Transactions

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A.S.M.E.

Steam-Boiler Performance and a Method of Comparison . . . . . . *E. G. Bailey* 367

The Locomotive Boiler . . . . *C. A. Brandt* 379

Design Factors Controlling the Dynamic Performance of Instruments . . . . *C. S. Draper and G. P. Bentley* 421

Thermostatic Bimetals . . . . *S. G. Eskin and J. R. Fritze* 433

The Significance of, and Suggested Limits for, the Stress in Pipe Lines Due to the Combined Effects of Pressure and Expansion . . . *D. B. Rossheim and A. R. C. Markl* 443

Properties and Performance of Plastic Bearing Materials . . . . *L. M. Tichvinsky* 461

Neoprene as a Spring Material . . . . *F. L. Yerzley* 469

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Steam-Boiler Performance and a Method of Comparison

By E. G. BAILEY,[1] NEW YORK, N. Y.

This paper presents data from two methods of determining rates of heat absorption in different sections of steam-boiler units. One by direct measurement of evaporation, first published in France in 1874, describing French locomotive-boiler tests, is compared with recent data on modern furnaces using the gas-temperature-weight method. A method of comparing data from all types, kinds, and sizes of boiler units at diverse rates of output is presented. It consists briefly of plotting "heat absorbed" in per cent of total heat available against "equivalent surface area," which results in all points, having the same Btu available per sq ft per hr, being on the same ordinate. Representative data from modern stationary and marine boilers are compared, together with the data from the French locomotive-boiler tests of 65 years' standing. Data on volumes of furnaces and entire boiler units are given together with draft losses and efficiencies.

THE three branches of steam engineering—locomotive, marine, and stationary—have developed their individual terms with respect to rates of boiler output and performance. Some of these terms are becoming less significant due to varieties of fuel, the addition of superheaters, economizers, air heaters, etc. However, there are common bases for comparison which can and should be used in ways that will be universally applicable to all forms of fuel-burning and heat-absorbing equipment. One of the principal purposes of this paper is to present a method which is believed to be of value, not only in comparing boilers of different types, but also in comparing more accurately the relative results from similar boilers.

As a basis for comparison with the modern locomotive boiler, which is being covered in another paper from the A.S.M.E. Railroad Division,[2] data are given in this paper on the performance of present-day stationary boilers. Because of the great diversity of boiler types, a really comprehensive treatment of the subject has not been possible within the space limitations. However, it is believed that the most value can be obtained by giving performance data from a few representative types of modern stationary boilers, and presenting them in a form which may be readily compared with similar data from locomotive or other boilers.

Coal-burning locomotives are fired by hand or stoker and none greatly exceeds an output of 100,000 lb of steam per hr. Many locomotives in the South and West are oil-fired. In order to make the results more directly comparable, the stationary-boiler data presented herewith are taken from boilers of a smaller size than is normally used in central-station practice and with working pressures approaching those now used in locomotives.

In addition to hand, stoker, and oil firing, one unit burning pulverized coal is included, since many stationary boilers are now designed for burning pulverized coal, oil, or gas without any change in the equipment. Some marine-boiler performance data are also given to show wherein conditions aboard ship have influenced the design and performance requirements.

While boiler efficiency is the final basis of comparison, with a complete heat balance to show the distribution of heat losses in unburned fuel, stack gases, excess air, etc., it is of further importance to know the distribution of heat absorbed by the boiler, superheater, economizer, and air heater and, from the surfaces of each, to know the average rates of heat absorption of these component parts. The absorption of radiant heat in the furnace, the effect of slag, cleanliness of heating surfaces, and the absorption of heat in different parts of the boiler sections are also factors of importance in connection with the design of furnaces and boilers, as well as in the operation of the fuel-burning and steam-generating equipment. The latter information is not obtainable from present standard tests.

Methods of Measuring Heat Absorption

An interesting study of the rates of heat absorption in different sections of the firebox and boiler tubes of a small locomotive boiler was made between 1860 and 1864 by the Northern Railway of France,[3] where a boiler divided into five sections was built and the evaporation was determined separately in each section.

Another method of determining the rate of heat absorption in different sections of furnaces and boilers was described by the author in a previous paper[4] before the Society. This method determines heat absorption by measuring the furnace gas temperatures and gas weights at consecutive stages in the travel of the gases, the difference in the heat content of the gases between any two points in the gas travel being the heat absorbed by radiation and convection by any surface between these points.

Results From Heat-Absorption Tests

A diagram of the French locomotive is shown in Fig. 1.[5] The firebox was 3 ft square at the grate, and there were 125 tubes about $1^7/_8$ in. diam and 12 ft 4 in. long. The five divisions of the boiler were arranged for continuous gas flow and separate water levels were maintained. The evaporation from each section was determined by a number of tests at different ratings. Three tests at the highest draft used have been selected for presentation and the essential data are given in Table 1. Tests A-1 and A-2 were made with briquettes and test A-3 with coke. Test A-2 was run with half of the tubes plugged and the total evaporation was about 87 per cent of that obtained in test A-1 or at about 62

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[1] Vice-President, The Babcock & Wilcox Company. Mem. A.S.M.E.

[2] "The Locomotive Boiler," by C. A. Brandt, Trans. A.S.M.E., vol. 62, July, 1940, pp. 379–419.

Contributed by the Power Division and presented at the Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

[3] "The Practical Physics of the Modern Steam Boiler," by F. J. Rowan, D. Van Nostrand Co., New York, 1903, p. 143.

"Permanent Way Rolling Stock and Technical Working of Railways," by C. H. F. Couche (translated from the French by J. N. Shoolbred and J. E. Wilson), vol. 3, Dulau and Co., London, 1878.

"Vaporisation Decroissante en Progression Geometrique," by Paul Havrez, *Annales du Gênie Civil*, vol. 3, 1874, pp. 520–528 and 545–564.

[4] "Modern Boiler Furnaces," by E. G. Bailey, Trans. A.S.M.E., vol. 61, Oct., 1939, pp. 561–569. Discussion, pp. 569–576.

[5] Reproduced from reference (3), by F. J. Rowan, Fig. 69.

TABLE 1 TEST DATA ON FIVE SECTIONS OF FRENCH LOCOMOTIVE BOILER

		A-1	A-2	A-3
1	TEST	A-1	A-2	A-3
2	BOILER		FIG 1	
3	RATED STEAM OUTPUT - LB./HR.			
4	STEAM OUTPUT ON TEST - LB./HR.	7934	6886	5830
5	STEAM PRESSURE - LB./SQ.IN. GA.	80	80	80
6	SATURATION TEMPERATURE - °F	324	324	324
7	STEAM TEMPERATURE SUPERHEATER OUTLET - °F	SATURATION		
8	FEED WATER TEMPERATURE - °F	60	60	60
9	AIR TEMPERATURE LEAVING AIR HEATER - °F	–	–	–
10	METHOD OF FIRING	HAND FIRED GRATE		
11	KIND OF FUEL	COAL BRIQUETTES		COKE
12	GRATE AREA OR FURNACE CROSS SECTION - SQ.FT.	9	9	9
13	LB. OF FUEL PER HR. DIVIDED BY ITEM 12	108.7	94.3	85.7
14	HEATING VALUE OF FUEL "AS FIRED" B.T.U./LB. ①	13000	13000	12000
27	TOTAL HEAT AVAILABLE ABOVE 80°F - MILLION B.T.U./HR. ①	11.69	10.14	8.34
28	LIBERATION - ITEM 27 DIVIDED BY ITEM 39 - B.T.U./CU.FT.HR.	354000	307000	253000
29	ITEM 27 DIVIDED BY ITEM 41 - B.T.U./SQ.FT. HR.			
	FIREBOX	152200	153700	108700
	1ST. DIVISION	45700	65300	32600
	2ND. "	26900	41400	19170
	3RD. "	19030	30300	13800
	4TH. "	14750	23900	10530
30	GAS TEMPERATURE LEAVING UNIT - °F	373	373	373
35	HEAT ABSORBED - CUMULATIVE DIVIDED BY ITEM 27 - %			
	FIREBOX	29.5	34.9	25.1
	1ST. DIVISION	53.6	56.3	51.5
	2ND. "	65.9	67.3	65.6
	3RD. "	73.5	73.9	74.1
	4TH. "	78.5	78.6	81.0
39	FURNACE VOLUME - CU. FT.	33	33	33
41	SURFACES - CUMULATIVE - SQ.FT.			
	FIREBOX	76.8	65.9	76.8
	1ST. DIVISION	256.0	155.4	256.0
	2ND. "	435.2	244.9	435.2
	3RD. "	614.4	334.4	614.4
	4TH. "	793.6	423.9	793.6
42	EQUIVALENT SURFACE AREA ($500 \times 10^6 \div$ ITEM 27) × ITEM 41 - SQ.FT.			
	FIREBOX	3285	3240	4610
	1ST. DIVISION	10950	7660	15330
	2ND. "	18670	12070	26100
	3RD. "	26300	16490	36850
	4TH. "	33950	20850	47600
43	VOLUME OF ENTIRE UNIT - CU. FT.	320	320	320
44	ITEM 27 DIVIDED BY ITEM 43 - B.T.U./CU.FT.HR.	36500	31700	26100
45	TOTAL DRAFT LOSS - INCHES OF WATER	3.94	3.94	3.94
47	OVERALL EFFICIENCY - %	72.2	72.3	72.9

① ESTIMATED BY ORROK - COMBUSTION APRIL - 1938

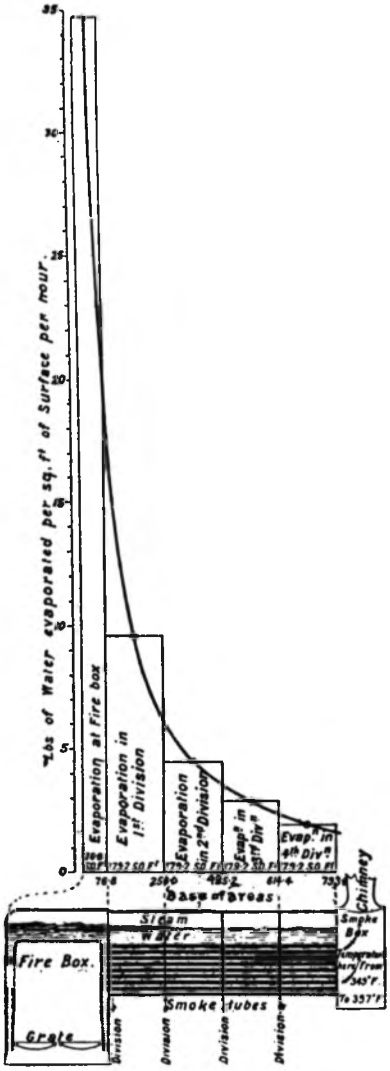

FIG. 1 DIAGRAMMATIC ARRANGEMENT OF LOCOMOTIVE BOILER (NORTHERN RAILWAY OF FRANCE), SHOWING DIVISIONS OF HEATING SURFACE AND RELATIVE EVAPORATION FROM EACH DIVISION

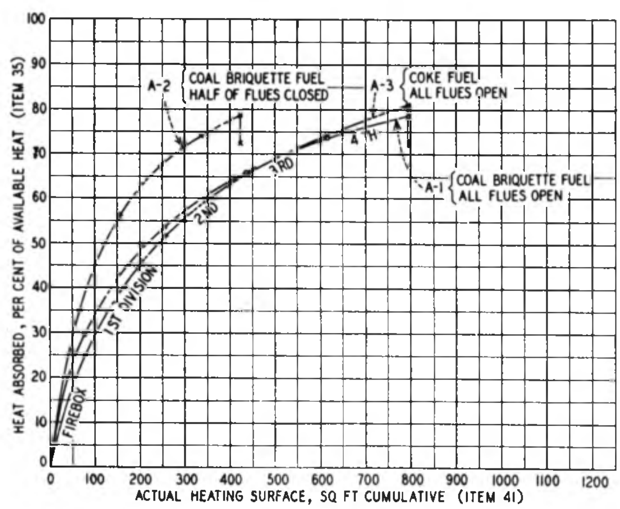

FIG. 2 HEAT ABSORPTION IN FRENCH LOCOMOTIVE BOILER PLOTTED AGAINST ACTUAL HEATING SURFACE

(Boiler of Fig. 1 and Table 1; tests A-1, A-2, and A-3.)

per cent higher rate on a basis of heat available per sq ft of active heating surface. With the same draft the coke gave a lower rate of evaporation. The heat absorbed, expressed as a percentage of the heat available for absorption, item 35, Table 1, is plotted in Fig. 2 against cumulative actual heat-absorbing surfaces, item 41. No useful comparison of these fuels and rates of output can be readily obtained from this plot because there is no way in which the rate of heat available or rate of heat absorption can be graphically indicated.

Tests B-1 and B-2 from Table 2 were obtained from the pulverized-coal-fired boiler of Fig. 3, rated at 100,000 lb of steam per hr. These tests are plotted in Fig. 4, which repeats test A-1, but to a scale suitable for plotting the B series. In this boiler the ash is removed in a dry form by blowing it from the flat water-cooled floor to the hopper at the end away from the burners. A certain amount of dry sponge ash accumulates on the water-cooled furnace walls and roof, which are not normally cleaned; the ash reaches an equilibrium by falling by its own weight from time to time in small amounts. The heat absorptions in the furnace and the different sections of the boiler tube banks, item 32, were obtained by the "gas-temperature-weight" method previously referred to, a brief detailed description of which may be in order. Test data are taken which include items 4 to 27, inclusive, of Table 2, the principal purpose being to determine the total heat input, item 24; the total heat available, item 27; and the total gas weight, item 20.

The total heat available, item 27, is the total heat from the fuel actually burned, plus the heat in the preheated air above 80 F, minus the latent heat in the moisture from the fuel and from the burning of hydrogen. In other words, it is the sensible heat in the gases of combustion which is available for absorption by radiation and convection by any heating surface over which it may pass.

Another very important determination is item 30, wherein the average true gas temperature is obtained at different points in the path of gases by means of the high-velocity thermocouple.[6] From these temperatures and the gas weights, the total heat absorbed in each section, item 32, is determined.

Item 36, Btu absorption per sq ft per hr for the different sections is also very interesting. The actual furnace heating surface is taken as the projected area or envelope of all water-cooled furnace areas, with no factors for cleanliness or different kinds of cooling surfaces except in the case of spaced tubes with refractory behind or between, where Hottel factors[7] are applied. Boiler, superheater, economizer, and air-heater heating surfaces are taken as the full tube circumference, plus any other exposed surfaces in contact with the actively flowing gases of combustion. No factors for tube arrangement, spacing, or mass flow are used because it is thought best to keep all units on an actual area and Btu available basis and to study the comparative results for the effect actually resulting from different forms of surface, fuel and ash characteristics, values of excess air and mass flow, etc., as determined from test data.

Many other data and calculations are included in the complete test, but only the essential ones pertinent to this paper are included in Table 2.

[6] "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.

[7] "Radiant Heat Transmission Between Surfaces Separated by Nonabsorbing Media," by H. C. Hottel, Trans. A.S.M.E., vol. 53, 1931, FSP-53-19b, pp. 265–271. Discussion, pp. 271–273.

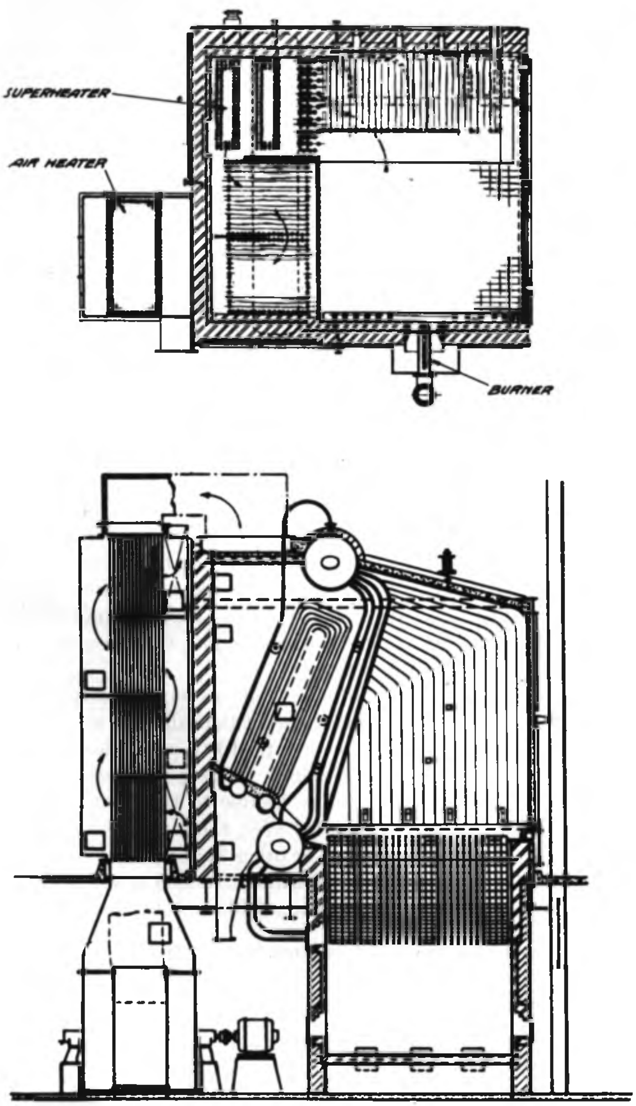

FIG. 3 PULVERIZED-COAL-FIRED BOILER FOR STATIONARY SERVICE WITH INTEGRAL WATER-COOLED FURNACE
(Data given in Table 2.)

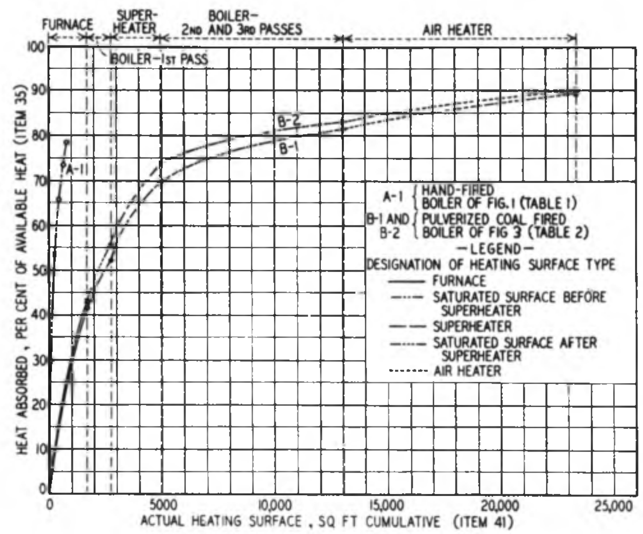

FIG. 4 HEAT ABSORPTION IN FRENCH LOCOMOTIVE BOILER AND IN PULVERIZED-COAL-FIRED BOILER PLOTTED AGAINST ACTUAL HEATING SURFACE
(Boiler of Fig. 3 and Table 2; tests B-1 and B-2, also comparative plot of curve A-1 from Fig. 2.)

During test B-1, the boiler produced 105,900 lb of steam per hr and, during test B-2, 83,300 lb per hr. The higher rating test shows a lower percentage of heat absorbed, but there is no indication from the plots themselves, as given in Fig. 4, as to the relative rates of output or the relative performance. Test A-1 plotted on the same scale of actual surface is still further from being on any useful basis of comparison with the other two curves in Fig. 4, and yet all three curves are plotted on basic units of great significance; viz., actual heating surface and per cent of available heat absorbed. Neither does one obtain any really satisfactory comparison of the relative results from these boilers at the different rates of output by examining all test data relating to them in Tables 1 and 2.

COMPARISON ON EQUIVALENT-SURFACE-AREA BASIS

In Fig. 5 the heat-absorption percentages of tests B-1 and B-2 are plotted to "equivalent surface area" instead of actual area. This simple expedient brings into the diagram a rate-of-input factor. The corresponding points of the lower-input test are plotted at higher scale values of equivalent surface areas in order to maintain relationship with tests of greater heat input such that all curve intersections with a given ordinate line shall have the same actual Btu available per sq ft value.

This equivalent surface area is given as item 42 in all tables. It is obtained by using the factor resulting from dividing 500,000,000 by the total Btu available for absorption, item 27, on any test regardless of the size of boiler unit or its rating, and then multiplying this factor by the actual area in sq ft of all cumulative

TABLE 2 TEST DATA ON PULVERIZED-COAL-FIRED AND

1 TEST	B-1					B-2				
2 BOILER			FIG 3					FIG 3		
3 RATED STEAM OUTPUT - LB/HR.			100000					100000		
4 STEAM OUTPUT ON TEST - LB/HR.			105900					83300		
5 STEAM PRESS. AT SUPERHEATER OUTLET - LB/IN² GA.			644					643		
6 SATURATION TEMPERATURE - °F			498					497		
7 STEAM TEMPERATURE AT SUPERHEATER OUTLET - °F			825					796		
8 FEEDWATER TEMPERATURE - °F			399					394		
9 AIR TEMPERATURE LEAVING AIR HEATER - °F			513					497		
10 METHOD OF FIRING			PULVERIZED COAL					PULVERIZED COAL		
11 KIND OF FUEL			SEMI-BITUMINOUS COAL					SEMI-BITUMINOUS COAL		
12 GRATE AREA OR FURNACE CROSS SECTION - SQ.FT.			342					342		
13 LBS. FUEL PER HR. DIVIDED BY ITEM 12			27.8					21.3		
14 HEATING VALUE OF FUEL "AS FIRED" - B.T.U./LB.			13950					13950		
15 SECTION OF UNIT	FURNACE	1ST. BOILER BANK	SUPERHEATER	2ND. & 3RD. BOILER BANK	AIR HEATER	FURNACE	1ST. BOILER BANK	SUPERHEATER	2ND. & 3RD. BOILER BANK	AIR HEATER
16 FLUE GAS ANALYSIS - LEAVING SECTION: CO_2 % BY VOL.	16.3			12.8	12.9	16.7			13.7	13.7
17 FLUE GAS ANALYSIS - LEAVING SECTION: O_2 % BY VOL.	3.7			7.1	7.2	3.2			6.4	6.4
18 FLUE GAS ANALYSIS - LEAVING SECTION: CO % BY VOL.	0.0			0.0	0.0	0.0			0.0	0.0
19 TOTAL COMBUSTION AIR LEAVING SECTION - %	121	121	121	150	151	117	117	117	143	143
20 TOTAL WEIGHT WET GAS LEAVING SECTION - LB/HR.	130500	130500	130500	159000	160000	97400	97400	97400	117300	117300
21 HEAT AVAILABLE ABOVE 80°F - ENTERING SECTION - BTU/LB GAS	1038	605	495	313	144, Lv - 80	1063	607	473	279	133, Lv - 74
22 MOISTURE BY WEIGHT IN FLUE GAS - %	4.3	4.3	4.3	3.7	3.7	4.3	4.3	4.3	3.8	3.8
23 ADIABATIC TEMPERATURE - °F	3610					3685				
24 TOTAL HEAT INPUT FROM FUEL - MILLION BTU/HR.	132.3					102.0				
25 TOTAL PREHEAT IN AIR ABOVE 80°F - MILLION BTU/HR.	10.6					7.3				
26 TOTAL LOSSES - LATENT HEAT, CARBON LOSS - MILLION BTU/HR.	7.4					5.7				
27 TOTAL HEAT AVAILABLE ABOVE 80°F - MILLION BTU/HR.	135.5					103.6				
28 LIBERATION - ITEM 27 DIVIDED BY ITEM 39 - BTU/CU.FT.HR.	22500					17180				
29 ITEM 27 DIVIDED BY ITEM 41 - BTU/SQ.FT.HR.	79900	49400	27500	10370	5800	61100	37800	21000	7940	4430
30 GAS TEMPERATURE LEAVING SECTION - °F	2280	1900	1268	647	398	2285	1825	1146	604	378
31 HEAT AVAILABLE IN GAS ENTERING - MILLION BTU/HR.	135.5	79.2	64.7	40.9	24.7	103.6	59.1	46.03	27.13	17.40
32 HEAT ABSORBED: INDIVIDUAL SECTION - MILLION BTU/HR.	56.3	14.5	23.8	16.2	10.6	44.5	13.07	18.90	9.73	7.30
33 HEAT ABSORBED: CUMULATIVE - MILLION BTU/HR.	56.3	70.8	94.6	110.8	121.4	44.5	57.57	76.47	86.20	93.50
34 HEAT ABSORBED: INDIVIDUAL SECTION DIVIDED BY ITEM 27 - %	41.6	10.7	17.5	11.9	7.8	43.0	12.6	18.2	9.4	7.1
35 HEAT ABSORBED: CUMULATIVE DIVIDED BY ITEM 27 - %	41.6	52.3	69.8	81.7	89.5	43.0	55.6	73.8	83.2	90.3
36 HEAT ABSORPTION: INDIVIDUAL SECTION - BTU/SQ.FT.HR.	33200	13900	10850	1995	1026	26200	12500	8610	1198	707
37 HEAT ABSORPTION CUMULATIVE - BTU/SQ.FT.HR.	33200	25900	19200	8480	5190	26200	21000	15500	6600	4000
38 EQUIVALENT HEAT RECEIVING SURFACE TEMPERATURE °F	2010	1960	1362			2080	1932	1280		
39 FURNACE VOLUME - CU.FT.	6030					6030				
40 SURFACES: INDIVIDUAL SECTION - SQ.FT.	1695	1043	2197	8118	10320	1695	1043	2197	8118	10320
41 SURFACES: CUMULATIVE - SQ.FT.	1695	2738	4935	13053	23373	1695	2738	4935	13053	23373
42 EQUIVALENT SURFACE AREA (500×10^6 ÷ ITEM 27) × ITEM 41 - SQ.FT.	6250	10100	18170	48200	86200	8190	13200	23800	63000	112600
43 VOLUME OF ENTIRE UNIT - CU.FT.			26000					26000		
44 ITEM 27 DIVIDED BY ITEM 43 - BTU/CU.FT.HR.			5220					3980		
45 DRAFT LOSS - IN. OF WATER	TOTAL TO BOILER OUTLET = 0.83				A.H. = 1.68	TOTAL TO BOILER OUTLET = 0.50				A.H. = 1.31
46 TOTAL DRAFT PLUS AIR PRESSURE - IN. OF WATER					6.6					3.81
47 OVERALL EFFICIENCY - %					83.7					84.4

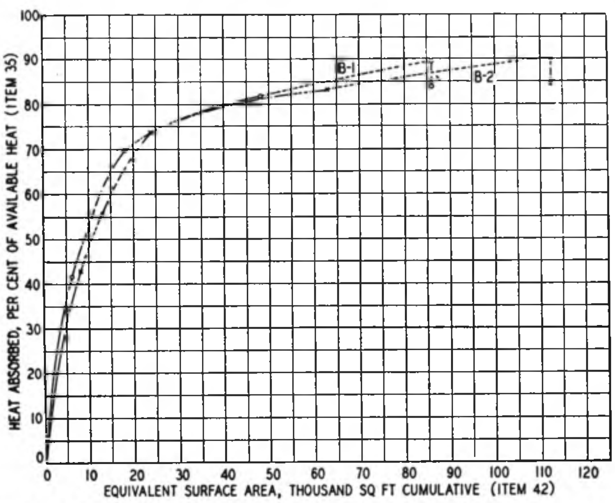

Fig. 5 Heat Absorption Plotted Against Equivalent Surface Area

(Boiler of Fig. 3 and Table 2; tests B-1 and B-2. For legend see Fig. 10.)

surface, item 41, under consideration, including the water-cooled portion of the furnace, and each successive additional portion of heating surface to include boiler tube bank, superheater, economizer, and air heater if present. Each cumulative actual area, item 41, should be multiplied by this factor and plotted as equivalent surface area, item 42, against the total percentage of heat absorbed, item 35, by the corresponding actual surface area.

The arbitrary value of 500,000,000 Btu available per hr was taken after much consideration. It is a nominal value corresponding to many central-station boiler units. Some are much larger and many are smaller. It really makes no essential difference so long as the same unit is used as a standard for all boilers to be compared. It is hoped that others applying this basis of comparison will use this same standard for the sake of uniformity. A unit actually operating at 500,000,000 Btu available input per hr will be plotted with the actual surfaces equal to the equivalent surfaces because the factor is unity.

When comparing tests at different ratings on the same boiler, or any tests on any boiler, it is apparent that all cases at a given value of equivalent surface have identical Btu available per sq ft of the total surface up to that point. If the percentage of heat absorbed is not the same, it may be due to the cleanliness or effectiveness of the surfaces passed over, the temperature, luminosity, or mass flow of the gases or other factors, some of which may not be known or predictable without further study or tests. One may easily determine the Btu available and the Btu absorbed per sq ft per hr for any point on any curve, viz., $\frac{500{,}000{,}000}{\text{Equiv surface}}$ = Btu available per sq ft per hr. This result multiplied by the per cent absorbed gives the average Btu absorbed per sq ft per hr for all surface cumulatively from the fuel bed or burner up to the point selected on the curve. For illustration test B-1, Fig. 5, the entrance to the superheater lies on approximately 10,000 equivalent surface, and therefore corresponds to 50,000 Btu available per sq ft per hr. The percentage absorbed is plotted at 52.3, hence the average rate of heat absorption up to this point is

ON OIL-FIRED STATIONARY BOILERS

C-1					C-2				
	SIMILAR TO FIG 3					SIMILAR TO FIG 3			
		160000					160000		
		156000					103000		
		459					451		
		463					461		
		568					555		
		221					218		
		482					438		
MECHANICAL ATOMIZING OIL BURNER					STEAM MECHANICAL ATOMIZING OIL BURNER				
		OIL					OIL		
		425					425		
		26.6					17.0		
		18020					18020		
FURNACE	1ST. BOILER BANK	SUPERHEATER	2ND & 3RD BOILER BANK	AIR HEATER	FURNACE	1ST. BOILER BANK	SUPERHEATER	2ND & 3RD BOILER BANK	AIR HEATER
13.9			13.1	12.8	14.0			13.2	12.9
4.3			5.4	5.6	4.0			5.1	5.1
0.0			0.0	0.0	0.0			0.0	0.0
125	125	125	134	135	123	123	123	131	131
201000	201000	201000	214000	216000	127500	127500	127500	135000	135000
1034	583	448	386	163.Lr-84	1036	515	380	316	134.Lr-62
5.8	5.8	5.8	5.4	5.4	6.1	6.1	6.1	5.8	5.8
3549					3545				
204.0					130.0				
17.4					10.1				
13.4					8.1				
208.0					132.0				
27800					17600				
127200	57400	48100	14380	6720	80800	36300	30500	9120	4270
2175	1725	1510	707	409	1940	1485	1260	600	325
208.0	117.5	90.5	78.2	37.7	132	65.7	48.5	40.34	20.0
90.5	27.0	12.3	40.5	17.4	66.3	17.2	8.16	20.34	10.1
90.5	117.5	129.8	170.3	187.7	66.3	83.5	91.66	112.0	122.1
43.5	13.0	5.9	19.5	8.4	50.2	13.0	6.2	15.4	7.7
43.5	56.5	62.4	81.9	90.3	50.2	63.2	69.4	84.8	92.5
55400	13500	17850	3990	1054	40600	8600	11820	2005	612
55400	32300	30000	11780	6070	40600	23000	21200	7750	3940
1541	1795	1233			1300	1580	990		
7494					7494				
1633	2000	690	10142	16500	1633	2000	690	10142	16500
1633	3633	4323	14465	30965	1633	3633	4323	14465	30965
3920	8740	10390	34780	74400	6180	13780	16400	54800	117300
		29170					29170		
		7140					4530		
TOTAL TO BOILER OUTLET = 1.35				A.H. = 3.14	TOTAL TO BOILER OUTLET = 0.52				A.H. = 1.13
				14.6					6.5
				83.4					86.2

point including all surface of the unit up to the entrance of the air heater plus about 46 per cent of the area of the air heater. However, as the gases pass completely through the air heater, the heat absorbed increases to 90.3 per cent, but the equivalent surface area is 112,600 sq ft, although the actual heating surface is the same as on the other test. At the same actual stages in the boiler unit, test B-2 shows higher percentages of heat absorption than the higher rating test B-1. This difference increases from 1.4 per cent leaving the furnace to 3.3 per cent entering the superheater and 4 per cent leaving the superheater, and then diminishes to 1.5 per cent entering the air heater and 0.8 per cent leaving the air heater. These are all taken from item 35 by subtracting the corresponding percentages from the two tests. This indicates that the first boiler tube bank and the superheater are more responsive to increased heat absorption due to increased heat input because of the temperature and mass-flow factors. The adiabatic or theoretical maximum combustion temperature of test B-2 is higher than on test B-1, hence this will account for some of the difference in heat absorbed in the furnace.

The total heat absorbed by the B-1 and B-2 tests is about 90 per cent and may seem unduly high, but it must be remembered that this is on a Btu available basis, which is more nearly like the net or low heating value often used abroad. The over-all efficiency of the unit by the A.S.M.E. method is shown as item 47 and plotted on the curves as a drop point under the last point on the available basis.

Comparison of Results and Boilers

The French locomotive tests are plotted on the equivalent-surface-area basis in Fig. 6. One cannot help but grasp immediately a realization of the difference between tests A-1 and A-2 due to the higher mass flow and better heat absorption resulting from plugging half of the tubes. There may also have been a lower excess air and higher adiabatic temperature in the firebox of test A-1, as it indicates a higher percentage of heat absorbed in the furnace, contrary to the relationship at the corresponding points of Fig. 5.

26,150 Btu per sq ft per hr. The slope of the curve at any individual section is a graphic index of the rate of heat absorption in that section.

In Fig. 5, test B-1 shows 89.5 per cent of the available heat absorbed up to the outlet of the air heater which has an equivalent surface area of 86,200 sq ft. Test B-2 shows only 86.8 per cent heat absorbed at the same equivalent area; this corresponds to a

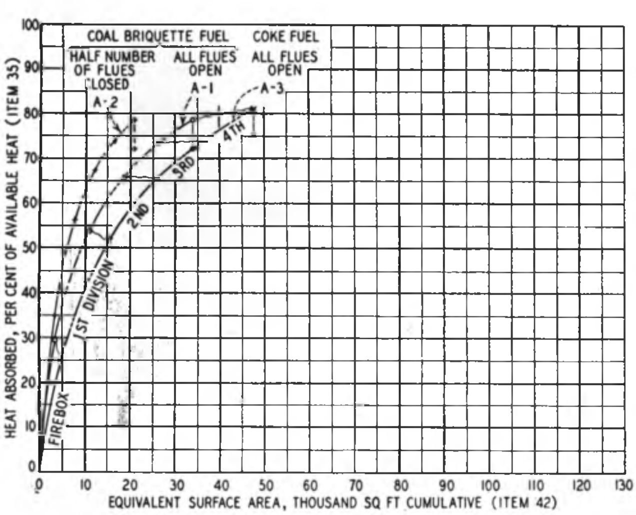

Fig. 6 Heat Absorption Plotted Against Equivalent Surface Area
(Boiler of Fig. 1 and Table 1; tests A-1, A-2, and A-3.)

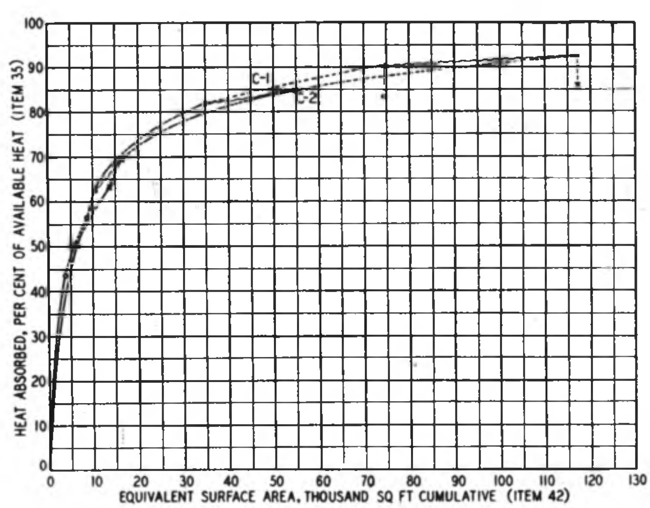

Fig. 7 Heat Absorption Plotted Against Equivalent Surface Area; Oil-Fired Boiler
(Boiler similar to Fig. 3, data given in Table 2; tests C-1 and C-2. For legend see Fig. 10.)

TABLE 3 DATA FROM ELEVEN BOILER TESTS MADE ACCORDING TO

1 TEST	D-1		D-2		E-1			E-2		
2 BOILER	SIMILAR TO FIG 8				FIG 8			FIG 8		
3 RATED STEAM OUTPUT - LB/HR.		22000		22000		13000			13000	
4 STEAM OUTPUT ON TEST - LB/HR.		23200		18000		10980			9970	
5 STEAM PRESSURE AT SUPERHEATER OUTLET - LB/SQ.IN.GA		600		600		401			395	
6 SATURATION TEMPERATURE - °F		494		492		449			449	
7 STEAM TEMPERATURE AT SUPERHEATER OUTLET - °F		872		837		766			750	
8 FEEDWATER TEMPERATURE - °F		350		350		312			313	
9 AIR TEMPERATURE LEAVING AIR HEATER - °F		391		386		305			330	
10 METHOD OF FIRING	MECHANICAL ATOMIZING OIL BURNER				STOKER			STOKER		
11 KIND OF FUEL		OIL		OIL	BITUMINOUS COAL			BITUMINOUS COAL		
12 GRATE AREA OR FURNACE CROSS SECTION - SQ.FT.		57.8		57.8		50			50	
13 LB. OF FUEL PER HOUR, DIVIDED BY ITEM 12		28.4		21.9		23			21	
14 HEATING VALUE OF FUEL "AS FIRED" - BTU/LB.		18400		18400		12127			12288	
15 SECTION OF UNIT	FURN. INC. BLR.	AIR HEATER	FURN. INC. BLR.	AIR HEATER	FURN. INC. BLR.	ECONOMIZER	AIR HEATER	FURN. INC. BLR.	ECONOMIZER	AIR HEATER
16 FLUE GAS ANALYSIS - LEAVING SECTION: CO_2 % BY VOL.	12.8		13.1				12.3			11.7
17 FLUE GAS ANALYSIS - LEAVING SECTION: O_2 " " "	4.4		4.7				7.2			7.2
18 FLUE GAS ANALYSIS - LEAVING SECTION: CO " " "	0.0		0.0				0.0			0.0
19 TOTAL COMBUSTION AIR LEAVING SECTION - %	125		128				150			150
20 TOTAL WEIGHT WET GAS LEAVING SECTION - LB/HR.	30000		23700							
21 HEAT AVAILABLE ABOVE 80°F ENTERING SECTION - BTU/LB. GAS	1092	136.5	1092	130.8						
22 MOISTURE BY WEIGHT IN FLUE GAS - %	6.75		6.75							
23 ADIABATIC TEMPERATURE - °F	3680		3680							
24 TOTAL HEAT INPUT FROM FUEL MILLION BTU/HR.	30.25		23.35		14.12			12.75		
25 TOTAL PREHEAT IN AIR ABOVE 80°F " " "	1.96		1.49		0.86			0.92		
26 TOTAL LOSSES, LATENT HEAT, UNBURNED CARBON " " "	1.90		1.49		0.92			0.73		
27 TOTAL HEAT AVAILABLE ABOVE 80°F " " "	30.31		23.35		14.06			12.94		
28 LIBERATION - ITEM 27 DIVIDED BY ITEM 39 - BTU/CU.FT.HR.	70000		53800		45400			41800		
29 ITEM 27 DIVIDED BY ITEM 41 - BTU/SQ.FT.HR	7680	4770	5910	3680	6210	4360	2750	5720	4020	2530
30 GAS TEMPERATURE LEAVING SECTION - °F	605	369	586	346	553	428	297	549	451	304
31										
32 HEAT ABSORBED: INDIVIDUAL SECTION - MILLION BTU/HR.		1.9		1.5		0.85	0.87		0.65	0.92
33 HEAT ABSORBED: CUMULATIVE " " "	26.2	28.1	20.3	21.8	11.4	12.25	13.12	10.35	11.0	11.92
34 HEAT ABSORBED: INDIVIDUAL SECTION DIVIDED BY ITEM 27-%		6.3		6.4		6.0	6.2		5.0	7.1
35 HEAT ABSORBED: CUMULATIVE " " " " %	86.3	92.6	87.0	93.4	81.2	87.2	93.4	79.9	84.9	92.0
36 HEAT ABSORPTION: INDIVIDUAL SECTION - BTU/SQ.FT.HR.		790		623		886	461		678	488
37 HEAT ABSORPTION: CUMULATIVE " " " "	6620	4420	5130	3430	5040	3800	2570	4570	3410	2335
38										
39 FURNACE VOLUME - CU.FT.	433		433		309			309		
40 SURFACES: INDIVIDUAL SECTION - SQ.FT.		2403		2403		959	1885		959	1885
41 SURFACES: CUMULATIVE " "	3951	6354	3951	6354	2263	3222	5107	2263	3222	5107
42 EQUIVALENT SURFACE AREA $(500\times10^6 \div \text{ITEM 27})\times\text{ITEM 41}$ - SQ.FT.	65200	104900	84800	136000	80500	114500	181500	87500	124000	197000
43 VOLUME OF ENTIRE UNIT - CU.FT.		2830		2830		2150			2150	
44 ITEM 27 DIVIDED BY ITEM 43 - BTU/CU.FT.HR.		10729		8250		6530			6020	
45 DRAFT LOSS INCHES OF WATER	0.06	0.24	-0.14	0.30	TOTAL TO OUTLET .05			TOTAL TO OUTLET .05		
46 TOTAL DRAFT LOSS PLUS AIR PRESSURE " " "		5.2		2.90			1.48			1.52
47 OVERALL EFFICIENCY - %		86.3		87.1			86.6			86.1

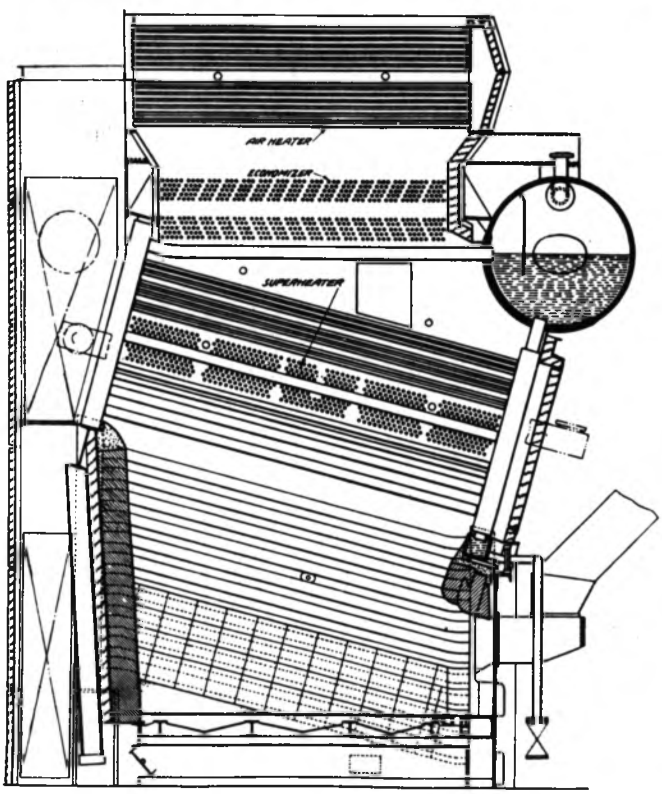

FIG. 8 STOKER-FIRED MARINE BOILER
(Data given in Table 3.)

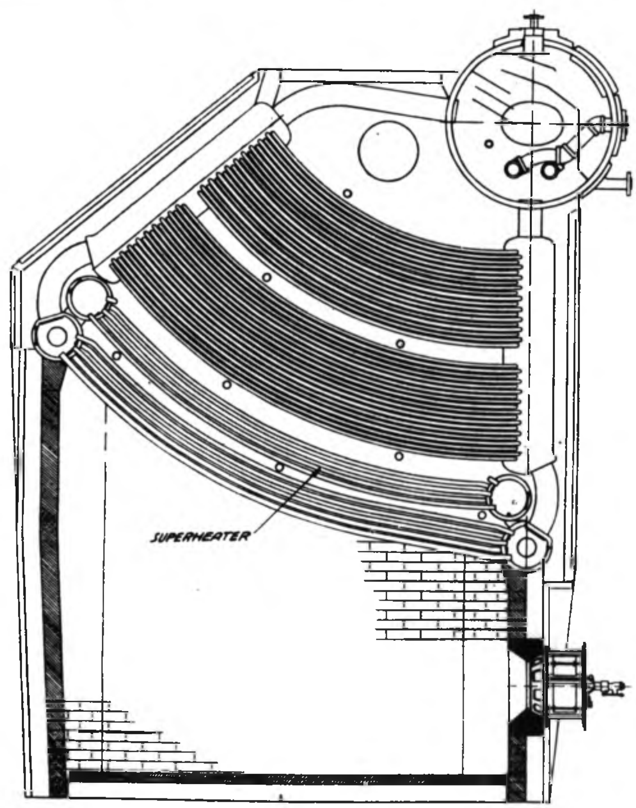

FIG. 9 OIL-FIRED MARINE BOILER
(Data given in Table 3.)

CONVENTIONAL STANDARD TEST METHODS

F-1	F-2	F-3	F-4	G-1		H-1	H-2
FIG 9				BENT TUBE		BENT TUBE	
67000	67000	67000	67000				
66070	50900	34280	17200		16450	27400	19500
309	305	311	301		224	125	121
428	426	426	424		400	354	352
695	679	663	602		536	422	420
211	216	210	211		216	100	94
121	116	113	129		276	74	73
MECHANICAL ATOMIZING OIL BURNER				STOKER		STOKER	
OIL	OIL	OIL	OIL	BITUMINOUS COAL		BITUMINOUS COAL	
71.7	71.7	71.7	71.7		86.5	93.5	93.5
70.3	53.3	36.2	17.5		23.0	37.1	26.9
18625	18625	18500	18625		10756	11883	11795
BOILER & SH				FURN. SH & BLR	AIR HEATER	FURN. SH & BLR	FURN. SH & BLR
12.5	13.3	12.2	10.9		14.2	12.9	13.7
4.1	3.0	4.4	6.3		6.1	6.19	5.2
0.0	0.0	0.0	0.0		0.0	0.02	0.01
123	116	125	141		141	140	132
92200	66100	47600	26100				
960	1007	942	834				
6.3	6.6	6.2	5.8				
3315	3445	3260	2965				
94.1	71.3	47.9	23.4	21.56		41.1	29.6
0.9	0.6	0.3	0.2	0.96			
6.6	5.3	3.4	1.85	1.90		2.8	2.6
88.4	66.6	44.8	21.75	20.62		38.3	27.0
208000	156800	105600	51200				
15800	11900	8020	3880	4550	3050	8500	6000
510	473	445	416				
					0.93		
78.3	59.7	40.2	19.5	18.15	19.08	32.0	22.85
					4.5		
88.6	89.6	89.5	89.6	87.8	92.3	83.4	84.6
					419		
14000	10670	7175	3480	4000	2820	7100	5070
424	424	424	424				
					2220		
5599	5599	5599	5599	4539	6759	4503	4503
31700	42000	62500	129000	110000	164000	58800	83500
2070	2070	2070	2070				
42700	32200	21700	10500				
2.53	1.56	0.98	0.30				
8.52	5.81	3.58	1.47				
83.2	83.8	83.8	83.2		84.1	77.5	77.1

In comparing tests A-1 and A-3, it is noted that, even though the rate of heat available is lower on coke-burning test A-3, the percentage of heat absorbed is less, while normally it would be greater. This again may be due to a lower adiabatic temperature, resulting from either higher excess air or perhaps unburned CO, which might have been very likely when burning coke. Another factor is luminosity, as the coke would have been different from the briquettes. Many desirable data are lacking from these tests.

Tests A-1 and B-1 are shown in Fig. 10 on the equivalent-surface-area basis. It is noted how much more useful is such comparison in contrast to that in Fig. 4, where they are both on the actual area basis.

A boiler similar to that of Fig. 3, but fired with oil, was tested by the gas-temperature-weight method and the essential data are given in Table 2 and plotted in Fig. 7. Characteristics similar to the pulverized-coal-fired tests B-1 and B-2 are noted, i.e., the heat absorbed at the lower rate of input tests C-2 are higher than on C-1 for corresponding locations, but lower on the same equivalent-surface-area basis.

The greater heat absorption in the oil-fired furnace, due largely to cleaner furnace walls, is brought out in Fig. 10, where curves B-1, coal-fired, and C-1, oil-fired, are plotted to the same equivalent-surface-area basis. The greatest difference is shown in the furnace, first boiler tube bank, and superheater. Beyond the superheater, the curves are substantially parallel, indicating about the same rates of heat transfer. The coal-fired boiler is operating at a 14 per cent lower Btu available per sq ft of total surface, and is 0.8 per cent lower in the percentage of available heat absorbed but is 0.3 per cent higher in over-all efficiency.

The French locomotive test A-1 follows the pulverized-coal-fired test B-1 very closely, until the superheater is reached on B-1 where it has a steeper slope and higher percentage rate of heat absorption between the 10,000 and 20,000 equivalent surface area. However, it loses out between the 20,000 and 32,000 equivalent surface area, where evidently the mass flow in the boiler tube bank is not as good as in the last two sections of the French locomotive. It is also noted that the saturation temperature of steam in the French boiler is 324 F, while in B-1 tests the saturation temperature is 498 F, causing an appreciable difference in the rate of heat absorption at these corresponding parts of the units.

The discussion so far has related to two kinds of boilers of three different sizes and with three methods of firing, hand, pulverized coal, and oil. Data from two methods of determining rates of heat absorption in different sections of the heating surface have been compared by a method which seems to have much value.

Table 3 includes data from eleven additional tests, the data being limited to the conventional standard test method. Certain data from these tests are plotted in Fig. 10 to show the percentage of available heat absorbed by the boiler and also by the economizer and air heater separately, where installed, and plotted against equivalent surface areas for comparison with the more

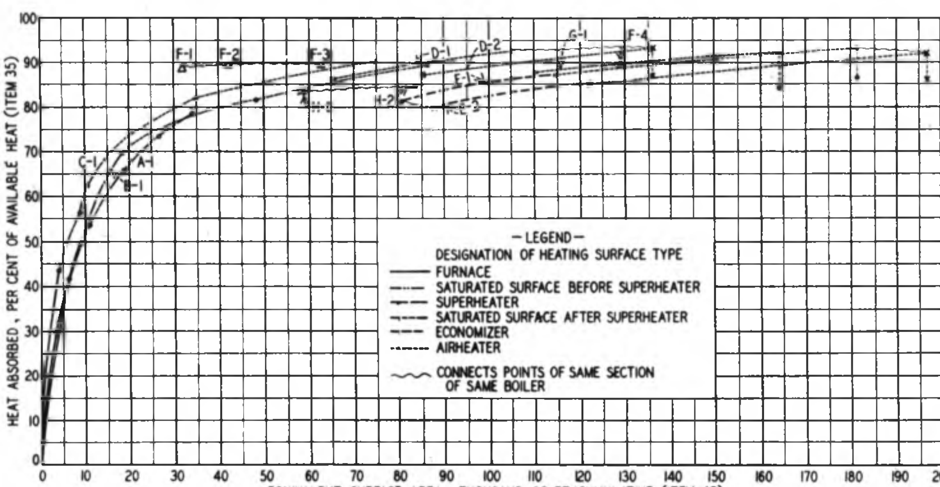

FIG. 10 COMPOSITE GRAPH OF TESTS FROM ALL BOILERS, SHOWING HEAT ABSORPTION PLOTTED AGAINST EQUIVALENT SURFACE AREA
(Data given in Tables 1, 2, and 3.)

complete tests of the A, B, and C series and with each other. This is done because it is believed to be the best known method for obtaining a comprehensive picture of the relative ratings and efficiencies of some representative units in the marine and stationary fields.

Marine Boilers

A typical marine boiler with spreader stoker for coal firing is shown in Fig. 8. It has a rated capacity of 13,000 lb of steam per hr at 400 lb pressure and 750 F. Two tests are given in Table 3 as E-1 and E-2 and are plotted in Fig. 10. While the boiler itself shows relatively low efficiency at lower rates of heat available than others plotted, it is designed with an economizer and air heater to give very good final efficiency.

Tests D-1 and D-2 are from an oil-fired marine boiler similar to Fig. 8, but of 22,000-lb capacity at 600-lb pressure and 850 F steam temperature. It is designed for high efficiency from the boiler itself and has an air heater also. It has less equivalent surface, smaller cubical content, and operates at higher draft loss than the stoker-fired marine boiler, and was operated at lower excess air.

Another marine boiler for Navy use is shown in Fig. 9. Some of the data published by Brierly[8] are reproduced as tests F-1 to F-4 in Table 3. This boiler is rated at 67,000 lb of steam per hr at 600-lb pressure and 700 F, but the tests reported are at 300-lb pressure to be more nearly comparable with present locomotive practice. As no sectional heat absorptions were obtained, only the over-all absorption by the boiler and superheater combined are plotted as triangles in Fig. 10. Even this serves well to show that this high-duty boiler has very good efficiency. It shows about 10 per cent better absorption of available heat than does the French locomotive boiler at the same high relative rating of about 15,000 Btu available per sq ft of total surface per hr. This is undoubtedly due largely to the better arrangement of heating surface and gas-mass flow as indicated by the higher draft loss. It is significant to note the uniformly high efficiency at the wide range of outputs. Tests on the boiler at 600-lb steam pressure show about $1^1/_2$ per cent lower efficiency than those given for 300 lb pressure, due to the difference in the saturation temperature of the heat-receiving surface.

