appreciated, and the data presented in the paper are inadequate to clarify some of the points which he has questioned. It is felt certain that additional and more thorough test data from the field will be helpful in learning more about furnace radiation. There is one point about the ash particles that certainly can be clarified by further field data. The author feels sure that there are very few, if any, particles of coal with no ash.

Dean Hitchcock's 38-year milestone makes one apologetic for not having accomplished more in that period of time. It now seems that the greatest hindrance in solving the slag problems of today arises from the fact that we learned too well the characteristics of slag in *fuel beds* and are only now beginning to realize that, when coal is burned in pulverized form, we must revise our methods of determining the ash-fusing temperature and approach the problem from a very different angle. We have

continued to obtain our laboratory data in a reducing atmosphere, and have then deposited the ash on furnace and boiler surfaces under varying conditions, often highly oxidizing with several hundred degrees difference between the two. We are well on the way toward making up for lost time, with a certainty that much more valuable progress is in sight.

Pulverized Coal-Tar Pitch as a Fuel

By E. W. CLARKE,[1] PITTSBURGH, PA.

This paper describes the method of adapting pulverizing equipment to burn fuel pitch, which is the hard residue remaining after the distillation of oils from coal tar when the maximum quantity of creosote has been removed. Early attempts to use this hard product as a fuel were beset with difficulties. However, a practical system was developed by the author in 1922 for pulverizing this fuel in a unit pulverizer and blowing it directly into the combustion space of a boiler through a nozzle designed specifically for its use. Since that time hard pitch has attained wide use as an economical and efficient fuel. The methods of firing fuel pitch are described by the author and specific examples of the results obtained are cited.

THE PRINCIPAL PRODUCTS of coal-tar distillation in the United States of America are road tar, creosote oil, and pitch. Not more than 2 per cent of the tar produced is used in making synthetic resins, dyes, and chemicals. The creosote-oil fraction varies from 15 to 60 per cent of the tar, depending upon the grade of pitch produced and the specifications of the wood-preserving industry in which the creosote is utilized. About 45 per cent of the creosote oil used in treating timber is consumed by the plants that treat railroad cross ties. Piling takes about 5 per cent, paving blocks 1 per cent, and construction timber 7 per cent of the creosote derived.

In 1908 about 55,000,000 gal of creosote were used for timber treatment. This steadily increased yearly to 108,000,000 gal in 1913, of which amount nearly two thirds were imported from Europe. When the outbreak of the World War reduced these importations, the American Tar Products Company, now the Tar and Chemical Division of the Koppers Company, was organized by the American Creosoting Company to supply its creosote requirements and to sell the surplus production in the American market. A ready market awaited the creosote but the opposite was true of the pitch, which is the residue left in the stills after distilling off the creosote from the tar.

This pitch is a liquid residue as produced in the stills and either flows by gravity or is pumped from them. At atmospheric temperatures it varies in character, depending upon the amount of oil distilled off from the tar, from a viscous, semisolid material to pitch hard enough for grinding. The point for stopping distillation is determined by the consistency of the pitch desired or by the specifications for the creosote, depending upon which is of the highest market value. Pitch comprises from 40 to 80 per cent of the tar, depending on the character of the tar and the hardness of the pitch.

The uses of distilled tars or pitches include saturant for roofing felts, protective coatings for iron and steel pipes, binder for aggregate in road construction by both hot and cold processes, surface treatment of roads by hot or cold applications, filler in block pavements, waterproofing walls below grade, built-up roofing, membrane waterproofing, saturating fibrous materials, binder in briquetting fuel, electric carbons, sealing and insulating compounds, core compounds, foundry facings, clay pigeons, and pitch coke.

Since the primary product of the newly organized American Tar Products Company was creosote oil, the bulk of the tar was distilled to give the highest creosote-oil yield, thus producing a hard pitch for which the demand was limited. This resulted in the need for a market to consume the hard pitch produced.

Early Experiments in Burning Pitch

Various attempts were made to use hard pitch as a special fuel. Pitch was mixed with coke breeze or noncoking coals to supply the cementing properties necessary for satisfactory coking. Pitch was mixed with coal and with ashes and used as fuel. The hard pitch had a softening point of 285 to 320 F. Attempts were made to liquefy it by heating and firing it with oil burners. However, all parts of the system had to be held at a temperature of about 500 F to maintain the pitch in a liquid condition. The suspended carbonaceous material, usually about 10 to 20 per cent, clogged up the oil burners and caused the abandonment of these experiments. To lessen these difficulties the liquid hard pitch was cut back with oil tar. At the Chicago plant of one tar distiller liquid hard pitch having a softening point of 280 to 300 F was cut back to a softening point of 145 F with oil tar and used for fuel under all its boilers and tar stills. The oil tar was obtained from gas companies operating water-gas sets. At the Philadelphia plant of the same company the boilers and tar stills were fired with a mixture of one part crushed pitch, having a softening point of 300 F, and two parts of run-of-mine coal burned on hand-fired grates. At the St. Louis plant of another tar distiller, a mixture of 40 per cent crushed hard pitch and 60 per cent coal screenings or coke breeze was used as fuel on Murphy stokers to heat boilers and tar stills.

The nuisance and added cost of these various methods of blending and mixing solid or liquid hard pitch with other fuels restricted such operations to the plants of the tar distillers and precluded the possibility of pitch making any headway in the open market as a fuel.

In 1922 the author took up the task of developing a better way to utilize hard pitch as a special fuel. At that time the opinion was generally held in the industry that the possibilities of burning pitch in pulverized form had been thoroughly investigated and that the difficulties were practically insurmountable. Prior to this, hard pitch had been pulverized for years with impact pulverizers with air separation and sold as the main constituent of core compound, which suggested the possibility of burning it as fuel, after running it through coal pulverizers. No difficulty was experienced in pulverizing it with the same type of machines as those used for core pitch. However, the heat radiated from the furnace caused the pulverized pitch to soften and to cake in the burners, soon plugging them shut.

Investigation of the patent literature showed that R. P. Perry and E. F. Stewart had been granted recent patents for air-cooled and water-cooled burners for pitch. Arrangements were made to borrow a small pulverizer and start experiments. The first step was to make burners such as those described in the Perry and Stewart patent and to conduct experiments with them. It was found that they had some tendency to clog when exposed to any considerable degree of radiant heat from the furnace. This suggested the idea that, if the burner could be located outside the furnace and the pulverized pitch could be discharged from it into

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[1] Chief Engineer, Tar and Chemical Division, Koppers Company.

Contributed by the Fuels Division and presented at the Spring Meeting of The American Society of Mechanical Engineers, held in New Orleans, La., Feb. 23–25, 1939.

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

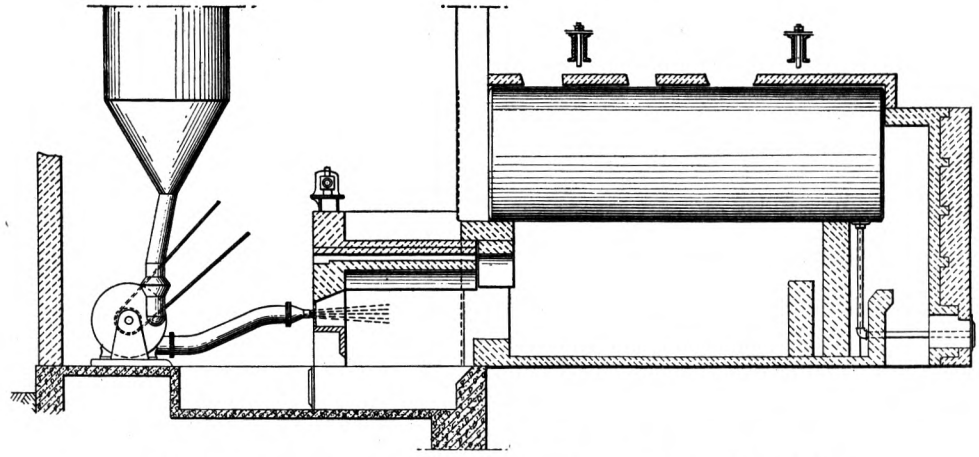


FIG. 1 PULVERIZING APPARATUS DESIGNED FOR BURNING FUEL PITCH

the furnace through an opening of sufficient size and shape so that the pitch would not impinge on any heated surfaces before being burned in suspension, the difficulties might be overcome. This was tried and found to be successful and a patent[2] was granted to the author on February 28, 1928, covering the apparatus and process. Fig. 1 shows the arrangement of the apparatus. The plants of the American Tar Products Company were then equipped to burn pulverized pitch instead of coal.

One thing that the hard pitch lacked was a name to identify it as a new type of fuel. The name "hard pitch," by which it was known to the industry, bore no relation to its use as fuel. Various designations were suggested, e.g., "burning pitch" or "furnace pitch." An excellent name was suggested by the oil industry's nomenclature and hard pitch was henceforth called "fuel pitch." Within a comparatively short time, fuel pitch became a stable article of commerce. The principal characteristics of fuel pitch are given in Table 1.

TABLE 1 CHARACTERISTICS OF FUEL PITCH

Proximate analysis	
Volatile matter, per cent	53.80
Fixed carbon, per cent	46.13
Sulphur, per cent	0.55
Ash, per cent	0.07
Ultimate analysis	
Carbon, per cent	93.71
Hydrogen, per cent	4.12
Nitrogen, per cent	0.98
Oxygen, per cent	0.65
Sulphur, per cent	0.44
Water, per cent	0.12
Ash, per cent	0.10

Softening point, 285 to 320 F
Calorific value per lb = 16,000 Btu
Weight by bulk = 50 lb per cu ft
Grindability, 150 Hardgrove

Fuel pitch can be pulverized in standard types of impact coal pulverizers with air separation. It requires only about one half the power that must be furnished for pulverizing coal and the wear on the pulverizers is negligible. The high volatile content of the pitch causes it to burn with a flame more like that of gas than pulverized coal, enabling the pitch to be ignited without preheating the furnace. Another characteristic is that dust explosions and puffs, which must be so carefully guarded against when burning pulverized coal, are entirely absent with fuel pitch.

Since pulverized pitch cannot be subjected to pressure without binding together, it cannot be used in central pulverizing plants where bins and screw conveyers are provided for the pulverized-fuel distribution. The type of equipment suitable is the unit pulverizer in which the pitch is pulverized and blown directly into the furnace. No preliminary drying is necessary since the moisture content of the pitch is usually less than 0.2 per cent. For the purpose of drying coal in unit pulverizers, it is customary to introduce heated air, but with pitch this is not only unnecessary but contrary to the nature of the material, since the heated air would soften the pitch in the mill and render it inoperative.

In proportioning combustion chambers, the same rules apply to pitch that determine the size for pulverized coal. Since the ash in pitch is about 0.10 per cent there is no difficulty with slag or ash removal. Furnaces fired with fuel pitch can be operated for months without shutting down for cleaning purposes.

MARKETS AND PRODUCTION METHODS

Because of the purity of the fuel, it can be used to replace gas and oil for chemical and metallurgical processes. Its low ash and sulphur content are especially desirable in the manufacture of highly alkaline materials by processes in which the combustion gases come in contact with the materials being produced; for example, high-purity chemical lime and sodium silicates.

A leading Midwestern manufacturer of lime found that the purity of the product from his rotary kilns was improved by firing them with fuel pitch instead of coal, due to the elimination of the ash from the coal, which acted as a source of contamination to the burned lime. Fuel pitch has been used in this plant for about thirteen years. Malleable-iron furnaces which had been fired with coal produced a stronger product when fuel pitch was substituted and this has proved to be another attractive market outlet. A furnace producing sodium silicate was successfully operated with fuel pitch which replaced oil at a saving in fuel cost. It is obvious that the tar distiller can demand premium prices if the use of pitch as fuel results in improved industrial products.

Fuel pitch is now produced either in flake form made on water-cooled drums and steel belts, or in crushed form processed by natural cooling in shallow open-topped steel tanks or bays built on the ground. From these it is reclaimed either by hand labor or mechanical means and then crushed to about $^3/_4$-in. size. From this point on to the pulverizers it is handled by the same type of conveying machinery and storage bins as used for coal. Neither crushed nor flaked pitch will solidify or bind together in bins or conveying systems, which facilitates its handling and makes it possible to ship it by trucks, cars, barges, or steamers without difficulty.

The heating values of fuel oil and fuel pitch are about 20,000 Btu per lb and 16,000 Btu per lb, respectively. On this basis,

[2] U. S. Patent No. 1,660,831.

the comparable values per ton for fuel pitch for various prices of fuel oil are as follows:

Fuel oil, cents per gallon	3	4	5	6	7
Fuel pitch, dollars per ton	6.85	9.12	11.36	14.71	16.00

Typical Fuel-Pitch Installation

The pioneer work with fuel pitch was done at a Wisconsin plant of the American Tar Products Company. After successfully demonstrating that pitch could be burned as a fuel commercially and equipping the boilers and stills at this tar-distilling plant for the purpose, a part of the surplus fuel pitch was sold for boiler fuel to a neighboring company. The boiler-room equipment at this plant included six 402-hp, 200-lb per sq in. pressure Stirling boilers, fitted with Foster superheaters which provided 100 F superheat.

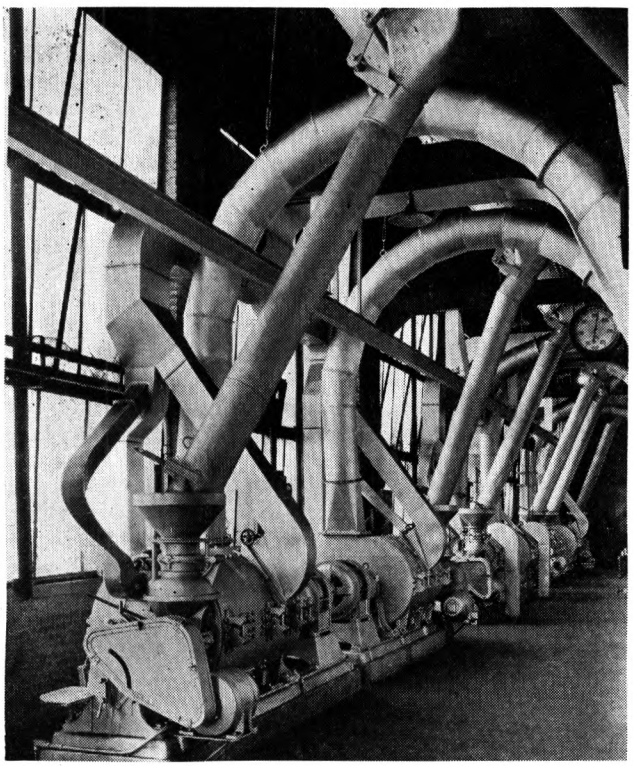


Fig. 2 A Typical Unit-Pulverizer Installation for Burning Fuel Pitch

The boilers were originally fired with stokers using coal as fuel. In February, 1926, the company purchased a unit pulverizer for one of the boilers and after removing the stoker remodeled the furnace by adding a small Dutch oven and lining the ashpit with firebrick.

After the first pulverizer had been installed and its operation compared with stoker firing, the savings were found to be of sufficient magnitude to warrant the installation of additional pulverizers. A second order for a unit pulverizer was placed five months after the first one was installed. In less than six months after the second pulverizer was ordered, a third machine was purchased so that three of the boilers were fired by unit pulverizers. The equipment permitted the use of either fuel pitch or coal as fuel. The pulverizers carried the boiler rating up to 200 per cent without difficulty.

The fuel pitch supplied to this plant at the outset had a softening point between 260 and 280 F, which is lower than that of the fuel pitch produced today. It had to be pulverized in machines where high friction losses were not prevalent and without the use of preheated air. Variable-speed control was used to drive the feed mechanism of the pulverizers in order to assure maximum flexibility. One pulverizer has handled 5500 tons of fuel pitch without the necessity of replacing any of the pulverizing elements. At the end of five months' operation, the second pulverizer had handled more than 2700 tons of fuel pitch without the casing of the machine having been opened.

Under average conditions the furnace temperature was about 2700 F; the CO_2, 14 per cent; the furnace draft about —0.10 in. water gage; and the stack temperature about 540 F at 175 per cent of boiler rating. The ash residue was cleaned out of the furnaces about once in every three months. The fuel pitch was very easy to ignite and a clear stack was readily maintained.

It will be realized from the foregoing that the maintenance costs on the pulverizers were most satisfyingly low. From a psychological standpoint, it is an interesting fact that the boiler-room operators preferred to burn pitch in the pulverizers rather than coal on the stokers.

The evaporation rate was 11.2 lb of water per lb of fuel pitch as fired. The fuel cost per 1000 lb of steam generated has been shown by carefully conducted test to be more than 23 per cent less than with coal fired on stokers. The pulverizers ran for hours at a time without attention. The additional capacity, supplied by the use of fuel pitch in the pulverizers, made possible the elimination from regular service of one of the boilers, even under the maximum demands for steam.

Some Observations on the Yield Point of Low-Carbon Steel

By JOSEPH WINLOCK[1] and RALPH W. E. LEITER,[2] PHILADELPHIA, PA.

A study has been made of the yield-point phenomena in low-carbon steel with the view of determining the mechanism of the transition of the metal from the elastic to the plastic state. Unlike copper and aluminum, low-carbon steel does not yield in tension until an excessively high elastic resistance has been built up. This high elastic resistance is the upper yield point. The upper yield point is believed to be of fundamental importance in the Lüders' line phenomena, and experimental evidence is given which supports this view.

THE peculiar behavior of low-carbon steel at the yield point has long been known and has been the subject of many researches both in this country and abroad. Of special importance has been the work of Nádai (1),[3] Siebel and Pomp (2), Sachs and Fiek (3), Rawdon (4), Kenyon and Burns (5), Gensamer (6), Gensamer and Mehl (7), Davenport and Bain (8), Fell (9), Deutler (10), Brinkman (11), and MacGregor (12).

The question of what is occurring during this singular behavior is not only of interest from a theoretical point of view, but is of great practical importance for at least two reasons: (*a*) The highly localized plastic flow, which occurs during the deep drawing of light-gage steel, gives rise to surface markings which seriously mar its appearance. If the forces causing the deformation are primarily tensile forces, these surface markings occur as depressions in the surface. Whereas, if the forces causing the deformation are primarily compressive, irregular lines of elevations occur. These surface markings are known as Lüders' lines, Hartmann lines, the Piobert effect, and, in the shop, as "stretcher strains." A photograph showing surface markings as depressions is shown in Fig. 1. And (*b*) a clear understanding of the significance and value of the yield point is, it would seem, essential to engineers engaged in the designing of structures.

The purpose of this paper is not an attempt to describe what causes the steel to behave in this peculiar manner, but an attempt to describe what the authors believe is taking place in the steel during its transition from the elastic to the plastic state. Its purpose, also, is to make some observations which will, it is hoped, serve to create a greater interest in this important field.

Deformation of Aluminum

In order to describe more clearly the phenomena which are occurring at the yield point of low-carbon steel, it will help if a description is first given of what occurs when aluminum is elastically and plastically deformed in tension.

There are, of course, two methods of deforming a metal in tension: (*a*) by actually hanging an increasing amount of dead weight on the metal; or (*b*) by deforming it at some given rate or rates by means of oil-pressure cylinders, screws, or the like. With the first method, the load is always present and obviously no drop in load can occur at any time. With the second method, the registered load is the resistance of the metal to the deformation being imposed upon it. The second method is the one most widely used, and is the one referred to throughout this paper.

At the instant deformation begins, the aluminum begins to offer a resistance to deformation and for a certain distance the load-deformation curve rises as a substantially straight line (13). So long as the curve remains substantially straight, the deformation is substantially elastic. But, when the deformation becomes so great that the resistance of the metal to elastic deformation is overcome, plastic deformation takes place. This is shown in Fig. 2. It is to be noted, too, that the metal exhibits a continually increasing resistance to deformation until the maximum load is reached; and also that the curve is smooth.

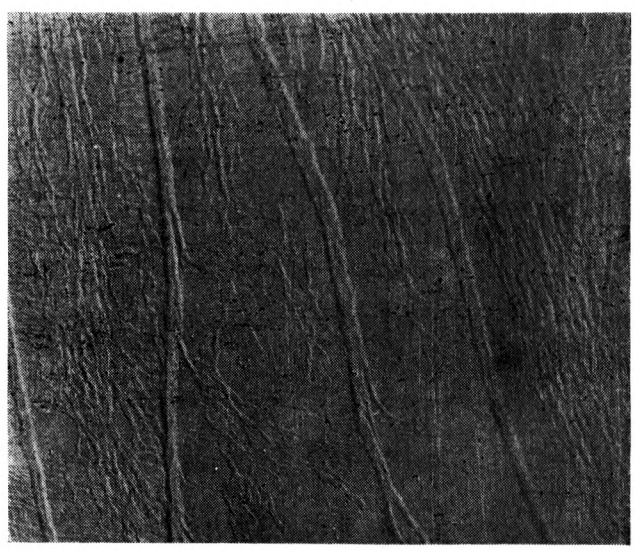


Fig. 1 Lüders' Lines or Stretcher Strains Shown as Surface Depressions

It should be further noted particularly that as the change from the substantially elastic state to the substantially plastic state takes place, the metal immediately offers a greater resistance to deformation, i.e., a higher load is recorded. In this case, the elongation is substantially uniform until the maximum load is reached.

Deformation of Low-Carbon Steel

When low-carbon steel is deformed, the metal appears to behave elastically in much the same manner, but when plastic deformation occurs there is a lowering of the resistance to deformation of the specimen and the registered load is lower. In testing parlance, this phenomenon is known as "the drop of the

[1] Chief Metallurgist, Edward G. Budd Manufacturing Company.

[2] Assistant Metallurgist, Edward G. Budd Manufacturing Company.

[3] Numbers in parentheses refer to the Bibliography.

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Discussion of this paper was closed January 10, 1939, and is published herewith directly following the paper.

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

beam." This is shown by the solid line in Fig. 3. Experiments have shown that substantial plastic deformation does not occur in a uniform manner as it does with aluminum, but occurs first at some point or points and, as the deformation proceeds, spreads until all of the metal has been so affected. The elongation required to produce complete plastic deformation varies with the speed of deformation, but under ordinary conditions is some 5 per cent.

Attempts have been made to analyze this behavior by mathematical equations designed to establish conditions which show when heterogeneous or homogeneous flow can take place. Attempts have also been made to explain the phenomenon by postulating that, unlike aluminum, a sudden weakening occurs in the metal, or, more scientifically, the metal temporarily exhibits a low rate of work strengthening.

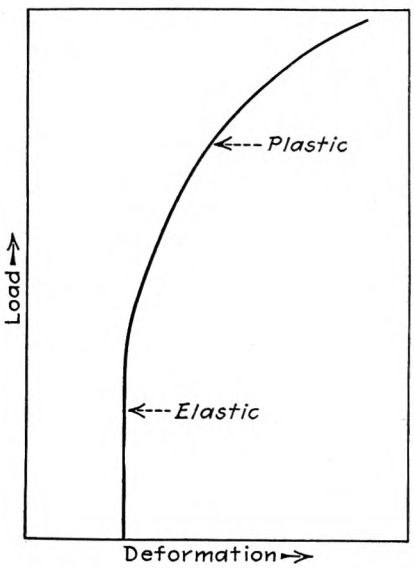


FIG. 2 LOAD-DEFORMATION CURVE FOR ALUMINUM

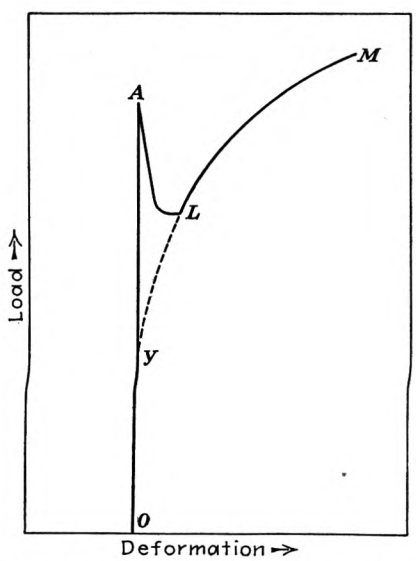


FIG. 3 LOAD-DEFORMATION CURVE FOR LOW-CARBON STEEL

None of these attempts is in accordance with the observations made in the authors' experiments. Mathematical equations cannot be established for this portion of the load-deformation diagram on the basis of homogeneous or heterogeneous flow because neither uniform nor nonuniform plastic flow is occurring! How can either of these be taking place when there are large portions of the metal which are not flowing at all? The authors believe also that there is not a sudden weakening but, indeed, a sudden strengthening of the metal undergoing plastic deformation.

It appears that, for steel to behave like aluminum, substantial plastic deformation should take place at the load Y in Fig. 3 and the load-deformation curve should be the curve $OYLM$. For some unexplained reason, however, the load Y is "overshot" and substantial plastic flow does not commence until the load A has been reached. This point has long been known as the "upper" yield point. Plastic deformation does not occur throughout the specimen at once, however, but first at one point. Therefore, at the instant plastic deformation commences, the plastically deformed metal finds itself overloaded, so to speak, with a resistance to further plastic deformation of only the load Y. The further behavior of this first increment of metal is dependent entirely upon its reaction to the precarious and unstable situation in which it finds itself. Because of the high load, it must deform plastically at a high speed, and it does so. The amount of metal forced to a great speed of deformation is governed by the amount of elasticity available in the other parts of the specimen and of the testing machine; and it is for these reasons that the downward path of the load-deformation curve veers away from the portion of the curve denoting elastic deformation.

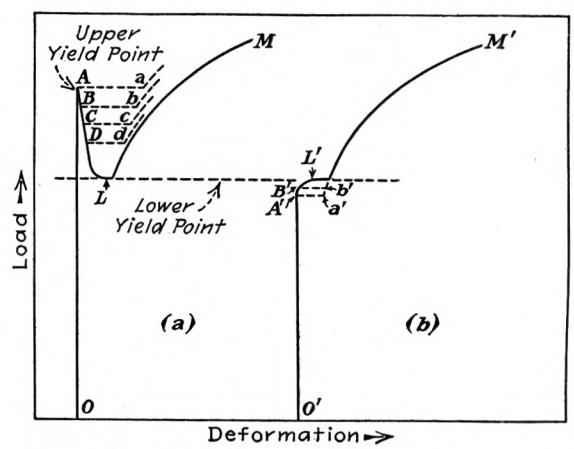


FIG. 4 LOAD-DEFORMATION CURVES WITH AND WITHOUT UPPER YIELD POINTS

The following is a more detailed explanation of what the authors believe occurs: When the load A, Fig. 4*a*, is reached, the first increment of metal deforms plastically at the high load which it has built up for itself while in the elastic state. And, since it now has the resistance to further plastic deformation of only the load Y of Fig. 3, it does what it is forced to do, namely, travel at a high speed along the dotted line A-a. At the next instant two other small increments, one on either side of the first, deform plastically at the load B an amount equal to B-b. The load on the specimen drops—first, because the metal has only the resistance to further plastic deformation of the load Y; second, because there is a concentration of stress at either end of the local constriction; third, because there is now present some metal which has been cold-worked, i.e., plastically deformed (the effect of this will be described in more detail later); and fourth because the increment's speed of travel is greater than the head speed of the testing machine.

The total load registered by the specimen never drops to the load Y but only to the load L, known as the "lower" yield point,

because a balance is reached between (*a*) the increase in resistance of the metal undergoing plastic deformation, i.e., work strengthening; (*b*) the increase in resistance of the metal because of the high speed of travel; and (*c*) the presence of metal in other parts of the gage length which has only been elastically deformed, and which must first be stressed to a value corresponding to the upper yield point before plastic deformation can occur in it too.

The extremely important point to emphasize is that, at the instant substantial plastic deformation is initiated, it is not necessary to have present the load which the metal has built up for itself while in the elastic state in order for further plastic deformation to occur. As opposed to this, the usual view, it seems, leads to as much information of the inherent properties of the metal as would be obtained by hanging a million-pound weight on a steel bar having a diameter of a quarter of an inch. After fracture, the bar would be considered (with a reduction in area of, say, 40 per cent) to have broken at a stress of the absurd figure of some 30,000,000 lb per sq in. This, of course, is true, but certainly no such stress as this was needed to cause the metal to fracture.

It has previously been considered that uniform elongation began when the load-deformation curve commenced to rise again at the termination of the yield-point elongation. But, if the authors' views are correct, uniform elongation does not begin until a load is reached at which all parts of the specimen have deformed the same amount that the first increment has deformed. In other words, the cross-sectional areas of the specimen taken when the end of the yield-point elongation has been reached should gradually increase from the smallest cross-sectional area of the metal first to deform plastically, up to the largest cross-sectional area of the metal last to deform plastically. And, furthermore, these areas should correspond to the elongations at the points *a*, *b*, *c*, *d*, and *L* of Fig. 4*a*. These points should not, of course, lie on the curve *LM* because of the greater speed at which these increments are deformed, and because of the short time these high loads existed.

Test on Steel Specimen

The following experiment was made in order to substantiate this. A highly polished and carefully annealed tensile specimen of low-carbon steel was prepared. The specimen was approximately 0.037 in. thick, had a uniform section 2.25 in. long and 0.5 in. wide, and the gage length was 2 in. The edges of the specimen were carefully smoothed and partially rounded to avoid any concentration of stress. It was then carefully aligned in a tension-testing machine (14) especially designed to give a constant head speed. The specimen was then pulled at the extremely low head speed of 0.0046 in. per min until the yield-point elongation had been completed. As is well known, Lüders' lines always start at the fillets of a test piece free from surface imperfections, notches, or the like, because of the inevitable concentration of stress at these points. For the purposes of this experiment, however, it was necessary to cause the Lüders' line to start near the middle of the gage length. Accordingly, two extremely small prick-punch marks were carefully made before the test on either side of the specimen at this point. These marks were, however, so small that an upper yield point was not removed from the load-deformation diagram.

During the test and while the load was dropping, pencil marks were made on the specimen at the boundaries of the spreading Lüders' line and at the same time were placed on the charted load-deformation curve. The specimen was then removed from the machine and careful measurements were made with point micrometers. After this, the specimen was replaced in the testing machine and, at the same head speed, was pulled a considerable distance beyond the yield-point elongation. The curve obtained is shown in Fig. 5.

TABLE 1

Location see Fig. 5	Width, in.	Thickness, in.	Area, sq in.	Change in area	Change in area, per cent
Original	0.4927	0.03740	0.01845	...	...
A	0.4885	0.03570	0.01743	0.00102	5.53
B	0.4890	0.03625	0.01775	0.00070	3.79
C	0.4890	0.03655	0.01788	0.00057	3.09
D	0.4890	0.03660	0.01790	0.00055	2.98
E	0.4880	0.03670	0.01791	0.00054	2.925
L	0.4880	0.03670	0.01791	0.00054	2.925

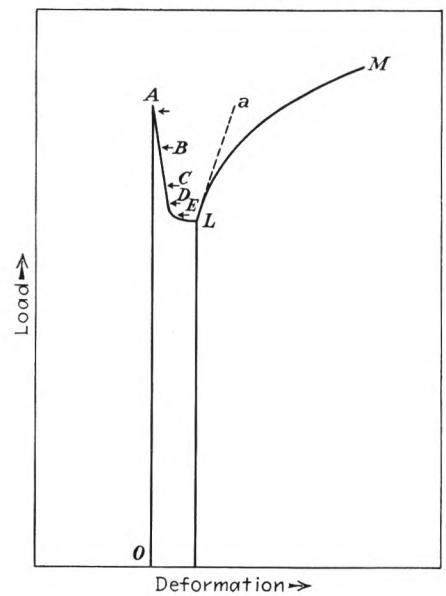


Fig. 5 Load-Deformation Curve for Data of Table 1

The measurements obtained are given in Table 1 and from these data the dotted line *L-a* in Fig. 5 was drawn. The small arrows indicate the loads which correspond to each set of measurements. The reason why this line does not coincide with the actual load-deformation curve has already been mentioned, viz., the greater speed at which these increments were deformed. These experimental data, it seems to the authors, definitely substantiate their views. These findings can be further supported by what is believed to be sound deduction of the existence of an upper yield point, and of the vital importance of this point in Lüders' line phenomena.

It is necessary, first, to describe a few experimental observations. As has been mentioned, the first substantial plastic deformation to occur is extremely local in character. That is, regions of only elastically deformed metal and regions of plastically deformed metal are present at the same time. This has been conclusively shown by placing extensometers on the elastically deformed regions and, after removing the load, noting that they return to zero. Careful visual observation of highly polished specimens show, also, no traces of plastic deformation in these regions.

It has also been conclusively shown by others and more recently by the present authors (14) that the horizontal portion of the load-deformation curve can be raised or lowered to an extraordinary degree by changing the speed of deformation. In spite of this, the same phenomena take place in exactly the same manner. The only difference is that the depth of the Lüders' lines is greater the higher the speed of deformation.

Now, since there are, during the stretching of a tensile bar, some regions which are only elastically deformed, these regions are not substantially in motion; that is, the elongation is taking

place only at the boundaries of the Lüders' line. And, since the horizontal portion of the curve can with changes in speed of deformation occur at almost any load level, an extremely pertinent observation, it seems, can be made: How is it possible for the only elastically deformed stationary regions to support the high loads produced by the deforming boundaries of their plastically deformed neighbors when these same elastically deformed regions would have yielded at a lower load had the speed of deformation been lower? In other words, how can the elastically deformed regions support, during a test carried out at a high speed, a load which is far greater than that which they could support if the test were carried out at a low speed? The absurdity of such a situation appears to be obvious. The only elastically deformed regions, therefore, must be able to withstand, in order to conform to this otherwise anomalous situation, a much higher load than that at which plastic deformation is occurring at their boundaries. If they could not, they too would yield at the same time. This higher load-carrying capacity of the elastically deformed regions results, the authors believe, is the phenomenon known as the upper yield point.

A further observation concerning the upper yield point follows: The load-deformation curve *OYALM* in Fig. 3 can always be obtained in annealed low-carbon steel of small or moderate grain size in either cylindrical or flat specimens if (*a*) the edges of the flat specimen are carefully rounded by polishing; if (*b*) the fillets on the specimens are large; if (*c*) great care is exercised in properly aligning the specimen in the testing machine; if (*d*) the surface is free from pits or scratches and other surface imperfections; and if (*e*) the specimen is free from any cold-worked material which might be present as a result of the machining operation, some local abrasion, or bending.

If, however, any or all of these precautions are not observed, an upper yield point is not obtained on the load-deformation curve. A localized spreading plastic deformation occurs, nevertheless, in exactly the same manner, but the resistance of the metal to the initiation of plastic deformation is lower, i.e., the registered loads are lower, and the loads at which the increments are deformed are lower. See Fig. 4*b*, lines *A'-a'* and *B'-b'*. And, as would be expected, the local constriction is less deep. Although the registered loads are lower, it is believed that the stresses are exactly the same as when an upper yield point is obtained. It is important to note from this figure two significant facts: (*a*) the horizontal portion of the curve, i.e., the load *L* is the same for a given steel and a given speed of deformation whether or not an upper yield point is obtained; and (*b*) the elastic limit does not coincide with the yield point as it does in Fig. 4*a*, but occurs at a lower load.

The Rate of Work Strengthening and the Stress-Strain Curve

The phrase "rate of work strengthening" is usually defined as: the slope of the stress-strain curve. The authors think that this definition is not a desirable one because work strengthening can only occur after plastic deformation has taken place. Indeed, it is plastic deformation which creates work strengthening. With this definition, however, the origin of the work-strengthening curve must be taken as the stress existing at the highest load which the steel can offer while in the elastic state. That is, the origin of the work-strengthening curve must be chosen as a stress existing when the metal is not in the plastic condition. According to this definition, then, the stress-strain curve through and somewhat beyond the yield-point elongation would be the curve *OYAPM* in Fig. 6. From this, it would be necessary to conclude that because the slope of the portion of the curve *AP* is low, the rate of work strengthening is low. It would seem that this definition leads to an erroneous conclusion.

The authors believe that the elastic condition must be totally divorced from the plastic condition in order to arrive at the proper rate of work strengthening. So long as there are regions of both conditions present at the same time, the true behavior of the metal in the plastic condition is obscured. As the authors see it, the influence of the upper yield point included in the definition should be removed. And the definition of the rate of work strengthening should be: The rate of increase of only the stress actually necessary to cause further plastic deformation in metal which has already been plastically deformed. To clarify, in Fig. 6 the rate of increase of stress actually necessary to cause an increment to deform plastically through the yield-point elongation is not the curve *AP* but the curve *YP*. That is, the stresses actually needed to cause the metal to deform through the yield-point elongation are the stresses S_1, S_2, S_3, ... The curve showing the rate of work strengthening then becomes the curve *YPM*. And it can be seen that the slope of the curve is greater during the yield-point elongation than at any other time. With this observation in mind, there is no necessity for postulating a negative rate of work strengthening (6) when the steel is tested in compression (in which no reduction of area is possible).

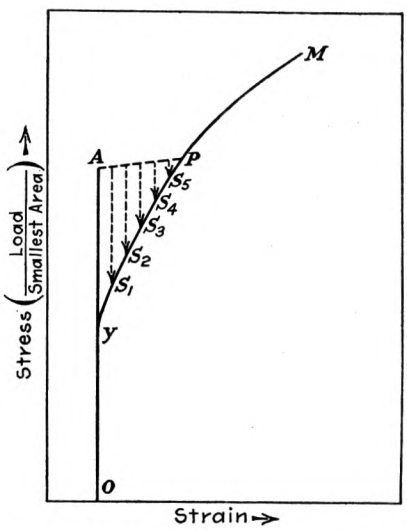


Fig. 6 Stress Strain in Relation to Work Strengthening

Other observations also indicate that this is correct. Hardness readings taken at the bottom of Lüders' lines formed at different speeds of deformation show the hardness curve to be rising in a normal manner. Of great importance, too, is the fact that if the steel is cold-worked slightly before testing in either tension or compression, all traces of the peculiar yield-point phenomenon completely disappear. The steel then behaves in exactly the same manner as aluminum, as shown by the curve *OYLM* in Fig. 3. That is, the load-deformation curve of cold-worked low-carbon steel is smooth (15), and substantial plastic deformation occurs at a much lower load.

Practical application of this has been made in the deep-drawing industry and special machines have been developed (16) to cold-work the steel a small amount before the deep-drawing operation is performed. In this condition, the steel does not show Lüders' lines (stretcher strains) after deep drawing and, because the deformation takes place uniformly, the deep-drawing properties are considerably improved.

The problem of what stress should be chosen for the yield point and the significance of the yield point raise questions which have not been given the attention they deserve. Experiments show that if an upper yield point is observed, the propor-

tional limit, the elastic limit, and the yield point occur at the same stress. On the other hand, if no upper yield point is observed, the elastic limit and the proportional limit occur at a considerably lower load, and the yield point, i.e., the horizontal portion of the curve, occurs at a slightly higher load than this.

What value, then, for yield point is important for engineering purposes: the stress at the elusive upper yield point (the stress at the drop of the beam); or, the lower yield point which, in ordinary testing procedure, is seldom noted? Should the horizontal portion of the curve be taken as the yield point when no upper yield point occurs? Is the stress at the load Y in Fig. 3 related to any other property of the steel, such as its fatigue strength? Of what significance to engineers is the upper yield point? Can it be obtained reliably? Is the absence of an observed upper yield point indicative of severe internal stress? (Locally cold-worked carefully prepared smooth specimens show no upper yield point.) Will a steel be thought to be of inferior quality because of a low proportional and elastic limit due only to failure to obtain an upper yield point? And, finally, what is the cause of the upper yield point anyway, i.e., the extremely high elastic strength of steel? These questions urgently need answers.

The authors wish to acknowledge the friendly and helpful advice and criticism of Dr. A. Nádai whose important work in this field is well known; and they wish to thank their colleague Dr. R. Eksergian for his interest and encouragement.

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16 U. S. Patents Nos. 1,649,704; 1,649,705; and 1,649,706.

Discussion

E. A. Davis.[4] In the latter part of this paper, the authors object to using the slope of the first portion of the stress-strain diagram, line A-P in Fig. 6, as a definition of the "rate of work strengthening." This objection, the writer believes, is entirely justified, for the rate of work strengthening should be a measure of the manner in which the stress is increased due to the effect of strain alone. If the material is such that the stress under which it yields is affected by changes in the speed of deformation, then these influences must be taken into consideration before the rate of work strengthening can be determined. Ordinarily, this can be avoided by stretching the specimen at a constant strain rate. In a like manner, if the properties of the material change with time (e.g., age hardening, annealing, etc.), the stress under which it yields will also change. This influence must be taken into account in measuring the rate of work strengthening. This may be a negligible factor in short-time tests, but it becomes important in evaluating the results of long-time tests.

In the case of mild steel or wherever localized yielding occurs, it is practically impossible to test any given fiber or element at a constant strain rate. For example, the speed of the movable head on the testing machine can be adjusted so as to give 5 per cent over-all strain in 5 min, yet each element may strain 5 per cent in less than 1 sec, and very little can be said about the uniformity of the rate of strain during this interval. If it were possible to select some infinitesimal gage length so that the strain inside this length would be uniform and, if this could be tested at a constant strain rate, then the slope of this stress-strain diagram should furnish a true "rate of work strengthening." This curve can be made to fit the ordinary curve in the region near M in Fig. 6 by choosing the strain rate the same as that in the ordinary test in this region. In the part of the diagram usually known as the yield-point elongation, the curve for the constant strain-rate test will be below the line A-P because of the slower strain rate. This curve should correspond in some way to the curve $OYPM$ shown in Fig. 6. Just what the shape of this curve should be does not seem obvious to the writer, and he would like to ask the authors if they will further explain their reason for drawing the curve between points Y and P as they have.

A. V. de Forest.[5] This paper again draws attention to the great difficulty of interpreting the yield-point phenomenon both from the point of view of the metallurgist and of the engineer. One reason for this is that, due to the limitations of our present test machines and method of recording the stress-strain diagram, there is no clear distinction between the idiosyncrasy of the machine and the properties of the metal under test.

In this work and in many previous papers, attention has been drawn to the upper yield point and the conditions necessary to measure its value. This is found to depend greatly on freedom from stress concentrations due to shape of the test specimen, preparation of its surface, and particularly on eccentricity of loading. It is never measured at all under commercial testing conditions where special precautions are not taken. The value of the lower yield point is similarly never measured because of imitations in the speed of response, both of the weighing mechanisms of the testing machine and the strain-measuring equipment. Furthermore, both points are sensitive to test speed and to temperature. It is, therefore, high time that the whole problem be reexamined and if possible, an agreement reached as to how this point shall be defined and measured for engineering

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[5] Professor, Massachusetts Institute of Technology, Cambridge, Mass. Mem. A.S.M.E.

purposes; how it is to be interpreted by engineers as a criterion of strength, and by metallurgists as a particular mode of deformation exhibited only under special conditions by special materials.

J. J. KANTER.[6] The authors showed in a previous paper[7] that the upper yield point of low-carbon steel sheet is influenced to an amazing degree by changing the rate of straining. They demonstrated a consistent rise in upper yield point with increase in straining rate for sheet steel about 0.036 in. thick of approximately 0.05 per cent carbon content prepared to five different grain sizes.

The fact that the upper yield point increases with increase in rate of straining suggests that the effective stress distribution at the yield point becomes increasingly homogeneous on a microscopic scale as the stress is applied. At stresses up to the upper yield point it is a fair approximation to say that the mode of deformation is essentially elastic and of uniform distribution. Beyond this point plastic deformation proceeds heterogeneously as the authors have demonstrated in Fig. 1. As stress increases the elastically deformed structure becomes increasingly less stable. Its existence as such becomes a transient condition. As the straining rate is increased, "snapshots" may be caught of this elastically deformed condition at somewhat higher loads, just prior to the plastic breakdown, which propagates at a rate governed to some degree by temperature.

Suppose the elastic straining were caused to proceed at a greater rate than that at which the propagation of plastic breakdown proceeds. Under this condition the microscopically homogeneous elastic-stress distribution might be preserved up to the rupture strength of the steel. Such a condition might well obtain in high-velocity tension impact tests, such as made by H. C. Mann[8] at the Watertown Arsenal, in which 0.35 per cent carbon steel exhibited a transition from plastic deformation of high energy absorption to brittle fracture of low energy absorption at impact velocities of about 30 ft per sec per in. of gage length. At striking velocities in excess of 30 ft per sec, it may well be that the upper yield point of the steel may be pressed to the point where it approaches the elastic cohesive limit of the material.

Consideration of the mathematical possibilities of dealing with the effect of straining velocities on yield point suggests at once the methods of statistical mechanics. We are dealing with a system, the elements of which are individually capable of undergoing an elastic type of deformation up to a cohesive limit. The energy necessary to reach this limit is, after a fashion, a measure of the individual yield point of the element involved. The upper yield point, which is measured for a particular aggregate of these elements subjected to a certain rate of straining, is a manifestation of a statistical average of states of strain. Due to the instability of the elastic state of deformation, once the stress has reached the point of instability, it is to be expected that but a fraction of the stressed elements simultaneously sustain the elastic strain necessary to attain the true upper yield point. The faster the rate of strain, the larger will be the proportion of elements simultaneously capable of sustaining the full elastic-cohesive effort without relaxation into slip. The kinetic oscillations of the particles comprising the steel conceivably contribute to the elastic instability which brings about yielding. Under rapid straining rates it is to be supposed that the kinetic oscillations are given a lesser opportunity to influence the elastic displacement and thus the unyielded steel is given a greater chance to persist at higher stresses.

If it be granted that the probability of kinetic disturbance underlies the inception of yielding in steel, it is to be inferred that the Maxwell-Boltzmann distribution law might be applied to correlating upper yield points with the straining rates at which they occur. In this law the chance of a number of eligible elements, out of a large number, being involved in an event is shown to be exponentially proportional to a quantity representing a ratio between the actual energy required to produce the event and the average energy available. In the case at hand, the event with which we are dealing is elastic deformation to the full cohesive effort without degradation to slip. The probability that is expressed by the exponential function should involve the proportionate occurrence of such an event. The exponential quantity will be that ratio between the observed upper yield point and the true elastic-cohesive effort at which the elements of the steel reach a brittle rupture.

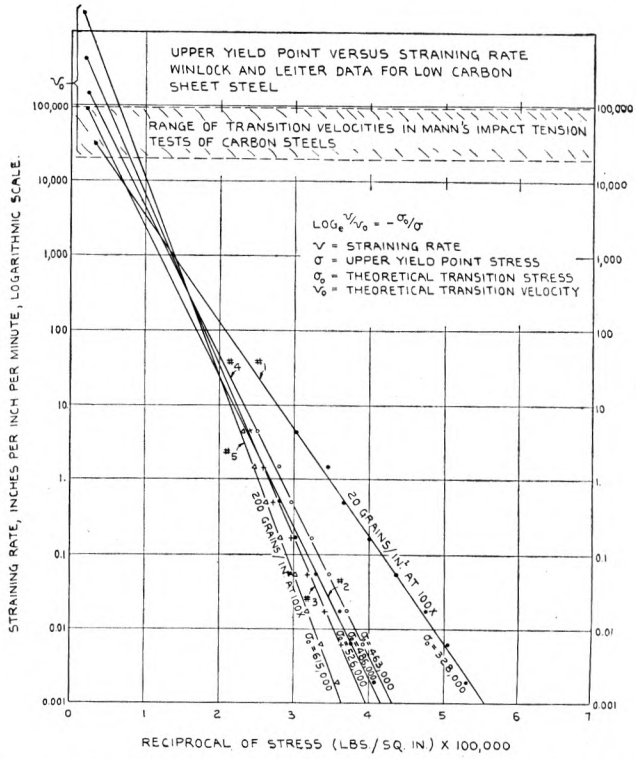


FIG. 7 CLOSE AGREEMENT BETWEEN STRAINING VELOCITY ESTIMATES AND RANGE OF TRANSITION VELOCITIES FROM MANN'S VARIABLE-VELOCITY IMPACT TESTS ON CARBON STEELS

The testing of this theory upon actual data suggests employing a semilogarithmic plot of log straining rate as a function of the reciprocal of the upper yield stress. A straight-line alignment upon such a chart fulfills the requirements of the Maxwell-Boltzmann distribution law. The straining rate V is, in effect, a measure of the total number of events which occur during a time interval, while the upper yield stress σ is the average of all conditions of stress acting. The true elastic-cohesive effort of which the steel elements are capable will be a constant, σ_0, and associated with yielding under this full effort will be a straining-rate constant, V_0. The equation relating upper yield point and straining rate is thus

$$V/V_0 = e^{-\sigma_0/\sigma}$$

[6] Research Metallurgist, Crane Company, Chicago, Ill.

[7] "Some Factors Affecting the Plastic Deformation of Sheet and Strip Steel and Their Relation to the Deep-Drawing Properties," by J. Winlock and R. W. E. Leiter, Trans. of the American Society of Metals, vol. 25, Mar., 1937, pp. 163–185.

[8] "High Velocity Tension Impact Tests," by H. C. Mann, A.S.T.M., vol. 36, part 2, 1936, p. 85.

The five sets of data for effect of straining rate on yield point which the authors have published are strikingly correlated by this expression. The range of values for the constant σ_0 is in accord with similar values arrived at through other considerations. Values for this same sort of constant may be estimated from the breaking strength of cold-worked wires. The measurements made by E. A. Davis[9] on the effect of loading rate upon upper yield point, together with the Winlock and Leiter results on effect of straining rate, afford another means of estimating the magnitude of constant σ_0. The ratio between loading rate and straining rate is a value dimensionally equivalent to σ_0. Computations of this ratio from the results given for materials of similar physical characteristics, at equivalent yield points, give values for σ_0 of the same magnitude as determined from the slopes of the log straining rate versus reciprocal yield-stress plot.

If the value of σ_0 is accepted as representative of a transition-point yield stress, the straining velocity, V_0, associated with the point may also be estimated. The values of V_0 so computed are in surprisingly good agreement with the range of transition velocities found in Mann's variable-velocity impact tests on carbon steels at Watertown Arsenal as charted on the semilog diagram. See Fig. 7.

These observations suggest that additional work should be done to correlate the effects of straining rates on yield points over wider ranges of velocity than have been reported upon. The semilog chart shows a considerable gap of straining velocities between those explored and those corresponding to the range of impact velocities which have been studied. It would be a matter of considerable theoretical as well as practical significance if the Maxwell-Boltzmann distribution equation could be demonstrated to fit the yield-stress–straining-rate data over the entire range from tensile-testing speeds to impact velocities.

Authors' Closure

The authors wish to thank Mr. Davis, Dr. de Forest, and Mr. Kanter for their very helpful discussions.

[9] "The Effect of Speed of Stretching and Rate of Loading on the Yielding of Mild Steel," by E. A. Davis, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 60, Dec., 1938, p. A-137.

The reasons for choosing YP (a continuation of the curve PM, Fig. 6), as the curve indicating the rate of work strengthening at the beginning of plastic deformation are as follows: (*a*) The curve $OYPM$ is the stress-strain curve of metals, such as aluminum and copper, which show no Lüder's-line phenomena. (*b*) Hardness readings taken at the bottom of Lüder's lines, formed at a different speed of deformation, show the hardness curve to be rising in a normal manner. (*c*) If the steel is cold-worked very slightly before testing, the stress-strain curve has the shape shown by the curve $OYPM$. (*d*) The slower the speed at which the test is carried out, the shorter the yield-point elongation and the lower the load at which the horizontal portion of the load-deformation curve occurs. (There is, however, a yield-point elongation at even the lowest speed of deformation.) At decreasing testing speeds, a continuation of the curve LM, Fig. 3, approaches the elastic portion of the curve. Even though its position is lowered, its general shape is not changed.