Tests G and H are presented as representative of small stoker-fired stationary boilers of different types, design not shown. Test G-1 has an air heater which shows a comparable rate of heat absorption with other air heaters within the same available heat-rate range.

Comparison of Size, Rating, and Draft Loss

Several items which may be of interest in comparison with locomotive boilers or others are given in the tables but are not specifically discussed in this paper. Coal burned per sq ft of grate is given and, in order to enable a comparison with oil and pulverized fuel to be drawn, the equivalent cross-sectional areas of the respective furnaces are given and also the lb fuel per hr divided by this area.

Btu per cu ft of furnace volume is also given, but it should be remembered that this is a very misleading factor when comparing furnaces of widely different size, even when burning the same kind of fuel. The total cubical contents of the boiler, including the outside dimensions of the casing, economizer, and air heater, may be of interest to some.

Higher draft loss results from higher gas velocities, which are necessary if saving in space and the increasing of rates of heat absorption by convection are to be attained. All arrangements of heating surface are not equally effective, and no conclusions are now attempted from the small amount of data given.

Summary

1 It is very desirable to know the actual rates of heat absorption in different elements of a boiler unit for the purpose of comparing effects of different rates of operation, different values of excess air, kinds of fuel, ash and slag behavior, and kinds of heat-absorbing surface.

2 The direct method of measuring heat absorption by evaporation calls for segregation of heat-absorbing surface which is either very difficult or entirely impossible, due to boiler construction and interference with circulation. The engineers of the Northern Railway of France deserve great credit for having done a very good piece of test work which checks very well with later work by other methods. The heat absorbed in some parts of a boiler unit may readily be determined by the direct method, such as in the case of superheaters, some economizers, and air heaters.

3 A second method is possible, wherein the heat absorbed in a section of the boiler unit, such as the furnace or a tube bank, results from calculation using average gas temperatures and gas weights entering and leaving the section, the difference, allowing for any combustion of fuel taking place within the zone being measured, being the heat absorbed.

4 Data obtained by either method is useful to engineers experienced in calculating boiler performance, but others often find it difficult to grasp an accurate and complete realization of the relative importance and comparisons of different parts of a boiler unit or one with another. A simple method of comparison is presented which consists of plotting a curve showing the relation between the heat absorbed, expressed in per cent of heat available, and the equivalent surface area wherein the heat was absorbed.

5 The data presented have been restricted to relatively small stationary boilers which would be comparable with representative marine boilers, two of which are given, and modern locomotive boilers, data from which are available from another paper.[2] These limited data are not sufficient to draw many conclusions regarding the factors of heat transfer by radiation and convection; however, some well-known facts are clearly demonstrated as being of outstanding significance when presented in this graphic form:

(a) Radiant heat produces high rates of heat transfer, as shown by the steep slope of the curve.

(b) Excess air and adiabatic temperature affect radiant heat greatly. This point is not fully brought out in this paper, because for the purpose of simplicity, furnace tests were selected with reasonably uniform adiabatic temperatures.

(c) High mass flow and high mean temperature difference increase rates of heat transfer. Draft loss, and fan power or its equivalent, are necessary to produce high rates of heat absorption by convection, and high efficiencies with limited surface areas.

(d) Surface is the principal requisite for heat absorption. It should be clean for high rates of heat absorption, but sometimes a covering of ash, slag, or other means is a desirable aid toward complete combustion and for protection against too high a rate of heat absorption and to pass higher temperatures on to other surfaces such as superheaters, which will bring the total absorption to a more economical result.

(e) Terms like Btu per cu ft per hr of furnace volume, Btu per front ft of furnace width, etc., should be used with caution, because they are not as fundamental in their importance as are some other factors. Pounds of oil per sq ft of

[8] "The Babcock & Wilcox High-Pressure Sectional Express Boiler," by Lieut-Com. R. C. Brierly, U.S.N.R., *Journal of the American Society of Naval Engineers*, vol. 43, no. 4, Nov., 1931, pp. 511–561.

boiler is a basically sound standard, much used in marine engineering, but it becomes less useful with the further use of furnace cooling, economizers, and air heaters. Btu available per sq ft per hr seems to be fundamentally sound, but it needs definition as to its application to furnace alone or to the entire unit. Item 29, Table 2, gives this factor for each cumulative section of the boiler unit. It seems now that this value, including all surface up to the entrance of the superheater, is likely to become more used. It also is useful when applied to the furnace alone.

6 Temperature of gases as measured at different points in the gas path is of great importance regardless of whether these temperatures are used in conjunction with gas weights to obtain heat absorption. A knowledge of temperatures at various points of the entire unit is essential in design and is useful in operation in connection with the behavior of ash and slag. The gas temperature might well be plotted to the equivalent surface area for comparison, but the excess air and air leakage at different points interfere with the same usefulness as is obtained from the heat-absorbed basis herein presented, hence in the interest of brevity for this paper such plots are omitted. Gas temperatures are given in the tables as item 30, and the adiabatic temperature as item 23. The adiabatic or theoretical temperature is useful as an index of the combined effect of excess air, moisture in fuel, and completeness of combustion.

7 Heat absorbed and equivalent surface are undoubtedly the most important basic factors for comparison in studying boiler performance.

Acknowledgment

Acknowledgment is made for valuable data and assistance from stoker manufacturers and from the author's associates.

Discussion

J. C. Hobbs.[9] Mr. Bailey's use of the French tests illustrates clearly the relations between cause and effect. Each part of the steam-generating equipment being set up independently, there can be no question about the heat transfer and the work done.

The curves show clearly not only the relationships indicated, but the slope of the curves, i.e., the increment heat absorbed by the increment heating surface, shows the punishment, if you please, being carried by the heating surfaces comprising different parts of the equipment.

The writer wishes to emphasize by brevity rather than by voluminous discussion, the great value of a uniform distribution of energy absorption over the entire surfaces in so far as it is possible, so that the apparatus may be more reliable and in the long run the operation more successful.

R. S. Julsrud.[10] It would appear from the writer's personal experience, that designers of steam-generating equipment, in their effort to secure a high efficiency in restricted furnace cavities fired by pulverized coal, are apt to lose sight of the physical and chemical limitations of the coal which makes available this heat.

Most coals have certain limitations; these may include low fusion temperature, generally resulting from high sulphur present in the fuel. Again high volatile content may result in coking and possible burning of the tips of pulverized-coal burners and extremely long flame travel with resultant high temperatures entering the convection surfaces. Also, low-volatile coals present the problem of maintaining ignition at low heat-output rates.

Our knowledge of heat-transfer rates to water-cooled surfaces, both bare and refractory-covered when relatively clean, is now fairly well established. However, when such surfaces have become covered with slag, whether fluid, sticky, or spongy, the heat absorption of the surfaces is considerably reduced, and instead of the predicted heat extraction taking place from the gases in the furnace, they enter the convection surface at considerably higher temperatures which may exceed the fusion point of the ash particles present in the gases. The result is a tendency for these sticky ash particles to build up at the heating surfaces. If this occurs on surfaces with restricted gas passages, the result is plugging of the gas passages and eventual outage of the unit. Again the presence of sulphur in coal, resulting in the formation of ferric sulphides when burned, may in furnaces operated at high heat liberation and temperatures result in a reducing atmosphere. It is also possible, but has not as yet been established, that reducing action of these pyrites results in attacking the surfaces exposed to this action. In any event, under high heat liberations these surfaces tend to wash and eventually fail. This action results in boiler outage and increased maintenance costs.

To attempt to reduce these failures, the surfaces are often built up to a greater thickness with a resultant lower heat-transfer rate and the possibility of high gas temperatures entering the generating tubes, as mentioned. Thus in eliminating difficulty in the furnace, it may be transferred to the convection surfaces.

Until such time as we have a better knowledge of the heat absorption of water-cooled surfaces when coated with ash in its various forms, it would appear that a greater degree of conservatism should be exercised in the design of these units.

A reduction of 1 per cent in the CO_2 at the boiler outlet will result in approximately 200 F drop in the furnace-gas temperatures. This may mean the difference between high furnace maintenance and outage as against continuous performance with somewhat lower efficiency. Moreover, high heat releases per cubic foot of furnace volume, which is a measure of furnace temperatures developed, can well be reduced for the same reason.

Such a design would make available to the consumer a wide variety of coals, attractively priced, now handicapped by their physical and chemical limitations.

E. B. Powell.[11] The author has made available an astonishingly simple, logical, and direct method of graphic comparison for evaluating heating-surface effectiveness. The writer would urge that he eliminate at once the only suggestion of complexity, i.e., the factor 500,000,000. Steam boilers of 500,000,000 Btu per hr input rate will soon be outmoded. The method of heating-surface analysis in this paper will serve for years to come. With such repetitive use certainly ahead, why not immediately eliminate all unnecessary effort, even that of dividing by 2, and adopt a more rational, permanently acceptable multiplier, 10^6 or 10^9. Also, to assure instantaneous interpretation of the chart, Fig. 10, the writer suggests that the term "equivalent surface" be defined merely as the square feet of surface per Btu of total heat liberated or made available to the steam-generating unit per hour, with a factor of 10^6 or 10^9 applied to the quotient to secure a number of convenient size. Furthermore, it is suggested that this formula, abbreviated, should be consistently included in the title of the abscissas.

In the closing paragraph, the author, in mentioning a deficiency of data for certain interpretative purposes, would seem to promise a subsequent paper discussing the influence of such factors as draft loss and arrangement of heating surface upon heat-transfer

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

effectiveness. The writer trusts he is correct in this inference. The author's clarifying treatment of all factors in steam-generating-unit heat transfer would be a tremendous asset.

In closing, it would be of interest to know whether corrections were applied to observed temperatures in order to allow for possible radiation error of the high-velocity thermocouple.

R. A. Sherman.[12] A knowledge of the performace of the various sections of a steam generator is essential to intelligent design for proper distribution of the heating surface to insure that the temperatures in various sections are of the proper order. The rate of heat absorption in the furnace governs the temperature of the gases which, for coal-fired furnaces, fixes the degree of fusion of the ash and, therefore, the type of coal which can be burned in a given furnace or the maximum rate of burning, or minimum excess air that can be maintained with a given furnace.

The Society is indebted to the author for the presentation of the data which he has obtained by gas-temperature measurements, as these permit an analysis of the performance of various sections of a boiler which would otherwise be impossible.

The writer wishes to emphasize the fact brought out by the author that the expression, "Btu per cu ft per hr" is of no significance in comparing the performance of various boilers or of furnace temperatures. Earlier in the year, at Battelle Memorial Institute, we conducted an investigation which indicated that the same expression had no significance in relation to the smoke emission. By a factual survey of 102 power and heating boilers burning up to 1200 lb of coal per hr, and in an experimental investigation of 22 boilers of the same type, no relation was found between the smoke emitted and the rate of heat liberation per cubic foot of furnace volume. Briefly, this may be explained by several facts:

1 It is the time of travel of the gases from the fuel bed to the entrance to the heating surface which determines whether the carbon will be burned before its temperature is lowered below its ignition temperature. Furnaces have volume because they have three dimensions but it is the length of travel which is important. Further, a large part of the volume is frequently not utilized and a large volume is not a definite assurance that the carbon will be burned.

2 Combustion of volatile matter without the liberation of carbon is rapid if the required air is mixed with the volatile and, with good mixing, little time or volume will be required for smokeless combustion. Without good mixing an infinite volume or length of flame travel will not avoid smoke.

3 Still other factors, in addition to the degree of mixing, such as the type of coal, the uniformity of distribution of the coal on the stoker, the excess air, the type of load, and the degree of intelligence and the ability of the operator are of greater importance in the elimination of smoke than the rate of heat liberation.

To return to the author's data, a retabulation of a few of the items given for boilers *A*, *B*, and *C* will emphasize the lack of meaning of the rate of heat liberation per cubic foot of furnace volume. The tabulation is as follows:

Boiler	*A*	*B*	*C*
Type	Locomotive	Coal	Oil
Furnace volume, cu ft	33	6030	7494
Furnace heating surface, sq ft	76.8	1695	1633
Ratio, surface/volume	2.33	0.28	0.21
Heat release, Btu per cu ft per hr	354000	22500	27800
Heat release, Btu per sq ft per hr	152200	79900	127200
Heat absorbed, Btu per sq ft per hr	44900	33200	55400

Judged from the relative rates of heat release per cubic foot of furnace volume in the locomotive and in the other two boilers, the temperatures might have been thought excessive in the locomotive boiler but, because of the high ratio of heating surface in the furnace walls to the total volume, the rate of heat absorption was high and the temperatures were probably no higher than in the other boilers. On the basis of the rates of heat liberation per square foot of furnace wall, the differences among the three boilers were much less and the rates of heat absorption per square foot of furnace wall were also of much the same order.

Basing the comparisons of boilers on the ratio of the heating surface to the heat available will bring various boilers more into line but variations will be due, as the author points out, to cleanliness of the surface and to the luminosity and emissivity of the flame. The emissivity of a flame increases as its depth increases owing to greater furnace dimensions, but the emission per square foot of flame surface cannot increase in the same order as the area decreases with respect to the volume.

A spherical furnace would be the least desirable shape of furnace because this gives the lowest ratio of surface to volume. The cubical box-like furnaces which we have in many present-day furnaces are also unfavorable for obtaining a large area of heat-absorbing surface. High, narrow furnaces are more favorable in presentation of surface; this principle can be extended by introduction of division walls of water tubes to obtain greater surface and lower the furnace temperature.

The adoption of the author's method of obtaining data on temperatures and gas weights by many other workers would be desirable. Further studies on the relation of the radiant emission from the flame to the dimensions of the furnace would also be valuable.

The writer agrees with a previous suggestion that the term "equivalent surface area" be replaced by the expression, "ratio of surface area to heat available," multiplied by a simple factor such as 10^6 on the ground that this is a more accurate expression and less likely to introduce confusion. Let us hope that we can mark this day as that on which the interment of the term, "Btu per cu ft of furnace volume," was completed, never to rise again.

John Van Brunt.[13] The method suggested by the author for comparing performance of furnaces, boilers, superheaters, economizers, and air heaters, comprising a complete steam-generating unit, is interesting and the selection of a unit of 500,000,000 Btu input for such comparisons serves the purpose perhaps as well as any other arbitrary value.

In the design of large steam-generating units it is necessary to determine the amount of heat absorbed by each portion of the entire unit by using actual fluid quantities and heat-transfer rates, etc. After such a unit is in operation, it is customary to check the calculated performance, and the basic design rates used for its determination.

The amount of heat to be absorbed by each division of the total surface is determined by the performance requirements and the conditions imposed by the purchaser and by the economic factors of each case, plus limitations dictated by the fuel characteristics and factors governing availability of the unit for continuous operation.

Given two units, for example, of 100,000 and 300,000 lb capacity, the smaller for 300 lb per sq in. pressure and 750 F steam temperature with 250 F feedwater, and the larger for 1200 lb pressure, 925 F steam temperature with 380 F feedwater temperature; there can be no purpose in comparing such units on the basis of a 500,000,000-Btu unit, as suggested by the author.

The per cent of heat absorbed in the furnaces in the various tests given in the paper are low, in fact, considerably lower than would be expected in a modern furnace. In tests A-1, A-2, and A-3, the low rate of absorption is probably due to poor combustion conditions and can be compared only with locomotive boilers.

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[13] Vice-President Engineering, Combustion Engineering Company, Inc., New York, N. Y. Mem. A.S.M.E.

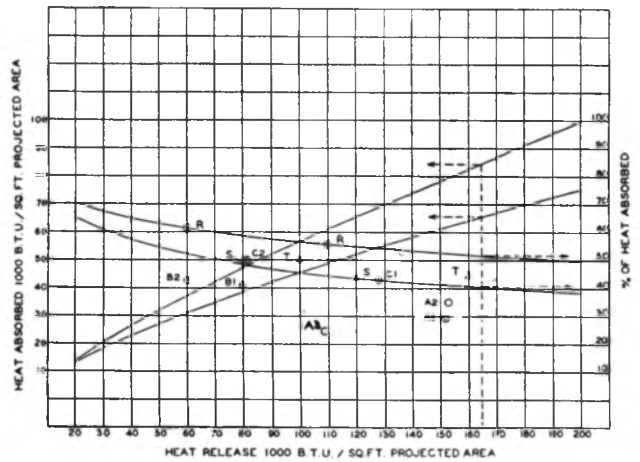

FIG. 11 APPROXIMATE RELATION OF HEAT RELEASED TO HEAT ABSORBED IN A PULVERIZED-COAL WATER-COOLED FURNACE

Fig. 11 of this discussion reproduced from Fig. 9 of the writer's paper,[14] shows the ratio of heat absorbed to heat released, both in Btu and in per cent. The curves sloping upward from the lower left-hand corner are read in Btu per lb per sq ft, to the left, and the curves sloping downward read percentage of heat absorbed or efficiency of furnace absorption, to the right.

The points from the tests in the author's paper are plotted against the percentage of heat absorbed. With the exception of C-1 and C-2, the points are well below the lower of the two efficiency curves. In a modern oil-burning furnace, the points C-1 and C-2 would be expected to fall near the upper efficiency curve. For comparison, the two points *R* are plotted from test results of a tangentially fired pulverized-coal slag-bottom furnace on a unit of 300,000 lb per hr capacity. The points *T* are from a 450,000 lb per hr tangentially fired pulverized-coal slag-bottom furnace and points *S* are from a vertically fired slagging-bottom furnace, pulverized-coal-fired. In all of these units, the furnace walls are composed of bare fin tubes.

The influence of both the amount and character of the ash and of the method of firing on the amount of heat absorbed is marked. It may be seen from the curves and from the points plotted from the tests that the per cent of heat absorbed in any furnace decreases as the rate of release per square foot increases.

Absorption below the lower curves would indicate inefficient use of the furnace wall surface, poor combustion, or walls covered with an excessive amount of ash. Waterwall tubes covered with refractory or refractory coatings will also fall below the lower curves.

AUTHOR'S CLOSURE

The comments of Messrs. Hobbs, Powell, and Sherman are very much appreciated as they fully grasped the importance and significance of this method of comparing the performance of furnaces and heat-absorbing surfaces in steam-boiler units. Mr. Powell has raised a question, in which Mr. Sherman concurs, i.e., whether the equivalent-surface area might be simplified to a factor of square feet of surface per Btu available. Mathematically this would seem to be simpler, but psychologically the author fears it would be more confusing because the logic of the factor is not quite so evident. For instance, assume a unit having 500,000,000 Btu available and, at the point to be plotted 10,000 sq ft of actual surface have been passed over; this factor would be 0.00002. The suggestion of multiplying this by 10^6 would leave it 20. This seems like a very inadequate figure to plot when it actually represents 10,000 sq ft of surface. Even multiplying this by 10^9 or 10^{10} seems in the author's opinion only to add to confusion rather than to simplify the method of comparison used in the paper.

There may well be some question concerning the choice of 500,000,000 Btu when some boilers may go as high as 1,000,000,000 and others may be only $^1/_5$ or $^1/_{10}$ this amount. The value of this entire method comes through standardization and while, as Mr. Powell says, the 500,000,000 may be outmoded, it is greatly to be doubted if the average boiler unit to be considered will vary so much from 500,000,000 Btu available that it cannot still be used as a perfectly logical standard. Certainly 100,000,000 Btu is not a satisfactory standard and 1,000,000,000 is applicable in only a few cases. As a matter of fact, it is felt that the odd number of 500,000,000 has some real value in that it makes one stop to think what it really is, while otherwise it might be an abstract factor and not carry the significance to the mind of the user, which condition now seems to prevail. Another point of significance is that most coordinate papers bring out the values on this basis very nicely, while figures $^1/_5$ or 2 times those used would not be as convenient to plot.

Mr. Julsrud mentions several matters pertaining to furnace design and the characteristics of ash which are really outside the scope of the paper as presented, since the data given were limited purposely to a range of boiler operation permitting a comparison with the sizes prevailing in locomotive practice, and also to bring out the method of comparison rather than to go into the many phases of slag, heat absorption, etc. As indicated by Mr. Powell, there is great opportunity in the future for all these matters to be studied through this method of comparison.

As a matter of fact, Mr. Van Brunt in Fig. 11 of his discussion has already compared some of the data on dry-ash coal-fired and oil-fired furnaces with data on slag-tap furnaces for which he has information.

Before replying to Mr. Van Brunt's discussion, the author checked with him and was assured that his "heat release 1000 Btu per sq ft projected area" is identical with the author's "heat available." It is significant to note that the lower curve of Mr. Van Brunt's Fig. 11 on the percentage basis is substantially identical with Mr. Orrok's curve, which the author reproduced on a similar basis and presented as Fig. 20 in his discussion[15] of Mr. Brandt's paper. We have found that not only do the points C-1 and C-2, which Mr. Van Brunt has plotted in his Fig. 11, fall on this curve of Mr. Orrok's, but they also are in close agreement with many other data obtained on oil-fired boilers having normally clean heat-absorbing surfaces and where the adiabatic temperature is close to 3600 F. Points B-1 and B-2 for the pulverized-coal-fired furnace with dry-ash removal are approximately 20 per cent lower in rate of heat absorption than are those of the Orrok curve on which fall the oil-fired tests. This indicates a certain amount of ash accumulation on the surface of such furnace, which also has been substantiated with many other data. As noted in the author's paper, all of these data, except those concerning the French locomotive, are based upon tests with the high-velocity thermocouple of the single-shield variety, and to answer Mr. Powell's question no correction has been made. It is now known, however, that many of these temperatures are likely to be from 50 to 75 deg low. This would indicate that the actual rates of heat absorption are still lower than those indicated by the *B* and *C* points and, therefore, closer to the French data which Mr. Van Brunt believes to be low. They are in the same region as Goss's locomotive tests of 1912, where heat absorption

[14] "Design of High-Capacity Boiler," by John Van Brunt, Trans. A.S.M.E., vol. 60, 1938, paper FSP-60-17, p. 487.

[15] "The Locomotive Boiler," by C. A. W. Brandt, discussion by E. G. Bailey, published on page 399 of this issue of the Transactions.

in the firebox alone was determined by direct evaporation without depending upon any gas temperatures. These data from Goss's tests are also given in Fig. 20 of the author's discussion[15] of Mr. Brandt's paper.

Mr. Van Brunt has questioned why all of these rates of heat absorption are so low in comparison with his points *R*, *S*, and *T*, taken in furnaces of the slag-tap type where the walls are apparently coated to some extent with slag. However, since Mr. Van Brunt does not give the fluid temperature of the ash or other data, it is beside the point to question them further than to state that any inaccuracies in the measurement of gas temperatures in these portions of boiler units may lead to very erroneous conclusions.

Dr. Mullikin has presented papers[16] before the American Institute of Physics showing comparative results from bare thermocouples of different sizes, single-shield high-velocity thermocouples, and multiple-shield high-velocity thermocouples. A great deal of field data were included. These data emphasize the importance of further studies in gas-temperature measurement.

Further evidence has been obtained showing that the multiple-shielded high-velocity thermocouple is more nearly correct than anything else that has yet been devised. In boiler installations where the heat absorbed in the air heater, economizer, and superheater can be determined separately, the gas temperatures entering the superheater may be determined by calculating back from an accurate measurement of the gas temperature leaving the air heater. Such data check very closely with the multiple-shielded high-velocity thermocouple.

Other confirmation on rates of heat absorption in furnaces has been obtained by a thermal probe wherein the rate of heat absorption has been determined by the rate of flow of a fluid and its rise in temperature through the element when inserted into different parts of furnaces.

Mr. Van Brunt states that there is no advantage gained in comparing a unit of 100,000 lb per hr output at 300 lb per sq in. pressure, 750 F steam temperature, and 250 F feedwater temperature, with a larger unit of 300,000 lb per hr output, 1200 lb per sq in. pressure, 925 F steam temperature, and 380 F feedwater temperature. If the fuel burned and furnace construction are the same, a comparison of heat absorption up to the entrance of the superheater will be well worth-while, for the temperature gradients are not sufficiently different to have any measurable effect upon the rate of heat absorbed up to that point. There will, of course, be a justifiable difference in the shapes of the curves through the superheater, later boiler bank or economizer, and the air heater. However, it is not beyond our experience to learn a great deal from even these comparisons. Of course, the real advantage comes in comparing units in which similar temperature gradients exist, in order to detect the effect of excess air, different kinds of fuel and ash, arrangement of surface, etc. The main purpose of the present paper was to show a simple method of making these comparisons, leaving it to the judgment of those using this method as to the value obtained by the variety of comparisons they might choose to make.

[16] "Gas-Temperature Measurement and the High-Velocity Thermocouple," by H. F. Mullikin, Symposium on Temperature Measurement and Control, American Institute of Physics, New York, N. Y., Nov. 2, 1939.

"Accuracy Tests of the High-Velocity Thermocouple," by H. F. Mullikin and W. J. Osborn, loc. cit.

These papers were published in abstract form in *Glass Industry*, vol. 20, no. 12, December, 1939, pp. 441–442.

The Locomotive Boiler

By C. A. Brandt,[1] New York, N. Y.

For many years increased traffic and the demand for high operating economy of the railroads resulted in the development of locomotives of high tractive effort capable of starting trains close to the limit of length for practical handling through yards and terminals. The ever-present demand for still higher transportation capacity of the railroads and competition have compelled the further development of locomotives to handle these long and heavy trains at high speeds. This has meant proportionately larger and more efficient boilers. The author discusses in this paper some of the problems of boiler designs and proportions that affect the efficiency and capacity of the conventional locomotive.

The public, which is served by the railroads, is continuously demanding higher speeds for both freight and passenger trains and this, in conjunction with a more intensive utilization of their motive power, is one of the many important problems confronting the managements of the American railroads today.

Until recent years the steam locomotive had been the principal power unit, but its supremacy is now being challenged by other forms of motive power, particularly the electric- and Diesel-driven locomotives. To meet this challenge the designers of steam locomotives are constantly studying the problem of building boilers of greater steam-generating capacity within the permissible limits of size and providing engines of lowest possible steam consumption per unit of power.

The fact that higher boiler capacity for minimum weight has been an ever-present problem throughout the years of locomotive development in America can best be illustrated by citing the evolution of the eight-coupled locomotive. The first of this design, or the 2-8-0, utilized 90 per cent of the total locomotive weight for adhesion. The demands for higher speeds, requiring greater steam-making capacity, led to the addition of truck axles to carry the heavier boilers, which resulted in the successive development of the 2-8-2, the 4-8-4, and last the 6-4-4-6 type high-speed locomotive, exhibited at the New York World's Fair this year.

The total weight of an early-design 2-8-0 type locomotive with 270,000 lb on the drivers and a tractive power of 67,500 lb was only 300,000 lb. To supply the steam for the high-speed 6-8-6 type locomotive with the same weight on drivers and the same tractive power requires a boiler of such size as to double the total weight of the locomotive to 600,000 lb, as is illustrated in Fig. 1.

As this and other modern locomotives are examples close to the maximum practical size that can be built, the important question is whether greater boiler efficiency is attainable, particularly at high-capacity operation, and what can be done to accomplish this.

The question of reducing the steam consumption of the engines is outside the scope of this paper, but the possibility of reducing the average steam consumption per ihp from an average of 18 lb to 13 lb is possible by the adoption of compound cylinders as reported from tests on French locomotives, as well as by the use of still higher superheat and valve gear adaptable for its utilization.

The paper will be confined to a discussion of those problems of design believed to be most essential in the advancement of the art. To enhance the value of this contribution, Tables 1 to 6 are included which give the principal dimensions and boiler ratios of representative locomotives of various types built in America in recent years. Locomotive test curves are also presented.

The subject matter will be centered on problems affecting the design of the conventional locomotive fire-tube boiler only, not because of any belief that this type of boiler is the final answer to the locomotive steam generator, but because at present most of the locomotives in the world are equipped with this type of boiler. The best solutions to some of the problems encountered in the design of large boilers have not yet been agreed upon principally because of the lack of reliable test data.

A committee of the Association of American Railroads (A.A.R.) submitted a report this year recommending a conventional type of boiler for a proposed 6400-hp high-speed locomotive, which is an acknowledgment that the fundamental principles of this boiler design have proved practical and are the best available at the present time.

Type	Wheel arrangement	Tractive power, lbs	Weight on drivers, lbs.	Total weight of loco, lbs.	Weight on drivers percent of total	Increase in weight of loco. to gain additional boiler power: Actual	Percent
2-8-0		67500	270000	300000	90		
2-8-2		67500	270000	360000	75	60000	20
4-8-2		67500	270000	400000	67	100000	33
4-8-4		67500	270000	450000	60	150000	50
6-8-6		67500	270000	600000	45	300000	100

TABLE SHOWING INCREASE IN THE TOTAL WEIGHT OF AN AVERAGE 8 COUPLED LOCOMOTIVE OF THE SAME WEIGHT ON DRIVERS AND SAME TRACTIVE POWER

Fig. 1 Increase in Total Weight of Eight-Coupled Locomotives Due to Larger Boiler Capacity

It is unnecessary to recite in detail the advantages of the conventional boiler; this has been done many times before. The virtues of the completely water-cooled radiant-heat-absorbing furnace, high gas velocities over the convection heating surfaces with forced draft, and high superheat, all originally inherent in the locomotive boiler, are now being recognized as essentials to efficient steam generation in other fields and are being rapidly adopted in stationary-power-plant boilers.

Locomotive boilers of designs radically different from conventional construction have been built in the past, but so far none has proved sufficiently practical for general railroad service. There will soon be placed in service in America a different type of steam generator for operation in conjunction with condensing steam turbines, and still other types are under consideration.

There has been a great increase in the size of locomotive boilers built in recent years in America; many of these have proved very efficient. However, much remains to be done to make possible the burning of more fuel per hour at higher combustion efficiency

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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.

with lower draft loss, greater heat-absorption efficiency, and higher superheat with less boiler weight per pound of steam produced.

Definition of Boiler Capacity and Existing Formulas

In designing a locomotive, the steam required for a given maximum horsepower capacity can be determined quite accurately from existing knowledge of steam consumption per ihp-hr, when initial and final steam conditions are specified. To this must be added the steam required for auxiliaries, train heating, air conditioning, and other uses. The boiler design must be such as to deliver this maximum steam output at a superheat as high and a back pressure as low as that figured on, or else the steam consumption per ihp will increase and the cylinder power fall below that required. To determine the excellence of a particular boiler design and compare it with others, it is necessary to have a standard measure of comparison.

It would appear logical that the weight of steam produced per pound of total weight of the boiler in service should provide a satisfactory measure when a definite steam pressure, superheat, and draft loss are specified. It is obvious that such a yardstick is of no value, unless a standard method of predetermining the maximum evaporative capacity is found which will satisfy all conditions in a reasonable way.

In reviewing the existing methods of calculating the maximum evaporating capacity of a locomotive boiler and the over-all boiler efficiency, the designer is confronted with the fact that formulas now generally used are inadequate for predetermining the result with accuracy, particularly when large boilers are involved.

In America the theory generally followed is that the steam-generating capacity[2] of a boiler is directly proportional to the amount of evaporating heating surface in square feet.

A set of evaporative values, giving the maximum quantity of steam in pounds per square foot of heating surface per hour, generated by the firebox and flues, was prepared some years ago by F. J. Cole.[3]

These values are generally used by the locomotive builders and railroads today.[2,3,4,5]

Any method of calculation that takes square feet of heating surface only into consideration must be inadequate. It is evident that the arrangement of the heating surfaces, their relationship to the grate area, furnace volume, gas area, firebox heating surface, and hydraulic depth and length of the flues must be considered to give approximately correct results. The reason is that it is these relationships which determine the boiler efficiency, the back pressure required to produce the steam, and the superheat, all of which have a great influence on the efficiency and power output of the cylinders.

Strahl proposed a formula in 1913[6] which takes into consideration the size of the grate area and its ratio to the evaporating heating surface

$$WS = \frac{a}{\frac{S}{R} + 7} \times S$$

where

WS = total steam produced, lb per hr
R = grate area, sq ft
S = evaporating heating surface, sq ft
a = coefficient for superheated locomotive, 778 lb per sq ft

This formula, with the coefficients established by Strahl, comes close to test results on boilers with proportions similar to those which Strahl used in his analysis, but leads to unsatisfactory results on boilers with different ratios. The Strahl formula may be modified to suit different arrangements by changing the coefficients to suit, but this is not an entirely sound procedure.

A method originated by Lawford H. Fry[7] for determining the maximum evaporating capacity of a boiler from the over-all boiler-efficiency curve appears to be a better approach. Fry suggested that, if the over-all boiler efficiency is plotted against pounds of coal or total heat in Btu fired per square foot of grate per hour, the relationship between the efficiency and rate of firing becomes a straight line. The correctness of this theory has been proved on tests examined. The fundamental equation established by Fry is

$$F = m - nG$$

where

F = boiler efficiency in per cent
G = dry coal fired, lb per sq ft of grate per hr
m = coefficient denoting theoretical efficiency at zero firing rate
n = coefficient determining slope of curve

If sufficient test data were available for all kinds of fuel and types of boilers so that the coefficients m and n or the origin and slope of the over-all boiler-efficiency curve could be predetermined with accuracy, together with the superheat and pressure at the outlet of the superheater, then the evaporation for any quantity of coal fired could be determined, as well as the maximum capacity.

It appears to the author that the Fry or other methods proposed do not satisfy the requirement as they stand, because draft loss and back pressure are not part of the picture.

The inadequacies of both the Cole and Strahl methods are apparent when a comparison is made between the data calculated by the different methods, and the actual test results as shown in the following table:

	Evaporation, lb per hr				
Locomotive	Strahl	Cole	Difference, per cent	Actual test	Difference, per cent
P.R.R. K4S No. 5341	48000	52150	+ 8.6	72000	+50
P.R.R. M1A No. 6706	53350[a]	67850[a]	+27.2	99095[a]	+85.7
N.Y.C. J3A	60700[a]	60500[a]	− 0.3	85000[a]	+40

[a] With feedwater heater.

The author called attention to this some twelve years ago.[8] As far as is known, however, no tests have been made of locomotives of greatly varying boiler ratios and types of fuels to establish a satisfactory formula which may be used universally and permit evaluation of the effect that varying proportions have upon the economics of boiler performance and costs.

Boiler Tests

In its 1936 report, the A.A.R. Committee on Locomotive Construction expressed the opinion that the Cole ratios were inadequate and recommended approval of plant tests to obtain neces-

[2] "Locomotive Data," The Baldwin Locomotive Works, eleventh edition, Philadelphia, Pa., 1939, pp. 21–23.

[3] "Locomotive Handbook," American Locomotive Company, Schenectady, N. Y., 1917, p. 58.

[4] "Potential Horsepower Formula Agreed to by American Locomotive Builders," Report of Federal Co-Ordinator of Transportation, Washington, D. C., Nov. 27, 1935, p. 48.

[5] "Horsepower and Tractive Effort of the Steam Locomotive," by A. I. Lipetz, Trans. A.S.M.E., vol. 55, paper RR-55-2, 1933, pp. 5–42.

[6] "Method of Determining the Capacity of Steam Locomotives," by Strahl, *Zeit. V.D.I.*, vol. 57, 1913, pp. 251–257, 326–332, 379–386, and 421–424.

[7] "A Study of the Locomotive Boiler," by Lawford H. Fry, Simmons-Boardman Publishing Corporation, New York, N. Y., 1924.

[8] "The Design and Proportion of Locomotive Boilers and Superheaters," by C. A. Brandt, Proceedings of the Canadian Railway Club, vol. 27, Feb., 1928, pp. 20–64.

sary data. The director of equipment research at that time, L. W. Wallace, made a report and recommended complete locomotive tests, but this program has never been carried out.

Many tests have been made by the Pennsylvania and New York Central railroads and the respective managements deserve the greatest praise for their valuable contributions to the art. Most of these tests, however, have been made on locomotives with relatively small grate areas and with approximately the same ratios of grate areas, gas areas, firebox volumes, etc. There are few data available on the effects of very large grates and large fireboxes.

The tests recommended by the Research Division of the A.A.R. were complete and necessarily expensive. It is believed by the author that adequate information as to the efficiency and maximum capacity of boilers with large grates and furnaces can be obtained with stationary blowdown tests of some three or four boilers with widely different boiler ratios. Such tests should be conducted with oil fuel and also with several different grades of coal, sufficient to establish fundamental data now lacking. With complete data on the quantities, qualities, etc., of the fuel, water, and air used, and the gases and steam produced, all losses could be segregated and closely determined.

This would permit the determination of coefficients m and n in Fry's formula. It is hoped that such a test program will be made possible as this matter is not one merely of academic importance, but is a vital item of railroad economics. The method of standing blowdown tests developed and used by the New York Central Railroad in recent years has proved very effective. Results of such tests are dependable since the boiler testing is separated from and is independent of the locomotive-engine performance. This facilitates the adjustments for setting the required operating conditions for each capacity test and the continuance of each test run for a sufficiently long period of time at constant rate, without interruption by locomotive- and test-plant running conditions.

The New York Central Railroad's method of blowdown test is conducted so that the steam exhausted from the cylinders through the nozzle is desuperheated to a temperature closely agreeing with that actually observed on road tests for equal capacities. The correctness of this test procedure has been proved by the fact that the front-end design and nozzle size established by such test have proved correct for best maximum performance in road service.

Gas Area in Relation to Boiler Efficiency and Capacity

It is well known that the efficiency of the boiler decreases with an increased firing rate. The rapid drop is mainly due to the high losses occurring in the form of unburned fuel escaping with the flue gases. The problem of greater fuel-burning capacity at higher efficiency of combustion is, therefore, the first item which should be considered and involves the arrangement and relative size of the gas area through the boiler, the grate area, combustion volume of the furnace, and the firebox heating surface, also, stoker construction, arrangement of firebrick arches, amount of air opening through the grates, and the introduction of secondary air above the fire bed.

In considering the matter of locomotive-boiler design, as in the case of any other structure, it is well to establish a base from which to start. This is difficult with a locomotive because the problem of its design involves a cycle of successive approximations to obtain maximum power and efficiency within the weight limitations.

In the past either the heating surface or the grate area has been the basis on which the size of the other component parts of the boiler has been determined. This is not the most logical procedure as the dimensions of these parts do not control the limit of size to which the boiler can be built, since they may be increased with the length of the boiler within considerable limits. The diameter and gas area through the boiler constitute the limiting factors because the height, width, and weight are fixed and cannot be exceeded. It is, therefore, more logical that this factor be made the basis on which the other parts are proportioned.

The capacity of the locomotive boiler is limited by its diameter because this determines the gas area of the flues through which all the gases of combustion must pass; in addition it must provide space for the superheater through which all the steam generated must flow. It determines the flue heating surface that can be installed per unit length of flue, the area of the steam-disengaging

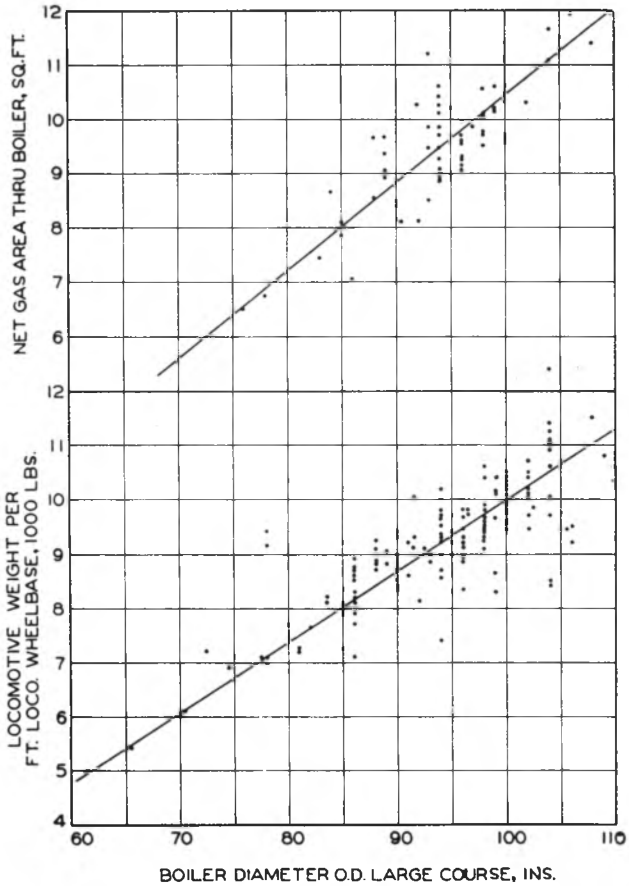

Fig. 2 Relation of Boiler Diameter to Total Weight of Locomotive per Foot of Wheel Base and Also Gas Area

surface, and the steam volume above the water level in the boiler. This last item is of the greatest importance and is, in reality, the limiting factor in high-output operation, as evidenced by the serious difficulties experienced with water carry-over into the superheater and cylinders in many boilers.

Within the clearance, the maximum diameter of the boiler is determined by the weight limits. In order to analyze this problem, the proportions of 165 different designs of modern locomotives have been studied to ascertain the relationship between the diameter of the boiler and the total weight of the locomotive. The curve, Fig. 2, shows this relationship. It is apparent that, while there are many factors which influence the weight, the boiler diameter has been sacrificed for other features of design in many cases. If, as an example, the 104-in. boiler is noted, the total locomotive weight per foot will vary from a minimum of 8400 lb to a maximum of 12,400 lb, or an increase in total locomotive weight of nearly 50 per cent for the same boiler diameter. The chart indicates that there is a uniform increase of 133 lb per ft of total locomotive weight for every 1-in. increase in the boiler diameter. This curve may facilitate studies in efforts to obtain the greatest possible diameter and gas area for allowable weights.

A relatively large gas area through the boiler in relation to the grate area and heating surface is not generally followed, as noted from a study of the ratios Nos. 30, 31, and 32, Tables 1 to 6.

That the gas area through the boiler is one of the most important items affecting the boiler capacity is not generally appreciated, but may be better understood if actual test figures are analyzed. From the test of the Pennsylvania Railroad, class M1A, locomotive No. 6706, column 10, Table 4, on which sufficient data are available to permit determination of the weight of the gases passing through the boiler, the data shown in Fig. 3 have been compiled. It is interesting to note that, at high firing rates, the gas velocity through the tubes near the back tube sheet reaches the high figure of 204 miles per hour. The great importance of eliminating unnecessary restrictions in the gas passages and the superheater units is apparent.

As the gas area is increased, the velocity and frictional resistance of the gases passing into and through the boiler flues are lowered. This will reduce the draft requirement and in turn the cylinder back pressure, enlarging the power output of the locomotive in direct proportion to the resultant increase in the mep. This is important when high power output is desired at high speed as the reduction of 1 lb of cylinder back pressure gives as much increase in power as a 4-lb gain in pressure on the steam inlet or admission side.

The importance of a large gas area through the boiler was recognized years ago. C. D. Young[9] pointed out in 1915 that from his experience the maximum evaporative capacity of a boiler was in direct ratio to the total gas area through the flues and was apparently limited to an amount of 7000 lb actual or 9100 lb equivalent evaporation per hr per sq ft of gas area for the engines tested up to that time. This has been bettered considerably on modern locomotives with type-E superheaters and more powerful draft arrangements to about 11,000 lb equivalent evaporation per sq ft of gas area. The test results given in the following table confirm this point:

Railroad	Class engine	Engine no.	Max equiv evap per hr, lb	Total net gas area through boiler, sq ft	Max lb of steam per hr equivalent evap per sq ft gas area
P.R.R.	K4S	1737	87,414	9.10	9605
P.R.R.	M1A	6706	123,870	9.71	12755
B.L.W.	3 cyl	60000	84,186	9.37	8985
P.R.R.	I1S	790	89,235	9.9	9015
N.Y.C.	J3A	5408	89,500	8.9	10055

To illustrate the limiting effect that the gas area has upon the evaporative capacity of a locomotive boiler, the test results from several locomotives have been plotted in Fig. 4. The relation between equivalent evaporation and back pressure is illustrated in Fig. 5.

A study of the relationship between the gas area and the boiler diameter of a number of modern boilers has been plotted in Fig. 2; it will be noted that full advantage has not been taken in many cases in providing the largest possible gas area. For some boiler diameters the gas area is as much as $31^1/_2$ per cent smaller than the maximum possible. This is perhaps due to a difference of opinion as to the most suitable distance between the top of the crown sheet and the inside of the boiler shell at the top; as to the water space between the combustion chamber and boiler shell, or the clearance between tube holes and tube-sheet flanges; and as to tube spacing. The type of combustion chamber, the thickness of the flues, and the superheater design also influence the gas area. Offhand, it may seem that these details are not of great consequence, but really they are very important.

The distance between the top of the crown sheet and the boiler shell determines the steam space and steam disengaging surface. It is obvious that the smaller this distance is made the greater will be the tube-sheet area. Information as to the most satisfactory relationship between the height of the crown sheet and other dimensions of the boiler is lacking, but this question deserves most careful consideration.

[9] Test Bulletin, No. 28, Pennsylvania Railroad, 1915, p. 35.

TOTAL COAL FIRED LBS. PER HOUR	TOTAL COMBUSTION GASES LBS. PER HOUR	VELOCITY OF GASES MILES PER HOUR SECTION A	B	C	D
8975	99000	135	46	35	107
23636	158000	263	92	70	204

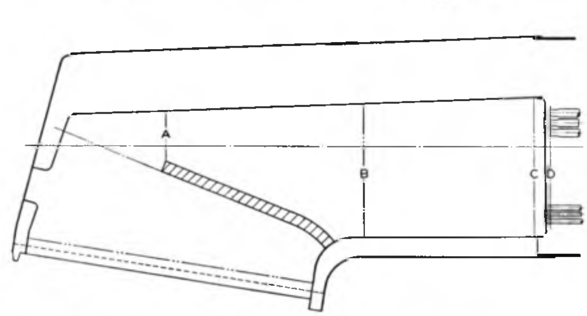

FIG. 3 GAS VELOCITIES THROUGH LOCOMOTIVE FIREBOX AND FLUES

A slight increase in the water space between the combustion chamber and boiler shell decreases the available gas area considerably. On some boilers, this water space has been made as large as 8 in., while very many large boilers are operating successfully with a water space of only 5 in., giving a large percentage increase in gas area. It appears that the smaller water space imparts a higher velocity to the water circulation at this point, minimizing mud collection which the large water space was supposed to eliminate.

The important influence that the superheater construction has upon the gas area is apparent and will be discussed in more detail under the subject of superheater design.

FIREBOX VOLUME IN RELATION TO COMBUSTION EFFICIENCY

Earlier in this paper, it was pointed out that the firebox volume is not subject to such limitations as the gas area. There are limitations to the width and height of the firebox which are important, but the length of the grate and firebox may be extended considerably beyond present general practice. This is fortunate as the most serious matter confronting locomotive designers today is the problem of improving the efficiency of combustion in the firebox, particularly at high-capacity operation. Incomplete combustion of fuel causes heavy losses in unburned fuel, high maintenance cost due to cinder cutting of firebox sheets, stay-bolt heads, tubes, and superheater. Damage due to fires along the right of way set by sparks, loss in good will to the general public due to smoke near railroad terminals, and loss of passenger traffic for the same cause constitute a serious challenge to designers and operators of steam locomotives.

Improvement in the combustion process of the locomotive furnace is a problem of primary importance and the proportions of the furnace should be such as not only to make possible complete combustion of the fuel with the elimination of smoke and cinders, but also to absorb sufficient radiant heat to cool the furnace gases to a temperature low enough to prevent slagging at the flue sheet and on the superheater units.

One would assume that sufficient data were available today to determine the correct volume of the firebox in relation to the grate area and heating surface. A study of Tables 1 to 6 indicates, however, that there is little agreement on this subject; great variations exist in these basic ratios. To cite an example: The important ratio of firebox volume to grate area ranges from a minimum of 3.70 to a maximum of 8.28 cu ft per sq ft of grate

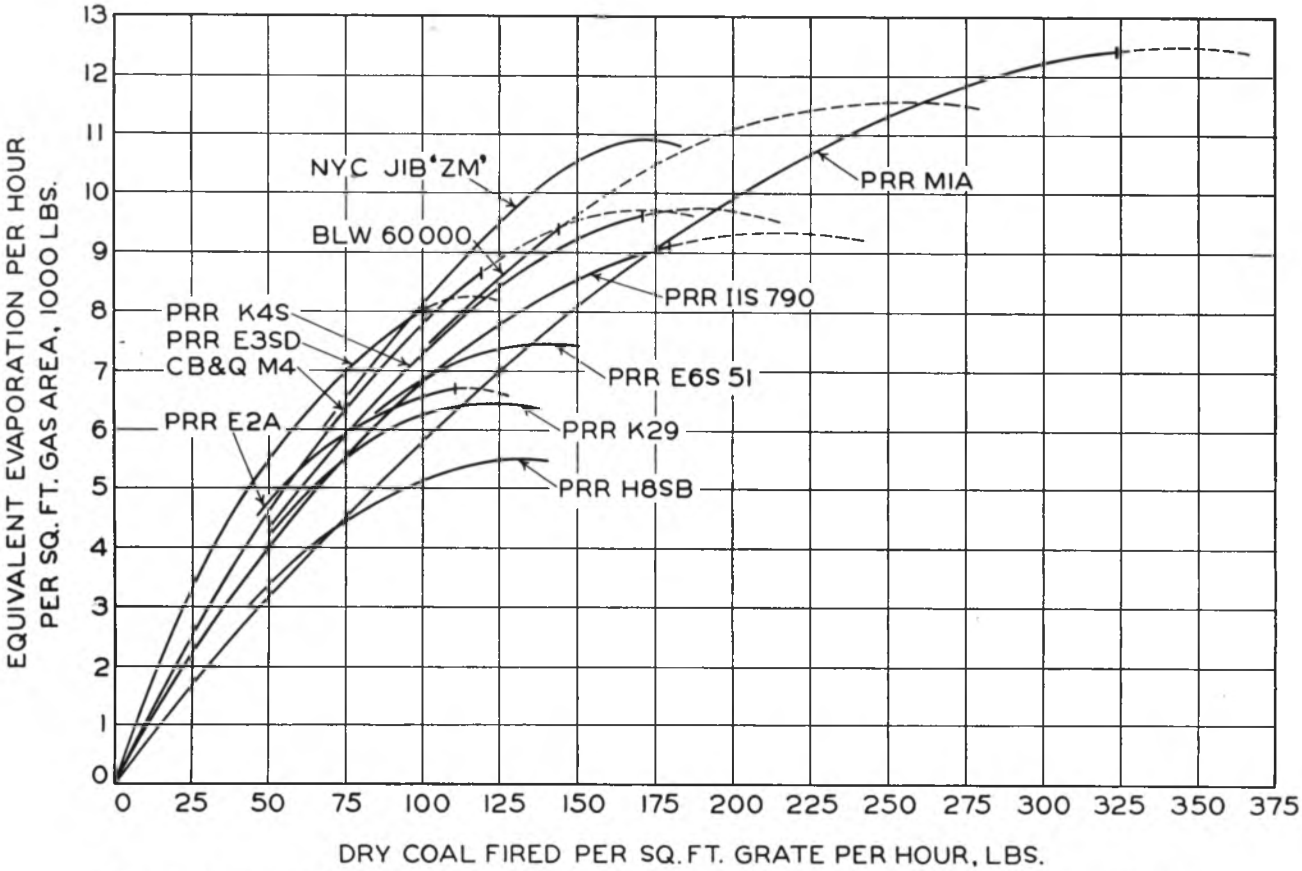

Fig. 4 Equivalent Evaporation per Square Foot of Gas Area Versus Coal per Square Foot of Grate

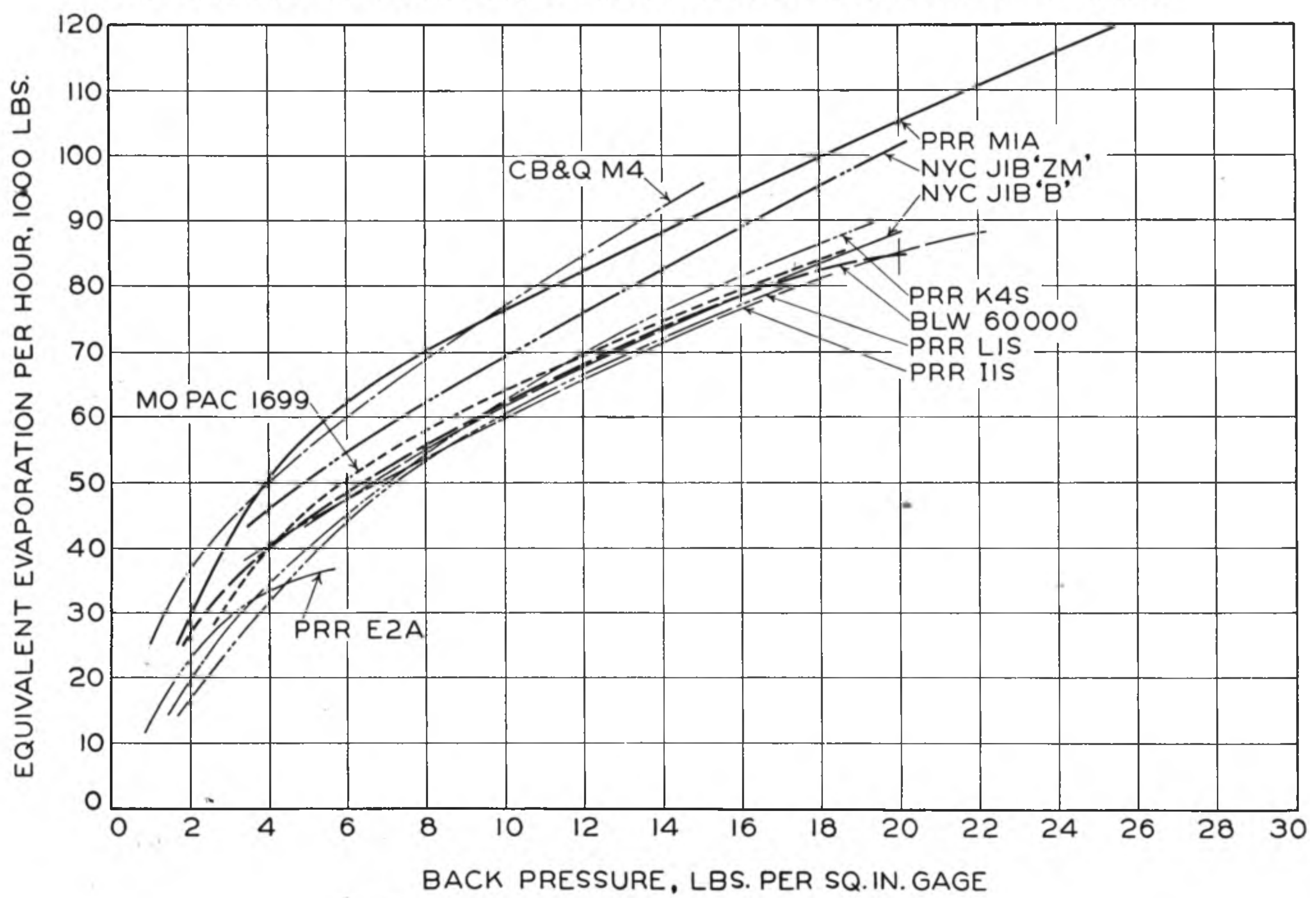

Fig. 5 Equivalent Evaporation Versus Back Pressure

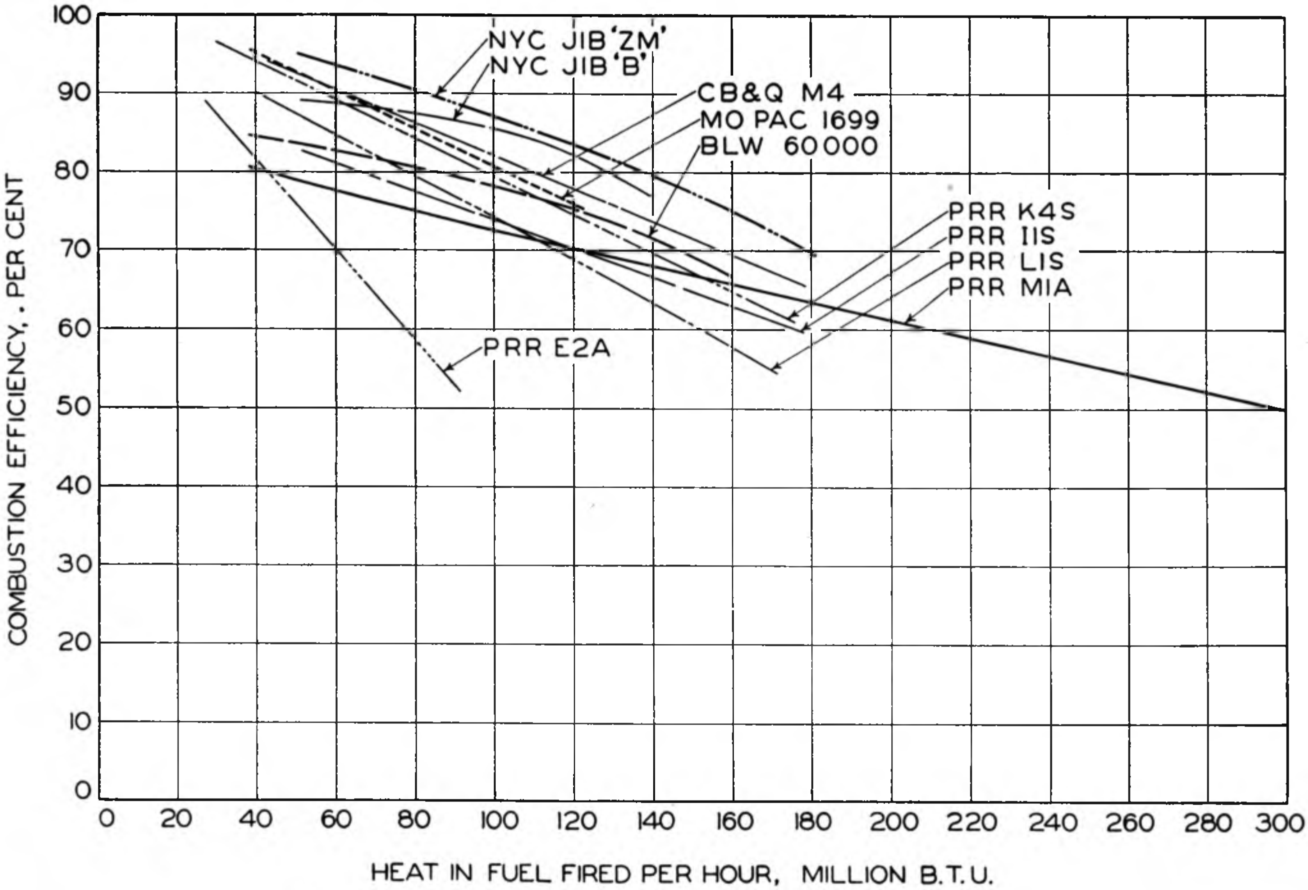

Fig. 6 Combustion Efficiency Versus Heat in Fuel Fired per Hour

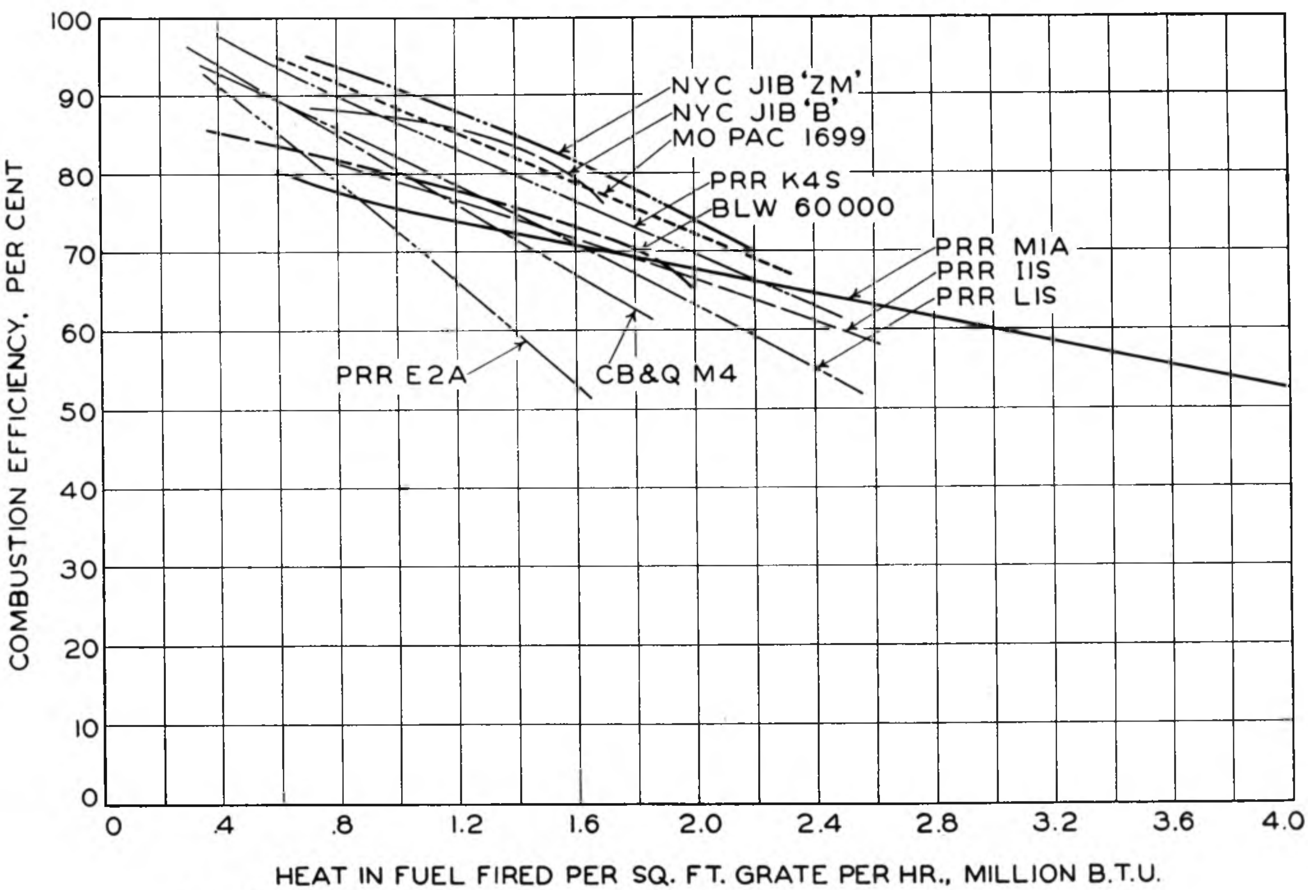

Fig. 7 Combustion Efficiency Versus Heat in Fuel Fired per Square Foot of Grate per Hour

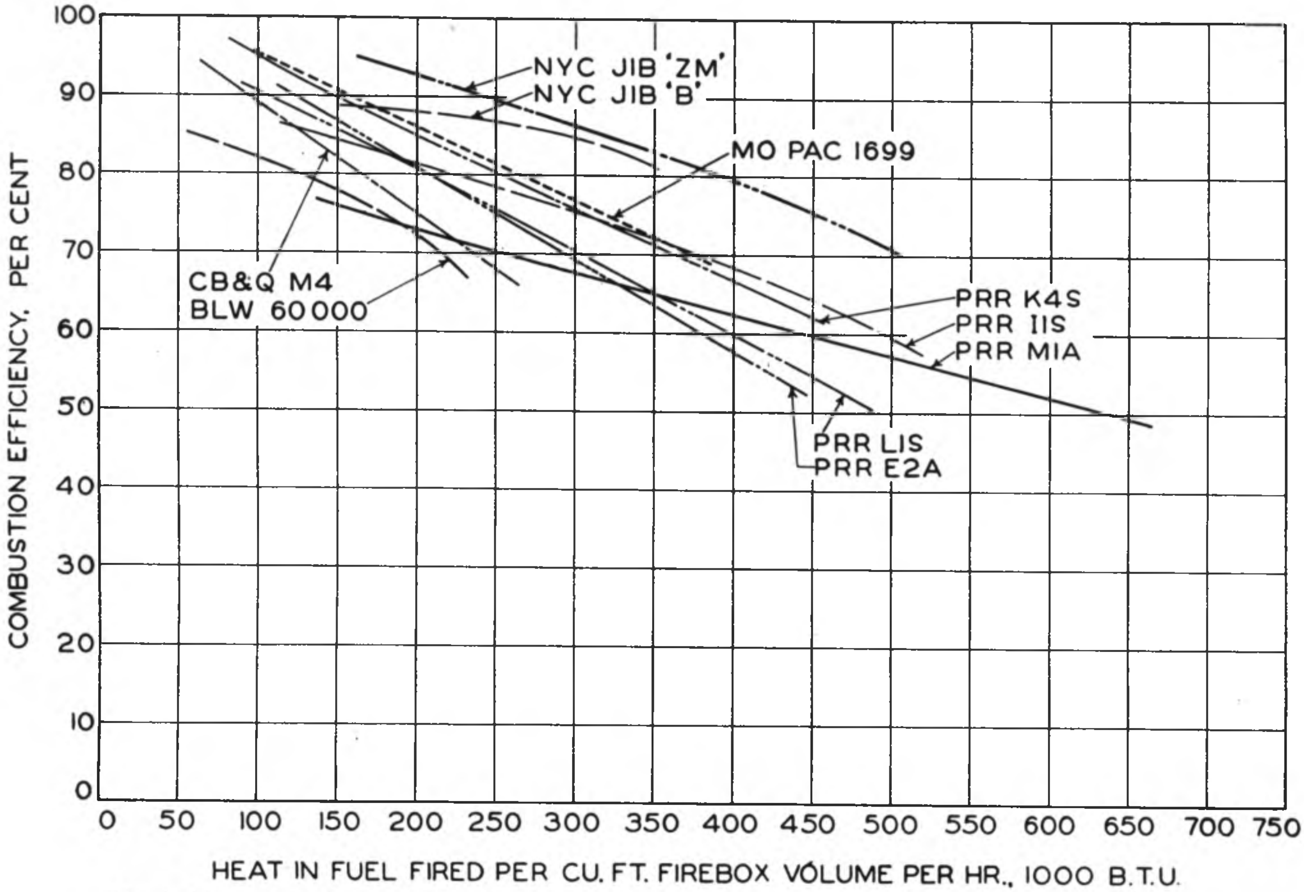

Fig. 8 Combustion Efficiency Versus Heat in Fuel Fired per Cubic Foot of Firebox Volume per Hour

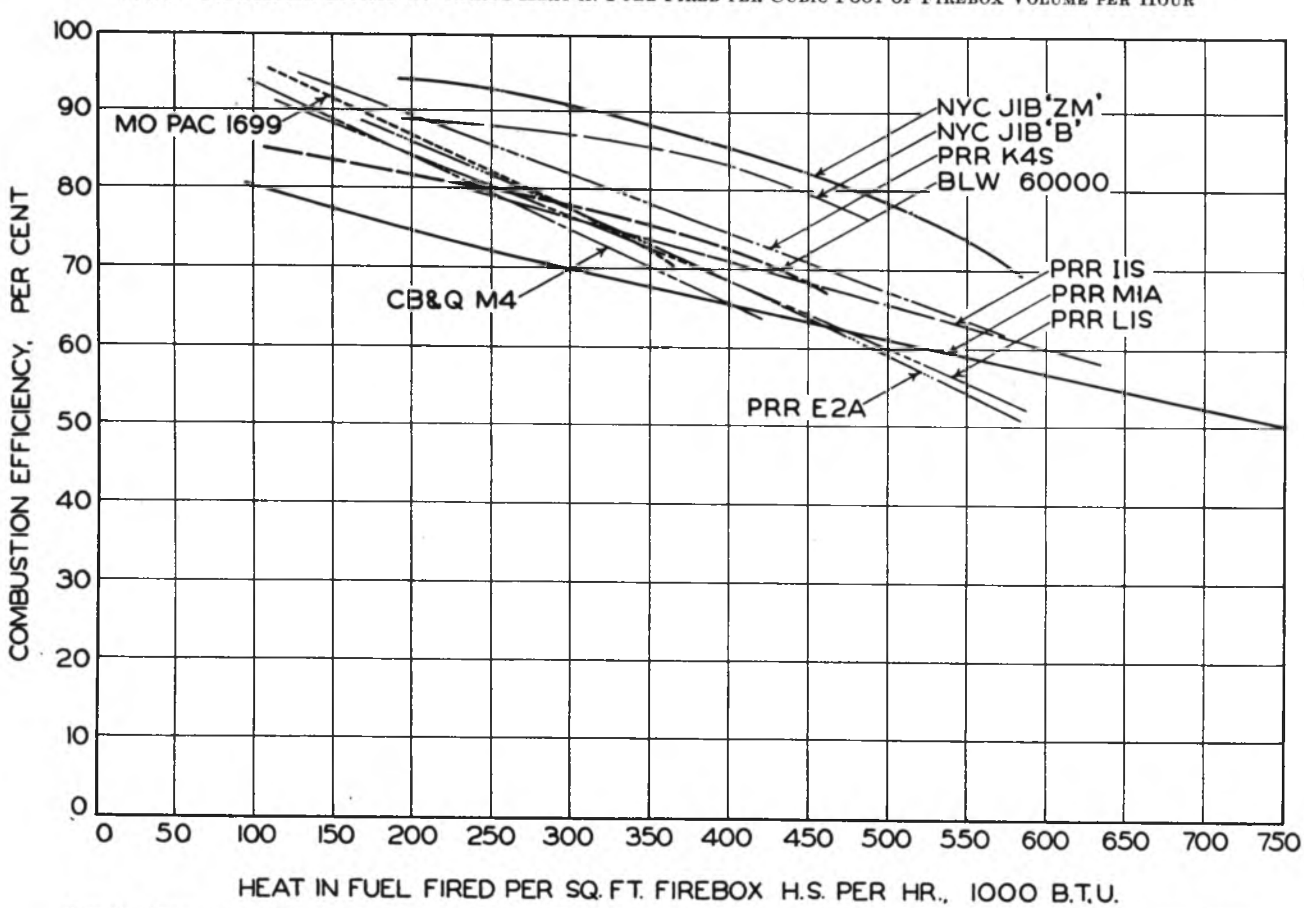

Fig. 9 Combustion Efficiency Versus Heat in Fuel Fired per Square Foot of Firebox Heating Surface per Hour

area. Compared on the basis of tractive power the furnace volume varies from a minimum of 4.04 to a maximum of 11.67 cu ft per 1000 lb of tractive power.

The difference in opinion among locomotive designers as to the correct size of firebox and combustion chamber is due to the absence of pertinent facts and test data to permit a scientific evaluation of the various factors. It is known that the first cost of the heating surfaces in the firebox and combustion chamber is very high as is also the cost of the maintenance and inspection when compared with the tube heating surface. What the relative cost of firebox and tube maintenance amounts to is unknown. Little information is available as to the cost of boiler maintenance, since few American railroads keep accounts to permit such a study. In the report of the Federal Co-Ordinator of Transportation, November, 1935, the estimated average cost of boiler maintenance is given as 30 per cent of the total cost of locomotive repairs. The firebox maintenance is the greatest part of this cost. As a large modern boiler will have from 5000 to 8000 stay bolts in the firebox which must be tested at regular intervals, the size of the firebox is a very important item when the question of intensive utilization of the locomotive is considered.

Against these items of cost should be balanced the great improvements in boiler efficiency and capacity which are generally believed to be obtained by the use of large fireboxes. The heat absorption of the firebox heating surfaces is from six to ten times as great as the average tube heating surface. Another important point not usually considered is that the radiant-heat absorption by the firebox is accomplished without the expenditure of any energy, while the heat absorption by convection in the tubes consumes considerable power to create the high draft required. The energy required to move the gases increases with the fourth power of the weight velocity. Only a few tests have been made which supply data permitting the calculation of combustion efficiency. These tests have been mainly on engines, the boiler ratios of which are very much alike, and thus the effect of greatly different boiler ratios is not clearly discernible from available data.

To illustrate the effect which different boiler ratios have upon combustion efficiency, the data from all tests available have been plotted in Figs. 6 to 9. The combustion efficiency is plotted against various factors as follows:

Fig. 6.... Heat in fuel fired per hr
Fig. 7.... Heat in fuel fired per hr per sq ft of grate area
Fig. 8.... Heat in fuel fired per cu ft firebox volume per hr
Fig. 9.... Heat in fuel fired per sq ft firebox heating surface per hr

It is, however, becoming more generally recognized that the larger the furnace volume of a locomotive, the greater is the opportunity for completing the processes of combustion. This is true only if the larger furnace volume is obtained in conjunction with maximum boiler diameter, furnace width and depth, and not by furnace length alone.

An examination of a typical boiler diagram, Fig. 3, shows the gas velocities at several points in the firebox, an explanation being given why the largest possible cross-sectional area through the firebox is very important. The velocity of the furnace gases in this case between the top of the arch and the crown sheet is as high as 260 miles per hour. It is remarkable that the small particles of coal being blown by the stoker jets into the center of a hurricane of such velocity are able to settle on the grate or have time to burn at all.

A particle of fine coal caught in the gas stream will pass through

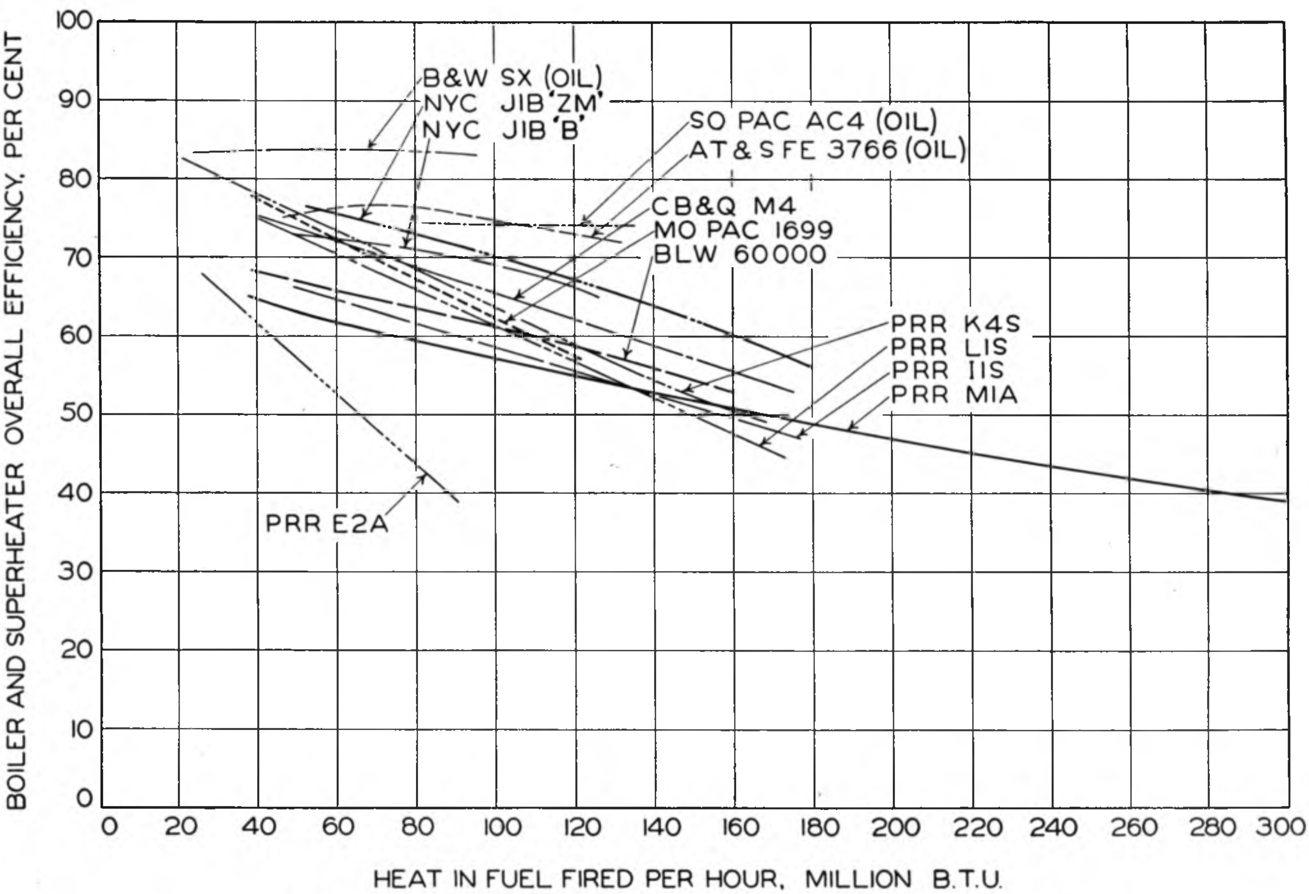

Fig. 10 Over-All Boiler Efficiency Versus Heat in Fuel Fired per Hour

this furnace in 0.1 sec and this is one of the reasons for the great loss of carbon at high-capacity operation. As a comparison, in stationary practice the velocity of the gases just before entering the tube bank is about 10 miles per hour at a maximum. A study of the designs of many recent boilers indicates that the width of the firebox, as well as the depth, could have been increased to give the advantage of a larger combustion volume under the firebrick arch, as well as a larger gas area between the top of the arch and the crown sheet.

A few modern boilers have been built with a firebox ring as low as 53 in. from the rail which with a horizontal grate would give a good depth of the firebox at the rear. On many locomotives with the grate sloping up toward the back end, the distance from the rail to the firebox ring is 72 in., or more, and thus there is a loss of 18 in. in the depth of the firebox. The author has not been able to get a satisfactory reason why sloping grates are still continued, especially since so many boilers have been built with horizontal grates and have given satisfactory service.

While discussing the subject of furnace volume, attention is called to the boiler-efficiency curves plotted in Fig. 10. Three efficiency curves from oil-fired boilers are shown. Two are locomotive boilers and one is a water-tube boiler, tested by the United States Navy.[10]

The following table gives certain pertinent data on maximum-capacity operation:

User	Santa Fe	Southern Pacific	U. S. Navy
Type	4-8-4	AC4	Marine
Engine number	3766	4108	Type SX
Furnace volume, cu ft	795	970	423.6
Gas area through boiler at furnace, sq ft	10.3	11.98	30.06
Maximum oil burned, lb per hr	7000	6800	5051
Maximum oil burned per cu ft furnace volume, lb per hr	8.8	7.0	11.9
Maximum oil burned per sq ft gas area	680	558	168
Maximum coal equivalent, lb per hr	10800	10500	7850

It is interesting to note that it was possible to burn 35 per cent more oil per cu ft of furnace volume with 8 per cent higher boiler efficiency in the Navy boiler than was attainable in the locomotive boiler. The gas area in the Navy boiler at the gas exit of the furnace, however, was three times greater than on the locomotive boiler, which again confirms the importance of gas area.

Absorption Efficiency of the Heating Surfaces

The absorption efficiency of the surfaces should be expressed as the total heat absorbed in per cent of the total heat available for absorption; that is, of the total heat drop from the furnace temperature to the metal temperature of the flues and superheater units at the smokebox. This is termed the true efficiency and is very high, about 90 per cent in a locomotive boiler, due to high gas velocity and cooling of the gases to within 75 F of the temperature of the water at low rating, and 200 F at high capacity.

On locomotives with type E superheaters the smokebox gases are from 100 to 150 deg lower than the steam temperature. The losses due to the sensible heat escaping in smokebox gases range from 19 to 25 per cent of the heat in the coal fired, at high rating, and the only hope of reducing these is by a radical change in the boiler design whereby the average temperature head could be increased by the use of a counterflow economizer. In this manner the smokebox gases could be lowered some 210 F which would give a 7 per cent increase in the over-all boiler efficiency.

The importance of an effective arrangement of the flue heating surfaces is most apparent at high capacity when 70 per cent of the liberated heat is delivered to the flues. The ratio of the net gas area of a flue to its gas-swept perimeter (sometimes called hydraulic depth) determines the gas-carrying capacity of the flues. The larger this ratio becomes the more gas will flow for equal pressure drop. This has been labeled the "flue capacity factor," or F.C.F.

The longer a flue of a given F.C.F. is made, the greater becomes its true efficiency and also its draft loss. Now if the length of the flue in inches is divided by the F.C.F. a value is obtained which expresses the true efficiency of any diameter and length of flue. This factor has been termed the "flue efficiency factor" or F.E.F. The larger it becomes the more efficient a flue is.

In Fig. 11 has been plotted a curve showing the true efficiency that will be obtained on any size and length of flue with a given flue efficiency factor, or F.E.F.; that is, the ratio of length to hydraulic depth.

In Fig. 11 is also shown the true efficiency of several sizes of flues for different lengths. The F.E.F. for a 2-in. inside-diameter flue 18 ft long is 432 and gives a true efficiency of 88 per cent. If this flue is made 25 ft long the F.E.F. becomes 600 and the true efficiency is increased to 94.5 per cent, but the draft loss is increased in the same ratio as the length. In America many boilers have been built with flues 22 ft to 25 ft long, but usually efforts are made to keep the flue lengths down by lengthening the combustion chamber where this can be done. It is evident from Fig. 11 that a larger-diameter flue of a greater length can be used to give the same true efficiency as a smaller flue of shorter length.

The heat-absorption efficiency of locomotive boilers increases with an increase in the flue efficiency factor of the tubes and flues and ranges usually between 78 and 88 per cent. The absorption efficiency of several boilers has been shown in Fig. 12; it is noted that there is a very small decrease in this efficiency as the rate of firing is increased.

The combined absorption and combustion efficiency gives the over-all efficiency of the boiler and superheater and these data are shown in Fig. 10 for the engines previously cited. The general proportions and boiler ratios of all the engines shown in this and previous curves are given in Tables 1 to 6, as follows:

Pennsylvania R.R.	E2A	Table 5	Column 8
Pennsylvania R.R.	K4S	Table 5	Column 6
Pennsylvania R.R.	L1S	Table 5	Column 7
Pennsylvania R.R.	I1S	Table 2	Column 1
Pennsylvania R.R.	M1A	Table 4	Column 10
Chicago, Bur. & Quincy	M4	Table 2	Column 6
Missouri Pacific R.R.	1699	Table 5	Column 9
Baldwin Loco. Works	60000	Table 2	Column 10
New York Central	J1B	Table 1	Column 3
Atchison, Topeka & Santa Fe	3766	Table 5	Column 4
Southern Pacific	AC7	Table 6	Column 10

Heat balances showing the heat absorbed by the boiler and superheater together with the various heat losses for two modern locomotives are shown in Figs. 13 and 14.

Before leaving the subject of heat absorption, a few words may be said as to the heat-absorbing capacity of the firebox. Cole proposed 55 lb of steam per sq ft as the maximum based on Professor Goss's test at Coatesville, the only test made on a locomotive where the firebox evaporation was determined separately. This value of evaporation rate has continued to be used up to now even though the size of the locomotive firebox has increased five to six times since the Coatesville tests. It is reasonable to suppose that the same evaporation rates cannot very well apply to a firebox with 800 sq ft of heating surface having siphons, circulators, etc., as to a firebox with only 150 sq ft.

Since the adoption of waterwalls in stationary boilers, the heat absorption due to radiant heat has received much attention and analysis.

The general behavior of radiant energy as expressed and defined in Stefan-Boltzmann's law, Prevost's law, Kirchhoff's

[10] "The B&W High-Pressure Sectional Express Boiler," by R. C. Brierly, *Journal of the American Society of Naval Engineers*, vol. 43, Nov., 1931, pp. 511–561.

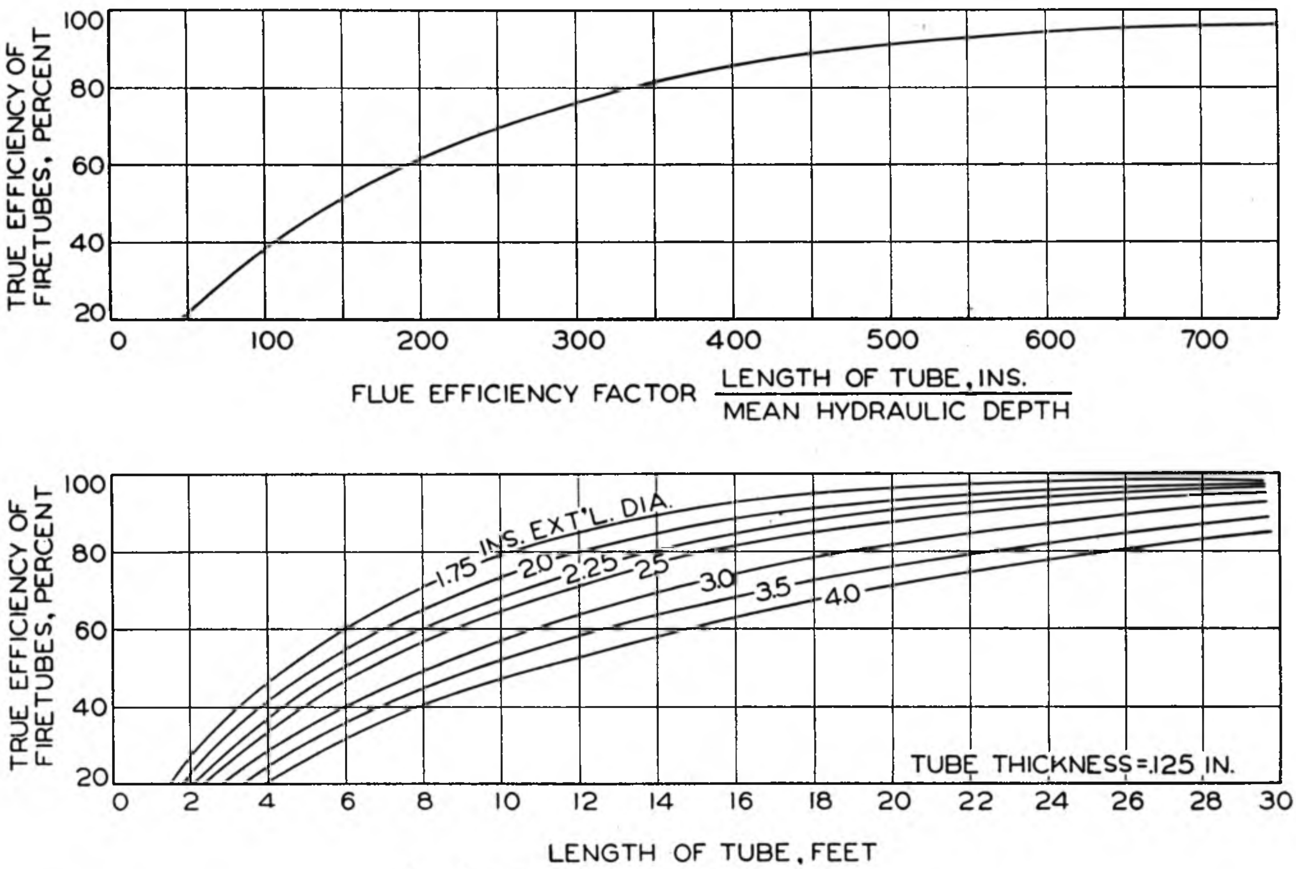

Fig. 11 True Absorption Efficiency of Flues

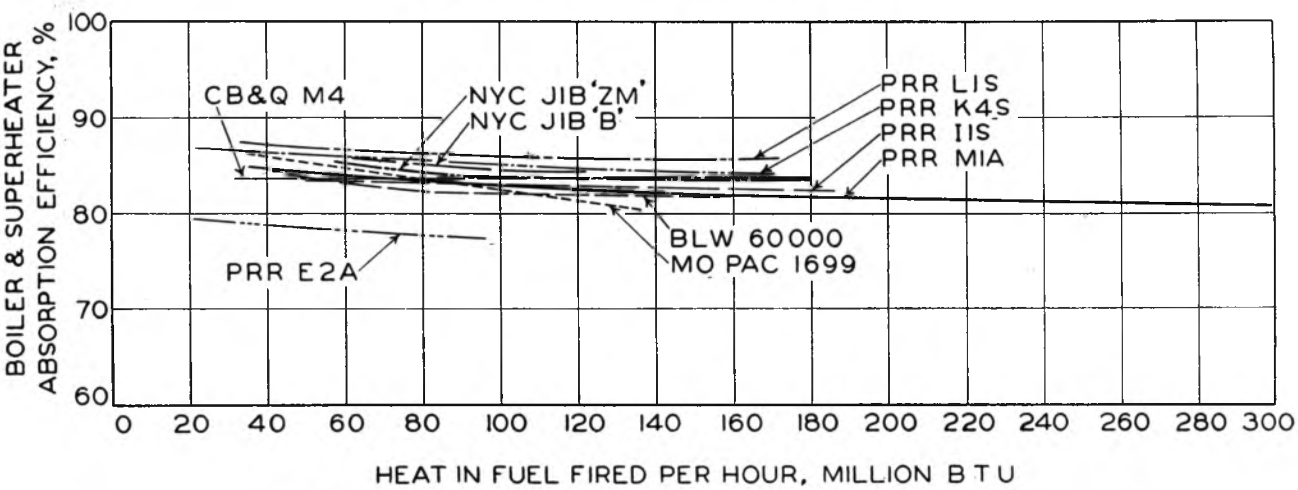

Fig. 12 Absorption Efficiency Versus Heat in Fuel Fired per Hour

law, Planck's formula, and studies by many other investigators, permits definition and determination of the coefficients of emission, absorption, reflection, and transmission of radiant heat. To apply these laws to locomotive fireboxes, additional test data will be required.

The proportion of the total heat liberated in the furnace, which is absorbed by the firebox heating surfaces has been termed "furnace absorption efficiency," and this determines the temperature of the combustion gases as they leave the firebox. An accurate knowledge of this temperature is important as it determines the design and performance of the superheater.

In tests where complete gas analyses have been made to determine the quantities of air used for combustion, it is possible to calculate the per cent of fuel fired that is actually burned and the total heat liberated in the furnace. A fraction of the liberated heat is absorbed by the firebox and the balance leaves as sensible heat in the combustion gases which, with the total known weight of gases, determines the mean temperature of these gases leaving the furnace. The furnace absorption efficiency can thus be stated in terms of the average temperature of the gases leaving the furnace.

Thus, if

t_1 = temperature of combustion gases leaving firebox, F
C_P = mean specific heat of combustion gases between t and 60 F
H_T = total heat liberated in firebox per lb of combustion gases
H_g = heat in 1 lb of combustion gas leaving furnace
H_F = heat absorbed in firebox in per cent of total liberated

then
$$H_g = C_P(t_1 - 60)$$

and the per cent of total heat liberated in the firebox that leaves the furnace in the gas is equal to

$$C_P \frac{(t_1 - 60)}{H_T} \times 100$$

The furnace absorption efficiency, therefore, is

$$H_F = \left[1 - \frac{C_P(t_1 - 60)}{H_T}\right] \times 100$$

The relation between the percentage of heat absorbed by the firebox and the temperature of gases leaving, depends upon the heat liberated per lb of combustion gas, which again depends upon the excess air used for combustion. If, therefore, an accurate measure of the gas temperature leaving the firebox is obtained with a reliable pyrometer the furnace absorption efficiency can be determined.

It has been demonstrated that the radiation from the flames is largely due to incandescent particles of fuel and ash in the flame. The quantity and distribution of this radiating material in the flames, together with the amount of firebox heating surface, determine the coefficient of emission or absorption. This explains why boilers fired with oil give higher superheat; the less luminous flames of the oil fire will give up a smaller percentage of heat to the firebox by radiation, and the gases entering the flues are therefore hotter.

Several formulas have been proposed for the determination of the furnace absorption efficiency. The author has checked a number of the formulas proposed by Hudson, Broido, Wohlenberg, Orrok, Munzinger, Bottomly, and Adloff, but sufficient test data are not available to confirm their methods closely enough for locomotive-firebox conditions. All formulas, however, show that the fraction of the total heat liberated, which is absorbed as radiant heat by the furnace walls, decreases as the total heat liberation is increased. It will range from about 50 to 55 per cent at low rating down to perhaps 25 to 30 per cent at high-capacity operation.

At low rating the heat absorption per square foot of firebox heating surface will be near the figure commonly used, or 55 lb of water evaporated per sq ft but at high capacity the heat absorption will approach a figure of 125 lb of water per sq ft of heating surface. From the studies made of this problem, it is doubtful if the percentage of heat absorption can be increased to much more than 50 to 55 per cent of the heat liberated, by increasing the amount of firebox heating surface, because the amount of radiant energy absorbed is limited by the effective surfaces of the fire bed and the luminosity of the flames.

Grate Area in Relation to Efficiency and Capacity

The size of the grate area should be in proportion to the maximum output expected of the boiler and the quality of the coal to be burned. With the exception of the Dakota and Wyoming lig-

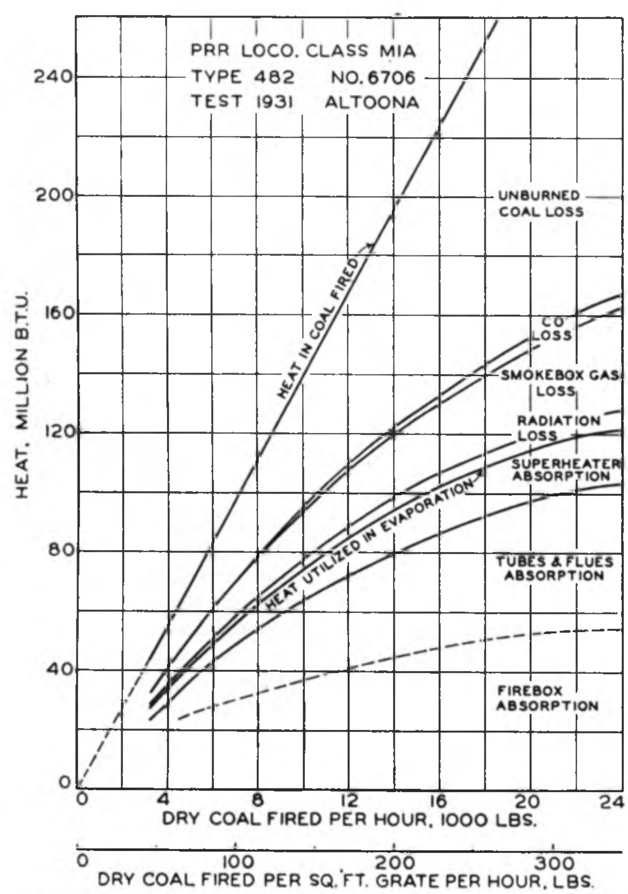

Fig. 13 Heat Balance Versus Dry Coal Fired per Square Foot of Grate per Hour; P.R.R., Class M1A

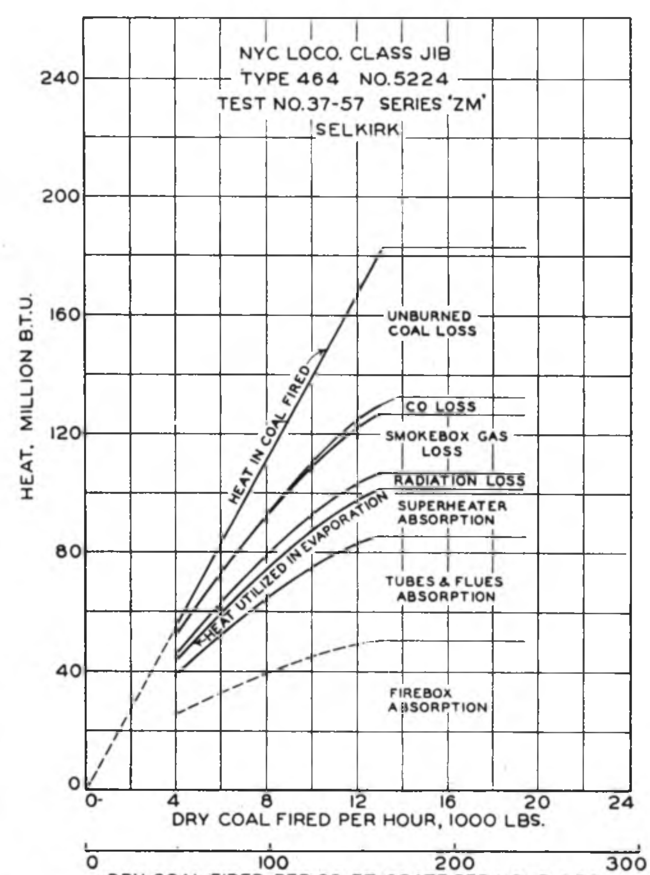

Fig. 14 Heat Balance Versus Dry Coal Fired per Square Foot of Grate per Hour; N.Y.C., Class J1B

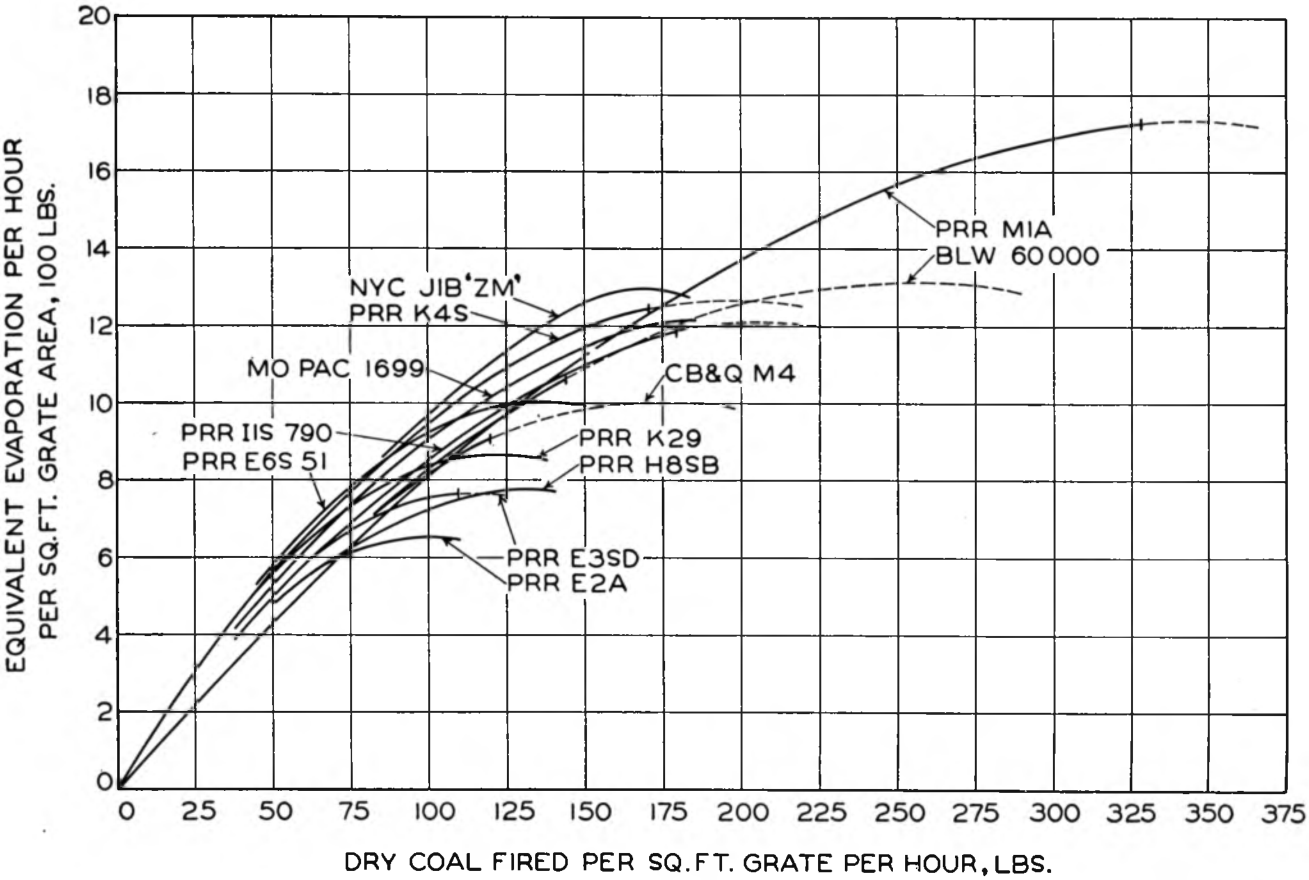

FIG. 15 EQUIVALENT EVAPORATION VERSUS COAL FIRED PER SQUARE FOOT OF GRATE PER HOUR

nites, the average maximum variation in the heating value of the coals mined east of the Mississippi is from 12,000 to 14,000 Btu per lb, and west of the Mississippi from 10,000 to 12,000 Btu per lb, or about 17 per cent for each class of coal.

If the grate-area ratios of modern locomotives, however, are studied it will be noted that they vary to a much greater extent than the heating value of the fuel, as shown in the following table:

	Ratio—Minimum	Ratio—Maximum	Range of difference, per cent
Grate area per 1000 lb tractive effort	0.78	2.32	1:2.97
Grate area per sq ft total heating surface	0.0119	0.0262	1:2.20
Grate area per sq ft firebox heating surface	0.101	0.353	1:3.50
Grate area per sq ft net gas area	7.1	13.8	1:1.94

It would appear that much thought and study must be given to this subject as the proper size of grate area is important in obtaining high furnace efficiency and capacity. If too small a grate is selected, the firing rate will be too high at maximum-capacity operation and, if too large, it is difficult to keep the grate covered at low capacity, which results in bare spots, a high per cent of excess air, reduction of boiler efficiency, and superheat. The service condition and the profile of the road must be carefully analyzed for best results.

The question of the percentage of air openings through the grates has been explored during the last few years. Recent tests on one railroad, where grates with air openings of 10, 24, and 32 per cent were tried, indicated a very small variation in boiler efficiency between the various percentages of air openings, or less than $1^1/_2$ per cent above and below the mean percentage.

The relation between equivalent evaporation per square foot of grate area and coal fired per square foot of grate area per hour has been plotted in Fig. 15.

Secondary Air for Combustion

The length of the firebox and the combustion chamber is of importance, but there is a question whether the function of the latter which gives it its name is operative, as there is no air admitted ahead of the arch on most locomotives to permit the carbon to burn completely in its passage through the combustion chamber.

Today, most modern locomotives in America are fired with a stoker which distributes the crushed coal with steam jets through the boiler backhead. The coal is thus thrown into the firebox through and against the gas stream at the point of its highest velocity. The heaviest coal lumps fall on the grate, but many of the fine coal particles are caught in the high-velocity gas stream and partially burned in suspension while traveling at high speed. During the short time the coal particles pass through the furnace, they must be heated to ignition temperature, the moisture driven off, and the volatile gas distilled and burned before the solid carbon is consumed. It is quite possible that most of the carbon now escaping could be completely burned in the large fireboxes if sufficient oxygen were admitted into the firebox and combustion chamber as secondary air. The beneficial effect of admitting secondary air above the fuel bed has been known for many years and recognized by authorities on combustion as very desirable, yet little has been done along this line.

This subject was investigated by Henry Kreisinger[11] as early

[11] "Combustion of Coal and Design of Furnaces," by Henry Kreisinger, C. E. Augustine, and F. K. Ovitz, Bulletin No. 135, United States Bureau of Mines, Washington, D. C., 1917.

as 1917. In these tests it was proved that, with a firing rate up to 185 lb per sq ft of grate and with the thickness of fuel bed varying from 6 to 12 in., all the oxygen in the air which entered through the grate and fuel bed was consumed at the level of approximately 4 in. above the grate.

A sufficient quantity of secondary air admitted above the fuel bed in such a manner as to create a turbulence in the gases would permit of an intimate mixing of the hydrocarbons thrown off by the coal, and provide the necessary oxygen for the complete combustion of volatile gases and carbon. Another very important result from the use of a correct amount of secondary air is that it will help to eliminate smoke which always has been a serious nuisance in locomotive-boiler operation.

Kreisinger states in his report,[11] "the ratio between the weight of air supplied and the weight of fuel gasified remains practically constant at about seven to one so that only about one half of the fifteen pounds of air required to burn the fuel should be supplied through the fuel bed. In order to complete the combustion of all the fuel fed into the furnace, the balance of the air should be admitted above the fuel bed to consume the volatile, as well as fixed carbon present in this zone, or it will escape unburned."

Any scheme to admit secondary air above the fire should be arranged so that the air is controlled in accordance with the demand, in order to keep the excess air at a minimum and not decrease the fuel-burning capacity of the furnace or decrease the over-all boiler efficiency by too much excess air. Several schemes have been prepared to admit secondary air above the fire. The simplest is that of applying air-inlet thimbles in the sides of the firebox and the effectiveness of this should be fully explored. Some studies have been made of the introduction of preheated air by means of blowers. Actual tests have been conducted on one railroad by using blowers without air preheating, but so far no definite advance has been made. It is possible that the cost of such a scheme may be justified from the standpoint of smoke elimination alone.

Front-End Design and Draft

Because of the very high draft required in a locomotive boiler, the proportion of the draft-making equipment in the smokebox has received much attention in recent years and a great many tests have been conducted to discover the most efficient front-end design.

It is well known that most of the large railroads in the United States have conducted many front-end tests in recent years. The best arrangements arrived at differ considerably in design. The important result, however, is that the efficiencies of the draft arrangements have been increased by handling greater quantities of gases with lower back pressure, and that it has been possible to increase the diameter of the stack to a considerable extent without causing the trailing of smoke, which is undesirable from the standpoint of signal observation.

Curves are given in Fig. 5, which show the equivalent evaporation plotted against back pressure. This curve, test *ZM*, shows the improvements gained by the use of an improved front end developed for the New York Central class J1B locomotives as compared with the standard front end.

Superheater

Of all the improvements made on the locomotive, since George Stephenson invented the first one, the superheater holds the first rank. Its great success has been due to the fact that it attacks the heat cycle at a point where the losses are the greatest, i.e., in the cylinders, by the elimination of cylinder condensation during admission and expansion and re-evaporation at exhaust.

Of the many tests made that have proved the beneficial effects of high superheat, those conducted by the Pennsylvania Railroad have been the most complete and conclusive. In his report[12] on the performance of the superheated passenger locomotive E6s, C. D. Young presented a most thorough analysis of the beneficial effects obtained by the use of superheated steam. The following summary is quoted therefrom:

"The application of the superheater to this locomotive increases its economy from a minimum of 23 per cent to a maximum of 46 per cent, the economy increasing with the increased power required of the locomotive.

"It was found that 30 per cent higher capacity was derived from the E6s locomotive when using superheated steam than with the same size and type of locomotive using saturated steam."

This result was obtained with a comparatively low superheat, of 200 to 225 F, as compared with present-day practice. This showed an average increase in economy of 1.15 to 2 per cent for each 10 F increase in superheat. Experience and tests have shown that this rate of increase in economy holds good up to the limits of present acceptable practice of 350 to 400 F superheat. That every effort must be made to maintain highest superheat temperature is apparent.