The authors agree with Dr. de Forest that the upper and lower yield points are sensitive to test speed and to temperature, but do not believe that the ability to obtain the lower yield point accurately is limited because of the speed of response, etc., of the testing machine, when ordinary testing machines and ordinary testing methods are used. (It is, of course, difficult with a beam-type machine to obtain accurately the load to which the resistance of the metal drops unless the yield point is of considerable magnitude, but with a hydraulic machine this value can be obtained readily.) The rigidity of the testing machine, etc., affects the position of the downward course of the load-deformation curve, as the authors pointed out in their paper, but the ordinary testing machine does not affect the value of the lower yield point *if a constant load is obtained during the yield-point elongation.*

The authors are especially indebted to Mr. Kanter for his carefully prepared and constructive discussion. It would be well to emphasize, however, that no increase in the magnitude of the elastic forces due to speed alone can take place. The raising of the upper yield point by speed must then be due either (*a*) to the load-deformation curve not being elastic up to the upper yield point, or (*b*) to some sort of locking of the crystalline grains which is affected by the speed of test.

Five Years' Progress of Oil and Gas Power

By HOWARD E. DEGLER,[1] AUSTIN, TEXAS

The last "Five Years' Progress of Oil and Gas Power" has been primarily that of constantly lowering the engine weight, decreasing the bulk, increasing the flexibility, and lowering the installed cost per horsepower. With these worth-while improvements also came better performance, lower fuel consumption, and a more quietly operating engine. These gradual changes and improvements have broadened the field of application to such an extent that, from the first development, running into hundreds of pounds per horsepower and turning over at from 100 to 150 rpm, engines have been reduced in weight and increased in speed to an extent that now offers possibilities for universal application. Undoubtedly, the Diesel engine is recognized today as one of the most economical and reliable prime movers known, and is fast taking its place in applications for which it seemed unsuited a few years ago.

THE LAST five years, 1933 to 1938, have been devoted to the development of many new lines of lightweight high-speed engines and the improvement of the established heavy-duty types of oil and gas engines. The total installed horsepower has been doubled during this period and the field of application of oil and gas power has been extended enormously. The best performance of modern Diesel engines will go as low as 0.35 lb of fuel per bhp-hr; in fact, one foreign 5500-hp engine uses only 0.329 lb per bhp-hr with 18,000-Btu fuel, corresponding to a thermal efficiency of 41 per cent. Several Diesel aircraft engines have shown the remarkable economy of 0.32 lb at full load and cruising speed.

Diesel-Type-Size Standardization

The Diesel engine has been subject to progressive refinement and to continued increases in size. The largest Diesel engine in the world was installed in 1933 in the Copenhagen, Denmark, electric-light plant. This unit is of the solid-injection two-stroke-cycle double-acting type; each of the eight cylinders has a 33-in. bore and a 59-in. stroke; and at 115 rpm the engine delivers 22,500 bhp. One manufacturer has offered to build a 40,000-hp engine; another engine of 80,000 hp has been proposed for installation on a ship. America's largest units are the 24 × 36-in. 7000-bhp eight-cylinder two-cycle double-acting HOR engines at Vernon, California, and Salt River, Arizona. Larger capacity per unit has been sacrificed by other builders in favor of the greater simplicity of construction of single-acting engines that results from freedom from stuffing boxes and crossheads, and greater simplicity of piston cooling and of valve-operating mechanism.

Above 1000-bhp and below 4000-bhp capacity, two-cycle engines are favored almost three to one; and above 4000 bhp this type is used exclusively. At present there are no four-cycle double-acting engines, but of the large two-cycle units about half are double-acting. Considering only capacities within the range of both types it is clearly a choice between the simplicity and accessibility of the single-acting engine and the smaller space requirements and smaller number of cylinders for a given power of the double-acting type. All large Diesel engines have vertical cylinders. In sizes between 300 and 1000 bhp, the use of the two- or four-cycle engine is a matter of choice; many builders make both types.

Marked advances have been made in standardization, not only in the general design of the Diesel engine, which has been toward simplification of lines and centralization of control, but also in the design of many of its component parts. Rotative speeds range from 2000 rpm for small high-speed engines to from 120 to 150 rpm for large heavy-duty Diesel engines, and piston speeds from 1500 fpm in small engines to 1000 fpm in large ones.

Gas-Engines and Gas-Burning Diesels

The gas engine has found new ways to make the most of the numerous pipe-line extensions bringing gas fuel to many parts of the country. Once principally used in steel mills, gas plants, and oil fields, gas-powered units now find acceptance in the general power field. The modern gas engine like its oil-engine counterpart is generally a vertical multicylinder unit operating at higher speeds. It is versatile, burning many gases including sewage-sludge gas and butane. The largest vertical gas engine is a 2200-bhp unit at Carlsbad, New Mexico, with eight 23 × 28-in. cylinders. It operates on the four-stroke cycle at 240 rpm. The effect of the present economic trend upon oil- and gas-engine design is shown by the introduction of an increased number of convertible Diesel-gas engines capable of operating on either liquid or gaseous fuels. The gas-injection two-cycle engine with improved scavenging features is giving as good fuel economy as the four-cycle types without the loss of any of its inherent advantages of simplicity, compactness, low first cost, and low maintenance.

One of the outstanding Diesel-engine developments made in recent years (1936) is a dual-fuel engine of the two-stroke cycle type (fundamentally air-injection with port scavenging) that will burn either oil or gas. The advantages of this engine are appreciated when it is understood that the unit can actually burn either fuel without making any major changes in the engine, and further, that either fuel is burned on the Diesel two-stroke cycle. With an eight-cylinder 1500-bhp dual-fuel Diesel engine installed in the municipal power plant at Lubbock, Texas, comparison of the cost of producing power with oil or gaseous fuel shows the advantage to be greatly in favor of the latter. With natural gas at 13.5 cents per 1000 cu ft, the average gas-fuel cost was $0.00172 per kwhr, while with oil as used by other Diesel engines in this same plant costing 3.1 cents per gallon, the cost was $0.0032 per kwhr. This amounts to a reduction of 46.2 per cent for the gas-burning Diesel engine.

Heavy-Duty Diesel Progress

Considerable progress has been made in the development and application of alloy materials and welding, and in precision manufacturing methods. Controlled pressure lubrication, more careful consideration of fuel-injection methods and combustion, and better scavenging have also contributed much to the perfection of modern engines. Diesel-engine installations in communities have been silenced and isolated to prevent noise and vibration. Adequate intake and exhaust silencers are available, as well as engine mountings using cork, helical springs, and the like. In general, more stress has been put on better filtration of

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

fuel oil, lubricating oil, and air before entering the engine. The near future will undoubtedly see the general application of the exhaust-gas turbosupercharging of four-cycle Diesel engines.

While there has been a trend in recent years toward lightweight high-speed engines, some manufacturers have not departed from the heavy-duty type of Diesel engine with which they have been so successful. Undoubtedly, they are not convinced that these newer trends will give the kind of performance which the more conservative designs are giving and which many users of Diesel equipment expect. Many of the Diesel engines now produced in the United States are being built under foreign patents, although American ideas of simplicity and accessibility are constantly modifying the designs. The greater use of large Diesel engines is one of the trends that is noticeable in this field; but America, because of conditions more favorable to the use of coal and natural gas, lags behind Europe in its adoption of the larger sizes. Weights per horsepower range from 25 to 300 lb with an average of 80 lb.

Other design trends, based upon unprecedented progress during this five-year period, indicate higher speeds, further weight reduction, greater unit output, supercharging, greater use of double-acting cylinders, more compactness, greater use of small engines, more gas-Diesel units, incorporation of accessories in the engine enclosure, the use of welding for frames and other parts, more complete enclosure, simplified control, higher mean effective pressure, increased use of normally wasted engine heat, combination of Diesel engines with existing steam plants to improve heat balance, wider application as auxiliary power to main generating units, also the starting, controlling, and stopping of Diesel units automatically in accordance with load requirements or with a minimum amount of personal supervision.

Some authorities doom the four-cycle engine, because the two-cycle type has no valves, has smaller cylinders, requires less floor space, weighs less, has lower first cost, and needs less maintenance. Others think it is not doomed, claiming that two-cycle units present greater cooling problems, are harder to lubricate, have lower mean effective pressures, require excellent scavenging equipment, frequently require supercharging, and present some cylinder-wear problems.

Each Diesel-engine manufacturer seems to have a different idea of the grade of fuel oil required for his product. Hence, ever-increasing demands have been made upon the petroleum industry to supply various grades of fuel oil for the many types of Diesel engines. During the last five years several engineering societies (The American Society of Mechanical Engineers, American Society for Testing Materials, and the Society of Automotive Engineers) have undertaken for the first time to prepare specifications for standard Diesel fuel oils for the mutual advantage of engine designers, manufacturers, and operators, and the petroleum industry. A tentative classification covering five grades of Diesel fuel was issued by the American Society for Testing Materials in September, 1936. Purchase of fuel according to this classification with resultant criticisms and comments should aid in developing a permanent series of specifications for Diesel fuel oil. The diversity of opinions regarding specification limits and the relative importance of the various items indicate the need for improved and continued cooperative work between engine builders, engine users, and oil refiners. In any case, for the good of the industry, the engine owner should be spared the trouble of purchasing fuel to too-rigid specifications.

Until the last five years, development of speed-regulating equipment for Diesel and other internal-combustion engines had lagged behind that of governors for the other two principal prime movers, steam turbines, and water turbines. The recent application of the hydraulic governor to Diesel and gas engines has resulted in close speed regulation and constant stability over the entire speed range. A hydraulic governor does not depend for its power on centrifugal force or inertia acting on a large mass, hence the size of the unit can be considerably reduced as compared to an equivalent mechanical governor for the same engine.

Smaller Diesels for Wider Applications

Many authorities see the greatest future of the Diesel engine in motive power such as that for portable industrial service, long-haul truck and bus transportation, lightweight high-speed trains, and other uses wherever its compactness and efficiency are effective. In marine work, the Diesel engine means low-cost operation and no fire hazard; in airplanes, greater dependability, reduced fuel cost, longer cruising range, and reduced fire hazard. But the automobile presents a different problem. Car owners in the United States today pay for easy starting, quick acceleration, surging speed, and smokeless exhaust but not fuel economy. Diesel engines have not proved as satisfactory for these desired characteristics as the present-day gasoline engine, but medium-compression spark-ignition oil engines approximating the Diesel principle may be used in the near future and make possible the wider adoption of Diesel engines for automobiles in this country.

Some of the long-established Diesel manufacturers devoted the last five years to the development of completely new lines of high-speed engines and to the improvement of their existing heavy-duty units. These companies and other manufacturers are producing high-speed Diesel engines that compare favorably with gasoline engines in power, speed, size, weight, ease of starting, performance, and reliability. Many of these units have been used in tractor, truck, and portable construction equipment; in 1937 over 80 per cent of the total number of engines manufactured had a capacity of less than 100 bhp. As compared with gasoline engines, the possibilities are evident when the following specifications of a typical high-speed Diesel engine are considered: Four cylinders, $3^3/_4 \times 4^1/_2$ in.; piston displacement, 199 cu in.; rating, 62 bhp at 2600 rpm; weight, 12 lb per bhp; and electric starting.

Not since the inception of railway travel has any mode of transportation aroused the public interest and captivated the imagination of railroad men so much as the first Burlington Diesel-powered *Zephyr*, the lightweight passenger train which on May 26, 1934, made its historic long-distance nonstop run from Denver to Chicago in 13 hr 5 min at an average speed of 77.6 mph.

Two developments which have been of great significance to the career of the high-speed Diesel engine, and have perhaps made that career possible, are the use of improved bearings of new alloys and the advance in the refining and compounding of lubricating oil. Not long ago all bearing metal was of babbitt, very satisfactory in its way for slower-speed heavier engines. The introduction of bearing metals containing silver and higher percentages of lead has permitted great increases in allowable unit bearing pressure on main, crank, and piston-pin bearings. Comparable results have been obtained in a different way by the use of "needle bearings," or roller bearings with rollers of very small diameter. Engine maintenance has been simplified by the use of what are termed "precision bearings," namely, thin steel bearing shells lined with a very thin layer of bearing metal. Such bearings require no fitting and when worn are thrown away and replaced.

A few years ago Diesel-engine lubricating oil was of the uncompounded mineral type, of either paraffinic or asphaltic base. Now there are addition agents available for the promotion of a number of desirable qualities. Some of these addition agents increase oiliness under normal conditions; others promote film

strength in order to preserve lubrication under abnormal conditions. Ring sticking, not long ago the cause of much trouble with high-speed units, has been materially decreased by the use of other additives. Unfortunately some of these additives are corrosive to some of the newer bearing alloys, but petroleum technologists are constantly at work on the problem and a solution is probably only a matter of time.

Auxiliary Combustion Chambers

With increasing speeds the principal design problem of any Diesel engine is in its injection and combustion system, and this has been the subject of much study and experimentation for the purpose of obtaining clean combustion and low rates of fuel consumption. Experience has shown that a high degree of turbulence in the combustion chamber just prior to fuel injection is necessary to obtain the desired results; however it was found that application of the methods employed for low-speed engines was not practicable in the case of high-speed units. Hence a new departure seemed necessary and accordingly various forms of auxiliary combustion chamber have been evolved to produce this turbulence. This chamber is generally located in the cylinder head and is completely water-cooled.

The auxiliary combustion chamber is in no sense a precombustion chamber, but contains all of the clearance volume, except that necessary immediately over the piston in order to secure sufficient mechanical clearance. Gratifying results have been obtained with this type of combustion chamber in that it has led to cleaner combustion and higher mean effective pressures. This design removes the spray valve from the center of the cylinder head, thereby permitting the use of one intake valve and one exhaust valve of large size instead of two of each, as is necessary with the centrally located spray valve if restriction of air flow is to be avoided. The location of the spray valve at the side of the cylinder head makes it accessible and shortens the length of the fuel line from the injection pump.

A departure from conservative ideas is represented by the development of the two-cycle high-speed opposed-piston Diesel engine. The compactness of the engine and its efficient scavenging will continue to be contributing factors in the wider application of these engines, especially for submarine and aircraft purposes.

Standardization of Engine Parts

An interesting feature of modern Diesel engines, especially high-speed types, is the increased use of parts made by specialty manufacturers. This applies not only to injection equipment and governors but also to bearings, valves, and, in the smaller engines, to pistons and cylinder liners. This makes possible the furnishing of parts made of superior materials at considerable savings in cost. In these smaller engines, drop-forged connecting rods are used. Their use effects a large saving in the weight of reciprocating parts and makes possible the safe operation of these engines at much higher speeds than could be attained with rods having ends of the marine type.

Trends in High-Speed Engines and Aircraft Engines

The future high-speed industrial Diesel engine will probably develop up to 150 bhp per cylinder, weigh from 10 to 15 lb per bhp, run at speeds up to 3000 rpm with piston speeds of 2000 fpm, and occupy less than $1/2$ cu ft per bhp. Present choice between four-cycle and two-cycle engines appears to favor the former. The argument that, because the four-stroke cycle has become the generally adopted system for high-speed gasoline engines, it must also be better for high-speed Diesel engines, is without value. Inasmuch as the simple two-cycle Diesel engine offers great possibilities with a high specific power output for a relatively small floor space, designers will consider more seriously ways and means of improving scavenging methods and lowering the fuel consumption of this type of engine. Design problems to be considered include: Higher compression ratios to insure a high temperature; increased speed of injection and good turbulence of the incoming fuel charge; a combustion chamber shaped to give rapid heating of the fuel spray; a long free fuel spray; cooling of the injection nozzle; use of the automatic or unit-injector type of fuel nozzle; and the all-important problem of scavenging.

The high-speed Diesel engine possesses many inherent qualities that render its application to aircraft purposes not only attractive but also desirable from the points of view of safety and economy. The German Junkers line of liquid-cooled opposed-piston Diesel engines has done more than all other engines put together to popularize the Diesel engine for aviation. Their first flight was made in 1929, but since that time development and successful application have been rapid. These engines are used extensively on the fast twin-engined (600 to 800 bhp each) ten-passenger airliners in daily service in Europe and on the flying boats which carry mail across the South Atlantic. It was with a Junkers four-engined flying boat, that the world's long-distance seaplane record was broken early in 1938 in a nonstop flight of over 5000 miles from England to Brazil. The weight of these engines is about $1^3/_4$ lb per bhp with Prestone (liquid) cooling.

At the present time the Junkers organization has its 1200-bhp engine on test and is proceeding with the construction of a 2000-bhp Diesel which may be ready for its tests in 1939. The 1200-bhp unit will be suitable for transatlantic and long-range aircraft, and in this respect will be the equivalent of our 1500-bhp gasoline engines because of smaller frontal area and lower fuel consumption. Paul H. Wilkinson, one of America's foremost Diesel aircraft-engine authorities states, "The 2000-hp engine will be far ahead of anything we have, or will have, in the United States. With a diameter of 39 in. and a weight of less than 1 lb per bhp, this powerful liquid-cooled two-cycle engine will be hard to beat. When its crankshafts are turning over at 3000 rpm there will be 144,000 power impulses per min from the 48 pistons of this square-type 24-cylinder engine—approximating the smooth power flow of the turbine."

With the exception of the work carried out by Guiberson in the air-cooled radial-engine field, aircraft Diesel development in the United States has not been supported by the government. The Guiberson Company has been fortunate in being able to continue this development, but its engines of 340 bhp are still on the restricted list. A modification of this engine is a 265-bhp unit recently adopted by the Ordnance Department to power medium-size tanks, for which service this type of engine has many decided advantages over gasoline engines. Other present-day American designers who have developed Diesel aircraft engines of considerable promise are Deschamps, Godfrey, and Lawrance. It is unfortunate that the government is not actively and financially supporting these developments which are so vital to our national defense and also to the safer and more economical operation of our commercial air lines.

Discussion

Oliver F. Allen.[2] Professor Degler has given a fair and constructive résumé of Diesel progress in this country during the last five years. In condensing such a report into the space and time available, an author must understress certain phases of the development which others might have elaborated upon.

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Without in any way criticizing his excellent paper, the writer wishes to call attention to some aspects of Diesel development which may justify a little more attention.

Considering developments in this country, in England, and on the European continent, it seems to the writer that the only outstanding progress has been made in land transportation.

Five years ago a few Diesels had been installed experimentally in busses and trucks in the United States. Only one or two Diesel manufacturers were then giving serious consideration to automotive applications. But at the Motor Truck Show, Port Authority Building, New York, last month, seven makes of commercial Diesels were shown for truck and bus service in sizes from about 100 hp up to 300 hp including both four- and two-cycle designs. There have been installed within the last few months, or are now on order, between 300 and 400 of the bus and truck engines of one make shown there.

In England, from a few trial installations five years ago, the use of Diesels in passenger vehicles has had such rapid growth that 5656 compression-ignition vehicles were registered in England at the end of October, 1937. This was an increase of over 2000 compared with a year before.

For many years, several hundred switching locomotives powered by Diesel engines have been in use here and abroad and Diesel-powered passenger trains were in regular service in Europe and in experimental service in this country prior to 1933 but, as the author points out, the long-distance runs of the light-weight passenger trains in 1934 inaugurated a development in passenger rail transportation by Diesel engines which has been growing by leaps and bounds ever since. This development is not limited to the United States although this country is the leader in long-distance high-speed Diesel passenger trains, some of which regularly reach schedule speeds exceeding 100 mph and go thousands of miles without changing locomotives.

There are now in service a considerable number of units of about 900 bhp at about 750 rpm of both the two- and four-cycle types. However, these weigh from $1^1/_2$ to 2 times as much per horsepower as the truck and bus engines. They are assembled to make many 1800-hp and some 3600-hp locomotives, and the designs of more powerful ones are already developed both here and in Europe.

Furthermore, in England there are already at least three Diesel-engine models which are now available for private automobiles.

Since 1933, land transportation by Diesel power has emerged from the experimental stage into a major industry.

The writer heartily agrees with what the author has said about the cleanliness and ease of operation of the gasoline passenger automobile as compared with the Diesel and, in this country where the cost of fuel is not a great factor, it is difficult to see how the Diesel pleasure vehicle can advance very rapidly. In Europe where the fuel cost is a much more important factor, there will be a more rapid development.

The development of double-acting Diesel engines has not been unqualifiedly successful and there is still considerable hesitation about using them except for special applications. On the one hand, the simplicity and other advantages of the single-acting type influence many users to restrict Diesel plants to such unit sizes as are available in single-acting engines; and on the other hand, recent improvements in small high-pressure high-temperature steam installations tend to decrease both the size of the unit and the size of station in which the steam plant is generally more attractive than the Diesel plant.

The author is perhaps too conservative in his reference to rotative and piston speeds. At least one European manufacturer offers four-cycle single-acting engines of over 2000 bhp at 250 rpm and 1180 fpm piston speed for regular power plants. Small light-weight engines are now designed for piston speeds up to 2000 fpm.

In his reference to gas engines, the author referred incidentally to the use of sewage gas in sewage treatment works. The number of these installations is increasing and their size often reaches well over a thousand horsepower in a single plant.

As is pointed out in the paper, more attention is being paid in this country to the supercharging of four-cycle engines than ever before. Attention is also being given to the supercharging of two-cycle engines. After all, the question of supercharging versus using a larger engine is primarily economic. If the combination of first cost and operating cost for a given application is less with the supercharged engine than with the conventional engine, the user will naturally take the supercharged combination, but if there is no economic gain he will prefer the simpler conventional design.

Closely affiliated with supercharging, as the term is ordinarily used, is the problem of better scavenging and higher volumetric efficiency with resultant higher mean effective pressures in conventional engines both two- and four-cycle. These are important aspects of employing higher pressures for supplying air to engine cylinders which have frequently been overlooked.

In referring to the two-cycle engines, it should not be overlooked that there are successful engines of this type both here and abroad using a combination of ports and poppet valves in the heads; in some designs the scavenging air is admitted through poppet valves and in others the exhaust is through the poppet valves.

As to fuel, there has been a tendency on the part of manufacturers to produce engines designed to use fuel having almost inflexible characteristics and therefore compelling the user to buy always a particular fuel for which the engine is designed. The continued growth of the Diesel-engine business would be facilitated by a policy on the part of designers and manufacturers of producing engines of greater flexibility in the matter of fuel requirements.

In the matter of improvements in speed control, it should be pointed out that there are Diesel central stations now in regular operation supplying thousands of secondary clocks which keep time accurately, and that both control equipment and Diesel engines are now available which will give the uniformity both in voltage and frequency needed for the best laboratories and equal to that supplied by the big steam stations.

The paper refers to auxiliary combustion chambers which are quite the fad now for high-speed engines. Precombustion chambers also continue to be used. In spite of the extensive use of some of these special designs, the plain open combustion chamber such as is used in the Junkers engine continues to be favored by many designers of good engines.

The standardization of injection equipment has been carried forward successfully by a few of the leading manufacturers as to their own product but, in view of the wide variety of pump designs, the many methods of controlling the metering and injection, as well as the wide divergence in nozzles and injection timing and pressures, it can hardly be said that injection equipment as a whole has been standardized.

The writer has felt for a long time that the life of the light-weight high-speed engine would be less than that of the old heavy-weight conventional engine, but some figures that have come to his attention during the last several months have made him doubtful. Now it rather looks as if the higher-speed light-weight unit, due to better design, better utilization of materials, and other characteristics, will have a maintenance cost per unit of power rather less than that of the old conventional engine.

In the matter of fuel economy, there seems to be very little

difference between the two- and the four-cycle engines of equally good design. For instance, the Junkers engines have a fuel economy quite equal to that of any conventional four-cycle engine.

PAUL H. WILKINSON.[3] The writer would like to add a few supplementary comments to the excellent paper of Professor Degler, which it is hoped will be of value in substantiating that section dealing with the Diesel aircraft engine. The author refers to the use of the Diesel in daily service in Europe, and to its flight across the ocean. Data which have recently been received from Deutsche Lufthansa, the only major international air line in the world using Diesel-engined aircraft, confirms his statements and should prove interesting inasmuch as they represent the result of several years' practical experience with the engine under varying conditions of air-line operation.

One interesting set of figures gives the mileage flown with Diesel-engined aircraft on its air lines. Starting with 1634 miles in 1931, there has been a substantial increase. In 1937, a distance of 1,166,023 miles was flown with the Diesel in regular scheduled operation. For 1938, the mileage should show a still greater increase inasmuch as they have further expanded their activities with this type of power plant.

With regard to the number of hours flown, Deutsche Lufthansa's figures show that up to the end of 1937 a total of 46,767 hr had been flown with aircraft equipped with Junkers Jumo 204 and Jumo 205 Diesels. The number of these engines in active service at the end of 1937 was 48, apart from engines in reserve. The fuel consumption obtained was 0.35 lb per hp per hr under regular air-line operating conditions.

While at first the overhaul period for the Diesel aircraft engines was every 200 hr, Deutsche Lufthansa states that this period has now been increased to 350 hr which compares favorably with the 500-hr period recently attained with the gasoline engine. It must be remembered that the Junkers Jumo Diesel cruises at approximately 80 per cent of its total power output compared with about 60 per cent power output for the gasoline engine, which accounts for most of the difference between their overhaul periods.

According to the latest information on the subject, Deutsche Lufthansa has found the Diesel to be satisfactory for short-range, medium-range, and long-range flights. With regard to short-range flights, approximately 28,000 miles a week were flown with the Diesel in 1938 over air routes in Europe, none of which exceeded 425 miles in length. On some of these routes stops were less than 50 miles apart which effectively discounts assertions that the Diesel cannot be used to advantage on any but long-range flights. The equipment used for the most part comprised standard Junkers Ju-86 ten-passenger air liners powered with two 600-hp water-cooled Junkers Jumo 205-C Diesels.

Flights that are now considered to be in the long-range category for gasoline-engined aircraft, are within the medium-range category for Diesel-engined aircraft. A typical example is the weekly air-mail service operated by Deutsche Lufthansa between Bathurst in British Gambia on the west coast of Africa and Natal in Brazil. Since 1936 this ocean flight of 1890 miles has been flown with Diesel-engined aircraft. In 1938 four twin-engined Dornier Do-18 flying boats powered with 600-hp Junkers Jumo 205-C Diesels with a pay-load capacity of 800 lb have been carrying the mail across the South Atlantic on regular schedule.

Numerous long-range flights with Diesel-engined aircraft have been carried out by Deutsche Lufthansa across the North Atlantic. This has given people in the United States an opportunity to see the Diesel in regular scheduled service. In the course of these transatlantic flights they completed 48 nonstop crossings over the 2400-mile route between New York and the Azores, a 100 per cent record which far exceeded the achievements of the gasoline-engined airplanes of Pan American Airways, Imperial Airways, and Air France in this field of endeavor.

The total mileage flown by Deutsche Lufthansa's Diesel-engined mail planes in the course of their survey flights across the North Atlantic amounted to 152,360 miles compared with 27,720 miles for Imperial Airways, 24,370 miles for Pan American Airways, and 8600 miles for Air France over the transatlantic route. The fastest eastbound trip of one of the four-engined Blohm & Voss Ha139 seaplanes with its four 600-hp Junkers 205-C Diesels was made in 11 hr 53 min at a speed of 202 mph. Westbound, the fastest trip took 13 hr 40 min at a speed of 176 mph. These seaplanes carry a crew of four and have a pay-load capacity of 1210 lb between New York and the Azores.

For transatlantic operations in 1939, Deutsche Lufthansa is taking delivery of a number of four-engined Dornier Do-26 flying boats powered with 600-hp Junkers Jumo 205-C Diesels. It is their intention to fly these airplanes nonstop between New York and Lisbon, covering the 3400 miles between these points in about 17½ hours. A crew of four and a pay load of 2500 lb, consisting of five or six passengers and 1000 lb of mail, can be carried on these flying boats which will take off from the water as is customary for passenger craft. The Dornier Do-26 flying boat cruises at 193 mph, and has a maximum flight range of 5600 miles.

This information received from one of the World's leading air lines relative to the performance of Diesel-engined aircraft, should be sufficient proof that the aircraft Diesel can do all that is claimed for it. In view of the tremendous strides being made with the engine abroad, it is to be hoped that the United States government will not continue to jeopardize our progress in aviation by "restricting" low-powered Diesel aircraft engines such as the Guiberson, or by withholding financial aid from those who are competent to build large engines suitable for long-range commercial and military aircraft.

EDGAR J. KATES.[4] The author is to be congratulated upon his success in handling the difficult job that was assigned to him. In the writer's opinion, two additional items of information deserve to be recorded.

First, in the matter of fuel-oil standardization, the A.S.M.E. Oil and Gas Power Division is entitled to recognition as having originated the movement the recent progress of which the author cites. In June, 1928, G. H. Michler of the Standard Oil Company of New Jersey read a paper on Diesel fuel-oil specifications at a meeting of the Oil and Gas Power Division held at The Pennsylvania State College. The discussion of this paper resulted in the formation of a subcommittee of the Oil and Gas Power Division. The committee functioned actively and one year later, again at an Oil and Gas Power Division meeting at the same place, Wiley H. Butler presented a progress report of the A.S.M.E. Research Committee on Diesel Fuel Oil Specifications (the committee into which the original committee had been transformed). The cooperative society work now in progress is a direct outgrowth of that start made by this Division.

The second matter which the writer wishes to mention is in connection with aircraft Diesel engines. Without attempting to detract in any way from the success and commercial progress that has been made by the Junkers engine, one should not forget that the Packard Company in this country designed and built a Diesel aircraft engine by means of which the world's first long-distance Diesel airplane flight was made in May, 1929. The author mentions the present-day American designers who have

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

developed Diesel aircraft engines of considerable promise. This would be an appropriate time also to pay tribute to the memory of Captain Woolson of the Packard Company who was responsible for the development of that first American aircraft engine and who died in the cause before his work was done.

H. W. BUKER.[5] With reference to the statement about "piston speeds of from 1500 fpm in small engines to 1000 fpm in large engines," the writer believes that 1200 fpm would be more nearly correct for the piston speeds of large engines. For example, the 22,500-hp engine in Copenhagen has a piston speed of 1130 fpm and an outside limit of 1200 fpm on big engines is not out of the way at the present time.

L. N. ROWLEY, JR.[6] With a paper as authoritative and comprehensive as this, it is difficult either to correct statements or to amplify them. However there are one or two small points that might bear further discussion.

First, with reference to the data on two- and four-cycle engines: The author states that "In sizes between 300 and 1000 bhp, the use of the two- or four-cycle engine is a matter of choice; many builders make both types." This might be construed to mean that in this size range, both types are more or less equally used. However, an analysis of annual tabulations of stationary Diesel-engine installations for 1936, 1937, and 1938 shows that the four-stroke cycle is used about twice as often as the two-stroke, while below 300 bhp the ratio is even greater. This is not meant as a recommendation for either type; it is merely a statement of fact based on present practice.

Mention of the combination of Diesel engines with existing steam plants is extremely interesting. Of course this combination may be equally effective in a newly designed plant. In either case the Diesel is a welcome aid to the power-plant designer faced with the problem of matching prime movers to power and heating or process loads. By adding to the number of possible combinations, it makes it easier to meet specific cases effectively and several Diesel-steam plants are demonstrating this fact at the present time. On the other hand, such a combination means a heavier investment in many cases and this added cost must be compared with possible operating economies before coming to a decision.

As a general thing more attention is given today to the accessory equipment that goes to make up a complete Diesel plant; technical developments in this field have kept pace with engine improvements. When accessory equipment is overlooked or given inadequate consideration, it is probably because the internal-combustion engine is considered as practically a complete power plant in itself. Present indications are that this kind of thinking is passing as it becomes generally recognized that careful selection of accessory equipment pays in operating efficiency and freedom from troubles.

Outstanding perhaps, are improvements in jacket-water cooling equipment. Cooling-tower size has been decreased for a given capacity and special designs for indoor or other close-quarter installations have been developed. The evaporative condenser, second cousin to the cooling tower and widely used in air-conditioning practice, has been applied to jacket-water cooling, with compactness and low first cost listed among its advantages.

Indicative of a trend away from past hit-or-miss methods of operation is the increasing use of instruments, particularly pyrometers.

Possibly stimulated by the efforts of the Oil and Gas Power Division to accumulate reliable data on performance and costs, plant operation is becoming more systematic. Maintenance is scheduled and maintenance needs and costs are carefully studied. More operating engineers are keeping comprehensive records of both cost and performance and these records are used as a guide to purchasing, operation, and maintenance.

M. P. CLEGHORN.[7] The author deserves commendation for the excellent paper he has presented. It gives a splendid review of the advances in this field in the last few years. It would seem, however, that in a discussion of this type mention might well be made of some of the places where advance has not been made as it should have been.

Much has been said regarding lower fuel consumption and better operating characteristics, but except for one brief sentence no mention was made regarding engine improvements for burning the low-grade fuels. As a matter of fact, engines are demanding better fuels which in turn mean more expensive fuels and this reacts against the present advantage of the Diesel engine in the small power plant.

Every time a new advance is made in the quality of a fuel, the engine manufacturer steps up the performance of his engines to take advantage of this new fuel, even though the fuel cost is greater. The cry for power-plant-Diesel performance to approach that of the automotive Diesel has led the engine builders to ignore the caution they should consider against still further narrowing the field of possible fuels.

The mere fact that fuels of high quality are becoming available generally for vehicle engines makes the engine purchaser feel that there will always be an adequate supply of such fuel and that the price should remain stable. But little account is taken of the probability that the demand for high-grade fuels will be greater for transportation needs and may overshadow the demand for the low-cost fuels for power plants.

The chemistry of fuels has already reached the point where practically all the low-grade stocks can be transformed into high-grade fuel, with a further curtailment of the supply of heavier grades. This can only result in an increased price charged for the lower-grade fuel. Further use of gas oils in furnaces and for household heating brings this fuel into a favorable sellers' market with a continuous trend toward higher prices. Any improvement therefore in the ability of the power-plant Diesels to handle low-grade fuels will relieve the pressure for high-grade fuels and slow up the gain in price.

At the present time high-grade Diesel fuel for trucks and busses is selling at retail in the Middle West for from 11.5 to 12.1 cents per gallon. Since the two main advantages of the Diesel engine are lower fuel consumption and lower price per gallon, the manufacturers of these engines may find themselves in an embarrassing position unless they can utilize satisfactorily the low-grade lower-priced fuels.

In view of these trends it would seem that the time is ripe to urge further development work on engines which will use fuel oils below the level of furnace oils, and to urge caution in choosing and installing power-plant engines having fuel requirements approaching those of the automotive Diesel. The rise in price of Diesel fuel oils, predicted for many years, apparently is facing us now.

O. D. TREIBER.[8] The summary of progress for the last five years in oil and gas power has a significant phase which seems to justify emphasis. This is the advent of the high-speed Diesel engine into American manufacturing. It is significant because American production methods with attending reduction in unit

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[6] *Power*, New York, N. Y.

[7] Professor, Head of Mechanical Engineering Department, Iowa State College, Ames, Iowa. Mem. A.S.M.E.

[8] Hercules Motors Corporation, Canton, Ohio. Mem. A.S.M.E.

costs open vast fields of applications heretofore impossible or impractical because of cost, weight, and size. Large-quantity low-unit-cost production requires a substantial investment in tooling together with carefully organized operations for quick and accurate duplication of parts and their assembly. However it makes economically possible the use of these engines in automotive equipment, tractors, and a myriad of industrial applications where the owners can enjoy the inherent low operating cost of a Diesel engine.

Low first cost is made possible not only by production methods but also by ratings at higher speeds which allow of reductions in weight and size to values which are comparable with gasoline engines. In fact the company with which the writer is associated has a line of four models and eight sizes all of which are dimensioned for interchangeable mountings with gasoline engines of corresponding sizes and horsepower.

Some idea of the significance of this important contribution of a "new tool" to the power-using world may be had from some comparisons. About fifteen years ago Diesel engines were sold for marine and stationary purposes at prices about a hundred times those of high-production gasoline engines stripped of accessories. Ten years ago Diesel prices were down to from 50 to 60 times gasoline-engine prices. The differential would buy gasoline for the life of many installations. Now with high-production tooling, prices of high-speed Diesels have been reduced to from a fifth to a tenth of the last low figure and to from 5 to 10 times those of gasoline engines.

Consequently, high-speed Diesel engines are now available for a great many applications not economically possible in the past, such as in trucks, tractors, motorboats (both pleasure and commercial), rail cars, industrial locomotives, power shovels and all kinds of excavating machinery, farm machinery, pumping plants, and electrical generating units. Even multiunit powerhouses of large aggregate capacity are feasible and practical.

It might be argued that high-speed engines will not last as long as the slow-speed units. This is granted (the higher the speed the shorter the life) but with the present knowledge of metallurgy and the mechanics of internal-combustion engines, their life is being lengthened sufficiently, it is believed, to warrant the use of high-speed Diesel engines even where requirements for continuity of power are most exacting.

The advantages are not only a great reduction in original investment but low costs of repairs and replacements. Furthermore the time required for overhauling and rebuilding is obviously much less on either a basis of unit power or engine unit than on large slow-speed engines.

There is no intention herein to depreciate the importance of large Diesel engines. They will continue to fulfill a requirement distinctly outstanding. But it is desired to call attention to the importance of the high-speed Diesel engine as a new power tool.

E. S. Dennison.[9] The writer would like to amplify the portion of the paper in which the author referred to the advantages that have accrued from the use of auxiliary combustion chambers. With small engines using the four-stroke cycle, the use of such chambers has made it possible to raise the mean effective pressure for rating the engine from about 80 lb, as it was some years ago with an open combustion chamber, to between 105 and 110 lb, now an almost equally conservative figure. It has also enabled the cylinder pressure in the high-speed engine to be brought down in one case at hand to a little under 700 psi whereas with open combustion chambers that pressure is often 1000 psi. It has enabled the pressure in fuel-injection systems to be brought down from between 6000 and 8000 psi for direct injection, to pressures from 4000 down to 2000 psi with an auxiliary combustion chamber. Furthermore it has made the timing requirements much less rigorous than they were, so much so that it is often possible to dispense with any adjustment of timing to correlate with speed.

Now in contrast to all that the auxiliary combustion chamber has done for the small high-speed four-cycle engine, it is well to question what it has done for the small two-cycle high-speed engine. In so far as the writer knows, auxiliary chambers are not now and have not been successfully applied to the latter and an interesting question today is whether they can be. As is well known, there is the unfortunate tendency of exposed metal parts in a two-cycle combustion chamber to burn away, and whether that will prevent the use of auxiliary chambers in these engines is an open question. The writer is a little pessimistic on that point.

Now if the two-cycle engine must use an open combustion chamber with high cylinder pressure, high injection pressure, and exacting requirements as to timing, the four-cycle type is left with a decided edge from the maintenance standpoint. As long as materials are used that are responsive at all to heat or to pressure and that have any tendency to wear, such wear is going to be more effective in the two- than in the four-cycle engine. Consequently the writer believes that although the small two-cycle engine may have an advantage as to weight perhaps and is about equal in economy, nevertheless, from the operator's standpoint the four-cycle will continue to have an advantage. These comparisons do not apply at all in the case of large engines where both types use an open combustion chamber.

Author's Closure

The author wishes to thank each of the gentlemen who have by their discussions contributed further information of interest to make the paper more complete, more interesting, and better historical material for future use.

He was glad to have Mr. Dennison's defense of the four-cycle high-speed Diesel engine with an auxiliary combustion chamber in comparison with the two-cycle engine and direct injection, and also to have Mr. Rowley's data showing that the relative use of the four-cycle and the two-cycle engine in that particular class was a two-to-one ratio rather than an even break.

Concerning preliminary papers on Diesel fuel-oil specifications and some of the related factors that engineers and engine users had been talking about over a period of years in this so-called preparatory fuel work that was being done, the author still believes that there was nothing definite accomplished along the lines of fuel standardization before the year 1933. However, the Oil and Gas Power Division of the A.S.M.E. does deserve much credit for having sponsored papers on this subject and contributing valuable suggestions during the preliminary years of uncertain Diesel-fuel requirements.

The discussion of Mr. Kates concerning the Packard engine was noteworthy, but the author is not personally familiar with developments previous to 1933. It might be said that the Packard was the forerunner of the Guiberson engine because the latter is also a radial aircraft type, but with many new features. Captain L. M. Woolson[10] indeed deserves considerable credit for his Packard Diesel aircraft-engine development and the author is glad that his name was mentioned in this meeting.

It is interesting to note that each of the discussers of this paper said something about fuels and their combustion. The author would like to reiterate in closing that these are the two important factors that designers, manufacturers, and users will continue to write and talk about in years to come, the two most important considerations in the progress of oil and gas power.

[9] Engineer in charge of Diesel-engine development, Electric Boat Company, Groton, Conn. Mem. A.S.M.E.

[10] "The Packard Diesel Aircraft-engine," by L. M. Woolson, *S.A.E. Journal*, April, 1930, and September, 1930.

Attack on Steel in High-Capacity Boilers as a Result of Overheating Due to Steam Blanketing

By EVERETT P. PARTRIDGE[1] and R. E. HALL,[2] PITTSBURGH, PA.

In recent years a number of failures have occurred in certain portions of boilers operating in the pressure range near 1400 psi. Many of these failures have occurred under conditions such that oxygen corrosion or scale could not be considered as the cause. Characteristically the tubes have shown internal attack along the top, generally with definite grooving and the production of black magnetic iron oxide.

The fundamental cause of this type of failure is steam blanketing. In a tube which is only slightly inclined to the horizontal, the steam inevitably tends to rise to the top and move along it. This introduces a considerable resistance to the flow of heat from the tube wall to the fluid, and causes the top of the tube to become more or less seriously overheated. Boiler water of normal alkalinity passing through the tube and touching the overheated surface is concentrated to an excessive degree, with resultant attack on the steel. As has been shown by Berl, the rate of oxidation of steel by water at high boiler temperatures is greatly accelerated by increase in concentration of sodium hydroxide.

Condensed case histories of occurrences of this type of failure in a number of high-pressure boilers are presented. These histories include failures in ash screens in dry-bottom furnaces, floor tubes in slag-bottom furnaces, and slag-screen and top-row tubes in straight-tube cross-drum boilers.

From certain of the case histories it is evident that the dissolution of the steel along the top of steam-blanketed tubes may be stopped or greatly retarded by eliminating caustic alkalinity from the boiler water. This, however, does not affect the fundamental cause, and may merely postpone trouble, as indicated by other case histories in which steam blanketing has resulted not in dissolution of the steel, but in serious cracking.

It is concluded that the best chance of obviating the difficulties which may result from steam blanketing is so to design the surfaces exposed to high rates of heat input that steam blanketing cannot occur, or where slightly inclined tubes must be used, to protect them so that the rate of heat input will be sufficiently low to avoid excessive overheating of the metal.

IN THE LAST few years, a number of new boilers in the 1200 to 1400-psi range have lost tubes located in positions inclined at only a small angle to the horizontal. These failures have occurred in the majority of cases under conditions which precluded ordinary scale formation or oxygen corrosion, and have been characterized by a tendency for the tubes either to waste away internally along the top or to crack in this region, without excessive deformation.

This type of failure has been attributed to a variety of causes, but no systematic correlation of the information available from plant experience has yet appeared. It is the purpose of this paper to present such a correlation of a number of typical case histories in the hope that the problem will thereby be clearly defined, so that appropriate steps may be taken to meet it.

A paper such as the present one would have been impossible without the cooperation of the engineers and executives of the various companies, who in many cases have released data and information hitherto unavailable for publication, even when, as in a few cases, they disagreed with the interpretation made by the writers. The factual statements are, it is hoped, beyond argument, although the discussion based upon them is freely offered for criticism.

Certain concepts essential to a proper appreciation of the details in the case histories are introduced first, and after the experiences at the individual plants have been presented, a comprehensive and consistent interpretation of the observed facts is attempted.

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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.

Fundamental Concepts

When heat is transmitted to a vertical or highly inclined tube in a boiler at rates ranging up to 100,000 Btu per sq ft per hr characteristic of contemporary design, no difficulty is ordinarily encountered. The steam bubbles tend to separate immediately from the tube wall and the highly turbulent flow of steam and water up through the tube maintains a high surface conductance of heat from the metal to the boiling water. No actual data covering the high-pressure range are known to the writers, but at atmospheric pressure the surface conductance for boiling water may vary from 1000 to as high as 4800 Btu per sq ft per hr per deg F. It is probable that, at the higher pressures also, the resistance to the flow of heat from the steel tube to the water is of the same small order of magnitude as the resistance of the steel wall of the tube itself. The temperature of the inside of the wall accordingly does not run much above that of the water in the tube, perhaps exceeding it by 100 F under the most severe conditions.

Film Boiling

At extremely high rates of heat input in laboratory apparatus, it is possible to produce on the heat-transfer surface a continuous layer of vapor which acts as an insulator and accordingly increases the temperature drop from the metal to the water. The experiments of Sauer, et al. (1)[3] at atmospheric pressure show, however, that the maximum rate of heat transfer corresponding to the

[3] Numbers in parentheses refer to the Bibliography.

transition from ordinary to film boiling for water at an iron surface was 413,000 Btu per sq ft per hr. McAdams[4] has suggested that film boiling is not likely to occur in an actual boiler even in the portions of the heat-absorbing surface exposed to the most severe radiation.

Steam Blanketing

Even though film boiling may not constitute a problem in steam generation, the insulating effect of a blanket of steam must be considered in the case of tubes which are inclined at only a slight angle to the horizontal, such as the ash screens in dry-bottom furnaces and the floor tubes in slag-bottom furnaces of contemporary design. In such tubes, the heat transfer is downward to the top of the tube, while all of the steam generated in it inevitably tends to rise and move along its ceiling.

Even with moderately high heat input, the steel at any given point is probably wetted frequently by intermittent splashing of water against the top, and its temperature is thereby kept comparatively low. High rates of radiation to a tube may, however, produce an almost continuous steam blanket, with a resultant great increase in temperature. When water does strike against the overheated metal, a sudden drop in temperature would be anticipated.

Considering tubes in various positions, from the horizontal to the vertical, it is obvious that increase in the inclination will tend to make less probable the development of a nearly continuous steam blanket. From another viewpoint, the heat input necessary to produce an overheating of 500 F in the wall of a tube will vary progressively from some lower value for a horizontal tube up to the probably much higher value necessary to establish film boiling in a vertical tube.

Lack of data for high boiler temperatures and pressures precludes more than a rough estimate of the temperature conditions set up in a steam-blanketed tube. However, the highest value measured by Poensgen (2) for the surface conductance from steam at 100 psi and 412 F to the wall of a 3.75-in. pipe at 388 F was 70.5 Btu per sq ft per hr per deg F. If an admittedly long guess is made that the surface conductance might be 150 for heat transfer from an ash-screen tube to steam at 1400 psi and 590 F flowing along its top, then a heat input to the top of the tube of 75,000 Btu per sq ft per hr would raise the temperature of the inner wall 500 F above that of the steam in equilibrium with water in the tube.

The difference in temperature distribution resulting from a surface conductance of 150 to steam in a blanketed horizontal tube on one hand and of 1500 to boiling water in a vertical tube on the other hand is shown graphically in Fig. 1. The temperatures indicated, even though obtained by simplified calculation from assumed values, are probably of the correct order of magnitude.

Change in Microstructure Due to Overheating

If the top of a tube be actually overheated as suggested in Fig. 1 for a sufficiently long period of time, a definite change will be produced in the microstructure of the steel; the iron carbide originally present in the customary lamellar grains of pearlite will form globular particles of cementite.[5] Table 1, from the work of Bailey (3) shows how the time necessary to develop this spheroidized structure increases with decrease in the temperature to which the steel is subjected. The lowest temperature at which spheroidization may develop over a long time is unknown. On the basis of creep tests, White, Clark, and Wilson (4) state that spheroidization begins to be noticeable in plain carbon steel after about 4000 hr at 1000 F when the stress is sufficiently great. A spheroidized region in a boiler tube may therefore be regarded as evidence that the metal temperature has exceeded this level.

TABLE 1 TIME REQUIRED TO SPHEROIDIZE ANNEALED LOW-CARBON STEEL (3)

Temperature		Time, hr
C	F	
600	1112	200
625	1157	75
650	1202	20.5
675	1247	7.5
700	1292	3

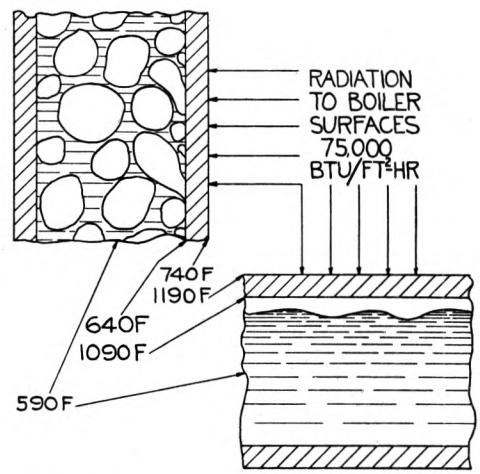


FIG. 1 COMPARISON OF TEMPERATURE CONDITIONS IN A VERTICAL AND IN A STEAM-BLANKETED HORIZONTAL TUBE
(Surface conductances of 1500 Btu per sq ft per hr per deg F from steel to boiling water and of 150 from steel to steam have been assumed.)

Excessive Local Concentration of Boiler Water

While steam blanketing tends directly toward overheating, indirectly it produces an equally serious effect. Normally, the concentration of dissolved substances in a boiler water will be low, but if this water is thrown intermittently against the excessively hot ceiling of a steam-blanketed tube, local concentration to any imaginable degree is possible.

Very soluble substances may be deposited as solids either on the top or at the water line on the sides of such tubes, decreasing correspondingly the concentration of these substances in the main part of the boiler water. The "hiding out" of phosphate and other soluble salts when boilers are operated at high ratings, followed by the reappearance of these substances in solution when the steam output is lowered has been observed in a number of boilers placed in service in recent years. The most potentially serious result of localized excessive evaporation is the high concentration of sodium hydroxide developed from a boiler water containing only 50 or 100 ppm of this substance.

Effect of Sodium Hydroxide on Oxidation of Steel by Water

What a concentrated solution of sodium hydroxide at 590 F will do to steel has been shown by Berl and van Taack (5) whose data are reproduced in Fig. 2. The vertical scale represents the extent to which carefully sized steel powder was oxidized when agitated for 7.5 hr in contact with solutions of sodium hydroxide of the concentrations shown along the horizontal scale. A minimum in the attack at 0.7 g per liter or 700 parts of NaOH per million parts of water is followed by a progressive increase at higher concentrations.

That water, even when free of dissolved oxygen is a powerful

[4] Private communication from W. H. McAdams.

[5] The change in structure is well illustrated in Fig. 5.

oxidizing agent with respect to steel is not always recognized, because as soon as the reaction

$$3Fe + 4H_2O \rightarrow Fe_3O_4 + 4H_2$$

commences, it is slowed down to an almost infinitesimal rate by the protective coating of Fe_3O_4 which is one of the products. Because the role of dissolved oxygen in helping to destroy the protective effect of this oxide coating has been recognized, mechanical deaeration has been so perfected in recent years that water containing only 0.005 ml per liter may be fed to boilers, and the use of chemical scavengers such as sodium sulphite and ferrous hydroxide has been developed to remove even this last trace. Every pound of water in a boiler, however, contains 0.89 lb of combined oxygen which is available to oxidize the steel surfaces whenever and wherever the water can penetrate the thin barrier of oxide.