This problem has become more and more difficult as the available heat for superheating has been reduced because of the high heat absorption in the large fireboxes of modern locomotives. To compensate for this, much larger superheaters have been found necessary and the introduction of the type E superheater was a natural development. This type has been used on most modern locomotives which have been built during the last fifteen years in America.

Because of the great influence that the superheater design has upon the efficiency and capacity of both the cylinders and the boilers, considerably more space could be devoted to the discussion of superheater design than is here available.

A great deal of research work and testing have been done with many different types of superheater units to discover the unit which will give the maximum superheat with the lowest possible draft loss and steam pressure drop, which is at the same time practical and keeps weight, first cost, and maintenance at a minimum.

The improved type E superheater is a very excellent arrangement as it gives a combination of maximum evaporating and superheating surfaces coupled with the greatest possible gas area through the boiler and steam area through the superheater. This assures high sustained superheat at low as well as high rating with minimum draft and pressure loss.

Feedwater Heating

A feature of great value in obtaining sustained capacity of the boiler, which may only be mentioned here, is the beneficial effect of feedwater heating. The increase in economy or power output of a locomotive ranges from 8 to 14 per cent through the recovery of the heat in the exhaust steam that otherwise would be wasted. The feedwater heater is now well recognized as necessary for economical locomotive operation.

Conclusion

In conclusion it may be said that the modern locomotive boiler is a very effective steam generator. Boilers[13] which can produce highly superheated steam at a rate of 1 lb equivalent evaporation per hr per 1.3 to 1.4 lb of total boiler weight, compare very favorably with stationary boilers of about the same capacity which weigh about four times as much, or 6 lb per lb of steam produced not including stack, breeching, brickwork, or draft fans.

[12] Test Bulletin No. 21, Nov. 4, 1912, pp. 82–89, 150–151.

[13] Class M1A, Pennsylvania Railroad; class J3, New York Central Railroad.

The weight of the locomotive boiler cited includes the boiler complete, superheater, smokebox, front end, stack, stoker, grate, and water.

While continued efforts must be made further to improve the boiler, clearly, it is the engine which deserves the most attention, as its thermal efficiency is so low that even a slight improvement is important. This may mean less simple mechanical parts than in the present reciprocating engine, but the mechanical-maintenance forces of the American railroads have proved to their great credit that very difficult mechanical problems, such as those connected with the maintenance of Diesel and electric locomotives, can be handled successfully.

This gives hope that apparatus and equipment proposed for the improvement of the steam locomotive will receive favorable consideration in the future, and that the hazards of added complications will not be considered as formidable as they once seemed.

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TABLE 1 GENERAL DIMENSIONS AND BOILER RATIOS OF SIX-COUPLED LOCOMOTIVES

	ITEM	UNIT MEASURE	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE			4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4
1	RAILROAD			C.P.R.	C.B.&Q.	N.Y.C.	N.Y.C.	CAN.NAT.	N.Y. N.H.&H.	B & O	A.T.& S.F.	C.M.ST. P.& P.	C.& N.W.
2	CLASS			H-1-C	S-4	J-1-B	STR. LINE J-3-A	K-5-A	I-5	V-2	ENG. NO. 3460	F-7	E-4
3	YEAR BUILT			1937	1930	1927	1938	1930	1937	1935	1937	1938	1938
4	BOILER PRESSURE	LBS. PER SQ. IN.		275	250	225	275	275	285	350	300	300	300
5	CYLINDER DIA. & STROKE	INCHES		22 X 30	25 X 28	25 X 28	22 1/2 X 29	23 X 28	22 X 30	19 X 28	23 1/2 X 29	23 1/2 X 30	25 X 29
6	DIA. DRIVERS	INCHES		75	78	79	79	80	80	84	84	84	84
7	WEIGHT ON DRIVERS	POUNDS		186800	207730	184800	201800	188600	193000	156000	211400	216000	216000
8	TOTAL WEIGHT OF ENGINE	POUNDS		354000	391880	348000	365500	356400	365300	294000	420400	415000	412000
9	TRACTIVE POWER	POUNDS		45300	47700	42400	43440	43300	44000	34000	49300	50300	55000
10	BOILER I.D. FIRST COURSE	INCHES		78 1/2	80 3/16	82 7/16	80 5/8	78	82 7/16	72 OD	86 5/16	82 1/2	88 3/16
11	BOILER O.D. LARGE COURSE	INCHES		90 1/8	94	87 5/8	91 1/2	85 7/8	93		91 7/16	94	94
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		18-3	19-0	20-6	19-0	19-1	18-0	25-0	21-0	19-0	19-0
13	COMBUSTION CHAMBER LENGTH	INCHES		27	36		43	31	42	36		44 1/2	31
14	GRATE AREA	SQ. FT.		80.8	87.9	81.5	82.0	73.7	77.1	61.8	98.5	96.5	90.7
15	TUBE & FLUE HEAT. SURF. (WATER SIDE)	SQ. FT.		3465	3878	4203	3827	3032	3335	2727	4395	3708	3472
16	FIREBOX HEATING SURFACE (WATER SIDE)	SQ. FT.		326	369	281	360	345	480	612	375	458	507
17	TOTAL EVAP. HEAT. SURF. (WATER SIDE)	SQ. FT.		3791	4247	4484	4187	3377	3815	3339	4770.	4166	3979
18	SUPERHEATER SURFACE-STEAM SIDE	SQ. FT.		1542	1830	1965	1745	1492	1042	880	2150	1695	1884
19	FIREBOX VOLUME	CU. FT.		427	494	353	452	362	448		483	503	560
20	GAS AREA THRU BOILER	SQ. FT.		8.38	9.00	9.66	8.90	7.04	8.92	4.61	9.10	9.14	8.07
21	TYPE OF SUPERHEATER			E	E	E	E	E	A	A	E	E	E
22	STEAM AREA THRU SUPERHEATER	SQ. IN.		60.6	65.6	64.2	61.8	52.2	54.7	44.8	71.2	63.6	67.7
23	MAX EVAP. CALCULATED	LBS. PER HR.		55960	61710	57770	60500	51600	64000	55480	64780	65400	65740
24	MAX EVAP. ON TEST	LBS. PER HR.				84800							
25	NUMBER & SIZE OF FLUES			171-3 1/2	184-3 1/2	182-3 1/2	183-3 1/2	146-3 1/2	48-5 1/2	27-5 1/2	200-3 1/2	164-3 3/4	196-3 1/2
26	NUMBER & SIZE OF TUBES			58-2 1/4	62-2 1/4	19-3 1/2 37-2 1/4	59-2 1/4	44-2 1/4	199-2 1/4	120-2 1/4	46-2 1/4	60-2 1/4	8-2
27	NUMBER & SIZE OF SUPERHEATER UNITS			86-1 3/16	93-1 3/16	91-1 3/16	93-1 3/16	74-1 3/16	48-1 1/2	27-1 3/4	101-1 3/16	84-1 1/4	102-1 3/16
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ. FT.	20/9	.185	.189	.228	.206	.163	.203	.136	.185	.182	.147
31	GAS AREA PER SQ. FT. OF GRATE	SQ. FT.	20/14	.104	.103	.119	.109	.096	.116	.075	.093	.095	.089
32	GAS AREA PER SQ. FT. OF TUBE AND FLUE H.S.	SQ. FT.	20/15	.00242	.00232	.00230	.00233	.00232	.00268	.00169	.00207	.00246	.00233
33	GRATE PER 1000 LBS. TRACT. POWER	SQ. FT.	14/9	1.79	1.85	1.92	1.89	1.71	1.75	1.82	2.00	1.92	1.65
34	GRATE PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	14/17	.0214	.0208	.0181	.0196	.0219	.0202	.0185	.0207	.0232	.0228
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU. FT.	19/9	9.43	10.60	8.34	10.40	8.35	10.20		9.80	10.00	10.20
36	FIREBOX VOLUME PER SQ. FT. OF GAS AREA	CU. FT.	19/20	51.0	54.9	36.6	50.8	51.4	50.2		53.1	55.1	69.4
37	FIREBOX VOLUME PER SQ. FT. OF GRATE	CU. FT.	19/14	5.29	5.61	4.34	5.51	4.91	5.81		4.90	5.22	6.16
38	FIREBOX VOLUME PER SQ. FT. TOTAL EVAP. H.S.	CU. FT.	19/17	113.0	116.2	78.5	108.0	107.5	117.5		101.1	121.0	141.0
39	FIREBOX H.S. PER SQ. FT. OF GRATE	SQ. FT.	16/14	4.03	4.20	3.45	4.40	4.68	6.23	9.90	3.81	4.75	5.59
40	FIREBOX H.S. PER SQ. FT. TOTAL EVAP. H.S. ×100	SQ. FT.	16/17 ×100	8.60	8.70	6.25	8.60	10.2	12.60	18.35	7.85	11.0	12.75
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ. FT.	17/9	83.6	89.0	105.9	96.3	77.8	86.6	98.1	96.8	82.9	72.4
42	TOTAL EVAP. H.S. PER SQ. FT. OF GRATE	SQ. FT.	17/14	46.9	48.3	55.1	51.1	45.8	49.5	54.0	48.5	43.3	43.8
43	SUPERHEATER H.S. PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	18/17	.407	.431	.438	.417	.442	.274	.264	.450	.406	.474
44	PERCENT GAS AREA THRU FLUES			84.8	85.0	80.1	85.5	86.1	51.0	42.6	89.0	86.1	98.3
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	7/9	4120	4350	4360	4640	4350	4390	4590	4285	4300	3925
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	8/9	7810	8200	8210	8425	8245	8300	8640	8550	8250	7490

TABLE 2 GENERAL DIMENSIONS AND BOILER RATIOS OF TEN-COUPLED LOCOMOTIVES

	ITEM	UNIT	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE	MEASURE		2-10-0	2-10-2	2-10-2	2-10-4	2-10-4	2-10-4	2-10-4	2-10-4	2-10-4	4-10-2
1	RAILROAD			P.R.R.	CAN. NAT.	N.Y.NH.&H.	CENT. VER.	C.P.R.	C.B.&Q.	C.&O.	K.C.S.	A.T.&S.F.	B.L.W.
2	CLASS			*790 I1S	T 4B	L1C	T 3A	T1B	*6320 M 4	T 1	J	5001	60000
3	YEAR BUILT			1922	1930	1929	1928	1938	1927	1930	1937	1938	1926
4	BOILER PRESSURE	LBS. PER SQ. IN.		250	275	200	250	285	250	260	310	310	350
5	CYLINDER DIA. & STROKE	INCHES		$30\frac{1}{2}$ X 32	24 X 28	30 X 32	27 X 32	25 X 32	31 X 32	29 X 34	27 X 34	30 X 34	1HP-2LP 27 X 32
6	DIA. DRIVERS	INCHES		62	57	63	60	63	64	69	70	74	$63\frac{1}{2}$
7	WEIGHT ON DRIVERS	POUNDS		352500	261040	301800	285000	309900	353820	373000	350000	371680	338400
8	TOTAL WEIGHT OF ENGINE	POUNDS		386100	344170	363325	419000	447000	512110	566000	509000	545260	457500
9	TRACTIVE POWER	POUNDS		90024	61600	77800	76800	76905	90000	91584	93300	93000	82500
10	BOILER I.D. FIRST COURSE	INCHES		82	$74\frac{5}{8}$	88	$84\frac{1}{2}$	$82\frac{1}{2}$	90	98	90	$92\frac{1}{4}$	$81\frac{3}{8}$
11	BOILER O.D. LARGE COURSE	INCHES		93	86	$92\frac{13}{16}$	94	$96\frac{1}{2}$	104	108	102	104	94
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		19-1	19-3	18-0	22-0	21-0	21-6	21-0	21-0	21-0	23-0
13	COMBUSTION CHAMBER LENGTH	INCHES		$42\frac{1}{4}$	37	—	48	54	$49\frac{1}{2}$	66	75	72	
14	GRATE AREA	SQ. FT.		70.0	66.7	82.0	84.4	93.5	106.5	121.7	107.0	121.5	82.5
15	TUBE & FLUE HEAT. SURF.	SQ. FT.		4303	3059	3951	4280	4642	5455	5990	4654	5443	4420
16	FIREBOX HEATING SURFACE WATER SIDE	SQ. FT.		287	347	454	423	412	449	645	500	632	342
17	TOTAL EVAP. HEAT. SURF.	SQ. FT.		4590	3406	4405	4703	5054	5904	6635	5154	6075	4762
18	SUPERHEATER SURFACE-STEAM SIDE	SQ. FT.		1575	1500	1945	2220	2032	2487	3030	2075	2675	1357
19	FIREBOX VOLUME	CU. FT.		364	386	422	458		670	826	722	745	683
20	GAS AREA THRU BOILER	SQ. FT.		9.90	7.05	9.95	8.70	9.75	11.10	12.75	10.35	11.55	9.37
21	TYPE OF SUPERHEATER			E	E	E	E	E	E	E	E	E	A
22	STEAM AREA THRU SUPERHEATER	SQ. IN.		49.0	52.2	71.9	69.1	65.1	78.3	97.3	70.4	89.6	56.9
23	MAX EVAP. CALCULATED	LBS. PER HR.		56660	51750	69000	65110	69120	78520	96210	74680	89140	60410
24	MAX EVAP. ON TEST	LBS. PER HR.		65257		65750			78710				69695
25	NUMBER & SIZE OF FLUES			170-$3\frac{1}{2}$	146-$3\frac{1}{2}$	204-$3\frac{1}{2}$	192-$3\frac{1}{2}$	196-$3\frac{1}{2}$	222-$3\frac{1}{2}$	275-$3\frac{1}{2}$	183-$3\frac{3}{4}$	249-$3\frac{1}{2}$	50-$5\frac{1}{2}$
26	NUMBER & SIZE OF TUBES			120-$2\frac{1}{4}$	44-$2\frac{1}{4}$	57-$2\frac{1}{4}$	33-$2\frac{1}{4}$	72-$2\frac{1}{4}$	87-$2\frac{1}{4}$	59-$2\frac{1}{4}$	73-$2\frac{1}{4}$	56-$2\frac{1}{4}$	206-$2\frac{1}{4}$
27	NUMBER & SIZE OF SUPERHEATER UNITS			85-$1\frac{1}{8}$	74-$1\frac{3}{16}$	102-$1\frac{3}{16}$	98-$1\frac{3}{16}$	98-$1\frac{3}{16}$	111-$1\frac{1}{16}$	138-$1\frac{3}{16}$	93-$1\frac{1}{4}$	127-$1\frac{3}{16}$	50-$1\frac{1}{2}$
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{20}{9}$	.110	.114	.128	.113	.127	.123	.139	.111	.124	.114
31	GAS AREA PER SQ. FT. OF GRATE	SQ. FT.	$\frac{20}{14}$	.141	.106	.122	.103	.104	.104	.105	.097	.095	.114
32	GAS AREA PER SQ. FT. OF TUBE AND FLUE H.S.	SQ. FT.	$\frac{20}{15}$	.00230	.00231	.00252	.00203	.00211	.00204	.00213	.00222	.00212	.00212
33	GRATE PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{14}{9}$	.78	1.08	1.06	1.10	1.22	1.18	1.33	1.15	1.31	1.00
34	GRATE PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	$\frac{14}{17}$	.0153	.0196	.0186	.0179	.0185	.0181	.0184	.0207	.0200	.0173
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU. FT.	$\frac{19}{9}$	4.04	6.26	5.42	5.96		7.44	9.02	7.75	8.01	8.29
36	FIREBOX VOLUME PER SQ. FT. OF GAS AREA	CU. FT.	$\frac{19}{20}$	36.8	54.8	42.4	52.6		60.3	64.9	69.8	64.5	73.0
37	FIREBOX VOLUME PER SQ. FT. OF GRATE	CU. FT.	$\frac{19}{14}$	5.20	5.78	5.15	5.43		6.28	6.79	6.75	6.13	8.28
38	FIREBOX VOLUME PER SQ. FT. TOTAL EVAP. H.S.	CU. FT.	$\frac{19}{17}$	79.3	113.5	95.8	97.4		113.5	124.7	140.0	112.5	144.0
39	FIREBOX H.S. PER SQ. FT. OF GRATE	SQ. FT.	$\frac{16}{14}$	4.10	5.20	5.54	5.02	4.41	4.22	5.31	4.67	5.20	4.15
40	FIREBOX H.S. PER SQ. FT. TOTAL EVAP. H.S. ×100	SQ. FT.	$\frac{16}{17}$×100	6.25	10.20	10.30	8.98	8.13	7.61	9.72	9.70	10.40	7.18
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{17}{9}$	51.0	55.3	56.7	61.2	65.8	65.6	72.4	55.3	65.4	57.7
42	TOTAL EVAP. H.S. PER SQ. FT. OF GRATE	SQ. FT.	$\frac{17}{14}$	65.5	51.1	53.7	55.8	54.1	55.4	54.5	48.2	50.0	57.8
43	SUPERHEATER H.S. PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	$\frac{18}{17}$	.343	.440	.441	.472	.401	.422	.457	.402	.440	.285
44	PERCENT GAS AREA THRU FLUES			73.2	86.1		91.7	83.7	83.2	89.6	84.8	89.5	51.6
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{7}{9}$	3920	4240	3870	3710	4020	3925	4070	3775	3985	4100
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{8}{9}$	4290	5290	4670	*5460	5814	5690	6185	5460	5870	5540

TABLE 3 GENERAL DIMENSIONS AND BOILER RATIOS OF EIGHT-COUPLED LOCOMOTIVES

	ITEM	UNIT MEASURE	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE			4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4
1	RAILROAD			N.C.&ST.L.	D.&R.G.W.	ST.L-SW.	L.V.	WABASH	C.&O.	CAN.NAT.	SOU.PAC.	G.T. NOR.	D.&R.G.W.
2	CLASS			J 2	M 64	L 1	T 2	O 1	J 3	U 2 D	G S 2	S 1	M 68
3	YEAR BUILT			1930	1929	1937	1931	1930	1935	1936	1936	1929	1937
4	BOILER PRESSURE	LBS. PER SQ. IN.		250	240	250	255	250	250	250	250	250	285
5	CYLINDER DIA. & STROKE	INCHES		25 X 30	27 X 30	26 X 30	26 X 32	27 X 32	$27\frac{1}{2}$ X 30	$25\frac{1}{2}$ X 30	27 X 30	28 X 30	26 X 30
6	DIA. DRIVERS	INCHES		70	70	70	70	70	72	73	$73\frac{1}{2}$	73	73
7	WEIGHT ON DRIVERS	POUNDS		220000	252000	248000	268000	274100	273000	237600	266500	273700	279172
8	TOTAL WEIGHT OF ENGINE	POUNDS		381000	408500	425500	422000	454090	477000	390000	448400	472120	479360
9	TRACTIVE POWER	POUNDS		57000	63700	61500	66700	70750	66960	56800	62200	67000	67200
10	BOILER I.D. FIRST COURSE	INCHES		79	$82\frac{15}{16}$	$84\frac{1}{2}$	$84\frac{1}{4}$	$86\frac{1}{2}$	90	$80\frac{7}{8}$	$84\frac{1}{8}$	$86\frac{1}{16}$	$90\frac{3}{16}$
11	BOILER O.D. LARGE COURSE	INCHES		92	96	98	98	100	100	90	96	98	100
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		20-6	22-0	20-0	21-6	21-0	21-0	21-6	21-6	22-0	21-0
13	COMBUSTION CHAMBER LENGTH	INCHES		54	42	54	54	50	54	$48\frac{1}{2}$	60	52	72
14	GRATE AREA	SQ. FT.		77.3	88.0	88.3	88.3	96.2	100.0	84.3	90.4	102.0	106.0
15	TUBE & FLUE HEAT. SURF. (WATER SIDE)	SQ. FT.		3751	4473	4259	4933	4694	5013	3805	4502	5004	4952
16	FIREBOX HEATING SURFACE (WATER SIDE)	SQ. FT.		444	446	469	508	495	525	415	350	401	555
17	TOTAL EVAP. HEAT. SURF. (WATER SIDE)	SQ. FT.		4195	4919	4728	5441	5189	5538	4220	4852	5405	5507
18	SUPERHEATER SURFACE - STEAM SIDE	SQ. FT.		1837	2229	1962	2256	2360	2342	1760	2086	2420	2336
19	FIREBOX VOLUME	CU. FT.		478	402		567		642	408	559		690
20	GAS AREA THRU BOILER	SQ. FT.		8.11	9.04	9.24	10.10	9.70	10.60	7.86	9.13	10.08	10.43
21	TYPE OF SUPERHEATER			E	E	E	E	E	E	E	E	E	E
22	STEAM AREA THRU SUPERHEATER	SQ. IN.		63.5	69.8	67.7	71.9	75.4	74.4	56.4	67.1	74.0	74.4
23	MAX EVAP. CALCULATED	LBS. PER HR.		63040	68280	70100	78500	74780	79640	60810	63575	70570	80800
24	MAX EVAP. ON TEST	LBS. PER HR.											
25	NUMBER & SIZE OF FLUES			$169-3\frac{1}{2}$	$195-3\frac{1}{2}$	$200-3\frac{1}{2}$	$202-3\frac{1}{2}$	$214-3\frac{1}{2}$	$220-3\frac{1}{2}$	$167-3\frac{1}{2}$	$198-3\frac{1}{2}$	$210-3\frac{1}{2}$	$222-3\frac{1}{2}$
26	NUMBER & SIZE OF TUBES			$49-2\frac{1}{4}$	$43-2\frac{1}{4}$	$52-2\frac{1}{4}$	$77-2\frac{1}{4}$	$49-2\frac{1}{4}$	$65-2\frac{1}{4}$	$42-2\frac{1}{4}$	$49-2\frac{1}{4}$	$61-2\frac{1}{4}$	$57-2\frac{1}{4}$
27	NUMBER & SIZE OF SUPERHEATER UNITS			$86-1\frac{3}{16}$	$99-1\frac{3}{16}$	$102-1\frac{3}{16}$	$102-1\frac{3}{16}$	$107-1\frac{3}{16}$	$112-1\frac{3}{16}$	$85-1\frac{3}{16}$	$101-1\frac{3}{16}$	$105-1\frac{3}{16}$	$112-1\frac{3}{16}$
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{20}{9}$	.142	.142	.151	.151	.137	.158	.139	.147	.150	.155
31	GAS AREA PER SQ. FT. OF GRATE	SQ. FT.	$\frac{20}{14}$	.105	.103	.105	.114	.101	.106	.093	.101	.099	.099
32	GAS AREA PER SQ. FT. OF TUBE AND FLUE H.S.	SQ. FT.	$\frac{20}{15}$	.00216	.00202	.00217	.00205	.00207	.00211	.00207	.00203	.00201	.00211
33	GRATE PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{14}{9}$	1.36	1.38	1.44	1.33	1.36	1.50	1.48	1.45	1.52	1.58
34	GRATE PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	$\frac{14}{17}$	.0184	.0179	.0187	.0163	.0186	.0188	.0200	.0186	.0188	.0192
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU. FT.	$\frac{19}{9}$	8.38	6.31		8.50		9.60	7.15	9.00		10.30
36	FIREBOX VOLUME PER SQ. FT. OF GAS AREA	CU. FT.	$\frac{19}{20}$	58.9	44.5		56.1		60.6	51.7	61.3		66.1
37	FIREBOX VOLUME PER SQ. FT. OF GRATE	CU. FT.	$\frac{19}{14}$	6.18	4.57		6.43		6.42	4.83	6.19		6.50
38	FIREBOX VOLUME PER SQ. FT. TOTAL EVAP. H.S.	CU. FT.	$\frac{19}{17}$	114.1	81.8		104.1		116.0	96.4	115.2		125.2
39	FIREBOX H.S. PER SQ. FT. OF GRATE	SQ. FT.	$\frac{16}{14}$	5.74	5.07	5.33	5.75	5.15	5.25	4.93	3.88	3.93	5.24
40	FIREBOX H.S. PER SQ. FT. TOTAL EVAP. H.S. ×100	SQ. FT.	$\frac{16}{17}\times 100$	10.60	9.05	9.92	9.33	9.52	9.50	9.84	7.20	7.40	10.07
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ. FT.	$\frac{17}{9}$	73.6	77.3	76.9	81.5	73.3	82.7	74.3	78.2	80.8	81.9
42	TOTAL EVAP. H.S. PER SQ. FT. OF GRATE	SQ. FT.	$\frac{17}{14}$	54.3	55.9	53.5	61.6	54.0	55.4	50.1	53.8	53.0	51.9
43	SUPERHEATER H.S. PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	$\frac{18}{17}$	.438	.453	.415	.414	.455	.424	.417	.429	.448	.424
44	PERCENT GAS AREA THRU FLUES			87.0	85.4	88.0	83.2	89.3	86.4	88.0	88.0	86.5	88.3
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{7}{9}$	3860	3950	4030	4020	3880	4077	4180	4280	4080	4150
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{8}{9}$	6680	6420	6920	6325	6420	6220	6865	7220	7050	7140

TABLE 4 GENERAL DIMENSIONS AND BOILER RATIOS OF EIGHT-COUPLED LOCOMOTIVES

	ITEM	UNIT MEASURE	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE			4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-2
1	RAILROAD			D.L.&W.	C.M.ST.P.&P.	C.B.&Q.	C.M.ST P.&P.	CAN.PAC.	L.V.	R.F.&P.	NOR.PAC.	U.P.	P.R.R.
2	CLASS			1631-1650	S 1	O 5	S 2	K 1	T3 5129	551-555	A 3	800-819	M1A 6706
3	YEAR BUILT			1934	1930	1938	1937	1928	1935	1936	1938	1937	1931
4	BOILER PRESSURE	LBS. PER SQ.IN.		250	230	250	285	275	275	275	260	300	250
5	CYLINDER DIA. & STROKE	INCHES		28X32	28X30	28X30	26X32	$25\frac{1}{2}$X30	27X30	27X30	28X31	$24\frac{1}{2}$X32	27X30
6	DIA. DRIVERS	INCHES		74	74	74	74	75	77	77	77	77	72
7	WEIGHT ON DRIVERS	POUNDS		274000	258814	282700	282320	249000	270100	277245	294000	270000	271000
8	TOTAL WEIGHT OF ENGINE	POUNDS		447000	450838	465300	490450	423000	441440	466040	491800	465000	390000
9	TRACTIVE POWER	POUNDS		72000	60000	67500	70800	60800	66500	66500	69800	63600	64550
10	BOILER I.D. FIRST COURSE	INCHES		$84\frac{1}{4}$	$86\frac{3}{16}$	$86\frac{1}{2}$	$90\frac{3}{16}$	$82\frac{1}{2}$	$84\frac{9}{16}$	86	$86\frac{3}{8}$	$86\frac{3}{16}$	82
11	BOILER O.D. LARGE COURSE	INCHES		95	100	100	100	$96\frac{1}{2}$	$98\frac{1}{4}$	$98\frac{1}{4}$	99	100	96
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		21-6	21-0	21-0	21-0	20-6	21-6	21-0	19-6	20-6	19-1
13	COMBUSTION CHAMBER LENGTH	INCHES		$66\frac{1}{16}$	60	40	72	60	48	78	$90\frac{1}{2}$	72	98
14	GRATE AREA	SQ.FT.		88.2	103.0	106.5	106.0	93.5	96.5	96.3	115.0	100.2	70
15	TUBE & FLUE HEAT. SURF. (WATER SIDE)	SQ.FT.		4992	4860	4804	4931	4509	4932	4827	4202	4118	4303
16	FIREBOX HEATING SURFACE (WATER SIDE)	SQ.FT.		496	540	433	578	422	507	551	553	479	403
17	TOTAL EVAP. HEAT. SURF. (WATER SIDE)	SQ.FT.		5488	5400	5237	5509	4931	5439	5378	4755	4597	4706
18	SUPERHEATER SURFACE - STEAM SIDE	SQ.FT.		2180	2403	2300	2236	2112	2056	2130	2026	1458	1630
19	FIREBOX VOLUME	CU.FT.		534	630	650	700	620	540	640	812	675	475
20	GAS AREA THRU BOILER	SQ.FT.		10.20	10.30	10.15	11.10	9.86	10.05	10.05	10.18	9.67	9.71
21	TYPE OF SUPERHEATER			E	E	E	E	E	E	E	E	A	E
22	STEAM AREA THRU SUPERHEATER	SQ.IN.		67.7	76.9	72.4	77.3	69.0	67.7	69.1	75.3	66.0	56.4
23	MAX EVAP. CALCULATED	LBS. PER HR.		76905	79050	72160	82000	69140	76990	79385	75280	68700	67850
24	MAX EVAP. ON TEST	LBS. PER HR.											99095
25	NUMBER & SIZE OF FLUES			202-$3\frac{1}{2}$	218-$3\frac{1}{2}$	218-$3\frac{1}{2}$	201-$3\frac{3}{4}$	196-$3\frac{1}{2}$	202-$3\frac{1}{2}$	205-$3\frac{1}{2}$	186-$3\frac{3}{4}$	58-$5\frac{1}{2}$	170-$3\frac{1}{2}$
26	NUMBER & SIZE OF TUBES			82-$2\frac{1}{4}$	56-$2\frac{1}{4}$	51-$2\frac{1}{4}$	66-$2\frac{1}{4}$	59-2 1/4 7-3 1/2	77-$2\frac{1}{4}$	73-$2\frac{1}{4}$	58-$2\frac{1}{4}$	201-$2\frac{1}{4}$	120-$2\frac{1}{4}$
27	NUMBER & SIZE OF SUPERHEATER UNITS			102-$1\frac{3}{16}$	109-$1\frac{3}{16}$	109-$1\frac{3}{16}$	102-$1\frac{1}{4}$	98-$1\frac{3}{16}$	102-$1\frac{3}{16}$	104-$1\frac{3}{16}$	94-$1\frac{1}{4}$	58-$1\frac{1}{2}$	85-$1\frac{3}{16}$
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{20}{9}$	.142	.172	.151	.157	.162	.151	.151	.145	.152	.151
31	GAS AREA PER SQ.FT. OF GRATE	SQ.FT.	$\frac{20}{14}$	.115	.100	.095	.105	.105	.104	.104	.089	.097	.139
32	GAS AREA PER SQ.FT. OF TUBE AND FLUE H.S.	SQ.FT.	$\frac{20}{15}$	.00204	.00212	.00211	.00225	.00219	.00204	.00208	.00242	.00235	.00226
33	GRATE PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{14}{9}$	1.23	1.72	1.58	1.50	1.54	1.45	1.45	1.65	1.58	1.09
34	GRATE PER SQ.FT. TOTAL EVAP. H.S.	SQ.FT.	$\frac{14}{17}$	.0161	.0191	.0204	.0192	.0190	.0177	.0179	.0242	.0218	.0149
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU.FT.	$\frac{19}{9}$	7.42	10.50	9.63	9.90	10.20	8.12	9.63	11.67	10.60	7.35
36	FIREBOX VOLUME PER SQ.FT. OF GAS AREA	CU.FT.	$\frac{19}{20}$	52.4	61.1	64.0	63.1	62.8	53.8	63.7	79.5	69.8	49.0
37	FIREBOX VOLUME PER SQ.FT. OF GRATE	CU.FT.	$\frac{19}{14}$	6.05	6.11	6.10	6.60	6.63	5.60	6.67	7.06	6.75	6.80
38	FIREBOX VOLUME PER SQ.FT. TOTAL EVAP. H.S.	CU.FT.	$\frac{19}{17}$	97.2	116.0	124.0	127.1	126.0	99.3	119.0	171.0	147.0	101.0
39	FIREBOX H.S. PER SQ.FT. OF GRATE	SQ.FT.	$\frac{16}{14}$	5.64	5.24	4.06	5.46	4.52	5.26	5.72	4.80	4.79	5.75
40	FIREBOX H.S. PER SQ.FT. TOTAL EVAP. H.S. ×100	SQ.FT.	$\frac{16}{17}\times100$	9.03	10.00	8.27	10.50	8.56	9.34	10.25	11.60	10.42	8.55
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{17}{9}$	76.2	90.0	77.5	77.8	81.1	81.6	81.0	68.1	72.3	72.8
42	TOTAL EVAP. H.S. PER SQ.FT. OF GRATE	SQ.FT.	$\frac{17}{14}$	62.2	52.4	49.2	52.0	52.7	56.4	56.0	41.4	46.0	67.2
43	SUPERHEATER H.S. PER SQ.FT. TOTAL EVAP. H.S.	SQ.FT.	$\frac{18}{17}$	.398	.445	.440	.406	.427	.378	.396	.422	.317	.346
44	PERCENT GAS AREA THRU FLUES			82.2	88.0	89.2	87.0	82.8	83.2	84.0	87.8	55.5	73.0
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{7}{9}$	3800	4315	4185	3990	4090	4065	4170	4220	4250	4200
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{8}{9}$	6210	7515	6890	6930	6960	6650	7010	7040	7320	6040

TABLE 5 GENERAL DIMENSIONS AND BOILER RATIOS OF EIGHT-COUPLED LOCOMOTIVES

	ITEM	UNIT MEASURE	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE			4-8-4	4-8-4	4-8-4	4-8-4	6-4-4-6	4-6 2	2-8-2	4-4-2	2-8-2	4-8-4
1	RAILROAD			GT. NOR.	SOU. PAC.	A.C.L.	A.T.&S.F.	P. R.R.	P. R.R.	P. R. R.	P. R. R.	MO. PAC.	U. P.
2	CLASS			S-2	G-S-3	R-1	3765	S-1	#1737 K-4-S	#1752 L-1-S	#5266 E-2-A	ENG. NO. 1699	820-834
3	YEAR BUILT			1930	1937	1938	1938	1939	1914	1914	1904	1925	1939
4	BOILER PRESSURE	LBS. PER SQ. IN.		225	280	275	300	300	205	205	205	200	300
5	CYLINDER DIA. & STROKE	INCHES		29X29	26 X32	27X30	28X32	4 CYL. 22X26	27X28	27 X30	20½X26	2-23X32 1-23X28	25X32
6	DIA. DRIVERS	INCHES		80	80	80	80	84	80	62	80	63	80
7	WEIGHT ON DRIVERS	POUNDS		247300	267300	263127	286890	281440	202880	235800	110001	244500	270000
8	TOTAL WEIGHT OF ENGINE	POUNDS		420900	460000	460270	499600	608170	309140	315600	184167	340000	483000
9	TRACTIVE POWER	POUNDS		58300	62800	63900	66000	76400	44460	61500	23880	65700	63800
10	BOILER I.D. FIRST COURSE	INCHES		82 11/16	84 1/4	84 1/2	88 1/4	91 1/16	76 5/8	76 5/8	65 5/8	88	86 3/16
11	BOILER O.D. LARGE COURSE	INCHES		94	96	98 1/4	102	102	89	89	73	92 13/16	100
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		22-0	21-6	21-0	21-0	22-0	19-1	19-1	15-1	19-0	19-0
13	COMBUSTION CHAMBER LENGTH	INCHES		60	80	72	64	113	36	36	—	33	90
14	GRATE AREA	SQ. FT.		97.7	90.4	97.8	108	132	69.3	70.0	55.5	66.3	100.2
15	TUBE & FLUE HEAT. SURF. (WATER SIDE)	SQ. FT.		4402	4502	4181	4851	5001	3729	3716	2471	3437	3971
16	FIREBOX HEATING SURFACE	SQ. FT.		379	385	568	552	660	304	299	157	363	499
17	TOTAL EVAP. HEAT. SURF.	SQ. FT.		4781	4887	4749	5403	5661	4033	4015	2628	3800	4470
18	SUPERHEATER SURFACE-STEAM SIDE	SQ. FT.		2265	2086	1497	2366	2085	908	908	—	1051	1900
19	FIREBOX VOLUME	CU. FT.			600	613	712	844	380	380	205	390	689
20	GAS AREA THRU BOILER	SQ. FT.		8.94	9.04	9.75	10.30	11.39	9.10	9.10	5.26	8.81	9.90
21	TYPE OF SUPERHEATER			E	E	A	E	A	A	A	—	A	E
22	STEAM AREA THRU SUPERHEATER	SQ. IN.		69.8	67.1	66.0	78.3	157.1	45.6	45.6	—	51.3	70.4
23	MAX EVAP. CALCULATED	LBS. PER HR.		63640	65650	74155	79675	85930	52150	51750	35570	52620	70380
24	MAX EVAP. ON TEST	LBS. PER HR.							65400	59085	30721	61680	
25	NUMBER & SIZE OF FLUES			195-3½	198-3½	58-5½	220-3½	69-5½	40-5½	40-5½	—	45-5½	184-3¾
26	NUMBER & SIZE OF TUBES			38-2¼	49-2¼	198-2¼	52-2¼	219-2¼	237-2¼	237-2¼	315-2	199-2¼	50-2¼
27	NUMBER & SIZE OF SUPERHEATER UNITS			99-1 3/16	101-1 3/16	58-1½	111-1 3/16	138-1½	40-1½	40-1½	—	45-1½	93-1¼
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ. FT.	20/9	.154	.144	.152	.156	.149	.204	.148	.220	.134	.155
31	GAS AREA PER SQ. FT. OF GRATE	SQ. FT.	20/14	.091	.100	.099	.095	.086	.132	.131	.095	.133	.099
32	GAS AREA PER SQ. FT. OF TUBE AND FLUE H.S.	SQ. FT.	20/15	.00203	.00201	.00233	.00212	.00228	.00242	.00246	.00213	.00256	.0025
33	GRATE PER 1000 LBS. TRACT. POWER	SQ. FT.	14/9	1.68	1.44	1.53	1.64	1.73	1.56	1.14	2.32	1.01	1.57
34	GRATE PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	14/17	.0205	.0185	.0206	.0199	.0233	.0172	.0175	.0211	.0175	.0224
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU. FT.	19/9	9.90	9.55	9.60	10.80	11.05	8.50	6.18	8.56	5.93	10.80
36	FIREBOX VOLUME PER SQ. FT. OF GAS AREA	CU. FT.	19/20		66.4	59.7	69.1	73.0	41.8	41.8	39.0	44.2	69.7
37	FIREBOX VOLUME PER SQ. FT. OF GRATE	CU. FT.	19/14		6.64	5.90	6.60	6.40	5.50	5.44	3.70	5.88	6.89
38	FIREBOX VOLUME PER SQ. FT. TOTAL EVAP. H.S.	CU. FT.	19/17		123.0	129.0	131.8	149.0	94.4	94.5	78.0	103.0	154.0
39	FIREBOX H.S. PER SQ. FT. OF GRATE	SQ. FT.	16/14		4.26	5.82	5.12	5.00	4.39	4.27	2.83	5.47	4.99
40	FIREBOX H.S. PER SQ. FT. TOTAL EVAP. H.S. ×100	SQ. FT.	16/17 ×100	7.93	7.87	12.00	10.20	11.67	7.53	7.45	5.97	9.55	11.19
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ. FT.	17/9	82.0	77.9	74.3	81.9	74.0	90.7	67.0	110.0	57.7	70.2
42	TOTAL EVAP. H.S. PER SQ. FT. OF GRATE	SQ. FT.	17/14	49.0	54.1	48.5	50.0	42.9	58.3	58.8	47.4	57.3	44.7
43	SUPERHEATER H.S. PER SQ. FT. TOTAL EVAP. H.S.	SQ. FT.	18/17	.473	.427	.316	.438	.369	.226	.227	—	.277	.425
44	PERCENT GAS AREA THRU FLUES			90.8	88.7	56.5	88.7	57.5	43.5	43.5	—	50.3	89.0
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	7/9	4240	4250	4120	4350	3680	4560	3830	4590	3720	4230
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	8/9	7210	7330	7200	7570	7960	6950	5125	7690	5170	7560

TABLE 6 GENERAL DIMENSIONS AND BOILER RATIOS OF ARTICULATED LOCOMOTIVES

	ITEM	UNIT MEASURE	RATIO	1	2	3	4	5	6	7	8	9	10
	TYPE			2-6-6-2	2-6-6-4	2-6-6-4	4-6-6-4	4-6-6-4	4-6-6-4	2-8-8-2	2-8-8-2	2-8-8-4	4-8-8-2
1	RAILROAD			C.&O.	S.A.L.	N.&W.	U.P.	NOR.PAC.	D.&R.G.W.	GT.NOR.	N.&W.	NOR.PAC.	SOU.PAC.
2	CLASS			H4A	R2	A	3900	Z6	L105	R2	Y6	Z5	AC7
3	YEAR BUILT			1927	1937	1937	1936	1936	1938	1929	1936	1930	1937
4	BOILER PRESSURE	LBS. PER SQ.IN.		210	230	275	255	250	255	240	300	250	250
5	CYLINDER DIA. & STROKE	INCHES		20X32	22X30	24X30	22X32	23X32	23X32	28X32	25X32 39X32	26X32	24X32
6	DIA. DRIVERS	INCHES		$56\frac{1}{4}$	69	70	69	69	70	63	57	63	$63\frac{1}{2}$
7	WEIGHT ON DRIVERS	POUNDS		379000	330000	430100	386000	435000	437939	544000	522850	558900	514800
8	TOTAL WEIGHT OF ENGINE	POUNDS		450500	480000	570000	566000	624500	641900	630750	582900	723400	639800
9	TRACTIVE POWER	POUNDS		81240	82300	104500	97400	104500	105000	146000	126838	140000	123400
10	BOILER I.D. FIRST COURSE	INCHES		87	$82\frac{5}{16}$	91	$96\frac{11}{16}$	$96\frac{3}{8}$	$92\frac{3}{4}$	98	$95\frac{9}{16}$	$103\frac{1}{4}$	$91\frac{7}{8}$
11	BOILER O.D. LARGE COURSE	INCHES		$93\frac{3}{4}$	96	$105\frac{1}{2}$	102	102	102	$111\frac{1}{8}$	$104\frac{1}{4}$	$110\frac{1}{4}$	$106\frac{1}{8}$
12	LENGTH OVER TUBE SHEETS	FEET & INCHES		24-0	24-0	24-1	22-0	23-0	22-0	24-0	$24\text{-}0\frac{3}{8}$	22-0	22-0
13	COMBUSTION CHAMBER LENGTH	INCHES		78	72	$115\frac{1}{2}$	86	89	$109\frac{1}{2}$	$84\frac{1}{16}$	$36\frac{5}{8}$	$72\frac{1}{2}$	68
14	GRATE AREA	SQ.FT.		72.2	96.3	122.0	108.2	152.3	136.5	126.0	106.2	182.0	139.0
15	TUBE & FLUE HEAT. SURF. (WATER SIDE)	SQ.FT.		4807	4914	6062	4756	4993	5563	7395	5207	6800	5990
16	FIREBOX HEATING SURFACE (WATER SIDE)	SQ.FT.		368	515	588	625	839	778	481	430	866	478
17	TOTAL EVAP. HEAT. SURF. (WATER SIDE)	SQ.FT.		5175	5429	6650	5381	5832	6341	7876	5637	7666	6468
18	SUPERHEATER SURFACE - STEAM SIDE	SQ.FT.		2246	2380	2703	1630	2090	2628	3515	1775	3224	2601
19	FIREBOX VOLUME	CU.FT.			532	843	862	1013	1060		630	1215	
20	GAS AREA THRU BOILER	SQ.FT.		9.36	9.13	10.95	10.46	11.05	11.20	12.50	10.72	13.61	11.75
21	TYPE OF SUPERHEATER			E	E	E	A	A	E	E	A	E	E
22	STEAM AREA THRU SUPERHEATER	SQ.IN.		63.5	68.4	80.3	68.3	83.2	79.7	99.4	68.3	98.7	79.7
23	MAX EVAP. CALCULATED	LBS. PER HR.		64450	74000	88610	81550	95000	98180	94060	71660	115070	84400
24	MAX EVAP. ON TEST	LBS. PER HR.											
25	NUMBER & SIZE OF FLUES			$180\text{-}3\frac{1}{2}$	$200\text{-}3\frac{1}{2}$	$239\text{-}3\frac{1}{2}$	$60\text{-}5\frac{1}{2}$	$73\text{-}5\frac{1}{2}$	$238\text{-}3\frac{1}{2}$	$281\text{-}3\frac{1}{2}$	$60\text{-}5\frac{1}{2}$	$280\text{-}3\frac{1}{2}$	$240\text{-}3\frac{1}{2}$
26	NUMBER & SIZE OF TUBES			15-3 1/2 38-2 1/4	$38\text{-}2\frac{1}{4}$	$57\text{-}2\frac{1}{4}$	$222\text{-}2\frac{1}{4}$	$192\text{-}2\frac{1}{4}$	$61\text{-}2\frac{1}{4}$	$88\text{-}2\frac{1}{4}$	$223\text{-}2\frac{1}{4}$	$92\text{-}2\frac{1}{4}$	$91\text{-}2\frac{1}{4}$
27	NUMBER & SIZE OF SUPERHEATER UNITS			$90\text{-}1\frac{3}{16}$	$103\text{-}1\frac{3}{16}$	$121\text{-}1\frac{3}{16}$	$60\text{-}1\frac{1}{2}$	$73\text{-}1\frac{1}{2}$	$120\text{-}1\frac{3}{16}$	$141\text{-}1\frac{3}{16}$	$60\text{-}1\frac{1}{2}$	$140\text{-}1\frac{3}{16}$	$120\text{-}1\frac{3}{16}$
28													
29													
30	GAS AREA PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{20}{9}$	.115	.111	.105	.107	.106	.107	.086	.085	.097	.095
31	GAS AREA PER SQ.FT. OF GRATE	SQ.FT.	$\frac{20}{14}$	.130	.095	.090	.097	.073	.082	.099	.101	.075	.084
32	GAS AREA PER SQ.FT. OF TUBE AND FLUE H.S.	SQ.FT.	$\frac{20}{15}$	.00195	.00186	.00181	.00220	.00222	.00201	.00169	.00206	.00200	.00196
33	GRATE PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{14}{9}$	.89	1.17	1.17	1.11	1.46	1.30	.86	.84	1.30	1.12
34	GRATE PER SQ.FT. TOTAL EVAP. H.S.	SQ.FT.	$\frac{14}{17}$	.0140	.0178	.0184	.0201	.0262	.0216	.0160	.0119	.0238	.0215
35	FIREBOX VOLUME PER 1000 LBS. TRACT. POWER	CU.FT.	$\frac{19}{9}$		6.46	8.06	8.86	9.69	10.10		4.97	8.68	
36	FIREBOX VOLUME PER SQ.FT. OF GAS AREA	CU.FT.	$\frac{19}{20}$		58.3	77.0	82.5	91.6	94.6		58.7	89.3	
37	FIREBOX VOLUME PER SQ.FT. OF GRATE	CU.FT.	$\frac{19}{14}$		5.53	6.90	7.96	6.64	7.76		5.92	6.67	
38	FIREBOX VOLUME PER SQ.FT. TOTAL EVAP. H.S.	CU.FT.	$\frac{19}{17}$		98.0	127.0	161.0	173.5	167.2		112.0	158.3	
39	FIREBOX H.S. PER SQ.FT. OF GRATE	SQ.FT.	$\frac{16}{14}$	5.10	5.35	4.82	5.77	5.51	5.70	3.82	4.05	4.76	3.44
40	FIREBOX H.S. PER SQ.FT. TOTAL EVAP. H.S. ×100	SQ.FT.	$\frac{16}{17}\times 100$	7.10	9.48	8.84	11.62	14.40	12.30	6.11	7.63	11.30	7.38
41	TOTAL EVAP. H.S. PER 1000 LBS. TRACT. POWER	SQ.FT.	$\frac{17}{9}$	63.6	66.0	63.6	55.4	55.8	60.3	54.0	44.4	54.8	52.5
42	TOTAL EVAP. H.S. PER SQ.FT. OF GRATE	SQ.FT.	$\frac{17}{14}$	71.6	56.4	54.5	49.7	38.2	46.5	62.5	52.9	42.2	46.6
43	SUPERHEATER H.S. PER SQ.FT. TOTAL EVAP. H.S.	SQ.FT.	$\frac{18}{17}$	.435	.438	.407	.303	.359	.415	446	.413	.420	402
44	PERCENT GAS AREA THRU FLUES			81.7	91.2	88.4	54.5	62.8	88.2	86.0	54.0	85.4	82.8
45	WEIGHT ON DRIVERS PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{7}{9}$	4660	4010	4125	3970	4160	4170	3720	4125	3985	4164
46	TOTAL WEIGHT OF ENGINE PER 1000 LBS. TRACT. POWER	POUNDS	$\frac{8}{9}$	5550	5835	5455	5820	5975	6100	4310	4600	5160	5180

Discussion

R. W. Anderson.[14] In his paper, the author clearly shows the importance of locomotive design ratios and successfully points out the limitations of formulas advocated by various locomotive-boiler authorities. Such limitations are primarily due to the lack of test data. For modern boilers, this in turn necessitates yet further approximations in the present inadequate formulas which must be used as a basis of design. Locomotive boilers are now being subjected to greater demands than ever before and the performance required will certainly not be lessened to ease the situation.

Weight and clearance dimensions immediately fix limitations which cannot be exceeded and which have the definite effect of hindrance in the design from its inception. However, in spite of these limitations and with the admittedly inadequate design formulas, filled as they are with estimated coefficients, boilers have been and are being built which generate more than the calculated quantity of steam, although the ratios between the grate areas, flue areas, firebox volumes, heating surfaces, etc., on successful boilers, vary within wide limits from the steam-generating viewpoint. However, when such boilers require more than the usual amount of attention considered necessary to keep them in proper condition, they can hardly be classified as being satisfactory for continuous operation.

All these conditions certainly point to the necessity of a central testing plant for such jobs. The experiments on the design formulas and ratios would no doubt take several years. Independent investigations and tests are often unsatisfactory due to lack of adherence to standard test codes and methods of collecting information.

The high draft-tube readings which are obtained in front-end experiments on some railroads lead others to try similar arrangements only to find that the new application could not survive transplanting. It would have been much cheaper to borrow the draft tubes, providing a standard measuring device for testing the existing arrangement.

If a series of boilers of various proportions were developed in such fashion, the major problem, namely, availability, is then only started. Modern locomotives have an exceptionally high first cost. Under present demands, boilers require considerably more than the usual amount of attention in roundhouses and shops. The metallurgist is being drawn further into this problem and it is his responsibility to determine the physical and chemical properties of the boiler steels which will be capable of withstanding present-day loads. He may be the one who will be required to tell the manufacturers to set the side sheets at a certain slope in order to prevent warped plates and leaky stay bolts. Continual firebox-repair work should not be a necessary evil attached to 300-lb-pressure boilers; to eliminate this condition a thorough study should be made of various furnace designs and the proper materials to use in their construction.

The engine portion of the locomotive has always received the greatest amount of attention from the designer's viewpoint. Consequently, the boiler is often changed to accommodate certain other parts which no doubt could have been altered to avoid affecting the boiler design. For instance, the trailer-truck design limits the grate and ashpan arrangement but, if the furnace design were given the attention it should have on modern locomotives and every effort made to have the trailer come within the limits specified, there would undoubtedly be a considerable saving realized in firebox, grate, and ashpan maintenance charges.

The performance of high-speed engines on long runs has created many problems in regard to lubrication and materials which must be solved. It will take a lot of coordinating to accomplish this.

E. G. Bailey.[15] The writer will confine his discussion to one phase of this paper, applying the data therein presented to the method of comparison of steam-boiler performance, covered by his paper[16] on that subject.

Throughout his paper the author has very properly mentioned the question of absorption efficiency and combustion efficiency, or total boiler and superheater over-all efficiency, which is in reality the product of the other two. The data given in his Figs. 13 and 14 are sufficiently complete to show wherein the difference lies between combustion efficiency and absorption efficiency, both relating to coal-fired boilers. However, in Fig. 10 he gives data of an oil-fired marine boiler which happens to be the same as boiler F, Table 3, in the writer's paper,[16] and also two oil-fired locomotive boilers, Southern Pacific AC4 and Santa Fe 3766. The difference between the efficiencies of these and

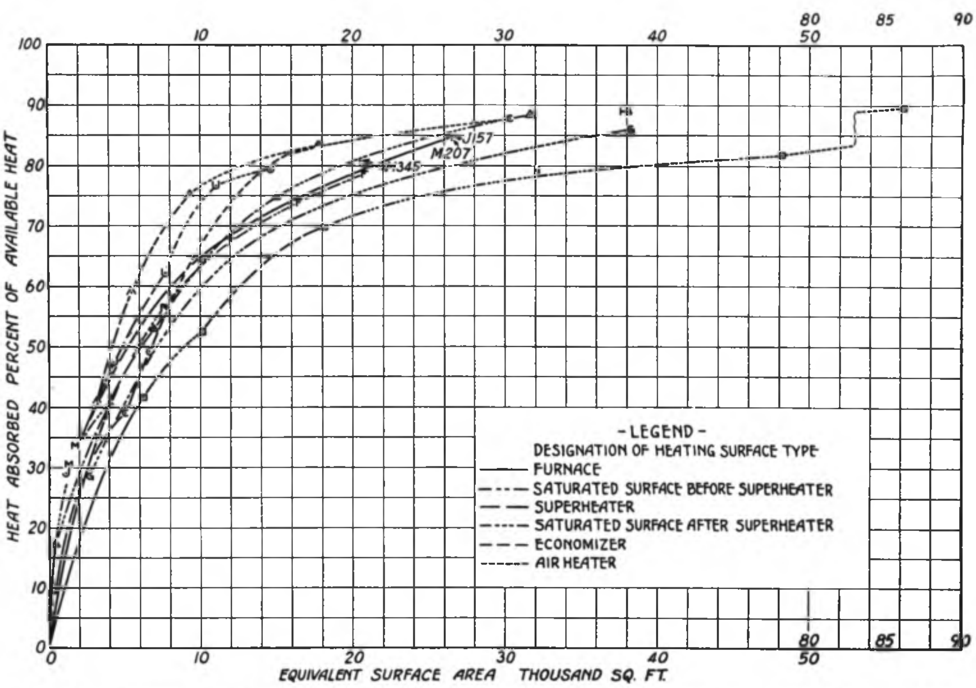

Fig. 16 Boiler Tests Showing Heat Absorption Plotted Against Equivalent-Surface Area

[14] Superintendent of Motive Power, Chicago, Milwaukee, St. Paul & Pacific Railroad Company, Milwaukee, Wis.