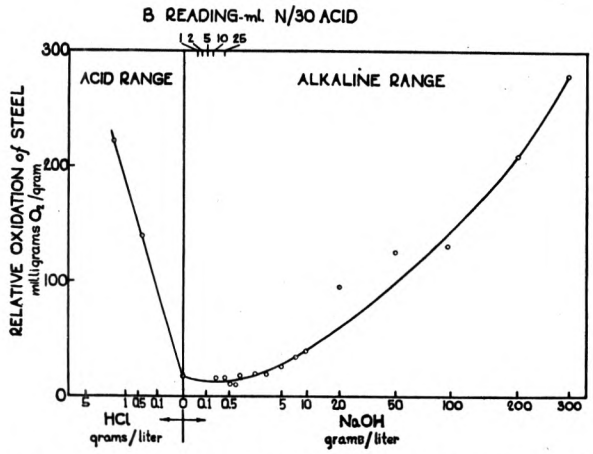


FIG. 2 EFFECT OF HYDROCHLORIC ACID AND OF SODIUM HYDROXIDE ON THE OXIDATION OF STEEL BY REACTION WITH WATER AT 590 F (FROM BERL AND VAN TAACK.)

(In each test five grams of sized steel powder was agitated in contact with solution in a bomb for 7.5 hr, and the extent of the reaction was measured by the degree of oxidation of the steel. Concentrations are plotted on a cube-root scale in order to show adequately the data over the wide range covered.)

The retarding effect of low concentrations and the accelerating effect of high concentrations of sodium hydroxide on the reaction of water with steel at high temperatures is believed to be due largely to the effect of this substance on the protective oxide, which may be regarded as a membrane full of minute pores through which water is always diffusing to the metal beneath to continue the reaction. A dilute solution of sodium hydroxide apparently helps to develop an oxide film which is less permeable, while a more concentrated solution tends to open it up, and a very concentrated one actually destroys it by forming soluble iron complexes.

Hydrogen Brittleness

Hydrogen, the second product of the reaction of water with steel, when produced in the atomic state at the surface of the metal, diffuses readily into it, causing a decided loss in ductility without decreasing the tensile strength. When high concentrations of sodium hydroxide produce accelerated attack of water on steel, it is therefore reasonable to expect that the steel may become more brittle, in the same manner that it does when pickled in acid (6).

Cracking as a Result of Thermally Induced Stresses

If the ceiling of a tube is repeatedly first overheated, because of the development of a steam blanket, and then suddenly quenched by water striking the surface, the reversed stresses set up in the steel may well lead to cracking of the stress-corrosion type.

Imperfect Circulation of Boiler Water

Steam blanketing may develop not only in nearly horizontal tubes receiving heat by direct radiation from the furnace, but also in tubes which, by reason of the sometimes unpredictable circulation in boilers, carry an inadequate flow of water. In straight-tube cross-drum boilers, for example, it is not uncommon to find the top-row tubes steam-blanketed near the uptake header, with definite evidence of a water line or even rope scale on the sides of the tubes.

CASE HISTORIES

From the preceding, it is evident that steam blanketing of tubes may lead not only to overheating, but also to accelerated attack on the steel due to the excessive concentration of normally alkaline boiler water, or to cracking by stress corrosion. These difficulties are not new. However, few case histories have been reported as completely as those of a 450-psi cross-drum boiler by Hanson (7), of 45-atm cross-drum boilers at Merseburg by Ziegler (8), and of the first 100-atm bent-tube boilers at Mannheim by Marguerre (9). A number of specific cases in which the authors believe that steam blanketing has constituted a problem will therefore be presented.

SLAG-SCREEN TUBES IN A STRAIGHT-TUBE CROSS-DRUM BOILER

South Amboy Station

The South Amboy Station of Jersey Central Power and Light Company comprises three 1400-psi straight-tube cross-drum boilers which went into service successively between Oct. 1, 1930 and Feb. 14, 1931. In these boilers, which are 40 tubes wide, the even-numbered tubes of the bottom row were originally bent downward, as indicated in Fig. 3, to form a slag screen, and the slope of the short, nearly horizontal section was 1.5 deg. Failure of these screen tubes along the top of the nearly horizontal section near the bend upward has occurred at intervals throughout the history of the station, while failure of the alternating straight tubes in the bottom row has been infrequent.

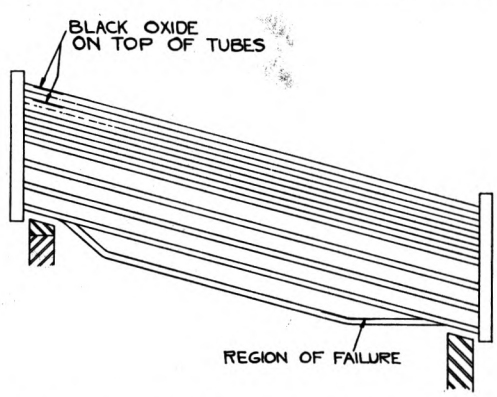


FIG. 3 ELEVATION OF GENERATING TUBES AT SOUTH AMBOY STATION

History of Failures. Interpretation of the early records is somewhat uncertain, because both oxygen corrosion and scale formation may have contributed to tube failures. When the plant started up and for some time thereafter, the pH of the boiler water was low and the oxygen content of the feed rose intermittently to 0.1 ml per liter, while considerable amounts of raw well water were introduced as make-up. After slightly less than two years of service, it was necessary to replace the en-

tire bottom row, both straight and drop-leg screen tubes, in boiler 1 toward the end of July, 1932, following individual failure of six tubes during the spring of that year.

Sporadic failure of screen tubes during the period from 1933 to 1936, always along the top of the nearly horizontal section, led to the decision to increase the slope of this section from 1.5 to 3 deg. On Apr. 25, 1936, all of the drop-leg tubes in boiler 1 were replaced with tubes rebent to give this increased slope. The 3-deg tubes were then installed in boiler 3, five on June 18, one on July 6, and the remaining fourteen on Aug. 23, 1936; and in boiler 2, two on June 9 and one on Oct. 26, 1936, and the entire twenty tubes on March 20, 1937.

In spite of the increased slope, replacement of tubes continued in the same discouraging manner, as indicated in Table 2. On

TABLE 2 TUBE REPLACEMENTS SUBSEQUENT TO INSTALLATION OF COMPLETE NEW SLAG SCREENS WITH 3-DEG SLOPE IN NEARLY HORIZONTAL SECTION

Date	Boiler	Tube	Time in service, months
8/13/37	3	38	11.7
1/26/38	3	16	17.0
		24	17.0
3/28/38	1	6	23.0
		8	23.0
4/26/38	1	4	24.0
7/1/38	3	16	5.0
		18	22.2
7/15/38	1	6	3.7
		8	3.7
8/25/38	1	4	4.0

July 1, 1938, tubes with a slope of 6 deg along the nearly horizontal section were installed as Nos. 16 and 18 in boiler 3. Shortly thereafter, however, it was decided to eliminate the drop-leg tubes altogether, and to replace them by straight tubes. This change was made in boiler 3 on Aug. 5, 1938.

Evidence of Overheating. One factor influencing the decision to abandon the slag screen was a study of tubes 6 and 8 removed from boiler 1 on March 28, 1938. Tube 6, which had developed a leak, and tube 8 which had not cracked but which was in a state of incipient failure, both showed typical attack along the ceiling of the horizontal section near the bend upward. The tube wall

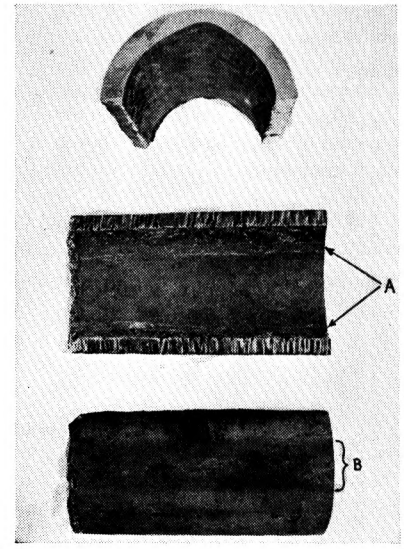


FIG. 4 APPEARANCE OF TOP OF NEARLY HORIZONTAL SECTION OF SCREEN TUBE FROM A 1400-LB STRAIGHT-TUBE CROSS-DRUM BOILER

(*A*, Water lines and corrosion grooves along ceiling of tube; *B*, red band on external surface of top half of tube.)

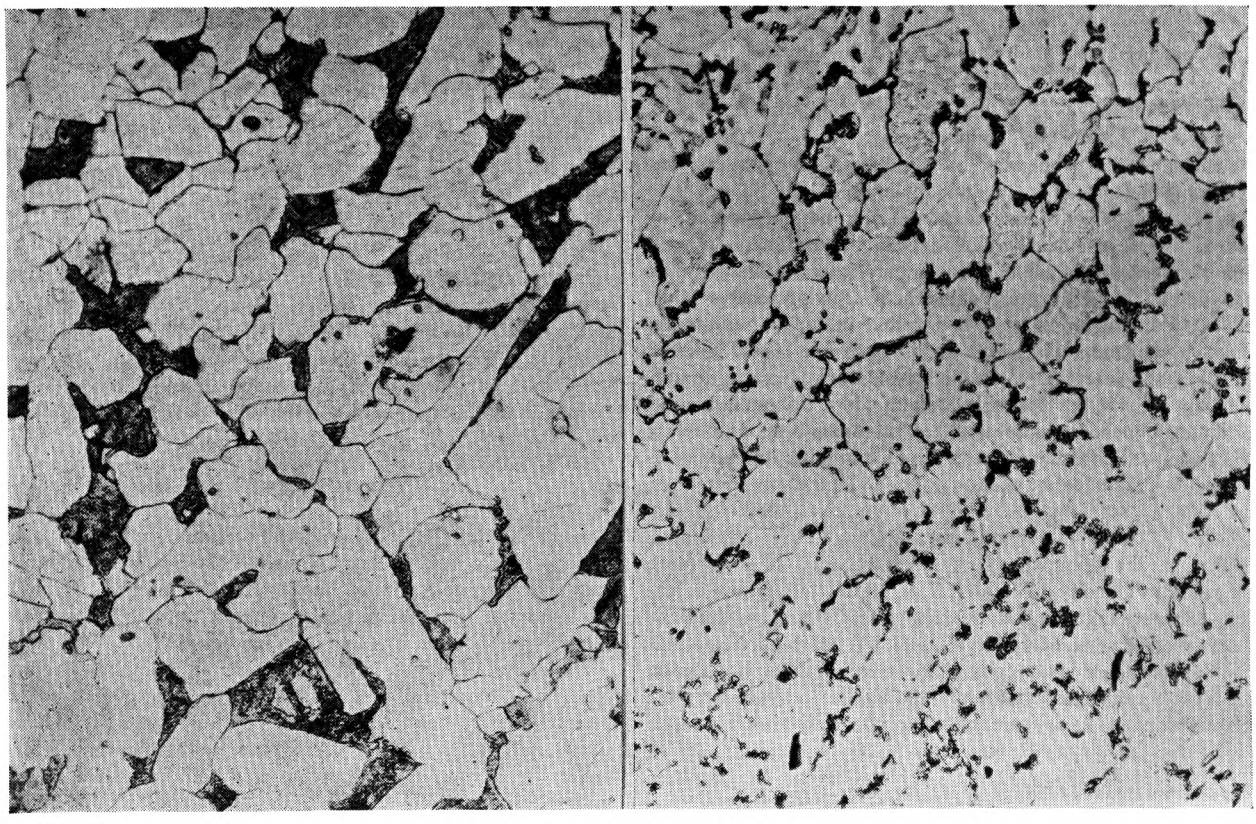


FIG. 5 MICROSTRUCTURE (×500) OF SPECIMENS OF TUBE

(Left: From side of tube, showing normal appearance. Right: From top of tube, showing severe overheating.)

had been thinned down as indicated in the cross section of Fig. 4, and at each side of the band of attack, which was covered by a thin relatively smooth coating of black magnetic iron oxide, there were narrow sharp grooves. The rest of the internal surface showed the typical appearance of old scale in the process of disintegration.

That the tops of the horizontal sections had been overheated was suggested by an external band of red oxide about 3 in. wide, which tapered off along the upturned portion of the tube. Conclusive evidence to this effect was obtained by metallographic examination of specimens cut respectively from the side and from the top of tube 8 at a point in the horizontal section 6 in. from the upward bend of the tube. While a microstructure typical of a seamless tube was observed in the side wall, extreme breakdown of pearlite and the formation of rounded particles of cementite characterized the top of the tube, as shown in Fig. 5.

Boiler-Water Alkalinity. An attempted correlation between the monthly average values for alkalinity and pH of the boiler water on one hand and replacement of screen tubes on the other

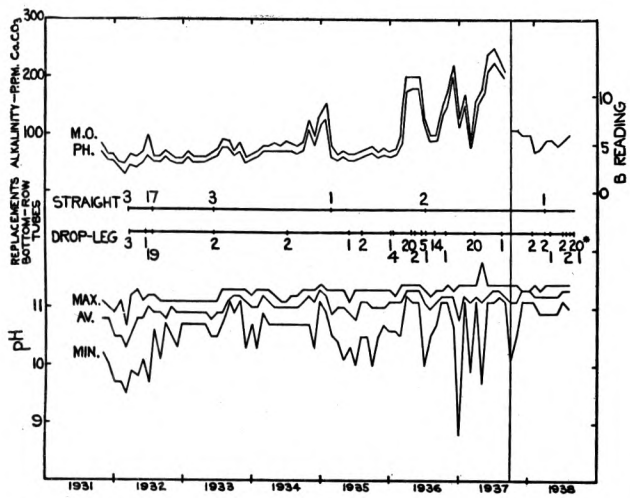


FIG. 6 RECORD OF SCREEN-TUBE REPLACEMENTS AND OF MONTHLY AVERAGE VALUES FOR ALKALINITY AND pH OF BOILER WATER

(Alkalinities are expressed as determined by titration to end-points of phenolphthalein (PH.) and methyl orange (M.O.), and by titration to end-point of phenolphthalein in the presence of an excess of barium chloride (B reading). While the results of the first two methods of titration of a boiler water do not allow calculation of the concentration of sodium hydroxide, multiplication of the B reading by 13.3 gives the approximate value for NaOH in ppm.)

in Fig. 6 suggests that the considerable increase in alkalinity of the boiler water early in 1936 may have accentuated the attack along the ceiling of the screen tubes and necessitated the immediate replacement of tubes which had been losing metal slowly during the preceding years. The maintenance of a moderate and carefully controlled alkalinity subsequent to Oct., 1937, did not, however, obviate the continued failure of the new screen tubes installed in all three boilers during 1936 and 1937.

That the replacement of the drop-leg tubes by straight tubes is likely to solve the problem is indicated in Fig. 6 by the fact that, while more than eighty of the former had been replaced since 1932 and prior to August, 1938, only seven of the straight tubes alternating with them in the bottom row had failed during this period, and most of these failures were due to external cutting by water and steam from leaking drop-leg tubes.

Attack in Upper-Row Tubes. While up to the present it has constituted a minor problem in comparison with the failure of screen tubes, some attack has also occurred, as indicated in Fig. 3, along the top internal surfaces toward the uptake header in the two upper rows of tubes in the boilers at South Amboy. During an inspection of boiler 3 in August, 1938, it was estimated that not more than 0.02 in. of metal had been removed. The attack apparently was more pronounced in the row next to the top than in the top row, but showed in all tubes in both rows. Only one tube in these rows has been replaced during the history of the plant, tube 16 of the top row of boiler 3 on Sept. 11, 1935, but it is apparent that the same condition exists here, though to a less serious degree, as in other straight-tube cross-drum boilers discussed in this paper.

TOP-ROW TUBES IN A STRAIGHT-TUBE CROSS-DRUM BOILER

San Antonio Station B

On Aug. 28, 1931, two top-row tubes failed in the 1250-psi boiler at Station B of San Antonio Public Service Company (10). This straight-tube cross-drum boiler had been operated intermittently for only about seven months, with B reading[6] held generally between 1 and 4, and phosphate between 20 and 60 ppm of PO_4, although occasional higher values were observed.

Failure occurred along the top of the tubes, near the uptake header. In addition to the two actual failures in the top row, many tubes in the upper four rows had been damaged, as attested by the considerable quantities of magnetic iron oxide in the top half of each tube. The attack was localized toward the right side of the tube bank; the iron oxide was thickest in the top-row tubes of headers 24, 25, and 26, and thinned out in 27 and from 23 back to 13, with practically none in 12. The tubes in the second row were in about the same condition as those in the first. In general, the oxide seemed to be thickest about 3 or 4 ft back from the uptake header, thinning out progressively toward the header.

In addition to the magnetic iron oxide still in place along the tops of the tubes, large quantities of this substance in the form of powder were found lying in the central portions of the waterwall

[6] B reading is the number of ml of $N/30$ acid required to titrate to the phenolphthalein end-point 100 ml of a boiler-water sample to which an excess of barium chloride has been added to precipitate buffer salts such as carbonate and phosphate.

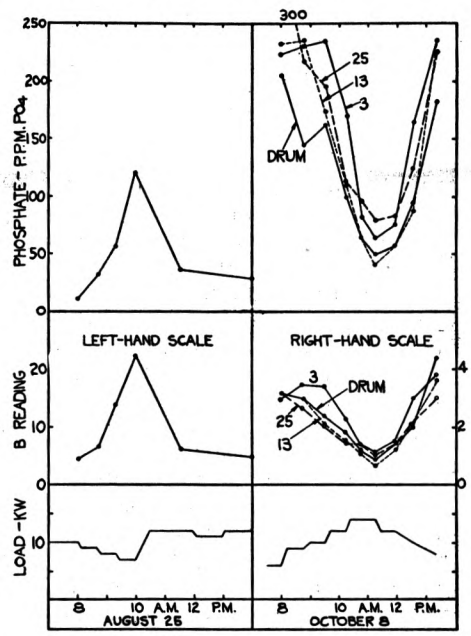


FIG. 7 INVERSE VARIATION OF BOILER-WATER CONCENTRATION WITH LOAD AT SAN ANTONIO STATION B

(Multiplication of the B reading by 13.3 gives the approximate value for NaOH in ppm.)

headers, apparently carried there and dropped by the circulation of the water.

"Hiding Out" of Soluble Compounds. For some time previous to the failure, the tendency of phosphate and alkalinity to disappear from the boiler water had been noted. Tests on Aug. 25, three days before the tubes let go, indicated that, when no addition of conditioning chemicals was made, the concentrations in the boiler water tended to vary inversely with the load. Although it did not seem credible at the time, the data, reproduced in Fig. 7, now may be interpreted as definite evidence of excessive local evaporation to substantial dryness in some portion of the boiler during operation at the higher loads, followed by return of the deposited substances to solution during subsequent operation at lower load. This hiding out of dissolved substances was observed repeatedly during later tests when samples were taken not only at the original sampling point in downtake header 3, but also in downtake headers 13 and 25, and from a perforated sampling line running the length of the drum. The data for Oct. 8 in Fig. 7 show how the concentrations changed in the same general manner with load at all four sampling points. As the load was raised progressively, both the alkalinity and the phosphate decreased; when the load was dropped they again increased.

Correlation Between Tube-Wall Temperature and Oxidation. Loss of dissolved substances from the boiler water during the test on Oct. 8 did not produce any consistent corresponding increase in the temperature of the top-row tubes as measured by thermocouples. However, during the period from Oct. 1, when the boiler was first returned to operation following the tube failures, until Oct. 16, there was a gradual increase in the tube-wall temperatures, with the highest values developing in the region where the heaviest formation of oxide had previously been observed. Inspection at this time showed that the oxide was developing as before, and that the largest amounts were in the tubes of header 23. Correlation of the temperature measurements with the observed condition of the tubes showed that the deposit of black oxide had accumulated only in those tubes which had exceeded 800 F during the operating period. Measurement of the internal diameter revealed that there had been little actual loss of metal from the tubes.

Changes in Baffling and Circulation. Experimentation with deflection baffles in the downtake headers to direct more of the flow into the upper rows, and with direct introduction of feed into the tops of the downtake nipples showed less promise during brief test runs than did a rearrangement of the baffles. The original monolithic baffle, resting on the top-row tubes and extending from the downtake headers nearly halfway along the tubes, was removed, and an alloy-steel baffle of similar extent was placed just under the superheater. At the same time a short baffle was installed, extending back from the front headers between the third and fourth rows from the top. The effect of these changes was to divert the upward flow of gases through the tube bank toward the rear, thus decreasing materially the heat input to the upper ends of the top three rows of tubes and increasing the heat input to the lower ends. During a run from Jan. 5 to 14, 1932, after this change had been made, the highest temperature observed in a tube wall at a load of 18,000 kw was 821 F, and the temperatures averaged approximately 200 F lower than during the previous run.

Extension of the short baffle 2 ft farther from the uptake header to still further deflect gas flow away from the upper ends of the tubes in the first three rows; installation of small restriction baffles in all downtake headers between rows 5 and 6 from the top; and replacement of the feed trough by a 4-in. pipe along the drum with 1-in. nipples extending downward into the tops of the downtake tubes solved the overheating problem early in March. At a load of 19,200 kw the maximum tube temperature was 613 F.

Recurrence of Trouble. An interesting postscript to this case was written into the record in 1938. For more than 6 years, from March, 1932, until July 22, 1938, the boiler had operated with no signs of any further difficulty with overheating of tubes. On the latter date the top-row tube of header 27 ruptured 6 in. from the uptake end. Inspection revealed black oxide deposits in this and adjacent tubes. At the end of the next operating period, from Aug. 5 to 20, the replacement tube had developed a coating of black oxide on the top quarter of its circumference extending over about 10 in. from 3 to 13 in. in from the end of the tube and increasing in thickness from the edges to a maximum of 0.25 in. at the center.

The reason for the reappearance of this old trouble was found in the removal of the end superheater tube immediately above the affected region. This tube had developed a leak and had been cut out and the header openings plugged during September, 1937, but all except the bottom two loops were left in place. Because the dead tube was found to be burning away, it was removed during the annual inspection in March, 1938, and the open space partially closed off by a vertical alloy-steel baffle running upward from the front edge of the inclined baffle under the superheater. Later, on May 25, the empty space toward the front of the boiler from this vertical baffle was partially filled with tile. Apparently, however, heat transmission to the top-row tubes was appreciably greater than when the dead superheater tube had been in place.

During the latter part of August the vertical metal baffle was removed, and the space formerly occupied by the bottom loop of the superheater was completely filled in with tile. When operation from Aug. 27 to Sept. 19 revealed continued though greatly decreased attack, a small protecting baffle was installed covering the upper ends of the top-row tubes in headers 25 to 27 for a distance of 27 in. from the headers. This effectively prevented serious overheating; the highest temperature observed was 720 F. While other changes may be made to reduce tube-wall temperature still further, the problem no longer seems serious.

Inclined Tubes in Generating Sections

Springdale Station

Failure in tubes in the generating section has occurred in the three 1300-psi boilers at Springdale Station of West Penn Power Company. These boilers, numbered 2, 4, and 6 respectively, are of the cross-drum sectional-header type with two-stage furnaces, and are rated to produce 500,000 lb of steam per hr.

The generating tubes are illustrated in elevation in Fig. 8. The first failure occurred in boiler 4 on Sept. 30, 1937, after about three weeks of service, when the top generating tube in header 17 ruptured along the top about 2 ft from the uptake end. During the subsequent 15 months, similar attack necessitated the replacement of 5 more top-row generating tubes and 2 division-wall tubes in boiler 4, 2 top-row and 1 division-wall tubes in

TABLE 3 COMPOSITION OF DEPOSIT FROM REGION OF FAILURE ALONG BOTTOM OF CORED TOP-ROW TUBE AT SPRINGDALE STATION

	Per cent
Sulphur trioxide (SO_3)	Trace
Carbon dioxide (CO_2)	0.0
Phosphorus pentoxide (P_2O_5)	1.7
Silica (SiO_2)	2.2
Iron oxide (Fe_3O_4)	82.9
Aluminum oxide (Al_2O_3)	Trace
Calcium oxide (CaO)	1.4
Magnesium oxide (MgO)	0.4
Copper (Cu)	8.9
Net ignition loss[a]	2.2

[a] Calculated from actual ignition gain of 3.5 per cent resulting from oxidation of Fe_3O_4 to Fe_2O_3 and of Cu to CuO.

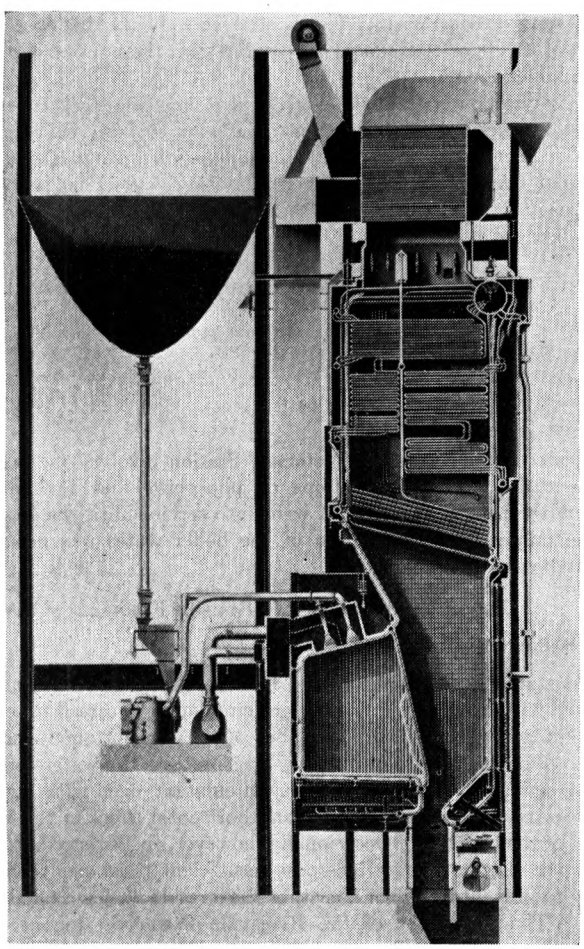


FIG. 8 CROSS SECTION OF 1300-LB BOILER AT SPRINGDALE STATION

boiler 6, and 1 screen tube in boiler 2. The type of failure is indicated in Fig. 9, top.

In the meantime, cores extending from the uptake headers to within about 18 in. of the downtake headers were installed in the top row of generating tubes in boiler 6 in an effort to obviate steam blanketing. This, after about 15 months' service, led to failure of these tubes along the bottom about 3 ft from the downtake end where a deposit of the composition shown in Table 3 accumulated, plugging the lower half of the annular space between the core and the tube wall. This deposit, almost entirely magnetic iron oxide and metallic copper, was partially loose material which had settled out of the water. The cross section of Fig. 9, bottom, shows the harder mass developed in place by oxidation of the tube wall. This was so hard and so firmly bonded to the metal that it was not loosened by turbining prior to removal of the tube.

Altogether, 12 of the 28 tubes of the top row have been replaced in boiler 6 as a result of this type of attack along the bottom toward the downtake end. The full-length cores have been removed, and other means of obviating steam blanketing are being tested.

The water conditions in the boilers at Springdale are indicated in Table 4.

TABLE 4 RANGE OF BOILER-WATER CONDITIONS IN 1300-PSI BOILERS AT SPRINGDALE STATION

	B Reading, ml			Phosphate, ppm PO_4		
Boiler	Maximum	minimum	Average	Maximum	minimum	Average
2	6.3	3.9	4.6	71	13	50
4	12.0	0.3	5.0	141	6	49
6	15.3	0.9	5.5	205	13	76

NOTE: Multiplication of the B reading by 13.3 gives the approximate concentration of NaOH in ppm.

Waterside Station

In the 1400-psi boilers which went into service at Waterside No. 2 Station in 1938, the roof of the furnace proper is formed by a straight-tube section, as indicated in Fig. 10. This section, 41 tubes wide, comprises 20 banks 3 tubes high, alternating with 21 banks 4 tubes high.

Boilers 4A and 4B had been in service several months when boiler 5A was started up about the middle of July, 1938. On Aug. 12, the top tube in the sixth header from the east end of 5A ruptured, and adjacent tubes were found slightly swollen. A month later, on Sept. 14, a second failure occurred, this time in the top tube in the twelfth header from the east end. Both of these headers are 4 tubes high.

Samples of the first tube failure showed little or no loss in wall thickness and little accumulation of oxide on the internal surface, but the steel was badly cracked both longitudinally and transversely over a considerable area, as illustrated in Fig. 11. In the case of the second failure, the wall thickness had been considerably reduced and a heavy coating of magnetic iron oxide was present, while the failure was rather localized, as shown in Fig.

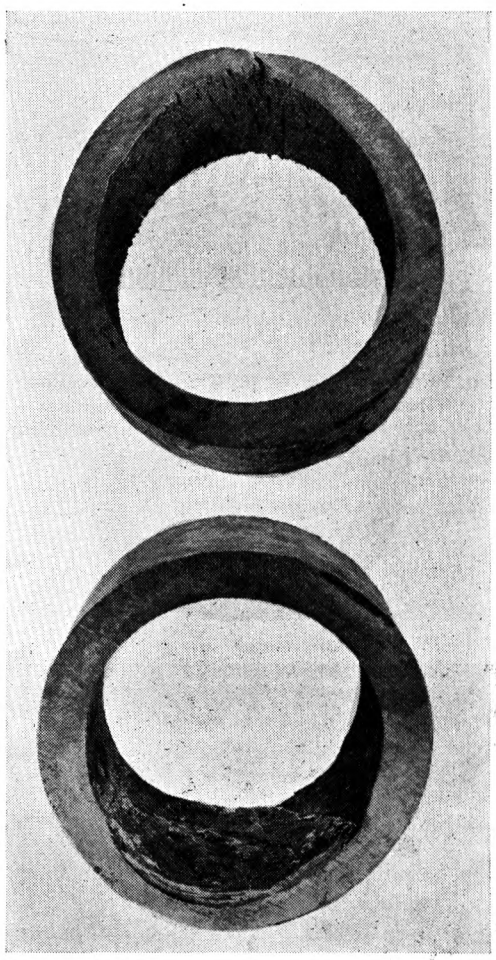


FIG. 9 CROSS SECTIONS OF FAILED TUBES FROM 1300-LB BOILERS AT SPRINGDALE STATION

(Top: Tube cracked along top as a result of steam blanketing. Bottom: Tube oxidized along bottom as a result of accumulation of sediment between core and tube wall.)

12. In both cases the damage was limited to the bottom half of the tube near the uptake header. Other tubes removed from the boiler, including one thought to be relatively undamaged, showed fine cracks in an early stage of development.

Microscopic examination of specimens from the damaged areas by F. E. Foster of the Research Bureau of the Consolidated Edison Company revealed that the cracks were intergranular, as is apparent in Fig. 13. This is believed to be the first observation of this type of cracking developing from the internal surface of a boiler tube.

The photomicrographs of Fig. 13 also show the steel to be almost completely decarburized near the inner surface, while spheroidized cementite, indicative of excessive overheating, is evident beyond this decarburized zone. In other specimens, various stages in the disappearance of the original lamellar pearlite could be observed.

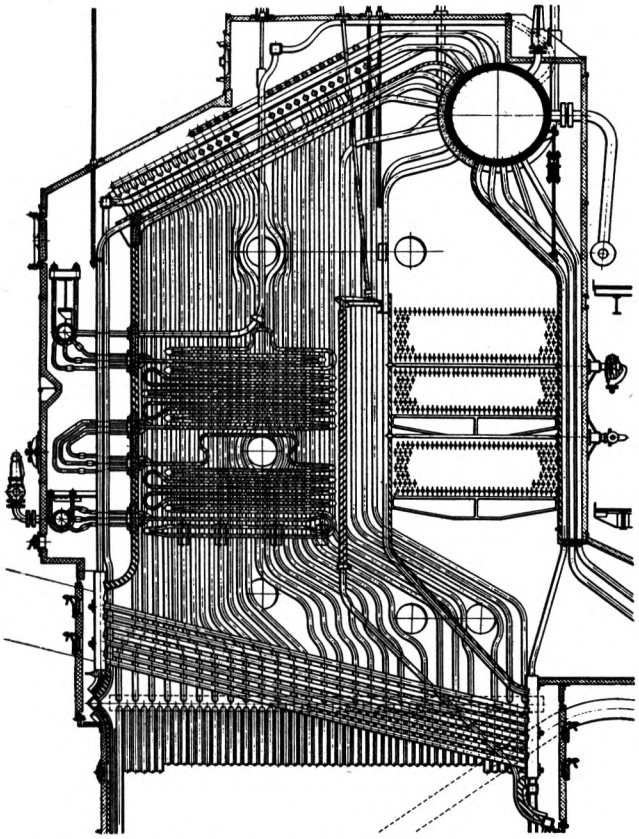


FIG. 10 CROSS SECTION OF HIGH-PRESSURE BOILERS
(Units Nos. 4 and 5, Waterside No. 2 Station.)

TABLE 5 RANGE OF BOILER-WATER CONDITIONS IN BOILER 5A AT WATERSIDE STATION JULY 19 TO SEPT. 14, 1938

	Concentration, ppm		
	OH	PO_4	pH
Maximum	88	160	11.4
Minimum	2	15	10.8
Average	29	51	11.2

Boiler-water control at Waterside Station involves the maintenance of low concentrations of phosphate and hydroxide. From the time that boiler 5A went into service until the second tube failure, the composition of the boiler water was normal, as indicated in Table 5.

FLOOR TUBES IN A SLAG-BOTTOM FURNACE

Rivesville Station

As stated by Caldwell (11), "The corrosion sometimes associated with overheating or deficient circulation in screen tubes is absent generally in the floor tubes of the slag-bottom furnace. The insulating nature of ordinary slag protects the water-cooled floor against overheating or faulty circulation, even with highly concentrated heat-release rates and horizontal tubes in the bottom of the furnace." Occasionally, however, the furnace bottom and the slag on it may interpose insufficient resistance to heat flow to prevent failure of floor tubes as a result of steam blanketing. The experience of the Rivesville Power Station of the Monongahela West Penn Public Service Company is a case in point.

The floor of the primary furnace in this 1300-psi straight-tube boiler contains 28 tubes about 18 ft long, nearly horizontal, with a slope of only about 1.5 deg. These tubes, which were studded over the top half, were originally covered with about 1 in. of

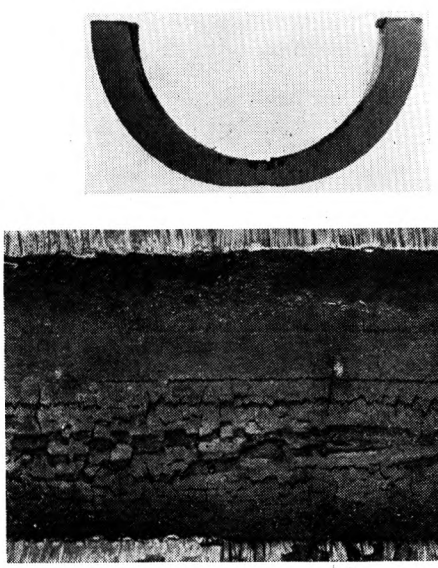


FIG. 11 CROSS SECTION AND INTERNAL APPEARANCE OF FIRST FAILURE OF A TOP-ROW TUBE AT WATERSIDE STATION

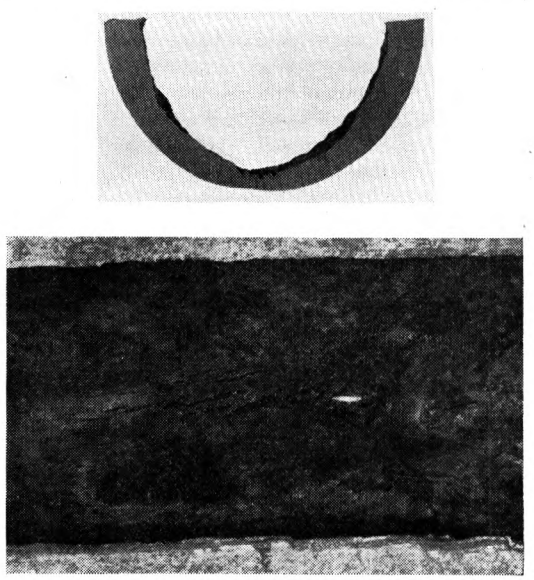


FIG. 12 CROSS SECTION AND INTERNAL APPEARANCE OF SECOND FAILURE OF A TOP-ROW TUBE AT WATERSIDE STATION

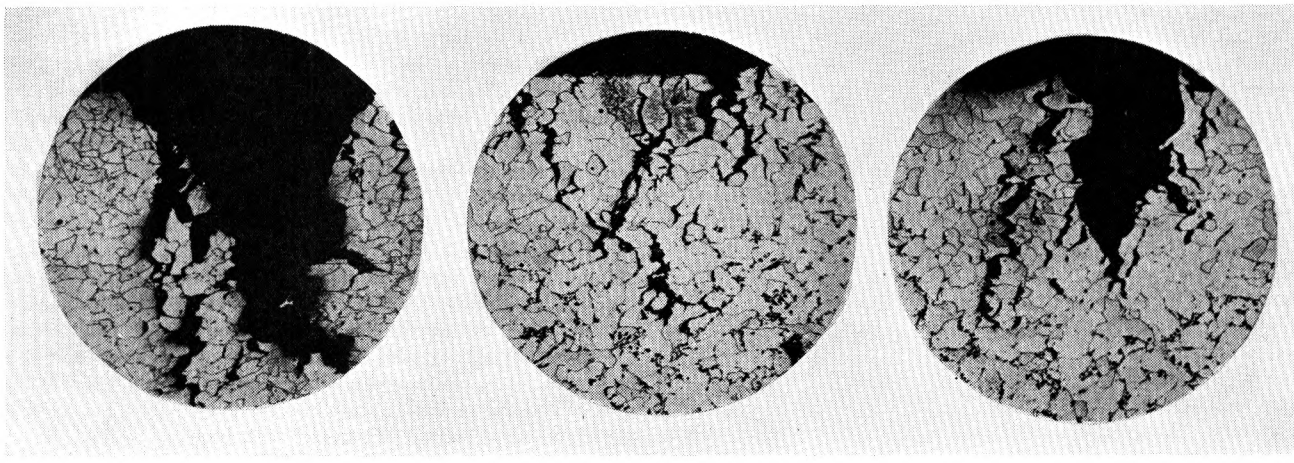


FIG. 13 INTERGRANULAR CRACKS DEVELOPED FROM INTERNAL SURFACE OF CRACKED TOP-ROW TUBES AT WATERSIDE STATION

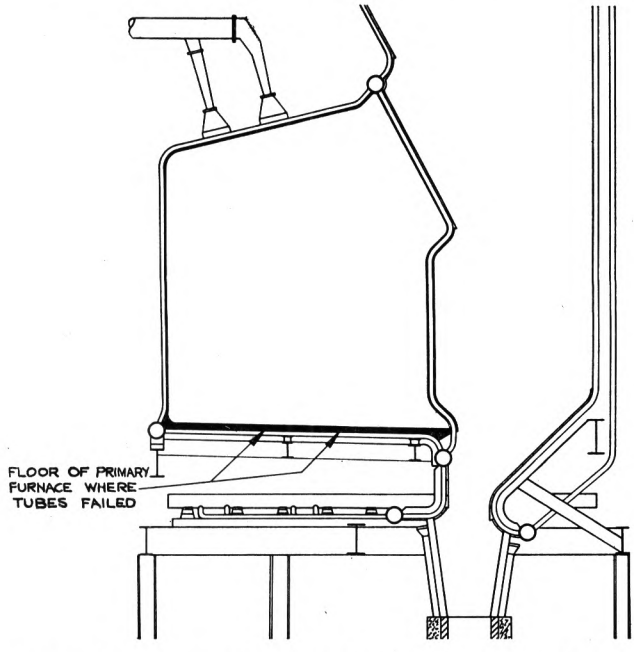


FIG. 14 CROSS SECTION OF 1300-LB BOILER AT RIVESVILLE STATION SHOWING FLOOR TUBES

chrome ore. Their location is indicated in the simplified cross section of Fig. 14.

The boiler went into service on Oct. 29, 1937. On Dec. 27, 1937, floor tubes 10 and 16 failed along the top, followed by tubes 14 and 18 on Dec. 30. Inspection revealed that the chrome-ore refractory had been worn away in the center of the floor, directly under the burners, exposing the ends of the studs, which were badly burned, some having lost as much as half the original metal. The center 20 tubes were then replaced, leaving only 4 of the original tubes at each side of the floor, the thickness of the layer of chrome ore was increased to 4 in., and provision was made to retain about 1 in. of molten slag on top of the chrome ore.

All of the 20 tubes which were removed had been attacked along the top, most of them for about two thirds of their length. Each tube showed sharp lines of demarcation along the sides, below which the steel was undamaged, but above which there were many deep pits. The actual failures were small holes in the tops of the tubes where the wall had been thinned until it yielded under the internal pressure.

Lying in the tubes and in the header into which they discharged was a large amount of loose black magnetic iron oxide, which was easily washed out.

The alkalinity and phosphate in the boiler water during the 2 months from the time the boiler was placed in service until the failures occurred were normal, as indicated in Table 6. During the year 1938 following the changes in the furnace floor, the alkalinity averaged somewhat higher than during the time prior to the failures, and the phosphate remained at about the same level, but no signs of further attack on the floor tubes were observed. Apparently, the considerable increase in the thermal insulation above the tubes so reduced the heat input to them as to minimize overheating and excessive localized concentration of the boiler water, even though the tubes, as a result of their nearly horizontal position, may still tend to be steam-blanketed.

TABLE 6 SUMMARY OF BOILER-WATER CONDITIONS IN 1300-PSI BOILER AT RIVESVILLE STATION PRIOR TO FAILURE OF FLOOR TUBES

	B Reading ml	Phosphate, ppm PO_4
Maximum	8.5	100+
Minimum	1.4	20
Average	4.9	70

One adverse result of decreasing the heat input to the floor tubes has been an increased temperature of the gases entering the tube banks, with a resultant tendency to choke up the boiler with slag, which at times makes it impossible to maintain superheat.

ASH SCREEN IN A DRY-BOTTOM FURNACE

Port Washington Station

The experience of Port Washington Station of Wisconsin Electric Power Company[7] with failure of screen tubes along the top has already been described in some detail (12) but will be reviewed here, together with supplementary information, because of the decided similarity between this case and the others described in this paper.

This boiler went into service in December, 1935. In October, 1936, the first tube failure in the ash screen was replaced. How serious a condition existed was realized in December. Four outages were necessary on Dec. 4, 17, and 26, 1936, and Jan. 3, 1937, to replace screen tubes which were found to have been grooved internally along the top. Newly installed tubes were thinned down to the point of leaking during 5 days of operation from Dec. 21 to 25. When a failure suddenly occurred on Jan. 12, it was decided to replace all questionable screen tubes.

[7] Formerly The Milwaukee Electric Railway and Light Company.

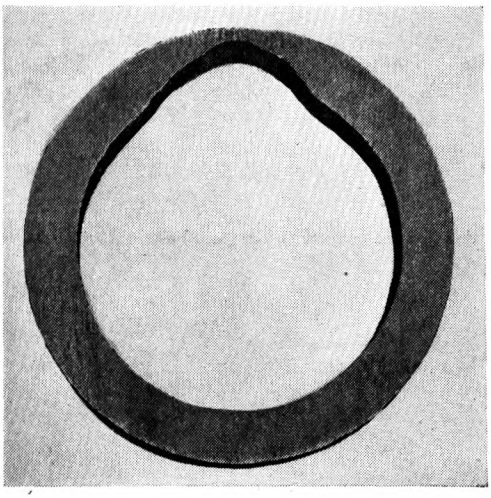


FIG. 15 CROSS SECTION OF SCREEN TUBE FROM PORT WASHINGTON SHOWING GROOVING ALONG TOP

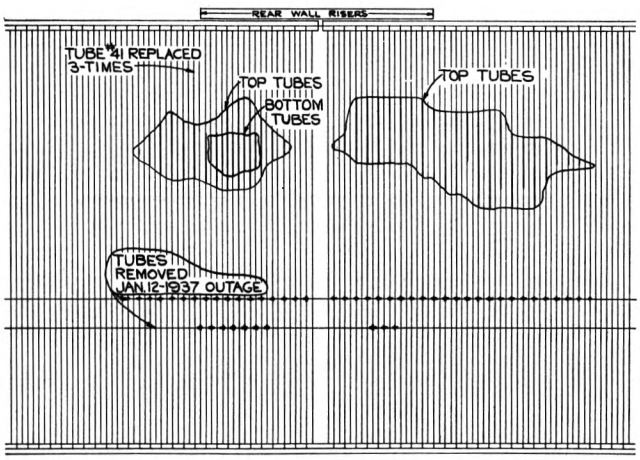


FIG. 16 AREAS OF ATTACK IN ASH SCREEN AT PORT WASHINGTON STATION

The typical appearance of the grooving of the screen tubes is indicated by the cross section in Fig. 15. Examination of the 41 top-row and 10 bottom-row tubes removed immediately after the outage on Jan. 12 demonstrated that the attack had been limited to two areas of the water screen about one third of the distance from the rear to the front headers, as shown in the plan sketch in Fig. 16. In the smaller left-hand area, 5 bottom-row as well as 14 top-row tubes had been damaged; in the larger right-hand area, the trouble was apparently confined to the top row, as the 3 bottom-row tubes removed showed no thinning along the top.

The replacement tubes were bent to give an increase of 0.5 deg on the average slope of 5 deg for the original design. Due to the natural sag of the tubes, the actual slope in the zone where trouble occurred was probably about half the average value.

Boiler-Water Conditions. In the search for some means of obviating further trouble, attention was concentrated on adjustment of the composition of the boiler water to maintain a protective coating on the tube surfaces, using the concentration of hydrogen in the steam as a measure of the rate at which the steel was being attacked. After the controlled formation of calcium-sulphate scale, on one hand, and the use of increased amounts of phosphate, on the other, had failed to limit the apparent rate of corrosion, it was observed that the lowest rates of hydrogen evolution frequently coincided with the lowest values for the alkalinity of the boiler water.

From December, 1935, when the boiler went into regular service, until the attack on the screen tubes first was recognized as a serious problem in December, 1936, the phenolphthalein alkalinity of the boiler water, expressed as ppm of NaOH, had been maintained generally between 20 and 30, and the phosphate concentration, expressed as ppm of Na_2HPO_4, was similarly controlled within the general limits of 50 and 70, as indicated in Fig. 17. After replacement of tubes during a week-end outage, the alkalinity was increased to an average value of 70 ppm for the period from Dec. 21 to 26. When the next week-end inspection revealed that, during the week, a new tube had been thinned to the point of leaking, the alkalinity was again raised and was held at a comparatively high level, ranging from 130 to 220 ppm of NaOH, until the failure of a tube on Jan. 12, 1937, led to the general replacement of tubes in the screen.

When the boiler was again placed in operation on Jan. 25, the alkalinity and phosphate were varied experimentally, while the rate of hydrogen evolution was watched as an indication of what

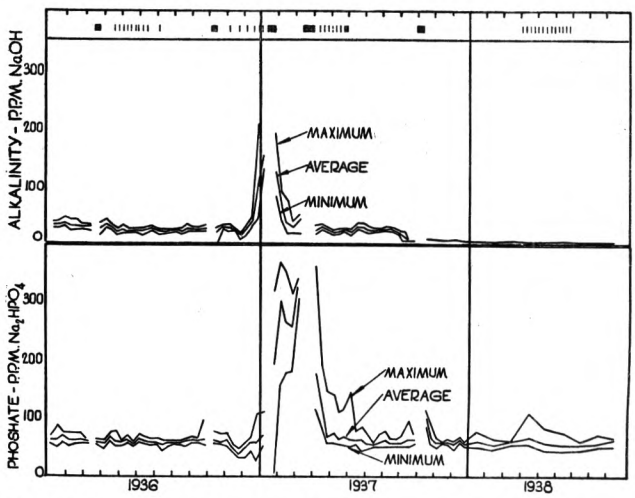


FIG. 17 RECORD OF ALKALINITY AND PHOSPHATE CONCENTRATION IN BOILER WATER AT PORT WASHINGTON

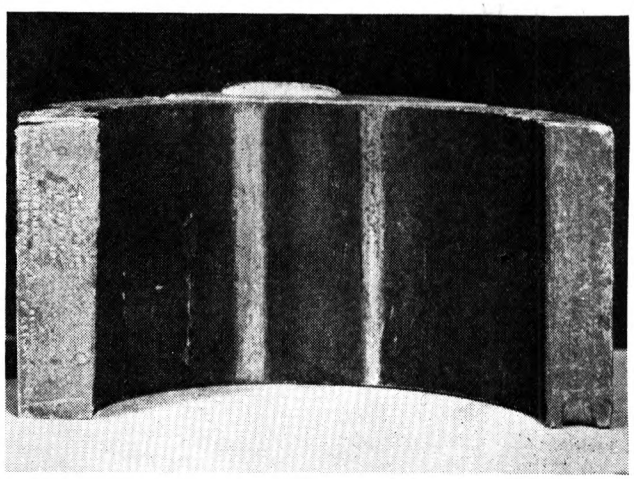


FIG. 18 APPEARANCE OF INTERNAL SURFACE OF TOP HALF OF SCREEN TUBE FROM PORT WASHINGTON AFTER OPERATION WITH LOW ALKALINITY IN BOILER WATER

(Central dark band is a thin continuous coating of magnetic iron oxide. Gray bands are exposed edges of thin layer of iron phosphate which covers remainder of surface, but is in turn covered by and intermixed with black magnetic iron oxide.)

was happening in the screen tubes. Alkalinity was progressively decreased until by the middle of February it was back in the range held during the major portion of 1936. In the meantime, phosphate concentrations were greatly increased, and held at a high level until the boiler was taken out of service in the middle of March. Specimen tubes removed at this time demonstrated that no appreciable attack had taken place since the January outage. Subsequently, the phosphate concentration was progressively lowered. For somewhat more than four months, from the latter part of April to the first part of September, both the alkalinity and the phosphate were controlled at the levels initially maintained during the first year of operation. A decision to reduce the alkalinity still further was put into practice about the middle of September, 1937, and since that time it has been held at an average value of 5 ppm of NaOH.

Removal of two more specimen tubes from the ash screen during the October, 1937, inspection confirmed the conclusion drawn

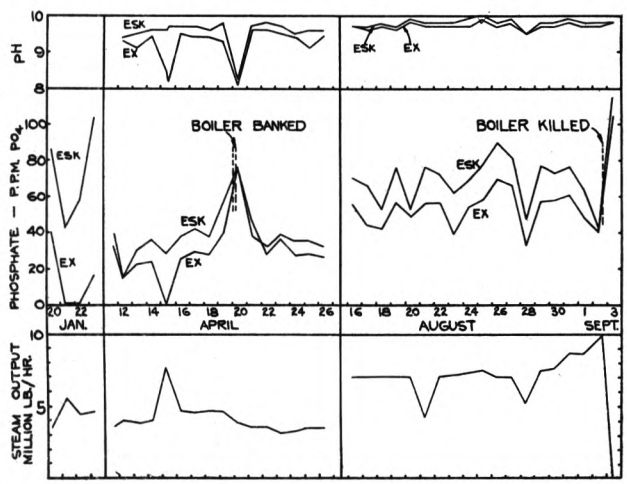


FIG. 19 COMPARISON OF TENDENCY OF SCREEN TUBES OF OLD AND OF NEW DESIGN TO STEAM-BLANKET

(Sample EX from special sampling line at top of tube of old design delivered substantially pure steam on Jan. 21 and 22 and on Apr. 15; during period from Aug. 16 to Sept. 3, no such steam blanketing was indicated by similar samples from line in tube of new design. Samples ESK came from skimmer in the east 48-in. drum.)

from the hydrogen measurements during the preceding few months that practically no attack had occurred. When four new tubes were installed in place of those removed previously and at this time for examination, the rate of hydrogen evolution went up, however, to 3 times its base value for about a week, indicating some reaction of the fresh metal surfaces with the boiler water. Pressure and load were decreased 20 per cent and the phosphate concentration was increased until the hydrogen evolution again came back to its base value corresponding to the loss of about 2 oz of iron per day from the entire unit, including the superheater.

Study of Internal Surfaces Under the Microscope. To ascertain the extent to which conditions at the internal surface of an ash-screen tube during the later period of freedom from attack differed from those during the period when failures were experienced, the polarizing microscope was applied to a study of the specimens illustrated respectively in Figs. 15 and 18. It was found that two substances were present on each of the two specimens, first the anticipated black magnetic oxide Fe_3O_4 and second, a crystalline substance similar to the natural mineral ludlamite,[8] which is regarded as a ferrous hydroxyphosphate.

[8] The crystalline substance present on the surface of the tubes was transparent, faintly blue-green, birefringent with minimum and maximum indexes of 1.655 and 1.700, and showed a definite cleavage apparently rectangular or nearly so.

Drewry has previously reported partial analyses (12) indicating that an iron phosphate was present on the internal surface of a tube removed during the later period of trouble-free operation. The microscopic study of the specimen shown in Fig. 18 has shown that the iron phosphate, which is readily dislodged from the tube in minute fragments comprising many crystals, is present over the entire internal surface of the specimen, with the exception of a dark band, 0.5 in. wide, along the top of the tube, where the steel is covered only by a thin, uniform film of iron oxide. The iron-phosphate coating is particularly evident in two gray bands, readily visible in Fig. 18 at either side of the central band. Elsewhere, the iron phosphate is partially covered by and intermixed with minute particles of Fe_3O_4, which give a black appearance to the surface.

That some slight attack had taken place along the central band at the top of the tube was evident even on visual examination. Under the microscope the surface appeared as smoothly rounded hummocks and valleys. The depth of the attack was, however, only of the order of 0.001 in.

Only a small amount of iron phosphate was found during micro-

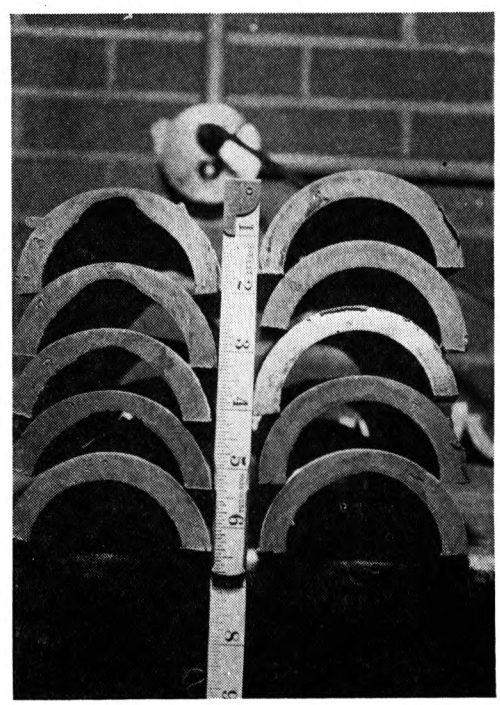


FIG. 20 CROSS SECTIONS OF ASH-SCREEN TUBE FROM LOGAN STATION SHOWING EXTENT OF GROOVING AT SUCCESSIVE 1-FOOT INTERVALS FROM SIDE-WALL HEADER

(Tube 4 SE, failed Jan. 31, 1938. Cross section at upper right is 1 ft from header, each succeeding cross section 1 ft farther away.)

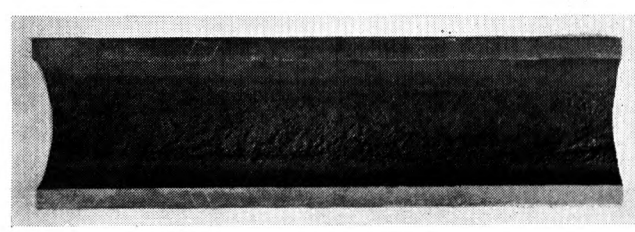


FIG. 21 INTERNAL ATTACK ON TOP OF ASH-SCREEN TUBE AT LOGAN STATION DURING OPERATION WITH NORMAL BOILER-WATER ALKALINITY

(Section 6 to 7 ft from side-wall header of original tube 4 SE, failed Jan. 31, 1938.)

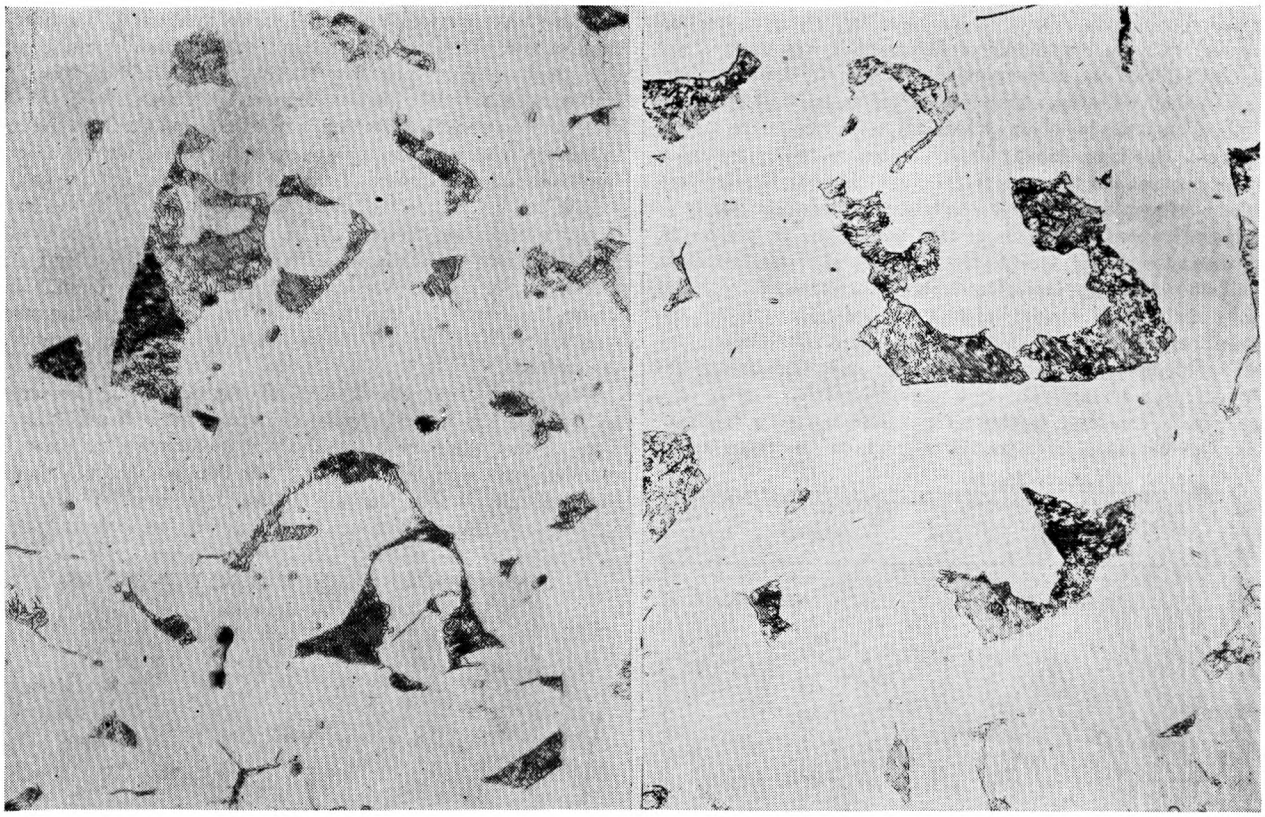


FIG. 22 STRUCTURE (×500) OF SIDE, LEFT, AND TOP, RIGHT, OF TUBE 4 SE FROM LOGAN STATION, 9 IN. FROM POINT OF FRACTURE
(Temperature at top of tube not sufficiently high to induce spheroidization of pearlite.)

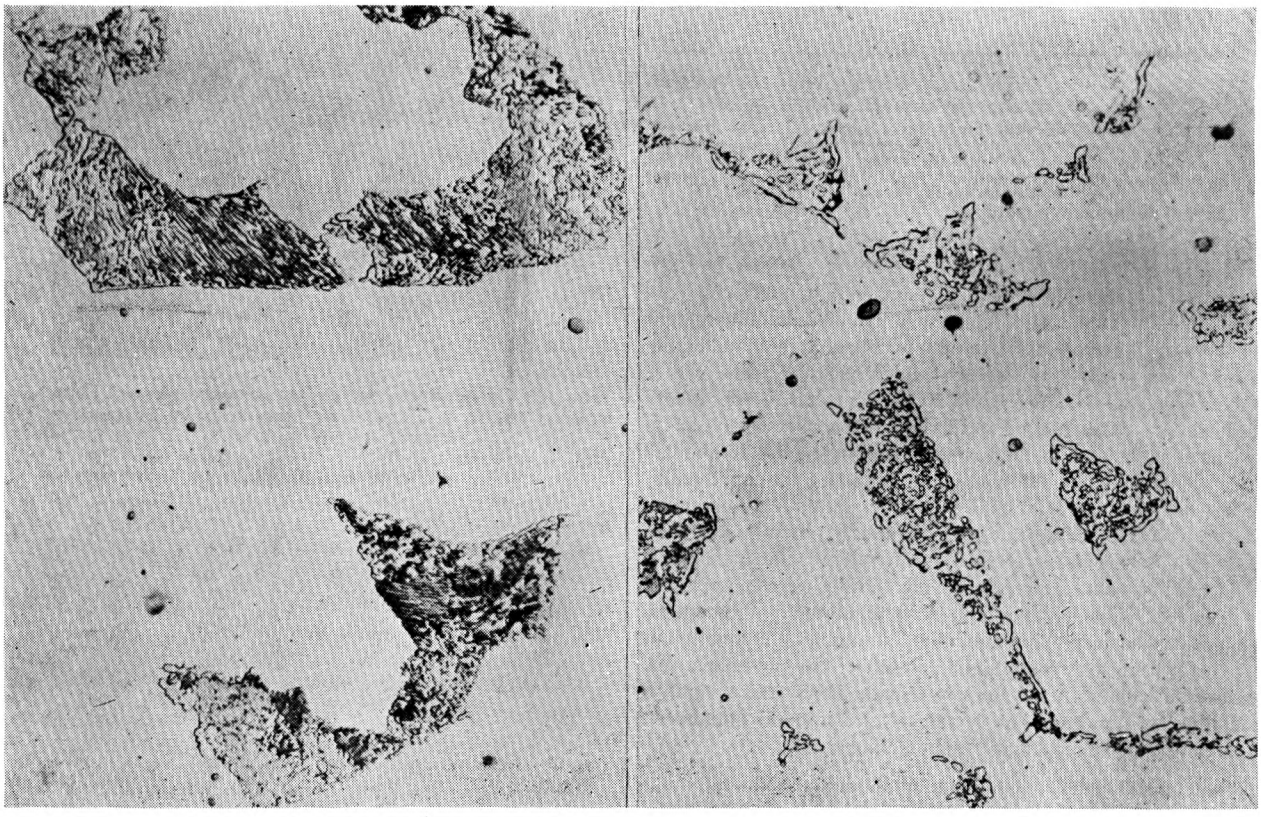


FIG. 23 STRUCTURE OF TOP OF TUBE 4 SE FROM LOGAN STATION. RIGHT, AT FRACTURE; LEFT, AT POINT 9 IN. FROM FRACTURE ×1000
(Spheroidization of pearlite definitely indicates excessive overheating in region where fracture developed.)

scopic examination of the specimen illustrated in Fig. 15, the internal surface of which was coated chiefly with black magnetic iron oxide. At the top of the tube where severe attack had occurred, this coating was thin and continuous, resembling that at the top of the practically undamaged specimen.

Logan Station

During October, 1937, a 1325-psi boiler went into operation at Logan Station of Appalachian Electric Power Company. This boiler, designed to produce 1,000,000 lb of steam per hr at a total temperature of 925 F, was operated intermittently at low loads during the fall of 1937. During a trial run from Dec. 5 to 8, inclusive, the output was raised to 725,000 lb per hr for a few hours, but averaged only 540,000. During the next operating period, from Dec. 15 to 25, the maximum and average outputs were, respectively, 550,000 and 440,000 lb per hr. Similar operation commencing Dec. 28, 1937, was terminated Jan. 7, 1938, by the development of a leak in a top-row screen tube, No. 2 from the southeast corner.

Study of Early Failures. The failure occurred about 6 ft from the side-wall header on the top of a portion of the tube that had been depressed approximately 7 in. by falling slag. Examination of this tube after removal showed that it had been attacked internally along a strip from 0.5 to 1.2 in. wide on the top of the tube. This region of attack extended over the portion of the tube between 4 and 9 ft from the side-wall header.

Inspection of the other top-row screen tubes after removal of handhole caps on the headers revealed similar corrosion, quite severe in the first six or eight tubes in each corner, but slight in the central portion of the furnace. To determine whether the tops of the tubes were steam-blanketed, a special sampling line consisting of a stainless-steel tube with an internal diameter of 0.09 in. was installed in the tube 17 NE, at the middle of the furnace. The end of the sampling line was turned upward and held in position close to the top of the tube at a point 6 ft from the side-wall header.

When the boiler was placed in operation on Jan. 20, comparison of the samples removed from the special sampling line EX, with those from the skimmer in the east 48-in. upper drum ESK, gave definite evidence of steam blanketing. As shown in the first section of Fig. 19, the phosphate concentrations of the EX samples from Jan. 20 to 23, inclusive, were always low compared to those of the ESK samples, and in two cases, substantially pure steam was issuing from the special sampling line. On Jan. 23 the run was terminated by the failure of a second top-row screen tube, 5 SE, along the top in the same manner as 2SE had failed previously.

An attempt to measure tube-wall temperatures by means of chromel-alumel thermocouples was made during the next test period, beginning Jan. 28. While the renewed tube 5 SE, which had been covered with protecting blocks, showed a temperature of only about 600 F during the brief life of the couples, the next tube 4 SE which had been left uncovered, simultaneously averaged 1250 F on the top and 640 F on the bottom, when the steam ouput of the boiler was 450,000 lb per hr. That the overheating indicated by the thermocouple was not fictitious was proved by the failure of this tube on Jan. 31 and by its appearance after removal and sectioning, illustrated in Fig. 20. Beginning at the upper right, the cross sections represent the condition of the top half of the tube at successive 1-ft intervals from the side-wall header. The grooving apparently began between 4 and 5 ft from the header, was deepest at about 6 ft, and continued to between 9 and 10 ft.

The appearance of the internal surface of the section from 6 to 7 ft from the side-wall header is shown in Fig. 21. The actual crack is about 3 in. from the 6-ft (left-hand) end. Specimens cut for metallographic examination from the side and from the top of the 7-ft end showed no significant difference in structure, as may be seen in Fig. 22, indicating that 9 in. from the point of failure the tube had not been overheated sufficiently to produce spheroidization. A specimen cut from the region of the fracture was, however, definitely spheroidized, as illustrated in Fig. 23, while the fine cracks near the main fracture at the external surface showed oxidation working inward along grain boundaries, as in Fig. 24.

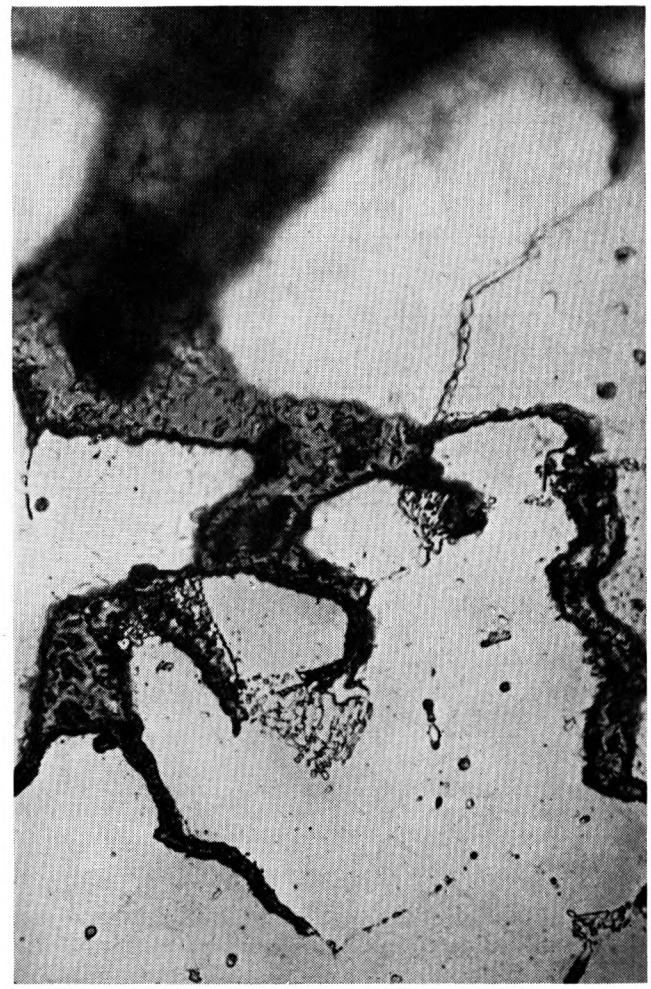


FIG. 24 SECTION NEAR EXTERNAL SURFACE OF CRACKED ASH-SCREEN TUBE FROM LOGAN STATION, SHOWING PENETRATION OF OXIDATION ALONG GRAIN BOUNDARIES, ×1000
(Note also accumulation of globular iron carbide at grain boundaries.)

Attempts to Obviate Attack. Since the most serious attack was evident in the screen tubes at the corners of the furnace, an attempt was made to improve circulation by removing the five top-row tubes in each corner, plugging tube holes 1, 2, 4, and 5, replacing each tube 3 with a new tube, and inserting baffles in the side-wall headers to isolate these sections from the central portions. In each of these corner sections there then were left a total of five tubes, one in the top row and four in the bottom, where previously there had been nine tubes. It was hoped that this change would nearly double water velocities through the remaining tubes.

While these changes were being made, the profile of the attack along the top of each tube on the south side was measured by a special calipering device, and series of thermocouples were peened into the tops of tubes 3 and 6 in each corner and tubes 17 NE

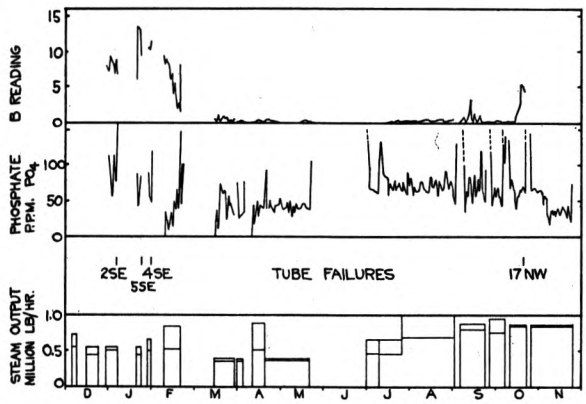


FIG. 25 RECORD OF LOAD, BOILER-WATER CONDITIONS AND FAILURES OF ASH-SCREEN TUBES AT LOGAN STATION

(Upper line for load indicates maximum, lower line average for each operating period. Multiplication of the B reading by 13.3 gives the approximate value for NaOH in ppm.)

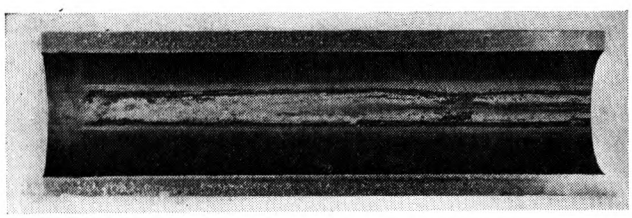


FIG. 26 INTERNAL APPEARANCE OF TOP OF ASH-SCREEN TUBE AT LOGAN STATION AFTER TWO MONTHS OF OPERATION WITH LOW BOILER-WATER ALKALINITY

(Replacement tube 6 NE, in service Mar. 14 to May 21, 1938. White deposit is iron phosphate.)

and SW at the center of the screen. When the boiler went into service on Feb. 9, however, the thermocouples failed within from 17 to 28 hr, before the steam output had exceeded 600,000 lb per hr. Although the maximum temperatures recorded in each tube during this brief period at the start ranged only from 640 to 730 F, appreciable attack occurred before the boiler was taken off the line on Feb. 21. On calipering, it was found that all tubes had lost some metal and that tubes 3 and 6 in both the SE and SW corners had suffered the most. The maximum decrease in wall thickness, approximately 0.1 in., had occurred in tube 3 SE about 3 ft from the side-wall header.

With the efforts to improve conditions by increasing velocity of flow apparently unsuccessful, two alternative procedures were urged: First, modification of boiler-water conditions in an attempt to stop the attack by chemical means, and second, modification of the design of the ash screen to eliminate or minimize steam blanketing along the top of the tubes. Since replacing the ash screen would involve rather considerable expense, and since the type of attack in question had apparently been obviated at Port Washington by decreasing the alkalinity of the boiler water (12), it was decided to maintain "Port Washington conditions" during the next operating period.

Operation With Low Alkalinity. As indicated in Fig. 25, the alkalinity was maintained at a low level not exceeding a B reading of 1 from March 14 to 28; in the succeeding periods from March 31 to April 4 and from April 11 to May 21 this maximum limit was further reduced to 0.25. During these periods the steam output did not exceed 500,000 lb per hr except on Apr. 14 and 15; a maximum of 880,000 was attained on the latter date. This relatively high rating again produced definite evidence of steam blanketing from the EX sampling line, at that time located in tube 3 NE. As shown in Fig. 19 the sample from this line on April 15 contained practically no phosphate and had a pH of only 8.2, while the samples taken simultaneously from the skimmer in the east 48-in. drum showed little change from the preceding day.

After this brief trial at higher ratings, the steam output was dropped to 460,000 and then to about 350,000 lb per hr for the rest of the period of operation, which terminated May 21. Measurements revealed little further loss of metal, but whether this was due more to low alkalinity or to low rating could not well be decided. The black bands previously observed along the tops of the tubes were still present, but were bordered and partially covered by a white deposit, evident in Fig. 26. This shows a portion of replacement tube 6 NE which had been installed new just before operation under Port Washington conditions was commenced on Mar. 14. To the eye, all of the surface except the band at the top was uniformly reddish brown. Examination under the polarizing microscope revealed that the surface of the black band was the familiar magnetic iron oxide, that the white deposit was iron phosphate similar to that previously observed in a tube from Port Washington, and that the reddish brown coating was chiefly finely divided metallic copper. Both the iron phosphate and the metallic copper could be readily rubbed off with a cloth, leaving the fundamental tightly bonded surface coating of Fe_3O_4 on the steel.

Change in Design of Ash-Screen Tubes. Before the boiler was again placed in service, the burners were swung out from the walls to decrease the diameter of the vortex in the furnace. At the same time, the entire east half of the ash screen was replaced, using tubes 3 instead of 3.5 in. in diameter, bent to give an increase in average slope from about 7.5 to about 12.5 deg. The change in slope is indicated in Fig 27. Replacement tubes of the original design were installed in 6 NW and 3, 6, 7, and 14 SW to allow direct comparison of the effect of tube design during the next operating period.

During the two months from June 30 to Sept. 2, water conditions were held closely the same in the east and west halves of the boiler, again with the low limit of alkalinity set by Port Washington. Toward the end of this period the steam output was progressively increased from a level of 700,000 to a maximum of 1,000,000 lb per hr, the rated capacity, just before the boiler was killed.

During this test period the EX sampling line was still located in tube 3 NE, but this was now of the new design. The representative data for Aug. 16 to Sept. 3 reproduced in Fig. 19 indi-

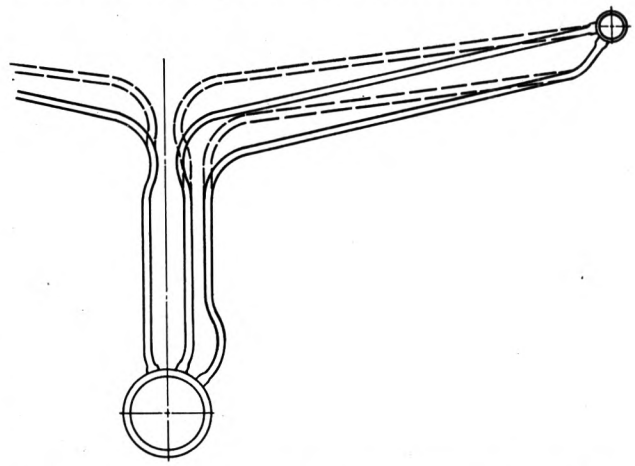


FIG. 27 CHANGE IN DESIGN OF ASH-SCREEN TUBES AT LOGAN STATION

(Broken lines: Original tubes, 3.5-in. diam, 7.5-deg slope. Solid lines: New tubes, 3-in. diam, 12.5-deg slope.)

cate a consistent dilution of the sample by the steam in the top of the tube, but not even at the maximum rating was there steam blanketing such as that shown for the old design of screen tube on Jan. 21 and 22 and on Apr. 15. This promise of better performance of the new tubes was substantiated when the boiler was inspected on Sept. 4. While all of the tubes of the old design in the west end showed the characteristic black band along the top, frequently edged and spotted with the white deposit of iron phosphate, the tubes of the new design in the east end were practically unmarked.

That the marks in the tubes in the west end were not merely carried over from previous operation was proved by the appearance of the replacement tubes of the old design. Thus 6 and 7 SW, new at the beginning of the test period, could not be distinguished from 8 SW, which had been in the boiler since it was erected.

While the appearance of the tubes of the old design attested continued steam blanketing, the actual attack during the two months under Port Washington conditions had been slight relative to that previously developed during the shorter early periods of operation with more alkaline boiler water. Reflecting the shift in the angle of the burners, tubes 8 to 10 SW lost more metal than those toward the corner or toward the center of the screen, and the maximum decrease of 0.05 in. in wall thickness occurred in tube 10 SW, 6 ft from the side-wall header.

Of the tubes of the new design in the east half of the screen, 12 SE was the only one to show a dark band along the top. This band, about 0.5 in. wide at its mid-point, extended from about 3 to about 6 ft from the side-wall header. The corresponding tube on the other side, 12 NE, showed a few discontinuous spots along the top. Measurement indicated an apparent maximum loss of 0.03 in. in wall thickness at one point in 12 SE and no loss greater than the uncertainty in measurement in 12 NE.

Effect of Increased Alkalinity on Tubes of New Design. The evidence now clearly indicated that the attack on the old design of tube had been greatly retarded by decreasing the alkalinity of the boiler water but that the fundamental cause of the attack, steam blanketing of the tops of the tubes, had been almost completely eliminated by the change in design. The next question was whether the new design would actually stand up when boiler-water alkalinity was again raised to the original operating level.

During the shutdown from Sept. 2 to 6, the deflector plates in the secondary-air passages of the upper burners were adjusted. Contrary to expectations, the change caused the flame to sweep down through the screen tubes during the period from Sept. 6 to 23. The plates were then changed back, restoring normal flame conditions during the operating period from Sept. 26 to Oct. 7. While the alkalinity of the boiler water was allowed to reach a B reading of 3 on one day during these two periods, it was not until shortly after the boiler again went on the line on Oct. 10, producing steam at a rate of about 840,000 lb per hr, that the alkalinity was progressively raised as indicated in Fig. 25. A B reading of 5.3, corresponding to a nominal concentration of 70 ppm of NaOH, was reached on Oct. 19 and held the following day. On Oct. 21 tube 17 NW, of the old design, located at the center of the furnace, developed a leak along the top which necessitated taking the boiler off the line.

Measurement of the screen tubes showed that while those of the old design in the NW quarter had lost quite generally about 0.15 in. of metal, with less severe attack indicated in the SW quarter, there had been no measurable attack on the tubes of the new design in either the NE or SE quarters. Complete replacement of the tubes in the west end with those of the new design was the result. This was carried out during a shutdown from Nov. 24 to Dec. 6, 1938. Further operation will be watched with no little interest.

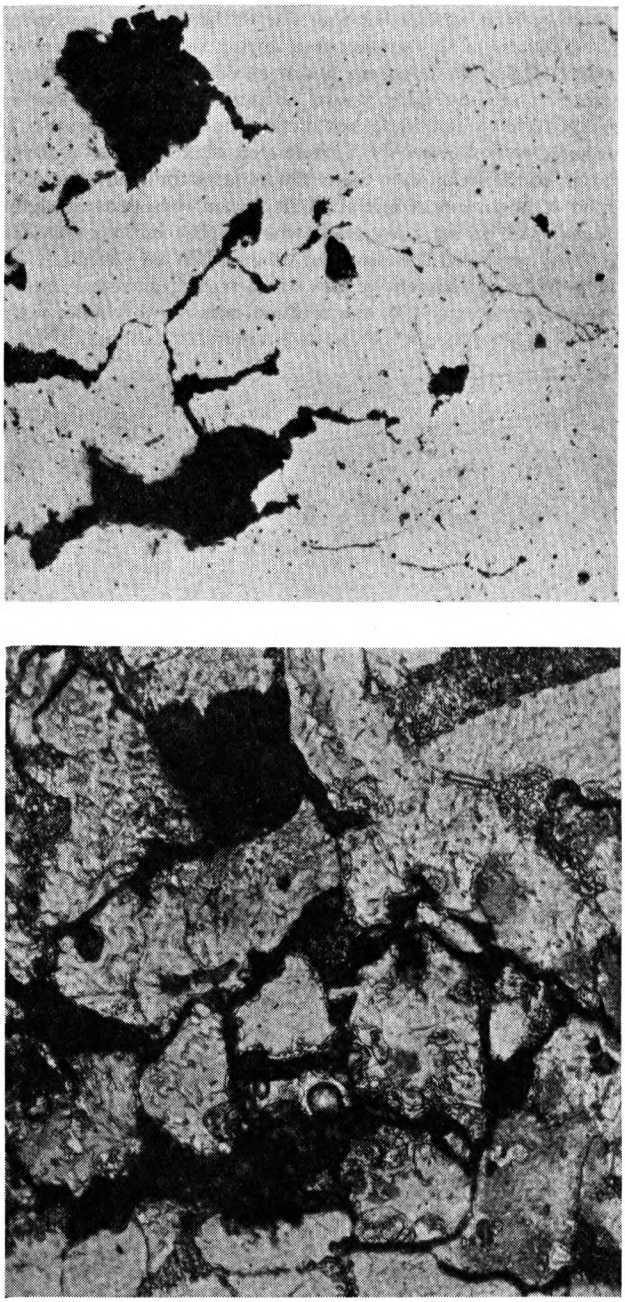


FIG. 28 STRUCTURE OF TOP OF TUBE 17 NW FROM LOGAN STATION, ×500

(Top: Unetched. Bottom: Etched with 4 per cent nital.)

Metallographic Examination of Failed Tube. Under the microscope, specimens from the area of failure in tube 17 NW revealed not only a structure indicative of excessive overheating, but also intergranular cracks developing from the internal surface. As may be seen from Fig. 28, the finer cracks in the unetched condition appeared only as faint lines but etching developed these into relatively broad black bands which seemed to follow the boundaries of the original grains.

Hydrogen Evolution as a Warning Signal. Theoretically the hydrogen content of the steam from a boiler should be a measure of the amount of boiler metal oxidized by reaction with the water. Actually the hydrogen measurements made on the steam from each end of the boiler at Logan Station daily since March 15,

1938 have correlated rather poorly with the other measurements of the attack on the screen tubes during the various operating periods. Thus the apparent hydrogen content was observed to increase markedly with steam output during one operating period at low alkalinity, but little attack was indicated by measurements of internal tube diameter. Likewise the hydrogen content of the steam went higher for a longer time during another period of operation at low alkalinity when there was negligible attack, than during the period when experimental increase in alkalinity produced the failure of tube 17 NW on Oct. 21.

The hydrogen content of the steam would, of course, be affected to some extent by the reaction between the boiler water

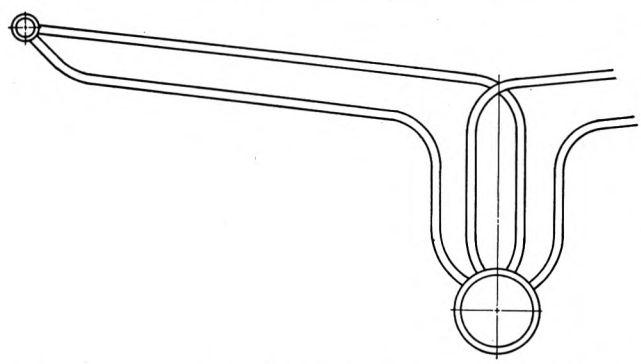


FIG. 29 ELEVATION OF ASH-SCREEN TUBES IN BOILER 5 AT ROUGE PLANT
(Tube diameter, 3 in.; slope, 7 deg.)

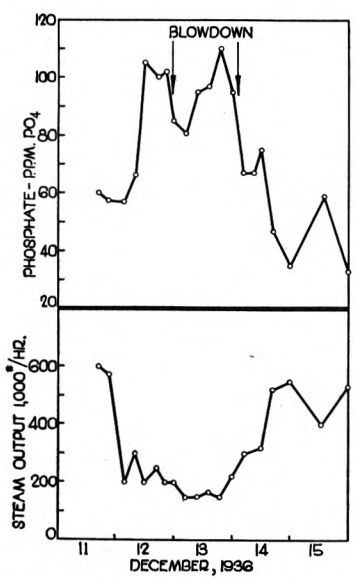


FIG. 30 INVERSE RELATION BETWEEN PHOSPHATE CONCENTRATION AND STEAM OUTPUT DUE TO "HIDING OUT"

and any excess of the ferrous sulphate used as an oxygen scavenger. However, 1 lb of iron combined in the form of ferrous sulphate would produce only 61 liters of hydrogen, as against 244 liters from 1 lb of metallic iron.[9]

[9] These values are based upon the equations:

$$3FeSO_4 + 6NaOH \rightarrow 3FeO \cdot H_2O + 3Na_2SO_4$$
$$3FeO \cdot H_2O \rightarrow Fe_3O_4 + 2H_2O + H_2$$
$$3Fe + 4H_2O \rightarrow Fe_3O_4 + 4H_2$$

The reactions of iron and of ferrous hydroxide with water to form Fe_3O_4 and H_2 have been observed experimentally by E. Berl and F. van Taack, *Archiv. für Wärmewirtschaft*, vol. 9, 1928, pp. 165–169.

Ford Motor Company

Power Plant No. 1 of the Rouge plant of the Ford Motor Company contains three 1350-psi dry-bottom boilers, which have experienced no difficulty with internal attack along the tops of the ash-screen tubes. Boilers 1 and 3, each with a rated output of 700,000 lb of steam per hr, have been in operation since 1931, and boiler 5, rated at 800,000 continuous, 900,000 for 2 hr, and actually operated on test at 1,040,000 lb per hr, went into service July 15, 1936.

Boiler 5 is in many respects similar to the Logan boiler. However, the screen tubes are 3 in., while the original tubes at Logan were 3.5 in. in diameter. The ash screen, which has a slope of approximately 7 deg, is shown in elevation in Fig. 29.

Initially the superheat on boiler 5 tended to run from 30 to 40 F too low and adjustment of the burners upward was necessary to attain 900 F. No such adjustment was necessary on the earlier boilers 1 and 3.

Boiler-Water Conditions. In boiler 5, as well as in the older high-pressure boilers at the Rouge plant, the water conditioning has always been carefully controlled. Only infrequently does the boiler water show concentrations which are outside of the control limits of from 10 to 20 ppm of OH and from 60 to 80 ppm of PO_4.

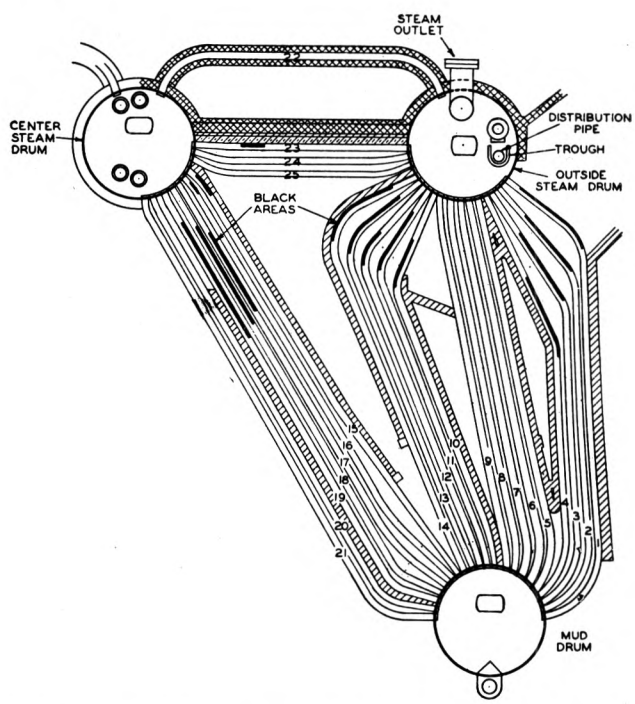


FIG. 31 LOCATION OF BLACK AREAS IN TUBES OF NO. 2 BOILER AT BEACON STREET
(Tubes 1 to 16 were inspected with a mirror. The lower ends of tubes 10 to 14 could not be seen because of bends. Tubes 17, 18, and 19 were removed and opened for examination.)

Although inspection has consistently shown all three high-pressure boilers to be free of attack in the ash-screen tubes, the boiler-water analyses show definite evidence of hiding out of phosphate during normal operation followed by its return to solution on reduction of load during the night or over a week end. The results of a test on boiler 5 are shown in Fig. 30.

This case has been included as representative of the considerable number of boilers in which no serious difficulty has yet been encountered, even though steam blanketing apparently is present.

Generating Tubes in a Bent-Tube Boiler

Beacon Street Heating Plant

The cracking of tubes in the double-set bent-tube boilers in the Beacon Street Heating Plant of The Detroit Edison Company has been reported previously (13), but is discussed here with some additional information. When the first two of these boilers went into service in September, 1926, they were the largest in the world with respect to heating surface. It was soon found, however, that the desired output of 435,000 lb of steam per hr from each boiler could not be reached without excessive carry-over. After trial of mechanical separators had failed to produce sufficiently dry steam, it was found by experiment that lowering the working water level achieved the desired result. Accordingly, during the summer of 1928, the water columns on each boiler were moved down 13 in. from their original location at the center line of the 54-in. drums.

For nearly eight years the boilers operated without difficulty. During inspections black areas were noted on the upper internal surfaces of many of the tubes, but little importance was attached to this condition until a No. 19 tube in boiler 2 developed a leak in February, 1936. When this was removed and sectioned, a thin, velvety-black coating of magnetic iron oxide was found on the high side, extending over a distance from 3 to 7 ft downward from the center drum, and in this area were many circumferential cracks. The tube wall had not been thinned, nor was it bulged.

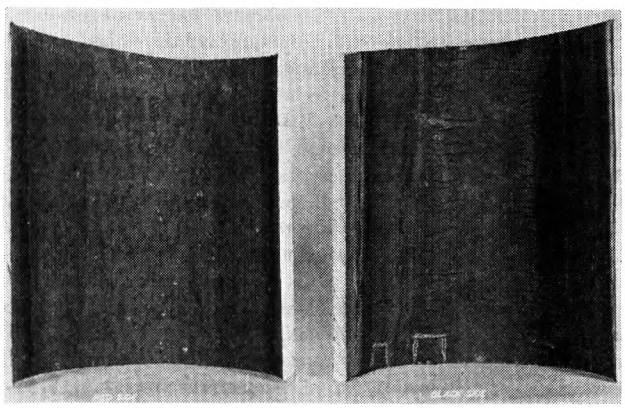


Fig. 32 Cracking as a Result of Steam Blanketing in a Bent-Tube Boiler at Beacon Street Heating Plant

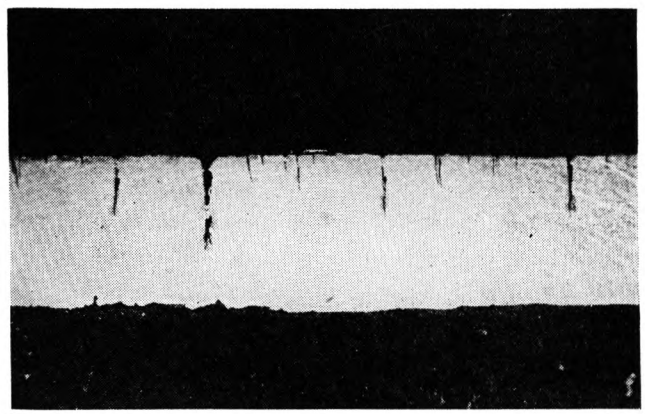


Fig. 33 Cracks Developed From Internal Surface of Boiler Tube at Beacon Street Heating Plant



Fig. 34 Decarburized Region Adjacent to Crack, ×500

However, along this portion of its length the tube was warped about 3 in. out of line, due to an apparent shortening of the black side.

Subsequent detailed examination revealed black areas as indicated in Fig. 31, in every case on the high side of the tube and in most of the tubes across the boiler. The No. 21 and No. 20 tubes, in front of the first baffle and closest to the furnace, were notably unaffected. The No. 19 tubes, however, not only showed well-defined black areas, but the majority were cracked in the same manner as the first tube removed. Fig. 32 shows the typical appearance of the undamaged front half and badly cracked rear half of one of these tubes.

Inspection revealed that boiler 1, and also boilers 4 and 5 installed respectively in 1929 and 1927, all showed black areas similar in appearance and location to those in boiler 2, but no cracks were apparent on visual examination of specimen tubes. A rough bending test caused the development of obvious cracks in the black areas but not in the normal areas of tubes from boiler 1. No cracks were revealed by this test in specimens from boilers 4 and 5.

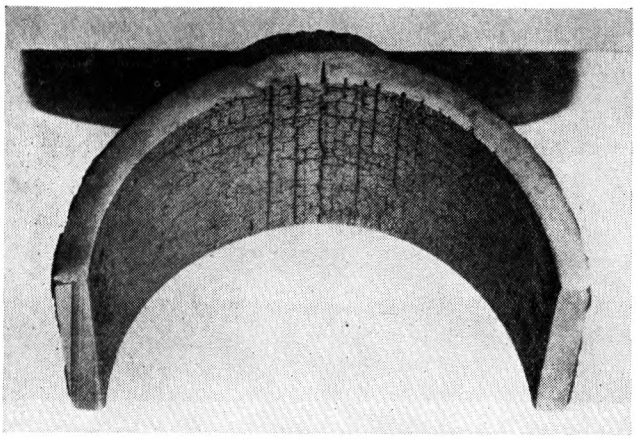


FIG. 35 LONGITUDINAL AND TRANSVERSE CRACKS IN TOP OF STEAM-CIRCULATING TUBE FROM 600-LB BOILER AT PIEDMONT PLANT

The typical character of the cracks is illustrated in Fig. 33. Microscopic examination showed them to be transcrystalline and revealed that the tube metal had been locally overheated. It was noted, moreover, that the inside surfaces of the cracks and the inside surface of the tube adjacent to the cracks were almost completely decarburized, as may be seen in Fig. 34.

Further evidence concerning the cause of the cracking was supplied by operating tests with thermocouples attached to tubes in the areas where the black-oxide coating had developed. It was found that, at the water level which had been employed, areas on tubes 15, 16, 17, 18, and 19 remained continuously at temperatures of from 600 to 1200 F, but that when the water level in the drum was raised, normal temperatures of from 360 to 460 F resulted. Phenomenally rapid and extreme changes in temperature were observed. Thus, on many occasions, raising or lowering the water level in the gage glass only $^1/_2$ in. produced a change of nearly 800 F in less than 1 min at some points on the high side of the tubes.

Overheating due to steam blanketing was recognized as the immediate cause of the difficulty. From the temperature studies, it was decided that the water level must be raised 9 in. toward the original level to establish adequate circulation through the tubes which had been affected. This again raised the problem of wet steam, which was solved by installing special baffles in the center steam drum and adding 184 new steam circulators between the center and the outside drums in each boiler.

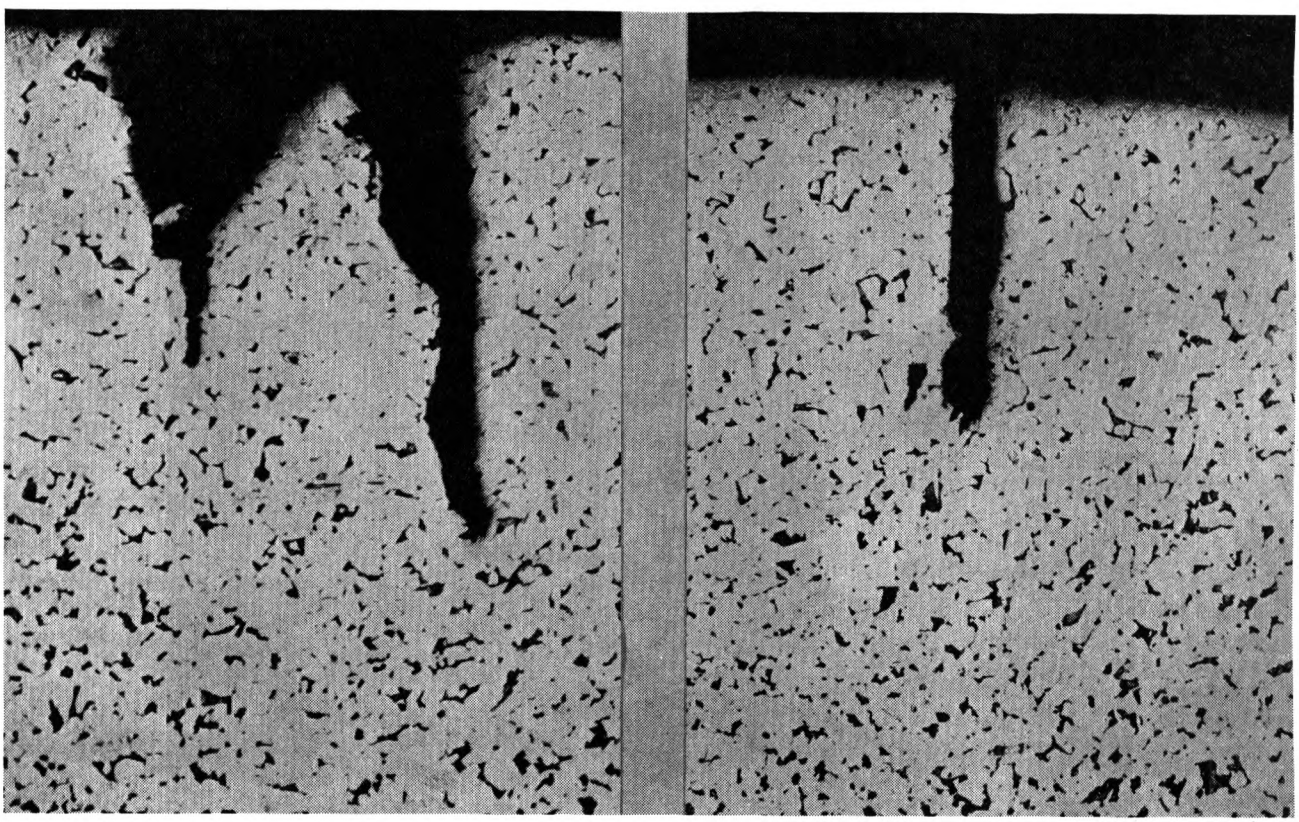


FIG. 36 CHARACTER OF CRACKS IN TOP OF STEAM-CIRCULATING TUBE

Cracking of Steam Circulators in a Bent-Tube Boiler

West Virginia Pulp and Paper Company

Boilers 22 and 23 in the Piedmont Plant of the West Virginia Pulp and Paper Company are 600-psi three-drum bent-tube units, which went into service respectively in December, 1934, and May, 1938. Cracking of steam-circulating tubes in each of these boilers has posed an interesting problem.

The first cracking in boiler 22 was observed in the spring of 1937, and all of the trouble developed in the second row of tubes from the top, at or close to the bend adjacent to the steam drum. Failure occurred first in the third and fourth tubes from one side and then in the fourth and fifth tubes from the other side. Efforts were made to continue these in service by welding up the first cracks observed but when further cracking led to the removal of tubes it was discovered that the internal surfaces were literally disintegrated. Fig. 35 shows the network of longitudinal and transverse cracks on the upper half of one of the tubes after pickling in acid to dissolve the thin coating of calcium phosphate present when it was removed from the boiler.

Under the microscope the tube wall had the typical appearance shown in Fig. 36, suggesting that the trouble was due to stress corrosion. The steel showed no signs of spheroidization.

By segregating the discharge from the wall tubes into the intermediate drum so that it was forced to pass out through the top row of circulators to the steam drum, conditions in the second row of circulators in boiler 22 were materially improved. This arrangement was accordingly installed in boiler 23. This did not prevent the development of the same type of cracking, which first became evident during the fall of 1938. In boiler 23, however, the failures occurred in the third instead of the second row from the top, and black oxide was evident on the ceilings of the tubes in the five lower rows.

A number of changes have been made to increase the ratio of water to steam in the circulating tubes. The third row of tubes, in which the cracks had developed, was removed and the holes were blanked; discharge from the upper waterwalls was divided between the top two rows; discharge from the lower waterwalls formerly entering the mud drum was routed into tubes delivering directly to the steam drum; and the baffling in the gas passage through the vertical tubes between the steam and mud drums was changed in an effort to increase the rate of downward flow of water through the tubes. While further modifications may be necessary for a final solution of the problem, recent operation indicates that the condition is much improved if not entirely cured.

The Relation Between Low pH, Pitting, and Stress Corrosion

Operation with little or no caustic alkalinity has proved effective in minimizing the attack of excessively concentrated boiler water on the tops of steam-blanketed tubes, as indicated in the preceding discussions of Port Washington and Logan Stations. While it is known that boilers fed with condensate containing consistently little dissolved oxygen may operate for years at a relatively low pH level with no difficulty from pitting or stress corrosion, nevertheless the experiences of a number of plants suggest that low alkalinity may contribute to these types of attack. As examples, the histories of Colfax Station of Duquesne Light Company and of Edgar Station of Boston Edison Company may be cited.

Pitting and Stress Corrosion at Colfax Station

At the time the first unit of seven 265-psi straight-tube cross-drum boilers was installed in Colfax Station of Duquesne Light Company in 1920, operation with pure distilled water was considered the ultimate answer to the then almost universal problem of boiler scale. With the exception of a few weeks during 1924, when soda ash was fed to the head tanks, and of some tests with sodium silicate on individual boilers, the plant was operated without chemical treatment until November, 1929. In the meantime a second unit of seven boilers had been added in 1922, a third unit of five in 1924 and 1925, and a fourth unit of three in 1927. While the design changed in detail from unit to unit, the general arrangement of the heat-absorbing surface is indicated in Fig. 37, which shows a boiler of the third unit. Boilers of the first two units were stoker-fired, while those of the third and fourth units were fired with pulverized coal.