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

[16] "Steam-Boiler Performance and a Method of Comparison," by E. G. Bailey, Trans. A.S.M.E., vol. 62, July, 1940, pp. 367–378.

TABLE 7

	SYMBOL	⊡	△					J		M-207		M-345		H	HB
1	TEST	B-1			F-1			N.Y.C. J-1-B		P.R.R. M-1-A		P.R.R. M-1-A		5-9-02	5-20-02
2	BOILER	STATIONARY BOILER FIG. 3			MARINE BOILER FIG. 9			LOCOMOTIVE		LOCOMOTIVE		LOCOMOTIVE		HOCKING VALLEY R.R.	
3	RATED STEAM OUTPUT - LB/HR.				67000			57700		67850		67850			
4	STEAM OUTPUT ON TEST - LB/HR.				66070			84800		80000		99095		17830	17200
5	STEAM PRESS. AT SUPERHEATER OUTLET - LB/IN² GA.				309			225		250		250		166	170
6	SATURATION TEMPERATURE - °F.				428			397		406		406		373	375
7	STEAM TEMPERATURE AT SUPERHEATER OUTLET - °F.				695									SATURATION	
8	FEEDWATER TEMPERATURE - °F.				211									63.5	72.4
9	AIR TEMPERATURE LEAVING AIR HEATER - °F.				121										
10	METHOD OF FIRING	PULV. COAL	MECHANICAL ATOMIZING OIL BURNER					STOKER		STOKER		STOKER		HAND FIRED	
11	KIND OF FUEL				OIL			BITUMINOUS COAL		BITUMINOUS COAL		BITUMINOUS COAL		BITUMINOUS COAL	
12	GRATE AREA OR FURNACE CROSS SECTION - SQ. FT.	.			71.7			81.5		70		70		31.08	30.10
13	LBS. FUEL PER HR. DIVIDED BY ITEM 12				70.3			157		207		345		94.5	84.7
14	HEATING VALUE OF FUEL "AS FIRED" - B.T.U./LB.				18625			13900 (DRY)		13900 (DRY)		13900 (DRY)		11898	12265
15	SECTION OF UNIT		FURNACE	1ST BOILER BANK	SUPERHEATER	INTERMEDIATE BOILER BANK	2ND BOILER BANK	FIREBOX	BOILER & SUPERHEATER	FIREBOX	BOILER & SUPERHEATER	FIREBOX	BOILER & SUPERHEATER	FIREBOX & BOILER	FIREBOX & BOILER
16	CO₂ % BY VOL.		12.5				12.5							12.2	9.4
17	FLUE GAS ANALYSIS LEAVING SECTION O₂ " " "	DATA GIVEN IN BAILEY PAPER TEST B-1 TABLE-2 FIG. 3	4.1				4.1							6.1	9.7
18	CO " " "		0.0				0.0							0.7	0.01
19	TOTAL COMBUSTION AIR LEAVING SECTION - %		123				123							140	183
20	TOTAL WEIGHT WET GAS LEAVING SECTION - LB/HR.		92200				92200							36800	38400
21	HEAT AVAILABLE ABOVE 80°F. - ENTERING SECTION - BTU/LB. GAS		960												
22	MOISTURE BY WEIGHT IN FLUE GAS - %		6.3				6.3							4.9	4.4
23	ADIABATIC TEMPERATURE - °F.		3315												
24	TOTAL HEAT INPUT FROM FUEL - MILLION B.T.U./HR.		94.1					180.8		208.5		333.6		34.9	31.2
25	TOTAL PREHEAT IN AIR ABOVE 80°F. - MILLION B.T.U./HR.		0.9												
26	TOTAL LOSSES - LATENT HEAT, CARBON LOSS - MILLION B.T.U./HR.		6.6					61.3		91.4		182.7		8.6	8.8
27	TOTAL HEAT AVAILABLE ABOVE 80°F. - MILLION B.T.U./HR.		88.4					119.5		117.1		150.9		26.3	22.4
28	LIBERATION - ITEM 27 DIVIDED BY ITEM 39 - BTU/CU. FT. HR.		208000					340000		246000		318000		179000	155000
29	ITEM 27 DIVIDED BY ITEM 41 - B.T.U./SQ. FT. HR.		1163000	123100	71600	33200	15780	425000	18800	290500	18950	374000	24400	15500	13200
30	GAS TEMPERATURE LEAVING SECTION - °F.		2820	2120	1732	1000	510							745	730
31	HEAT AVAILABLE IN GAS ENTERING - MILLION B.T.U./HR.		88.4	74.75	52.55	41.85	22.50	119.5	84.9	117.1	78.1	150.9	104.9	26.3	22.4
32	HEAT ABSORBED: INDIVIDUAL SECTION - MILLION B.T.U./HR.		15.12	20.7	10.7	19.35	12.4	34.6		39.0		46.0			
33	HEAT ABSORBED: CUMULATIVE - MILLION B.T.U./HR.		15.12	35.82	46.55	63.9	78.3	34.6	101.1	39.0	98.7	46.0	122	20.8	19.9
34	HEAT ABSORBED: INDIVIDUAL SECTION DIVIDED BY ITEM 27 - %		17.1	23.4	12.1	22.0	14.0	29.0		33.5		30.5			
35	HEAT ABSORBED: CUMULATIVE DIVIDED BY ITEM 27 - %		17.1	40.5	52.6	74.6	88.6	29.0	84.7	33.5	84.3	30.5	80.9	79.1	88.9
36	HEAT ABSORPTION: INDIVIDUAL SECTION - BTU/SQ. FT. HR.		199000	32300	20700	13550	7240	123000		97000		114000			
37	HEAT ABSORPTION: CUMULATIVE - B.T.U./SQ. FT. HR.		199000	50200	37700	24800	14000	123000	15900	97000	15900	114000	19700	12300	11700
38	EQUIVALENT HEAT RECEIVING SURFACE TEMPERATURE - °F.														
39	FURNACE VOLUME - CU. FT.		424					353		475		475		147	144
40	SURFACES: INDIVIDUAL SECTION - SQ. FT.		76.0	640	516	1428	2939	281		403		403		[illegible]	[illegible]
41	SURFACES: CUMULATIVE - SQ. FT.		76.0	716	1232	2660	5599	281	6356	403	6188	403	6188	1690	1697
42	EQUIVALENT SURFACE AREA (500×10⁶ ÷ ITEM 27) × ITEM 41 - SQ. FT.		430	4060	6980	15000	31700	1176	26500	1718	26500	1340	20500	32100	37800
43	VOLUME OF ENTIRE UNIT - CU. FT.						2070							813	828
44	ITEM 27 DIVIDED BY ITEM 43 - B.T.U./CU. FT. HR.						42700							32400	27000
45	DRAFT LOSS - INCHES OF WATER						2.53							2.03	1.96
46	TOTAL DRAFT PLUS AIR PRESSURE - INCHES OF WATER						6.52							3.78	4.12
47	OVERALL EFFICIENCY - %		16.1	38.1	49.5	70.0	83.2	19.1	56.0	18.7	47.3	13.8	36.6	59.6	63.9

the oil-fired marine boiler is outstanding, and further data as to the breakdown of this efficiency difference would be interesting.

Fig. 16 of this discussion includes a typical pulverized-coal-fired stationary boiler, as represented by test B-1, Table 2 of the paper,[16] and test F-1 for an oil-fired marine boiler from Table 3 of the same source. The latter has been calculated to show the percentage of heat absorbed in the different sections of the unit, although the official efficiency tests cover only the over-all efficiency, as is plotted in Fig. 10.[16] In addition to the data which were plotted from Pennsylvania Railroad class M1A, the author's Fig. 13, and the New York Central class J1B, Fig. 14, the writer has also gone back to the data of Goss's tests at Coatesville in 1912 and selected the one with the most representative maximum rating giving complete results on a coal-fired radial-

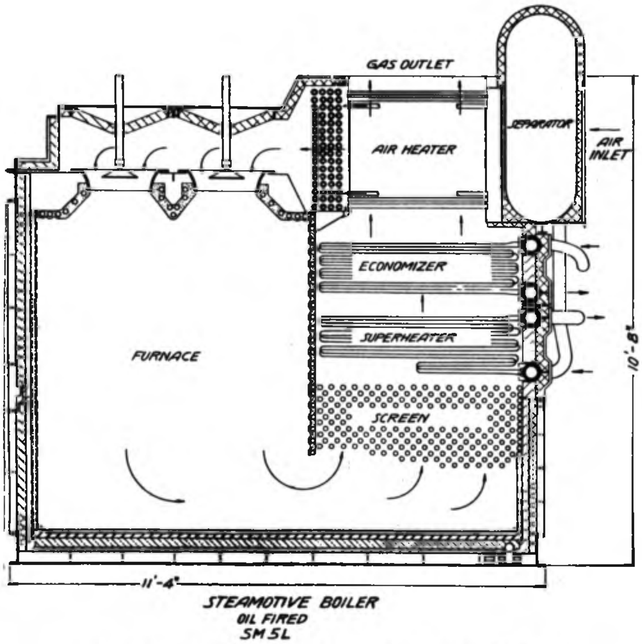

FIG. 17 OIL-FIRED STEAMOTIVE BOILER, SM5L

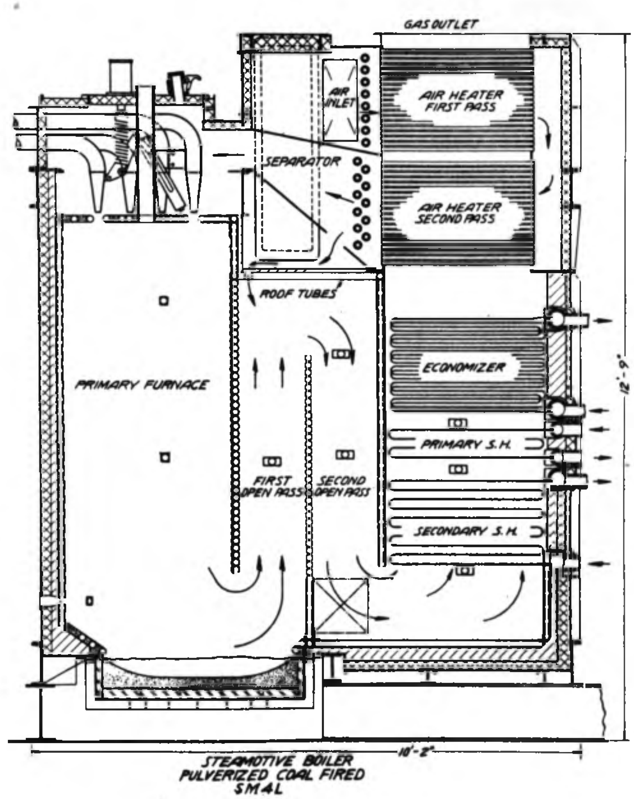

FIG. 18 PULVERIZED-COAL-FIRED STEAMOTIVE BOILER, SM4L

TABLE 7 (*Continued*)

F	G		Λ				U				• (1ST HOUR OF RUN)					C (16TH HOUR OF RUN)					
A-2	COATESVILLE A-112R			6-10-35				3-31-38					11-24-39					11-24-39			1
NORTHERN RAILWAY OF FRANCE FIG.1	RADIAL STAY LOCO.		STEAMOTIVE AG 1L				STEAMOTIVE SM 5L FIG 17				STEAMOTIVE SM 4L FIG 18					STEAMOTIVE SM 4L FIG 18					2
				21000				40000					20000					20000			3
	28547			22050				44350					14100					13700			4
	214			1339				1476					1190					1150			5
	393			583				595					568					563			6
	SATURATION			914				903					869					901			7
	55			346				365					213					211			8
				493				362					533					565			9
HAND FIRED	HAND FIRED		STEAM MECHANICAL ATOMIZING OIL BURNER				STEAM MECHANICAL ATOMIZING OIL BURNER				PULVERIZED COAL					PULVERIZED COAL					10
BRIQUETTES	BITUMINOUS COAL		OIL				OIL				BITUMINOUS COAL					BITUMINOUS COAL					11
	58.07			12.25				23.0					6.5					6.5			12
	73.4			149				156					262					262			13
	13065			18470				18570					12965					12965			14
	FIREBOX	BOILER	FURN & BOILER	SUPERHEATER	ECONOMIZER	AIR HEATER	FURN & SCREEN	SUPERHEATER	ECONOMIZER	AIR HEATER	FURNACE TO S.H. INLET	SECONDARY SUPERHEATER	PRIMARY SUPERHEATER	ECONOMIZER	AIR HEATER	FURNACE TO S.H. INLET	SECONDARY SUPERHEATER	PRIMARY SUPERHEATER	ECONOMIZER	AIR HEATER	15
DATA GIVEN IN BAILEY PAPER TEST A-2 TABLE-1 FIG.1		9.7				13.8				13.2					15.4					15.5	16
		8.8				3.1				3.6					2.8					2.6	17
		0.3				0.0				0.0					0.0					0.0	18
		16.8	115.7			115.7	120			120	114.5				114.5	113				113	19
	56400	56400	30900				62600	62600	62600	62600	20200				20200	20100				20100	20
			1091				1037	560			1069					1084					21
	3.5		5.5				7.18				5.3				5.3	5.3				5.3	22
			3720				3510				3665					3705					23
	55.8		33.8				66.75				21.9					22.1					24
			2.93				4.03				1.42					1.50					25
	16.8		2.98				5.88				1.72					1.80					26
	39.0		33.75				64.9				21.6					21.8					27
	107700		374000				358000				385000					388000					28
	161600	13100	152600	90750	53400	27900	117200	65000	45300	34300	97700	74700	62700	30900	16320	98600	75400	63200	31200	16470	29
		615	2370	1730	1110	784	2025	1370	1000	891	2230	1900	1650	750	540	2440	2075	1810	815	584	30
	39.0		33.75	19.82			64.9	34.19			21.6	12.10	10.07	8.595	3.555	21.8	13.35	11.12	9.52	3.89	31
	11.1	22.44	13.93	5.94	.546	2.75	30.71	12.28	6.58	1.89		2.03	1.475	5.04	1.13		2.23	1.6	5.63	1.22	32
	11.1	33.54	13.93	19.87	25.33	28.08	30.71	42.99	49.57	51.46	9.50	11.53	13.005	18.045	19.175	8.45	10.68	12.28	17.91	19.13	33
	28.5	57.5	41.2	17.6	16.2	8.2	47.3	18.9	10.2	2.9		9.4	6.8	23.4	5.2		10.2	7.3	25.9	5.6	34
	28.5	86.0	41.2	58.8	75.0	83.2	47.3	66.2	76.4	79.3	44.0	53.4	60.2	83.6	88.8	38.8	49.0	56.3	82.1	87.7	35
	51700	8120	63000	39400	21000	4760	55500	27600	15250	4090		29700	26650	14280	1806		32600	28900	16000	1950	36
	51700	11200	63000	53400	40000	23200	55500	43100	34600	27100	43000	39800	37800	25900	14480	38300	36900	35700	25700	14470	37
																					38
	362		90.4				181				56.2	(121.2 CU. FT. TO ENTRANCE OF S.H.)				56.2					39
	214.7	2769	221.2	150.7	260.0	578.0	553.2	445.4	432.4	462.0		68.4	55.3	353.0	626.0		68.4	55.3	353.0	626.0	40
	214.7	2984	221.2	371.9	631.9	1209.9	553.2	998.6	1431.0	1893.0	221.0	289.4	344.7	697.7	1323.7	221.0	289.4	344.7	697.7	1323.7	41
	2755	38300	3250	5470	9300	17800	4270	7700	11030	14600	5120	6700	7980	16150	30620	5070	6640	7900	16000	30380	42
		1340				425				677					5124					5124	43
		29100				79500				95800					42100					42500	44
						24.8				11.3					6.1					8.9	45
		6.4				44.9				29.7					14.10					16.6	46
	19.9	60.1	37.1	52.9	67.5	74.9	44.3	62.0	71.6	74.3	40.8	49.5	55.9	77.5	82.4	35.8	45.1	52.0	75.9	81.0	47

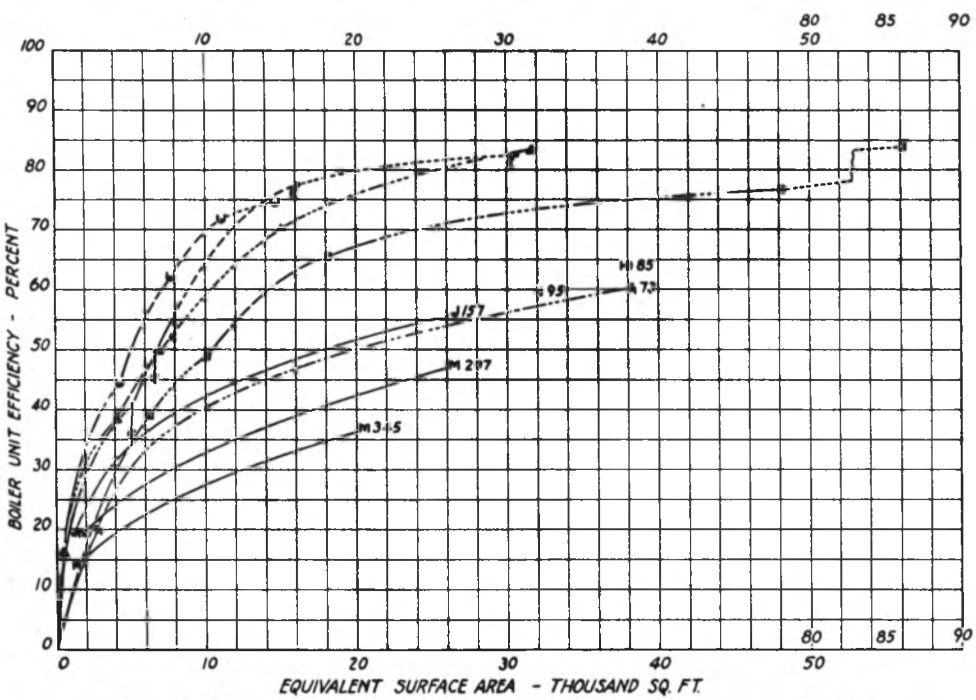

FIG. 19 CURVES SHOWING OVER-ALL EFFICIENCY OF UNITS LISTED IN TABLE 7

stay-bolt firebox, in which Dr. Goss determined the heat absorption in the firebox separately. From Hitchcock,[17] the writer has also selected two road tests on locomotives conducted in 1902, from which sufficient reliable data were obtained to plot the absorption efficiency and the over-all efficiency, but without measured distribution between the firebox and the flues. Both the Goss and Hitchcock tests were without superheaters. Also from the paper,[16] test A-2 of the French locomotive, with one half the tubes plugged, falls directly in line as to rate of output with the Pennsylvania class M1A at 345 lb of coal fired per sq ft of grate per hr. The other point of Pennsylvania class M1A plotted is taken to duplicate exactly the equivalent-surface area of the New

[17] "Road Tests of Consolidation Freight Locomotives," by E. A. Hitchcock, Trans. A.S.M.E., vol. 25, 1904, pp. 550–588.

York Central class J1B at its maximum rate. It is to be noted that they come closely together and are quite consistent, while there is an indication that the tests on the French locomotive with its small firebox showed somewhat less efficiencies.

Superposed on these curves are also data from the oil-fired Steamotive boiler as tested at Schenectady and reported in a paper[18] by E. G. Bailey, A. R. Smith, and P. S. Dickey; from the oil-fired Steamotive boiler on the Union Pacific steam turboelectric locomotive which has been described[19] recently; as well as from tests on pulverized-coal-fired Steamotive boiler, conducted at Erie, Pa., during 1939.

The Steamotive generators are high-pressure high-temperature units, as indicated in Table 7 of this discussion, and the slopes of the curves after the gases enter the superheater and economizer show very high rates of heat absorption due largely to the high mass flow and the suitable arrangement of the tubes with respect to gas velocity and gas temperature. A cross section of the Union Pacific oil-fired Steamotive boiler is shown in Fig. 17 and the pulverized-coal-fired unit in Fig. 18.

Fig. 19 gives data from most of the units listed in Table 7 on the over-all efficiency basis. This includes boiler, superheater, economizer, and air heater.

The pulverized-coal-fired stationary and Steamotive units, as well as the oil-fired marine and Steamotive units show only the normal difference between the absorption efficiency and the total efficiency, but the coal-burning locomotives show a marked diminution in efficiency due largely to cinder loss, which increases in general with the increased rate of firing. However, it is noted that the New York Central at 157 lb of coal fired per sq ft of grate per hr comes within the same range of over-all efficiencies as were obtained from Goss's tests at 73 lb of coal per sq ft, and the Hocking Valley tests at 85 and 95 lb, respectively.

In the author's Figs. 13 and 14, by means of dotted lines, he shows the heat absorbed in the firebox. His treatment of that subject in the text indicates definitely that he has the correct conception of the variation of this figure with rates of heat available, rather than the use of a constant figure of 55 lb of steam per sq ft of firebox per hr, which was evidently based on Goss's tests, as reproduced herewith. Fig. 20 expresses per cent of heat absorbed in the firebox as a function of the Btu available per square foot of actual heating surface. This gives data from Goss and the French locomotive tests as plotted points. It also gives a curve, calculated from the Hudson-Orrok formula[20] for which a basis of 13,900 Btu of coal with 11.8 lb of air per lb of coal, has been selected, which corresponds closely to 19 per cent excess air and 3600 F adiabatic temperature. Broido's average curve also is shown within the range in which it is given in his paper[21] before the Society. It would seem from this that the author's dotted lines of Figs. 13 and 14 may be somewhat high, and the curves of Figs. 16 and 19 of this discussion are plotted on the basis of being taken from the Orrok curve. One could readily plot in Fig. 16 the curves from Brandt's data as shown in Fig. 20. This would indicate a relatively higher rate of heat absorption in the firebox than seems to exist on other types of furnaces and, even at the same rates of Btu available, they are markedly higher than is indicated by Goss and the French locomotive, both of which were based upon actual weights of water evaporated by the firebox surface.

It is doubtful if this question of distribution of heat absorption between the firebox and the flues of a locomotive boiler can be accurately determined by gas-temperature measurement. The difference in the size of the flues, the resistance, gas velocity, and stratification of both quality of gas and temperature make it difficult, even though a device could be used which would get the true gas temperature. The cold surroundings of the tube sheet and flues will influence any thermocouple readings, even with the high-velocity method, to the extent that the data would be questioned. Therefore, it seems that the method used by

[18] "Steamotive," by E. G. Bailey, A. R. Smith, and P. S. Dickey, *Mechanical Engineering*, vol. 58, December, 1936, pp. 771–780.

[19] "Progress in Railway Mechanical Engineering, 1938-1939," Report of Committee RR6, Railroad Division, A.S.M.E., *Mechanical Engineering*, vol. 62, December, 1939, pp. 863–875.

[20] "Estimation of Radiant-Heat Exchange in Boiler Furnaces," by G. A. Orrok and N. C. Artsay, *Combustion*, vol. 9, no. 10, April, 1938, pp. 37–42.

[21] "Radiation in Boiler Furnaces," by B. N. Broido, Transactions A.S.M.E., vol. 47, 1925, pp. 1123–1147.

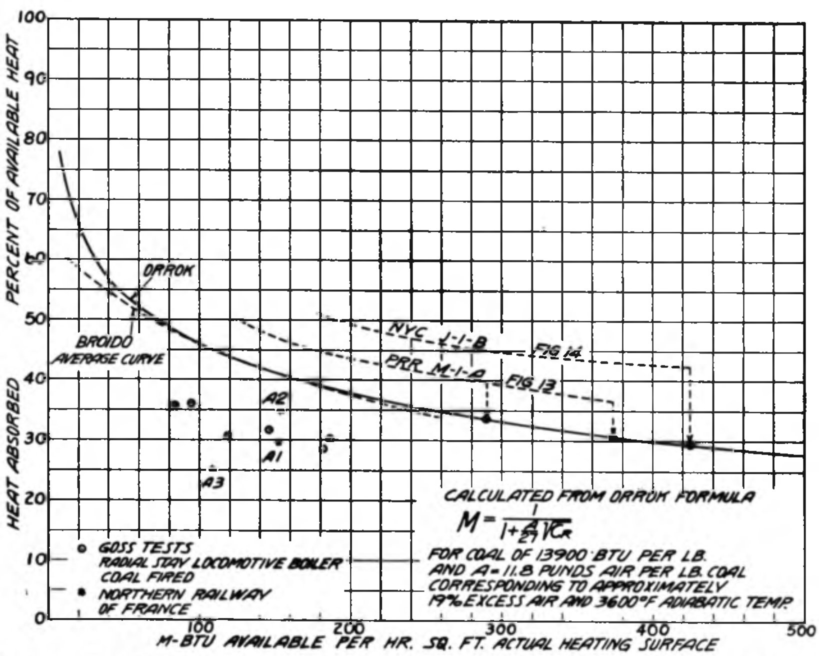

Fig. 20 Curve Showing Per Cent of Heat Absorbed in Firebox as Function of Btu Available per Square Foot of Actual Heating Surface

Goss and the French locomotive may be the only one in which such distribution can definitely be determined. Knowledge of this, however, only affects the detailed slope of the curve at the lower portion, when plotted on the equivalent-surface basis. The upper points on both the absorption-efficiency basis and over-all efficiency basis are equally reliable from well-conducted tests and show the important part of the over-all efficiency and the rate of heat input.

The author brings out clearly the limited area for gas flow entering the tubes and flues of the locomotive boiler. On a water-tube arrangement of heating surface, such as used in the Steamotive, this area can be made anything desired within reasonable limits. The designs now operating have more than twice the area on a given capacity basis than is possible on the locomotive units as shown by the author.

A further study of draft loss through the respective units is also of interest. The different tests in the writer's paper[16] give the over-all draft loss including the air heater, if any, burners, or fuel bed, draft loss through boiler settings, superheater, economizer, and air heaters, if any, as the total pressure drop. It would be helpful if in his closure the author could supply the data in a form which might be compared more readily than that as given in Fig. 5, showing back pressure.

J. M. Duncan.[22] The author reminds us that, in current locomotive practice, combustion rates and heat releases are far higher than any realized commercially in stationary plants. If industrial and central-station heat-power engineers could approach these rates, they would advance their art by materially reducing capital costs.

Pennsylvania Railroad locomotive class M1A appears to be designed for a heat liberation of 250,000 Btu per cu ft of furnace volume per hr, burning 121 lb of coal per sq ft of grate area per hr. Figs. 8 and 12 of the paper indicate that under these conditions combustion is substantially complete, the dry flue-gas loss being moderate, but the unburned-coal loss is 30 per cent. The furnace heat absorption is 41 per cent of the heat liberated, and the average heat absorption in the furnace is about 87,000 Btu per sq ft of furnace walls per hr. Data on furnace temperatures and excess air are not given. The writer calculates the exit furnace temperature at 1800 F and the excess air at 70 per cent. These figures may be considerably in error as essential data are lacking. Unburned-coal losses for other locomotives are above and below this 30 per cent figure at the same heat-release rate. At this rate, New York Central locomotive J1B has a loss of 12 per cent.

The author's objective is to develop means for increasing locomotive capacity and efficiency. Any increase in efficiency accomplished within existing weight and space limitations will, if present fuel-burning ability is maintained, automatically increase locomotive capacity. Two opportunities for increased efficiency lie in reduction of dry-flue-gas loss and reduction of unburned-coal loss. As pointed out by the author, convection-heat transfer is efficient. Improvements must be sought in furnace conditions.

If the underfeed or traveling-grate stoker can be adapted to locomotive service, will not the unburned-coal loss be automatically almost eliminated? The very fact that such large losses are tolerated indicates that the sprinkler type is accepted as indispensable for the large modern locomotive. But, if these other types can be adapted to this service, important advantages will be realized in addition to reduction of unburned-coal loss. This is principally because with these other types furnace absorption efficiency can be greatly increased.

An increase in furnace-absorption efficiency would reduce dry-gas loss almost proportionally if excess air is unchanged. If the increase is accomplished while reducing excess air, a further gain will be made. The following relations are calculated for a New River coal, assuming complete combustion with no air preheat:

Furnace exit, F	—Furnace absorption efficiency, per cent— 20 per cent excess air	80 per cent excess air
1500	63	46
2000	49	26
2500	34	4

These figures indicate that excess air cannot be materially reduced without incurring slag accumulations in flues with many coals which must be burned on railroads, unless furnace absorption efficiency can be greatly improved. The writer believes that the improvement must be brought about by a radical change in stoker design. This appears to be the author's conclusion when he says:

"From the studies made of this problem it is doubtful if the percentage of heat absorption can be increased to much more than 50 to 55 per cent of the heat liberated, by increasing the amount of firebox heating surface, because the amount of radiant energy absorbed is limited by the effective surface of the fire bed and the luminosity of the flames."

In regard to the effect of smoky flames the author says:

"It has been demonstrated that the radiation from the flames is largely due to incandescent particles of fuel and ash in the flame. The quantity and distribution of this radiating material in the flames, together with the amount of firebox heating surface, determine the coefficient of emission or absorption. This explains why boilers fired with oil give higher superheat; the less luminous flames of the oil fire will give up a smaller percentage of heat to the firebox by radiation, and the gases entering the flues are therefore hotter."

These remarks doubtless apply to a comparison of pulverized-coal and oil flames. But the facts cited tend to divert our attention from the controlling influence of luminous flames in reducing over-all radiant-heat absorption in stoker-fired furnaces wherever the grate is not obscured by an arch. The luminous flame will, it is true, lose more heat than the nonluminous flame (if high turbulence is maintained) but it will at the same time hide the incandescent and much hotter stoker fire from the heat-absorbing walls to a great extent. The author's remarks do apply to the stoker-fired-locomotive furnace shown in Fig. 3, where the luminous flames created by the sprinkler-type stoker are of advantage because that stoker requires an arch to reduce the unburned-coal loss. In such a furnace clear flames would be a disadvantage. Data of Fig. 3 do in fact indicate the presence of highly luminous flames. The velocity at section *C* is materially lower than at section *B* although the gas areas are practically equal. If flames were completely transparent, temperatures would be nearly equal throughout this section of the furnace.

The importance of clear furnace atmospheres in underfeed-stoker-fired furnaces was forcefully impressed upon the writer when observing some tests using a high-volatile coal. Operators had been unable to obtain smokeless combustion at moderately high rates. Boiler exit temperatures were higher than they should have been although excess air was less than guaranteed. A test operator then took charge for higher rate tests. While maintaining the same percentage excess air as before, he produced a transparent furnace atmosphere at considerably higher combustion rates and the excess boiler-exit temperatures were reduced. It was obvious that a large increase in the percentage of furnace heat absorption had been obtained. This must have been due to the removal of smoke and luminosity from the flames, thus allowing the fuel bed to "see" the furnace waterwalls and roof.

[22] Petersburg, Va. Mem. A.S.M.E.

During those tests it occurred to the writer that furnace heat absorption and over-all boiler performance could have been increased in spite of smoky furnace atmosphere if the "hydraulic mean radius" of the furnace were reduced by installing tubes spaced, say, on three- or four-foot centers throughout the upper portion of the furnace. (The vertical screen used to divide many modern powdered-coal furnaces, e.g., Fisk Street, appears to be a development of this sort, perhaps made more necessary by the essentially opaque nature of the flames in such furnaces.) The degree to which this could be carried out without prejudice to the furnace function depends upon the effect upon combustion. Has the author data upon the relation between furnace temperature and completeness of combustion—of carbon and hydrocarbons as well as of CO?

The sprinkler-type stoker seems inherently incapable of realizing high radiant-heat absorption. Its fuel bed is probably lower in surface temperature than that of the underfeed or traveling-grate type. Heating of green coal should not be spread over the fuel bed if a hot surface and a smokeless furnace atmosphere are desired. The other types of stokers confine the distillation of green coal to a small section of the fuel bed and so leave a large area of highly incandescent coke to radiate to the furnace walls; they also produce less smoke and consume it more rapidly. Such stokers do not require the arch shown in Fig. 3, which completely obscures the fuel bed from most of the furnace walls even if smoke is not produced.

Furnace-gas turbulence has been recognized as desirable to assist in attaining complete combustion. The writer believes it should be sought also in order to insure maximum radiant heat transfer at all times that an absolutely clear furnace atmosphere is not maintained. Evidently the gas velocities shown in Fig. 3 bring a high degree of turbulence. The temperature drop indicated by velocity drop is several times what could be caused by convection heat transfer. Smoky flames require high turbulence successively to turn new envelopes toward cold furnace walls if high radiant heat transfer is to be realized. The importance of turbulence therefore is great even if combustion (of CO) is complete without it, and especially where small excess-air ratios tend to cause hazy or smoky conditions in the furnace. But the sprinkler type of stoker has some critical rate of furnace-gas turbulence above which unburned-coal losses become prohibitive. Turbulence in furnace gases would not adversely affect the underfeed or traveling-grate stoker in this respect.

Thus the underfeed and traveling-grate stoker appear to be capable of increasing locomotive efficiency, and therefore capacity, by increasing furnace-absorption efficiency and reducing unburned-coal loss. Stationary-plant experience does not yet extend economical coal-burning capacities to the rates evident from an inspection of the author's Tables 1 to 6. However, as the author has noted, furnace-wall area is not the controlling factor in radiant heat transfer. The controlling factor is the area of the hot body, and even more, its temperature. Thus a larger grate area could with advantage be adopted, sacrificing furnace-wall area as necessary to obtain it, provided the furnace wall could see the fuel bed so installed. This larger grate area would also be helpful in reducing such slight tendency as there would be (slight, that is, compared to unburned losses now accepted in locomotive operation) for fuel to blow off the grate. In this connection, it would be interesting to know whether the higher combustion efficiency shown in Fig. 8 for locomotive NYC-J1B is due to the lower rate of coal combustion per square foot of grate area, which is inferred from the data given in Table 1.

It is suggested that a design for high efficiency and capacity should aim at:

1 Feeding coal in such manner as to throw no fines up into the furnace atmosphere

2 Confining distillation of volatile constituents to a small area of the fuel bed.

3 Maintaining a large incandescent coke area within full view of the furnace walls.

4 Operating at high rates with about 20 per cent excess air, with little or no smoke and a nearly nonluminous flame.

5 High turbulence in the furnace, with furnace designed for smallest practicable volume-area ratio to increase furnace heat absorption if smoky flames cannot be avoided.

Such a design should operate with little or no unburned-coal loss and with very low dry-flue-gas loss. If applied to locomotive PRR-M1A a reduction in fuel consumption of over 30 per cent should be realized. Designed capacity would in that event be realized with a fuel-burning rate of about 80 lb per sq ft of grate area per hr. Reduction to this moderate rate would, as shown by the author's data, considerably reduce engine back pressure, still further increasing efficiency and capacity. These possibilities are undoubtedly known to locomotive designers. The writer would appreciate knowing the principal difficulties which have stood in the way of their realization so far.

The author's paper suggests that there is a great opportunity for the application of rational design methods to locomotive-boiler practice. It is evident that a large amount of data awaits systematic analysis, following which the several combinations of furnace, stoker, and boiler indicated by that analysis as worthy, could be tested at moderate cost to achieve a fine commercial design.

R. Eksergian.[23] Not only has the author given a comprehensive contribution on locomotive-boiler characteristics and limitations, but he has pointed out some pertinent factors which have not been clearly recognized before. One outstanding factor is the tremendous influence of increased gas area through the boiler on evaporation capacity and its tendency to sustain fair efficiencies through a wide range of firing rates. The slope of the boiler efficiency against firing rate is particularly significant as to the maximum evaporation capacity. On the basis of a linear variation of boiler efficiency against firing rate, we must necessarily have a parabolic evaporation curve against firing rate. The maximum evaporation occurs, then, at a firing rate at which the boiler efficiency is just one half the efficiency at zero firing rate. This means that, to obtain high firing rates at maximum evaporation, the slope of the efficiency curve against firing rate must be flat. Pennsylvania Railroad locomotive M1A shows this to a predominant degree and, with this characteristic, we note the corresponding large gas area through the boiler as compared with some other types.

One major feature of the advantage of the steam locomotive over all other types of motive-power units is its considerable overload capacity. Locomotives under normal operation do not perform at firing rates corresponding to their maximum evaporative capacity. The normal rating may either be limited to a definite over-all efficiency or to a firing rate corresponding to some percentage of the firing rate at maximum evaporation. In a well-designed boiler, both aspects are compatible. Mr. Fry[7] assumes the boiler efficiency to vary as a straight line against firing rate and arrives at a parabolic curve for the evaporative capacity against firing rate. This assumption is well borne out in this paper, Figs. 4 and 10. On this basis if F is the over-all boiler efficiency and x the firing rate in pounds of fuel per hour

$$F = m - nx$$

and, if H is the heating value per pound of fuel, the total evaporation is

[23] Consulting Engineer, Edward G. Budd Manufacturing Company, Philadelphia, Pa. Fellow A.S.M.E.

$$E = KFHx = KH(mx - nx^2)$$

The maximum evaporation occurs at a firing rate x_m such that $\frac{dE}{dx} = 0$; that is, $m - 2nx = 0$

$$\therefore x_m = \frac{m}{2n} \quad \text{and} \quad E_{max} = K\frac{Hm^2}{4n}$$

If the normal firing rate is taken at some percentage k of the maximum firing rate x_m, then $x = kx_m = \frac{km}{2n}$ and, since the ratio of normal to maximum firing rate is

$$\frac{E}{E_{max}} = \frac{mx - nx^2}{m^2/4n}$$

$$\therefore \quad E = (2 - k)kE_{max}$$

With $k = 0.5$ then $E = 0.75\ E_{max}$. If $k = 0.66$, then $E = 0.89\ E_{max}$. This shows that firing rates exceeding 50 per cent of the firing rate, corresponding to maximum evaporation, rapidly approach the maximum capacity of the boiler.

Another limitation of firing rates is dependent upon maintaining a fair over-all boiler efficiency F. In fact, if we limit this over-all efficiency to F_0, then

$$x_0 = \frac{m - F_0}{n}$$

$$\therefore k = 2\left(1 - \frac{F_0}{m}\right)$$

which gives the per cent firing rate relative to that at maximum evaporation corresponding to the limited efficiency F_0.

For example referring to the author's Figs. 4 and 10, the K4S locomotive, which is representative of good average performance and, with a boiler efficiency limited to 65 per cent, the corresponding firing rate is roughly 50 per cent of the maximum. Considering this as a normal firing rate, the maximum evaporative capacity is roughly 33 per cent greater. But with this overload the boiler efficiency rapidly falls.

With Pennsylvania Railroad locomotive M1A, the relatively small slope permits a large overload capacity but at the expense of relatively poor efficiencies at the lower firing rates when compared with the K4S, and similar types.

It would seem, therefore, that boiler proportions which permit higher over-all efficiencies than are found in the M1A, but with less slope of the efficiency-firing-rate curve as with the K4S type, are more desirable, thus giving higher efficiencies at normal firing rates with greater overload evaporative capacity.

It is of interest to note that, in normal locomotive operation, the heat in fuel fired per cubic foot of firebox volume is in the order of 250,000 Btu per hr. In the M1A this reaches over 650,000 Btu per hr at a combustion efficiency of 50 per cent. The Velox boiler attains the latter value at a combustion efficiency around 90 per cent, by means of injecting the combustion air under pressure. The pressure in the firebox is in the order of 30 lb per sq in., effected through a compressor driven by an exhaust-gas turbine. It appears that, in the orthodox locomotive boiler, greater combustion efficiency can be obtained through turbulence of the injection air above the grate as suggested by the author.

In the conclusion to the paper, he cites that it is the engine which deserves the most attention as its thermal efficiency is so low that even a slight improvement is important. This is true from two aspects, engines can work on improved cycles, i.e., with higher pressures and corresponding temperatures, provided throttling losses can be reduced by more efficient cylinder design, and by improved cylinder performance at high speeds.

On the other hand, it is important to point out that, based on the availability of steam supplied to the engine at orthodox boiler pressures, engine efficiencies are remarkably high. The following analysis of cylinder performance with orthodox boiler-pressure and temperature limitations is of interest. The data relate to a 4-6-2 type locomotive with cutoff at 25.5 per cent.

The steam conditions are given in the following table:

Pressure, gage, lb per sq in.	Temp, F	Superheat F	Enthalpy	Entropy
212	748	356	$H_1 = 1395$	$\phi = 1.727$
12	361	117	$H_2 = 1220$	$\phi = 1.782$
	$H_1 - H_2 = 175$		$\phi_1 - \phi_2 = 0.055$	

Steam consumption:

$$\frac{2545}{H_1 - H_2} = 14.5 \text{ lb per hr}$$

Isentropic expansion:

$$\left.\begin{matrix} H_1 = 1395 & H_2' = 1175 \\ \phi_1 = 1.727 & \phi_2' = 1.727 \end{matrix}\right\} \quad H_1 - H_2' = 220$$

Thus, the cylinder efficiency based on the available energy supplied is, $E_c = \frac{175}{220} = 80$ per cent; the cylinder losses amount to 20 per cent of the available energy across the engine. However, these losses have a much smaller ratio, when considering the availability of the power plant as a whole, based on the availability of the combustion gases. The largest loss from an availability aspect occurs in the locomotive boiler itself due to the large differences in flame and gas temperatures and steam temperatures in orthodox locomotives. The largest increase of the unavailable energy components is due to heat transfer. Raising the boiler pressure and thereby the steam temperature to pressures exceeding 600 to 700 lb per sq in. reduces this loss, and results in marked improvement in over-all thermal efficiencies.

Thus, cutting down the unavailable heat-transfer losses in the boiler really amounts to increasing the Rankine efficiency of the engine by operating on a higher pressure-temperature cycle. To accomplish this in any marked degree requires a radical change in boiler design and cylinder design as well, such as including compounding and other features, resulting in a greatly modified type of locomotive.

Another opportunity for increasing the over-all availability is in lowering the back pressures at the entrance to the exhaust. The author has pointed out the significance of this feature in connection with the boiler design.

J. R. Jackson.[24] The performances of the steam boiler as described by the author are predicated on the evaporation of water into steam and involve the factors of fuel and water. In locomotive practice, the control of the grades of fuel supplied is fairly well in hand on most railroads, but the supply of water, as regards the foaming characteristics, is largely as nature supplies it. While it is common practice to treat locomotive-boiler waters chemically to reduce scale and corrosion within the boilers, no practicable method of treatment for the control of foaming within the boiler is known. So-called antifoam compounds aid somewhat in foam control with certain waters, and it is common practice to control foaming by blowing boilers at terminals and on the line. However, in territory west of the Mississippi River, we experience a seasonal foaming condition of the surface boiler-water supply which constitutes a serious operating problem.

[24] Engineer of Tests, Missouri Pacific Railroad Company, St. Louis, Mo. Mem. A.S.M.E.

It was for the purpose of making some studies of the actual conditions of foaming within a locomotive boiler that the Missouri Pacific Railroad in collaboration with the Electro-Chemical Engineering Corporation undertook a series of tests and observations at St. Louis, in August, 1939. For these tests a locomotive boiler was fitted with bull's-eye sight glasses in the dome cover and the interior of the boiler was illuminated with special 1000-w electric lamps to permit making observations and taking motion pictures of the interior of the boiler within and immediately under the dome. A series of standing locomotive tests with different arrangements of steam driers and foam-controlling devices was carried out. The motion-picture record disclosed the following:

(*a*) The relatively rapid movement of the surface water in the boiler in a forward direction, the rate of movement being more or less proportional to the rate of evaporation.

(*b*) The swelling of the surface water with the opening of the main throttle.

(*c*) The development of a spongy or bubbly condition or state of the surface water accompanied progressively by a decrease in the steam space under the roof sheet at the dome entrance; the increase in the velocity of steam movement from the firebox end of the boiler toward the dome; the picking off of the expanded or frothy surface water at the dome restriction; the actual splashing of the water particles and foam carried by the high-velocity steam against the dry-pipe extension and up into the dome against the dome cover; and finally the actual carry-over of water and foam into the dry pipe.

(*d*) The instantaneous subsidence of the expanded and foamy surface water within the boiler with the closure of the main throttle.

(*e*) The control of foaming and prevention of carry-over through the medium of overflow into a foam-collapsing trough, located within the boiler at the surface of the water immediately under the dome and extending along the center line of the boiler toward the rear, and discharge from the trough through an electromatic blowoff connection to the outside of the boiler.

The standing tests were later supplemented by road tests between St. Louis and Kansas City during which additional motion pictures were taken of the interior of the boiler.

L. B. Jones.[25] As a matter of interest, the writer will review the history of the development of the boiler referred to in the paper as the Pennsylvania M1A, which is shown to have exceptionally high evaporation capacity.

The first of these boilers to be built was not successful, so much water being carried over with the steam that no superheat was obtained. After some experiments, it was found that the performance was considerably improved by lowering the water level, the indications being that, if the steam space could be increased sufficiently, the trouble would be overcome. The boiler was accordingly redesigned, with a lower crown sheet and the dome barrel course raised slightly. The steam space was increased from 150 cu ft to 199 cu ft which, expressed in per cent of the total cubical content of the boiler, amounted to an increase from 16.5 to 21 per cent. The total evaporative heating surface was reduced from 4499 sq ft to 4319 sq ft, and the superheating surface was reduced from 2283 sq ft to 2052 sq ft. In spite of these reductions in surface area, the evaporative capacity of the boiler was increased, and a steam temperature of 730 F was maintained at very high rates of evaporation.

The author's comments on the necessity for liberal fire-tube area are deserving of special attention. It is pointed out that the energy required to move the furnace gases through the flues increases with the fourth power of the weight velocity, which involves the expenditure of power in the shape of cylinder back pressure at the exhaust nozzle. It is also pointed out that a reduction of 1 lb back pressure is equivalent to an increase of 4 lb in steam-chest pressure for average rates of working. When we consider the relatively low value of tube heating surface as compared with firebox heating surface, it becomes evident that the locomotive as a whole can be materially benefited by an attack on the space between the front and back flue sheets, in a way which will provide greatly increased flue-gas area, with a possible accompanying increase in superheat area, without doing any damage because of reduced tube evaporating surface, which is not needed.

We are inclined to take issue with the author on one point, namely, that the gas area can be increased at the expense of water space in the water legs of the firebox. While many boilers are operating with water space as low as 5 in., we believe that boiler-maintenance experience, compared with boilers having more liberal water legs, will eloquently support the wider space, with its accompanying longer stay bolts and reduced angularity of flexure. Important as flue-gas area is conceded to be, it should be obtained in other ways than by sacrificing water space along the sides of the firebox.

In Figs. 6 to 10, inclusive, the author has tabulated some interesting data on boiler and combustion efficiencies. Considerable danger is involved in comparing boiler efficiencies from tests made at different times and places, and with different fuels, especially at the high rates of combustion necessary in a locomotive boiler. This is particularly true of the various Pennsylvania locomotives reported. The test data for all these locomotives were obtained at the locomotive test plant, but the tests were made at different times, using different methods of firing, and somewhat different fuels, with the result that the relative boiler efficiencies plotted in the diagrams do not faithfully reflect the true characteristics of the boilers. We have found that, when it is desired to compare conditions involving boiler efficiency, it is necessary to confine the source of fuel to one mine room or entry, and otherwise safeguard the handling of the fuel from the mine to the locomotive firebox. Two fuels of equivalent Btu content may show widely different boiler efficiencies in the same boiler because of texture and friability, as well as other variables.

W. G. Knight.[26] The author's deductions as to the limitation of existing methods for calculating boiler efficiency are certainly correct. The writer feels that Mr. Fry's formula[7] would no doubt be adequate if more data were available for the value of the constants involved.

We have run standing tests on the Bangor and Aroostook Railroad which would bear out the New York Central contention that front-end design and nozzle size can be tested in this manner and properly adjusted for road service.

The curve in Fig. 2 of the paper is interesting and has special value due to its being drawn from data concerning so many engines. Boiler diameter and gas area, of new power especially, could be more clearly considered after a study of data of this kind.

When considering the distance from the top of the crown sheet to the boiler shell, it is advisable to do everything possible to increase heating surface in order to produce maximum boiler capacity and keep within the weight limitations. However, because of the general tendency of engineers to carry high water rather than low, it is the writer's belief that too high a crown sheet will decrease dryness of the steam, lower the superheat, and otherwise affect lubrication and general performance.

We have found on the Bangor and Aroostook that a ratio of

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[26] Mechanical Superintendent, Bangor and Aroostook Railroad Company, Derby, Me.

from 5 to 6, firebox volume to grate area, is decidedly advantageous as the road characteristics demand high boiler capacities for freight service. It is also possible with large fireboxes to adjust the arch to decrease the high velocity of gases at point *A* in Fig. 3 and bring them down to a value equal to or below the velocity in the tubes near the back tube sheet.

During the last 5 years, we have tried several designs and a number of different percentages of air openings in grates, our latest experiment being made on an application with 60 per cent free air opening. As yet, we have not definitely proved its advantage since we are still changing the draft appliances and ashpan arrangement to achieve better combustion efficiency. We feel that a gain will result from using this large percentage opening.

Much has been said in the last year or so on the subject of secondary air for combustion. Whether it would aid the older locomotives without considerable change in their design is a question. It would seem at present that equal efficiency can be obtained by an increased grate area and proper changes in draft appliances.

The general trend of all roads recently has been toward much more efficient draft appliances. We have tried to keep pace with the ever-increasing demand for more horsepower by improving front-end design and decreasing back pressure. The improvements made along these lines have produced satisfactory results.

A. I. Lipetz.[27] The locomotive boiler as part of the steam locomotive has the value advantage of automatic self-regulation; in other words, to a certain limit it adjusts its evaporation to the demand of steam. This automatic feature is actuated by the back pressure from the engine. For a given back pressure the amount of steam exhausted by the engine through the nozzle and the draft of the boiler is fixed. A certain proportion of the combustion gases is then pushed through the flues and tubes of the boiler from the firebox into the front smokebox. This evaporates a certain amount of steam, which in turn is exhausted by the engine. Thus the cycle is completed and then repeated. If the speed and cutoff of the engine are varied, in accordance with the demand upon the locomotive, the back pressure and the evaporation are varied—this is the automatic self-regulation feature.

By assuming a certain back pressure and the resulting draft, and figuring the amount of fuel burned in the firebox, it would be possible to calculate the amount of heat which is thus generated. Being familiar with the laws of heat transmission, and depending upon the gas velocities through the various sections as explained by the author, it would be equally possible to evaluate the amount of steam evaporation. For another back pressure and another fire rate, a different amount of steam evaporation would follow; thus the curve of steam evaporation, as a function of coal firing and back pressure, could be established.

It is evident that this way of figuring, although logical in itself, would involve a great many calculations on the basis of laws of natural phenomena, which are not always known. It is therefore simpler and more reliable to establish these curves by tests with locomotive boilers and to plot the curves, as has been done by the author in Figs. 4 and 5 of the paper. Likewise from tests, the combustion efficiencies can be plotted, the boiler and superheater over-all efficiencies can be figured, and, for further study, depending upon the length of tubes, mean hydraulic depth, etc., the heat absorption and various other factors can be established.

The value of this paper consists of giving us concisely a complete series of curves from reliable tests as material for study. The author did not attempt to draw any conclusions except to say that the present locomotive boiler is a very efficient steam generator, when compared to stationary boilers, especially taking into consideration its simplicity and weight.

However, the writer would like to call attention to the possibilities obtainable from a proper review of the author's curves. From the foregoing statement of the automatic regulation of the boiler cycle, it is obvious that, in application to a locomotive boiler with the same principal fundamentals of design, the efficiency and production of the boiler should be more or less uniform, depending upon certain ratios. These can be established from tests. In analyzing Figs. 7, 9, and 10, especially the latter, it is evident that the efficiency curves are straight lines, as is well known from the investigations of Professor Goss and Mr. Fry.[7] Fig. 12 is still more interesting because it narrows the system of straight lines for a great variety of locomotive boilers tested in Altoona and in actual service on different roads. It also brings out the fact that the Pennsylvania M1A locomotive seems to be represented by a line about the average of all others.

This, together with Tables 1 to 6 given in the paper, permits making a comparison of the various ratios tabulated therein. Thus it can be seen that item 31, which is the gas area per square foot of grate, varies for these locomotives and for these locomotives only, between narrow limits of 1.14 and 1.41 (columns 3 and 4 in Table 1; columns 1, 6, and 10 in Table 2; column 10 in Table 4; columns 6, 7, and 9 in Table 5). For the class M1A locomotive it is 1.39. Further, the ratios of firebox heating surface to the total evaporating heating surface (item 40), of the total evaporating heating surface in relation to the grate area (item 42), and of the superheater heating surface in relation to the total evaporating heating surface (item 43), vary also between limits which represent the best results of American practice.

That these ratios are important and essential has been known for a long time by a great many investigators of American and European locomotives.[8,28,29] They are usually included in articles describing new locomotives. This paper provides an opportunity to study these figures and to establish such ratios as we might consider to be essential. Tables 1 to 6 will permit establishing rules at least for the following boiler ratios: The ratio of the gas area through the boiler to the grate area; the firebox heating surface to the total evaporating heating surface; the total evaporating heating surface to the grate area; the superheater heating surface to total evaporating heating surface.

Of course, variations in these ratios are always necessary, depending upon the grade of fuel and local conditions. However, the writer does not believe that the difference in local conditions justifies the great variations in locomotive design which prevail. Certainly, a certain degree of standardization can be introduced for locomotives on American railroads.

T. C. McBride.[30] This paper indicates the inconsistencies in the proportions of existing boilers and demonstrates the need for further tests to establish the best proportions. The Cole method[8] for determining the evaporation to be expected from locomotive boilers, having the evaporation values based only on heating surface, is condemned as inadequate. Nevertheless, it is stated that "today these values are generally used by the locomotive builders and railroads." With no comparable substitute method in sight, an effort should be made to adapt the old and almost universally used method to present conditions.

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[28] "Die Eisenbahntechnik der Gegenwart," by E. Brückmann, Berlin, 1920, Appendix.

[29] "Handbuch des Dampflokomotivbaues," by Prof. Dr. Igel, Berlin, 1923, pp. 73–86.

[30] Consulting Engineer, Railway Division, Worthington Pump & Machinery Corporation, Harrison, N. J. Mem. A.S.M.E.

It is necessary to reach an understanding of the various terms used throughout the paper to indicate capacity, evaporation, etc. Referring to the first table of the paper, the Strahl formula need not be considered because it is not used in this country. The "Cole evaporation" of this table might be defined as Cole's conception of the maximum evaporation which boilers of about the year 1910 could be depended upon to develop when in operation on the road. Dr. Lipetz, in his 1932 paper[5] on locomotive ratios, endorsed such a definition but applied it to boilers built 20 years later. It is generally conceded now that this evaporation presumed by Cole was very high for the boilers of about 1910 and was not realized until some years later with boilers of better proportions; but today that is immaterial if proper coefficients are applied. The "actual test evaporation" of this table is the ultimate evaporation possible of attainment under the favorable conditions of a standing test of short duration, after elaborate preparations had been made. Often a number of attempts have been made in an endeavor to obtain the highest possible figure. These could not be depended upon for road operation. The Cole evaporation should therefore be lower than the actual test evaporation; the fact that it is lower does not indicate inadequacy but rather confirms its adequacy for the purpose for which it was intended.

The Cole evaporation of the Pennsylvania Railroad K4S locomotive is stated in this table to be 52,150 lb per hr. The actual test evaporation of 65,400 lb given in Table 5, column 6 for the K4S of 1914 should be compared to the Cole value. This was only 25 per cent greater than the Cole, certainly not too great a margin for dependable operation on the road, as implied in the definition of Cole evaporation. The much higher actual test evaporation stated in the table for the Pennsylvania Railroad K4S; the actual test for the Pennsylvania Railroad M1A, 46 per cent higher than the Cole; and the actual test for the New York Central J3A, 45 per cent higher than the Cole, all follow developments and improvements which place these boilers in a later class than was contemplated by Cole. They require the application to Cole evaporation values of a coefficient, such as the "evaporation coefficient" β, of Dr. Lipetz' paper.[5]

However, the maximum evaporation of the greatest importance is that now demanded from modern boilers in operation on the road. A fairly accurate estimate of this evaporation may be obtained by using the pumps of the Worthington heaters as water meters. The pump speed, noted when the locomotive is operating at the highest power output likely to be demanded of it, indicates demands for water up to 115 per cent of the Cole evaporation, that is, an evaporation coefficient of 1.15 for coal burners, with a still higher demand for oil burners.

Twenty years ago, injectors having a capacity 75 per cent of the Cole evaporation were quite usual, at least in the eastern part of the United States. At that time, one 75 per cent injector supplied enough water for most of the operation. Coal burners in the West were operated at a higher evaporation, and oil burners at a still higher rate, but even then the latter did not attain the Cole evaporation. The increase in the evaporation coefficient from not much over 0.75 to 1.15 and the higher pressure and temperature of superheated steam are measures of the progress made in 20 years in feedwater heaters, stokers, better proportions of boilers, and front ends.

Returning to the first table of the paper and applying an evaporation coefficient of about 1.15 to the Cole evaporation of the modern or modernized boilers, it is noted that the Cole value, as modernized by the coefficient, takes its proper position in relation to the actual test evaporation of the table.

While only the proportions of the boiler, principally the firebox proportion, had to be considered, it was not difficult to follow the increasing evaporation coefficient and to predict the evaporation to be expected and the amount of water the boiler would require. But, with the appearance of much more efficient front ends, an additional coefficient, presumably a draft coefficient, is needed.

The author rightly urges more tests to aid in the determination of the best proportions for the boilers of locomotives. However, it is possible that much useful information could be obtained from data of tests already available, if curves were to be drawn showing the relation between boiler efficiency and percentage of Cole evaporation. The boiler showing the highest efficiency at the Cole evaporation would then represent the best proportions, at least as a first approximation. Corrections for differences in draft and other items might be necessary. Comparison of the height and slope of the efficiency curves would indicate which boiler proportion gave the highest efficiency at the higher capacities and therefore promised the highest practical capacity; that is the information desired.

Existing conditions are forcing rapid developments in the boilers of locomotives, especially in capacity. Prompt collection and analysis of all available data are urgently needed. In the writer's opinion useful conclusions can be reached in the shortest time by the determination of the coefficients and the methods reviewed in this discussion.

C. L. MEISTER.[31] Railroad mechanical officials know that in these days of faster train speeds their locomotives must have ample steam capacity to take care of, not only the long-sustained runs, but all the auxiliary demands requiring steam; and careful design of the boiler is the first essential.

This paper gives many timely suggestions: Ample combustion space; reduction at the vital points of the gas flow and the necessity of large gas areas; the relation of boiler diameter to total weight; grate area; flue efficiency factor; front-end design; and others.

A copy of this paper should be in the files of every railroad mechanical officer, and studied not only by the chief, but by his draftsmen as well. It should be carefully studied by locomotive builders, who so often are prone to copy and perpetuate older designs with slight improvements here and there, based on road-performance experience in lieu of scientifically conducted tests.

Although the author has correctly stated: "The superheater holds first rank of all the improvements made on the locomotive since George Stephenson's time," he has refrained from going into detail on this phase of the subject. The writer would like to have further information on the following matters:

1 What is the practical limit to superheat temperatures on the present type of reciprocating locomotive, and what effect has superheat as high as 350 to 400 F, as used in Europe, on the economy and maintenance of locomotives?

2 What are the important problems affecting the design and materials of boilers and superheaters in connection with high steam temperatures?

3 On the basis of final economy would the greatest advantage be obtained by the use of higher boiler pressures or higher superheat, or is there a combination of pressures and superheats which would give the best ultimate economy?

4 In Table 5, column 5, relating to the Pennsylvania Railroad class S1 locomotive, it is noted that this engine is equipped with a type "A" superheater having a large steam area as compared with other types. Is this a special type which permits such a great increase in the steam area, and what should be the advantage, if any, of using such a design?

5 The author points out the desirability and effectiveness of type "E" superheater design in modern boilers, and a review of

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the tables indicates that this type of superheater has generally been applied. It would be interesting to have a concise statement of the principal differences in the various superheater designs and the improvements which can be obtained by the use of the type "E" superheater.

6 Other features on which further enlightenment would be advantageous are the problem of unburned coal, due to large gas velocities, and the location of the stoker. Would not locating the stoker at the front, instead of back, of the firebox be an advantage even if it meant reducing the coal sizes so as to be more easily handled? Would the problem of high gas velocities and resultant losses caused by unburned coal be improved by the use of pulverized coal?

7 In the paper and conclusion, the author refers to the competitive condition of Diesel and electric locomotives. Has a real economic analysis been made of the comparative economics of the modern steam locomotive with that of the Diesel or electric operation, both as to availability, first cost, operating cost, and maintenance?

R. M. Ostermann.[32] The author's wide experience in designing superheaters for locomotives, and proportioning them properly for best over-all performance, has perhaps suggested to him that it might be superfluous to discuss the limitations of locomotive-superheater design in a more detailed manner; his discussion on superheaters is brief indeed. And yet, it would seem, since the superheater is such an integral part of the locomotive boiler, and its design is so bound up with the design and performance of the boiler itself, that locomotive designers as well as the users thereof would have been interested in a more thorough exposition of the various factors which not only determine boiler capacity and boiler efficiency, but also the capacity and efficiency of the superheater, the steam pressure loss which it creates, and the added draft requirement which is the result of its installation. The practice of locating the superheater heating surfaces within the flues has evolved from several past trials of other arrangements, and now can be considered a virtual standard wherever steam locomotives are used. The practice has become standard because superheating the steam either in the smokebox or in the firebox was found to be impractical for a number of reasons. Thus, one is confronted with one general limitation of the locomotive superheater in that it is heated by gases, the temperature of which is that available within the locomotive-boiler flues. Such temperatures generally range between 600 and 2000 F.

The author discusses at length the factors which affect the heat absorption of the evaporative heating surfaces of the boiler barrel when in contact with gases of such temperature. These same factors, of course, also influence the superheating capacity because the latter depends upon the mass velocity of the gases flowing over the superheater units, and the temperature difference between the gases and the steam to be superheated, from point to point in the steam path.

There is, however, one essential difference between the mechanics of heat transfer, as between the gases and the water, and as between the gases and steam. The speed of water circulation along the flues does not affect the total heat-transfer resistance on the water side of the flue nearly as much as the speed of the steam moving within the superheater-unit pipes affects the heat transfer between it and the gas. Experiments, covering such heat transfer, are on record and reveal that the specific heat-transfer resistance from the superheater tube to the steam varies inversely with the 0.75 power of the steam velocity. Whenever the velocity of the steam flow is reduced in the presence of an undiminished heating effect of the gases, heat is impounded in the metal of the pipe and its temperature is raised until the adverse effect of the lowered steam velocity upon the heat transfer between the pipe and the steam is compensated by a larger temperature difference between the two. On the other hand, the rise of the metal temperature causes a smaller temperature head between the gas and the metal, and a lessened heat transfer per square foot of superheater heating surface takes place unless compensation is made through a greater mass velocity of the gas flow. This eases up the specific heat transfer from the gas to the metal but, incidentally, also increases the draft loss.

Quite naturally it would be desirable to build superheaters in locomotives which cause as little pressure drop of the steam as possible, for obvious reasons. However, if the physical laws of heat transfer are kept clearly in mind it is recognized that the superheater capacity, the evaporative capacity of the boiler barrel, the draft loss through the boiler barrel, the heat-absorption efficiency, and the pressure loss in the superheater are all interlinked with the size of the available tube-sheet area.

Locomotive superheaters may well be designed for special low pressure drops, but superheating capacity or draft loss or evaporative capacity have to be sacrificed, and such sacrifices may affect the maximum power output of the locomotive more than the pressure drop through the superheater, which, after all, is not a thermal loss. This emphasizes the author's statement that reduction of back pressure is relatively far more essential than a reduction of superheater pressure drop, pound for pound. Resort may be had to the so-called single-loop superheater which halves the steam velocity, and the tubes may be made of special alloy steel so as to operate them safely enough with the resulting higher metal temperature. However, this would result in a lowered superheating capacity. The steam velocity and pressure drop may also be reduced by increasing the diameter of the superheater-unit pipes but, in that event, either evaporative heating surface must be sacrificed or the net gas area through the boiler reduced. Such limitations do not exist in other types of boilers in which the superheater may be placed at will into the gas stream where it is of a higher temperature than the one prevailing in the flues of a locomotive boiler, and, therefore, where the length of the steam path can be correspondingly shortened. At such a point in the gas path, it may also be possible to dimension the net gas area more liberally and without creating any undue draft loss. In such boilers, low-pressure-drop superheaters can then be more easily provided for than in locomotive boilers.

The superheater research to which the author refers, actually produced several types of superheater units usable with $5^1/_2$-in. flues, which are fundamentally different from the well-known double-loop unit, and which, by reason of this difference, are capable of producing, with the maximum gas load through the flue, a superheating effect which is about 20 per cent greater than with the double-loop design of unit. Unfortunately, however, this type of unit is sufficiently more complex in its structure and sufficiently heavier than the double-loop unit as to make it relatively impractical for the highest boiler pressures which are now used on steam locomotives. It is, nevertheless, interesting to mention briefly that the substantially increased heat transfer to the superheater heating surface of the unit mentioned was only partially obtained at the expense of the evaporating capacity of the flue. In other words, in spite of an increased heat transfer to the superheater heating surface, the total heat absorption of the flue-unit combination could be improved; the draft loss was no greater than it had been through the flue with the double-loop unit. This rather gratifying result was obtained by utilizing counterflow between the steam and the gases to a greater extent than is possible with the double-loop-unit construction, and also by special means designed to prevent the temperature of the gases in contact with the superheater heating surface from dropping as

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much as the temperature of the gases in touch with the water-cooled surface of the flue.

With respect to the draft loss through the barrel of the boiler, it should be remembered that it is only a part of the total draft loss through the furnace and boiler as a whole. Thus it must be decided from case to case how much justification there is to decrease the hydraulic depth of the boiler barrel as a whole without running the risk of overloading the draft-making front-end arrangement, and thereby unduly increasing the back pressure of the locomotive. Due to the interaction of the superheater proportions with those of the front end, of the boiler barrel, and of the furnace arrangement—not to speak of differences in fuel—it is impossible to expect that a given standard of superheater design perform in exactly the same manner in locomotives with a wide variety of boiler proportions and operating conditions. Yet, in the interests of maintenance, it is imperative that certain standards of superheater design be adhered to, and it is this phase of the author's subject which the writer particularly wished to bring out.

N. A. Pownall.[33] There is no doubt but that present formulas are inadequate for determining the probable maximum power of a modern locomotive. This is particularly true since the advent of such capacity- and efficiency-increasing appliances as the superheater, feedwater heater, siphon, etc., as well as the effects of variations in exhaust-tip size, stack diameter, front-end appliances, air openings through grates, and various features of boiler design mentioned in the paper. It will be of considerable value if a formula can be developed which will make it possible with fair accuracy to predict the power of a certain design of boiler. It is to be hoped that, by means of the increasingly used standing-test method, such a formula may ultimately be developed, at the same time establishing the best method of boiler design as regards furnace volume, lower gas velocity, greater steam space, etc., as discussed by the author. Research work of this nature will be necessary, for there are certain changes, which are even now being made, that thus far have produced unpredictable effects. For example, opening up the exhaust tip is desirable from the standpoint of reducing energy wasted in back pressure and yet, if this is carried too far, though the engine may still steam well, temperatures may have been lowered so as to affect adversely the superheat, with a resulting negative economy. Any drop in temperature of the superheated steam is attended by a positive rise in fuel consumption.

The steam space above the water level is certainly important. Treated water is now considered a necessity for locomotive boilers, and any antiscale treatment will necessarily increase the foaming tendency of the water, resulting in a greater volume of water bubbles in the steam space, which, because of space limitation, is relatively small for the tremendous steam-producing rate of a modern locomotive. Careful study of the boiler design, in order to increase steam-liberating surface and, by gaining a little here and a little there, to increase steam space above normal water level, will be of considerable value in reducing water carry-over. It is noted that experiments are now under way with a trough or box directly under the dome which increases the steam space at the main-throttle location and offers promise of some relief.

It is astounding to learn that the velocity of gases through the back of flues exceeds 200 mph, and over the arch 260 mph. Therefore, it is not difficult to realize why losses in unburned fuel and stack losses are so high, why stay-bolt heads are cut by cinders, and why there is excessive slagging of flue sheets when the boiler is worked at high rates. As a result, ample gas area is very important, though here again space limitation interferes. Advantage must be taken of small gains here and there; such as, decreasing the space between the combustion chamber and the boiler shell. If water is kept fully treated, and the boiler systematically blown out in service and given thorough washings, this decrease may be made with no bad results.

The cost of boiler maintenance is stated to be 30 per cent of the total cost of locomotive repairs. This is too high. Years ago, the writer had occasion to keep records of the repair costs, divided between the boiler and items not pertaining to the boiler, and found the percentage of boiler repairs to vary from 9 per cent on the best treated water division to 38 per cent on the worst untreated water division. With proper water treatment (and it is prehistoric for any railroad to be without it), this percentage should be kept below 15.

As the author states, at the high combustion rates it is quite important to have maximum possible firebox heating surface and furnace volume, thus reducing the loss of sensible heat up the stack by reducing front-end temperatures; a 210 deg decrease in front-end temperature causes a 7 per cent fuel saving. However, this must not be carried to the point of causing reduction in the temperature of superheat.

Few people realize the high horsepower obtained from this boiler plant on wheels, many 4-8-2 and 4-8-4 type locomotives being capable of developing from 4000 to 5000 hp. We must also pay a tribute to the feedwater heater which heats the water for this 5000 hp up to 210 or 220 F with apparatus of a size that can hardly be seen on the locomotive. Yet under the restricted conditions, we are still striving by intelligent and seemingly insignificant changes in design to develop this high power at even greater efficiency.

Unfortunately trains have to run light, and the engine and boiler do not perform as well as under a heavier loading. The locomotive must be so designed as to operate reliably under all conditions occurring on the territory to which it is assigned. It is preferable and desirable to have a locomotive of general utility but, in view of the many factors entering into the economical development of maximum power, it may be advisable, where it can be done, to adapt design to specific operating conditions.

Our experience bears out the author's remarks concerning front-end design and draft. In this connection, it may be of interest to describe some changes that were made on 4-8-2 class freight locomotives. These engines have 27 × 32-in. cylinders, 245 lb per sq in. boiler pressure, 70-in. drivers, 69,400 lb tractive power, total heating surface 4620 sq ft, grate area 84.2 sq ft, and are equipped with type "E" superheaters, firebox siphons, and feedwater heaters. According to Cole's ratios, their normal expected horsepower was 3214. When received, they developed 3650 ihp, illustrating the inadequacy of this formula for ordinary modern design.

The grates were a finger-bar type with 43 per cent air opening, master mechanic's front end with $6^1/_2$-in-diam round tip, and 19-in-diam stack, and these were changed to 14.9 per cent air opening tuyère-type grates, a Kiesel front end with six-opening 53-in. star-shaped tip, and a $25^1/_2$-in-diam stack. Combustion was better, excess air was considerably reduced which entails a fuel saving, back pressure was reduced, draft was noticeably improved by reason of less impedance to flow of gas in the front end, and the larger stack evidently contributed considerably to greater power. The locomotive developed consistently over 4900 ihp, which confirms the author's feeling that a new formula is needed to measure the power of modern locomotives and the effect of changes in the important parts of the boiler.

We have built locomotives, making use (possibly inadvertently) of correct principles which have resulted in unexpectedly high power; yet now and then we overlook some unrecognized, important feature and are disappointed in the result.

[33] Assistant to Superintendent of Motive Power, Wabash Railway Company, Decatur, Ill.

Mr. Brandt's paper is very comprehensive, should give designers plenty of food for thought, and the writer feels that in the foregoing he has added little, but has tried to confirm Mr. Brandt's reasoning and conclusions.

F. P. Roesch.[34] The writer prefers to leave questions of locomotive design and construction details to those better qualified, and confine his comments to accessories which are not considered as a part of the boiler proper but, nevertheless, have a material effect upon both boiler and locomotive performance.

The Ashpan. During Federal Administration of railroads, a committee from Section 3, Mechanical, American Railway Association (A.R.A.), among other suggestions tending toward fuel economy, reported[35] as follows:

"Many of the older engines are still equipped with ashpans having insufficient air openings, a defect readily detected with a U-tube draft gage. There should be no indication of vacuum in the ashpan. Correction of this condition recently permitted an increase of $^1/_2$ in. in the nozzle diam of several locomotives and not only effected a substantial saving in fuel, but greatly increased the efficiency of the locomotives."

Prior to that time, tests indicated the desirability of free air openings into the ashpan, having a combined clear opening of not less than 14 per cent of the total grate area. This was based on the total air openings through the grates and its relation to total tube area. But apparently these recommendations, as well as those of the American Railway Association committee have either been forgotten or are disregarded for items of presumably greater importance.

Perhaps it may be well to check into the matter to decide the point. A definite amount of air is always required in the combustion of various kinds of coal, as pointed out by the author. Where the area of the supply ports for such air is restricted below requirements, the velocity of supply must be increased. To obtain this velocity we must increase the intensity of the draft. This is usually done by reducing the area of the exhaust-nozzle opening, thereby increasing the cylinder back pressure, so when the cycle is completed, we find that what should be free air has been obtained at the expense of cylinder horsepower.

At this point the writer wishes to make it clear that we are not speaking of test-plant methods in this discussion, but of conditions and methods obtaining in general road service, where a reported engine failure through any cause is a serious matter and any course is often thought justified which will eliminate such reports. At any rate, an engine failure is something real that everybody hears about, while decreased locomotive efficiency is only revealed by the indicator.

However, decreased locomotive efficiency is not the only evil resulting from a restricted air supply. On the contrary, many others such as decreased combustion efficiency, increased stack loss, cinder cutting, etc., all follow in its wake, as will be shown later.

Locomotive Grates. During Federal control a Fuel Conservation Department was organized. One of its aims was to check and reduce waste. In connection with that job, it was noticed when the fire was laid preparatory to firing up a locomotive, that from 500 to 2000 lb of the smaller-sized coals fell through the openings in the grates into the pit over which the locomotive stood. As such coal was afterward covered with ashes falling through the same grate openings, recovery of the unburned coal presented some difficulty and, therefore, was seldom salvaged. To overcome this waste, it was recommended that the grates be covered with perforated sheets of heavy paper before the coal was introduced. This fuel loss received wide publicity. Therefore, when the railroads were returned to corporate operation, many of them applied the logical remedy, where conditions warranted, namely, better-fitted grates with smaller openings.

But little is known of the physicomechanical reactions or behavior of the fuel bed in a locomotive firebox at relatively high rates of combustion. So little in fact, that many of our theories as to what takes place are based on such an insecure foundation that conclusions reached are questionable. The chemical reactions are known and serve as a guide toward improving the combustion of such coal as is actually burned, but this is not what we are concerned about at present. What we need is more definite information covering stack loss, draft currents, both as to direction and velocity, between the fuel bed and the underside of the arch, etc., conditions which can only be observed through small openings cut through the sides, or larger openings in the backhead, covered with some transparent substance. We recommend the latter, as the range of vision through the sides is too limited, although their addition to backhead windows will be helpful.

It has been the writer's privilege to observe some of the physicomechanical behavior of the fuel bed through side holes during tests on a test plant, and later through a large covered opening through the backhead. In the latter case, however, the locomotive was equipped with forced draft. Nevertheless, the behavior of the fire in both instances was quite similar, at least enough so to enable us to set up certain theories, which later practice proved to be sound.

These observations indicated that, while the gases moved in the direction as generally pictured, viz., from front to rear, their velocity above the surface of the fire bed was relatively low until they reached a zone immediately under the rear end of the arch. Here the movement of the particles of coal entrained in the gas currents indicated a point of excessive turbulence and high upward velocity, Fig. 3 of the paper.

This was quite interesting, but the most interesting phenomenon was the behavior of the small lumps of coal lifted above the fire bed by the action of the air jets passing upward through the fire. It was particularly noticeable that in many instances one lump of coal was projected high and far enough to enter the zone of high turbulence, where it disappeared, while another lump of apparently the same size and rising from the fuel bed within 6 or 8 in. from the first one, was only lifted about 1 ft or so and then fell back and was burned. As both were presumably subject to the same pressure or velocity influence, the question excited much interest. Further observations indicated that the same phenomenon occurred at other points. In fact, the fuel bed had the appearance of a group of miniature volcanoes spouting fire and ashes, some far up and others only high enough to overflow the rim.

After the fires were dumped, the reason for all this became clear, viz., difference in size of air openings through the grates. On this observation was based the theory that the lifting action of air jets passing upward through the fire bed was in proportion to the squares of their areas. This theory has been proved and the foregoing observations are largely responsible for the present general use of what are called restricted grates. In these it is not so much a question of the ratio of air openings to total grate area, as a question of the size of the openings, their spacing, and the fit of the bars against each other and against the side and center carrier bars. We do feel, however, that the total air openings need not be much in excess of the total tube area. Or to put it another way, knowing how much air is required to burn 1 lb of coal, air openings can be proportioned accordingly, thus mechani-

[34] Vice-President, The Standard Stoker Company, Chicago, Ill. Mem. A.S.M.E.

[35] "Ash Pans," Report of Committee on Fuel Economy, Proceedings, Section 3, Mechanical, American Railway Association, 1919, p. 481.

cally restricting the supply needed, instead of leaving the matter to the judgment, or lack of judgment, on the part of the fireman.

Needless to say, the general adoption of restricted grates was hailed by locomotive-stoker manufacturers, as it fully supported their contention that stack loss was not a question of coal size as fired, but rather one of air openings through grates and draft influence. Furthermore, stack loss far antedated mechanical stokers and, according to a University of Illinois bulletin,[36] lump coal breaks down during the process of combustion (naturally) and thus accounts for the excess in cinders collected above the amount of fines in the coal as fired. As a matter of fact, a higher heat release is obtained from small-sized coal as fired, other conditions being right, than from lump coal which must be broken down during its combustion.[36]

Draft Streams. In a preceding paragraph, the statement was made that the velocity of the gas streams, moving from the forward part of the firebox to the rear, was relatively slow. In support of this statement, we offer the fact that while the locomotive is in operation it requires a greater distributor-jet pressure to blow a lump of coal from the front of the firebox and presumably with the draft, than it does to blow an equal lump from the rear to the front, and again presumably against the draft. This fact came to light only recently with the introduction of the so-called front-delivery stoker as developed by the Baltimore & Ohio Railroad. However, an approach to the problem with an open mind and no fixed convictions as to what really happens in a locomotive firebox when coal is being burned at relatively high rates, enables one to arrive at a logical solution, namely, the momentum imparted to any mass, in this instance a lump of coal, is proportionate to the velocity or force of the steam jet impinging against it. This should hold true regardless of whether the mass is projected from the rear to the front or from the front to the rear of the firebox. The velocity of the jet is as the pressure. Therefore, it follows, if higher jet pressures are required to blow the coal rearwardly and presumably with the draft stream, it must be subject to other resistance than that of gravity.

This we find the author has clearly indicated in Fig. 3 and subsequent analysis. Reasoning along these indicated lines, we draw the conclusion that the initial momentum is gradually reduced and, when the mass reaches the zone of greatest turbulence and high velocity, it no longer has sufficient momentum to carry it through that zone unless the initial velocity is increased through higher jet pressures. On the other hand, where the coal is introduced at the rear of the firebox and blown forward, the introduction takes place within the high-velocity zone, therefore the mass even at a lower initial velocity still has sufficient momentum to carry it through this zone and deliver it to the forward part of the fire bed.

This theory if accepted, raises another interesting problem. Tests on the Baltimore & Ohio Railroad have definitely shown the advantage of the front-delivery stoker; for instance, a 50 per cent decrease in stack loss, 15.9 per cent reduction in fuel per 1000 gross-ton-miles, and 15.5 per cent increased evaporation per lb of coal, together with the practical elimination of smoke. There is no question as to the correctness of the published data, as the reason for this improved performance is self-evident. However, as the comparisons were made between the front-delivery stoker, which delivers the coal at a point about 14 in. above the normal grate level, and one effecting delivery at a much higher level, there is the possibility that the same performance differences may not obtain where the coal is introduced with a rear-delivery stoker, firing from a level corresponding to that of the front-delivery stoker, for the following reasons:

The statement has been made that the lifting action of the air jets passing upward through the fire bed was as the square of their areas, the velocity remaining the same. In the following, it must be understood that the figures are not at all accurate since, owing to the resistance and varying thicknesses of the fuel bed, accuracy is out of the question, but the comparisons can nevertheless be accepted as valid.

Assuming the author's data as given in Fig. 3 to be based on a grate having an area of 100 sq ft, we find from established tests that an approximate draft equal to $2^1/_2$ in. of water is required to burn coal at the rate of 90 lb per sq ft of grate per hr,[37] and this, neglecting fuel-bed resistance, results in a velocity efflux of 103.30 ft per sec.[38] In contrast, at the higher rates, viz., 236 lb per sq ft of grate, we must have a draft equal to 4 in.,[37] with a corresponding velocity of approximately 130.43 ft per sec.[38] However, in the first instance, due to the expansion of gases, the velocity is increased to 198 ft, and in the second to 386 ft where the gas stream passes over the arch.[8] Therefore, where the fuel is introduced at the lower levels, a lesser amount is caught up by the higher-velocity streams.

Since the velocity of the gas stream, where it passes over the arch, is proportionate to the volume to be moved and the area of the opening between the top of the arch and the crown sheet, it would appear that any increase in this area would decrease correspondingly the velocity at that point. This brings us to a consideration of the setting of the arch.

The Brick Arch. The value of the brick arch is well known and requires no further discussion. Recommended practice[39] suggests a gas area over the arch equal to from 110 to 120 per cent of the minimum net tube and flue area. This is to increase the length of flame travel and reduce the stack loss. However, according to the author, this results in a gas velocity at that point of 198 ft per sec at low rates, and 386 ft at high rates of combustion, while the corresponding velocity through the tubes and flues is 157 and 300 ft per sec, respectively.

The writer believes in a long arch but, as has been shown, the entrainment of the smaller particles of coal is influenced by the velocity of the gas streams. Theoretically, any reduction in such velocity should in turn affect entrainment and, while theory does not always check with practice, it does appear reasonable to assume that some modification in arch application which would tend to equalize gas velocities might be worth our consideration. As a suggestion based on tests conducted by the writer, we propose, beginning at the rear, longitudinal rectangular openings between the arch and both side sheets about 4 in. wide and 36 in. long, leaving the center of the arch to the recommended length. With such an arrangement much of the fines entrained in the gas streams would come directly in contact with the water-cooled firebox sheets, cooling such particles and preventing their adherence to the tube sheet in the form of honeycomb.

We feel this discussion would not be complete without further reference to the mechanical locomotive stoker, as it is only through mechanical firing that the full rating of modern locomotive boilers can be obtained. This being the case, would it not be well when designing such boilers, to give consideration to stoker requirements as to ready and appropriate application?

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[36] "Comparative Tests of Six Sizes of Illinois Coal in a Mikado Locomotive," by E. C. Schmidt, J. M. Snodgrass, and O. S. Beyer, Jr., University of Illinois Engineering Experiment Station, Bulletin No. 101, Sept., 1917.

[37] "A Study of the Locomotive Front End, Including Tests of Front-End Model," by E. G. Young, University of Illinois Engineering Experiment Station, Bulletin No. 256, May, 1933.

[38] "Mechanical Draft," by W. B. Snow, B. F. Sturtevant Company, Boston, Mass., 1910.

[39] Report of Committee on Locomotive Construction, Proceedings, American Association of Railroads, 1936, pp. 277–436.

We refer particularly to the type and design as developed by the Baltimore & Ohio Railroad, a type which may tend to revolutionize mechanical firing. This design and its prototype which, however, fires from the rear are the only mechanical stokers satisfying the requirements as to secondary air as discussed by the author under "secondary air for combustion."

No doubt this has escaped general notice, but an examination of either type will reveal this fact. While the method used is more in the nature of a heat exchanger, yet it does supply preheated "top air" and always in direct proportion to combustion requirements. Gas analyses verify this statement. The apparatus simply consists of a vertical grate surrounding that part of the stoker-delivery member extending upward through the locomotive grates.

These vertical grates, termed "protecting grates," extend above the fuel bed and absorb heat from it. This heat is in turn absorbed by the free air contacting the inner surfaces, and is in turn drawn into the firebox and over the fuel bed by the draft through $^1/_2$-in. holes drilled radially and spaced so as to give perfect diffusion. The number of drilled holes is always proportional to the total grate area and to the volume of air supplied through the fuel bed, thereby maintaining a fairly even balance. There can be no question but that this supplementary air so admitted contributes materially to the splendid results obtained on the Baltimore & Ohio as reflected in the boiler and fuel performance.

We have so little factual knowledge of what actually transpires in a locomotive firebox at moderate or high rates of combustion, that, in view of the importance of the subject, it would appear that almost any expenditure for tests would be warranted. Such tests, however, should be made on a plant where road conditions could be simulated. While tests[38] conducted by the University of Illinois indicated no material difference as between constant-flow and pulsating draft, many engineers feel that these are inconclusive, hence the suggestion to simulate road conditions. So unless tests are made, and such tests duplicate as far as possible actual road conditions, we will still find ourselves where we started.

To show to what extent the modern locomotive boiler depends upon the mechanical stoker, we might refer to the large simple "Mallets" built for the Northern Pacific some few years ago. These locomotives burn a subbituminous coal and therefore require a relatively large grate. The proportions as to grate area desired and steam output of the boiler were worked out by the engineering department of the Northern Pacific in collaboration with the locomotive builder. The latter, however, would not agree to conform to the dimensions requested (referring to grate area) until the stoker company agreed to build a stoker with sufficient capacity to meet maximum requirements, and would guarantee to distribute the amount of coal fired evenly over the grate. The maximum requirements were estimated at 45,000 lb of coal per hr. While this rate has never been required, nevertheless there are times when a rate of 30,000 lb of coal per hr is actually reached. The firebox first laid down was 20 ft long and $9^1/_2$ ft wide, or a grate of 190 sq ft. This was later reduced to about 184 sq ft. Needless to say, the requirements were met by the stoker company, otherwise the locomotives would never have been built.

L. A. Shipman.[40] In comparing steam-boiler performance on an efficiency basis, we have two vital factors to consider (1) water and (2) coal.

In the case of locomotive M1A tested by the Pennsylvania Railroad, referred to in the paper, wherein burning rates of 345 lb of coal per sq ft of grate area per hr were obtained, one of the main fuel problems, as indicated by the 50 per cent stack waste, was keeping the fuel on the grate until it was burned. This problem can be solved at least partially by fuel selection and sizing.

In connection with the matter of water circulation and foaming, the importance of the water conditions as they affect the efficiencies of the different tests should receive special attention.

While it has been mentioned that forced circulation of water was used in the Steamotive tests, the fact that practically distilled water was used in both Steamotive tests and in the Navy test, as against the regular water-tank water as used in the New York Central and Pennsylvania Railroad tests, was not mentioned. While this fact certainly affects the efficiencies of the test, it also singles out those who have developed the equipment and its methods of operation so that distilled water is economically possible for special distinction.

Because of the fact that there was a difference in the water, in the water circulation, natural against forced, and in the fact that there was a difference in the percentages of the total fuel consumed because of the high stack loss of the conventional locomotive at high burning rates, the performances are not on an equivalent basis for comparison although the facts presented definitely point out the tremendous improvements possible in fuel economies for which those contributing to the new designs should be highly complimented.

L. K. Sillcox.[41] The locomotive boiler and its operation can scarcely be discussed without referring to the fact that only a limited number of test data are available. The author is fortunate in having access to new information which heretofore has not been reported. Nevertheless he still is compelled to call attention to the need for vastly more information which can only be derived through further boiler research.

No entirely satisfactory method of defining locomotive capacity has yet been derived which will establish its capacity as limited by the features of furnace, boiler, engines, and rail, acting individually. Boiler calculations are the least exact of all the elements involved, although the investigations of Cole, Fry, Strahl, and others have all contributed in important measure to our total knowledge of the subject.

The author presents a table which shows the failure of Strahl and Cole to estimate the actual evaporative capacity of three locomotive types. In the case of the Pennsylvania Railroad class M1A, Strahl estimates capacity at 54 per cent of test values, Cole at 68 per cent. Perhaps, in justice to Strahl and Cole, it should be pointed out that, when evaporating 99,095 lb of water per hr, the locomotive was fired at the rate of 337 lb of coal per hr per sq ft of grate. No such overload is contemplated in the application of the formulas. At this high coal rate, an efficiency of boiler and furnace of but 37.5 per cent was obtained. Yet the absorption efficiency of the boiler was over 80 per cent. This is illustrative of the mass of operating details which must be explained in every case if test conclusions are to bear all their possible significance.

It has been remarked that, even with the stated excessive and uneconomical firing rate, absorption efficiency did not suffer greatly. This would indicate that high gas velocities may not always be objectionable, in so far as boiler performance is concerned. In fact, whereas a gas velocity of 204 mph in the tubes is quoted as very high, velocities of 445 mph are sought in the Velox steam generator, and blowers are introduced to attain this end. There is a vast difference in function and design, of course, but the Velox generator, one of which is now installed in a loco-

[40] Combustion Engineer, Southern Coal & Coke Company, Inc., Knoxville, Tenn. Mem. A.S.M.E.

[41] Vice-President, The New York Air Brake Company, Watertown, N. Y. Mem. A.S.M.E.

motive of the Paris, Lyons and Mediterranée Railway in France, occupies much less space and represents much less weight than a conventional locomotive boiler of the same capacity. While we are continuing improvements in the conventional firetube locomotive boiler, which no doubt will remain the standard type for general service for many years to come, we cannot afford to confine our attention to this construction to the extent that we are blind to possibilities originating in other fields. Concentration of more boiler capacity within less space and providing this with considerably reduced weight, offers attractive possibilities for investigation by locomotive builders, with available fuel types ever in mind.

It is the limitations imposed by compliance with familiar and proved proportioning of locomotive parts and strict adherence to conventional practices that renders difficult the fitting of required power into the loading restrictions and clearance limits of American railways and calls for a firing rate such as that proposed by the A.A.R. committee appointed to investigate the requirements in a locomotive capable of hauling 1000-ton trains at 100 mph, namely, 152 lb of coal per sq ft of grate per hr under normal conditions. Combustion efficiency would be higher if the unit firing rate were lower, yet weight limitations govern the permissible size of grate which may be practicably employed; approximately 128 sq ft in this case.

The coal-fired steam-locomotive boiler has advanced rapidly and far. It compares phenomenally with central-station boilers in capacity per unit of weight. Present railway-operating practices have taxed its capacity in all practicable sizes yet the demands are unchecked. Perhaps we must search out new approaches to an old problem, replacing well-thumbed pages of past operating manuals and removing the inherent difficulties which have retarded progress in the use of such promising expedients as pulverized coal and the water-tube boiler.

L. W. Wallace.[42] At the outset of this paper, a statement is made to the effect that the public is continually demanding higher speeds for both freight and passenger trains. It may be that such an apparent demand is being taken too literally. Is it higher speed, in terms of miles per hour, that the public wants or is it a reduction in the total elapsed time required to travel between two points? There is a real and significant difference between speed in miles per hour and total elapsed time. The total elapsed time experienced on many runs may be reduced materially without increasing significantly the sustained speed in miles per hour for any major portion of the run. This may be done by reducing the waiting time at stations and elsewhere through more expeditious servicing and handling of trains. It is the waiting period at terminals and elsewhere which causes the largest amount of restlessness and complaint on the part of the traveling public. It would appear that this phase of train operations has not been given the analytical attention which its importance warrants.

The difference between high speeds in miles per hour and a reduction in total elapsed time between two points, realized through more expeditious train servicing and handling, may make a major difference in locomotive design, its first cost, and the cost of maintenance and operation. What this may mean is reflected by the comparison of the 2-8-0 and 6-8-6 locomotive types. This comparison discloses that the weight of the locomotive had to be doubled in order to obtain sufficient boiler capacity. All of which raises the question, "What price is paid for high speeds in miles per hour as contrasted with the price which may be paid for a reduction in total elapsed time?"

The comparison of the locomotives referred to adds emphasis to this significant statement made by the author, ". . . the important question is whether greater boiler efficiency is attainable, particularly at high-capacity operation, and what can be done to accomplish this."

The following pertinent comment also appears in the paper: "The subject matter will be centered on problems affecting the design of the conventional locomotive fire-tube boiler only, not because of any belief that this type of boiler is the final answer to the locomotive steam generator. . . ."

The writer was delighted that the author disavowed any belief that the typical locomotive boiler is the final answer. In my judgment it is not. It may well be that if a broadly conceived and executed line of creative research were undertaken, the final result would be some form of boiler closely akin to the flash type. It seems evident that something different from the present typical locomotive boiler must come, as it is not sound economics to double the weight of a locomotive in order to obtain adequate steam capacity.

Again to quote: "However, much remains to be done to make possible the burning of more fuel per hour at higher combustion efficiency with lower draft loss, greater heat-absorption efficiency, and higher superheat with less boiler weight per pound of steam produced." This is a farseeing, correct, and timely statement of the problem confronting the railroad industry. It is fairly obvious that one approach to relieving the present situation is to supply the steam required for auxiliaries, train heating, air conditioning, and for other purposes through a steam-generating unit other than the locomotive boiler. However, this alone will not give the final answer.

The author states: "The method of standing blowdown tests developed and used by the New York Central Railroad in recent years has proved very effective." This is undoubtedly true. Moreover, it is the writer's thought that considerable valuable work in certain directions can be accomplished by the use of scale models. This means the application of the law of similitude which has been so successfully used in connection with research programs covering a wide range of difficult technical problems. There is no apparent reason why the method cannot be applied successfully to the solution of many railroad problems. The method has the advantage of affording a more definite control of test conditions than for full-scale operations and, furthermore, a material reduction in cost of experimental work may be realized.

The locomotive boiler presents a difficult problem. A comprehensive approach to its solution has been too long delayed. The solution lies in the direction of broadly conceived lines of research coupled with a daring and courageous attitude.

Author's Closure

It is an honor for the author to have the privilege of replying to the many valuable contributions made to his paper, "The Locomotive Boiler."

The interest that this subject has aroused among those responsible for the design and operation of steam locomotives testifies to the importance of further improvements to the coal-burning steam locomotive.

Since the preparation of this paper the European war has started, reminding us that the development of the steam locomotive must not be neglected, and it is pertinent that this subject be again emphasized.

It is often heard today that the railroads, and particularly the coal-burning steam locomotive, are out of date, and will be replaced by airplanes and oil-driven power units on railroads and highways.

Such opinion, if shared by responsible men in government and business, may have a paralyzing effect upon efforts to improve the steam locomotive. Those who are able to see beyond the

[42] Director, Division of Engineering and Research, Crane Company, Chicago, Ill. Mem. A.S.M.E.

present phase in the development of transportation, and are devoting efforts and thought to the perfection of the coal-burning steam locomotive should be encouraged in their efforts.

The struggle now raging in Europe is being fought on land, air, and water with engines of war driven by gasoline and oil power, and one of the major problems of this war is the control of adequate supply of petroleum.

That the world's supply of oil is definitely limited is believed to be true by many eminent authorities. A late estimate of the available oil supply in continental United States, prepared by the Committee on Petroleum Reserve, of the American Petroleum Institute, gives the total available supply as 17,348,146,000 barrels in 1939. The annual production in 1938 was 1,213,186,000 barrels per year which would indicate that our petroleum supply in the United States may approach exhaustion in fourteen years.

That this definitely limited supply of petroleum, absolutely necessary for operation of naval vessels, airplanes, tanks, and trucks, will have to be conserved for our national defense in the future appears reasonable.

The use of oil for the production of power, in places where coal, abundantly available, can be utilized to good advantage, appears unwise from the viewpoint of future national welfare.

The coal-burning steam locomotive has hauled the bulk of the country's traffic in the past. Its gradual replacement now taking place, at a rapidly accelerating pace, with oil-using Diesel locomotives and gasoline power units on highways and in the air, may find this country only a few years hence in a position of having a transportation machine that has cost billions of dollars, but without suitable, or too costly fuel to drive it.

A review of the many discussions reveals a general agreement with the author's thesis, that there is room for improvement and necessity for coordination and clarification of principles with further research and tests.

The opportunity to present a paper on the same occasion as Mr. Bailey is an honor that is very much appreciated by the author. Mr. Bailey's paper on steam-boiler performance[16] is of great fundamental value, and the discussion that he presented on the subject of locomotive boilers is an expression of his theory on "methods of comparison of boiler performance." His discussion, read in conjunction with his own paper just referred to, is so very complete that there is little that can be added for further clarification.

The author in his paper referred to a new type of locomotive boiler soon to be tried out. The boiler referred to is that illustrated and analyzed by Mr. Bailey in his discussion. The results obtained with this boiler, as to efficiency, capacity, and weight, may point the way to further developments on coal-burning steam locomotives.

The author regrets his inability to supply more detailed information on the oil-fired tests of the Southern Pacific, as requested, as these data are not obtainable.

Mr. Bailey suggests that the author supply additional data on the subject of draft loss through various portions of locomotive boilers under different ratings so that the relationship between output and efficiency versus draft loss could be analyzed. It is regretted that sufficient data are not available at the present time to furnish this information.

Mr. Duncan's discussion reveals his interest and insight into the problems under consideration, and deserves a more detailed answer than space will permit at this time. He suggests that if an underfeed or traveling stoker could be adapted to locomotive service a great improvement would result in furnace efficiency. It may be noted here that several designs of both of these types of stokers have been made.

The Crawford underfeed mechanical stoker was built and tested out quite extensively on the Pennsylvania Railroad in 1913 and 1914, and many engines were so equipped. It is believed that there are no underfeed stokers on locomotives now in service. The author has no knowledge of any traveling-grate stokers actually tested, although at least one design was made.

During some thirty years of experiments with different types of stokers the sprinkler type, due to its simplicity and reliability, is the only one that has survived the test of service.

The reason that underfeed and traveling-grate stokers have not been perfected and adapted for locomotive service is due to the fact that they require more room, and weigh more than the existing locomotive limitations permit.

Mr. Eksergian's discussion has amplified and supported the author in several important observations made and it would be superfluous to add anything further to the points touched upon by him, except to express appreciation for his carefully thought out comments.

Any contribution made by Mr. Jones, engineer of test and chief of the locomotive test plant, Pennsylvania Railroad, where most of the tests quoted by the author were made, is of real interest to all. He is eminently qualified to discuss this subject, and his reference to the M1A locomotive with enlarged steam space, emphasizing the improvement possible even with a small change in design, is valued highly.

While on this subject it should be noted that most of the test-plant tests are run with particular attention to keeping the water level below a full glass, or three gages of water, and also without foaming condition of the water. In road service it is known that many times the water level is carried above the highest gage cock and badly foaming water is encountered.

In some cases, on large and long boilers, it has been found that the distance from the water level to the top of the boiler barrel at the dome is as small as ten inches with the water level at the top gage cock. When ascending a heavy grade, say 2 per cent, with engine working at maximum capacity, the water level is usually carried to a full glass of water. When engine is tipping over the grade the water level near the dome will rise as much as 8 in., thus reducing the steam space to nearly zero.

This condition indicates the importance of the use of steam separators in the dome.

The carefully conducted tests of locomotive performance that are frequently being made by Mr. Knight on his railroad are an example of the alertness with which present-day railroad management explores every avenue to improve railway efficiency. His comments based on actual experiences are, therefore, of much interest.

The information pertaining to fuel tests with different grades of coal, furnished by Mr. Knight and quoted elsewhere in this summary, is very valuable. His reference to tests made with different percentages of air openings through the grates is most interesting, particularly the trial with grates having 60 per cent free air opening. Further information on these tests will be of interest.

Mr. Lipetz' contribution is of particular value because of his thorough knowledge of the fundamentals involved. He calls attention to the possibilities obtainable from the proper study of the curves of the test data presented in conjunction with the boiler ratios given in the table.

The data shown represent the results of all the pertinent and complete tests made. Many road tests have been analyzed by the author, but most of them usually lack sufficient test data to permit an accurate analysis of the performance of the locomotive under different conditions of load and speed.

Some road tests analyzed have accurate data on the cylinder performance, drawbar pull, and horsepower, but lack data on the fuel and water consumption or general boiler performance. Other tests have been made with accurate data on coal and water consumption, but without dynamometer or indicator cards to measure the work performed.

The reason for the many incomplete tests run in the past is due to the desire to get a quick picture of the general performance rather than a critical analysis of design; also, the question of the cost of tests is important, particularly when one railroad must bear such cost alone.

The advantage of coordinated research is appreciated by all concerned, and perhaps some day this may be brought about.

Mr. McBride suggests that a prompt collection and analysis of all available data are urgently needed in connection with the problem of proportioning locomotives. The author agrees with this and it was the primary purpose of his paper to make available all the pertinent and accurate test data available just for such analysis and study.

However, the author disagrees with Mr. McBride as to the accuracy of methods suggested by him as they use arbitrarily determined coefficients which apply to empirical formulas, the fundamental sufficiency of which is in question.

We need not be concerned here with a rule-of-thumb method for determining the approximate maximum evaporation of a boiler for the purpose of setting the size of the stoker, injector, etc., but rather a critical analysis of the effect that definite changes in the proportions of various component parts of the boiler have upon boiler efficiency, capacity, first cost, and cost of maintenance.

When this is determined, accurate formulas for calculating the capacity and efficiency of the boiler will follow.

The several questions raised by Mr. Meister, regarding the influence of superheater design upon locomotive economy, are important, and the author will endeavor to answer them.

The first question pertains to the practical limit of steam temperatures on present type of reciprocating steam locomotive; also, its effect on economy and maintenance.

Modern locomotives in the United States, Canada, and Europe operate with steam temperatures of 750 F, and on many locomotives total steam temperature of 800 F, at maximum rate, is not uncommon. With 250 lb boiler pressure this total temperature is equal to a superheat of 350 to 400 deg.

This steam temperature will give a metal temperature of the superheater units of from 875 to 900 F, and units made of open-hearth, low-carbon steel have proved satisfactory for these temperatures. Heat-resisting low-alloy steels are also available to give additional life if desired.

The foregoing statement is predicated on the basis that the units are kept clean from scale on the inside, as it is well known that even a thin layer of scale will decrease the heat transfer from metal to steam and raise the metal temperature.

As far as other parts of the locomotive in contact with superheated steam are concerned the problem of high steam temperatures has been successfully solved by the use of cast steel for the superheater header, throttle, steam pipes, and cylinders. The single-seated multiple-disk valve throttle has proved successful for this service. The lubrication problem has also been satisfactorily solved by the use of the modern high-temperature lubricating oils and forced-feed lubricators.

The question of economy gained by the use of high superheat is answered on page 391 of the author's paper. A reduction of as much as 46 per cent in steam consumption was gained with superheat up to 225 F as compared with saturated-steam operation.

It has been definitely shown on many tests that for each 10 F increase in superheat there would be a reduction in steam consumption of not less than 1 per cent up to the limit of the present practice of 800 F total temperature. This is due to the raising of the total heat content of the steam at admission, and increasing the total heat drop, or the Rankine-cycle efficiency.

The second question raised is a rather broad one and is perhaps best answered by the replies to the other more detailed questions in Mr. Meister's discussion.

The third question, inquiring as to what combination of pressures and superheat will give the best ultimate economy, can best be answered by stating that from a thermal standpoint the higher the pressure and superheat become, the greater will be the Rankine-cycle efficiency of the prime mover. From a practical standpoint it appears that 300 lb gage is the highest to which the conventional stayed locomotive boiler should be built considering material and practice of today. An increase above this pressure would not be of any advantage from an economy standpoint when used in a single-expansion reciprocating noncondensing engine with dimensions of present-day locomotives equipped with piston valves and Walschaerts or similar type of valve gears. With water-tube boilers of types now on the horizon it appears entirely practical to raise the boiler pressure considerably above 300 lb per sq in. and maintain up to 800 F steam temperature.

With the advent of a practical cylinder design having poppet valves and cam-operated valve gear, like the Franklin gear, referred to by Mr. Jones in his remarks at this meeting, it appears that a reciprocating engine has arrived that is not only well suited for handling high pressure and superheat from a practical point of view, but also capable of utilizing the increased amount of energy supplied with each pound of steam with greater efficiency.

The superheater on the 4-4-4-4 type locomotive, referred to in question 4 of Mr. Meister's discussion, is a large-flue superheater with the same size of superheater flue and arrangement as the type "A." The only difference is that each unit consists of two single loops joined to one set of header connections instead of one double-loop unit as with the type "A."

The heating surface of the flue, and superheater, and gas area in each flue, is the same as the type "A," but the steam area through each unit is twice as large. This results in a reduction of the steam velocity in the unit to one half of that of the double loop for any given rating. The pressure drop through the superheater is reduced thereby, but because the lower steam velocity reduces the rate of heat transfer the superheat will also be less.

In the early days of the development of the locomotive fire-tube superheater in the United States, some thirty years ago, a number of locomotives were equipped with single-loop superheaters having two single loops of $1^1/_2$ in. pipe in $5^1/_2$ in. flue.

In view of the desire to keep pressure drop down to a minimum interest in single-loop superheaters has been renewed.

The problem to be solved with the single-loop superheater is the relatively greater differential expansion of the pipes—this is receiving careful study.

On the basis of an increase of 350 deg in the steam temperature from the inlet side of the superheater unit to the outlet side there would be a difference in the average temperature of the metal in the pipes of each unit loop of 175 F.

As the two ends of each unit loop are fixed the large difference in the expansion of the unit pipes sets up a heavy strain on the unit joints that may cause them to leak and finally break.

The Superheater Company has experimented and tested many different types of single-loop superheater units, most of them based upon the principle of placing the bulk of the superheating surface of the unit so that the steam flows in counterflow with the gases, and only one pipe is used for the return of the steam to the front end in uniflow with the gases.

Example of such units are the "H" type, 5P, 6P, and 7P units. All of these types of units are subjected to the same differential temperature stresses. In the "HA" type unit this has been compensated by the use of a short section of double-loop pipe which acts like an expansion joint thus taking care of the differential expansion.

In the standard double-loop superheaters, both type "A" and type "E," there is sufficient flexibility to distribute and absorb this differential expansion without setting up undue stresses that cause trouble.