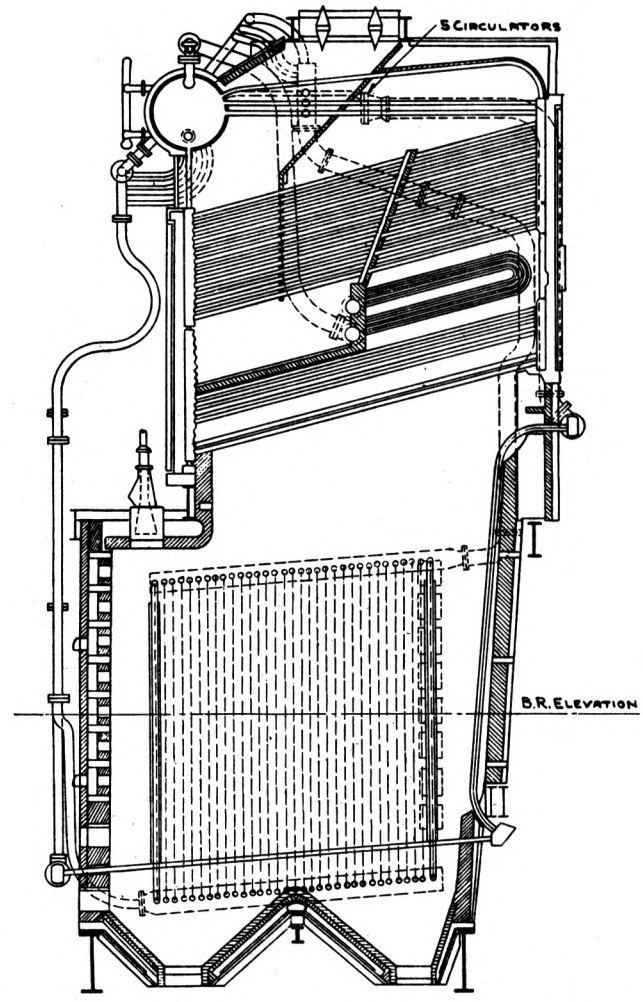


Fig. 37 Cross Section of Third-Unit Boiler at Colfax Station

Efforts to Reduce Oxygen Content of Feedwater. According to the information available when Colfax Station was designed, serious corrosion was not anticipated if the oxygen content of the feedwater was kept below 0.5 ml per liter. In retrospect, it is not surprising that this amount of oxygen introduced into boiler water at a low pH should have caused noticeable corrosion during the early operation of the plant. This led to the practice of covering the surfaces of the drums with a special paint. In 1923 and 1924 pitting of the tubes in the boilers of the first unit was recognized as a problem, and oxygen as a primary factor. Painting of the generating tubes, started late in 1925, proved ineffective.

A small deaerator installed to treat the 2 per cent make-up from the evaporators and auxiliary supplies went into service

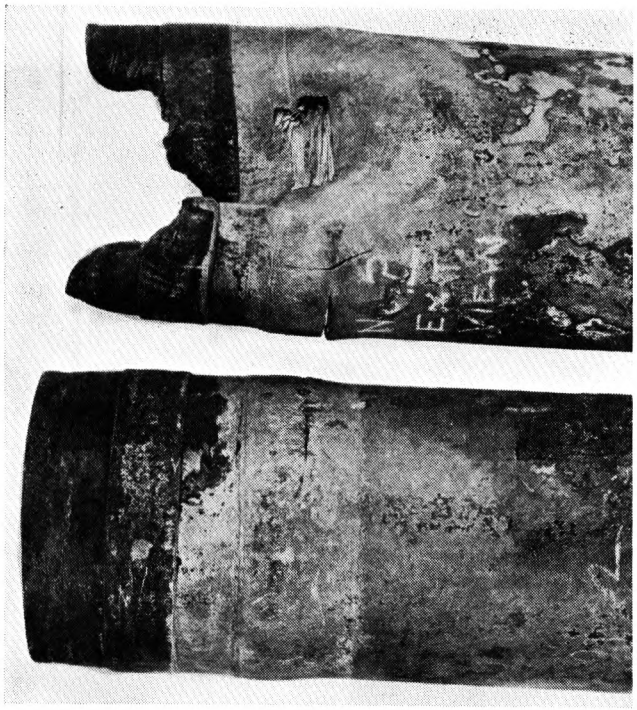


FIG. 38*a* EXTERNAL VIEWS OF CRACKED NIPPLE



FIG. 38*b* INTERNAL VIEWS OF CRACKED NIPPLE

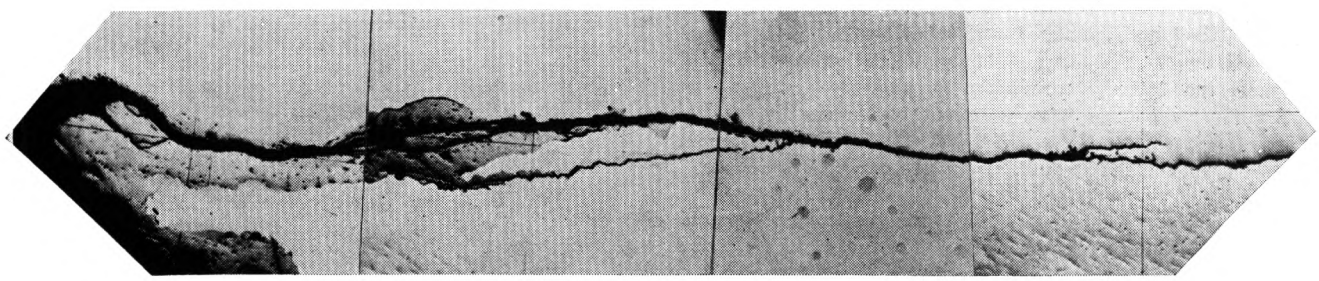


FIG. 40 TYPICAL CRACK IN HEADER, ×20

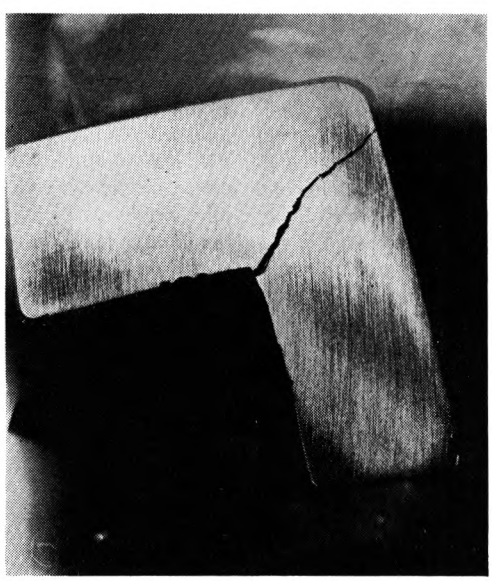


FIG. 39 STRESS-CORROSION CRACK IN CORNER OF HEADER

in April, 1924, and in September, 1925, closed heaters were substituted for the barometric condensers which had served as feedwater heaters, while possible sources of air leakage into the system were closely checked. As a result, the oxygen content of the feed was reduced from 0.5 to a maximum of 0.2 ml per liter (14) during full-load operation of the first and second units. A second deaerator was installed in 1927 with the fourth unit of boilers. In 1930 the small deaerator installed in 1924 was replaced by a larger one to deaerate surge return to the condensate system.

Cracking Due to Stress Corrosion. Not only did pitting continue in spite of these efforts, but nipples in the downtake headers and the headers themselves began to crack with increasing frequency as a result of what would now be called stress corrosion, which may have been developing for years. Figs. 38 and 39 illustrate typical cracking in a nipple and a header, respectively. The character of the cracks in the headers is shown by the photomicrograph of Fig. 40. All cracks developed from the internal surface.

Beginning in 1928, as a precautionary measure, all of the downtake nipples on the boilers at Colfax Station, except those in the newly installed fourth unit, were replaced with particular care to minimize deformation during rolling in.

In the boilers of the first and second units, which were equipped with umbrella-type feed nozzles in the drums, the cracking had occurred chiefly in the nipples in the downtake headers immediately below these nozzles. Boilers in the third and fourth units, equipped with perforated feed pipes extending the length of the drum below the water level, developed cracking of the nipples more at random. It was found however that the oxygen content of the boiler water was lower in the first and second units, where the entering feedwater was partially deaerated by spraying through the steam space,[10] than in the other units. The combined advantages of deaeration and distribution were secured in the boilers of the first two units by substitution of troughs for the feed nozzles during the period from May, 1929, to November, 1930.

Chemical Conditioning of Boiler Water. It should be remembered that during all of these efforts to reduce the oxygen content of the feedwater and to distribute the feed, the water in the boilers contained only what dissolved substances might be introduced by evaporator carry-over or condenser leakage, so that the pH inevitably remained low, ranging between 7 and 9. In November, 1929, however, after some tests on a single boiler, the feed of trisodium phosphate to all boilers was started for the combined purposes of preventing the formation of a thin but dense scale on the heat-transfer surfaces, and of minimizing corrosion. While control on the basis of maintaining from 25 to 50 ppm of PO_4 in the boiler water did eliminate scale, the average pH of the water in the boilers was increased only to about 8.8, and corrosion and cracking continued, although to a decreased extent.

In June, 1932, the trisodium phosphate was supplemented by caustic soda to maintain between 17 and 34 ppm of OH in the boiler water. Installation of steam seals on the surge tanks the following March reduced the dissolved oxygen in the feedwater to the range from 0.02 to 0.05 ml per liter, as determined by the standard Winkler procedure, and the feed of sodium sulphite was commenced in May, 1933. No failures due to cracking have occurred since 1933.

At the time the feed of sodium sulphite was started, corrosion was occurring primarily at the uptake ends of the tubes in the lower deck and chiefly on the lower half of the tubes. In the stoker-fired boilers of the first two units, the treatment with sodium sulphite substantially stopped this corrosion which, however, persisted in the pulverized-fuel boilers operated at somewhat higher ratings. It was found, as a result of a series of tests begun during the latter part of 1935, that ferrous hydroxide effectively inhibited this type of corrosion. As a result, it has been progressively substituted for sodium sulphite on all pulverized-fuel boilers. The reason for the improved results with ferrous hydroxide in these boilers is not known. During the last two years, there has been practically no indication of active corrosion at Colfax Station, although occasional failures of previously pitted tubes have occurred.

Summary of Replacements Due to Corrosion. Condensed in Table 7 is a record of replacements of generating tubes, steam circulators, and headers on account of internal corrosion. Many of the tube replacements were made before actual failure, particularly those in the third unit in 1930 and 1932. Not included in the table are a considerable number of tubes removed primarily for observation in the early years of the station, and the complete replacement of all downtake nipples in the first three units, which was started in 1928.

A number of interesting features are apparent in Table 7. Thus 10 out of 11 cases of header cracking occurred in the first unit, with the remaining one in the third unit. Cracking of nipples was distributed chiefly between the first and third units, with 16 in the former, 14 in the latter, and only 4 in the second unit.

The third unit has undergone by far the most extensive replace-

[10] The oxygen survey given in (14) indicated only from 0.04 to 0.06 ml of oxygen per liter in the boiler water at various locations when the entering feedwater contained 0.2, and the saturated steam 0.24 ml per liter.

TABLE 7 TUBE AND HEADER REPLACEMENTS AT COLFAX STATION DUE TO INTERNAL CORROSION

Unit	Boiler	1920	1921	1922	1923	1924	1925	1926	1927	1928	1929	1930	1931	1932	1933	1934	1935	1936	1937	1920 – 1937 Total
				N	N	G N	N	N	S N	G S H N	G S N	G S	G S H	G S H	G S H	G S	G S	G S	G S	G S H N
1	1		(2/9)	1		1					1 2	14			1					16 0 0 2
	2	(12/8)			1	5					6		4	1	2 1					19 0 1 5
	3		(2/7)							1		4								1 0 1 0
	4		(1/4)				1		1		2	5 2		5		2				8 0 3 5
	5		(5/20)						1											0 0 1 0
	6	(12/8)									5	1		1	2					8 0 3 5
	8	(12/22)		1		1					3	1		1						5 1 1 0
Total				2	1	4 1	1		2	0 0 1 11	25 1 5	4	5 0 1	5 0 2	5 0 1	2				48 1 10 14
2	7		(10/18)																	0 0 0 0
	9				(4/16)						1	17 5		2	1			10		21 5 0 2
	10		(11/16)								2									0 0 0 2
	11		(2/27)								1	1	1	9						9 2 0 1
	12		(11/26)													1		1		1 0 0 0
	13					(2/16)					5									0 4 0 0
	14		(12/2)							1				1						0 0 1 0
Total										4	1 5 0	17 4	0 1 0	12 0 0	1 0 0	0 1		11		42 8 0 4
3	15					(9/17)		1			1	47			7		1		2	10 47 1 1
	16					(10/7)			4		5	40		89 5	2	4	4			61 48 0 4
	18						(6/21)			1	4	46		26	41	-20	2	2	1	90 51 0 0
	20						(7/18)					47	1	167 20	2	5 2				175 69 0 0
	22						(8/7)		9			47			5		1	2		15 47 0 2
Total								1	15	0 1 0 0	0 9 1	227	1	282 25	57	52 2	8	4	5	587 282 1 14
4	17								(12/11)					5	1		1			7 0 0 0
	19								(12/17)	1 1					1					2 1 0 0
	21								(10/21)							1				1 0 0 0
Total										1 1 0 0				5	2	1	1			10 1 0 0
Total for all four units				2	1	4 1	1	1	2 15	1 2 1 15	24 15 4	21 281	6 1 1	282 25 2	65 1	55 5	9	15	5	485 275 11 34

G - Generating tubes.
S - Steam circulators.
H - Headers.
N - Nipples.

Dates in parentheses indicate when respective boilers were placed in service.

ment of tubes due to corrosion, with 367 out of a total of 465 generating tubes, and 262 out of a total of 273 steam circulators. The general replacement of the latter in 1930 in all the boilers of the third unit may have been due chiefly to overheating produced by secondary combustion in the upper part of the furnace adjacent to these necessarily steam-blanketed tubes. This view is substantiated by analyses which show that the top halves of the tubes were coated primarily with iron oxide, while the deposit on the lower halves comprised typical scale-forming constituents.

The experience at Colfax Station might be summarized briefly by the statement that, after reduction of dissolved oxygen in the feedwater and maintenance of higher alkalinities and oxygen-scavenging chemicals in the boiler water, pitting and stress corrosion have apparently been stopped.

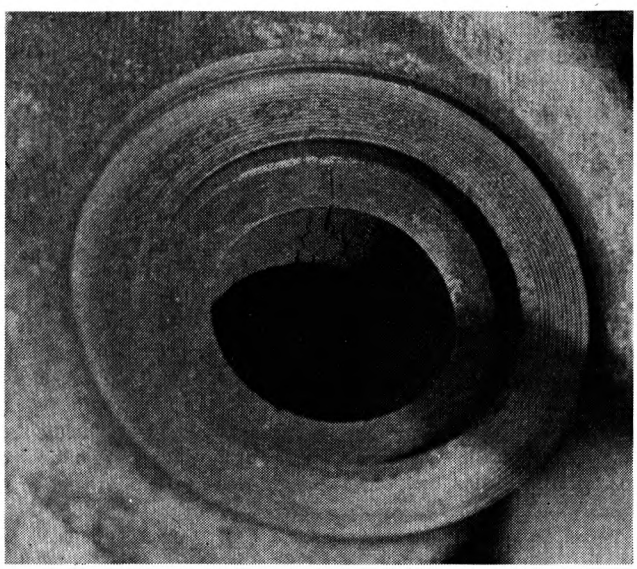


FIG. 41 CRACKS IN WATER COLUMN AT LOWER CONNECTION, BOILER 6, EDGAR STATION

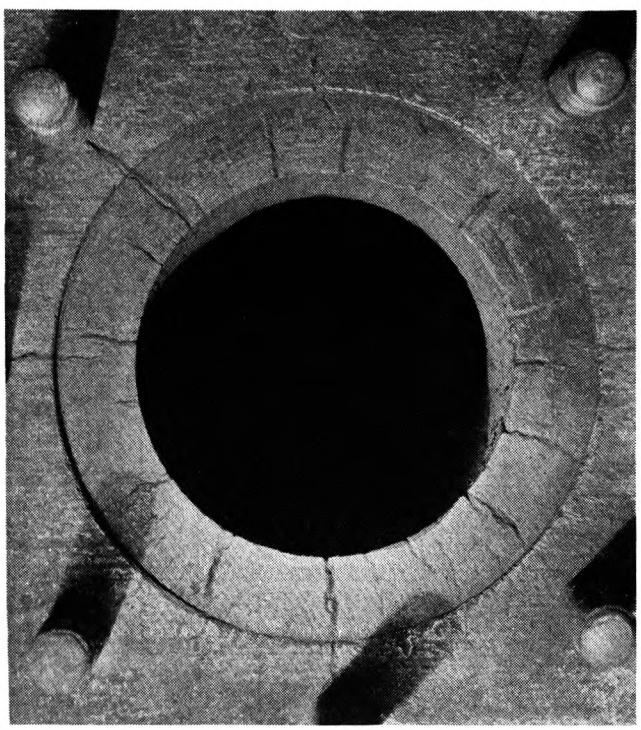


FIG. 42 CRACKS IN DRUM AT FEED-LINE CONNECTION, BOILER 5, EDGAR STATION

STRESS CORROSION AT EDGAR STATION

In Edgar Station of Boston Edison Company there are five high-pressure boilers. No. 3, installed in 1925 to operate at 1200 psi, was one of the pioneer units in this pressure range. Boilers 5 to 8, designed for 1400 psi, went into service successively from 1927 to 1929.

Late in 1935, it was found that the water columns on boiler 6 were cracked at the bottom companion joint, as indicated in Fig. 41. The same condition was then observed on boiler 5. In no case, however, were any cracks found in the corresponding joints at the upper end of the water columns.

The following year, cracks were discovered in boiler 3 in the cutouts for the feedwater connections, where the drum itself formed part of the feed line. Later inspections showed similar

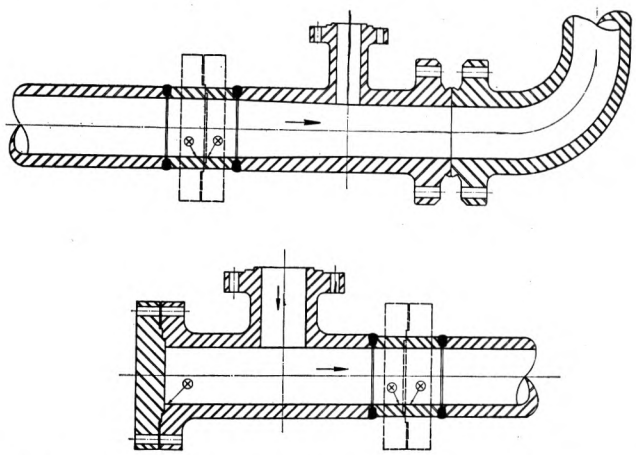


FIG. 43 LOCATION OF CRACKS AT BOTTOM OF FLANGED CONNECTIONS AT ENDS OF SUPERHEATER INLET AND OUTLET HEADERS, BOILER 5, EDGAR STATION

cracking in boiler 5, both in the drum at the feedwater connections and in the flanged joints of the superheater headers, and in boiler 6, in the Van Stone laps of the supply and riser tubes to the waterwalls.

The most serious condition was that in the drum of boiler 5. Fig. 42 shows how the cracks had developed to the stud circle on the inside surface. This boiler was removed from service in February, 1938, for rebuilding. The details of the inspection made at this time and of the changes made to obviate further cracking have been given by White (15).

Both repeated stress and contact with water at a low pH seemed to be important factors in the development of the cracks in the various locations. The feedwater to the drum, comprising condensate and evaporated make-up, was substantially neutral, with a pH between 6.9 and 7.2, and a temperature between 250 and 350 F. As a result of the intermittent introduction of feedwater by the automatic regulator, the temperature in the wall of the hole in the drum must have varied repeatedly between this range and the temperature of the water in the drum, about 585 F, presumably setting up localized thermal stresses in the steel.

On both the inlet and the outlet headers of the superheaters, tees are connected by flanged companion joints, as indicated in Fig. 43. Cracks were found in these joints at the bottom only, where a definite water line showed that condensate had col-

lected at various intervals. No cracks were found in the side outlets of the tees, which were in a position to drain readily, or in the Sargol joint on the outlet fitting.

In the water columns, normal condensation kept the bottom connections in contact with water which showed pH values as low as 7.3 when samples were withdrawn slowly to avoid flow of water from the drum back into the column.

The only cracking where water at a higher pH may have been in contact with steel was in the Van Stone laps on the supply and riser tubes to the waterwalls of boiler 6. Since the supply tubes take off from the drum, it might be expected that boiler water with a pH of 11 or higher would pass through them. Short-circuiting of feedwater down the supply tubes is, however, possible. No means are available for checking the actual pH of the water in this circuit.

The concentration of dissolved oxygen which may have contributed to the stress corrosion is not known. Although mechanical deaeration eliminated oxygen from the feedwater to the point where it could not be detected in the standard Winkler test, some trouble with pitting was encountered prior to the adoption of sodium sulphite as a chemical scavenger during September, 1932.

Interpretation of Evidence

By piecing together evidence from the various case histories a reasonably complete picture of steam blanketing and its effects may be sketched. The most direct proof of the development of a steam blanket in a slightly inclined tube is afforded by the experiments at Logan in which substantially pure steam issued from a special sampling line located with its intake at the top of one of the original ash-screen tubes, Fig. 19. The overheating which must result from high heat input to a blanketed tube was observed by means of thermocouples at San Antonio and at Beacon Street, and was indicated by the microstructure of the steel at South Amboy (Fig. 5) and at Logan (Fig. 23). The hiding out measured at San Antonio (Fig. 7) and Rouge (Fig. 30) may reasonably be ascribed to localized evaporation on the ceiling or at the water line along the sides of steam-blanketed tubes.[11]

Grooving and Cracking of Tubes

Grooving such as occurred at South Amboy (Fig. 4), Port Washington (Fig. 15), and Logan (Fig. 20) is believed by the authors to develop where a relatively constant condition of steam blanketing causes excessive localized concentration of a normally alkaline boiler water along the ceiling of a highly overheated tube. Cracking such as was experienced at Beacon Street (Fig. 32) and at Piedmont (Fig. 35), on the other hand, is attributed to repeated quenching of overheated steel by intermittent contact with boiler water.

It is interesting to note that grooving occurred in the high-pressure boilers at Mannheim, Germany, in 1929 (16). The published photographs show tube walls thinned in the same manner as those illustrated in the present paper.

An early case of cracking of upper-row tubes near the uptake headers occurred in 1925 at Hell Gate Station of the United Electric Light and Power Company. The study of tube temperatures and the application of cores to prevent overheating has been briefly described elsewhere (17).

Effect of Boiler-Water Composition

Maintenance of low controlled alkalinity characteristic of the best contemporary operation did not prevent failures of steam-blanketed tubes at South Amboy, San Antonio, Springdale, Waterside, Rivesville, Port Washington, Logan, Beacon Street, or Piedmont. Increasing the concentration of sodium hydroxide aggravated the trouble at South Amboy (Fig. 6) and at Port Washington (Fig. 17), while substantial elimination of this substance from the boiler water at Port Washington and at Logan minimized grooving (Figs. 17 and 25).

On the other hand, Rouge is representative of many plants which have experienced no difficulty when operating with normal low concentrations of sodium hydroxide in the boiler water, while stress-corrosion cracking has apparently been promoted by the absence of this constituent at certain localities in the boilers at Edgar and during the earlier operation at Colfax.

Formation of an iron phosphate, which takes place only to an insignificant extent in boilers operated with definite though low concentrations of sodium hydroxide, apparently is promoted by elimination of this caustic alkalinity. Both at Port Washington (Fig. 18) and at Logan (Fig. 26) considerable amounts of a white iron phosphate were found admixed with black Fe_3O_4 on the tube surfaces and exposed along the steam-blanketed strips on the tube ceilings. The authors believe this to be merely a casual result, with no particular significance as far as prevention of attack is concerned.

Effect of Tube Slope

The influence of tube slope is shown most clearly by the fact that the drop-leg tubes at South Amboy failed frequently, and always along the top of the nearly horizontal section, while the straight inclined tubes with which they alternated were practically free from trouble except as they were cut externally by leaks from the drop-leg tubes. At Logan the increase in slope of the ash-screen tubes was accompanied by a decrease in diameter; to what extent the resulting improvement may have been due to possible increase in velocity of flow can scarcely be guessed, but the writers believe that this effect was less important than the increased slope.

Effect of Circulation

So little is known about the rate at which water actually circulates through individual tubes that little can be done to evaluate the influence of this factor. The attempt at Logan to double the velocity in the tubes at the corners of the ash screen failed to produce any marked improvement, and the modifications with respect to flow at San Antonio apparently had no great effect.

In straight-tube sectional-header boilers in general, the circulation appears to be downward through the upper row or rows of tubes (7, 8). In most cases, any damage from the resultant steam blanketing is confined to the tops of the tubes near the uptake header, as at San Antonio, and at Springdale before insertion of cores. The location of the failures on the bottom of the tubes at Waterside is difficult to explain, unless it is assumed that at times little water was passing through them. The attack on the bottom of the cored tubes at Springdale suggests the same condition.

With respect to ash-screen tubes, even if the rate of flow were greatly increased, as by forced circulation, steam blanketing might still occur. In this connection, the authors have had the privilege of examining a specimen of studded tube from a forced-circulation once-through boiler which showed pits on the internal surface precisely under the studs on the portion of the wall toward the combustion chamber. Apparently the studs had served as channels for localized high heat input to the tube wall.

Effect of Rate of Heat Input

Heat absorption sufficient to develop steam blanketing in a specific tube may not actually cause any serious damage to the steel. Progressive increase in heat transmission must, however, increase tube temperature and localized concentration of boiler

[11] Hiding out is also indicated in the control reports of a number of other boilers of contemporary design.

water, thus accelerating attack. Unfortunately, there is little basis for determining the maximum rate at which heat may be safely absorbed by a steam-blanketed tube.

The experimentation at San Antonio revealed that as long as the temperature of the tube wall did not exceed 800 F, oxidation was slow. The practical solution of the problem at this plant, it will be recalled, was baffling of the gas flow to shift the zone of heat absorption away from the upper ends of the tubes.

Swinging the burners out slightly from the walls at Logan moved the region of most severe attack from the sides toward the center of the ash screen. Adjustments of this sort may have played a critical part in determining whether other plants would or would not encounter trouble as a result of steam blanketing. For example, the shift of the burners upward on boiler 5 at Rouge in order to achieve the desired superheat may have helped to obviate the failure of ash-screen tubes.

The three 1300-psi boilers at Lakeside Station of Wisconsin Electric Power Company represent the pioneer application of ash screens in the United States. What the actual heat transmission to the tubes in these screens may be is not known, but the fact that no failures have occurred in spite of probable steam blanketing suggests a lower input to these tubes than to those in more recent installations of similar type. It is interesting to note from the discussion by Anderson (18) that some development of black oxide took place at Lakeside at particular points in both the boiler and fin tubes where the heat transfer was the highest.

Stopping the failure of floor tubes at Rivesville by increasing the thickness of the floor insulation is perhaps the most direct example of the effect of rate of heat input on the damage to steam-blanketed tubes.

Intergranular Cracking

While in the tubes at Beacon Street and at Piedmont which were presumably subjected to repeated quenching, the cracks were characteristic of stress corrosion (Figs. 33 and 36), the intergranular cracks observed first at Waterside (Fig. 13) and later at Logan (Fig. 28) suggest the possibility of embrittlement by sodium hydroxide or by hydrogen. However, intergranular cracks produced in low-carbon steel during long-time creep tests at 1000 F in air by White, Clark, and Wilson (19) and cracks developed in Armco iron at 1100 F in air as well as in hydrogen by Thielemann and Parker (20) are similar to those in Fig. 13. The latter authors attributed the intergranular character of the cracks in their experiments, which also included low-carbon and alloy steels, primarily to long-continued load above the equicohesive temperature, and expressed the belief that intergranular cracking preceded intergranular oxidation rather than resulting from it. The similarity of prolonged overheating of a tube in a boiler to a creep test under low stress is obvious. Whether or not intergranular corrosion by the boiler water, analogous to that which sometimes develops in riveted seams, was a factor in the formation of the cracks at Waterside and at Logan, excessively high temperature seems to have been the fundamental cause.

Significance of Metallic Copper

Metallic copper present in deposits of magnetic iron oxide has sometimes been regarded as the principal cause of the process which produced the oxide (21). The authors believe, however, that the copper is merely an incidental product of the fundamental reaction of steel with boiler water, and if supplied to the boiler by attack on condensers, valves, pumps, or other auxiliary equipment, will be found chiefly wherever stress, steam blanketing, or other influence accelerates reaction between the water and the steel.

Black Oxide and Hydrogen as Warning Signals

Steam blanketing leaves its distinctive mark in a tube before the conditions produced by it become dangerous. This will usually appear as a dark band along the top of a tube, sometimes dull and sooty, sometimes scintillating with the light reflected from the faces of small but well-formed crystals of Fe_3O_4. The presence of such a band or area does not necessarily mean immediate danger, but it should be considered as a warning signal. When loose, powdery Fe_3O_4, which may be admixed with metallic copper, is found in any appreciable amount either adhering to tube surfaces or lying in tubes or headers, the condition should be regarded as serious.

Hydrogen in steam has been used as a measure of the attack on tubes under superheating conditions by Fellows (22) and more recently by Potter, Solberg, and Hawkins (23). As a measure of the over-all attack in the boiler it proved useful at Port Washington but yielded rather unreliable results at Logan. While the technique of accurate determination of the hydrogen content of a representative sample of steam is not simple, this type of measurement applied to saturated steam should tell whether or not any serious oxidation of steel by boiler water is taking place.

Definition of the Problem

Difficulties caused by steam blanketing in various types of boilers of four different manufacturers in nine plants have been described in this paper. The intention has not been to raise a great alarm, but to bring what seems to be a general problem clearly into focus, so that it may be discussed and solved to the advantage of both the manufacturer and the purchaser of contemporary high-capacity boilers.

How can the difficulties resulting from steam-blanketing of slightly inclined tubes in high-pressure boilers best be prevented? One answer would be to eliminate the steam blanketing by eliminating the slightly inclined tubes; another, to minimize steam blanketing by controlling heat input or improving circulation; and still a third, to modify the composition of the boiler water so that the rate of attack, even on overheated steel, would be insignificant.

A boiler without slightly inclined tubes is a problem for the designer. Where a boiler is already in service, extensive changes may be justifiable neither from the mechanical nor from the economic viewpoint. It may be possible, however, to redistribute heat absorption or improve circulation sufficiently to minimize the damage from steam blanketing, even though the latter may still persist.

To reduce the heat input over any appreciable fraction of the heat-absorbing surface of a boiler apparently runs counter to the current trend toward smaller furnace volume and higher heat release. The question may well be asked, however, whether a small increment in the initial cost of a boiler guaranteed for a specific output of steam might not save both the manufacturer and the purchaser several times this amount in money and in the money equivalent of the time during which the boiler is unavailable for service due to necessary modifications.

Over against the factor of design, which fundamentally determines whether steam blanketing will or will not occur in a boiler, and if so, to what extent the blanketed areas will be heated above normal temperature levels, must be set the factor of boiler-water conditions. By eliminating sodium hydroxide from a boiler water and operating only with the alkalinity contributed by phosphate, the wasting away of metal in steam-blanketed tubes may be greatly reduced, although attack will still continue at a slow rate. Control of the pH of the boiler water can scarcely, however, be expected to reduce the temperature of the overheated tube wall which, if quenched repeatedly by water striking it, may ultimately crack. Operation with zero caustic alka-

linity seems less a solution to the problem of steam blanketing than an expedient which, in competent hands, will allow operation until the necessary fundamental changes can be made.

The writers believe that the real solution of the difficulties described in this paper lies in the prevention of steam blanketing in tubes subjected to high heat input, or conversely, limitation of the heat input to tubes which are likely to develop a steam blanket. Once this is accomplished, the question of what alkalinity should be maintained in the boiler water is no longer conditioned by what must be done to save overheated steel from attack by excessive local concentration of the boiler water, but instead by what will give maximum protection to the boiler as a whole.

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Discussion

J. B. Romer.[12] The authors have presented considerable data that they have gathered on the question of high-temperature corrosion of steel by steam and the effect of boiler-water conditioning, particularly along the lines reported by M. K. Drewry (12) on this general problem.

Their paper may be summed up to the effect, that overheated tubes, irrespective of angle, in contact with steam corroded severely and that with the possible exception of a special treatment involving excess ferrous iron, there is no known feedwater treatment which adequately takes care of this situation.

The writer is in agreement with this summation and under these conditions the answer must lie in adequately taking care of this situation through the medium of design.

[12] The Babcock & Wilcox Co., Barberton, Ohio.

The microstructures as presented by the authors indicate that the nonoxidized metal, at the point of corrosion, had been at temperatures well above 1000 F and quoting from the authors:

> Correlation of the temperature measurements with the observed condition of the tubes showed that the deposit of black oxide had accumulated only in those tubes which had exceeded 800 F during the operating period. Measurement of the internal diameter revealed that there had been little actual loss of metal from the tubes.

These two items place the oxidation limit, of steel in steam, at or above 1000 F.

Straub in his paper[13] indicates that in laboratory tests the limit may be as low as 750 F—this when certain chemistry is adhered to. That control of the chemistry controls the extent of the oxidation is not only extremely interesting but it is also valuable information.

We should not lose sight of the effect of sodium sulphate. Straub shows that a decided decrease in the hydrogen generated and in the amount of corrosion observed follows as the result of an increased content of sodium sulphate.

This is quite significant and it is also unexpected that this particular problem should show up the advantage of the old caustic-soda antidote, sodium sulphate.

While the writer agrees that the answer to the formation of black magnetic oxide of iron in certain tubes at temperatures above about 950 to 1000 F is not one of proper water treatment, he wishes to point out that the feedwater chemist cannot dodge his two major problems:

1 To produce scale-free water at maximum operating temperatures by such methods that will

2 Produce water which will not damage the material of the boiler.

He should not raise the bugaboo that "lack of circulation is the cause of certain types of scale."

While there may be evidence indicating or pointing in the direction that certain types of scale form under these conditions of "dryness" there are other equally if not more definite indications that this dryness is the result of the formation of a type of sludge the porosity of which is insufficient to permit water to reach and cool the tube wall adequately, and further that this sludge formation may and does occur in the presence of ample circulation.

The second job, namely, to produce water that will not damage the material of the boiler generally involves controlling the pH. Experiences over a long period of time indicate that water having a low pH is excessively reactive toward iron and steel and unless this reactivity can be satisfied without using up or depleting the excess soluble ferrous ions then the boiler metal may suffer from some form of corrosion or corrosion fatigue.

In connection with the hiding out of soluble solids the writer would like to inquire as to a condition frequently encountered in modern boiler-water-conditioning practice. In many instances it is well known that the feedwater contains both calcium and magnesium apparently as carbonates or bicarbonates. Within the boiler these are treated with phosphate for the purpose of changing the calcium salts from a form of relatively low solubility to a more stable material supposedly of the sludge type. The blowdown from such units, even when the concentration of soluble salts is normal, is often crystal clear. Under these circumstances, what becomes of the sludge which is being formed in the boiler?

Another question which is quite pertinent to this problem is: Can dry areas be produced by water conditioning? In a number of instances sludge has accumulated in areas as shown in Fig. 1 and resulted in starvation of certain sections, thereby producing dry areas.

Among the case histories quoted by the authors are four with which the writer is familiar, namely, South Amboy, Rivesville, Edgar, and Colfax. At the present time it is considered that the first two of these have been satisfactorily taken care of by suitable changes in design. The third has been taken care of by changes in design which simultaneously involve changes in pH and the fourth case has been taken care of by changes in the pH alone.

The authors raise the question as to the possible secondary dangers as the result of maintaining low pH and point out the experiences at both Colfax and Edgar in this connection. Although the latter case may be of the stress-corrosion type, it is well known that low pH lowers the stress which may be applied to a pressure vessel to a marked degree, that many failures of boiler metal parts in the past have been directly due to low pH of the boiler waters, and that proper correcting of this value has eliminated the trouble.

Authors' Closure

That the reader does not always get out of a paper what the writer tried to put into it is illustrated by the remarks of Mr. Romer. His discussion implies that tube slope is not a factor in steam blanketing, and that oxidation of the ceiling of a steam-blanketed tube may possibly be avoided by maintaining excess ferrous iron in the boiler water. Without wishing to seem dogmatic, the authors again call attention to the improvement in conditions at South Amboy and at Logan following an increase in tube slope, and to the fact that the use of ferrous hydroxide at Colfax minimized only ordinary corrosion due to dissolved oxygen, rather than the oxidation due to steam blanketing, which was the chief concern of the paper.

Professor Straub and Mr. Nelson, in their paper[13] presented on the same program as that of the authors, concluded from laboratory studies of steam-blanketing that:

1 "This type of corrosion occurs only in partially dry areas.

2 "If the metal temperature is below 750 F, sodium hydroxide appears to be the active constituent in the boiler water causing the corrosion.

3 "If the metal temperature is below 750 F, the corrosion may be controlled by modifying the boiler water as follows: (*a*) reducing the causticity to zero, and (*b*) adding other salts, such as sulphate, silicate, etc.

4 "If the metal temperature is above 750 F, corrosion takes place in the absence of sodium hydroxide and chemical treatments effective at the lower temperature are no longer protective."

Unless one is willing to grant or to guarantee that the ceiling of a steam-blanketed tube will fail to exceed 750 F or some temperature not far above this, there is little basis either in the laboratory tests of Straub and Nelson or the operating experiences cited in the present paper for assuming that ferrous hydroxide or sodium sulphate or anything else added to the boiler water would prevent damage to the steel.

As the writers noted in the second sentence of the paper, neither the insulating effect of scale nor the acceleration of ordinary corrosion by dissolved oxygen are factors in the type of damage which is the chief subject of the paper. In most of the cases described, the amount of sludge found on the boiler surfaces was literally microscopic, and in no case was it sufficient to be a factor in the major problem. The authors hope, therefore, that, without evading any responsibility, they may postpone to some more appropriate forum a response to Mr. Romer's comments and questions relative to scale and sludge.

[13] "Corrosion in Partially Dry Steam Generating Tubes," by F. G. Straub and E. E. Nelson, *Mechanical Engineering*, vol. 61, 1939, pp. 199–202.

Deflection of Helical Springs Under Transverse Loadings

By W. E. BURDICK,[1] F. S. CHAPLIN,[2] AND W. L. SHEPPARD[3]

This paper presents an analytical approach to the problem of the transverse deflection of a helical closely coiled spring under the action of a force perpendicular to its axis and an end moment. Supporting test data are included to substantiate the formulas within established limits. Expressions for the stresses under this loading in combination with an axial force are derived for the critical points on the spring.

THE USE of helical closely coiled spring suspensions for vibrating masses has not only necessitated a careful consideration of the characteristics of the spring under normal axial loading but in addition, where a thrust acts in a direction perpendicular to the axis of the helix, equal regard should be given to the properties of the coil under transverse loadings. This condition is evident in the case of those reciprocating mechanisms where an unbalanced force lies in the horizontal plane; also, it exists in certain types of vehicular suspensions in which a helical spring is utilized to take lateral or fore-and-aft reactions in addition to gravity loadings. The latter application is best exemplified in certain recently constructed types of railway-car trucks incorporating this particular capacity of a helical spring to cushion loads in more than one plane.

It is the authors' intention to present herein an analytical approach to the problem of a helical spring with one end resting on a rigid foundation and the other subjected to a transverse loading alone or in combination with an end moment. The investigation includes a derivation of the relationship for the resulting transverse deflection at the free end and attendant stresses under such a condition. Comparison will be made with actual test results made with a wide variety of spring sizes and loadings to demonstrate the validity of the expression for deflection and establish the approximate range of application. The stress derivations are dependent on the ability of the stock itself to follow Hooke's law of proportionality for a material in bending and torsion, with appropriate factors for stress concentration due to curvature and may be accepted with no more reservations than are accorded the present approved method of investigation for the axial loading condition.

The only specific information available on this subject which the authors were able to find was a paper by Marty.[4] In this case, the investigation was mainly concerned with a uniformly distributed transverse load over the length of the spring as occurs under centrifugal loading. This author expressed an effective flexural and torsional rigidity of a helical spring analogous to a beam as a function of certain of its dimensions, and stated the relationship for stress and deflection but without derivations or supporting test data. Timoshenko[5] has evaluated the equivalent column characteristics of a helical spring as an axially loaded member for purposes of determining its stability.

Nomenclature

The nomenclature used in this paper is as follows:

- h = height of helix
- ϕ = slope of end coil
- D = diameter of helix
- R = radius of helix
- d = bar diameter
- P = axial loading
- L = transverse loading
- M_0 = end moment on coil
- n = effective number of coils = $\left(\frac{\text{solid spring height}}{d} - 1\right)$
- f = pitch of helix = (h/n)
- θ = angular coordinate of any point on the helix
- M_t = torsional moment on any cross section of the bar
- J = polar moment of inertia of this section
- τ_t = shear stress on this section due to M_t
- M_x = bending moment on this section about the axis perpendicular to the axis of the helix
- I_x = moment of inertia about this axis
- σ_x = bending stress on this section due to M_x
- M_y = bending moment on this section about the axis parallel to the axis of the helix
- I_y = moment of inertia about this axis
- σ_y = bending stress on this section due to M_y
- S_x = shear load on this section perpendicular to axis of helix
- S_y = shear load on this section parallel to axis of helix
- Q = axial load on this section
- τ = shear stress
- E = modulus of elasticity in tension and compression
- G = modulus of elasticity in torsion
- Δ = vertical deflection of coil
- δ = transverse deflection of coil
- V = total strain energy
- C_v = vertical spring constant
- C_l = transverse spring constant

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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.

[4] "On the Calculation of Helical Springs Loaded Transversely" (French), by R. Marty, Comptes Rendus, vol. 192, July 11, 1932, p. 105.

[5] "Theory of Elastic Stability," by S. Timoshenko, McGraw-Hill Book Company, Inc., New York, N. Y., 1936, pp. 165–167.

Deflection

Any analysis covering the exact deflection of a helical spring necessarily involves consideration of that portion of the deflection introduced by the end coils. Such an analysis introduces complex quantities, and for simplification only that portion of the helix will be considered which involves the full bar diameter.

Consider a helical spring with initially parallel ends supported on a rigid foundation, as shown in Fig. 1*a*. Assume the helix to extend along the center line of the bar from the first full diameter of the bar on one end to the corresponding point on the other end. This gives a helix height equal to the full distance

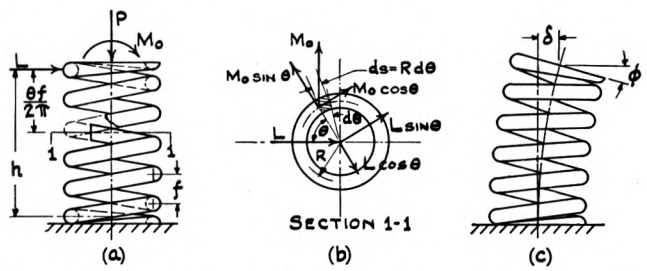


Fig. 1 Helical Spring With Initially Parallel End Supported on a Rigid Foundation

between coil ends less one bar diameter. Therefore, the number of turns in the helix n will be one less than the solid spring height divided by the bar diameter. Assume the spring to be initially loaded axially with P and consider the effect of superimposing a lateral load L and a moment M_0. Let L intersect the helix axis and lie in a horizontal plane containing the highest point on the helix. Let the transverse end couple M_0 have its vector perpendicular to the plane containing the helix axis and L. Since L is independent of P, only the deflections due to L and M_0 will be considered. The transverse deflection resulting from the application of P will be considered in the Appendix. The helix is developed by the continuous function $R\theta$ and the pitch f. The origin of θ will be taken at its coincidence with L.

The general equation for the strain energy[6] is expressed as

$$V = \frac{1}{2}\int_0^s \left(\frac{M^2_x}{EI_x} + \frac{M_y^2}{EI_y} + \frac{M_t^2}{GJ} + \frac{KS_y^2}{AG} + \frac{KS_x^2}{AG} + \frac{Q^2}{AE}\right) ds \quad \ldots\ldots [1]$$

This form of strain-energy equation neglects the effect due to curvature, but is sufficiently accurate for a first approximation.

In order to maintain the linear relationships required by the Castigliano theorem the influence of the couple $P\delta$ due to deflection is also disregarded. Since the strain energy of direct shear and compression is relatively small, only that portion of the energy equation dealing with bending and torsion will be considered. The slope and deflection may be expressed by

$$\delta = \frac{\partial V}{\partial L} \quad \text{and} \quad \phi = \frac{\partial V}{\partial M_0}$$

Hence, from Equation [1]

$$\delta = \int_0^s \left(\frac{M_x}{EI_x}\frac{\partial M_x}{\partial L} + \frac{M_y}{EI_y}\frac{\partial M_y}{\partial L} + \frac{M_t}{GJ}\frac{\partial M_t}{\partial L}\right) ds \quad \ldots\ldots [2]$$

$$\phi = \int_0^s \left(\frac{M_x}{EI_x}\frac{\partial M_x}{\partial M_0} + \frac{M_y}{EI_y}\frac{\partial M_y}{\partial M_0} + \frac{M_t}{GJ}\frac{\partial M_t}{\partial M_0}\right) ds \quad \ldots\ldots [3]$$

[6] "Strength of Materials," by S. Timoshenko, D. Van Nostrand Co., Inc., New York, N. Y., 1930, pp. 318–336.

For closely coiled springs, the pitch angle is small, and with sufficient accuracy, the reactions due to L and M_0 on any section with coordinate θ may be designated as follows:

1 A bending moment M_x in the vertical plane

$$M_x = L\frac{\theta f}{2\pi}\sin\theta + M_0 \sin\theta \quad \ldots\ldots [4a]$$

2 A bending moment M_y in the horizontal plane

$$M_y = LR\sin\theta \quad \ldots\ldots [4b]$$

3 A torsional moment M_t

$$M_t = L\frac{\theta f}{2\pi}\cos\theta + M_0\cos\theta \quad \ldots\ldots [4c]$$

In addition, there are the following reactions which have been disregarded as influencing deflections:

A direct horizontal shear

$$S_x = L\cos\theta \quad \ldots\ldots [4d]$$

A direct compressive load

$$Q = L\sin\theta \quad \ldots\ldots [4e]$$

The partial derivatives of Equations [4*a*], [4*b*], and [4*c*] with respect to L and M_0 are

$$\frac{\partial M_x}{\partial L} = \frac{\theta f}{2\pi}\sin\theta; \quad \frac{\partial M_y}{\partial L} = R\sin\theta; \quad \frac{\partial M_t}{\partial L} = \frac{\theta f}{2\pi}\cos\theta;$$

$$\frac{\partial M_x}{\partial M_0} = \sin\theta; \quad \frac{\partial M_y}{\partial M_0} = 0; \quad \frac{\partial M_t}{\partial M_0} = \cos\theta$$

Since $ds = Rd\theta$, Equations [2] and [3], with n = number of turns and $s = 2\pi nR$, become

$$\delta = \int_0^{2\pi n}\left(\frac{Lf^2}{4\pi^2 EI_x}\theta^2\sin^2\theta + \frac{M_0 f}{2\pi EI_x}\theta\sin^2\theta + \frac{LR^2}{EI_y}\sin^2\theta + \frac{Lf^2}{4\pi^2 GJ}\theta^2\cos^2\theta + \frac{M_0 f}{2\pi GJ}\theta\cos^2\theta\right) Rd\theta \quad \ldots\ldots [4]$$

$$\phi = \int_0^{2\pi n}\left(\frac{Lf}{2\pi EI_x}\theta\sin^2\theta + \frac{M_0}{EI_x}\sin^2\theta + \frac{Lf}{2\pi GJ}\theta\cos^2\theta + \frac{M_0}{GJ}\cos^2\theta\right) Rd\theta \quad \ldots\ldots [5]$$

The integration of Equations [4] and [5] with simplifying approximations, is shown in the appendix.

Hence with $f = (h/n)$

$$\delta = Rh\pi n\left(\frac{Lh}{3} + \frac{M_0}{2}\right)\left(\frac{1}{EI_x} + \frac{1}{GJ}\right) + \frac{LR^3\pi n}{EI_y} \quad \ldots\ldots [6]$$

$$\phi = R\pi n\left(\frac{Lh}{2} + M_0\right)\left(\frac{1}{EI_x} + \frac{1}{GJ}\right) \quad \ldots\ldots [7]$$

Expressing M_0 in terms of ϕ and L from Equation [7], and substituting in Equation [6], the deflection is

$$\delta = \frac{LRh^2\pi n}{12}\left(\frac{1}{GJ} + \frac{1}{EI_x}\right) + \frac{LR^3\pi n}{EI_y} + \frac{\phi h}{2} \quad \ldots\ldots [8]$$

In the case where the ends of the spring remain parallel after deflection ($\phi = 0$), and noting that from symmetry $M_0 = -Lh/2$, both Equations [6] and [8] become

$$\delta = \frac{LRh^2\pi n}{12}\left(\frac{1}{GJ} + \frac{1}{EI_x}\right) + \frac{LR^3\pi n}{EI_y} \quad \ldots\ldots [9]$$

For a round bar steel spring with $E = 29{,}500{,}000$ lb per sq in., and $G = 11{,}500{,}000$ lb per sq in., the general Equations [6] and [8] for the lateral deflection become

$$\delta = \frac{Dn}{10^6 d^4}(0.825h^2L + 1.238M_0h + 0.271D^2L) \quad [10]$$

and

$$\delta = \frac{LDn}{10^6 d^4}(0.271D^2 + 0.206h^2) + \frac{\phi h}{2} \quad [11]$$

and for a spring whose ends remain parallel after deflection

$$\delta = \frac{LDn}{10^6 d^4}(0.271D^2 + 0.206h^2) \quad [12]$$

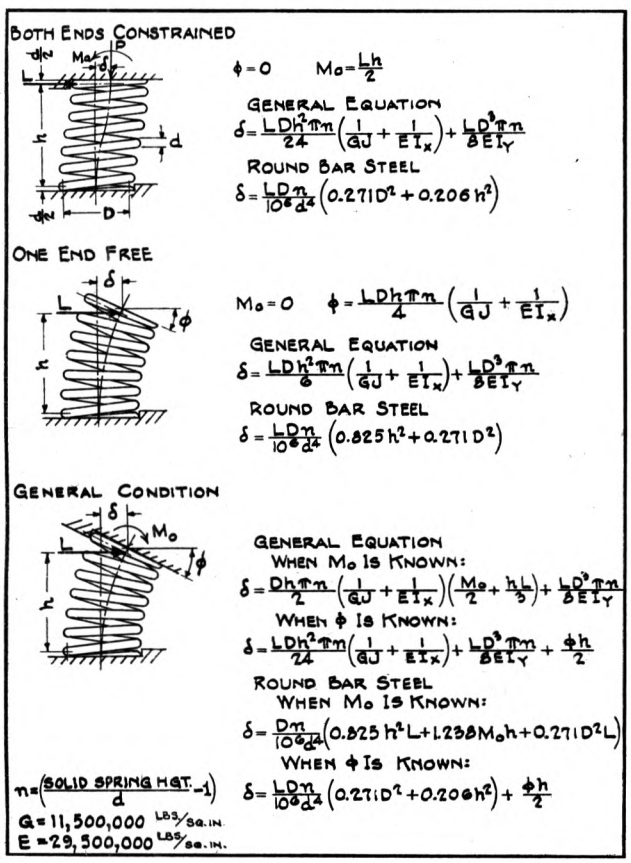


FIG. 2 TRANSVERSE-DEFLECTION FORMULAS FOR CLOSELY COILED HELICAL SPRINGS

The transverse-deflection formulas are shown with illustrative diagrams in Fig. 2.

STRESSES

The maximum stresses imposed by the loading described previously will now be discussed. As in the case of the deflections, the small influence of the helix angle has been neglected. The maximum shear stress under axial loading, including Wahl's[7] concentration factor, is given by the formula

$$\tau = \frac{16PR}{\pi d^3}\left(\frac{4c-1}{4c-4} + \frac{0.615}{c}\right) \quad [13]$$

where $c = D/d$.

[7] "Stresses in Heavy Closely Coiled Helical Springs," by A. M. Wahl, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-17, pp. 185–200.

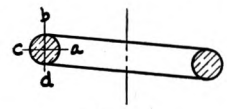


FIG. 3

This stress occurs on the inside of the coil as shown at a in Fig. 3, and is constant for all values of θ. The portion $\frac{16PR}{\pi d^3}\left(\frac{0.615}{c}\right)$ is due to direct shear P and will not be affected by the transverse loading. The remainder, $\frac{16PR}{\pi d^3}\left(\frac{4c-1}{4c-4}\right)$ is due to the torque PR and will be increased by any augment in the torque due to L and M_0. The expression for torque, that is Equation [4c], with the addition of PR is

$$M_t = \left(\frac{L\theta f}{2\pi} + M_0\right)\cos\theta + PR \quad [14]$$

It is obviously a maximum when the expression $[(L\theta f/2\pi) + M_0]$ is a maximum and $\cos\theta = \pm 1$. Also $[(L\theta f/2\pi) + M_0]$ will be maximum when θ is either 0 or a maximum, depending on the relative values and signs of L and M_0.

Thus, the maximum shear stress will occur on the inside of the coil in the plane containing L and P, shown in Fig. 1. Its magnitude will be

$$\tau = \left[\left(\frac{L\theta f}{2\pi} + M_0\right)_{\max} + PR\right]\frac{16}{\pi d^3}\left(\frac{4c-1}{4c-4}\right) + \frac{16PR}{\pi d^3}\left(\frac{0.615}{c}\right) \quad [15]$$

Disregarding the direct horizontal shear and compressive load due to L given by Equations [4d] and [4e], there remain the bending stresses due to M_x and M_y. An examination of the expressions for these moments shows every term to contain $\sin\theta$ and they are accordingly zero for the points already investigated. The moments M_x and M_y attain their maximum values when $\sin\theta = \pm 1$, i.e., at that section of the coil intersected by a plane which includes the axis of the helix and lies at right angles to (the direction of) L.

Thus, $M_x = [(L\theta f/2\pi) + M_0]\sin\theta$ and reaches a maximum value a quarter turn from the maximum for M_t. The moment $M_y = LR\sin\theta$ and is likewise a maximum twice every turn. Where M_x and M_y are maximum, M_t has the value of PR.

The maximum stress due to M_x occurs at the extreme fibers of the bar section at points b and d shown in Fig. 3. At these points the torsional and bending stresses are not increased by any stress concentration.

Therefore, at b the stresses will be

$$\tau_t = \frac{16PR}{\pi d^3} \quad [16]$$

$$\sigma_x = \frac{32[(L\theta f/2\pi) + M_0]}{\pi d^3} \quad [17]$$

The maximum principal stress

$$\sigma_{\max} = \frac{16}{\pi d^3}\left[\left(\frac{L\theta f}{2\pi} + M_0\right) + \sqrt{(PR)^2 + \left(\frac{L\theta f}{2\pi} + M_0\right)^2}\right] \quad [18]$$

and the maximum shear stress

$$\tau_{\max} = \frac{16}{\pi d^3}\sqrt{(PR)^2 + \left(\frac{L\theta f}{2\pi} + M_0\right)^2} \quad [19]$$

The maximum stress due to M_y occurs at the inside of the bar at a shown in Fig. 3, where it combines with a torsional shear.

At a, the stresses will be

$$\tau = \frac{16PRk}{\pi d^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

where $$k = \left(\frac{4c-1}{4c-4} + \frac{0.615}{c}\right), \text{ and}$$

$$\sigma_y = \frac{32LRq}{\pi d^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots [21]$$

From the curved-beam theory, $q = (4c-1)/(4c-4)$. The maximum principal stress

$$\sigma_{max} = \frac{16}{\pi d^3}\left[LRq + \sqrt{(PRk)^2 + (LRq)^2}\right] \quad \ldots\ldots [22]$$

and the maximum shear stress

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(PRk)^2 + (LRq)^2} \quad \ldots\ldots\ldots [23]$$

In general it will be found for springs of normal proportions, that the maximum shear stress will be more critical than the maximum direct stress. Therefore, the critical stress will occur on the inside of the coil where M_t is a maximum at which point it will have the value

$$\tau = \left[\left(\frac{L\theta f}{2\pi} + M_0\right)_{max} + PR\right]\frac{16}{\pi d^3}\left(\frac{4c-1}{4c-4}\right) + \frac{16PR}{\pi d^3}\left(\frac{0.615}{c}\right) \quad \ldots\ldots [15]$$

It will be noted that the maximum moment occurs on the end coil. Obviously, if this coil were tapered, the maximum moment would occur on a reduced section and some provision may be necessary for its protection.

Discussion of Other Contributing Factors

As previously mentioned, Timoshenko[6] has discussed the stability of helical springs under axial loading. It is obvious that the addition of a transverse load will decrease the axial critical load. A complete investigation and discussion of stability was considered to be beyond the scope of this paper.

TABLE 1 CHARACTERISTICS OF SPRINGS TESTED

Spring no.	Mean diameter D in.	Bar diameter (d) in.	Free height, in.	Solid height, in.	Axial capacity, lb	$\frac{D}{d}$
1	5.812	1.25	9.219	6.594	10200	4.65
2	3.672	0.734	9.406	6.156	3700	5.00
3	6.726	1.241	15.250	8.72	12200	5.43
4	3.875	0.750	14.656	8.81	4050	5.16
5	0.566	0.123	0.798	0.553	139	4.60
6	0.569	0.123	1.211	0.841	121	4.63
7	0.572	0.123	1.664	1.108	121	4.65
8	0.752	0.123	1.313	0.715	98	6.12
9	0.752	0.123	2.078	1.102	98	6.12
10	0.752	0.123	2.801	1.455	96	6.12

The precise equation, including the effect of the helix angle, expressing a relationship for transverse deflection is shown in the Appendix. Numerical calculations on two springs, whose helix angle was approximately 5 deg, indicated an error of less than 0.5 per cent.

Description of Test Procedure

In order to demonstrate the validity of the foregoing formulas, tests were conducted on several sets of helical springs to determine the transverse-load-deflection rates. While the original tests were conducted on large springs of the railway type, the procedure was cumbersome due to the magnitude of the vertical loads required. Accordingly, to facilitate experimental work, subsequent tests were conducted on small springs, accurately coiled and with the ends ground. These springs were furnished by the Railway Steel Spring Company.

The pertinent characteristics of all springs tested for lateral deflection properties in conjunction with this paper are shown in Table 1. It will be noted that the experimental springs covered a wide range of ratios of height to diameter, and the spring indexes varied from 4.5 to 6. The experimental apparatus was essentially the same for both the large and small coils, employing in each case four identical helical springs with ground ends seated at the four corners of a rectangular base plate and surmounted by a similar plate on which the vertical load was placed. The assembled coils were then loaded transversely by a turnbuckle and tension rod in series with a load-measuring device lying in a horizontal plane and affixed, one end to a rigid foundation and the other to some point on the top plate according to whether pure translation or combined translation and rotation was to be applied to the top plate.



Fig. 4 Apparatus Used for Testing High-Capacity Springs

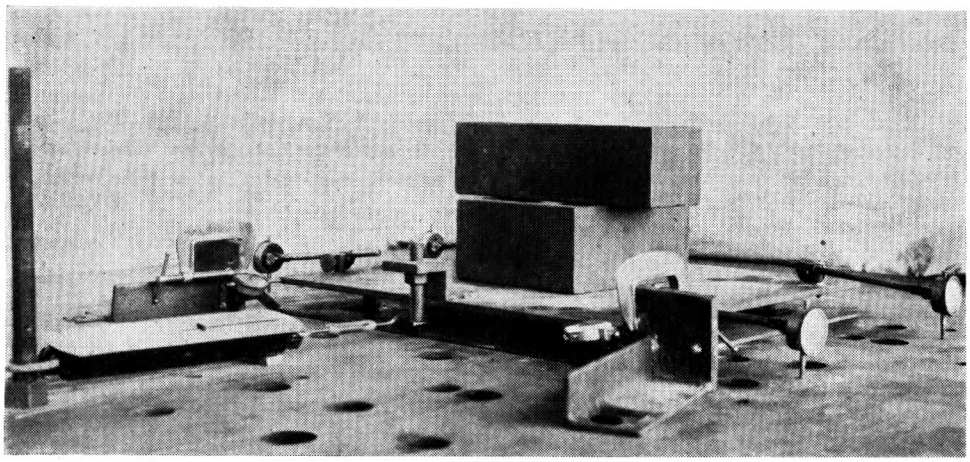


FIG. 5 APPARATUS FOR TESTING SMALL SPRINGS

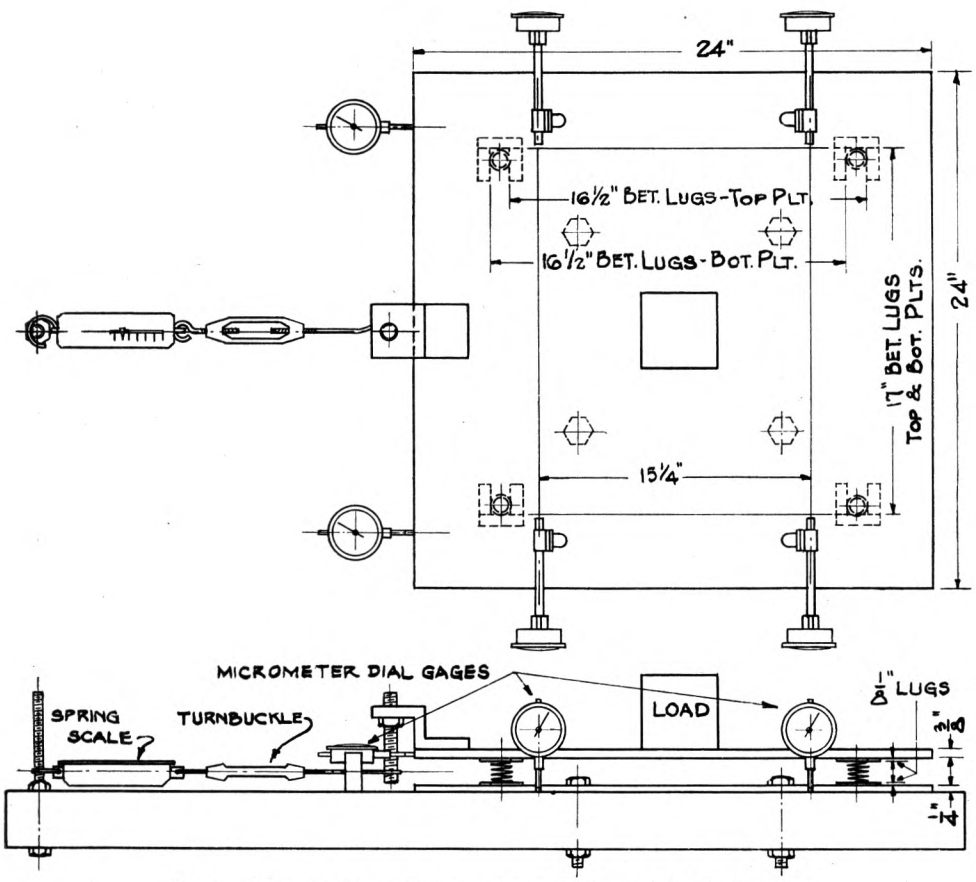


FIG. 6 ARRANGEMENT OF APPARATUS FOR TESTING SMALL SPRINGS

In the case of the large testing apparatus, which is shown in Fig. 4 the springs were retained at both extremities by set bolts tapped in an annular ring welded to each plate around the spring seat and abutting the end coil at the quarter-circle points. A slotted lug was provided at the midpoint of the underside of the top plate, which afforded an opportunity for varying the height of transverse-load application.

As shown in Figs. 5 and 6, the small springs were retained in their position by cleats of the same thickness as the bar diameter of the spring; the cleats were welded to the plates on such a side of the coil as to prevent horizontal movement of the ends under transverse loading. An auxiliary cleat was welded at right angles to this, thus forming an angle which thereby nested the end coil on two points of its circumference. An adjustable threaded rod affixed to the front edge of the top plate, in this case, provided a means of varying the height of transverse load application.

On both apparatus the amount of transverse load was determined by mounting a calibrated ring gage or accurate spring scale, according to the magnitude of the loads, in tandem with the turnbuckle. Micrometer dials or, wherever necessary, surface gages, were used to observe all necessary deflections. Lateral deflections were taken on either side of the center line of the

TABLE 2 COMPARISON OF ACTUAL AND CALCULATED TRANSVERSE DEFLECTIONS

	25 per cent axial capacity			50 per cent axial capacity			75 per cent axial capacity		
Spring no.	Working height, in.	Actual δ/L, in. per lb	Calculated δ/L, in. per lb	Working height, in.	Actual δ/L, in. per lb	Calculated δ/L in. per lb	Working height, in.	Actual δ/L, in. per lb	Calculated δ/L, in. per lb
1	7.301	0.00028	0.000214	6.679	0.000213	0.000196	6.039	0.00018	0.000176
2	7.860	0.00175	0.001560	7.047	0.001570	0.001310	6.234	0.00145	0.00111
3	12.384	0.00118	0.000827	10.760	0.001000	0.000704	9.141	0.00088	0.000562
4	13.076	0.01610	0.005650	....	Unstable	.....	...	Unstable	.....
5	0.612	0.00162	0.001425	0.554	0.001340	0.001305	0.512	0.00117	0.001227
6	0.934	0.00495	0.004160	0.884	0.004350	0.003600	0.764	0.00415	0.003020
7	1.378	0.01480	0.009550	1.226	0.014800	0.007960	1.143	0.01540	0.007160
8	1.048	0.00615	0.006000	0.902	0.005250	0.005080	0.771	0.00400	0.004370
9	1.720	0.03150	0.020000	1.480	0.026700	0.015870	1.262	0.02450	0.012630
10	2.347	0.11200	0.045700	....	Unstable	.....	...	Unstable	.....

top plate to insure an average value of this reading, while vertical deflections were taken at the four corners of the assembly to determine the change in spring heights between the front and rear sets of springs under rotational loading.

All springs were tested for transverse deflections under gravity loadings of about 25, 50, and 75 per cent of their axial capacity, except in those few cases where the instability of the coil prevented it. Springs Nos. 1, 2, and 3 were transversely loaded at the level of the top plate. It was then recognized that the slope ϕ contributed an appreciable increment to the deflection. Therefore, springs Nos. 4 to 10, inclusive, were loaded at the height of the point of contraflexure of the springs for the purpose of eliminating the end slope. Sufficient increments of transverse load were applied to establish the true rate of transverse deflection, although for obvious reasons the loading was not pursued to the point of instability of the spring. Upon release of the lateral load, no appreciable lag was observed.

Discussion of Test Results

Examination of the results, as shown in Table 2 indicates good agreement between the actual and the calculated values for a majority of the springs tested. However, a discrepancy exists in the case of certain springs. The approximations occurring in the formula are discussed elsewhere in this paper, together with their effect on the exact evaluation of δ. It will suffice at this point to consider wherein the actual conditions of testing depart from the original assumptions, and to what degree they influence the results obtained.

In order to simulate the general method of a helical-spring installation involving no rigid attachment of the end coils to their base, the testing arrangement made no provision for a constraining moment other than that offered by the loaded plate. In addition, the transverse loading was applied at whatever point of tangency happened to be established between the end coil and the restraining cleat (or set bolt). This procedure, while duplicating practical application, upset certain premises on which the theory was established.

(*a*) The end moment M_0 acted not only upon that portion of the spring contained in the helix proper, but also over the diminishing section of the tapered end coil, thus producing a finite deflection for this portion of the spring which was disregarded in the general equation.

(*b*) Likewise, the transverse loading L, unless the spring possessed exactly n plus $1/2$ coils (where n is a whole number) acted at a point within the tapered region of the end coil, and thereby added its component of transverse deflection in excess of the calculated value.

The magnitude of the error in the calculated value as the axial loading progressed from 25 to 75 per cent capacity was increased or decreased according to which of two effects was prevalent. (1) When the spring was well within the limits of stability under load, the decrease in M_0 as the working height diminished meant a corresponding decrease in its distorting effect on the end coil as just described in paragraph (*a*). (2) If the spring approached an unstable condition as the gravity load increased,

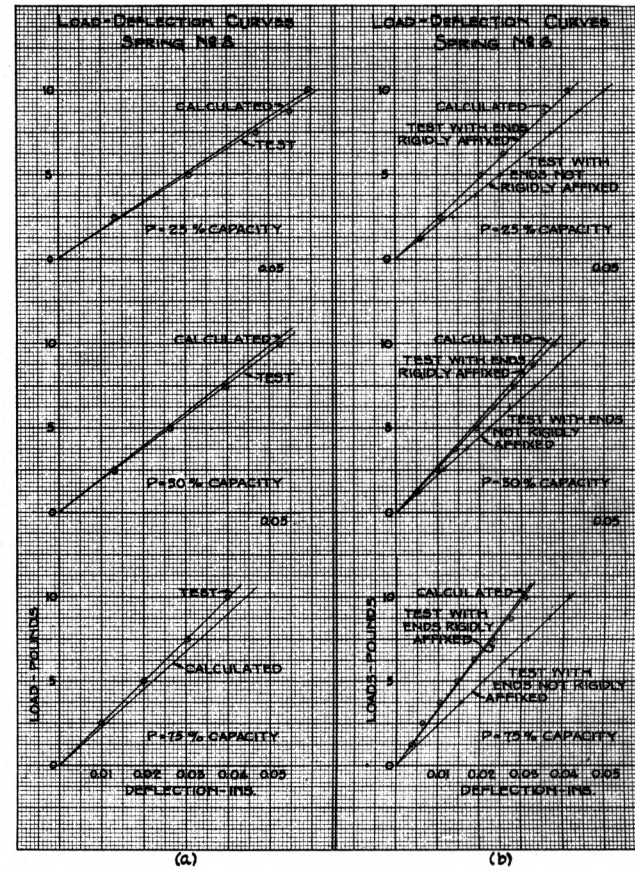


Fig. 7 Transverse-Deflection Characteristics of Two Representative Springs

a given transverse loading produced a greater deflection than would occur for the same transverse loading in combination with a smaller axial-load application.

The actual transverse-deflection characteristics of a representative spring (i.e., spring no. 8) is shown in Fig. 7*a* with the calculated curve for the extent of the test. Similar characteristics for spring No. 6 is shown in Fig. 7*b*. This latter spring was subsequently affixed rigidly to steel base plates at its extreme coil faces and almost perfect agreement with the calculated values was obtained. Examination of the actual plotted points in Figs. 7*a* and 7*b* indicates that there is no measurable deviation from the straight line. This apparently indicates that the omission of the influence of the couple $P\delta$ is justified.

Conclusions

A plot of the ratio of actual to calculated deflection versus the ratio of working height to the mean diameter D, on the basis of the actual tests, is shown in Fig. 8.

As previously noted, the springs were chosen to give a wide

variation of the latter ratio in order to establish the range of application of these formulas. It appears from the results of these tests that the equations may be used with reasonable accuracy when this ratio is less than 1.5. For the majority of engineering applications of compression springs subjected to transverse loadings, this ratio will not be exceeded. With springs in excess of this ratio there is the possible danger of approaching the limit of stability.

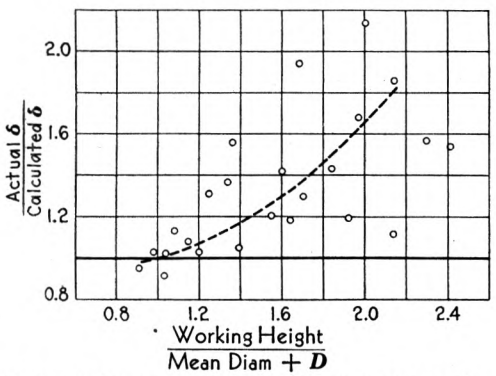


FIG. 8 RATIO OF ACTUAL TRANSVERSE DEFLECTION TO CALCULATED TRANSVERSE DEFLECTION VERSUS THE RATIO OF WORKING HEIGHT TO MEAN DIAMETER OF SPRINGS

The extent to which the instability and end-coil conditions weaken the validity of the expressions, can only be qualitatively appraised at this time.

Since the scope of this paper was to establish the expression for lateral deflection within the stable limits and did not consider the influence of the end coil, it is hoped that this research effort will furnish the impetus for further investigation into the subject.

Appendix

The integrals involved in Equations [4] and [5] are as follows:

$$\int_0^{2\pi n} \theta^2 \sin^2 \theta d\theta = \left[\frac{\theta^3}{6} - \frac{\theta^2}{4}\sin 2\theta - \frac{\theta}{4}\cos 2\theta + \frac{1}{8}\sin 2\theta\right]_0^{2\pi n} \approx \frac{4\pi^3 n^3}{3} \quad [24a]$$

$$\int_0^{2\pi n} \theta \sin^2 \theta d\theta = \left[\frac{\theta^2}{4} - \frac{\theta}{4}\sin 2\theta - \frac{1}{8}\cos 2\theta\right]_0^{2\pi n} \approx \pi^2 n^2 \quad [24b]$$

$$\int_0^{2\pi n} \sin^2 \theta d\theta = \left[\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right]_0^{2\pi n} \approx \pi n \quad [24c]$$

$$\int_0^{2\pi n} \theta^2 \cos^2 \theta d\theta = \left[\frac{\theta^3}{6} + \frac{\theta^2}{4}\sin 2\theta + \frac{\theta}{4}\cos 2\theta - \frac{1}{8}\sin 2\theta\right]_0^{2\pi n} \approx \frac{4\pi^3 n^3}{3} \quad [24d]$$

$$\int_0^{2\pi n} \theta \cos^2 \theta d\theta = \left[\frac{\theta^2}{4} + \frac{\theta}{4}\sin 2\theta + \frac{1}{8}\cos 2\theta\right]_0^{2\pi n} \approx \pi^2 n^2 \quad [24e]$$

$$\int_0^{2\pi n} \cos^2 \theta d\theta = \left[\frac{\theta}{2} + \frac{1}{4}\sin 2\theta\right]_0^{2\pi n} \approx \pi n \quad [24f]$$

In evaluating the foregoing integrals, only the first terms in the brackets have been retained. It will be seen that the balance of the terms diminish rapidly from the first and the evaluation of their maxima was found to affect the calculated deflection less than 1 per cent with n between 3 and 4. With larger values of n the error decreases.

The foregoing analysis deals only with the deflection and slope, resulting from the application of a lateral force and end couple, to a spring initially loaded with an axial force. It is to be noted that the axial force itself may produce a lateral deflection and slope, and by the reciprocal theorem the lateral force and moment may produce a vertical deflection. The following investigation will disclose this effect.

Referring to Fig. 1, let P, only, act on the spring, and apply a dummy load $L = 0$. The only coupling between the two, neglecting the helix angle, lies in the moment M_t, or

$$M_t = PR + \frac{L\theta f}{2\pi}\cos\theta \quad \text{and} \quad \frac{dM_t}{dL} = \frac{\theta f}{2\pi}\cos\theta$$

Hence

$$\delta_P = \int_0^{2\pi n} \frac{PR^2 f}{2\pi GJ}\theta\cos\theta d\theta$$

$$\delta_P = \frac{PR^2 h}{2\pi GJn}\left[2\pi n \sin 2\pi n + \cos 2\pi n - 1\right] \quad [25]$$

It will be seen that when n is a whole number the value of the bracket (and hence the deflection) is zero. At other values of n the deflection is expressed as follows:

n = whole number plus:	δ_p equals:
1/4 turn	$\frac{PR^2h}{GJ}\left(1 - \frac{1}{2\pi n}\right)$
1/2 turn	$-\frac{PR^2h}{GJ}\left(\frac{1}{\pi n}\right)$
3/4 turn	$-\frac{PR^2h}{GJ}\left(1 + \frac{1}{2\pi n}\right)$

To determine the end slope induced by P, the dummy moment $M_0 = 0$ may be applied, from which it will be found that

$$\phi_{Pc} = \frac{PR^2}{GJ}\sin 2\pi n \quad [26]$$

Similarly the vertical deflection induced by the application of L and M_0 will be found to be

$$\Delta_L = \frac{LR^2h}{2\pi GJn}\left[2\pi n \sin 2\pi n + \cos 2\pi n - 1\right] \quad [27]$$

$$\Delta_{M_0} = \frac{M_0 R^2}{GJ}\sin 2\pi n \quad [28]$$

An evaluation of these reciprocal relationships involves an end-coil analysis to disclose the total effective number of turns so that the equations of the form ($An \sin Bn$) and ($An \cos Bn$) may be determined accurately. This is also true for a precise evaluation of the sinusoidal terms in Equations [24a] to [24f], inclusive. Such an analysis lies beyond the scope of this paper.

It is sufficient, however, to point out that with multiple-spring suspensions, it is possible to dispose the various coils so that their lateral deflection components may counteract.

The point has been raised regarding the lateral deflection under the action of axial loading, inasmuch as it indicates a field of profitable investigation in its bearing on the stable limits of spring loading.

The derivation of the transverse deflection in the body of the paper disregarded the helix angle. Including the effect of the helix angle [$\alpha = \tan^{-1}(f/\pi d)$] gives the following moments:

$$M_x = L\frac{\theta f}{2\pi}\sin\theta + M_0\sin\theta$$

$$M_y = \left(L\frac{\theta f}{2\pi}\cos\theta + M_0\cos\theta\right)\sin\alpha + LR\sin\theta\cos\alpha$$

and

$$M_t = \left(L\frac{\theta f}{2\pi}\cos\theta + M_0\cos\theta\right)\cos\alpha - LR\sin\theta\sin\alpha$$

Also

$$ds = \frac{Rd\theta}{\cos\alpha}$$

By rewriting Equations [4] and [5] and proceeding as in the previous case, the more exact expression for Equation [9] becomes

$$\delta = \frac{LRh^2\pi n}{12}\left(\frac{\cos\alpha}{GJ} + \frac{1}{EI_x\cos\alpha}\right) + \frac{LR^3\pi n}{EI_y}\cos\alpha - \frac{LR^2h}{2}\left(\frac{1}{GJ} + \frac{1}{EI_y}\right)\sin\alpha \quad [29]$$

for the deflection of a spring whose ends remain parallel.

ACKNOWLEDGMENTS

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R. Ebenbach, of the E. G. Budd Manufacturing Company, who assisted in conducting the tests.

E. G. Hallquist, vice-president of General Steel Castings Corporation, who placed the facilities of this plant at the authors' disposal for conducting the tests on the large springs.

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The authors are grateful to R. Eksergian, of the E. G. Budd Manufacturing Company, for his personal assistance and contributions in the preparation of this paper.

Discussion

E. T. P. NEUBAUER.[8] In this paper valuable data are presented on the problem of spring mountings for the isolation of machinery vibration. In order effectively to isolate the vibration of heavy machinery it is necessary to have the critical or resonance frequency of all the various degrees of freedom well below the operating frequency of the machine. This requires a knowledge of the transverse deflection of springs as well as the longitudinal.

A. M. WAHL.[9] The authors have contributed interesting test data which, in conjunction with the analytical work, should be of considerable interest to spring designers.

The writer notes that in deriving the formula for lateral deflection, as mentioned in the paper, the authors have neglected the effect on the axial load. This would be in such a direction as to increase the deflection above the calculated value using the formulas given. Since most of the test deflections were higher than the calculated (in some cases twice as high), it is suggested that a part at least of the discrepancy between calculated and test deflections may be accounted for by the fact that this effect was neglected.

A formula to take into account the increased lateral deflection due to presence of an axial load on a spring laterally loaded (with both ends constrained) has been derived by E. Rausch.[10] This lateral deflection is

$$\delta_{max} = \alpha\delta_n$$

where δ_n = nominal deflection, figured by neglecting the axial load (this is given by Equation [12] of the paper, which agrees with the formula derived previously by Rausch).

The term α is a factor greater than unity which takes into account the increased deflection due to the axial load

$$\alpha = \frac{1 - \dfrac{\dfrac{\delta_1}{h}}{\left(2.6 + \dfrac{\delta_1}{h}\right)\left(1 + 0.77\dfrac{h^2}{D^2}\right)}}{1 - \dfrac{\delta_1}{2.6\,h}\left(1 + K\dfrac{h^2}{D^2} - \dfrac{\dfrac{\delta_1}{h}}{2.6 + \dfrac{\delta_1}{h}}\right)} \quad [30]$$

where

$$K = \frac{0.83 + 0.77\dfrac{h^2}{D^2}}{1 + 0.77\dfrac{h^2}{D^2}} \times 0.93$$

where

δ_1 = deflection under the axial load
h = working height of spring
D = mean diameter

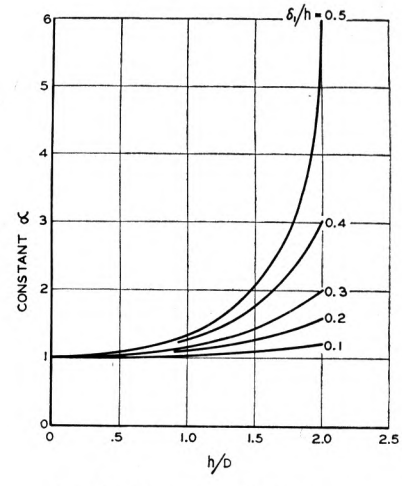


FIG. 9 CURVE FOR FINDING α

Curves given by Rausch for the application of this formula indicate that, where the ratio of working height h to mean diameter D is between 1 and 2, the factor α may be as high as 3,

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[9] Research Engineer, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Mem. A.S.M.E.

[10] "Die Steifigkeit von Schraubenfedern senkrecht zur Federachse," by E. Rausch, *Zeit. V.D.I.*, vol. 78, Aug. 11, 1934, p. 964.

where the axial deflection δ_1 due to the axial load is around 40 per cent of the working height. This effect decreases as the axial load decreases. The constant, α, is plotted in Fig. 9 (given by Rausch) for various ratios of δ_1/h (for ease in calculation). Of course, as indicated by the authors, the question of stability must also be considered.

It would be of interest if this formula could be applied to the springs tested by the authors to see whether or not better agreement could be had between test and calculated results.

Authors' Closure

The authors wish to thank Mr. Wahl for calling their attention to the paper by Professor Rausch, which was overlooked in preparation of this paper. Professor Rausch's original abstract[11] is quite complete and the authors are gratified to note that his formula based on both ends constrained parallel agrees with Equation [12], although obtained through a different mathematical approach.

Equation [30] for the correction factor α was given by Professor Rausch in collaboration with Dr. A. Röver and appeared in a supplement[10] to the original paper. However, there was no proof or derivation given and the authors were therefore unable to check it. For this reason the authors thought it advisable to derive a correction factor which was based on the properties of a column.

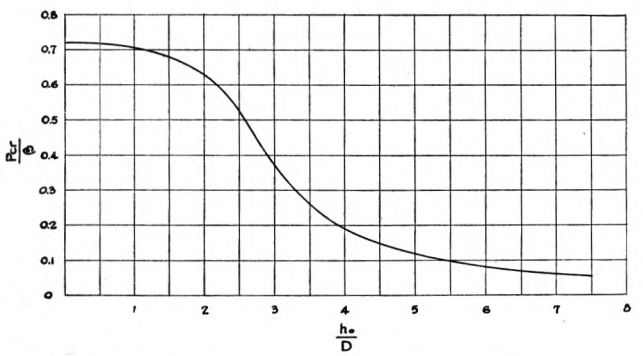


Fig. 10 Curve Showing Ratio of Critical Load to Compressive Rigidity Versus Ratio of Free-Spring Height to Mean Diameter

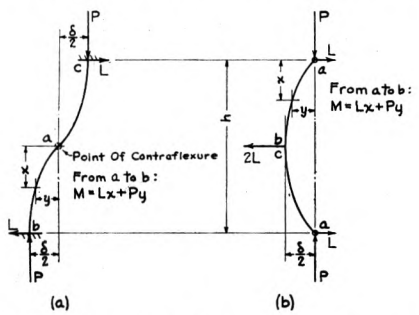


Fig. 11 Showing Similarity Between Column With Ends Constrained Parallel and Pin-Ended Column

It is true that the axial load influences the transverse deflection of the spring. The magnitude of this influence depends on the ratio of the imposed axial load to the critical load at which buckling of the spring occurs (without regard to transverse forces).

The buckling resistance of helical springs has been reviewed by Dr. Timoshenko.[5] For pin-ended connections the critical axial load may be obtained from the curve shown in Fig. 10 which is reproduced from Timoshenko's review,[5] using the values of G and E given previously in this paper. The compressive rigidity β may be found from

$$\beta = \frac{GIh_0}{\pi n R^3} = C_v h_0 \quad \ldots\ldots\ldots\ldots\ldots\ldots [31]$$

where h_0 = free spring height of helix.

From Fig. 11 it will be noted that the deflection curve of a column with parallel ends free to move laterally satisfies the conditions for a pin-ended column, since the bending moment acting on each half of the column (a) is duplicated on each half of the curve (b). It will be recognized that the curve, Fig. 11(a) represents the axis of the deflected spring.

The deflection for a pin-ended column, Fig. 11(b), transversely loaded at the center line has been derived by Dr. Timoshenko[12] and is expressed as

$$\delta = \delta_0 \lambda \quad \ldots\ldots\ldots\ldots\ldots\ldots [32]$$

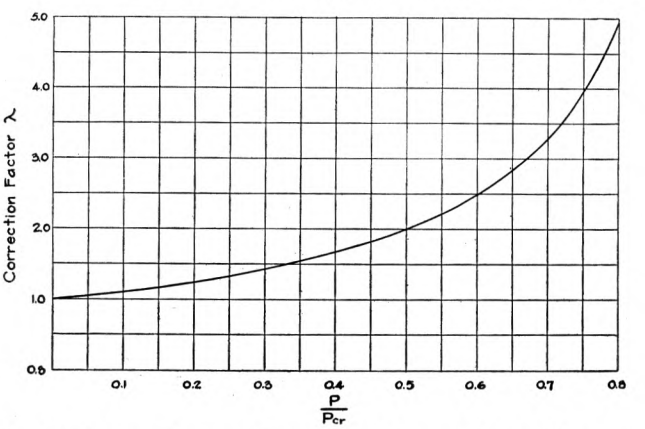


Fig. 12 Curve Showing Correction Factor Versus Ratio of Axial Load to Critical Load

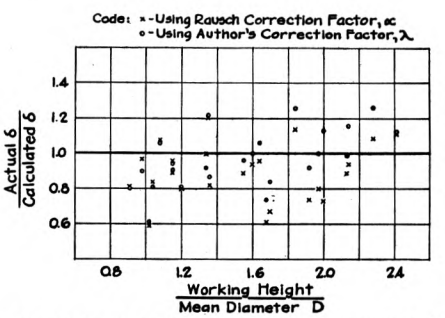


Fig. 13 Ratio of Actual Transverse Deflection to Corrected Calculated Transverse Deflection Versus Ratio of Working Height to Mean Height of Springs

where

$\lambda = \dfrac{3(\tan u - u)}{u^3}$ and δ_0 = deflection due to the transverse load acting alone.

$$u = \frac{\pi}{2}\sqrt{\frac{P}{P_{cr}}} \quad \ldots\ldots\ldots\ldots\ldots [33]$$

Fig. 12 illustrates the correction factor λ expressed as a function of $\dfrac{P}{P_{cr}}$. To find the correction factor it is only necessary to deter-

[11] "Die Steifigkeit von Schraubenfedern senkrecht zur Federachse," by E. Rausch, *Zeit. V.D.I.*, vol. 78, March 24, 1934, p. 388.

[12] "Theory of Elastic Stability," by S. Timoshenko, McGraw-Hill Book Company, Inc., New York, N. Y., 1936, p. 4.

mine the critical load P_{cr} from Fig. 10, from which the ratio $\frac{P}{P_{cr}}$ may be obtained. The correction factor may then be determined directly from Fig. 12.

Examination of Equation [9] shows that the first term represents the portion of the deflection due to the action of bending moments, while the second term represents the portion due to shear forces alone. Since only the former is affected by the column action, Equations [9] and [12] may be written with the correction factor λ as

$$\delta = \lambda \frac{LRh^2\pi n}{12}\left(\frac{1}{GJ} + \frac{1}{EI_x}\right) + \frac{LR^3\pi n}{EIy} \quad \ldots\ldots [9a]$$

$$\delta = \frac{LDn}{10^6\, d^4}(0.271D^2 + 0.206h^2\lambda) \quad \ldots\ldots [12a]$$

The authors have recalculated the deflection ratios of the experimental springs based both on the Rausch correction factor and on the factor λ obtained as just outlined. Fig. 13 illustrates the results plotted similarly to Fig. 8. It will be noted that a much better agreement is obtained when the correction factors are applied; the majority of the calculating values agreeing within 20 per cent of the actual.

A treatment of the correction factors applied to the general case of a spring the ends of which do not remain parallel is too complex to be treated here.

Studies in Boundary Lubrication

Variables Influencing the Coefficient of Static Friction Between Clean and Lubricated Metal Surfaces

By W. E. CAMPBELL,[1] SUMMIT, N. J.

A thorough survey has been made of the literature on static-friction measurement. It is shown that there are large discrepancies among the results of different investigators and it is concluded that the differences are largely due to lack of control of experimental technique. A complete bibliography is presented.

An experimental study of several of the more important variables influencing static friction is described. The variables studied are the effect of load, vibration, and cleanliness of the surfaces. An inclined-plane and a horizontal-plate apparatus are used. It is concluded that the method of cleaning the surfaces is the factor which has the greatest influence upon the results and it is shown that invisible films which may be deposited upon the surfaces during the cleaning process can exert large effects upon static-friction measurements.

A method of cleaning is described which involves polishing the surfaces with moist abrasive. This method yields high and reproducible values for static friction between unlubricated surfaces. It is shown that sulphide and oxide films on steel, copper, and brass produce a lowering in static friction and that the amount of lowering depends upon the thickness of the film.

A comparison is made between the static and the kinetic values obtained on several bearing metals lubricated with typical oils. It is shown that the static values are in general considerably higher than the kinetic values, and it is concluded that the differences which are observed are due in large part to viscosity effects in the kinetic measurements.

An analysis of static-friction measurements on typical lubricating oils leads to the conclusion that moisture and corrosion effects are responsible for many of the phenomena observed. It is concluded that static-friction measurements do not lend themselves to routine testing of oils, but that they can be used to considerable advantage for studying single variables influencing boundary lubrication.

DURING the last few years increasing attention has been given to a study of the factors which govern lubrication in the region where loads and speeds are such that the film of oil separating the rubbing surfaces is no longer continuous. Under these conditions the coefficient of friction for constant operating conditions is no longer a simple linear function of the viscosity (1).[2] The composite effect of the new variables entering into the problem when operation takes place under incomplete film or boundary conditions has been assigned the vague term "oiliness," but no really satisfactory definition has been proposed, nor has much attempt been made to study the separate variables in a quantitative manner. A comprehensive review of the results and findings of investigators of oiliness is given in a paper by Herschel (2), which includes descriptions of testing machines commonly employed. More recently a complete discussion of the subject of boundary lubrication, containing a list of the various factors which may contribute to oiliness, has been given by Hersey (3). In general the important factors may be summarized as follows:

1 The adsorptive force operating at the metal-liquid interface, which may vary with the nature of the metal and of the liquid.

2 The structure of the molecules adsorbed at the interface.

3 Solid films formed on the surface during operation, such as adsorbed graphite films, or oxide, sulphide, and chloride films formed by chemical action between lubricant and metal.

4 The structure of the bearing surfaces.

5 The increase, with increasing pressure, of the viscosity of the lubricant.

A large number of oiliness investigations have been carried out using kinetic-friction measurements, and have been concerned primarily with the development of a machine for routine rating of oils. Kinetic measurements, though they have the advantage that they may be made under conditions closely approximating those obtaining in service (28), yield results complicated by viscosity effects (26). The static-friction method, on the other hand, has the advantage of giving results which are independent of viscosity (4, 5, 10*b*). It involves the use of simple apparatus and lends itself much better to the study of very small quantities of pure compounds under controlled atmospheric conditions.

The present investigation was originally undertaken with the object of studying the effect of molecular structure on boundary lubrication, and the static-friction method appeared to be the most suitable choice for such a study. Difficulties were encountered in the early stages in reproducing the measurements of earlier workers, and reference to the literature indicated considerable discrepancies among the data of different observers using similar methods. It was therefore found necessary to make a detailed study of the variables entering into static-friction measurements, which will be described in the experimental section of this paper.

Rayleigh (6) appears to have been the first to use static-friction measurements for studying boundary lubrication effects. He estimated that the thickness of the films between the surfaces was 1000A or less, and observed that lubrication by these films could not be treated hydrodynamically. Shortly afterward Langmuir (7) showed that static friction between glass surfaces could be lowered from a value between 1.1 and 1.7 to 0.10 by a monomolecular film of a fatty acid transferred to the glass from a water surface. Deeley (4) and Hyde (8) were able to show marked differences between lubricating oils which correlated with their efficiencies as judged by practical experience, and by the Lanchester worm-gear testing machine. Hardy and his collaborators (10) have carried out studies of a high degree of accuracy on a large number of pure organic compounds, and have obtained ex-

[1] Bell Telephone Laboratories.

[2] Numbers in parentheses refer to the Bibliography.

Contributed by the Special Research Committee on Lubrication and presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, held in Detroit, Mich., May 17–21, 1937.

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

TABLE 1 VARIABILITY OF RESULTS OBTAINED BY DIFFERENT OBSERVERS FOR THE COEFFICIENT OF STATIC FRICTION BETWEEN METAL SURFACES

Observer	Brass-brass, μ	Steel-steel, μ
Rennie (14)	0.175	0.146
Jacob (15)	0.07	..
Hardy (10)	..	0.82
Tomlinson (16)	0.63	0.41
Deeley (4)	..	0.17

ceedingly reproducible results which indicate relations of astonishing simplicity between coefficient of friction and molecular weight for members of a given homologous series. Woog (11), Wilharm (12), Wilson and Barnard (5), and more recently Rhodes and his coworkers (13), have demonstrated the value of static-friction measurements between lubricated surfaces for the analysis of oiliness phenomena.

It is to be expected that the measurement of a property such as static friction, which is controlled largely by interfacial forces, would be profoundly influenced by very small variations in the state of the surfaces. A study of the literature bears this out, and indicates that the attainment of reproducible and significant static-friction measurements requires the development of an elaborate and time-consuming experimental technique. There is considerable difference of opinion, however, regarding the extent of control necessary for any given experimental variable, such as preparation of surface, control of temperature, and purity of atmosphere.

Some idea of the spread of comparable data in the literature can be seen from Table 1, in which are given results for dry static friction between similar metal combinations obtained by several independent workers.

The lack of agreement is probably due in large measure to the presence of different amounts of grease or other films on the surfaces. That Deeley's surfaces were contaminated is indicated by the fact that the friction rose when the surfaces were breathed upon, an effect of moisture on friction which is characteristic of greasy surfaces, but which Hardy (10*a*), and the present author have found to be absent for clean surfaces. Deeley also gives figures showing a continuous drop in friction coefficient with load for clean plane surfaces, which is contrary to the law of solid friction, but which is in agreement with results obtained by Hardy and Bircumshaw (10*h*) for thin invisible films of lubricants on steel.

In view of the variability of the results for dry friction, it is not surprising that those for lubricated surfaces are in poor agreement. Coordination of results of different investigators is further complicated by the fact that most of the work reported covers a wide variety of loads, and is concerned for the most part with commercial lubricating oils of indefinite compositions from widely different sources.

Only one group of investigators (10) has made a thorough study of the variables entering into measurements of static friction. In their early papers it is claimed that reproducible values can be obtained only by using highly purified chemical compounds and by carrying out the measurements in a very pure dry atmosphere. Woog (11), Deeley (4), and Hyde (8), however, have published data indicating a high degree of precision on surfaces lubricated with commercial lubricating oils and exposed to laboratory atmospheric conditions.

Woog (11) lays stress on the sensitivity of measurements to vibration and to minute irregularities in the surfaces. Wilharm (12) also found elaborate precautions necessary to minimize vibration effects, but other investigators (5, 17) appear to have obtained reproducible values in the presence of considerable vibration from a driving motor. There is fairly general agreement regarding the necessity for highly polished surfaces, but there is also evidence that the coefficient of friction is independent of the roughness of the surfaces (10*a*).

No work has been done on the possible effect of invisible films of oxide and sulphide which may be present on the surfaces before application of lubricant, although it is well known that such films form during and immediately after the polishing process (18), and Dow (19) has shown that oxide films on brass produce a friction-lowering effect.

There is some difference of opinion regarding the effect of load. Deeley (20), in describing the technique to be employed in using his machine, assumes that the coefficient of friction is independent of the load, but other workers (17*a*, 17*b*), using the same machine, have published data indicating a decrease in the static coefficient of friction up to a limiting load of from 50 to 100 psi, when it becomes constant. Other publications (10*h*, 21) confirm this latter finding and indicate that the coefficient of friction is constant with increasing load for a spherical slider on a plane surface. Hardy and Bircumshaw (10*h*) explain the friction results they observe for small loads and plane surfaces, where the friction is considerably higher than it is above the limiting load, as being due to orientation of molecules by attractive forces at the metal-liquid interface. They consider that these orientation effects extend to a distance of the order of 7μ from the surfaces. The work of Bulkley (22) and of Bastow and Bowden (23) indicates that the length of such oriented chains cannot exceed 0.01μ.

The effect of temperature on the static coefficient of friction appears to be rather complicated. The majority of observers assume no temperature effect over the range of temperatures from 20 to 35 C. This assumption has been clearly demonstrated for pure organic lubricants (10*d*), but seems less certain for commercial lubricating oils in the light of more recent work (10*j*, 10*k*, 13*b*), which shows that the coefficient of static friction may remain constant, decrease, or increase according to the nature of the adsorbable components in the oil.

Hardy (10*g*) has found that if μ_{a-a} and μ_{b-b} are the coefficients of dry friction for metal a against metal a and metal b against metal b, respectively, then the coefficient of static friction between metal a and metal b will be $(\mu_{a-a} + \mu_{b-b})/2$ and that the same law holds true for lubricated surfaces. Tomlinson (16) has confirmed this law, which Hardy calls the mean-value rule, for clean metal surfaces. He finds that it follows by considering friction to be the resultant of the attractive and repulsive forces between atoms, since the former is considerably greater than the latter and a less rapidly decreasing function of the distance between atoms. Similar views are expressed by Shaw (24). However, the data of other observers (8, 13*a*) on static friction between lubricated surfaces show little confirming evidence of such a rule,[3] and it is very uncertain whether it can be generally applied.

Hardy (10*h*) has also noted that large changes in friction may take place with time of contact between lubricated surfaces. The time during which an equilibrium value is approached he calls the latent period, which is always one of falling friction at loads above the limiting load. The length of the latent period is dependent on the molecular structure of the lubricant employed, being large for long-chain molecules having polar groups, such as fatty acids and alcohols, and negligibly small for nonpolar molecules such as straight-chain saturated hydrocarbons. This latent period is considered to be the time of orientation of molecules in the adsorbed layer.

Many of the measurements recorded in the literature have been made with the Deeley machine (20), in which the friction is measured between a stationary three-legged slider and a disk which is started slowly rotating from a position of rest so that both static and kinetic measurements can be made with the same apparatus. Although it is not always clear how the readings are

[3] Rhodes and Allen (13*a*), for example, find that the coefficient of static friction is the same for surfaces of steel, brass, or chromium, lubricated with the same material.

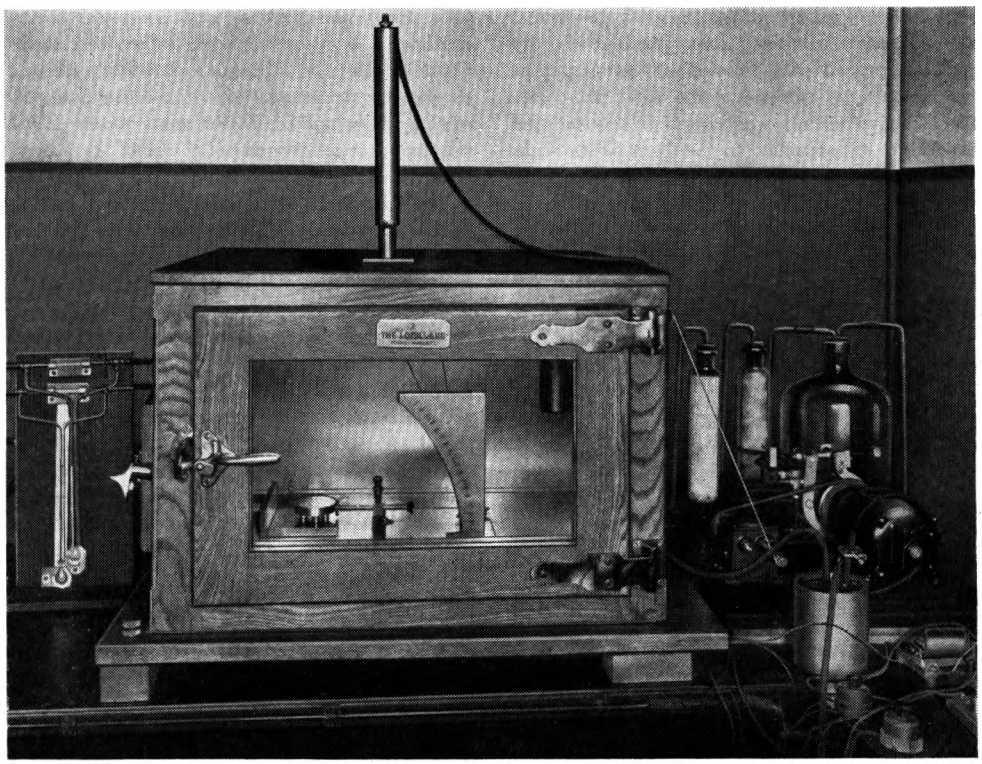


FIG. 1 INCLINED-PLANE APPARATUS

taken, it is likely that most of the values obtained with this machine are kinetic values. Deeley assumes that the static and kinetic coefficients of friction between lubricated surfaces are equal under boundary conditions, and recommends taking the latter value as being more reproducible, and Dover and Helmers (17*c*), using Herschel's modification of Deeley's machine (25), state that their kinetic measurements are independent of viscosity. On the other hand, the results of McKee (1) for a journal-bearing machine, and Bridgeman (26, 27) for a modified Herschel machine, indicate that the coefficient of friction is a function of viscosity practically down to zero speed, and that single measurements of kinetic friction at a given speed cannot be used as a direct measure of oiliness.

On only one point is there fairly complete agreement, and that is that static friction between lubricated surfaces in the boundary region takes place between thin adsorbed films of oriented molecules, although there is considerable difference of opinion, as has already been noted, regarding the thickness of these films. It is also fairly well established that long-chain molecules having a polar group at one end of the chain will be good boundary lubricants, but it is certain that high adsorptive power alone is not a measure of oiliness (5, 10*b*, 11) since substances like water or glycerin, though strongly adsorbed on metals, definitely produce no lowering effect on the static friction between metal surfaces.