In view of the desirability of improving the tractive power of steam locomotives at high speeds the question of low pressure drop is important, and careful attention is given to the design of superheater, dry pipes, throttles, steam pipes, and cylinders to gain this end.

The question of reducing the pressure drop through the dry pipe, superheater header, throttle, steam pipe, and valves presents no difficulty from the standpoint of design as these steam passages can be increased considerably above present practice without interfering with the fundamental design of the locomotive, or reducing the useful life of these parts.

An undue increase in the steam area through the superheater units, however, may under some conditions adversely affect the power output of the locomotive as well as the life of the units.

To reduce pressure drop to a minimum and at the same time maintain high superheat for maximum economy in the use of steam involves not only the steam passages from the boiler through the superheater to the steam chest, the valves, and exhaust passages but also the correct proportions of the cylinders to the weight on drivers.

Experience with superheater engines during the last thirty years has shown that a factor of adhesion of 4 is about the minimum for satisfactory locomotive operation with superheaters of normal pressure drop, and it is only under difficult rail conditions that such locomotives have been found to slip at high speeds.

If locomotives with factor of adhesion near the lower limit have the superheater changed to single loop, cutting the pressure drop in half, and thereby raising the mean effective pressure at high speed, such engines might prove very slippery at high speeds when using ordinary valve gears.

Under such conditions the engine must be throttled, the cylinder diameters reduced, or the initial boiler pressure cut, and this will cancel the improvement that was originally sought to be gained by the lower pressure drop at high output, and one is back where he started.

For locomotives that are undercylindered, and have a factor of adhesion larger than 4, a considerable improvement may be gained in decreased pressure drop through the use of the single-loop superheater.

Engines designed fifteen or more years ago were usually equipped with superheaters smaller than is present practice.

Many of these older locomotives have been rebuilt and equipped with more efficient and powerful front ends, stokers, better grates, and valve gears. This has permitted such modernized engines to be forced to much higher rates of evaporation than originally possible and, logically, the superheater should be enlarged correspondingly both as to surface and steam area.

The greatest gain in increased boiler efficiency, higher superheat and low pressure will be obtained by installing type "E" superheaters. Whenever this is not expedient a good improvement can be obtained by the installation of a superheater unit like the "HA," which gives some 50 F higher superheat, without any increase in the pressure drop.

Another solution may be that of using a single-loop type "A" superheater. This would give less superheat, but only one third of the pressure drop of the double-loop type "A" with the same number of units.

In his fifth question Mr. Meister observes that the majority of modern locomotives have been designed with type "E" superheaters, and inquires regarding the improvements that will result from the use of this design as compared with the older type "A" superheater, or other types of superheater and boiler designs.

The type "E" superheater is a fire-tube superheater with a header in the front end of the same general design as the type "A." The difference lies in the arrangement, also the size of the superheater flues and units. The superheater flues in the type "E" superheater may vary in size from $3^1/_4$ in. OD to 4 in. OD depending upon the length of the flues. The size of the flue is selected so as to give the maximum efficiency with minimum draft loss.

Each unit is a double loop with one loop of superheater pipe in each of two superheater flues. The unit pipes are of such size as to give the highest superheat with the lowest pressure drop and draft loss. There are a number of open small tubes that do not contain superheater units.

Due to the use of the smaller size of superheater flues in the type "E" superheater as compared with the larger flues used in the type "A" it is possible to gain from 8 to 10 per cent in the total flue evaporating heating surface of the boiler for the same length of flues; this in addition to a gain of from 3 to 6 per cent in the total free gas area through the boiler.

The heating surface of the type "E" superheater is increased from 35 to 50 per cent over the type "A" superheater thus giving from 60 to 80 degrees higher superheat at maximum capacity for equal conditions of operation, and an even higher difference at the lower rates of working because of the greater surface and the more efficient disposition of same.

The steam area through the superheater is usually greater than the type "A," and is ample for normal low pressure drop.

The foregoing physical improvements in the amount and effectiveness of the heating surfaces result in a decided improvement in boiler efficiency and superheat with a resultant improvement in the over-all efficiency and capacity of the locomotive of upward of 15 per cent as compared with an old-style large-flue superheater.

With a properly proportioned grate and firebox the type "E" superheater-equipped locomotive boiler represents the most efficient concentration of boiler power that has so far been proposed for use in the conventional boiler and has proved practical in every respect.

In addition to the improvements in efficiency and capacity resulting from the use of the type "E" design there are several other practical advantages, such as a reduction in the number of unit header connecting joints to keep tight, and easier locomotive to draft because of the more even distribution of the same size of flues over the tube sheets, a larger space between the flues at the back tube sheet providing more water space and better circulation, which is of considerable importance in flue and tube-sheet maintenance.

The relative small-diameter flues permit the use of thinner flues as compared with the large flue superheater. This, together with the uniform size and thickness of the flues, equalizes the expansion strains reducing flue leakage and flue-sheet cracking.

Regarding Mr. Meister's sixth question, whether better results would be obtained with the stoker located in the front of the firebox. This question is very completely answered by Mr. Roesch's discussion on this paper; and the question of burning pulverized coal is touched upon in another place.

Mr. Meister's seventh question is of great interest to railway management. Many studies have been made on this subject, but it is apparent that sufficient experience has not yet been had to permit an accurate analysis, as suggested by Mr. Meister, from which definite conclusions could be made. This subject is of such large scope that it alone would require a separate treatise.

Mr. Ostermann's discussion on the high economy that results from the use of high superheat is valuable, and brings out the great importance that correct principles of superheater design have upon the satisfactory operation of locomotives.

The record of locomotive construction throughout the world, during the last thirty years, amply sustains the conclusion that the fire-tube superheater has proved the only practical design for the conventional locomotive boiler so far devised to obtain high superheat. Many other types have been tried, but without success.

This record, however, although gratifying, has not stopped the research work of The Superheater Company as they for many years have been engaged in a systematic and complete program of superheater design.

Every practical form of superheater unit so far conceived has been tested by The Superheater Company to determine the form of unit that would give the maximum superheat and highest evaporation, combined with minimum draft loss, minimum pressure drop through the superheater, low weight and cost, and, at the same time, be easy to maintain.

The author's replies to several of the questions raised by Mr. Meister in his discussion will answer some of the points raised by Mr. Ostermann.

Mr. Pownall adds a valuable contribution. Based on years of experience in this field he has confirmed the importance of testing, also modification of designs to meet specific operating conditions. His work in the development and improvement of the locomotive front-end construction is well known.

Mr. Pownall suggests that the cost of boiler repairs, stated by the author to be about 30 per cent of the total cost of repairs, should not exceed 15 per cent.

The 30 per cent figure was quoted from the report of the Federal Co-Ordinator of Transportation on comparative cost of steam-locomotive repairs, dated November 7, 1935, page 82, and being of great interest is quoted below:

Boiler Repair Costs

In response to the question: "Do you have data separated so as to show what proportion of the total costs of repairs is properly chargeable to locomotive boilers? If the answer is 'yes' state the percentage," eight roads stated that they kept the data separated and showed it varied from 19.2 to 37.4 per cent of the total cost of locomotive repairs; one hundred and ten roads did not have the separation, and ninety seven made estimates that ran from one to sixty-five per cent. A summarization of their estimates showed:

Number of roads	Estimated proportion of total cost of locomotive repairs due to boilers, per cent
17	25
15	20
13	30
8	15
6	35
5	40
5	29
3	10
3	19
3	60
2	18
2	27
2	31
2	65
1	1
1	6
1	12
1	13.5
1	16
1	22
1	26
1	33
1	44
1	49
1	62
97	

Of the roads making an estimate, 68 of them, aggregating 70 per cent of their estimates, ranged from 20 to 40 per cent, so it is probably fair to assume that a mean between these extremes, or 30 per cent, would be a fair general approximation. An effort was made to develop to what extent the treatment of water had affected the cost of locomotive repairs. However, since there is such a limited knowledge with respect to what the boiler repairs cost, it was found impracticable to get much of value in this connection.

From the foregoing it will be noted that only five class 1 railroads out of 165 reporting had a boiler cost below 15 per cent of the total cost.

Mr. Roesch has submitted an interesting discussion on the subject of the great importance to locomotive-boiler operation of the problems and phenomenon of burning fuel in firebox. In the past this subject has been given much thought and study, and there is no question but that further research is necessary, also, that available test data should be published and interpreted.

There is one point in Mr. Roesch's discussion upon which further thought may be given. He states, "Needless to say, the general adoption of restricted grates was hailed by locomotive-stoker manufacturers as it fully supported their contention that stack loss was not a question of coal size as fired but rather one of air openings through the grates and draft influence."

The author feels that this statement cannot be accepted as it stands without further explanations and qualifications as it is contrary to his own experience as to the effect that the size or per cent of fineness of coal has upon stack loss or unburned fuel loss.

The author has personally conducted many fuel tests which showed that the greater the percentage of fineness in the coal as fired, the greater the stack loss, based on exactly identical locomotives, heating value of fuel, and method of firing. Many tests conducted, some within recent years, have proved this to be so.

With the permission of W. G. Knight, mechanical superintendent, Bangor and Aroostook Railroad Company, the following abstract is quoted from the test report of a recent test most carefully conducted on this road.

"In 1939 a thorough investigation was made on the Bangor and Aroostook in connection with noted increase in coal consumption.

"Every cargo of coal that this Company buys is analyzed, and a careful study made indicated that the chemical composition of the coal was good, and it was, therefore, necessary to look elsewhere for the cause.

"Stack losses were known to be excessive and tests with dynamometer cars were made under the following conditions:

"Two carloads of coal of approximately 80 tons were screened and the amount of fines removed from the coal would be everything below the $^1/_2$" screen mesh. This coal was run through the stoker and rescreened again. From this test approximately 28 per cent fines would pass through the $^1/_2$ in. screen.

"Two engines, identically of the same construction, one being hand-fired, and the other stoker-fired, were used for the test.

"A coal car was used behind the locomotive to catch all cinders that could be obtained for the purpose of getting samples to analyze their heat value.

"The results showed among other things that by using screened coal containing 1.24 per cent slack the saving obtained when compared with the same coal having different percentage of slack was found to be as follows:

Slack, per cent	Fuel saving lb per M.G.T.M. in per cent
12.0	8
23.8	12.2
28.0	16.0

The conditions of the locomotive and test were identical in all comparisons.

"It was further found from these experiments that the difference from the coal consumption of run-of-mine coal having 25 per cent to 30 per cent fines as compared with the 15 per cent fines amounted to approximately 10 per cent.

"Results from operation extending over a year on coal having 15 per cent or less slack have even shown on average operation slightly better than 10 per cent saving."

The foregoing quoted test results on the Bangor and Aroostook are confirmed by recently conducted fuel tests on the New York Central Railroad.

These tests were run for the specific purpose of determining the qualities of different grades of fuel, under identical conditions, of a modern 4-6-4 type locomotive, on standing blowdown tests.

Permission has been granted to quote the following conclusions relating to the subject of the effect of fineness and slack in fuel.

"Preliminary results of tests on fuel from six different mines in which the percentage of slack in the coal varied from straight egg coal with no slack (what is commonly known as 100 per cent egg) to 40 per cent egg with 60 per cent screenings or 60 per cent slack.

"The test showed that where the slack content has been increased the coal rate was correspondingly increased for identical rates of evaporation for practically all the fuel fired.

"Coals of practically the same chemical composition and some slight difference in the B.t.u. value will result in an inferior boiler performance as the use and percentage of screenings is increased.

"The use of increased percentage of screenings, resulting in a lower boiler evaporative capacity, will affect the locomotive performance. This decrease in the boiler performance may be of sufficient magnitude to reduce the locomotive performance required for heavy trains on fast schedules 10 per cent."

From the foregoing it would appear that this question is of great importance and is worthy of the very careful study that it is now receiving at the hands of the various railway administrations.

Mr. Sillcox's comments are of great value because of the prominent part that he has taken in the development of the steam locomotive. His reference to a new approach to the use of pulverized coal and water-tube boilers is particularly timely.

Many studies have been made of this, and different types of water-tube boilers have been built in this country and in Europe. So far only a few locomotive tube boilers have operated successfully. These have been confined to good feedwater territories.

For high availability, that is, long periods of continuous service that we strive for today, it is necessary, no matter what the construction of the water-tube boiler may be, to use only fully treated water; or in conjunction with condensing operation of the engine when pure scale-free boiler feedwater will be assured.

Mr. Sillcox's reference to the use of pulverized fuel is also important. Based on various tests with burning pulverized fuel it appears practical to design a locomotive firebox for the conventional standard boiler, or water-tube boiler, in such a manner as to satisfactorily burn pulverized coal at rates that will satisfy and exceed any demands so far placed on the modern locomotive.

To do so it will necessitate pulverizing the coal very finely as this will increase the speed of the combustion process by a more rapid contact between the molecules of carbon and oxygen.

At maximum rate it should appear possible to obtain a combustion efficiency of 95 instead of 60 per cent, or less, with the present standard method of firing. This increase in the combustion efficiency, in addition to the other advantages that will accrue from burning of pulverized fuel, must pay for the total cost of pulverizing the coal.

The discussion by Mr. Wallace is welcomed indeed in view of the intimate knowledge which he has had of research problems affecting locomotives and railway operation. His comments on the necessity of reducing the delay at terminals are pertinent as every minute saved on stops and slow orders will decrease the elapsed time between terminals.

Mr. Wallace again stresses the necessity of a coordinated program of test and research for the further improvement of the steam locomotive, and that the development of a more efficient and lighter boiler must go hand in hand with the development of the prime mover as the steam consumption of the engine must be reduced substantially below the present practice.

Whether this new prime mover will take the form of a reciprocating or rotating steam engine, or turbine, can only be answered by future developments, but, in any case, it is apparent that condensing operation is a very desirable thing not only to reduce the specific steam consumption of the prime mover, but also to furnish pure condensate for the boiler feedwater.

The most recent practical development now in sight is the perfection of the Franklin improved cylinder arrangement with poppet-type steam and exhaust valves driven with a cam valve gear.

Such an arrangement is now being successfully used in the United States, and has shown a remarkable improvement in power development at high speeds.

Design Factors Controlling the Dynamic Performance of Instruments

By C. S. DRAPER,[1] CAMBRIDGE, MASS., and G. P. BENTLEY,[2] WALTHAM, MASS.

This paper presents a generalized treatment of instruments which have a single movable index, controlled by the magnitude of a single actuating quantity. Such an instrument acts as a system with one degree of freedom and is studied by the well-known methods developed for such systems. Instruments which include servomechanisms cannot be treated as systems with one degree of freedom and are therefore excluded from consideration. From the standpoint of static indications, it is shown that the essential properties of an instrument can be specified in terms of range, scale sensitivity, calibration error, environmental error, and uncertainty. The definition and use of each of these quantities is discussed.

It is shown that the response of an instrument to rapid changes in the actuating quantity can be calculated if two parameters called instrument characteristics are known. One of these parameters has the dimension of time and is called the characteristic time. The second parameter is nondimensional and is called the damping ratio. Nondimensional curves, showing instrument responses to three typical variations in the actuating quantity, are plotted for a series of values of the damping ratio. These curves are used in an illustrative example to show how the values of the instrument characteristics, required to keep dynamic errors below a given limit, can be determined in a practical case.

INSTRUMENTS must be designed to meet many types of service. Although individual instances will often present problems which call for particular construction details or special materials, certain aspects of instrumentation can be generalized to include the requirements of many practical cases. Such generalizations are useful not only in writing specifications for equipment but also in the analysis and discussion of instrument problems. The system which is outlined herewith has been developed for teaching and design purposes at the Massachusetts Institute of Technology. Emphasis is placed on the points of similarity between instrument problems rather than on the differences which appear when a comprehensive scheme of terminology is applied. For a classification of instruments from the standpoint of their use in industry the reader is referred to the works of Behar (1),[3] and Smith and Fairchild (2, 3).

Reduced to the simplest terms, an instrument is so often an arrangement in which the position of a pointer or index is controlled by some physical quantity, that instruments of other types may be considered as special cases. In the operating range of an instrument, it is usually intended that a continuous motion of the index will follow a continuous change in the actuating quantity which controls the instrument. A device in which this result has been achieved by design falls into the category of systems with 1 deg of freedom and will be called an "elementary instrument." Unless otherwise specified, the term "instrument" will always refer to an elementary instrument. One of the authors and G. V. Schliestett (4) have discussed in generalized form the mathematical analysis of elementary instruments. The analysis divides naturally into two cases, depending upon the way in which the actuating quantity changes. If the actuating quantity is constant, the index will become stationary and the instrument subjected to "static conditions." On the other hand, if the actuating quantity is varying so rapidly that the index does not follow the changes exactly, the instrument is operating under "dynamic conditions." Practical problems deal with the properties required in an instrument to meet certain static requirements and to perform satisfactorily under given dynamic conditions. The present paper suggests a method of specifying the essential features of an instrument in terms of characteristics which can be conveniently applied either for design purposes or for determining the suitability of an instrument for use in a given situation.

Static Characteristics

The essential external features of an elementary instrument are shown in Fig. 1. From the standpoint of performance, the

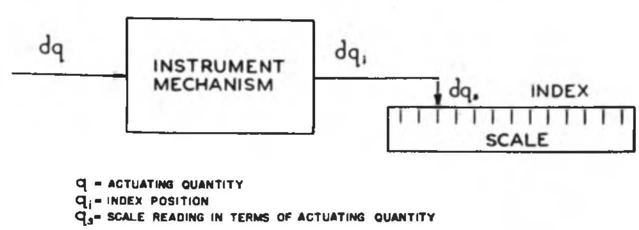

Fig. 1 Essential Features of an Elementary Instrument

mechanical details inside the case of an instrument are unimportant. It is sufficient to know that the instrument contains a mechanism of some sort which produces a change in an index when a change occurs in the actuating quantity. If the instrument belongs to the general class called "meters," a scale is attached to the structure which directly associates each position of the index with some measure of the actuating quantity. Other instruments may not include a scale and so would be disqualified as meters. However, there is always the possibility of interpreting index positions in terms of the actuating quantity by means of tables or calibration curves. For purposes of discussion, any means capable of performing the function of identifying some state of the actuating quantity with each position of the index will be called a "scale."

The general nature of an instrument mechanism is determined when the actuating quantity is given. The requirements of static performance for the mechanism and the scale may be specified in terms of the range and the scale sensitivity. Range is the difference between the highest and lowest magnitudes of the actuating quantity which the instrument is to measure.

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[3] Numbers in parentheses refer to the Bibliography.

Contributed by the Machine Design Committee of the Machine Shop Practice Division and presented at the Annual Meeting Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

Scale sensitivity is the ratio of a small change in scale reading to the corresponding small change in index position, i.e.

$$\text{Scale sensitivity} = S_s = dq_s/dq_i \ldots\ldots\ldots\ldots [1]$$

In general S_s may be different for different parts of the scale or it may be constant throughout the range. A vapor-pressure thermometer, using a pressure gage of uniform sensitivity as the indicating system, will have a temperature scale of varying scale sensitivity.

For static conditions, an instrument with a scale perfectly suited to the mechanism, with no friction or hysteresis in the mechanism and not affected by changes in physical quantities other than the actuating quantity, would be completely specified by the range and scale sensitivity. Such an instrument would be perfectly accurate in the sense that the scale reading would always be equal to the actual value of the actuating quantity. This condition may not be fulfilled but, in any case, a small change in the scale reading may be related to a small change in the actuating quantity by the equation

$$dq_s = Cdq \ldots\ldots\ldots\ldots [2]$$

where C is the calibration coefficient. Obviously the condition for an ideal direct-reading instrument is for the calibration coefficient to be unity. Keeping in mind that C may vary over the range in practical cases, the scale reading corresponding to a given magnitude of q can be found by integrating Equation [2] to obtain

$$q_s = \overline{C}q + K \ldots\ldots\ldots\ldots [3]$$

where K is the constant of integration and $\overline{C}$ is a proper average value of the calibration coefficient. If the zero adjustment of the instrument is set so that the scale reading is correct ($q_s = q$) at some part of the scale, the constant of integration can be eliminated from Equation [3]. The actual instrument performance can then be expressed by means of a plot of

$$\overline{C} = q_s/q \ldots\ldots\ldots\ldots [4]$$

as a function of the scale reading.

In practice the variations of the calibration coefficient over the range of an instrument are so small that a plot of static corrections is more convenient for studying performance than a plot of the calibration coefficient itself. If the static correction is defined as the difference between the actual magnitude of the actuating quantity and the scale reading, Equation [3] gives

$$\text{Static correction} = q_{\text{corr}} = q - q_s = q_s\left(\frac{1-\overline{C}}{\overline{C}}\right) \ldots\ldots [5]$$

or, expressed in terms of the fractional static correction

$$\text{Fractional static correction} = q_{\text{corr}}/q_s = (1-\overline{C})/\overline{C} \ldots\ldots [6]$$

The fractional static correction expressed as a percentage is a measure of the inaccuracy of the scale reading in any particular case, but it is inconvenient for generalized discussions, since it varies with both the correction and the scale reading. For specification purposes, it is better to express the correction as a percentage of the highest reading on the scale. In this form the correction will be called the full-scale fractional static correction and will be given the symbol Δq_s, where

$$\Delta q_s = q_{\text{corr}}/Q_s \ldots\ldots\ldots\ldots [7]$$

and Q_s is the difference between highest and lowest scale readings or total scale range.

Analysis of Static Corrections

Many instrument mechanisms are affected to some extent by changes in physical quantities other than the actuating quantity. These physical quantities together make up the environment of an instrument. The process of marking off the scale to read directly in terms of the actuating quantity is called calibration and must be carried out under some particular state of the environment which establishes the calibration conditions.

Static corrections can be broken down on the basis of the source responsible for each component. These components and a corresponding symbol in terms of full-scale fractional static correction are as follows:

1 Calibration error (Δq_c) is due to improper fitting of the scale to the instrument under calibration conditons.

2 Environmental error (Δq_e) is due to variations of the environment from calibration conditions.

3 Uncertainty (Δq_u) is due to imperfections in the instrument mechanism such as rubbing friction between solid parts, hysteresis in elastic members, etc.

Calibration errors often appear when instrument mechanisms are attached to independently graduated scales without special adjustments. The slow changes due to wear and aging effects may also produce calibration errors. Such errors may be found by experiments carried out under calibration conditions. Environmental error can sometimes be calculated but can always be determined by experiment. For any particular reading of a given instrument, the calibration error and the environmental error can be combined into a correction which removes both these errors when applied to the reading. In addition to the method of correction, environmental errors can be eliminated by designing the instrument mechanism to be unaffected by changes in the environment. An instrument constructed in this fashion is said to be compensated.

Uncertainties differ from the errors just considered in that they cannot be exactly determined for a particular scale reading and applied as a correction. Uncertainty appears when a scale reading is made from a pointer of finite width. Rubbing friction between solid parts in the mechanism is a second typical cause of uncertainty. Due to this friction, some finite change in the actuating quantity must occur before a detectable motion of the index appears. From this standpoint, uncertainty, as the word is used here, is similar to ultimate sensitivity as this term is accepted by Behar (1).

In general, it is only possible to set an upper limit for the uncertainty associated with a particular scale reading. Smith (3) has applied the term "dead zone" to a similar concept. According to Smith: "The dead zone is the extreme range of values of the variable possible without moving the indicator from a given position, in the absence of abnormal vibration; the dead zone is ordinarily twice the frictional error." Actually the uncertainty in any particular scale reading may be anything from zero to a maximum value determined by friction or the pointer width. The exact value of the uncertainty in a given instance will depend upon the previous history of the scale reading, condition of the instrument mechanism, vibration, etc. Although it might be possible to estimate the uncertainty in particular cases, the usual procedure is to accept the maximum uncertainty as a fundamental limitation on the accuracy of an instrument.

Dynamic Errors

Rapid changes in the actuating quantity are accompanied by differences between instantaneous scale readings and the true values of the actuating quantity which occur at the same time. For any particular case, the static correction may be responsible for part of such differences but there will, in general, be another component peculiarly due to the fact that the actuating quantity is changing. This latter component will be called the "dynamic error." Fig. 2 graphically illustrates the concepts of dynamic error and the corresponding time lag for a case with negligible static corrections. From Fig. 2, dynamic error is the difference between q and q_s at a given instant, while time lag is the time interval between the instant the actuating quantity passes through a certain value and the instant at which the scale reading reaches this value.

Many authors have studied the problem of dynamic errors. In the field of industrial instruments, Behar (1), and Smith and Fairchild (3) have summarized the methods of attack and have given typical results. The work of Draper and Schliestett (4), in reducing the analysis of dynamic errors to generalized nondimensional forms suited for engineering use has already been mentioned. The discussion which follows deals with a scheme for analyzing the dynamic errors of an elementary instrument

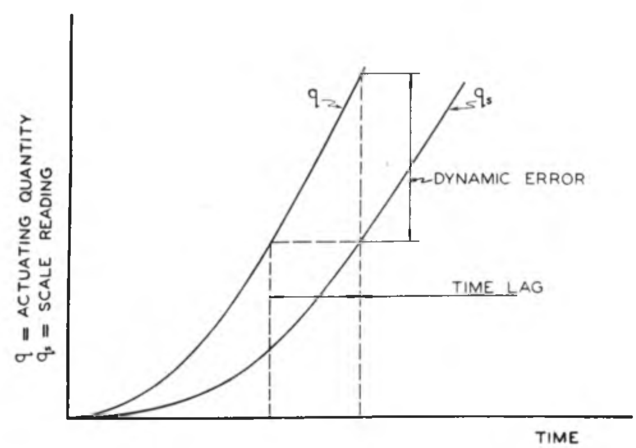

Fig. 2 Dynamic Error and Time Lag

in terms of two characteristic parameters. One of these parameters has the dimensions of time and will be called the "characteristic time." The other parameter is dimensionless and may be either the damping ratio or the characteristic damping number. In certain cases, the undamped natural frequency may conveniently be used as the time parameter. Methods for determining these parameters in actual cases have been described (4). The treatment which is outlined in this reference will be reviewed briefly in the present paper and extended to show how the characteristic parameters may be applied to engineering problems.

For any given instance, the dynamic error will vary with time in a manner dependent upon the characteristics of the instrument used and upon the way in which the actuating quantity changes. This means that each set of circumstances will require special treatment for an exact determination of dynamic errors. The amount of work necessary can be reduced by approximating actual variations in the actuating quantity by means of changes falling into three basic types. In the first of these types, the actuating quantity changes so rapidly between two constant values that the change is completed before the instrument reading has changed appreciably. A change of this sort is called a "step function." The second type occurs when the actuating quantity varies directly with time and will be called a "linear change." The third type is a continuous sinusoidal variation which may actually be one component of a more complicated periodic change. It will be shown how the instrument characteristics can be simply applied to predicting the dynamic errors which occur with each type of variation in the actuating quantity.

Generalized Equation of Motion

Motion of an instrument index is determined by the balance between effects of four types which, for purposes of discussion, will be considered as generalized forces, as follows:

1 Restoring forces which tend to move the index toward some equilibrium position with a force magnitude proportional to the divergence of the index from that position.

2 Damping forces which resist motion of the index with a resultant magnitude proportional to the index velocity. This assumption yields useful results for many cases in which it represents an approximation.

3 Inertia forces which are proportional to acceleration of the index.

4 Forces which depend upon the magnitude of the actuating quantity. These forces are responsible for initiating changes in the index position, and for this reason, the mathematical expression which specifies the actuating quantity as a function of time will be called the "forcing function."

It has been shown (4), that the first three of these forces may be expressed in terms of the scale reading to give the equation of motion in the form

$$(a_2)d^2q_s/dt^2 + (a_1)dq_s/dt + (a_0)q_s = a_0q \ldots\ldots\ldots\ldots [8]$$

This equation can be solved by well-known methods (5) for the case in which the coefficients a_0, a_1, and a_2 are constants. When these coefficients vary either with time or with the scale reading, a solution is usually so difficult to handle that the assumption of constant coefficients is made even for cases where it is not true. The resulting solutions will not be exactly correct but will often give a picture sufficiently close to physical facts for engineering purposes. These solutions will show the way in which the scale reading varies with time for a particular instrument under given conditions. The actual value of the actuating quantity at any instant can be found from the forcing function so the dynamic error is completely determined as the difference $(q - q_s)$ or on a fractional basis as $(1 - q_s/q)$.

Two types of problems involving dynamic errors appear in practice. In the first type the coefficients associated with a given instrument are known and it is desired to find the dynamic errors existing when a particular forcing function is applied. Cases of this nature are simple in that it is only necessary to carry through straightforward solutions of Equation [8]. Problems of the second type start with requirements on the magnitude of dynamic errors accompanying certain forcing functions and lead to values of the coefficients (instrument design constants) a_0, a_1, and a_2 as the desired results. The object of generalizing the analysis of dynamic errors is to reduce the amount of effort required for solution of either type of dynamic-error problem. The method of attack is to divide Equation [8] through by a_2, so that the coefficient of the first-derivative term has the dimensions of reciprocal time, and the coefficient of the term in q_s has the dimensions of reciprocal time squared. The coefficient of the first-derivative term can be used to establish a characteristic time for an instrument. It will be shown that one additional parameter in the form of a dimensionless ratio will completely specify a given instrument. The dimensions of the actuating quantity in any particular case are immaterial since time is the only dimension involved in the characteristics. The characteristics completely define the equation of motion so it follows that all instruments with the same characteristics will have the same dynamic errors under identical forcing functions and initial conditions. Because of this fact, it is possible to plot families of nondimensional curves for the dynamic response of instruments to the three forcing-function types which have been chosen as representative practical cases. With these curves available, it is possible to solve a wide range of dynamic-error problems by a simple inspection of the curves. The mathematical background necessary for proper use of the instrument characteristics is outlined and the use of nondimensional performance curves is illustrated by examples.

Solution of Equation of Motion in Terms of Instrument Characteristics

Equation [8] has solutions which give the variation of the scale reading q_s with time. These solutions have the general form

$$q_s = q_{s_{(tr)}} + q_{s_{(ss)}} \qquad [9]$$

The first term on the right-hand side is called the "transient" and is the solution of Equation [8] with the forcing function taken as zero. The second term is any solution of the complete equation for a given forcing function and is called the "steady-state" solution. Physically, the steady-state solution gives the behavior of the instrument after a particular forcing function has continued indefinitely. The transient represents the period of accommodation required for the instrument to change from one steady-state condition to another.

Mathematical details required to solve Equation [8] are discussed in any course on the differential calculus but the solutions thus obtained are usually left in a form which is unnecessarily clumsy for engineering use. The treatment which follows is intended to reduce the work required to obtain quantitative results in dynamic-error problems. It is found that the transient component of the solution may be either oscillatory or nonoscillatory, depending upon the relative magnitudes of the coefficients in Equation [8]. From the standpoint of analysis, a knowledge of the quantities chosen for characteristics should make it possible to determine the dynamic behavior of an instrument directly from generalized nondimensional curves for either oscillatory or nonoscillatory transients. Reasons for the choice of characteristic time and the damping ratio, as these parameters are defined, will appear as the analysis is developed.

Instrument Transients

The nature of the characteristics can best be explained from a study of the transient component in the solutions of Equation [8]. If the forcing function is placed equal to zero and each term is divided by a_2, Equation [8] becomes

$$d^2q_s/dt^2 + (a_1/a_2)dq_s/dt + (a_0/a_2)q_s = 0 \qquad [10]$$

The exponential function, the form of which is unchanged by differentiation, is particularly suited for the solution of Equation [10]. Two arbitrary constants must appear in the solution, since the original equation is of the second order. It follows that the solution may be written in the form

$$q_s = Ae^{-t/\tau_1} + Be^{-t/\tau_2} \qquad [11]$$

The symbols τ_1 and τ_2 must represent quantities with the dimension of time, since the exponents must be without dimensions. Substitution of the expression given by Equation [11] into Equation [10] shows that τ_1 and τ_2 must be expressible in terms of the coefficients a_0, a_1, and a_2 by the relationships

$$1/\tau_1 = (a_1/2a_2) - \sqrt{(a_1/2a_2)^2 - (a_0/a_2)} \qquad [12]$$

$$1/\tau_2 = (a_1/2a_2) + \sqrt{(a_1/2a_2)^2 - (a_0/a_2)} \qquad [13]$$

The first term on the right-hand side of these equations is always real, but the quantity under the radical may be either real or imaginary, depending upon the magnitude of $(a_1/2a_2)^2$ with respect to the magnitude of (a_0/a_2). If the second of these quantities is greater than the first, the radical terms become imaginary and the corresponding motion for the transient part of the expression for q_s is oscillatory. In this case Equation [11] is usually changed from the exponential form to a trigonometric form which is more convenient for practical use. If $(a_1/2a_2)^2$ is greater than (a_0/a_2), the radical term is real and Equation [11] can be used as written. A third case exists which represents the transition condition between oscillatory and nonoscillatory transients.

For maximum usefulness the quantities chosen as instrument characteristics should be easily applied to any case whether the transient is oscillatory or nonoscillatory. The undamped natural period of an instrument is suitable for use when the transient is oscillatory or the forcing function is sinusoidal, but is difficult to interpret directly when the transient is nonoscillatory or forcing functions are not sinusoidal. On the other hand, the quantity $(2a_2/a_1)$ has the dimension of time and is so intimately associated both with the damping of transients and steady-state dynamic errors that it is particularly suitable for use as one of the instrument characteristics. For this reason the characteristic time will be defined as

$$\tau = (2a_2/a_1) \qquad [14]$$

To specify completely the behavior of an instrument under dynamic conditions, another parameter is required in addition to τ. This parameter is the second instrument characteristic and can always be chosen as a dimensionless ratio. The ratio between τ_1 and τ_2, as these quantities are defined by Equations [12] and [13], is the most obvious choice for a dimensionless instrument characteristic. This ratio is especially useful for nonoscillatory transients and will be called the "characteristic damping number."

$$\nu = \tau_1/\tau_2 (\tau_1 > \tau_2) \qquad [15]$$

When the transient becomes oscillatory, the characteristic damping number becomes a complex quantity which is difficult to interpret for practical purposes. A dimensionless ratio which can be conveniently applied in any case is the damping ratio. This ratio is defined as the ratio of the coefficient a_1 which actually exists to the coefficient a_{1c} which must exist for the condition which divides oscillatory from nonoscillatory transients. Mathematically this critically aperiodic condition is reached when

$$(a_1/2a_2)^2 = a_0/a_2 \qquad [16]$$

For this case

$$\tau_1 = \tau_2 = \tau \qquad [17]$$

and

$$\text{Damping ratio} = \zeta = a_1/a_{1c} = \nu = \tau_1/\tau_2 = 1 \qquad [18]$$

It follows that when ζ is less than unity, the transient will be oscillatory, while the transient will be nonoscillatory for values of ζ greater than unity. The characteristic damping number will be unity when the transient is critically aperiodic, greater than unity for nonoscillatory transients, and complex for oscillatory transients.

In order to solve dynamic-error problems, certain definitions, in addition to those already discussed, are convenient. The "circular natural frequency" is defined as

$$\text{Circular natural frequency} = \omega = \sqrt{a_0/a_2 - (a_1/2a_2)^2} \qquad [19]$$

$$\omega = 2\pi n = 2\pi/T \qquad [20]$$

In Equation [20] n is the natural frequency and T is the natural period. When the coefficient a_1 is zero, the quantities thus

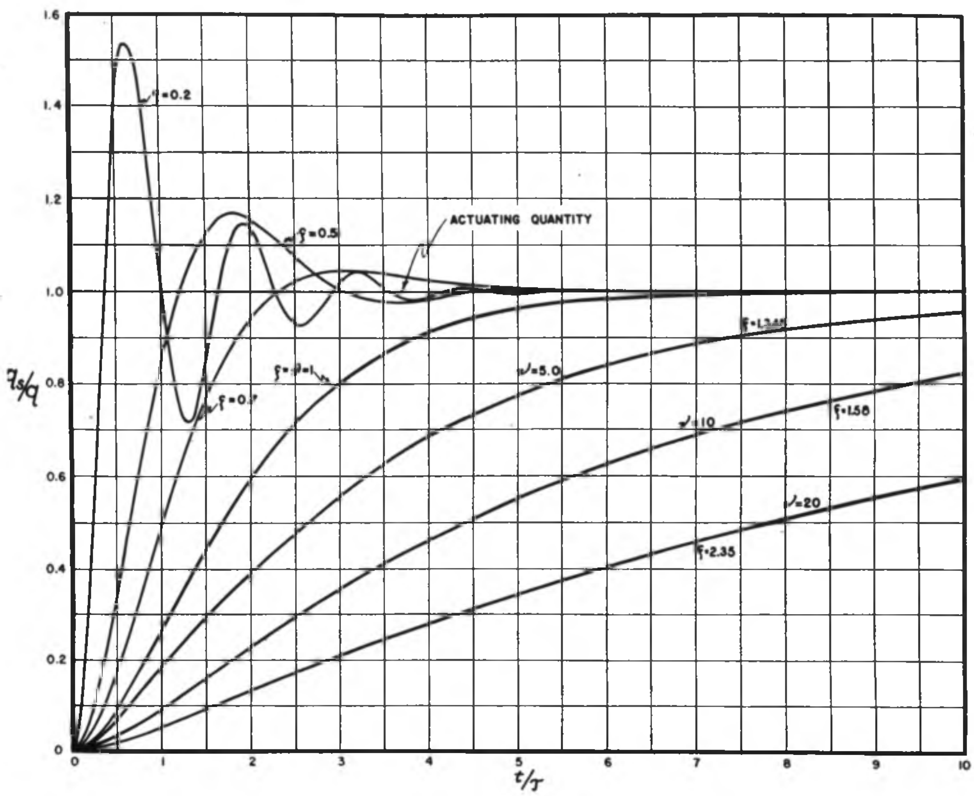

FIG. 3 INSTRUMENT RESPONSE TO A STEP FUNCTION

defined become the undamped natural circular frequency, the undamped natural frequency, and the undamped natural period, respectively. The undamped condition is indicated by a subscript n on each of the quantities so that

$$\omega_n = \sqrt{a_0/a_2} = 2\pi n_n = 2\pi/T_n \quad [21]$$

and

$$\omega = \omega_n \sqrt{1-\zeta^2} \quad [22]$$

A convenient way of remembering the relationships between the instrument characteristics and the other essential quantities involved in the solution of dynamic-error problems is to write the original differential equation of motion in terms of the derived quantities. The result of this procedure is as follows

$$d^2q_s/dt^2 + (a_1/a_2)dq_s/dt + (a_0/a_2)q_s = (a_0/a_2)q \quad [23]$$

$$d^2q_s/dt^2 + 2\zeta\omega_n(dq_s/dt) + \omega_n^2 q_s = \omega_n^2 q \quad [24]$$

$$d^2q_s/dt^2 + (2/\tau)dq_s/dt + 1/(\zeta\tau)^2 q_s = 1/(\zeta\tau)^2 q \quad [25]$$

$$d^2q_s/dt^2 + (\nu+1)/\tau_1(dq_s/dt) + (\nu/\tau_1^2)q_s = (\nu/\tau_1^2)q \quad [26]$$

Written in terms of the instrument characteristics, the transient can be conveniently expressed in three forms, depending upon the value of ζ.

Case 1 Oscillatory transient, ζ is less than unity, ν is complex

$$q_s = e^{-t/\tau}\left\{A \sin\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau)\right] + B\cos\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau)\right]\right\} \quad [27]$$

or

$$q_s = Ae^{-t/\tau}\cos\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau) + \varphi\right] \quad [28]$$

Case 2 Critically aperiodic, $\zeta = \nu = 1$

$$q_s = Ae^{-t/\tau} + Bte^{-t/\tau} \quad [29]$$

Case 3 Nonoscillatory, $\zeta > 1$, $\nu > 1$

$$q_s = Ae^{-[2/(\nu+1)]t/\tau} + Be^{-[2\nu/(\nu+1)]t/\tau} \quad [30]$$

where

$$\zeta = (\nu+1)/2\sqrt{\nu} \quad [31]$$

In Equations [27], [28], [29], and [30], the symbols A, B, and φ represent "fitting constants" to be used for fitting the transient equations to combinations of particular steady-state solutions and given initial conditions.

DYNAMIC ERRORS PRODUCED BY A STEP FUNCTION

The step function will be treated as the first typical forcing function. For analytical purposes, a step function can be considered a case in which the scale reading is constant at zero under a zero magnitude of the actuating quantity, before the instant taken as the initial instant, followed by an instantaneous change of the forcing function to some constant value at the initial instant. Mathematically these circumstances can be expressed as

$$t = 0 \quad q = q_s = 0 \quad dq/dt = dq_s/dt = 0$$
$$t > 0 \quad q = q_a \quad dq/dt = 0$$

When the fitting constants of Equations [27] to [30] are adjusted so that the general transient equations meet the particular initial conditions of the step function, the results are

$\zeta < 1$

$$q_s/q_a = 1 - e^{-t/\tau}\left\{\left(\zeta/\sqrt{1-\zeta^2}\right)\sin\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau)\right] + \cos\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau)\right]\right\} \quad [32]$$

$$q_s/q_a = 1 - e^{-t/\tau}\cos\left[\left\{\sqrt{1-\zeta^2}/\zeta\right\}(t/\tau) - \sin^{-1}\zeta\right] \quad [33]$$

$\zeta = \nu = 1$

$$q_s/q_a = 1 - e^{-t/\tau}(1 + t/\tau) \qquad [34]$$

$\zeta > 1, \ \nu > 1$

$$q_s/q_a = 1 - [\nu/(\nu - 1)]e^{-[2/(\nu+1)](t/\tau)} + [1/(\nu - 1)]e^{-[2\nu/(\nu+1)](t/\tau)} \qquad [35]$$

The general nature of these transients is so well known that it is unnecessary to include a detailed mathematical discussion. From an engineering standpoint, the important point is that a family of curves can be plotted with t/τ as the independent variable and q_s/q_a as the dependent variable, each curve of the family being based on a constant value of ζ. Fig. 3 shows such a family of curves for values of ζ between 0.2 and 2.4. These curves can be applied to instruments of any type which obey the original differential equation. A useful generalization which is apparent from an inspection of Fig. 3 is that the transient is substantially over in a time interval equal to three times the characteristic time, if the damping ratio is in the range between 0.2 and 1. An example showing how the curves of Fig. 3 may be applied to practical engineering problems will be discussed later.

Dynamic Errors Due to a Linear Change of the Actuating Quantity

In many practical cases, the change in the actuating quantity can be approximated as a linear increase or decrease with time. Mathematically this is equivalent to using an expression

$$q = Mt \qquad [36]$$

as the forcing function. M is the constant which can be adjusted to make Equation [36] fit the physical facts as closely as possible. The complete solution for q_s, as a function of time, depends not only upon the instrument and the forcing function but also requires that the initial conditions be specified. Out of an infinite number of possible choices, the case in which the reading is constant at zero, when the linear forcing function is applied, is most useful for generalized discussions of instrument performance. The case in which the initial reading is not zero but is constant at some other value may obviously be handled by a simple substitution of $q - q_0$ and $q_s - q_{s0}$ as variables to replace q and q_s. The equations resulting from use of the forcing function $q = Mt$ and $q_s = 0$, $dq_s/dt = 0$ when $t = 0$ are

$\zeta < 1$

$$q_s/M\tau = t/\tau - 2\zeta^2 + e^{-t/\tau}\left\{2\zeta^2 \cos\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right)(t/\tau) + \frac{\zeta(2\zeta^2 - 1)}{\sqrt{1-\zeta^2}} \sin\left(\frac{\sqrt{1-\zeta^2}}{\zeta}\right)(t/\tau)\right\} \qquad [37]$$

$\zeta = \nu = 1$

$$q_s/M\tau = t/\tau - 2 + e^{-t/\tau}(2 + t/\tau) \qquad [38]$$

$\zeta > 1, \quad \nu > 1$

$$q_s/M\tau = t/\tau - (\nu + 1)^2/2\nu + [\nu(\nu + 1)/2(\nu - 1)]e^{-[2/(\nu+1)](t/\tau)} - \left[\frac{\nu + 1}{2\nu(\nu - 1)}\right]e^{-[2\nu/(\nu+1)](t/\tau)} \qquad [39]$$

These equations have been made nondimensional by using the amount that the actuating quantity changes in one characteristic time as the unit of scale reading. By this procedure, the equations are simplified and a single family of curves will take care of all possible cases, involving a linear forcing function acting on an instrument with a constant reading at the initial instant.

With τ used as the time unit, the time scale is not uniform as in the normal case when the damping is varied for a system which is otherwise unchanged. This effect is liable to be confusing unless it is properly interpreted. Therefore, the curves of Fig. 4 have been plotted for an instrument with constant undamped natural frequency and varying damping. By using the undamped natural period as the unit of time and the change of the actuating quantity occurring in a time interval equal to the undamped natural period, the distance associated with a given time interval or a given change in scale reading is the same for each one of the family of curves. This makes interpretation of the curves in terms of the physical facts somewhat simpler

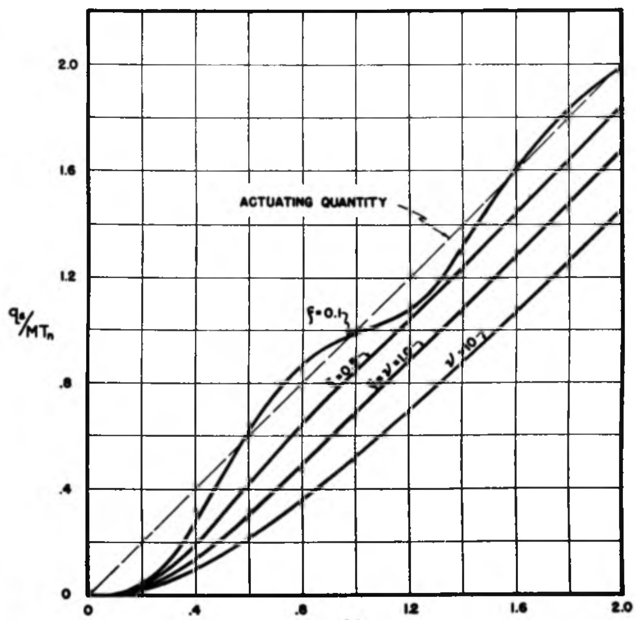

Fig. 4 Instrument Response to a Linear Forcing Function

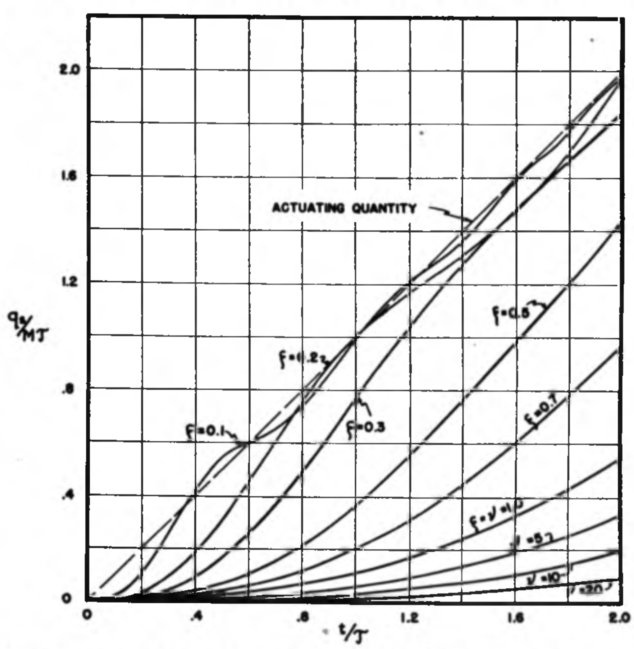

Fig. 5 Instrument Response to a Linear Forcing Function

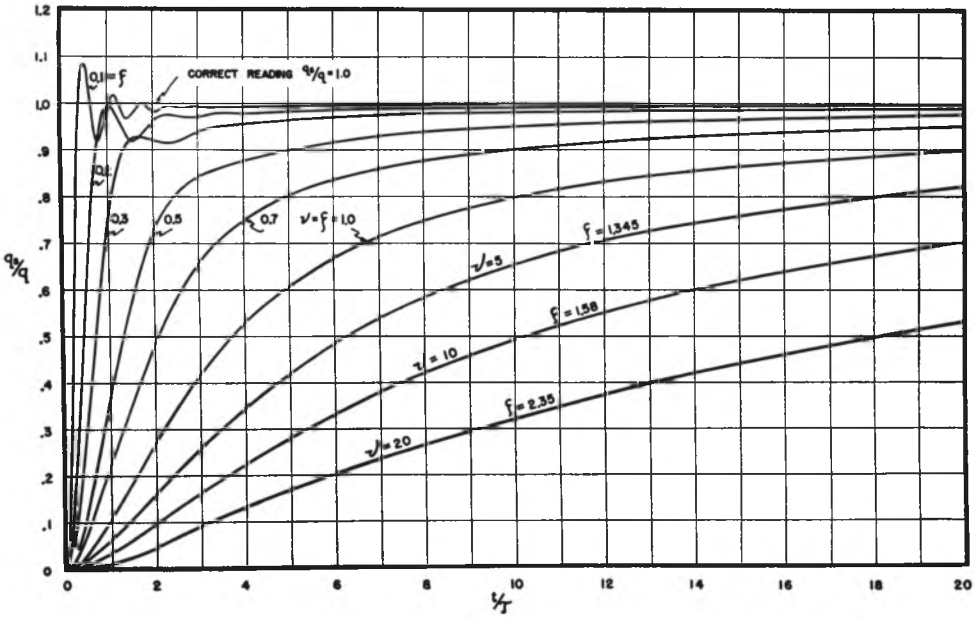

Fig. 6 Linear Forcing Function; Variation With Time of Actual Reading as Fraction of True Value of Actuating Quantity

than if Equations [37], [38], and [39] had been plotted directly. The curves of Fig. 4 show that, as the transient dies out, the instrument reading differs from the true instantaneous value of the actuating quantity by an amount which tends to a constant value of $2\zeta^2 M_T$ for oscillatory transients, $2M_T$ for the critically aperiodic case and $[(\nu + 1)^2/2\nu]M_T$ for the nonoscillatory case. This is an example of a steady-state dynamic error which is constant and corresponds to constant time lag in terms of T_n units. Fig. 5 shows the curves of Fig. 4 plotted directly from Equations [37], [38], and [39]. The effect of damping on the characteristic time is shown by the expanded abscissa scale as the damping ratio is increased.

Instrument engineering problems are usually concerned with the dynamic error rather than with the actual instrument reading at any instant. For generality, it is convenient to express the actual reading in terms of the correct reading at any instant. The correct reading will always be the instantaneous value of the actuating quantity and in terms of M_T units is equal to t/τ. It follows that

$$q_o/q = (q_o/M_T)[1/(t/\tau)] \qquad [40]$$

The instantaneous value of the dynamic error, expressed as a fraction of the true value of the actuating quantity, will be

$$\text{Dynamic error} = 1 - q_o/q \qquad [41]$$

Curves showing the fractional reading and the dynamic error as a function of t/τ for values of ζ from 0.1 to 2.35 are plotted in Fig. 6. The use of this family of curves will be illustrated later.

Steady-State Dynamic Errors Due to a Sinusoidal Variation of the Actuating Quantity

Many practical situations may be represented in terms of periodic changes in the actuating quantity. In cases where the disturbance exactly repeats itself, a Fourier series of sines or cosines will accurately describe the physical facts. Even if

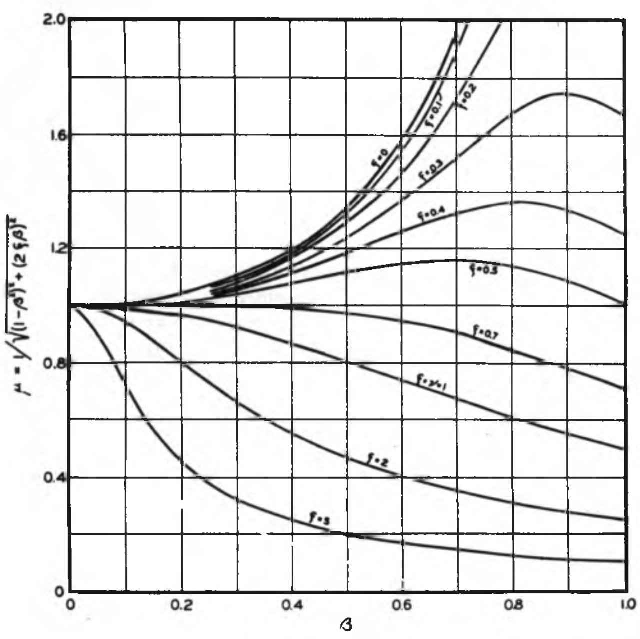

Fig. 7 Amplitude Ratio as Function of Frequency Ratio

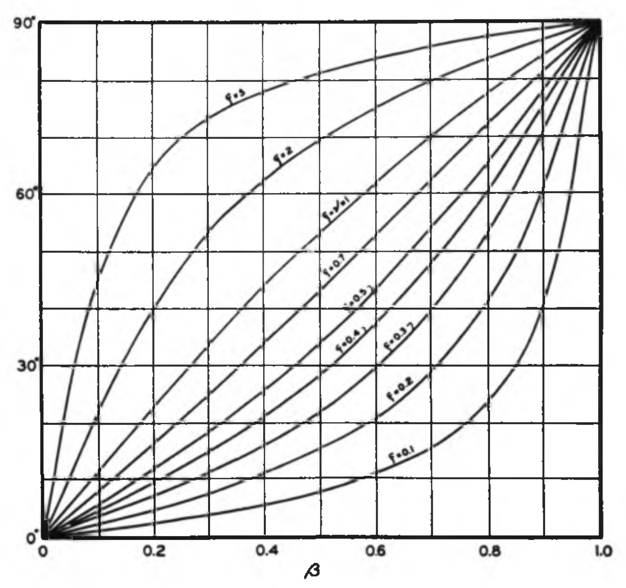

Fig. 8 Lag Angle as a Function of Frequency Ratio

this condition is not precisely fulfilled, satisfactory results can often be obtained by the use of a sinusoidal forcing function to find the corresponding steady-state solution of Equation [24]. Thus if the forcing function is

$$q = q_a \sin \omega_f t \qquad [42]$$

the steady-state solution as found by conventional methods (5) can be written as

$$q_s/q_a = \frac{1}{\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2}} \sin(\omega_f t - \varphi) \qquad [43]$$

In Equation [43]

$$\omega_f = 2\pi n_f = 2\pi/T_f = \text{circular forcing frequency} \qquad [44]$$

$$\beta = \omega_f/\omega_n = \text{frequency ratio} \qquad [45]$$

$$\varphi = \tan^{-1}(2\zeta\beta)/(1-\beta^2) = \text{lag angle} \qquad [46]$$

The expression

$$(q_s/q_a)_a = \mu = 1/\sqrt{(1-\beta^2)^2 + (2\zeta\beta)^2} = \text{amplitude ratio} \qquad [47]$$

is very useful in connection with instrument problems involving periodic changes. A family of curves of μ against β is given in Fig. 7. Practical instrument problems usually involve values of the frequency ratio less than unity so only this range has been covered in Fig. 7. A family of curves showing the lag angle as a function of β for the same range as that of Fig. 7 is given in Fig. 8.