EXPERIMENTAL PROCEDURE

Two forms of apparatus, differing mainly in the method of application of a tangential force to a sliding member, were used. The first apparatus, an inclined-plane machine which is essentially the same as that used by Wilson and Barnard (5), is shown in Fig. 1. A polished metal plate, upon which was placed a three-legged slider, rested against a stop on the plane. The plane was tilted by a motor operating through a reducing gear, and when slip occurred between slider and plate, a switch operating a solenoid brake on the motor was thrown, so that the movement of the plane was sharply halted. The fraction of the right angle through which the plane had moved was read to three significant figures by means of a vernier attachment, and the coefficient of friction calculated from this angle. The rate of raising the plane for most of the measurements was 12 cm per min measured at the moving end. This rate was chosen after it had been shown that the static coefficient of friction between lubricated steel surfaces was independent of the rate of raising the plane over the range of speeds from 5 to 20 cm per min. The apparatus was contained inside a metal-lined box through which a current of air, cleaned and dried by passage through sulphuric acid, soda lime, dehydrite, cotton, and glass wool, was passed at a rate of 500 liters per hr. Under these conditions a volume of air equal to that of the box was passed every 15 min and the relative humidity was maintained between 0.1 and 1.0 per cent. The slider was manipulated between readings through a small side door at the left of the box. The box and contents were insulated from vibration by means of layers of sponge rubber and tissue paper. For some of the work slight movement of the slider brought it against a contact attached to the stop on the base plate, closing a relay circuit which operated the solenoid brake on the motor. This method was not always convenient, particularly in certain cases when slip was very slow, and it was found that equally accurate results could be obtained by observing visually the opening of a gap between the back of the slider and a brass disk attached to the base plate.

The second form of apparatus, a horizontal-plate apparatus, which is similar to that used by Hardy, is shown in Fig. 2. The plate carrying the slider was placed upon a brass leveling platform, and the tractive force was applied tangentially by means of a light thread attached to a corner of the slider and passing over a pulley. The load was applied by running water into a light glass bucket attached to the other end of the thread. The plate was prevented from slipping by a step at one end of the leveling platform. The assembly was mounted inside a glass desiccator of 10

liters capacity, bored with holes to admit of the passage of clean dry air during the course of a run. The air was purified as previously described and was passed at a rate of 100 liters per hr. Slip was generally detected by observing the movement of a galvanometer needle when a circuit was closed by contact between a polished strip of metal attached to the front of the slider and a point ground on the end of the feeler of a depth gage set in the wall of the chamber. Successive readings were taken by

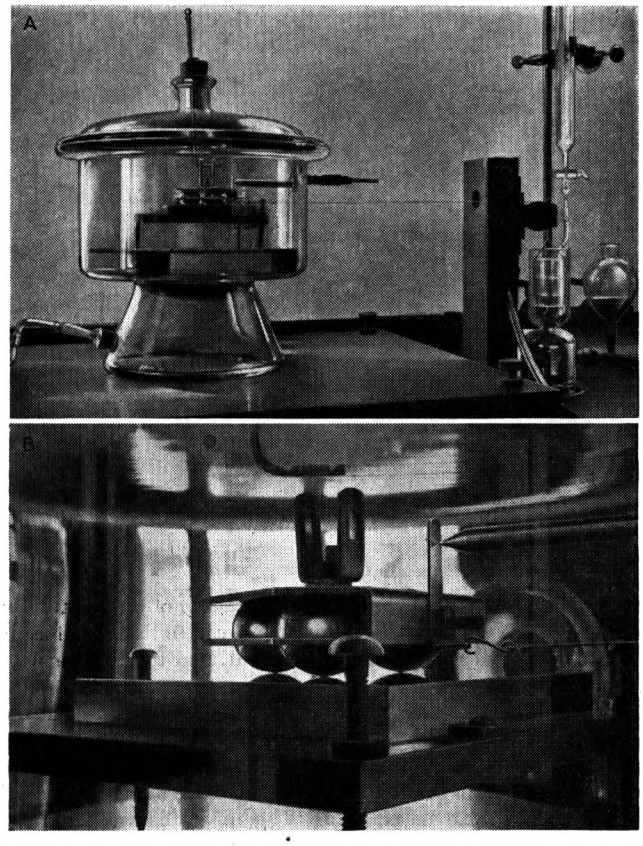


FIG. 2 HORIZONTAL-PLATE APPARATUS

drawing the feeler back 0.010 in. In some cases, however, when experimental conditions were such as to make the electrical contact uncertain, the slip was observed by watching the closing of the gap between the point of the feeler and its shadow on the polished metal strip.

The slider shown in Fig. 1 was the type used in the earlier experiments. The pegs were of speculum metal or steel, and the radius of curvature of their polished ends was 1 in. The mass was 370 g. Sliders of the type shown in Fig. 2 B were generally used. They consisted of three ball bearings at the apexes of a triangle of 2-in. sides, held between two triangular steel plates. By loosening a screw, the balls could be turned so that each of a series of readings could be made on an unscratched surface of the ball if desired. The sliders were of two sizes, fitted with balls of 1 in. and $^1/_2$ in. diam, respectively. Their masses varied from 290 to 300 and 125 to 135 g, respectively. The steel balls were hardened-tool-steel ball bearings and were obtained from the Atlas Ball Company, Philadelphia. The plates were of hardened steel. Two most frequently used were toolmaker's flats obtained from Pratt and Whitney. Others were cut to $3^1/_2 \times 3^1/_2 \times {}^1/_2$ in. from ground flat stock and were hardened, lapped, and polished.

The most important factor to be controlled in order to obtain significant results is the cleanliness of the surfaces. Fatty films only a molecule thick not only can lower the friction between unlubricated surfaces enormously but can lower considerably the value obtained on subsequent flooding with lubricant. The work carried out in these laboratories has shown that the only method of cleaning which removes the last traces of lubricant used in the previous test consists in polishing off the film with moist abrasive on a metallurgical lap.

The data in Table 2 indicate the large effects which an invisible film can produce on the friction between dry and lubricated surfaces. Before each run the surfaces, after solvent treatment, were cleaned by polishing with wet broadcloth sprinkled with magnesium oxide used for metallographic polishing. For the first four runs water was removed from the surfaces, after polishing and scrubbing under water, by rinsing with alcohol and drying in air. For runs 5 to 9 the water was removed by means of a clean linen cloth. Application of a test, to be described later, of the cleanliness of the surfaces, which consisted in the observation of their tendency to be wetted with water, showed that the alcohol-rinsed surfaces were clean, whereas those dried by wiping with the cloth were covered with an invisible contaminating grease film.

TABLE 2 EFFECT OF CLEANING ON COEFFICIENT OF FRICTION BETWEEN SPECULUM METAL AND STEEL

Treatment of surfaces	Run no.	Lubricant	μ	Per cent average deviation from mean
Polished under water, rinsed with alcohol, dried in air	1	None	0.45	3.6
	2	Acto oil	0.28	9.0
	3	S.A.E.-10 automobile oil	0.21	5.9
	4	Acto oil + 1 per cent oleic acid	0.16	3.9
Polished under water, dried with clean cloth	5	None	0.13	11.2
	6	None	0.08	8.4
	7	Acto oil	0.11	5.0
	8	S.A.E.-10 automobile oil	0.10	..
	9	Acto oil + 1 per cent oleic acid	0.16	6.8

It will be observed that the coefficient of friction between the surfaces rinsed with alcohol and dried in air after abrasion (run 1) is more than four times greater than that obtained when the surfaces were dried with a cloth (runs 5 and 6). The effect of the film deposited by the cloth is almost as great for the lubricated surfaces; the results for the two mineral oils are considerably higher when they lubricate the clean steel surfaces (runs 2 and 3) than when they lubricate the cloth-treated surfaces (runs 7 and 8). It will also be noted that the difference between the white Acto oil and the automobile oil is masked by the presence of a contaminating film, and that the effect of addition of oleic acid to the white oil is actually reversed (runs 4 and 9).

The values of the coefficient of friction are the means of ten determinations. It will be observed that the spread of the readings averaged is not lowered by improving the cleaning technique. This was rather disappointing since many published results (4, 10*c*, 11) indicated that static-friction determinations reproducible to three or four significant figures were obtainable.

Although the cleaning method giving the higher values in Table 2 gave more promising results than had been obtained hitherto, the reproducibility of dry values was still rather poor (μ = 0.4 to 0.6 for speculum on steel). After testing various cleaning methods, the following was adopted: The surfaces were first scrubbed with cotton soaked in benzene and acetone, and were then polished on wet broadcloth sprinkled with levigated alumina; the hands of the operator were covered with welt cotton gloves. When the surfaces appeared to be clean they were scrubbed with cotton under running tap water, washed with redistilled ethyl alcohol, and dried by resting the slider or plate on the lip of a glass beaker, cleaned by flaming, set under a bell jar through which was passed a current of clean dry air. The

sliders were harder to clean than the plates, particularly those having balls held between two steel plates, because contaminating material became lodged in crevices and became deposited on the lower surfaces by the alcohol rinse. These sliders were therefore frequently dismantled, the parts cleaned with solvents, reassembled, and extracted with ether in a large Soxhlet-type extractor before the final cleaning by abrasion. All handling of sliders and plates, except during the actual polishing, was done with crucible tongs which had been cleaned by heating to redness. Alcohol was found to be the best solvent to use for removing the water after polishing. In general, absolute alcohol as received left the surfaces clean, but it was found safer to use redistilled alcohol, which was stored in carefully cleaned Pyrex bottles. Other solvents invariably left a film of grease on the surfaces unless they were freshly distilled.

The plates were tested for freedom from grease by wetting with water and draining in a vertical position. When the surface is clean the water will remain in a continuous film until it has become so thin by drainage and evaporation as to show interference colors at the upper edge. A straight line of interference colors will then proceed down the plate. Any drying at the vertical edges should also be preceded by the appearance of interference colors.

It is rather difficult to tell when the surface of a ball is clean by the wetting test, particularly if the radius of curvature is small. The following method has been found the most satisfactory: After rinsing in redistilled alcohol and drying in air, a drop of water is placed on one ball. If the ball is clean, it will wet all over and will be somewhat duller in luster than the two unwetted balls. After about twenty seconds, a circular dry spot, fringed with interference colors, appears near the upper part of the curved surface and spreads fairly rapidly over the ball. If any contamination is present, the water will flash back immediately to the boundary between the ball and the base of the slider, leaving it dry all over.

No method of solvent cleaning alone, such as has been frequently used (5, 13, 20), was found to be satisfactory. Water invariably drew into droplets on steel surfaces cleaned in this manner and the dry coefficient of friction ran between 0.2 and 0.3, for a given slider-plate combination, whereas for surfaces cleaned as previously described, the dry coefficient ran between 0.75 and 0.80.

The necessity for a cleaning technique which provides surfaces free from organic contamination cannot be too strongly emphasized. The work carried out in these laboratories indicates that failure to use adequate cleaning methods is the most important single factor contributing to the lack of uniformity of published results on static friction. Hardy stressed the need for clean metal surfaces in static-friction work as early as 1919. The results of the present work are in full agreement with his findings on this point, and the friction coefficient for steel on steel checks closely his value for steel surfaces cleaned by a similar technique (10*j*).

The plates and sliders were dried for about half an hour after rinsing with alcohol, and were then transferred to the chamber for measurement. Before flooding the surfaces with oil, a few dry-friction measurements were always made as an additional check on the cleanliness of the surfaces. In Table 3 a series of typical results for the coefficient of friction between steel surfaces cleaned in this manner is given. Sliders 6 and 7 were fitted with 1-in-diam balls, and slider 9 with $^1/_2$-in-diam balls. Plates 1 and 5 were toolmaker's flats and plates 6 and 7 were made from ground flat stock. Each of the values given in the table is the mean of at least five readings. The first ten values were obtained using the horizontal-plate apparatus and are picked at random from more than one hundred obtained on surfaces in equilibrium with dry air (relative humidity less than 0.1 per cent) over a 2-year period. Determinations 11 to 14 were obtained using the inclined-plane apparatus in an atmosphere of approximately 1 per cent relative humidity, and determinations 14 to 18 were obtained using the horizontal-plate apparatus in an atmosphere of 70 per cent humidity.

TABLE 3 COEFFICIENT OF STATIC-FRICTION MEASUREMENTS BETWEEN STEEL SURFACES

Determination no.	Normal load, grams	Slider no.	Plate no.	μ	Per cent average deviation from mean
1	295	6	5	0.75	1.4
2	295	6	5	0.82	2.3
3	295	6	6	0.80	1.9
4	295	6	6	0.77	3.4
5	295	6	7	0.75	1.1
6	295	7	5	0.79	3.0
7	295	7	6	0.80	1.3
8	1700	9	5	0.74	2.0
9	133	9	6	0.80	4.1
10	133	9	6	0.82	2.5
11	230	6	1	0.79	1.1
12	470	7	1	0.84	1.7
13	450	7	5	0.80	1.8
14	104	9	5	0.80	1.8
15	295	6	6	0.78	2.0
16	295	6	7	0.74	2.1
17	295	7	6	0.77	3.0
18	295	7	7	0.76	2.5

It will be noticed that over the range of loads tested, the law of friction is obeyed, and that, as would be expected (10*a*), the coefficient of friction is independent of the humidity of the surrounding air.

The grand average of several hundred values obtained over a period of four years for a large number of slider-plate combinations is 0.78, with a maximum deviation of 0.06. For a given slider-plate combination, the maximum deviation is generally of the order of 0.03. This spread is not as low as would be expected from the literature (4, 10*c*, 11), but no higher reproducibility was obtainable, no matter what care was taken.

It had been assumed that the results were independent of temperature. In case temperature effects could account for fluctuations, runs were made at constant temperature with no improvement in reproducibility. It is possible that the extent and nature of irregularities in the surface may be the limiting factor, although no significant difference was observable between the spread of the results obtained on the toolmaker's flat which was optically plane, and of those on the square plates which were flat to only 0.001 in.

The method used for insulating against vibration reduced general building vibration fairly efficiently, but for measurements on the inclined-plane apparatus, an appreciable amount of vibration was transmitted from the motor through the string. In order to determine the effect of this vibration on the reproducibility and on the mean value of the friction readings, runs were made in which readings were taken alternately using the motor lift and a hydraulic lift mounted on top of the chamber. In all, ten readings for each method of lifting the plane were made in a given run, and the means are given (runs 1 and 2 of Table 4). For runs 3 and 4 the chamber and contents were mounted on a vibration-free mounting suspended by steel springs from the ceiling and the hydraulic lift was used.

TABLE 4 EFFECT OF VIBRATION ON COEFFICIENT OF STATIC FRICTION BETWEEN LUBRICATED STEEL SURFACES

Run no.	Surfaces	Method of lifting plane	μ	Per cent average deviation from mean
1	Steel, Acto oil, steel	Motor	0.233	6.5
		Hydraulic	0.262	5.5
2	Steel, oleic acid, steel	Motor	0.102	3.2
		Hydraulic	0.126	5.9
3	Steel, Acto oil, steel	Hydraulic	0.273	10.5
4	Steel, Acto oil, steel	Hydraulic	0.255	6.3

These results indicate that vibration from the motor exercises a small lowering effect on the coefficient of friction. This lowering,

TABLE 5 FRICTION BETWEEN SULPHIDED COPPER SURFACES FOR VARYING FILM THICKNESS

Run no.	Color of film	Equivalent air-film thickness cm × 10^{-5}	μ
1	No film	..	1.17
1	Slightly darkened	1.0	1.07
1	Rose I	1.2	0.91
2	Blue I	1.6	0.88
2	Yellow II	2.8	0.77
2	Purple II	3.9	0.78
2	Blue II	4.3	0.73
2	Blue green II	4.9	0.73
3	Light blue II	4.6	0.78

however, is scarcely greater than experimental error and is negligible for practical purposes. It is also clear from Table 3 that vibration effects were not responsible for the lack of reproducibility in earlier values—in fact the highest spread was found in run 3, which was carried out under practically vibrationless conditions.

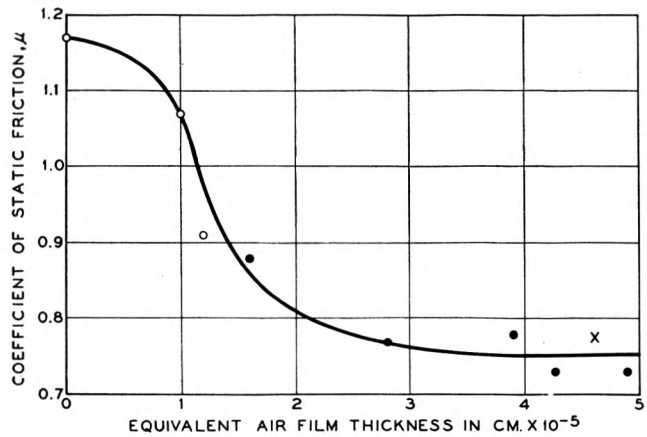


FIG. 3 RELATION BETWEEN COEFFICIENT OF STATIC FRICTION AND THICKNESS OF TARNISH FILM ON COPPER

In the early work there had been indications that the presence of thin tarnish films, such as oxide and sulphide, formed on the surfaces while drying in laboratory air, might be responsible for irregularities in both dry- and lubricated-friction results. Since also the formation of corrosion films plays an important part in the functioning of extreme-pressure lubricants, a series of experiments was carried out to determine the effect of oxide and sulphide films of various thickness on the static friction between steel, brass, and copper surfaces.

The sulphide films were made by immersion of slider and plate for varying times in 0.02 per cent sodium-sulphide solution, and the oxide films were made by heating in air at temperatures from 100 to 500 C.

A preliminary survey indicated that in general the very thin invisible or barely visible films produced little or no effect, but that the friction dropped sharply when a certain limiting thickness of film was reached. For example, the invisible oxide film formed on steel by a 5-min immersion in hot chromic acid and on brass by heating for 1 hr in air at 110 C gave a friction lowering only slightly greater than experimental error, whereas thicker films produced a reduction in friction of the order of 50 per cent. Some results obtained for sulphide films on copper over a wide range of thickness, which illustrate this point clearly, are given in Table 5 and plotted in Fig. 3. The procedure for each run was to dip slider and plate in the sodium-sulphide solution until a given uniform tarnish color was obtained. They were then rinsed with water and alcohol, dried, and used for a friction determination. After testing for cleanliness by the water-wetting test, they were redipped in the sodium-sulphide solution and the friction redetermined. The results given are the means of ten readings. The thickness of the films was not measured directly. It has been shown by Constable (29), however, that the equivalent air-film thickness, which is the thickness of an air film between glass plates giving the same color as the tarnish film under consideration, is directly proportional to the measured thickness. Air-film thicknesses are therefore given in the third column of Table 5 and provide a relative measure of the thickness of the film.

It will be observed that at the pressures used in these experiments (mass of slider = 297 g, diameter of balls = 1 in.)[4] no pronounced drop in friction takes place until the film is thick enough to show well-defined interference colors. The friction then drops sharply to a relatively constant value for thicker films. Similar results were obtained for oxide on copper. The film produced by the action of sodium sulphide on steel appears to be an exception to this behavior in that a sharp drop in friction was measured for an invisible film in this case.

Table 6 gives results for the friction between surfaces of steel, brass, and copper coated by a variety of solid films of thickness large enough that little or no breakdown took place. The unlubricated surfaces reacted clean to the water-wetting test after each set of ten readings, the means of which are given in the table. The results recorded in this table as well as those of Table 5 were obtained using the horizontal-plate apparatus.

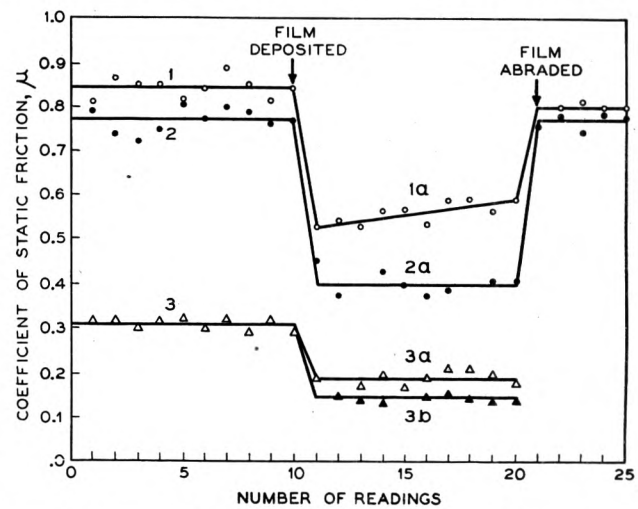


FIG. 4 EFFECT OF SULPHIDE FILMS ON BRASS AND STEEL ON THE COEFFICIENT OF STATIC FRICTION

1 Brass
2 Steel
3 Steel, Acto oil
1a Sulphided brass
2a Sulphided steel
3a Sulphided steel, Acto oil
3b Oxidized Steel, Acto oil

TABLE 6 FRICTION-REDUCING EFFECT OF SOLID FILMS ON STEEL, BRASS, AND COPPER

Metal combination	Treatment of surfaces	Static coefficient of friction, μ: Untreated surfaces	Static coefficient of friction, μ: Treated surfaces
Steel-steel	Oxide	0.78	0.27
Steel-steel	Sulphide	0.78	0.39
Brass-brass	Sulphide	0.88	0.57
Copper-copper	Sulphide	1.21	0.74
Copper-copper	Oxide	1.21	0.76
Steel-steel	Oleic acid	0.78	0.11
Steel-graphite	Oleic acid	0.21	0.09
Steel-steel	Acto oil	0.78	0.32
Steel-steel	Oxide, Acto oil	0.78	0.19
Steel-steel	Sulphide, Acto oil	0.78	0.16

It will be observed that a substantial drop in friction results from the presence of the solid films.

[4] Assuming Hertzian surfaces the average pressure is of the order of 1.5×10^4 psi.

The values for the lubricated solid films are lower than those for the metals lubricated by the same oil, but the reduction is not as large nor as consistent as that which would follow from application of the empirical rules given by Hardy and his coworkers.

Some of the results of Table 6 are given in more detail in Fig. 4. The figures plotted are those obtained for individual readings and a good idea of the reproducibility of the results in the tables may be gained from a study of the curves. The upward trend in the values for sulphided brass indicates some breakdown of the film on the balls; similar but considerably steeper curves were obtained by Dow (19) for friction as a function of the number of slips for thin layers of oxide on brass. He interprets the rise in friction as due to the abrasive action of thin oxide films. His results, however, are all considerably lower than those obtained in the present work, and it is believed that they may be interpreted as being due to the removal of an oxide film contaminated with a thin layer of fatty material.

The last series of readings for curves 1 and 2 in Fig. 4 were taken after the slider had been rubbed back and forth on the plate to produce three areas of clean but roughened metal surface of approximately 1 sq cm. A spot of clean metal approximately 0.04 sq cm in area was formed on each ball. The readings were taken on these abraded areas. The results are in good agreement with the values obtained for polished surfaces and offer additional evidence that the coefficient of friction is independent of the area of contact and of the roughness of the surfaces (10) for solid friction.

In order to find out to what extent the static-friction method could be used to differentiate among a variety of oils and in order to gain some idea as to how the static and kinetic methods of friction testing would compare, studies were made on a series of commercial lubricants of widely differing chemical properties and viscosities on several different bearing metals, after these same combinations had been measured by the Herschel disk friction machine. The static-friction measurements on the bearing alloys were made using the inclined-plane apparatus, and the kinetic measurements were made by Winslow Herschel in the friction and lubrication section of the U. S. Bureau of Standards. After the kinetic measurements were completed, the $^1/_2$-in. balls were removed from the slider and fitted into an aluminum frame so that they were no longer free to rotate, giving a slider mass of 75 g. The plates were the same disks that had been used for the kinetic-friction measurements. The results for steel were obtained using the slider with 1-in. steel balls described earlier, on the toolmaker's flat, using the horizontal-plate apparatus. The surfaces were completely flooded with oil after cleaning, and the plate was rotated through an angle of about 20 deg after each reading. The slider was pressed in place and agitated gently on the plate before a measurement, which was taken 5 min later. The plane was lifted using the motor.

The composition of the alloys was as follows:

	Per cent		Per cent		Per cent
A.S.T.M. No. 1	tin 91	copper	4.5	antimony	4.5
A.S.T.M. No. 8	lead 80	tin	5	antimony	15
A.S.T.M. No. 1	lead 83	tin	2	antimony	15
Frary metal	lead 96.9	tin	0.18	barium	1.71
	calcium 0.70	mercury	0.33	zinc	0.11

The following oils were tested:

	Viscosity, poises at 30 C
Atlantic spindle oil	0.33
Liberty aero oil	8.92
Castor oil	4.73
Lard oil	0.73
Atlantic spindle oil + 2 per cent oleic acid	0.34

The results of the tests are given in Table 7, and show the means of ten readings in the case of the static measurements. These results are represented in Fig. 5, and are reproducible to ±0.01.

TABLE 7 COMPARISON OF THE COEFFICIENT OF STATIC FRICTION BETWEEN STEEL AND VARIOUS LUBRICATED MATERIALS WITH THE COEFFICIENT OF KINETIC FRICTION BETWEEN THE SAME SURFACES

μ_s = Coefficient of static friction (inclined-plane apparatus)
μ_k = Coefficient of kinetic friction (Herschel oiliness machine)

	A.S.T.M. —No. 1—		A.S.T.M. —No. 8—		A.S.T.M. —No. 10—		Frary metal		Steel
Oil	μ_s	μ_k	μ_s	μ_k	μ_s	μ_k	μ_s	μ_k	μ_s
Atlantic spindle	0.23	0.16	0.17	0.14	0.25	0.13	0.19	0.12	0.23
Liberty aero	0.22	0.10	0.23	0.09	0.27	0.085	0.22	0.095	0.22
Castor	0.15	0.06	0.11	0.065	0.12	0.06	0.15	0.06	0.15
Lard	0.08	0.11	0.09	0.07	0.10	0.055	0.09	0.05	0.11
Atlantic spindle + 2 per cent oleic	0.085	..	0.08	..	0.11	..	0.05	..	0.17

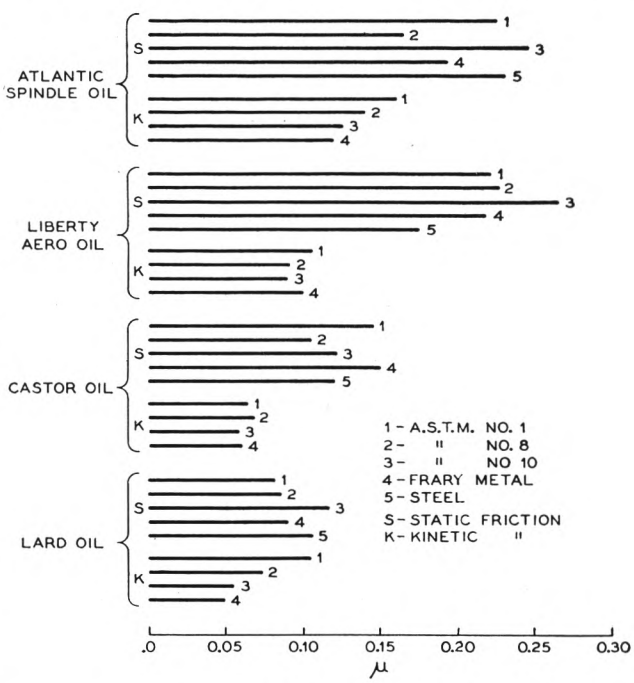


FIG. 5 REPRESENTATION OF COMPARATIVE DATA ON STATIC AND KINETIC FRICTION FOR SEVERAL OILS AND BEARING MATERIALS

It will be noticed that the kinetic values are in general considerably lower than the static values and that for the two mineral oils kinetic values for the low-viscosity Atlantic spindle are higher than those for the high-viscosity Liberty aero, whereas the static values for these two oils are practically the same. These differences are believed to indicate that the kinetic values are influenced by viscosity effects. With one exception, the kinetic values for the two fatty oils are similar. Here, it is believed, corrosion effects enter as an additional complicating factor.

The kinetic values show some tendency to rate the bearing materials in a given order. It would be expected that some similar effect of the bearing material would be apparent in the static values (10*g*, 16), but none is present. Here again corrosion effects may be in part responsible, but it is clear from other measurements made in these laboratories on a variety of materials, under experimental conditions closely approximating those of Hardy, that the quantitative relations describing the effect of the solid surface given by him do not hold generally.

In many of the runs a pronounced downward trend of values, somewhat reminiscent of the latent period mentioned earlier in this paper, was noticed. This latent period (from 1 to 2 hr) was considerably longer than that described by Hardy and Bircumshaw (10*h*), and was noticed for the relatively nonpolar mineral

as well as for the fatty oils, although it tended to be shorter in the latter case. In general the procedure adopted was to continue taking readings until a steady value for two successive means of ten was reached. It was found that the steady value could be reached immediately by rubbing in the lubricant with a linen pad extracted free from friction-lowering impurities, and then rubbing off all but a thin smear of oil. Some evidence was found that this rubbing-in technique gave more reproducible values than the standard method just described, and in one or two cases, where fatty oils were concerned (e.g., lard oil on A.S.T.M. No. 1), a difference greater than experimental error was found between the value obtained by the standard method and the rubbing-in value. This might be due to the presence of a corrosion layer at the interface in the case of the standard value, but the two methods were not investigated in sufficient detail to draw any definite conclusion. It is believed that the downward trend was due in part to the presence of an adsorbed moisture film at the metal-liquid interface. Moisture effects on static-friction measurements have been investigated in considerable detail and will be described in a later paper.

For certain of the runs in Table 7 the effect of an increased load was determined for alternate readings after the steady stage had been reached. The results are given in Table 8.

TABLE 8 EFFECT OF LOAD ON THE COEFFICIENT OF STATIC FRICTION BETWEEN LUBRICATED METAL SURFACES

Lubricant	Metal combination	Load, gr	μ
Atlantic spindle oil	Steel—A.S.T.M. No. 8	75	0.168
		175	0.176
Lard oil	Steel—A.S.T.M. No. 8	75	0.085
		175	0.094
Castor oil	Steel—A.S.T.M. No. 8	75	0.103
		175	0.110
Liberty aero oil	Steel-steel	292	0.22
		717	0.23

These results indicate that under the conditions of test there is a slight increase in the static coefficient with load. The effect, however, is not greater than experimental error and was not investigated in further detail.

Although clear-cut differentiation was obtained among a series of oils, it is clear that the static-friction method is not suitable as a routine test for oiliness. At best it can only give information regarding the state of surfaces at the beginning of motion, and the elaborate precautions required to obtain consistent results provide a further limitation on its use. However, when proper precautions are observed, it provides a useful supplement to other oiliness-testing methods, and can be used to good effect as a research tool for fundamental investigations of the factors operating at the solid-liquid interface in boundary lubrication.

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Stresses in Helical Compression Springs—Present Status of the Problem

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Since Dr. Röver, in 1913, first formulated the modern theory of the stress distribution in a helical spring, including the effect of bar curvature, pitch angle, direct-shear stress, etc., a number of investigators have treated the problem. Wahl, Göhner, Adams, Vogt, Perkins, and Keysor have all presented interesting analyses. Most of these vary somewhat in the detailed method of treatment, but they all arrive at substantially the same result, in terms of the maximum fiber stress in the section. Wahl, in a notable series of experiments, demonstrated that these maximum stresses postulated by the theory are actually present in the spring.

Thus it would appear that we have a satisfactory working theory. But practical spring men feel that there must be something wrong; the theory, when applied to certain standard spring designs with very satisfactory service records, predict stresses which seem far too high to be compatible with good endurance in service. This feeling is supported by the results of a few endurance tests, which indicate rather clearly that the maximum stresses predicted by the theory, though undoubtedly present, do not correctly index the actual endurance.

In the author's opinion, all the evidence points to the conclusion that the problem of design stresses in helical springs is not one for the mathematician, but must be worked out in the laboratories, by actual experiment.

IT WAS as long ago as 1913 that Röver published an article (1)[2] in which was announced, for the first time, a first approximation to the modern theory of the stress in helical springs. Up to that time the Reuleaux formula (2) had been the universal standard; it assumed that the only stress present was pure torsion. Röver's theory recognized not only the torsional stress, but also the direct-shear stress, the bending stress due to the pitch angle, and the variation in torsional stress from point to point of the section, due to the curvature of the bar.

Röver developed his theory in three steps. He first worked out an approximate correction factor for the maximum value of the stress augment due to the bar curvature, plus the direct-shear stress; the pitch angle was taken as zero. He then amplified this correction factor to include the bending stress due to the pitch angle. Both of these formulas are approximate, and both are based on the assumption that the center of torsional rotation coincides with the geometrical axis of the bar. This is incorrect, as the shear moments over the section would not be in statical equilibrium. The center of torsion is displaced a small amount toward the spring axis.

Röver's third step was to develop a more exact but very complicated solution, based on elastic theory, for a particular value of the spring index. This development is therefore of value only as a check on his approximate formula. The latter gives a result differing by only 1 per cent from the more exact solution, for the numerical case examined.

Considering the importance of Röver's theory, it is surprising that it attracted so little attention. It seems to have been practically unnoticed for 15 years, when Wahl made his experiments at the Westinghouse laboratories (3).

This research of Wahl's was magnificently simple in conception and execution. He mounted a half coil of a spring in a testing machine, in such a way that the stress distribution accurately simulated that in the body coils of a complete helical spring, and measured the strains at the innermost helical element (the locus of maximum stress); then reduced these strains to corresponding shear stresses. He also took measurements at the outermost element (locus of minimum stress) and checked these with measurements on the outermost element of complete springs. Due to space limitations, it was impossible to check the complete springs at the innermost element.

Supplementing his experimental work, Wahl announced a revised stress formula. It takes into account the stress differentials due to bar curvature and to direct shear, but not the pitch angle. The correction for bar curvature recognizes the displacement of the center of torsion from the geometrical center of the bar, in accordance with curved-bar theory. Therefore Wahl's formula is presumably more accurate than Röver's first approximation. As to the experimental results, Wahl did not report any numerical figures; however, he did show a graphical plot of his results, together with the theoretical curve for the test springs based on his formula. He ran the tests on two specimens, and as the theoretical stress was a few per cent greater than the experimental result on one, and a few per cent less on the other, we can consider that he got substantial agreement.

Another interesting contribution to the subject was that of Adams (4), who developed an extremely complicated set of equations for the torsional stress. However, it is difficult to appraise accurately the value of Adams' work. The author himself states that "the results derived in the.....analysis are rigorous." But the analysis is not given in full; in an editorial footnote it is stated that the integrations are very difficult and lengthy, and cannot be reproduced, but that they have been examined and checked. Adams gives his final result (a correction factor for the stress augment due to bar curvature) in three separate equations. Then he arbitrarily inserts in the equation another correction factor, representing the effect of direct-shear and bending stresses, and he names certain values for this factor, for various index values. He says he has "shown (this) elsewhere." His formula for displacement of the axis of torsion gives about the same numerical value as the form used by Wahl, for an index of 4. His final correction factors agree closely with Wahl's for very small and rather large index values; however, for intermediate values there is a difference of several per cent. Thus Adams was

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

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

TABLE 1 FORMULAS USED FOR COMPARING STRESSES IN HELICAL COMPRESSION SPRINGS

Author	Formula	Stress Components included	Value of coefficient for $c = 4$
Reuleaux	$\frac{8PM}{\pi d^3}$	A	1.000
Rover (1)	$\frac{8PM}{\pi d^3}\left\{\frac{c}{c-1}+\frac{1}{4c}\right\}$	A-B-C	1.396
Rover (2)	$\frac{8PM}{\pi d^3}\left\{\frac{7}{13}\left[\frac{c}{c-\cos^2\phi}+\frac{\sin^2\phi}{4c}\right]\sin\phi + \sqrt{\left[\frac{c}{c-\cos^2\phi}+\frac{1+\sin^2\phi}{4c}\right]^2\cos^2\phi+\left[\frac{c}{c-\cos^2\phi}+\frac{\sin^2\phi}{4c}\right]^2\sin^2\phi}\right\}$	A-B-C-D-E	1.450
Modification of (2)	$\frac{8PM}{\pi d^3}\left\{\frac{7}{13}\left[\frac{c}{c-\cos^2\phi}+\frac{\sin^2\phi}{4c}\right]\sin\phi+\sqrt{\left[\frac{c}{c-\cos^2\phi}+\frac{1}{4c}\right]^2+\left[\frac{\sin\phi\cos\phi}{4c}\right]^2}\right\}$	A-B-C-D-E	1.450
Wahl	$\frac{8PM}{\pi d^3}\left\{\frac{4c-1}{4c-4}+\frac{0.615}{c}\right\}$	A-B-C	1.404
Adams	$\frac{8PM}{\pi d^3}\left\{\frac{d^3}{16K}\cdot\frac{d-2x}{M-d}\right\}C_1$ where $x=\frac{d}{4}\left\{2c-\frac{1}{c-\sqrt{c^2-1}}\right\}$ $K=\frac{d^2}{12}\left\{\frac{d(c^2+1)}{\sqrt{c^2-1}}+6x\right\}-\frac{M}{6}\left\{M^2\left[\frac{c^2}{\sqrt{c^2-1}}-1\right]+6x\left[M-x\left[1-\sqrt{1-\frac{1}{c^2}}\right]\right]\right\}$ C_1 is a factor to allow for effect of combined stresses.	Doubtful	abt. 1.44
Göhner (1) for S_T	$\frac{8PM}{\pi d^3}\left\{\frac{c}{c-\cos^2\phi}+\frac{\cos^2\phi}{4c}+\frac{\cos^4\phi}{16c^2}+\left[\frac{0.62\cos^2\phi}{c}+\frac{0.53\cos^4\phi}{c^2-c\cos^2\phi}\right]\tan^2\phi\right\}\cos\phi$	A-B-C	1.324
Göhner (1) for S_B	$\frac{16PM}{\pi d^3}\left\{1+\frac{0.87\cos^2\phi}{c}+\frac{0.64\cos^4\phi}{c^2-c\cos^2\phi}+\frac{1}{4c}\right\}\sin\phi$	D-E	1.324
Göhner (2) for S_T	$\frac{8PM}{\pi d^3}\left\{\frac{c}{c-\cos^2\phi}+\frac{\cos^2\phi}{4c}-\frac{\cos^4\phi}{8c^2}\right\}\cos\phi$	A-B-C	
Göhner (3) for S_T	$\frac{8PM}{\pi d^3}\left\{\frac{c}{c-\cos^2\phi}+\frac{\cos^2\phi}{4c}\right\}\cos\phi$	A-B-C	
Göhner (4) for S_T	$\frac{8PM}{\pi d^3}\left\{1+\frac{5}{4c}+\frac{7}{8c^2}+\frac{1}{c^3}\right\}\cos\phi$	A-B-C	1.326
Göhner (4) for S_B	$\frac{16PM}{\pi d^3}\left\{1+\frac{1.12}{c}+\frac{0.64}{c^2}\right\}\sin\phi$	D-E	1.326
Vogt	(for Compression Springs) $\frac{8PM}{\pi d^3}\left\{\frac{0.9875c}{c-1}\right\}$	A-B-C-D-E	1.317
Vogt	(for Extension Springs) $\frac{8PM}{\pi d^3}\left\{\frac{1.0875c}{c-1}\right\}$	A-B-C-D-E	1.450
Perkins for S_T	$\frac{8PM}{\pi d^3}\left\{\left[1-\frac{\sin^2\phi}{2}\right]\left[\frac{2c+1.1\cos^2\phi}{2c-1.4\cos^2\phi}+\frac{0.3\cos^6\phi}{c^2(c-\cos^2\phi)}\right]-\left[\frac{\sin^2\phi}{4}\right]\left[\frac{2c-\cos^2\phi}{c-\cos^2\phi}\right]\frac{1}{\cos\phi}\right\}\cos\phi$	A-B-C	1.320
Perkins for S_B	$\frac{8PM}{\pi d^3}\left\{\frac{\sin^2\phi}{2c}+\frac{2c^2-\cos^4\phi}{c(c-\cos^2\phi)}\right\}\sin\phi$	D-E-F	1.320

SYMBOL NOTATION FOR STRESS COMPONENTS INCLUDED.

A- Pure Torsional Stress
B- Correction for Bar Curvature
C- Direct Shear Stress
D- Bending Stress
E- Direct Tension or Compression
F- Secondary Shear Stress

able to check Wahl's experimental results to about 1.25 per cent; Wahl's test springs were of index 3.

The next contribution we find is that of Göhner. In 1930 and 1931 Göhner published three articles developing the pure theory of the stress distribution in helical springs. In March, 1932, he published a fourth article (5) in which he announced a series of simplified formulas based on his earlier work and in convenient form for practical spring calculations. The stress formula is stated in four different forms, each successive form being a further simplification.

In the year following Göhner's work, Vogt (6) made some ingenious simplifications of the equations developed by Röver as his second step; as explained before, this step includes the corrections for bar curvature, direct shear, and pitch angle. There-

fore Vogt's equations, like Röver's, involve the error of taking the center of torsion at the geometrical axis of the bar.

Vogt makes the interesting suggestion that the stress equation be stated in terms of tensional stress instead of torsional, and in his own paper his equations are so stated. Probably something could be said on both sides of this question, but a discussion of the subject does not come within the scope of the present paper.

At the December, 1933, meeting of the A.S.M.E., Professor H. C. Perkins of Cornell University presented a paper (7) in which he developed a very interesting set of equations. Perkins followed the method used by Röver in his accurate solution for a particular index value; but, he generalized the loading on the spring to cover a combination of an axial force and an axial couple, and he generalized his solution to cover any numerical value of index.

In the following year, Professor Perkins gave us a second paper, in which he gave stress equations for the case of an eccentrically loaded helical spring.

It is necessary to mention one other name. At the A.S.M.E. meeting in December, 1934, Keysor (8) gave us a discussion of the problem from a new standpoint. All the previous investigators had discussed a condition in which the stress distribution was the same in every section of the bar. Keysor, in considering the problem of the number of inactive coils, decided that any assumptions as to axial loading were not justified, and investigated the general case of a helical spring of standard end conformation compressed between two parallel plates. He found that the effective loading was in general nonaxial, the eccentricity varying with the spring index, the number of total coils, and the number of fractional coils. For certain combinations of these variables the eccentricity becomes zero.

While the Bibliography given with this paper is by no means complete, it probably covers the more important contributions to the subject. Now it will be interesting to look at the various formulas that have been mentioned, and compare the results they give.

In Table 1 are listed the formulas, their authors, the stress components accounted for by each, and the numerical value of the correction factor from each formula for an index value of 4. For ready comparison, all the formulas are reduced to a standard form, consisting of the Reuleaux formula times a correction factor, and a common notation is employed. The latter is

- S_b = maximum tension or compression stress (bending plus direct tension or compression)
- S_t = maximum shear stress (torsion plus direct shear)
- S_c = maximum combined stress, in terms of shear stress
- d = diameter of spring bar
- M = mean diameter of helix
- P = applied load, compressing the spring
- c = spring index = M/d
- ϕ = pitch angle.

In order to calculate the numerical values of the correction factors, it is necessary, where the formulas involve the pitch angle ϕ, to assign a particular value therefor. As the fiber stress is a maximum when the spring is completely compressed or solid, and this solid stress is usually the starting point for design calculations, it seems logical to use as typical the value of the pitch angle when the spring is solid. This value is

$$\phi = \tan^{-1} d/\pi M = \tan^{-1} 1/\pi c \quad [1]$$

Listed first in Table 1 is the Reuleaux formula, which probably needs no introduction to engineers. It is the well-known

$$S = \frac{8PM}{\pi d^3} \quad [2]$$

in which only pure torsional stress is considered.

Next come the several Röver formulas. It is easy to see how Röver arrived at his first and simplest form. Neglecting the pitch angle and the displacement of the axis of torsion, he assumed that the stress augment at any point on the periphery of the bar varied inversely as its distance from the cylindrical axis; its maximum value, at the innermost point of the periphery, was

$$S = S_0 \frac{M/2}{(M-d)/2} \quad [3]$$

where S_0 is the mean stress as given by the Reuleaux formula. The fractional term obviously reduces to the first term in the Röver equation $c/(c-1)$. For the direct shear at the innermost point on the cross section, Röver wrote $(P/2)/(\pi d^2/4)$, which, when the term $8PM/\pi d^3$ is factored out, becomes the second term in his correction factor, that is, $1/4c$.

Röver then modifies his formula to give effect to the influence of the pitch angle. Space will not permit a detailed description of the method used. The two formulas developed, for torsion-shear and bending-tension stresses, respectively, are

$$S_t = \frac{8PM}{\pi d^3}\left(\frac{c}{c-\cos^2\phi} + \frac{1+\sin^2\phi}{4c}\right)\cos\phi \quad [4]$$

$$S_b = \frac{16PM}{\pi d^3}\left(\frac{c}{c-\cos^2\phi} + \frac{\sin^2\phi}{4c}\right)\sin\phi \quad [5]$$

Röver then combines these in the classic form (taking Poisson's ratio as $m = 3^1/_3$)

$$S_c = 0.35S_b + 0.65\sqrt{(S_b^2 + 4S_t^2)} \quad [6]$$

Substituting the two values, and rewriting in terms of torsion-shear stress, we obtain the third Röver formula given in Table 1.

This formula can be rigidly transformed into the fourth form. The latter can be further simplified very greatly, with only very minor losses in accuracy. However, a discussion of these does not lie within the scope of this paper.

Reference to the last column in Table 1 shows that for index 4 the value of the correction coefficient has been increased about 4 per cent by including the effect of the pitch angle.

Going now to Wahl's formula, this is quite widely known and used, and a full description of the derivation is readily available; therefore, only a brief statement is necessary here. Wahl used for the displacement of the axis of rotation the well-known approximation $d^2/8M$. He then found that the Röver factor for torsional stress augment, that is, $c/(c-1)$, must be multiplied by another factor

$$\frac{(d/2) - (d^2/8M)}{d/2}$$

and becomes

$$\left(\frac{c}{c-1}\right)\left[\frac{(d/2)-(d^2/8M)}{d/2}\right] = \left(\frac{c}{c-1}\right)\left(1-\frac{1}{4c}\right) = \frac{4c-1}{4c-4} \quad [7]$$

For the direct-shear stress at the innermost point of the cross section, Wahl used the expression $1.23P/(\pi d^2/4)$, and his complete correction factor becomes

$$\frac{4c-1}{4c-4} + \frac{0.615}{c}$$

It may be noted that the first term of Wahl's correction factor yields considerably lower numerical values than Röver's and the second term gives higher values. The two values for the complete expressions, as shown in Table 1 differ by less than 1 per cent for $c = 4$.

Wahl made no allowance for the pitch angle, stating that "it can be shown that a change in slope from zero to 10 deg does not change the maximum shear stress more than a few per cent."

It is impossible to discuss Adams' formulas in detail for the reasons already given. A glance at Table 1 will reveal their complex character. As one of the terms in the final equation is an arbitrary constant, it was necessary to read off the representative numerical value of the correction factor, for index 4, from Adams' graphic chart. As the latter was too small to serve this purpose properly, the tabulated value is only approximate.

Göhner's equations are very interesting. In their simplified forms they are quite easy to manipulate, and calculations for several index values show that for "solid" pitch angles the error in these simplified forms is trifling. This is illustrated by the two values given in Table 1.

Next we come to Vogt's equations. In his simplification of the Röver formulas, Vogt wrote the equation for combined stress

$$S_c = \pm 0.36 S_b + 0.64 \sqrt{(S_b^2 + 4S_t^2)} \quad [8]$$

with the double sign for the first term, and he said that the plus sign applied to extension springs and the minus sign to compression springs.

This procedure seems to be quite correct. In a compression spring, the bending component acts to bend the bar inward (in the direction of the curvature), and the resultant stress at the innermost fiber of the section is compressive. Also the $P \sin \phi$ component of the direct load is obviously a compressive force. According to the authorities, for combined torsion and compression stresses the equation must be written with a minus sign for the first term.

Röver ignored this distinction; in combining his own formulas he wrote the result as in Table 1. Therefore, while Vogt's extension-spring formula agrees with Röver's, his compression-spring formula gives a value about 10 per cent lower. Apparently Adams missed this point also. Göhner did not give a combined form for his own equations; in working out the values given in Table 1 Vogt's practice was followed.

Perkins' equations are quite interesting, as their accuracy is probably of a very high order. As already stated, Perkins followed Röver's method, except that he generalized it to cover any value of the index, instead of one particular value. He found that a "secondary shear stress" accompanies the bending, opposing the stress in torsion. However, its magnitude is very small.

The final equations for torsion-shear and bending-tension stresses are each stated in several successive steps, like those of Adams; however, Perkins' equations can be combined quite readily into the forms given in Table 1. Consolidation of these two into a single equation for the combined stress would yield a rather complicated expression, which is therefore omitted; for an index value of 4 the combined stress coefficient is 1.320, substantially the same as given by the Göhner and Vogt formulas.

Perkins' equations gave effect to an axial couple as well as an axial force; but in order to make his formulas comparative with the others in Table 1, the terms involving the axial couple have been omitted.

Finally, we have Keysor's paper, dealing primarily with stress augments due to the eccentricity of the effective loading. Keysor's complete analysis is not available at the present writing, and in any event there would be no comparison between his formula and those already dealt with. Keysor used Adams' correction coefficient for bar curvature, direct shear, etc., inserting a symbol in his formula to represent them.

But it should be carefully noted that Keysor's analysis predicts a stress augment, due to eccentricity of loading alone, of as much as 28 per cent in certain designs (springs of very few total coils), and that in the outside coil of the standard A.A.R. freight-car bolster spring this augment is about 8 per cent, superimposed on the stress augments already discussed. Some experimental tests have been made, and a more elaborate research is now in progress, the results from which so far seem to confirm Keysor's theory.

From the comparisons given in Table 1, we might reasonably conclude that we have a very satisfactory theory covering our stress problem. Now let us look for a moment at the practical aspect. In the ten years since Wahl published the results of his research, these results have become quite generally known to spring engineers and designers. While they are coming more and more into use, many practical spring men still regard them with a certain amount of skepticism. It is necessary to inquire into the reasons for this attitude.

What these practical men tell us is, briefly, that our correction factors, when applied to certain widely used spring designs, indicate stresses that look rather absurd. Consider, as an example, the outer coil unit of the standard A.A.R. freight-car bolster spring. Literally millions of these springs are riding American freight cars today. The maximum fiber stress at solid height, by the Reuleaux formula, is about 92,000 lb per sq in. The correction coefficient from Wahl's formula, less about $4\frac{1}{4}$ per cent for the effect of the pitch angle, is about 1.4, making the maximum stress 129,000 lb per sq in. If Keysor's theory be correct, we must now add to this about 8 per cent for load eccentricity, and we arrive at a stress of about 140,000 lb per sq in. in torsion. This is well above the usual yield strength of the material.

It is true that these springs rarely or never go solid in service, but they will frequently come close to it. Yet, if made under reasonably good manufacturing conditions, they will give reasonably good service. There are other spring designs in common use, in which stress conditions are more extreme.

The A.S.M.E. Special Research Committee on Mechanical Springs felt that this practical argument merited some investigation. Here we must distinguish carefully between the intrinsic correctness of the Röver-Wahl theory, and its practical application to spring design. There is no doubt that the maximum stresses predicted by the theory are actually present in the spring; Wahl's experimental work proved that beyond any reasonable doubt. The real question is: Are these the critical stresses which index the life of the spring in service, or in an endurance test? Only the test itself can furnish the answer to this question.

Accordingly the Committee arranged, in connection with its extensive fatigue-test program at Wright Field, to have two groups of springs made up from the same rolling and heat of steel, and otherwise alike in every respect except as to the spring index and related dimensions; one group was of index 5 and the other of index 3. The answer we want is a formula for maximum fiber stress, which, when applied to the results of the fatigue tests, will indicate the same endurance limit for both groups.

The test results are shown graphically in Fig. 1. Datum points for the individual runs are plotted in duplicate, with stresses calculated from the Reuleaux formula and alternatively with the Wahl correction factor. Probable S–N curves are plotted for each of the groups, likewise in duplicate. These curves were calculated from the test data by statistical methods, based on the assumption that the S–N curve for ferrous metals is of the form

$$(N - a)(S - b) = k \quad [9]$$

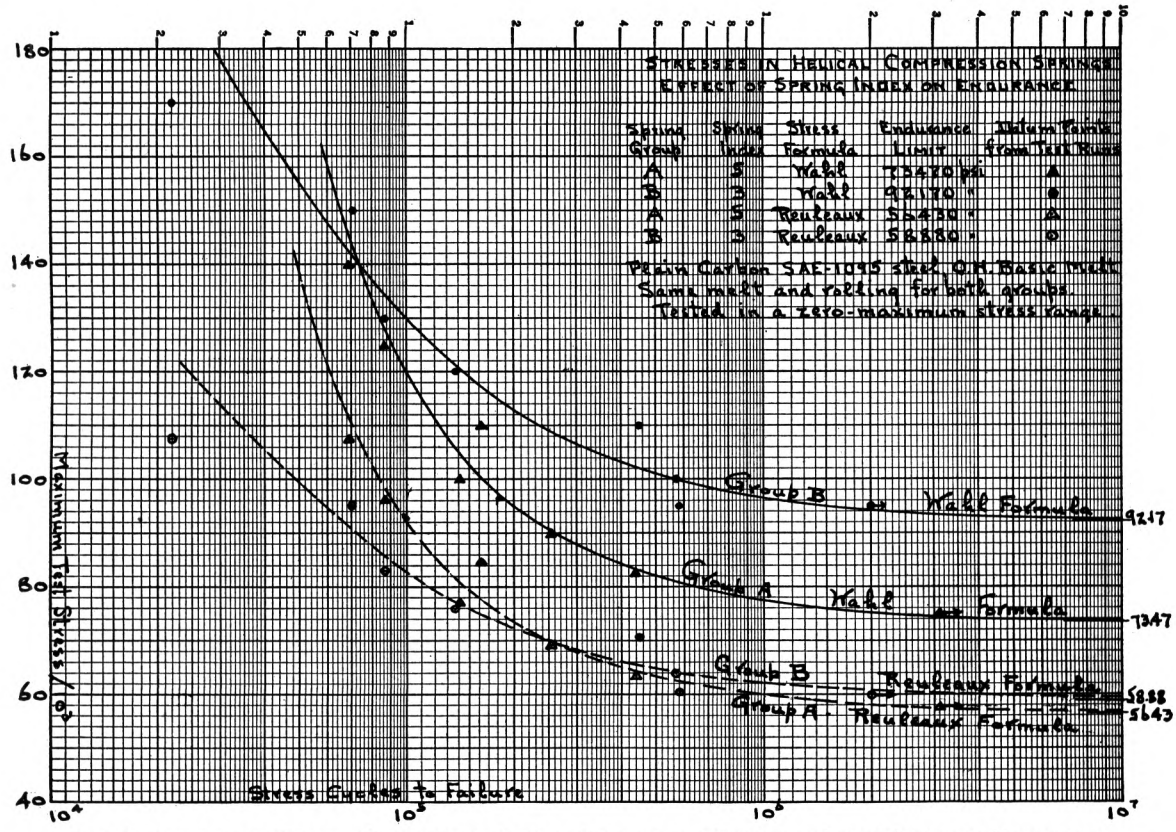


FIG. 1 STRESSES IN HELICAL COMPRESSION SPRINGS SHOWING THE EFFECT OF SPRING INDEX ON ENDURANCE

and that the value of b can be interpreted as the endurance limit. The correlation between the test data and the curves as plotted is quite good for both groups.

A study of Fig. 1 discloses a rather puzzling state of affairs. The two Wahl curves intersect at a rather high stress. Springs in this and higher stress regions are given a quite severe "cold setting" in manufacture, and a possible interpretation of the results might be that under these conditions the springs are very sensitive to local stress augments, and perhaps that they are being subjected to augments not provided for in the Wahl correction factor, such as eccentricity of loading and distortion occasioned by the cold setting; there are several possibilities. As the stress is reduced the two Wahl graphs diverge rapidly, and give predicted endurance limits of 92,170 lb per sq in. for group B (index 3) and 73,470 lb per sq in. for group A (index 5).

For the Reuleaux curves the situation is reversed. They are far apart at high stresses, converge rapidly as the stress is reduced, and the endurance limits are 58,880 and 56,430 lb per sq in., respectively. The apparent conclusion is that for moderately stressed springs local stress augments have little or no effect on endurance. This does not seem quite reasonable, but we can find nothing wrong with the test procedure and data. Further tests along the same lines are projected.

We then got F. P. Zimmerli interested in the problem, and he ran some tests on cold-wound wire springs of different index values. The results are given in Table 2, with an attempt at an analysis thereof.

TABLE 2 TEST RESULTS ON COLD-WOUND WIRE SPRINGS

Spring	Index	Wahl correction factor	Endurance limit, Wahl formula	Endurance limit, Reuleaux	Revised correction factor	Revised endurance limit
A	3.5	1.476	100×10^3	67.75×10^3	1.262	85.50×10^3
B	4.55	1.346	94×10^3	69.84×10^3	1.190	83.11×10^3
C	7.00	1.213	93×10^3	76.67×10^3	1.117	85.64×10^3
D	9.10	1.161	91×10^3	78.38×10^3	1.089	85.36×10^3
E	11.90	1.121	90×10^3	80.29×10^3	1.067	85.67×10^3

The revised correction factors in Table 2 are coefficients, the decimal portions of which have been reduced by a fixed percentage from those of the corresponding Wahl factors. The fixed percentage was established by cut and try, the object being to arrive at a set of revised endurance limits for the various springs which would show a minimum systematic deviation from a mean. The percentage arrived at was 45 per cent. So we might state, as a hypothesis for discussion, that the effective maximum fiber stress in a helical spring is obtained by using a correction factor, the decimal portion of which is 55 per cent of the decimal portion of the Wahl factor.

Mr. Wahl calls this 55 per cent the "sensitivity index," and he attacks the problem along a line of argument, quite different from the foregoing, in a paper[3] being presented coincidentally with this one. He has given a good deal of time and thought to the subject, and what he has to say will be more pertinent than any theory the present writer is able to offer at the moment.

But at least it seems obvious that our theory of stress in helical compression springs, despite the great amount of work that has been done on it and the very consistent results of the mathematicians, will have to be drastically modified to check with what we find in the experimental laboratory, and in actual spring service.

Here the author pauses, somewhat perplexed even to suggest a promising line of attack. Consider for a moment the intricate stress pattern in a helical spring. It includes:

Pure torsional stress.
Direct shear stress.
Bending stress.
Direct tensile or compressive stress.

[3] "Analysis of Effect of Wire Curvature on Allowable Stresses in Helical Springs," by A. M. Wahl, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 61, 1939, p. A-25.

A stress increment or decrement, nonsymmetrical with the bar axis, due to curvature of the bar.

A stress increment or decrement, nonsymmetrical with respect to the cylindrical axis, due to eccentricity of the effective loading.

In springs which have been cold-set, entrapped stresses, of which little is known.

Possibly some kind of stress augment about two turns from the tip of the bar. Endurance-test failures show a tendency to localize in this region. This augment may be related to the load eccentricity.

At the Committee meeting last December, Dr. A. W. deForest remarked to the author that in view of the complex stresses present in a helical spring, it seemed hopeless to expect to solve them. Perhaps the best answer is that we can reasonably expect to find a definite consistency in the stress pattern. The proportions of a helical spring can only vary in a very limited number of ways, therefore, our stress pattern should only vary in a correspondingly limited number of ways. This means that while we may never attain to full understanding of the absolute stresses present, we should be able to find out a great deal about the relative stresses.

Stated in this form, the problem is no longer one for the mathematicians; it becomes definitely the problem of the research engineer. The function of this paper is to describe the problem, rather than to suggest solutions. However, the author has for some time been convinced that we must look primarily to the experimental laboratory for our code of design for helical springs.

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7 "Analysis of Stress in a Helical Spring of Circular Wire," by H. C. Perkins, presented at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, December, 1933, unpublished.

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Discussion

A. M. WAHL.[4] From Table 1 of the author's paper, it would appear that a value of the coefficient of 1.326 (for a spring of index 4) would be obtained by using Göhner's formula, while by using the writer's formula a coefficient of 1.404 is found, the difference between these values being about 6 per cent. However, it should be mentioned that this difference is based on the use of the maximum-strain theory in combining the shear-stress component S_t with the bending stress S_b, the use of this theory being originally suggested by Röver (1)[5] and later used by Vogt (6). If the stress components S_b and S_t had been combined according to the shear-energy (von Mises-Hencky) theory, this difference would have been only about $1^1/_2$ per cent. There are considerable fatigue test data on spring steels which tend to support the shear-energy theory rather than the maximum-strain theory. Moore and Kommers[6] report a ratio between the endurance limit in reversed torsion and that in reversed bending of 0.52 for a 1.2 per cent carbon steel, heat-treated; for a 0.93 per cent carbon steel, heat-treated, a value of 0.53; and a value of 0.50 and 0.56 for quenched and drawn chrome-nickel steels, the average for a large number of alloy steels being 0.58. Also Ludwik[7] reports a ratio of 0.575 between the torsion and bending endurance limits for a chrome-nickel steel. These values are all quite close to the value of 0.577 predicted on the basis of the shear-energy theory, and are considerably different from the value of 0.77 predicted on the basis of the maximum-strain theory. Dr. Koerber, director of the K. W. Institute for Eisenforschung in Germany, in a lecture before the Fifth International Congress for Applied Mechanics also stated that the shear-energy theory appears to yield the best results when compared with the results of fatigue tests on steels made at their laboratories. For these reasons, there seems to be at present a distinct trend toward the use of this theory by engineers interested in strength of materials.

Using the shear-energy theory as a basis, it may be shown that the equivalent stress is given by

$$S_e = S_t \sqrt{1 + \frac{S_b^2}{3S_t^2}} \quad \ldots\ldots\ldots\ldots\ldots [10]$$

where S_b and S_t have the same meanings as given in the author's paper and S_e is the equivalent torsion stress.

Using this formula, taking a pitch angle $\phi = \tan^{-1}\frac{1}{\pi c}$ as was done by the author, and calculating S_b and S_t from Göhner's Equations [3] and [4] given in Table 1 of the author's paper, the value of the coefficient for $c = 4$ becomes 1.384. This differs by about 1.5 per cent with the writer's value of 1.404, obtained by neglecting the pitch angle.

Another theory which is quite widely used in industry is the maximum-shear theory, according to which the ratio between the torsion and bending endurance limits should be 0.50 (which is somewhat below the average values obtained by tests). On the basis of this theory the equivalent stress is

$$S_e = S_t \sqrt{1 + \frac{S_b^2}{4S_t^2}} \quad \ldots\ldots\ldots\ldots\ldots [11]$$

Using Göhner's Equation [4] of Table 1 as before for calculating S_b and S_t, a coefficient of 1.383 is found for a spring of index 4, which is practically the same as that obtained previously by using the shear-energy theory.

These examples show that, if either the maximum-shear or shear-energy theories is taken as a basis, the simple formula previously proposed by the writer, in which the pitch angle was neglected, will give results for most practical springs within less than 2 per cent, as compared with those obtained by Göhner's equations, the derivation of which is based on the more exact methods of the theory of elasticity. For these reasons, it is the writer's opinion that for the present at least the simpler formula may be used with sufficient accuracy for practical work.

Since, at present, no theory for combining the various stress components present in a helical spring can be regarded as definitely established experimentally, it would seem logical to compare individually the stress components figured from the different formulas. The most important component in helical springs is the shear stress due to the axial load. Using Göhner's formula

[4] Research Laboratories, Mechanics Division, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Mem. A.S.M.E.

[5] Numbers in parentheses refer to the Bibliography in the author's paper.

[6] "Fatigue of Metals," by H. F. Moore and J. B. Kommers, McGraw-Hill Book Co., Inc., New York, 1927, p. 147.

[7] "Kerb und Korrosionsdauerfestigkeit," by P. Ludwik, *Metallwirtschaft*, vol. 10, no. 37, September, 1931, pp. 705–710.

[4] of Table 1, for $c = 4$ and $\phi = \tan^{-1}\frac{1}{\pi c}$, the stress S_t becomes $\frac{8PM}{\pi d^3} \times 1.379$. Compared with the writer's value of 1.404 this is a difference of less than 2 per cent. Even smaller differences will be found for larger indexes. As far as the bending stresses S_b are concerned, there is little question about these since they may be figured by well-known theory. The formulas given by Göhner in Table 1 for S_b may be shown to be in close agreement with the results calculated by ordinary curved-bar theory,[8] taking the moment equal to $\frac{PM}{2}$ sin ϕ and adding on the direct stress due to the compression component of the axial load.

These examples indicate that, if elastic conditions are assumed, the formulas available will yield stresses in helical springs which are very close to the actual values. However, in most cases, when compressing the spring solid (as is usually the case during manufacture), the peak stress as given by the formula exceeds the proportional limit or even the yield point in torsion. In such cases yielding occurs and the actual maximum stress is less than the calculated. When the load is released, a residual stress of opposite sign is set up, so that when the load comes on again the peak stress is reduced by the amount of this residual stress.

Referring to the freight-car bolster springs cited by the author as an example, the stress of 140,000 lb per sq in., mentioned would not actually exist when the spring is compressed solid. Instead, yielding of the material would take place at localized regions near the inside of the coil so that the peak stress would not appreciably exceed the yield point. Then too, in usual service as mentioned by the author, the springs are subjected to relatively few cycles of the solid stress, although they may be subject to a great many cycles of a smaller stress. In the writer's opinion there is a definite need for fatigue-test data for cases where there are relatively few cycles of a maximum stress, but in between these cycles a relatively large number of cycles of a moderate stress takes place. When such data are available, the performance of these freight-car springs mentioned may be explained on a rational basis.

The results of the tests carried out on the springs at Wright Field seem rather surprising, since they indicate that neither the stress augment due to curvature nor to direct shear has any effect on the endurance limit. This result may be due to a combination of several factors, including: (*a*) The eccentricity of loading in the two sets of springs of different indexes may have been different and this may tend to mask the effect of bar curvature; (*b*) the material may not be fully sensitive to stress-concentration effects; (*c*) the effect of heat-treatment may be different in the springs of smaller index. In this connection, mention should be made that the sensitivity effect of 55 per cent discussed by the author is not the same as the sensitivity index of that used in the writer's concurrent paper (3).[9] In this paper "sensitivity index" is defined as

$$q = \frac{K_f - 1}{K_c - 1} \quad [12]$$

where K_f = actual fatigue-strength reduction factor and K_c = theoretical stress-concentration factor due only to curvature. The definition given by Equation [12] is in line with that used by research workers on fatigue of metals in this country and in Germany. Using the methods developed in paper (3), the results of Zimmerli's tests (mentioned by the author) may be explained, using a sensitivity index of 1 (defined in this manner). This means that the full sensitivity of the material to stress-concentration effects due to curvature are assumed in figuring the variable component of the stress, the static component being calculated by neglecting stress-concentration effects due to curvature.

The writer fully agrees with the author that a great many more laboratory tests must be carried out before a satisfactory code of design for helical springs will be available. However, it is his belief that analytical work carried out in conjunction with the fatigue tests will be of value in correlating and explaining the experimental results.

Author's Closure

The author notes with pleasure Dr. Wahl's extensive discussion. The point made by Dr. Wahl in connection with the use of the maximum-strain theory is probably well taken. Had the author the task of preparing this paper again, he would probably refrain from any attempt to combine the bending- and shearing-stress components into a single formula, except perhaps in the cases of Röver and Vogt, who themselves suggested combined forms. Apparently the correct theory for combined stresses is still a matter of controversy. In any event, it is of no great importance as related to the subject under discussion. Comparison among the various stress formulas can just as well be made on the basis of shear-stress components only. Dr. Wahl seems to feel the same way about it.

Coming now to the discussion of actual test results, it is the author's belief that Dr. Wahl's argument rather begs the question. The statement made in the paper was not that the stress of 140,000 lb per sq in. was actually present in the standard A.A.R. bolster spring, but that this was the stress predicted by the formulas. Obviously, if this stress is beyond the elastic range of the material, plastic deformation will occur, and the figured stress will not be attained. The author does not believe that Dr. Wahl will contend for a moment that such a condition would not have a detrimental effect on the life of the spring. Indeed, there is very satisfactory evidence from the Wright Field tests that the spring endurance is affected to a marked extent by the amount of "cold setting" in initial tests.

The situation with regard to this problem urgently calls for further experimental work. Assuming that the tests at Wright Field can be continued (regarding which there is some doubt), plans are now being made to run some more tests on the effect of spring index on endurance, in the near future.

[8] "Strength of Materials—Part II," by Stephen Timoshenko, D. Van Nostrand Co., New York, 1930, p. 427.

[9] The effect of sensitivity is treated in the author's bibliographical reference (3).

Discussion

Chlorination of Condenser Cooling Water[1]

H. K. ALDINGER.[2] The author's treatise on chlorination of condenser cooling water presents a very interesting and valuable engineering problem. While this article confines itself principally to the chlorination of circulating water for steam condensers, it should be equally interesting to anyone using process water for other purposes, such as bleacheries, chemical plants, food-product plants, and breweries.

In discussing the author's treatise, the writer takes the liberty to depart somewhat from the main issue, that is, chlorination of cooling water for condensers, because he believes that the slime control in water is of equal importance for any other industrial purpose.

Most operating engineers of power and other plants have to deal more or less with slime and scale-forming conditions in such units as condenser tubes, boilers, air washers, tanks, and pipes. In the case of condenser tubes, as the author states, the economic losses due to slime and scale formation are more pronounced in low-temperature heat exchangers than in high-temperature exchangers.

While chlorination of circulating water in large plants may be of decided advantage, it is doubtful whether it would pay for itself in small plants where a condenser can be cleaned easily over week-end shutdowns, unless the slime and scale formation becomes excessive and retards considerably the heat transfer. Such a condition would occur mostly in the spring and summer months, and where the loss in vacuum would amount to 1 in. or more; however, the warmer circulating water in summertime in itself would reduce the vacuum somewhat. Of course, slime conditions in condenser tubes exist mostly with such plants where they take cooling water from rivers, lakes, or open ponds. It hardly occurs where the cooling water is recirculated through a cooling tower and the make-up water is obtained either from deep-well or city-water supply, as is the case with many small plants.

From the writer's observations, slime conditions are caused by algae in the water. Algae in turn are caused only when vegetable matter or pollen is deposited on or into the water, and have sufficient time to disintegrate under the influence of air and sunlight; they are, of course, most annoying in spring and summer months. Pure cellulose, such as wood and cotton fibers, are the best propagators for the development of algae. However, pollen from grass and other plants furnish their share.

These algae troubles are not nearly as bad in condensers as they are where open water from such sources as ponds and reservoirs is used for processing work, as for instance in bleacheries, food-product plants, and chemical plants. Even filtering will not entirely eliminate these algae, and their propagation is very rapid. The writer's studies have been made with water used mostly for finishing cotton products (bleachery); however, it makes very little difference whether reservoir water is used in bleacheries or in a condenser, slime conditions will occur as long as it is infested with algae. While the writer does not claim to have found a cure for such slime formation, he made some experiments for its prevention by exterminating algae with chemicals other than chlorine.

Of course, in using river, lake, or pond water, the algae are carried in with the water to the plant. It would not be practical to cover a lake or pond in order to prevent vegetable matters or any cellulose from entering the water. On the other hand, a reservoir or tank could be covered, provided that algae are not brought in from outside sources with the water entering the reservoir.

The writer does not intend to discuss the histology of cells. There are many books written on it, and a short and comprehensive treatise on algae can be found in Van Nostrand's Scientific Encyclopedia.[3]

Algae are generally referred to in industrial plants as reservoir or pond scum or waterbloom. They can be found in varied forms in stagnant and putrid water, especially in large reservoirs, ditches, ponds, tanks, and open wells where much vegetation exists and around cotton mills where flying lint and organic dust can easily settle on the water. Algae are classified under the group of Thallophyta.

In the case of algae, the pigment enclosing the nucleus contains chlorophyll, a green coloring matter, while fungi do not. Algae are divided into four classes, when classified by their color, but we are interested here only in the first and second group, that is blue-green algae (Cyanophyceae) and green algae (chlorophyceae). The other two groups occur mostly in sea water. The chlorophyll is what gives the algae its characteristic green or blue color.

While the individual cell of the protoplast of an algae is complex in structure, it can be considered generally to consist of three principal parts: (1) the nucleus; (2) the cytosome or cell body, which envelops the nucleus and is composed of cytoplasm; and (3) the outer cell wall or shell surrounding the cytoplasm. It is the cytoplasm of the unit cell which, when aggregated in large masses, becomes a soft slimy substance of considerable thickness and size and causes all the trouble in condenser and process work.

Figs. 1*D*, 1*B*, and 1*C* show this clearly. Figs. 2 and 3 show water infested with algae, while Fig. 4 shows the same water filtered.

The writer has developed, by photosynthesis, some large specimens of cytosome as shown in the test tubes in Figs. 1*A*, 1*B*, and 1*C*.

The process by which the cytosome develops and grows is still a hypothetical one as far as the writer knows. However, he assumes that under the influence of light, water, and chlorophyll, a great amount of CO_2 is formed in some way, probably by decay. This CO_2 is evidently taken up again by the cells, and in some complex way is transformed into glucose and starch, probably under the influence of some form of amino acids. Of course, this is purely the writer's conclusion, and is based on the starchy character of the cytosome.

A quite similar process can be observed in the fermentation of wine or in making vinegar from apples where a similar slimy skin of considerable thickness is formed and it is from this process the writer got his idea. In the case of either algae or vinegar there is no cellulose present in the cytoplasm. (The cell wall surround-

[1] Published as paper FSP-60-16, by R. B. Martin, in the August, 1938, issue of the A.S.M.E. Transactions, pp. 475–483.

[2] Mechanical Engineer, Fulton Bag & Cotton Mills, Atlanta, Ga. Mem. A.S.M.E.

[3] "Van Nostrand's Scientific Encyclopedia," D. Van Nostrand Company, Inc., New York, N. Y., 1938.

ing the cytosome or cytoplasm consists of cellulose, but the cytoplasm itself does not contain any cellulose.)

It is not the algae as such which gives all the trouble in condensers or other apparatus, but rather it is the slimy mass or scum inherent with algae. In bleacheries, for instance, the algae itself may cause thousands of yards of goods to be damaged by staining the goods in some way or another, mostly green or brown.

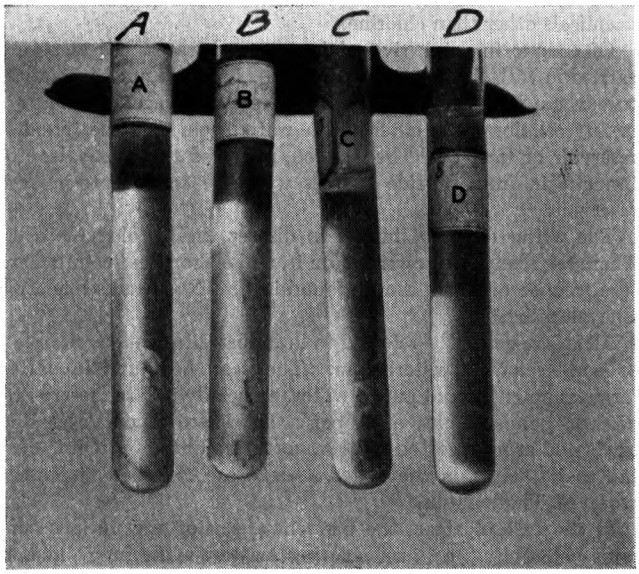


FIG. 1 ALGAE BEFORE AND AFTER TREATING WITH HYPOCHLORITE (*A*, *B*, and *C* show algae developed by photosynthesis; *D* is a sample of the same water treated with hypochlorite, and shows the algae dead at the bottom of the tube.)

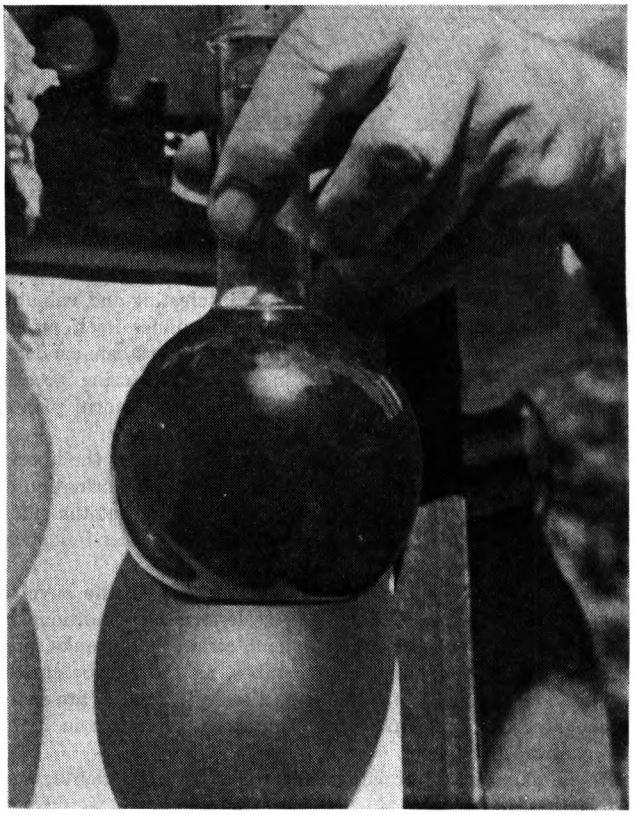


FIG. 3 ANOTHER WATER SAMPLE INFESTED WITH ALGAE

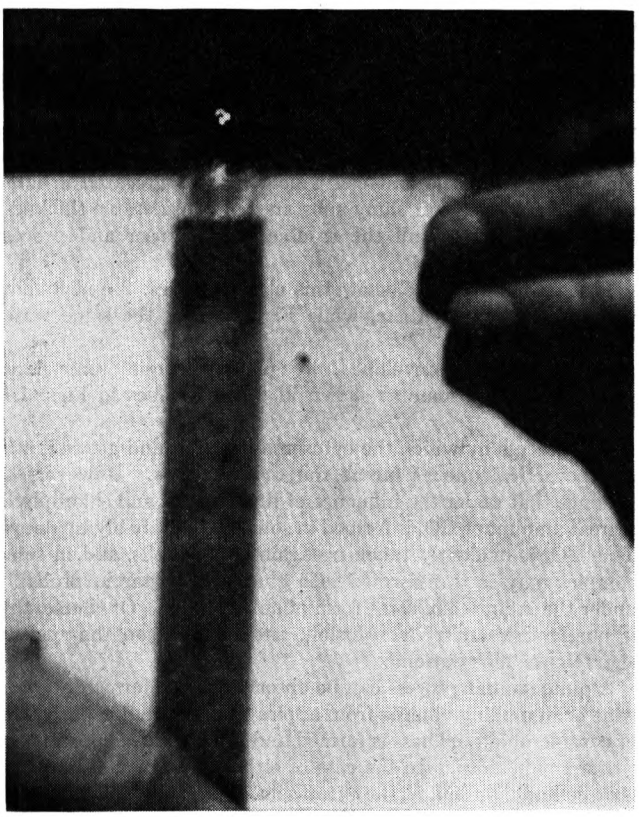


FIG. 2 WATER INFESTED WITH ALGAE

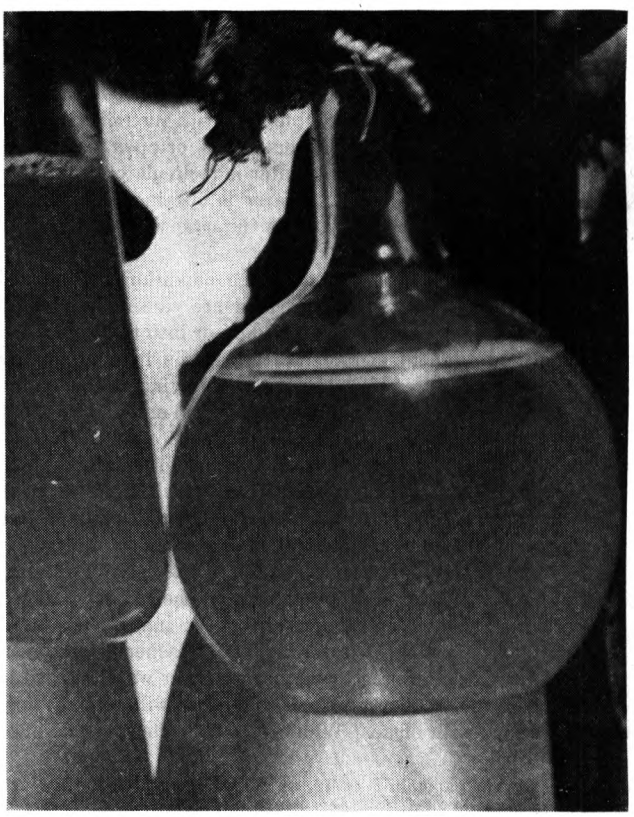


FIG. 4 THE WATER, SHOWN IN FIG. 3, AFTER FILTERING

Such staining is called "possum tack," and is caused when the cloth lays in kiers or bins for some length of time. Small puddles of water containing some algae form between the layers of goods. The water naturally drains in the bin, and the cloth, acting as a filter, holds the algae back. Thus the algae are deposited on the wet cloth, and form the characteristic streaks and stains.

With the common gravity or pressure filter, it is not quite possible to free the water of all algae unless the water is coagulated before and possibly after filtering. And even if the water were thoroughly free from algae, the fuzz and lint from the cloth, or the cloth itself, would only too soon create new algae if allowed to stand for any length of time in the bins, unless the cloth has a high residual chlorine content after bleaching which, however, is not desirable.

The writer conducted some tests with samples of bleached cotton cloth. The first sample was immersed in common clear well water. Algae developed in six days. The second sample was immersed in reservoir water already containing algae and was then exposed to sunlight. In two days the water and cloth sample were green and alive with floating cells, and had an oily and fishy smell. The writer immersed the third sample in city water; this sample held out 10 days before showing traces of algae. The fourth sample was immersed in city water and a 10 per cent hypochlorite solution; it was 20 days before algae developed in this sample.

There is no question but that chlorine or chlorides will inhibit or retard the formation of algae for some length of time and also kill algae already in the water, provided it is supplied continuously to the water in allowable and effective quantities. But as soon as this supply is stopped, only for a short time, the algae will appear again under favorable conditions, even within a few hours.

In inland industries, we have to deal mostly with the blue-green or green algae (waterbloom), which are the ones held together by a gelatinous sheath. The writer has no experience with the treatment of water with direct chlorine gas, but only with hypochlorite derived from the solution of chloride of lime. By calculation, there is about 39 per cent of chlorine present, but only 28.3 per cent is active with the oxygen radical, or approximately 11 grains per 1 lb of water. The writer experimented with a 4-Twaddell solution, but the results were rather disappointing. Figs. 1*A*, 1*B*, 1*C*, and 1*D* show such a treatment with the hypochlorite and while it arrested the further growth of algae, it destroyed neither the algae nor the gelatinous film. On the contrary, it seems to fix the film. Furthermore, the writer found that hypochlorite did not destroy the free swimming cells or gametes, and amoeba are always present where there are algae.

Like in all stagnant water loaded with algae, other organisms of the higher order can be found sooner or later. The writer does not know their names or origin, but supposes that these are carried from the air into the water, and probably are attached to vegetable matters or cotton fibers. Some may be mosquito larvae, which we call wigglers.

The wigglers, which are brownish red in color and develop to about $^3/_8$ in. in length, seem to be the chief enemy of algae. They seem to be especially fond of algae and incessantly eat a great mass of it. Unfortunately, a yellowish brown amoeba attacks and kills the wigglers.

One could conclude that if enough wigglers were present in the water they would destroy the algae and all that would be necessary, before using the water, would be to strain off the wigglers. The writer has not experimented along that line but found that hypochlorite has very little effect on these bugs unless used for a long time and in a strong solution which would eventually also kill the algae.

Somewhat disappointing as it was to use hypochlorite for condensers, air washers, and process water, the writer experimented with some other methods for directly eliminating the algae in the water, since filters let as much as 1 per cent in volume of the algae through; however, a strong solution of hypochlorite eventually arrested the growth of the algae, as shown in Fig. 1*D* where it can be seen the water is fairly clear and the algae dead at the bottom, it took nearly a full-strength solution of hypochlorite. Of course, neither in condensers nor in a bleachery can such a strong solution of hypochlorite be used without affecting the metal.

The writer first experimented with coagulation. Fig. 5*A* shows filtered water, containing some algae, which was treated with alum.

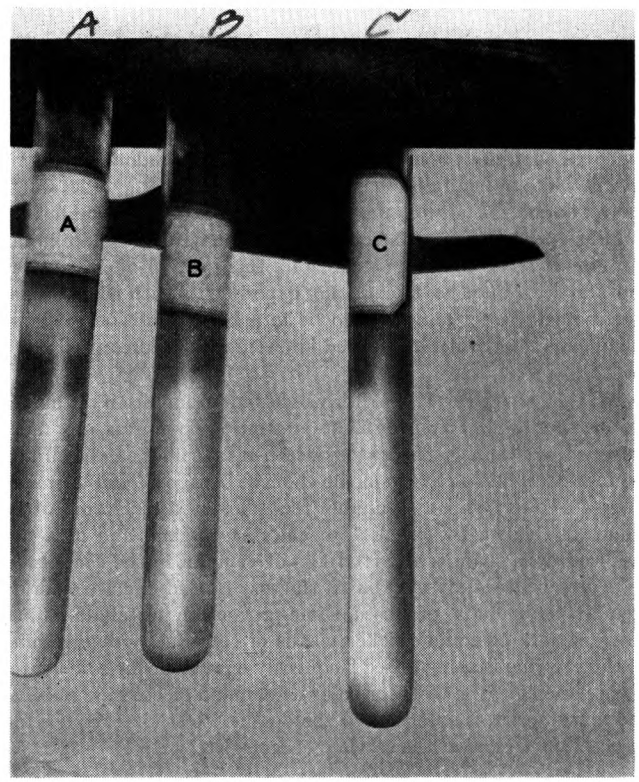


FIG. 5 FILTERED WATER, CONTAINING SOME ALGAE, TREATED WITH (*A*) ALUM, (*B*) ALUM WITH BICARBONATE OF SODA ADDED, AND (*C*) TRISODIUM PHOSPHATE AND HYPOCHLORITE

Fig. 5*B* shows the same water except bicarbonate of soda was added to the alum; the precipitate can be seen on the bottom of the test tube. The water was cleared perfectly, although it took some time.

Fig. 5*C* shows a precipitate made of trisodium phosphate (Na_3PO_4) and hypochlorite. It is this solution to which the writer would like to call the reader's attention, since it worked remarkably well and has many advantages. So far the writer tested this $Na_3PO_4 \times Ca(CLO)_2$ solution only in test tubes, but believes he has found a cheap, quick, and very effective coagulant that not only precipitates and actually kills the algae quickly, but also is noncorrosive to metal; furthermore, it precipitates all the hardness-producing salts, especially in well water, and actually softens the water. In other words, this combination of trisodium phosphate and hypochlorite not only is a strong coagulant but also a disinfectant and water softener.

Cloth samples in such treated water did not show any algae formation after 30 days of exposure. The strength of this test solution was 1 per cent Na_3PO_4 and 2 per cent $Ca(CLO)_2$ by volume.

The writer also found that this solution killed all other bacteria and insects in the water at least three times quicker than hypochlorite alone. There is only one drawback to that solution and that is that it intensified the smell of the dead algae; however, this could probably be overcome by aeration.

Of course, Na_3PO_4 has been and still is used for treatment of boiler feedwater, but the writer is not aware that it has been recommended as a coagulant and disinfectant for water in general and especially for algae extermination.

The cost is nominal—about $2^1/_2$ cents per lb for the trisodium phosphate and about 2 cents per lb for the chloride of lime. According to the strength of such a solution required, it would amount only to a few cents per 1000 gal of water.

But the main value, and the point the writer is coming to, is that it absolutely prevents any and all slime formation of the algae; also, it is practically noncorrosive and protects iron instead of attacking it. This last statement is based on the limited experimentation conducted by the writer. However, the writer believes that the prevention of slime formation alone would be worth while to justify further experimentation.

No doubt the author may already know of these materials, and has probably tried them; however, the writer would like the author to relate his experience along this line. But in case the author should not have made any experiments with the material mentioned, he certainly should be in a better position to do so than the writer, with his limited knowledge and apparatus.

W. A. CARTER.[4] The writer would like to inquire if, as an aid to the operation of chlorine slime-prevention apparatus, any more rapid means than the laboratory death-rate studies can be offered to indicate when a change in the chlorine dosage should be made. There is often an unseasonable requirement for chlorine other than that shown by the test for chlorine demand and it is not realized until there is a partial loss of vacuum in the condenser. The death-rate study may be satisfactory in studying the chlorine requirements for a proposed slime-prevention installation, but there should be some quicker means of continually determining if the chlorine is being dosed frequently enough or for sufficiently long periods. The tests for chlorine demand or chlorine residual do not afford this answer, and the death-rate studies require too much time as well as the services of an expert.

ALEX D. BAILEY.[5] The writer would like to carry the history for this subject back beyond that which the author mentions. His attention was called to it by the paper on the treatment of water for condensing purposes with chlorine by L. L. Robinson,[6] who gave the origin of the idea and reports on tests conducted at the Hackney Electricity Works in England. About the same time a paper on the same subject was presented by H. W. Coulson,[7] who also reported on results at the Hackney Electricity Works.

As a result of this work, the engineers of the Commonwealth Edison Company, Chicago, took the matter up with representatives of Wallace & Tiernan Products and with the Electro Bleaching Gas Company. The representatives contacted did not know of the work done in England, but an agreement was made between the three organizations to conduct tests on the condenser of unit No. 3 at Northwest Station of the Commonwealth Edison Company. Twelve tubes were segregated in each of the two passes of the condenser and the water supplying these tubes was chlorinated. The test was started January 27, 1924, and continued until April 18, 1924. Although the results were quite satisfactory, it was decided to run a second test similar to the first, but using new tubes. This test was started April 20, 1924, and continued to June 21, 1924, and at that time the tubes were removed, split open, and comparisons were made with tubes using untreated water.

As a result of these tests, it was decided to treat all the water for this condenser, injecting the chlorine into the suction of the circulating pump. This test was started September 8, 1924, and ran until the end of October, the chlorine dosages varying from 0.25 to 0.45 ppm. In these tests, chlorine was fed continuously and the fact was established that the growth of slime could be prohibited, although it was not satisfactorily proved that it could be destroyed after it had formed. Because of the large amount of chlorine required and the high cost, which was then from 8 to 10 cents per lb, it was decided that the improvement would not warrant the expense of a permanent installation and its operation.

In the Transactions of the National Electric Light Association[8] there is a description of these tests, which is a part of the report of the Condenser Subcommittee of the Prime Movers Committee.

Shortly after the Crawford Station of the Commonwealth Edison Company was started, trouble was experienced with the sliming of the generator air coolers on unit No. 3, and chlorinating equipment was installed. This consisted of two chlorinators supplied by Wallace & Tiernan, one of which was of the vacuum type, intermittent feed; the other was of the old-style continuous feed. This equipment went into service August 16, 1926. The intermittent machine, probably the first of its type, proved unsatisfactory. It was subsequently rebuilt and put in service January 31, 1927, after which tests were continued until August of that year.

About this time, Dr. Enslow of the Chlorine Institute was contacted and he became quite interested in the results. On June 4, 1928, he came to Chicago and a meeting was held with representatives of Wallace & Tiernan, Electro Bleaching Gas, and the Commonwealth Edison Company. At this meeting it was agreed that the chlorination of all the circulating water for a large condenser should again be attempted using the intermittent process as this would reduce the amount of chlorine required. In the meantime the cost of chlorine had been considerably reduced. A very careful record of the performance of condensers for No. 1 unit at Crawford Station using raw circulating water was maintained for the balance of 1928, and arrangements were made to install intermittent chlorinating equipment on these condensers to be ready for service about the beginning of 1929. Those tests were successful and led to the permanent installation of chlorinating equipment on additional units.

M. S. MAURER.[9] Several parts of the author's paper express ideas which seem contrary to the writer's experience.

In the minimal chlorine death-rate studies, it appears that saline suspensions were used for the bacteria. These studies would seem to be of little practical value as they do not show the effect of the presence of organic matter. This is of major importance, for when the chlorine is added to a circulating water, it reacts with the organic matter first before acting on the bacteria.

It was shown by the studies in connection with the use of chloramines at Williamsburgh, that unless the organisms were

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[5] Chief Operating Engineer, Commonwealth Edison Company, Chicago, Ill. Fellow A.S.M.E.

[6] "Chemical Treatment of Condensing Water," *Electrical Times*, June 22, 1922, p. 607.

[7] "A New Use for Chlorine," by H. W. Coulson, a paper delivered before a meeting of The Institution of Mechanical Engineers and the Chemical Engineering Group of the Society of Chemical Industry and later published in *Chemical Trade Journal* and *Chemical Engineer*.

[8] Vol. 83, 1926, p. 1009.

[9] Williamsburgh Power Plant Corporation, Brooklyn, N. Y.

freshly isolated and suspended in water taken from the circulating system, little if anything was accomplished by the laboratory work. It is well known that changes in the organic matter caused by bacterial growth will take place which affect the chlorine demand and therefore the germicidal efficiency. Thus, the time elapsing between taking the samples and running the tests is of great importance.

With regard to the ideas expressed by the author as to the formation of chloramines by the reaction of chlorine with ammonia occurring naturally in water, it appears to be generally recognized that chloramines are not formed from natural ammonia. We have examined a number of waters on the eastern seaboard in which the natural-ammonia content as determined by both direct Nesslerization and distillation was more than sufficient to form chloramines and in every instance, when an ammonium salt was added to the water, the chlorine demand was lowered. This is direct proof that the natural ammonia in these cases, did not form chloramines. It is true that chlorine will act more rapidly, in water free of organic matter, than chloramines. This property is lost, however, when the water contains sufficient organic matter to support the growth of bacteria in large numbers, as is the case in slime-forming waters. In this instance, chlorine reacts more rapidly with the organic matter than with the bacteria and a smaller amount of chlorine remains available for bactericidal action than where chloramines are used.

The author states that the formation of chloramines is of advantage in waters of abnormally high organic chlorine demand as it retards the combination of chlorine with the organic matter, thus requiring a lower quantity of chlorine to obtain a given residual. If this be true of high-organic-demand water then the same advantage must hold true in waters which have any organic demand whatever. As previously stated, all slime-forming waters have an organic content, and actual practice demonstrates that chloramines do have this definite advantage over straight chlorine in slime-forming waters.

In cases where the natural ammonia is high, unless an ammonium compound is added or in other words, a true chloramine treatment used, chlorine is at a decided disadvantage as the chlorine demand will probably be so high as to make a straight-chlorine treatment uneconomical.

The large generating station in the East referred to is the Williamsburgh plant of The Brooklyn Manhattan Transit Company. The author is entirely wrong in his statement as to the water conditions at this plant. The natural ammonia content is not low as stated, but high, and would be more than sufficient to form chloramines if they could be formed from natural ammonia. During the period when the author's company was trying out a straight-chlorine treatment at Williamsburgh, it was impossible to obtain slime control. However, by the addition of ammonium sulphate to this same water, the chlorine demand has been lowered and satisfactory slime control obtained. It would be interesting to know what reason can be ascribed as to why it is necessary to add ammonium salts at this plant, to get a chloramine treatment when the natural ammonia content is high.

In the cases cited in the Middle West and a small plant in the East, where the natural ammonia content is so high that excessive quantities of chlorine would be necessary to satisfy the demand due to the formation of chloramines, the writer is quite sure that the author is mistaken in the statement that chloramines would be formed, because if they were it would be unnecessary to add sufficient chlorine to combine with all the ammonia and only enough would need be introduced to form chloramines in amounts to give the desired germicidal action.

In waters containing natural ammonia there is usually an organic demand. It would seem the cases mentioned should be particularly adapted to the use of chloramines and were an ammonium compound to be added to any of these waters, the chlorine demand would drop and a more efficient bactericidal treatment obtained than if sufficient chlorine were added to satisfy the demand and give a germicidal residual.

With regard to the question of intermittent treatment, as interpreted in the past by the author's company, which necessitates complicated and expensive program equipment, it has been demonstrated by four large companies here in the East, namely, The Brooklyn Manhattan Transit Company, The Brooklyn Edison, The Interboro Rapid Transit, and the Public Service of New Jersey, that a half-hour dosage, once in 24 hr, is decidedly superior to short frequently repeated dosages. This is in accord with general bactericidal practice, the reasons for which are well known and it seems to be doubtful whether in any case the short frequent dosage would be preferable.

The mechanism of using a bactericidal treatment for slime control with the condenser in service is to use the circulating water as a carrier for the germicide which is to act on the organic growths on the tube surfaces. In the case of chlorine or allied chlorine compounds, the character of the water influences the amount of germicide available to kill the slime-forming organisms on the tubes. This germicidal residual is ordinarily determined with orthotolidin, and the orthotolidin is added immediately when the sample is taken. This immediate residual is not necessarily the germicidal residual available to kill the bacteria. At least 5 or preferably 10 minutes should elapse before the orthotolidin is added. The reason for this is that chlorine reacts first with the organic matter before reacting with the bacteria. This also applies to the organic matter in the layer of slime as well as the organic matter in the water. The sequence of events is the chlorine or chlorine compound first reacts with the organic matter in the water, then with any bacteria in the water and any residual remaining will then act first on the organic matter in the slime on the tube and last on the organisms in the slime layer. This is why the immediate residual has little significance and also one of the reasons for the half-hour dosage.

However, the most important thing in which we are interested is the slime layer on the tubes. This slime is very high in organic matter compared to the water from which it originates, and, as the author states, chloramines are superior to chlorine in the presence of high organic demand; it follows, due to this fact that chloramines will be advantageous over chlorine in acting on the actual layer of slime. This difference in action on organic matter results from chloramines being a much weaker oxidizing agent than chlorine.

Any water exhibiting an organic chlorine demand and which will form slime can be regarded as polluted. It is stated in the paper that chloramines are much slower in their action in slime control than chlorine, yet it seems to be generally recognized that in polluted water this is not so and the two are approximately equal in speed of killing in the presence of the organic matter, which is always present in slime-forming circulating water. It is difficult to reconcile some of the ideas the author expresses about chloramines with what has been published by his company on this same subject. For example, in Wallace and Tiernan's[10] Technical Publication No. 11 it is pointed out that the germicidal value of chloramine is almost three times that of chlorine. Further advantages of chloramine over chlorine are shown in Wallace and Tiernan's[10] Research Publication No. 327 where it is also pointed out that the use of chloramine is more specific than straight chlorine and that Wallace and Tiernan[10] found chloramines a great improvement over straight chlorine when used on slime-forming bacteria; also in acting on slime-

[10] Wallace and Tiernan Products, Inc., Belleville, N. J.

forming bacteria, chloramines require only one fourth the dose that chlorine required for the same length of contact and that, for the same dosage as chlorine, chloramines required only one fourth the period of contact.

This same publication also points out that the difference between chlorine and chloramine is that chlorine is a strong oxidizing agent and chloramine a very weak oxidizing agent; therefore, when bacteria are to be killed in the presence of a chlorine demand, the use of chloramines results in a larger proportion being available as a germicide than is the case with straight chlorine.

Further advantages of chloramine over chlorine are also shown in Wallace and Tiernan's[10] Research Publication No. 305.

The use of chloramines for the elimination of slime, at Williamsburgh and several other places, has indicated that they have the advantage over chlorine as stated in the Wallace and Tiernan[10] publications and that the ideas expressed by the author do not seem to be in accord with the results obtained.

Author's Closure

Mr. Aldinger has presented interesting details of a particular water treatment, in the handling of a type of problem common in the process industries. In the paper presented in December, 1937, the author was dealing with but one phase of the broad subject of water treatment, only that involved in the chlorination of very large volumes of cooling-water flow, as contrasted with the relatively smaller volumes of water that require sterilization in the recirculated spray pond or cooling-water systems, or in industrial-process water plants.

As was pointed out in the opening remarks of the paper, the present-day industrial uses of chlorination are founded on the accomplishments of chlorine in the field of drinking-water sterilization. An extensive review of the literature, including very old and more recent United States and foreign patents, discloses innumerable applications of a wide variety of chemicals and of special mechanical devices, having as objectives a method of bacteriological treatment that would prove to be reliable under any one of the many conditions common to the water-treatment field. These many alternate methods have not advanced into the realm of practical application, since in each case one or more important features have been found lacking, that have made chlorination so universally utilized at the present time. These features include availability of the chemical in convenient form, the extremely low cost now characteristic of chlorine, the development and advance in design of reliable proportioning and control apparatus, and the simple tests available for checking satisfactory chlorinator operation.

At the present time, in addition to the practically uniform use of chlorine in sterilization of drinking-water supplies, chlorination is also in general use in the smaller cooling-water plants, and in industrial-process water treatments. These smaller applications involve continuous rather than intermittent use of either chlorine or chloramines.

It has been the author's experience that slime growths are equally as troublesome, if not more troublesome in the smaller power and process water plants, and represent proportionately greater economic losses. It is true however that less frequently do these losses reach the same magnitude.

Wide experience is valuable in deciding upon the best and most economical method of accomplishing the desired result, without the risk of adverse reactions in the water-treatment plant, or in subsequent process steps, and in deciding whether continuous or intermittent applications of chlorine or of chloramines can best be utilized. The selection of type of treatment is dependent upon the nature of the contaminating organisms, the physical details of the treatment plant and the water distribution, and upon the nature of the process in which these treated waters are to be used.

Mr. Carter has inquired for a more rapid means by which it may be determined that the slime-prevention treatment is under proper control. Coincidentally with presentation of the original paper in December, 1937, experiments were undertaken and developments are now well under way, which involve electrometric means of analysis of water and slime-forming tendencies. It is proposed to report these developments more fully at a later date.

Mr. Bailey has called attention to an error on the part of the author, in failing through oversight to include in the paper presented in December, 1937, details of the Chicago experiments, a report of which had appeared in 1926 in the Transactions of the National Electric Light Association. This omission was corrected in an expanded version of the paper, which has received widespread distribution in reprint form. The author welcomes this opportunity to make a further apology.

Mr. Maurer's discussion is based on limited experience, and it is readily understandable that conclusions such as expressed might be reached without having a sufficiently broad background to permit of judging the relative merits of chlorine and of chloramines for a specific application. These conclusions fail to take into consideration basic differences in results obtained by these sterilization methods, as they may be made use of in drinking-water treatments, in the general industrial-process and paper-mill water-treatment field, as against the more specific applications in the condenser-cooling-water treatment.

Careful reading and unclouded interpretation of the original paper will serve to answer the questions raised. As experience is extended, the observations made in the paper will be confirmed, and a better understanding will be obtained of the various factors which must be taken into consideration in deciding in what types of applications the continuous or intermittent uses of chlorine or chloramines may prove to be most advantageous.