General Design of an Instrument to Meet Given Specifications

The ideas previously discussed are particularly useful for translating performance specifications for an instrument into terms of mechanical details. In practice every instance will present problems requiring special treatment, but the general procedure can be illustrated by an example which might reasonably occur as an actual case. This example will deal with the preliminary design of a recording pressure indicator for slow-speed steam engines or air pumps. Specifications start with the requirement that the instrument must be able to operate for reasonable periods of time under the conditions of pressure, temperature, vibration, etc., which exist in service. Beyond this necessity of satisfactory mechanical operation, the desired properties of the indicator can be defined in terms of static specifications and dynamic specifications. The specifications assumed for the present example are listed as follows:

Static specifications:

- Range—20 lb per sq in. below atmospheric pressure to 160 lb per sq in. above atmospheric pressure
- Scale sensitivity—50 lb per sq in. per in. (constant over range)
- Calibration error—2 lb per sq in. (maximum)
- Environmental error—1 per cent of range per 10 C temperature change
- Uncertainty—1 per cent of range (maximum).

Dynamic specifications:

- Step function—Indicator must give reading within 10 per cent of actual pressure, 0.05 sec after sudden pressure change (q_s/q_a between 0.90 and 1.10; $t = 0.05$ sec)
- Linear change—Gage must follow pressure increasing linearly with time with dynamic error of less than 10 per cent at end of range, when pressure change is such that range would be covered in 0.07 sec (q_s/q_a between 0.90 and 1.10; $t = 0.07$ sec)
- Sinusoidal change—Gage must follow sinusoidal pressure changes with maximum frequency of 4 cycles per sec within an amplitude error of less than 10 per cent and a phase-angle lag of less than 10 deg (μ between 0.90 and 1.10; φ less than 10 deg).

Static specifications determine several of the indicator properties. The range of 180 lb per sq in. and scale sensitivity of 50 lb per sq in. per in. require a scale length of $180/50 = 3.6$ in. The pressure-sensitive element and the indicating system must operate over the range and have a spring-stiffness constant within $2/180 = 1.1$ per cent if the calibration-error specification is not to be exceeded.

The idea of using sensitivities associated with the component systems of an elementary instrument, as developed (4), is useful in laying out preliminary designs. In the present case of a pressure indicator, the converter can be taken as the system which receives pressure changes and produces changes in the position of some mechanical part which can be used to operate the indicating system. The corresponding sensitivities are

$$\text{Converter sensitivity} = S_c = \frac{\text{small change in position of converter output element}}{\text{small change in applied pressure}} \qquad [48]$$

$$\text{Indicating-system sensitivity} = S_i = \frac{\text{small change in index position}}{\text{small change in element connected to converter}} \qquad [49]$$

From (4) for a direct-reading instrument

$$S_c S_i S_s = 1 \qquad [50]$$

The converter of the indicator could be a spring-and-piston system, working in a cylinder, a metallic bellows, a diaphragm, a Bourdon tube, or any other means for converting pressure changes into mechanical displacements. If a spring-and-piston system is chosen as the converter with a converter sensitivity of 0.001 in. per lb per sq in., the necessary indicating-system sensitivity can be found by use of Equation [50], and the specified scale sensitivity as

$$S_i = 1/S_c S_s = 1/(0.001)(50) = 20 \text{ in. per in.} \qquad [51]$$

Uncertainty is the result of imperfections in the indicator mechanism and the width of the recorded line. In the present case, the recording point must produce a line so fine and the mechanical details be so well worked out that the maximum uncertainty is plus or minus 1.8 lb per sq in.

The difficulty of meeting the environmental-error specification can be reduced by establishing calibration conditions as near actual service conditions as possible. In case the indicator temperature varies widely in service, it might be necessary to introduce differential expansion elements to reduce the effect of temperature upon the instrument readings. A problem of this nature must usually be handled by special methods to suit given cases.

Dynamic specifications determine the instrument characteristics τ and ζ. Theoretically, there is an infinite number of characteristic combinations which will satisfy the dynamic specifications. Practical mechanical considerations will determine the particular combination to be selected. For an engine indicator, as in many simple instruments, no benefit is to be gained by using a damping ratio in excess of unity ($\zeta = 1$). On the other hand, values of the damping ratio less than 0.1 ($\zeta < 0.1$) will permit excessive natural-frequency components in the index motion.

With these facts in mind, it is reasonable to survey the range of possibilities by selecting damping-ratio values of 0.2, 0.5, 0.7, and 1 for trial purposes. The dynamic specifications will then place limiting values on the ordinates of Figs. 3, 6, 7, and 8, so that critical values of t/τ can be found for each value of the damping ratio. The spring stiffness of the mechanism has already been determined by the value of S_c. From the standpoint of design it is often desired to find the maximum permissible inertia or mass of the instrument parts. For a constant value of the spring stiffness, this means that the natural undamped period of the system will be a maximum. For a given damping ratio, the characteristic time is proportional to the undamped natural period, so the general problem can be resolved into finding the minimum value of t/τ for which the dynamic-error specification is not exceeded. For steady-state sinusoidal changes, a minimum value of the frequency ratio β corresponds to a minimum value of t/τ for the other cases.

Table 1 gives a summary of the numerical work required to solve the problem discussed in the preceding paragraphs. For low values of ζ where oscillation is noticeable, the minimum value of t/τ must be selected so that the specified ordinate limits are not exceeded for any greater value of t/τ. For example, in Fig. 3 the curve for $\zeta = 0.2$ crosses the 10 per cent error line several times, but the critical value is the last crossing at $t/\tau = 2.1$.

TABLE 1 SUMMARY OF NUMERICAL WORK USED TO SOLVE TYPICAL PROBLEM

(a) Step-function analysis (Fig. 3)
$t = 0.05 \quad q_s/q_a = 0.90 - 1.10$

ζ	$(t/\tau)_{\text{crit}}$	$\tau_{\text{crit}} = \dfrac{0.05}{t/\tau\ \text{crit}}$
0.2	2.1	0.0238
0.5	2.3	0.0217
0.7	1.8	0.0278
1.0	3.8	0.0131

(b) Linear-pressure-increase analysis (Fig. 6)
$t = 0.07 \quad q_s/q_a = 0.9 - 1.10$

ζ	$(t/\tau)_{\text{crit}}$	$\tau_{\text{crit}} = \dfrac{0.07}{t/\tau\ \text{crit}}$
0.2	0.7	0.1000
0.5	5.0	0.0140
0.7	10.0	0.0070
1.0	20.0	0.0035

(c) Frequency-response analysis (Figs. 7 and 8)
$\omega_f = 4 \times 2\pi \quad T_f = 0.25$ sec $\quad \mu = 0.90 - 1.10 \quad \varphi < 10$ deg

ζ	Frequency criterion, β_{crit}	Phase criterion, β_{crit}	Critical, T_n	τ_{crit}
0.2	0.32	0.375	0.0800	0.0638
0.5	0.47	0.175	0.0440	0.0140
0.7	0.72	0.125	0.0308	0.0070
1.0	0.33	0.082	0.0207	0.0033

TABLE 2 SUMMARY OF POSSIBLE INSTRUMENT DESIGN CONSTANTS

(Determined by minimum value of τ from a, b, c, Table 1)

ζ	$\tau_{\max}$	$T_{n_{\max}}$	$N_{N(\min)}$ (cycles per sec)	Critical specification
0.2	0.0238	0.0299	33.5	Step function (a)
0.5	0.0140	0.0440	22.7	Phase angle (c)
0.7	0.0070	0.0308	32.5	Linear change (b)
1.0	0.0033	0.0207	48.3	Phase angle (c)

Values of ζ and τ, which meet the dynamic specifications are listed in Table 1. In order to satisfy all the conditions, the shortest characteristic time required for any of the three forcing functions with a given value of ζ must be chosen. Table 2 is a summary of the critical conditions established in Table 1. The critical values of the characteristic time have been converted into values of undamped natural frequency and undamped natural period by the relation

$$T_n = 2\pi\zeta\tau = 1/n_n \qquad [52]$$

From Table 2, it is seen that, for low values of the damping ratio, the step function establishes the critical condition, since the low damping permits several natural-frequency oscillations before the reading will remain below the specified error limit. As the damping ratio is increased, the transient component of the step-function response becomes of reduced importance, while the steady-state time lag for the other two forcing functions is increased. In the present case, the steady-state time lag, during a linear change, and the phase-angle lag for a sinusoidal change introduce limitations which are almost identical.

In general as the steady-state-lag specifications are made more severe (specified lags are made smaller), the required undamped natural frequency is increased for a given value of ζ, or the damping ratio must be decreased for a given natural frequency. For the step function, the reverse is true, since the response to such a change is more accurate, as ζ is increased to about 0.7. It follows that the characteristics selected for an instrument must be a compromise between conflicting trends. In many cases, it is mechanically easier to construct an instrument with a low rather than a high undamped natural frequency. This will probably be true for a spring-and-piston engine indicator like that of the present discussion, so the results summarized in Table 2, which are based on the assumption of a critical lower limit to the undamped natural frequency, represent at least one reasonable engineering solution to the original problem. It follows that an indicator with a damping ratio of 0.5 and a characteristic time of 0.0140 sec will be satisfactory. The corresponding undamped natural frequency will be 23 cycles per sec.

After an analysis of dynamic performance has been carried out along the lines suggested in the foregoing discussion, it may be found very difficult to obtain the required characteristics in practice. For example, an engine indicator would probably not have a damping ratio of 0.5 without including some special damping system in the mechanism. In such a case, the original specifications must be relaxed or the expense of building a more than usually complicated instrument accepted. It is evident that the dynamic-error specifications, which can actually be fulfilled by a given indicator, may be found by using the dynamic-performance curves in a process which reverses the method described.

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Discussion

K. J. DeJuhasz.[4] This paper represents a noteworthy endeavor to coordinate the scattered information on the dynamic behavior of instruments into a coherent whole and make it more readily available and understandable for students and practicing engineers. In view of the magnitude of the task, it is no detraction from the value of this paper to state that this objective has been only partially attained. The following remarks are intended to complement the paper in some respects and to call attention to some aspects in which further elucidation would be desirable.

The title of the paper suggests a broader scope than is actually the case, as the treatment does not include discussion of seismographic-type instruments which depend for their function on a low natural period of the actuating member, such as seismographs, vibrographs, and torsiographs. A more detailed treatment of Coulomb friction and backlash also would be desirable, as these enter into the functioning of most engineering instruments.

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The writer questions the usefulness of the concept of "time lag" when dealing with transient changes. The "lag angle" in measuring a sinusoidally varying actuating quantity is a well-established and useful concept. The time lag of a transient change has no meaning in cases in which the scale reading cannot follow rapidly enough the increase and decrease of the actuating quantity, hence, some values of the actuating quantity are not indicated at all by the scale reading. Furthermore, it may happen that a scale reading is shown before the actuating quantity attains the corresponding value, i.e., the time lag is negative. This would be tantamount to saying that the instrument knew beforehand, and predicted what was going to occur in the future, which is of course absurd. Therefore the writer suggests that the concept of time lag be dropped altogether when dealing with transient changes, and the performance of the instrument be characterized solely by the "dynamic error," i.e., the difference between the actuating quantity and the scale reading at a given instant, which has a real meaning in all cases. Attention should be focused on the concept of "maximum dynamic error," i.e., the maximum difference between actuating quantity and scale reading.

It would be highly desirable to elucidate the various abstract concepts.

As examples, the writer suggests:

(*a*) The undamped natural period of an instrument is equal to the period of a simple pendulum having a mass equal to that of the instrument, and a length equal to its static deflection under its weight.

(*b*) The maximum dynamic error decreases with the decrease of the natural period.

(*c*) The critical damping is brought about by a damping force which is just sufficient to dissipate the potential energy of the instrument.

Possibly these definitions could be further elucidated or improved, in order to reduce to the minimum the mental effort necessary for their understanding.

The writer cannot call to mind any instrument in which the concept of "error after the elapse of a given time" would be of advantage, or could be stipulated in advance. In most cases the requirements are as follows:

(*a*) That the natural period and damping be as low as possible. This is the case with engine indicators, in which the "maximum error" is proportional to the natural period, and the superimposed waves can be allowed for by simple inspection, or more accurately, deducted from the curve by graphical differentiation. An uncertain amount of damping (or friction) is far more difficult to take into account.

(*b*) That the damping be the lowest causing the instrument to "dead-beat," i.e., critical damping. This is the case with quick-weighing scales, in which the overshooting of the pointer is prevented by a dashpot. In such cases it is farfetched to stipulate, and hardly feasible to design in advance, a definite damping ratio. Such dashpots are designed by a rough advance calculation, relying on the fortunate fact that the final adjustment can be readily made by means of a needle valve.

In view of the ever-increasing importance of instrumentation, and corresponding attention given to the theory and design of instruments, further investigations in this field are greatly to be desired, and the authors' paper is welcomed for stimulating thought in these problems.

E. S. Smith.[5] Figs. 7 and 8 and their context, dealing with "steady-state dynamic errors due to a sinusoidal variation of the actuating quantity," offhand appear to this writer to comprise the novel portions of the paper and as such should be useful to engineers working in this art. Nothing that appears anywhere in this discussion is to be taken as minimizing the value of the authors' contribution which unquestionably involved the expenditure of a considerable amount of effort with a corresponding reduction in the work which others in this special field must do.

The treatment is not in fact general, as stated in the summary, since it is not applicable to common servo-operated measuring and controlling instruments which move at constant speed from one position to the next or are intermittently actuated; instead, it seems to be limited to a special (lineal-law) sort of self-operated meter. The treatment does not fit even the ordinary flowmeter which responds to a differential pressure which increases with the square of the velocity of flow.

The terminology of the paper unfortunately differs radically from any of those in common use in this country. The American Society of Mechanical Engineers' Committee on Industrial Instruments and Regulators is endeavoring to develop an authoritative and generally acceptable terminology which should do much to lessen the effort required in digesting papers such as this one, in which the points of possible originality may be obscured by unusual expressions.

As to discussion in the present paper, of papers (2, 3) by Fairchild and the writer and the use of the term "dead zone" in paper (3), it is regretted that such discussion was not made as such, at the time the papers in question were presented, so that the usual closure could have covered it adequately. In view of such belated discussion, the following brief answer may be in order: The papers were presented with controlling (or regulating) as well as measuring in mind, and the treatment of measuring instruments was intended to be suitable for the metering portions of controlling (or regulating) instruments. They were presented to cover the basic theory involved in an introductory manner and with the use of the unavoidable minimum of mathematics; however, such papers are believed to be more than mere "classifications" and to cover much of the material in the present paper. In this paper, the use of numerous Greek letters and the presentation of lengthy mathematical restatements of generally known material are objectionable, in so far as they exceed a reasonable requirement of completeness. However, such repetition may be defensible as making such material more readily available to mechanical engineers.

The term "dead zone" was intended by Fairchild and the writer to cover the maximum disturbance of the measured variable which cannot affect the meter and, hence, alter the position of a controller, instead of referring to the error of a particular reading. Such a definition of this expression still appears to be substantially correct, with usual ratios of starting friction to running friction, although an attempt was made to attain clarity rather than to sponsor an unassailable definition.

The DeJuhasz treatment of the engine indicator in his book[6] on that subject appears to the writer to be essentially similar to that of the present authors on the linear change of the actuating quantity, although it includes a more limited treatment of the amplitude of the response with sinusoidal variations with, offhand, no treatment of lag in such case.

The authors might well test the usefulness of their two criteria on the design of a commercially needed, reliable, and accurate pressure indicator for hydraulic transmissions of the multiple-piston type using, e.g., a working pressure of about 1500 lb per sq in., a frequency of 12,000 cycles per min, and an unsymmetrical wave form (rounded tops and V-troughs).

[5] Patent Agent, C. J. Tagliabue Mfg. Co., Brooklyn, N. Y. Mem. A.S.M.E.

[6] "The Engine Indicator," by J. K. DeJuhasz, Instruments Publishing Company, Pittsburgh, Pa., 1934.

AUTHORS' CLOSURE

Professor DeJuhasz and Mr. Smith have based their major criticism upon their opinions that the principles discussed in the paper could not be applied to instrument problems of certain types and have advanced examples to prove their claim. The authors feel that this criticism is due to a lack of understanding of the manner in which generalized reasoning in terms of nondimensional variables can be applied to specific problems. When such a method is used, a large part of the mathematical work can be carried out once and for all to produce results in the form of curves which may be applied to any physical case described by the differential equation used as the basis for the analysis. To use such curves intelligently and to be able to handle cases which may fall outside the range of available plots, it is only necessary for an engineer to have a general understanding of the steps followed in order to obtain the final results. For this reason, the analysis has been presented in some detail at the risk of "lengthy mathematical restatements of generally known material." It was the authors' intention to offer the possibility of a saving in time and effort on the part of workers in the general field of instrumentation.

In considering an instrument problem, it is only necessary for the practicing engineer to satisfy himself that the behavior of a given instrument can be described by a second-order differential equation with constant coefficients. When this requirement is fulfilled, ability to use the curves of the present paper will provide an engineer with specific information directly applicable to many problems which could be duplicated from papers similar to references (2, 3), cited by Mr. Smith, only after many hours of detail work. Mr. Smith shows some recognition of this fact in his statement, ". . . such repetition may be defensible as making such material more readily available to mechanical engineers."

The particular objection of Professor DeJuhasz that the method of analysis described in the paper is not general, because it does not apply to the seismographic type of instruments for measuring vibration, is definitely unjustified. As a matter of fact, the method of analysis as presented was largely worked out during the design of a line of vibration-measuring equipment for aircraft which has been commercially available for some time. A detailed discussion of the manner in which the methods explained in the present paper can be applied to vibration-measuring instruments was given by C. S. Draper and Walter Wrigley at the 1940 Annual Meeting of the Institute of the Aeronautical Sciences.[7]

Mr. Smith states that the method of analysis discussed in the present paper does not apply to the case of an instrument such as a flowmeter which involves a square law between the actuating quantity and the reading of an indicator which responds to pressure changes. As a matter of fact, the authors anticipated this objection by defining the sensitivities in terms of differential changes. Where nonlinear laws are required to express the relationship between the actuating quantity and the index position, the coefficients of Equation [8] are actually differential coefficients which can be treated as constants over small ranges of operation. The sentences, which follow Equation [8] in the paper, discuss the case in which the analysis will not produce accurate results due to variations of the coefficients with scale position. In practice, the analysis has been found adequate to handle the case of dynamic errors in air-speed meters for aircraft when relatively small fractional changes in speed are involved. The assumptions are similar to those used by electrical engineers in treating circuits containing vacuum tubes and are subject to the same limitation, namely, that the results will have increasing errors as the magnitude of the changes is increased.

The contention that the analysis does not include the case of systems which contain servomechanisms is only partially correct. Actually, the courses on theory and practice of servomechanisms taught by Professors Hazen and Brown at the Massachusetts Institute of Technology use the same general method of nondimensional analysis as that described in the present paper. For analytical purposes it is convenient to consider any servomechanism as made up of a series of systems with 1 deg of freedom. Frequently, the analysis of the performance of a complete servosystem involves only a second-order differential equation. Even where equations of higher order than the second are required, it has been found helpful to use the ideas of damping ratio and undamped natural frequency. The example of a servosystem of the relay type, as suggested by Mr. Smith, may be analyzed in a particularly simple fashion by use of the ideas discussed in the present case, but the method requires the application of a sequence of forcing functions acting upon the individual 1-deg-of-freedom systems (elementary instruments) which make up the servomechanism.

The criticism of the authors for failing to use a system of terminology which as yet has not been published cannot be given much weight. Surely it is granted that, until a system is unanimously adopted, the opportunity to present the arguments in favor of one which has been evolved through many exhaustive tests should not be denied. The terminology used in this paper has been developed over a period of about 10 years by the students and teachers interested in instrumentation at the Massachusetts Institute of Technology and is certainly familiar to and endorsed by a considerable number of engineers who at present are working in the field of aircraft instruments and vibration analysis. The reason underlying development of the terminology has been the lack of a generally accepted system which could be applied to dynamic problems of any type. The authors' system has facilitated the solution of many problems. Furthermore, the time and effort required to teach or to acquire a knowledge of instrument analysis have been reduced, as compared with the work required by an approach based entirely upon the contemporary treatments available in the literature.

The authors feel that the use of Greek letters in the list of symbols is a definite advantage in forming a systematic nomenclature. The general scheme is to use Greek letters for nondimensional quantities and quantities with the dimensions of time, small Roman letters for variables or directly measured quantities, and capital Roman letters for undetermined coefficients. This system has certainly justified its use by reducing the time required for an engineer to understand the results of the analysis from an unfamiliar project.

The authors disagree with Professor DeJuhasz that the quantity defined as "time lag" is unimportant. Time lag is the essential design variable in any instrument which must disregard small instantaneous variations of a quantity and indicate an average value. For example, the artificial horizon used as an aircraft instrument must have a time lag of such magnitude that it will substantially disregard deviations of the resultant-force vector from the true vertical. The requirement of a short time lag in a rate-of-climb meter used for maintaining level flight is obvious. Another application of the concept of time lag occurs when an elementary instrument is used to detect changes in the physical quantity actuating an automatic control system, short time lag being essential in order to start the system functioning soon after a change has occurred. The importance of "error after the elapse of a given time" is also apparent in this case. The existence of a negative time lag, as suggested by Professor

[7] "Instruments for Measuring Low-Frequency Accelerations in Flight," by C. S. Draper and Walter Wrigley. Presented at the Meeting of the Institute of the Aeronautical Sciences, Jan. 25, 1940. To be published in an early issue of the *Journal of the Aeronautical Sciences*.

DeJuhasz, is a physical possibility and is merely an indication that the elementary instrument is oscillating.

The concept of time lag, as the term was used in the paper, is useful and should certainly be retained in any comprehensive system of instrument analysis. On the other hand, the term "time lag" appears to be unsatisfactory and should probably be replaced by other words to describe the same concept. Professor Brown of the Massachusetts Institute of Technology has suggested the term "response delay" to replace time lag. This term is descriptive of the concept involved and will be adopted by the authors for future work.

A discussion of Coulomb friction and backlash effects as suggested by Professor DeJuhasz should certainly be included in any complete instrument analysis. One of the authors has included these effects in teaching instrumentation for some years. A reasonable generalized treatment would have been too long for inclusion in the paper. A study of Coulomb friction is included by Den Hartog in his book (5).

The use of analogies has not been found particularly helpful. It is better to allow an individual to apply theoretical results to whatever electrical, mechanical, acoustical, hydraulic, or other system which he judges to be most readily understandable. For this reason, the authors have omitted any attempt to make particular applications of the generalized results in a fashion which might be useful in explaining the theory to a given group of listeners. Mechanical analogies are readily accessible in books on mechanical vibrations, while electrical analogies are discussed in many electrical-engineering texts.

The authors have had some difficulty in recognizing the usefulness of the pendulum analogy suggested by Professor DeJuhasz, inasmuch as the mass of an instrument is largely case structure, which is not the mass of interest in this discussion. Furthermore the period of a pendulum is independent of the pendulum mass. Of course, any pendulum of a length equal to the static deflection of a 1-deg-of-freedom system will have the same period as the system, but the advantage of this type of analogy is open to question.

The maximum dynamic error normally decreases with decrease of the natural period, but it is possible to show that this statement is not general. For example, if an instrument period is originally longer than the fundamental forcing frequency, reducing the natural period will cause the system to pass through resonance when the dynamic errors increase.

The statement that the natural period should be as low as possible and the damping near critical is unnecessarily restricting. One of the particular advantages of a generalized set of curves is that, when mechanical considerations limit the natural frequency which may be realized in practice, and hence limit β, a satisfactory instrument may be obtained by a suitable choice of the damping ratio. These considerations are clearly brought out by a study of Fig. 7 of the paper, and further by the following discussion of the instrument problem proposed by Mr. Smith. As noted in the paper, there is usually an infinite variety of combinations of β and ζ which will satisfy specified instrument performance conditions. Therefore, it is the designer's responsibility to select that combination leading to the simplest and most reliable mechanical construction.

The hydraulic pressure indicator, proposed by Mr. Smith as a design problem, cannot be analyzed by any methods without a full performance specification. The word "accurate" is not an engineering specification per se, it must include the acceptable numerical tolerance range. It is this lack of reduction of instrument specifications to numerical terms that leads to many of the instrument designer's headaches. It is also not quite clear whether Mr. Smith intended the quantity to be measured accurately as the mean steady working pressure or the instantaneous variations of pressure throughout the cycle. If the latter were the problem in mind, the unsymmetrical wave form must be broken down into a Fourier series. The predominant term will be the fundamental frequency of 200 cycles per sec. To indicate the pressure variations to any degree of accuracy whatsoever, an instrument must respond to frequencies as high as the fundamental, with an accuracy at least equal to the numerically specified tolerance. When higher frequency components are present in the forcing function, the frequency range for linear response must extend to higher frequencies, if the response is to be a true picture of the forcing function.

In the particular problem at hand, where higher harmonics will be present in the instrument forcing function and no numerical accuracy has been specified, a fundamental response accuracy of at least 5 per cent might reasonably be assumed. Furthermore, to avoid resonant magnification by higher frequency components, μ (Fig. 7) must be kept reasonably close to unity by selection of a ζ of approximately 0.6. From Fig. 7, a β of 0.25 and a ζ of 0.6 will fulfill the accuracy requirements for the fundamental up to the fourth harmonic and will record within 15 per cent accuracy. These harmonic components will undergo a phase shift which is linear with frequency within the frequency range over which the magnitude response is satisfactory (Fig. 8). This means that the first four harmonic terms of the Fourier series will be reasonably well recorded with respect to the magnitude, and the shift along the time scale, due to phase shift, will be the same for all these components, thereby retaining the true wave form of the forcing function. For $\beta = 0.25$ a natural frequency of the indicator of 800 cycles per sec is necessary. It should be noted that the use of critical damping in this case would reduce the accuracy of amplitude recording, Fig. 2, of the paper.

If the desired indicator were to measure the mean pressure only, then β must have some value in excess of unity. This is similar to the case of a seismograph and has been covered in earlier papers.[8] For an indicator reading to oscillate less than 5 per cent, any value of ζ can be used if β is greater than 5 (natural undamped frequency is less than 40 cycles per sec).

The methods of this paper permit the critical instrument constants β and ζ to be evaluated quickly. The problem of achieving these constants in the instrument is the responsibility of the mechanical designer, but it is certain that the desired response cannot be obtained unless the graphically obtained constants β and ζ are actually realized in the completed indicator.

[8] For example, reference (4), Fig. 5.

Thermostatic Bimetals

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The use of thermostatic bimetals has increased rapidly during the last fifteen years. In this period, considerable study has been given the subject. The present paper reviews the available information dealing with the theory and practice in the application of bimetals. A lack of essential data concerning various pertinent properties of the material is indicated. Basic principles are thoroughly considered, as in the analysis of the mechanics by Professor Timoshenko and in the properties of nickel-iron alloys. However, a considerable fund of useful data is also presented in the form of curves and tables which give the properties of representative bimetals.

Introduction

During the last few years, the demand for automatic methods of control in regulating devices has expanded in proportion to the ease and comfort enjoyed by mankind. All concerned have increasingly realized the importance and convenience of automatic regulation and the complete satisfaction with which it may be used. Many automatic regulating devices, in which temperature can be used for controlling the regulation, depend for their successful operation on thermostatic bimetal.

In general, automatic control effects considerable savings in fuel cost. It also acts to safeguard the equipment on which it is used. Such equipment normally is left to operate unattended with full confidence that nothing is likely to happen to it. Whether in factory or home, whether to engineer or housewife, automatic control creates a peace of mind in our complex civilization which cannot otherwise be obtained.

Since the use of thermostatic bimetal has so greatly increased in recent years and is so vital to much automatic regulation, considerable work has been done on its theory and application. In this paper, the authors present a review of the available information on thermostatic bimetals which at present is scattered throughout numerous publications.

Definition of Thermostatic Bimetal

Thermostatic bimetal is a metallurgical product made from two metals which have widely different coefficients of thermal expansion and which have been firmly bonded at their face of contact. When subjected to a change in temperature, the composite material will change shape, which change of shape is utilized for control purposes. In addition to having widely different coefficients of expansion, the components must have mechanical properties suitable to minimize hysteresis and provide adequate strength. For example, lead cannot be used as a component. The two component metals are usually of equal thickness and joined together in a firm bond by brazing or welding.

History and Manufacture

Thermostatic bimetals were known for many years as laboratory curiosities. Their first known application was in balance-wheel compensators, supposed to have been used as early as 1775. As far back as 1858, Wilson obtained U. S. patent No. 24,896 covering a thermostatic bimetal. In 1863, in France, Villarceau published an analysis of thermostatic bimetal in the Annales de L'Observatoire Imperial de Paris. Wilson's patent covered a bimetal consisting of brass and steel and, since the difference in the coefficients of expansion of the two metals was not very large, the movement of the bimetal was so limited that its practical value was small. With the discovery by C. E. Guillaume in 1898–1899 of the low-expanding alloy known as "invar," consisting of 36 per cent nickel and the remainder iron, a new impetus was given the work on bimetals.

At the present time, the manufacture of bimetals is still an art rather than a science and the processes involved constitute closely guarded secrets. In one plant thermostatic bimetals were made by brazing two blocks of the component metals in a furnace in which a reducing atmosphere was maintained to prevent oxidation of the surfaces which were to be joined together. Sometimes blocks were used which were $1/2$ in. thick, 4 in. wide, and 12 in. long. Prior to brazing, one of the blocks was machined on both surfaces to a thickness of exactly $1/2$ in., while the other block was machined only on one surface. While the blocks were being brazed in the furnace, they were subjected to pressure by means of a hydraulic press to obtain a sound joint. It was very important to keep the metal surfaces clean before brazing. The blocks were usually kept in carbon paper, which prevented their oxidizing.

After the brazing operation, the exposed unfinished side of one of the blocks was machined to produce a total thickness of exactly 1 in., thereby assuring that each component was of equal thickness before reduction by rolling. The blocks were then rolled down to the desired thickness, annealing being required at a number of stages. Special rolling mills were used for this work and those in the last stages needed to be extremely accurate to finish the product to within very close tolerances.

In general, the thickness variation of bimetals cannot exceed plus or minus 0.001 in. for thickness above 0.040 in. and plus or minus $2^1/_2$ per cent below this thickness. These close tolerances are necessary to obtain uniform performance since the thickness enters into the deflection equation.

It is interesting to note that, in most cases, the final thickness of each component is very close to $1/2$ of the total thickness, even though in some bimetals the hardness of the separate components differs materially.

During the rolling processes, it is necessary to anneal the strips a number of times which it has been found most convenient to do by heating the strips electrically in the flat condition.

Properties of Invar and Related Alloys

As defined, thermostatic bimetals consist of two metals, having widely different coefficients of thermal expansion, which are joined together at their surfaces. The effectiveness of a given bimetal will depend upon the difference in thermal expansivity of the two components. Although numerous alloys now on the market have varying degrees of high expansivity, there are only a few alloys with thermal expansion sufficiently low for commercial use in bimetals.

In making a study of the iron-nickel series, Guillaume discovered that some of these alloys possessed low expansivity at

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[2] Engineer in charge of materials, Edison General Electric Appliance Company, Inc.

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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.

ordinary temperatures, the composition corresponding to minimum expansivity being in the vicinity of 36 per cent nickel. This alloy maintained its dimensions so nearly constant with ordinary variations in atmospheric temperature that he gave it the appropriate name "invar."

Since Guillaume's work, the entire iron-nickel series has been so explored that the engineer can now use an alloy with any desired coefficient of linear expansion between zero and that of nickel. With such a range of low-expansion alloys available, the development of bimetals followed as a matter of course.

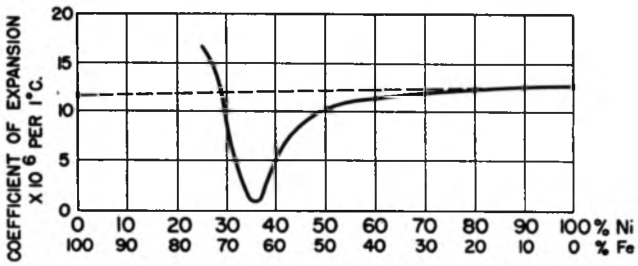

Fig. 1 True Coefficient of Linear Expansion at 20 C of Typical Nickel-Iron Alloys, Containing 0.4 Per Cent Manganese and 0.1 Per Cent Carbon (Guillaume)

In Fig. 1, the coefficient of linear expansion at room temperature of iron-nickel alloys is plotted against nickel content (1).[3] The composition corresponding to the minimum point on the curve is that of invar. As the nickel content varies from the invar point the expansion coefficient increases continuously but not uniformly. Fig. 1 is representative of typical alloys containing 0.4 per cent manganese and 0.1 per cent carbon which have been cooled in air after being hot-rolled. The point on the left edge of the grid shows the value of the coefficient of expansion of pure iron as 11.9×10^{-6} per C, while on the right edge is the point giving the value of pure nickel as 13×10^{-6} per C. The dotted line on the curve connecting the values of the coefficients for pure iron and pure nickel, represents the theoretical results obtained from the law of mixture in a condition stable at low temperatures. However, the true coefficients of the various mixtures are far from following this law, as the curve shows.

The point of low expansivity shown in Fig. 1 holds true only in an alloy in which the proportion of elements other than iron and nickel are also fixed, as has been stated. The position of the minimum is changed considerably when other elements are present.

Fig. 2 shows the change in the expansivity of nickel steels produced by the addition of 1 per cent manganese or chromium.[4] The solid curves give the change in the expansivity due to the addition of the foregoing elements, while the dotted curve gives the actual expansivity of nickel steels. The effect of these additions is to increase the expansivity of invar very considerably.

All thermal or mechanical treatment given such an alloy changes its expansivity. When such an alloy has been heated and then slowly cooled, its expansivity rises. However, its expansivity falls if the cooling has been rapid. When such an alloy is cold-rolled or drawn, its expansivity will decrease still more. By first quenching and then drawing a rod to the limit, it is possible to reduce the expansivity to such an extent as to reach a negative value. Reheating at 100 C for several hours brings the value up to substantially zero. However in processing invar for bimetal applications, advantage is not taken of this property, since reheating to higher temperatures will cause its expansivity to revert to the higher minimum.

[3] Numbers in parentheses refer to the Bibliography.

[4] Bibliography (1), Fig. 24, p. 38.

In Fig. 3 are reproduced curves showing the variation of the true coefficient of linear expansion with temperature for nickel steels varying in composition from 33.8 per cent nickel to 64.8 per cent nickel.[5] The values of the coefficient of expansion are plotted against temperature. The curves show invar (35.4 per cent nickel) to be the most desirable for temperatures from 100 to 300 F. From this point up, the coefficient of expansion greatly increases, thus making such an alloy useless for bimetal applications at higher temperatures if uniform rates of movement are desired over the entire range. Fig. 3 also shows that the alloy of 42 per cent nickel, although having a considerably higher coefficient of thermal expansion than some other alloys at room temperatures, develops only a slight increase in the coefficient with an increase in the temperature and maintains a uniform coefficient over a rather wide range of temperature. It has been found that alloys of 40 and 42 per cent nickel are particularly suitable for bimetals operating in the medium and high temperature ranges and they are used extensively for this purpose.

In an attempt to explain the expansion anomaly of invar several theories were evolved. Some of the earlier views attributed the nonexpansion to an inner molecular change. Another theory expressed the view that the nonexpansibility was caused by the high nickel content bringing the temperature of the A_3 transformation to the vicinity of room temperature, which made the

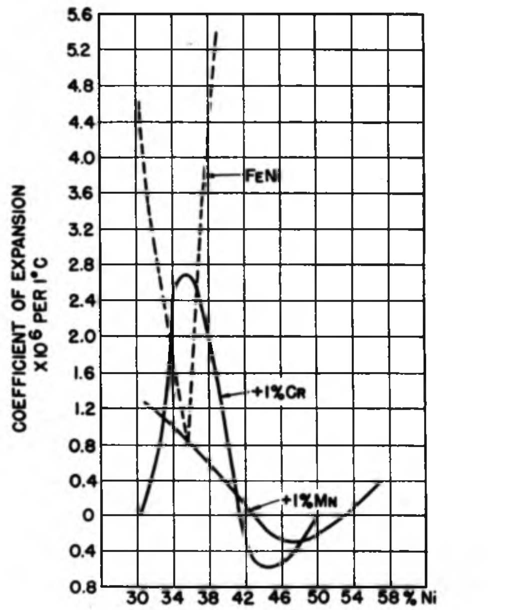

Fig. 2 Effect Upon Linear Expansivity of Nickel Steels Produced by Adding 1 Per Cent Manganese or Chromium (Guillaume)

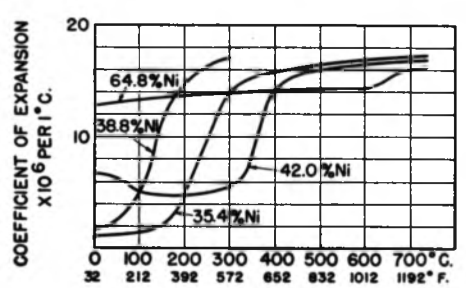

Fig. 3 Variation of True Coefficient of Linear Expansion With Temperature in Various Nickel Steels (Chevenard)

[5] Bibliography (1), Fig. 28, p. 44.

transformation of γ into α progressive. However, it was shown that the position of Ac_3 for 36 per cent nickel is about +400 C and that of Ar_3 about —100 C, thus making impossible an equilibrium at ordinary temperature and the invar would be in a decidedly unstable condition.

Benedicks (2) in developing a theory for the expansion of invar, reasoned that the alloy was in reality a two-phase system instead of being homogeneous. He proved his theory by a careful microstudy of the alloy which led him to four conclusions:

1 Strictly local, deep-etching spots occurred regularly throughout the metal.

2 These spots were often more concentrated at definite places, at which points considerably more iron goes into solution.

3 In an unetched surface, a darker, finely divided structure could be made visible in red light.

4 The deep-etching spots of invar presented the same general characteristics as those observed in a larger proportion in a 30.5 per cent nickel alloy where a two-phase microstructure was already known to occur.

From the foregoing evidence, Benedicks concluded that invar possessed a duplex structure. As he was certain that invar contained two phases, he reasoned that, since every transformation between two phases requires a definite time for its completion, by heating invar quickly, sufficient time would not elapse for the transformation between the two phases to take place. In that event, the initial expansion would be a normal one and the coefficient would lie between that of iron (12×10^{-6}) and that of nickel (13×10^{-6}). However, when sufficient time is allowed,

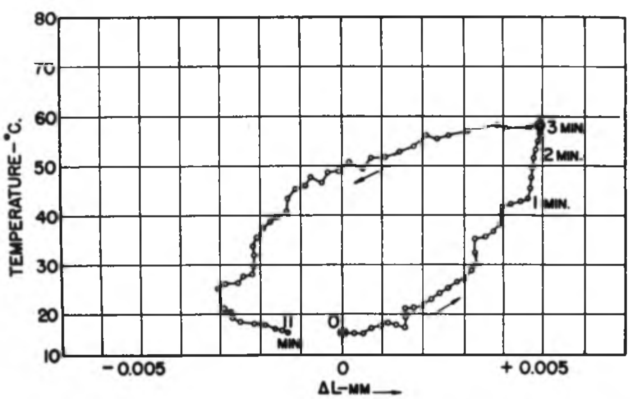

FIG. 4 EXPANSION HYSTERESIS OF INVAR (BENEDICKS)

the change caused by the two-phase transformation will become noticeable. In other words, since invar is a two-phase alloy, quick heating should show a thermal hysteresis, while if invar were a one-phase alloy, no thermal hysteresis would be detected. Consequently, the high-expansion coefficient should be expected to occur at quick temperature changes if the alloy had two phases.

To prove his theory, Benedicks conducted an experiment in which the thermal expansion of invar was measured as a function of time. He developed special equipment whereby measurements of expansions were made with an apparatus which amplified the movement 14,000 times, time being measured simultaneously.

Fig. 4 shows the results of Benedicks' experiments.[6] As expected, the thermal hysteresis was very pronounced. For a given temperature change (42 C), a sample of invar wire had an initial expansion coefficient of 13×10^{-6} or about the normal value for iron or nickel. The coefficient then diminished continually until, after 3 min heating, a considerable contraction had set in. Due to this, the resulting length diminished so much after a lapse of about 11 min as to give an expansion coefficient of 0.2×10^{-6} to 0.6×10^{-6}. This is the same magnitude established for invar by previous workers (0.4×10^{-6}).

In other words, invar upon heating first expands the same average amount as the separate constituents would expand for the same temperature change and then, after a lapse of a few minutes, contracts to show a coefficient of expansion considerably less than that of the two elements.

In applications of thermostatic bimetals where rapid rates of heating or cooling are encountered, this temperature-time hysteresis of invar may affect the deflection of bimetals by an appreciable amount. The present methods of testing bimetals result in data giving the "static" (constant-temperature-deflection) properties only. In order to detect this hysteresis effect, a "dynamic" (temperature-time-deflection) test must be used.

SUMMARY OF PROPERTIES OF INVAR

Invar is a nickel steel made either in the open hearth, crucible, or electric furnace, containing about 36 per cent nickel, 0.1 per cent carbon, and 0.5 per cent or less of manganese with negligible quantities of sulphur, phosphorus, and other elements.

It can be forged, rolled, turned, filed, and drawn into wire. It is very strong and ductile and takes a high polish, giving an excellent surface on which extremely fine lines may be ruled. It is generally ferromagnetic but becomes paramagnetic in the vicinity of 165 C.

Some of the physical properties of invar in the hot-rolled or forged conditions are given in Table 1 (3).

TABLE 1[a] PHYSICAL PROPERTIES OF INVAR

Melting point	1425 C (2600 F) (melts sharply)
Density	8 g per cc
Tensile strength	65000–85000 lb per sq in.
Yield point	40000–60000 lb per sq in.
Elastic limit	20000–30000 lb per sq in.
Elongation	30–45 per cent
Reduction in area	55–70 per cent
Scleroscope hardness	19
Brinell hardness	160
Modulus of elasticity in tension	21400000 lb per sq in.
Thermoelastic coefficient[b] (thermal coefficient of Young's modulus)	500×10^{-6} per C
Electrical resistance	75–85 microhm-cm
Thermal coefficient of electrical resistance	1.2×10^{-3} per C
Specific heat between 25–100 C	0.123 cal per g per C
Thermal conductivity 20–100 C	0.025 cal per sec per cm² per cm thickness per C temperature difference
Linear coefficient of expansion (annealed)	1.7×10^{-6} per deg C

[a] Bibliography (3).
[b] W. G. Brombacher (21) gives 480×10^{-6} per C, in range —50 to 50 C.

The amounts of carbon, manganese, and chromium present appear to exercise considerable influence on the expansion. Above 200 C, the expansion of invar is nearly that of steel.

MATERIALS FOR HIGH-EXPANDING SIDE

It has been pointed out that commercially useful bimetals must consist of two metals having widely different coefficients of thermal expansion. In the previous section, the properties of low-expansion alloys have been described in detail. Materials for the high-expansion side, in addition to having a high coefficient of expansion, should have good brazing or welding properties, should develop high elastic properties as a result of cold-working, and should have high heat-resisting properties.

Initially, brass used in combination with invar resulted in a satisfactory bimetal for low-temperature applications up to approximately 300 F. Further development led to the adoption of monel for high-temperature bimetals, but the deflection constant when using monel was rather low. To meet the requirements for high-temperature, high-deflection bimetals of good constancy,

[6] Bibliography (2), Fig. 63, p. 141.

various nickel-chrome alloys have been adopted for the high-expanding side. Representative analyses contained from 18 to 27 per cent nickel and from 3 to 11 per cent chromium.

Developments in high-temperature bimetals have been mainly in connection with the high-expanding side. Most of the patents taken out recently on bimetals of improved properties cover modifications of the high-expanding side made to obtain the desired performance.

Mechanics of Thermostatic Bimetals

Efficient and economical design of elements of mechanisms requires, first, the knowledge of stresses to which these elements will be subjected under various conditions of service; and second, data giving the physical properties of materials which may be used for these elements. This is particularly the case when controlling or indicating devices are designed, employing thermostatic bimetals, since thermal and residual stresses are present in bimetals in addition to the mechanically induced stresses. Where cost may be a primary consideration, the size of bimetal elements has to be considered, since bimetals are relatively expensive due to the involved fabrication processes. Consequently, a knowledge of the properties of bimetals and an understanding of the stress conditions in sensitive elements made of this material are essential to the most advantageous use thereof.

The first experimental investigation known to the authors of the thermal properties of bimetal strips was made in the laboratories of the General Electric Company. The results of this investigation, published in 1920 (4), were that:

(*a*) For strips, the deflection with change in temperature is given by the relation

$$D \approx \frac{(T_2 - T_1)\, L^2}{t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where D = deflection
$T_2 - T_1$ = change in temperature of the strip
L = length of strip
t = thickness of strip

The relation given by Equation [1] is in agreement with more recent investigations and theory.

(*b*) The deflection is not affected by changes in width of the strip. Later investigations showed this to be generally true if the width of the strip is less than 20 times its thickness. If the width of the strip is greater than this ratio, cross buckling may take place, reducing the deflection of the strip.

(*c*) The force exerted by the strip varies as the square of the temperature difference and is not affected by changes in length, a conclusion which is erroneous. A simple analysis of the mechanics involved, using the relation of Equation [1], requires that the force vary directly as the change in temperature, inversely as the strip length, directly as the square of the thickness, and directly as the width.

In 1925, S. Timoshenko published a complete analysis (5) of the mechanics of thermostatic bimetals. In this analysis, he developed, in a clear and concise manner, the equations giving the change in curvature of a bimetal strip upon a rise in temperature. The relative effects of variation in the thickness of the two components of the bimetal strip and the difference in the values of their moduli of elasticity were shown; equations being derived for thermal stresses in heated bimetallic strips. In 1934, T. A. Rich presented a similar analysis (6), using a well-planned group of diagrams for visualization of the stress conditions which produce a change in curvature of a bimetallic strip upon an increase in its temperature. The following analysis of the mechanics of bimetals follows closely the analysis of Timoshenko (5).

The method of visualization (6) introduced by Rich has been used in a modified form in Fig. 5 to show the effect of expansion stresses in the two components of an invar-brass bimetal strip on its curvature when heated from a temperature T_1 to a temperature T_2. Assuming that the two separate components of the strip are of the same length at some temperature T_1, the lengths will increase in both cases when the temperature is raised to T_2. The invar will increase in length only slightly, while the brass will expand considerably. Next, the lengths of the two components are equalized by elongating the invar strip with the tensile forces P_1, and compressing the brass strip with oppositely applied compression forces P_2. While under the action of these two sets of forces, the two separate strips are joined over their surfaces by either brazing or welding. The bimetal strip thus

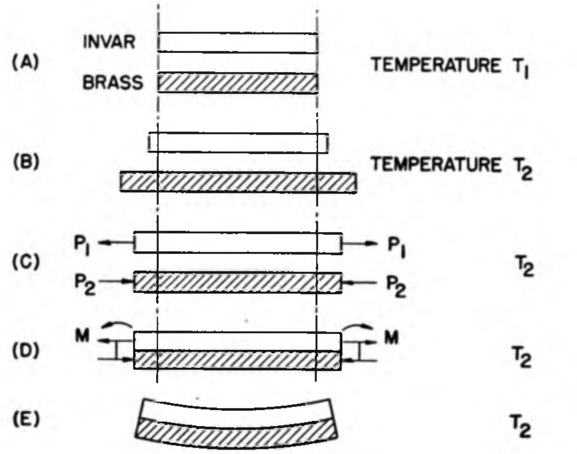

Fig. 5 Diagrams of Forces Which Produce Change in Shape of Bimetal Strip Upon Heating

formed will be straight as long as it remains under the action of the forces P_1 and P_2 applied at the ends. Upon the removal of forces P_1 and P_2, the strip will assume the form of an arc of a circle due to the action of internal moments M, which, in the straight strip, were balanced by external moments of the same magnitude, resulting from the forces P_1 and P_2.

The expression for the radius of curvature of a bimetallic strip subjected to a temperature rise of $T_2 - T_1$ deg can be obtained from the condition that, at the joint of the two components of the strip, the unit elongation is equal in each of the components. The following derivation will be carried through on the basis that the thicknesses of the two components are not equal, which makes it possible to determine the effect of differences in the relative thickness of the components. The width of the strip will be taken equal to unity, since, within certain limits, the width does not affect the deflection. The following notation will be used:

P = force
M = moment of force
α_1 = coefficient of expansion of component having low expansion
α_2 = coefficient of expansion of component having high expansion
a_1 = thickness of the low-expansion component
a_2 = thickness of the high-expansion component
E_1 = modulus of elasticity of the low-expansion component
E_2 = modulus of elasticity of the high-expansion component
I_1 = moment of inertia of the low-expansion component
I_2 = moment of inertia of the high-expansion component
t = total thickness of strip
ρ = radius of curvature, due to temperature rise $T_2 - T_1$

m = ratio of thicknesses of low- to high-expansion components

n = ratio of moduli of elasticity of low- to high-expansion components

Since no external forces are present in a freely supported heated bimetallic strip, all internal forces must be in equilibrium at every section of the strip. This condition makes it possible to establish the relations

$$P_1 = P_2 = P \qquad [2]$$

and

$$M = M_1 + M_2 = \frac{Pt}{2} \qquad [3]$$

Since

$$\rho = \frac{E_1I_1}{M_1} = \frac{E_2I_2}{M_2} \qquad [4]$$

$$\frac{Pt}{2} = \frac{E_1I_1 + E_2I_2}{\rho} \qquad [5]$$

The force P can now be eliminated from Equation [5] by determining its value from the condition that unit elongation of the two components is equal at the joint. This results in the equality

$$\alpha_1(T_2 - T_1) + \frac{P_1}{E_1a_1} + \frac{a_1}{2\rho} = \alpha_2(T_2 - T_1) - \frac{P_2}{E_2a_2} - \frac{a_2}{2\rho} \qquad [6]$$

By substituting the value of $P_1 = P_2 = P$ from Equation [6] in Equation [5], the expression for the radius of curvature is obtained

$$\frac{1}{\rho} = \frac{(\alpha_2 - \alpha_1)(T_2 - T_1)}{\dfrac{t}{2} + \dfrac{2(E_1I_1 + E_2I_2)}{t}\left(\dfrac{1}{E_1a_1} + \dfrac{1}{E_2a_2}\right)} \qquad [7]$$

Replacing moments of inertia I_1 and I_2 with their respective values $\frac{a_1^3}{12}$ and $\frac{a_2^3}{12}$, considering unit width, and taking the ratio of thicknesses equal to m and ratio of moduli of elasticity equal to n, Equation [7] reduces to the form

$$\frac{1}{\rho} = \frac{6(\alpha_2 - \alpha_1)(T_2 - T_1)(1 + m)^2}{t\left[3(1 + m)^2 + (1 + mn)\left(m^2 + \dfrac{1}{mn}\right)\right]} \qquad [8]$$

From Equation [8], the effect of the variation in the thickness of the two components of the strip may be determined. Generally the thicknesses of the two components are the same so that $m = 1$. Equation [8] thus simplifies into more useful form

$$\frac{1}{\rho} = \frac{24(\alpha_2 - \alpha_1)(T_2 - T_1)}{t\left(14 + n + \dfrac{1}{n}\right)} \qquad [9]$$

Equation [9] shows that the ratio n of the moduli of elasticity can vary over a rather wide range without affecting to any extent the radius of curvature of a heated bimetallic strip. In most bimetals, the moduli of elasticity of the two components are of the same order of magnitude so that the ratio n can be taken equal to unity. Equation [9] with this approximation reduces to its final form

$$\frac{1}{\rho} = \frac{3}{2}\frac{(\alpha_2 - \alpha_1)(T_2 - T_1)}{t} \qquad [10]$$

From Equation [10] it is possible to determine the deflection of strips of various forms supported in a number of different manners. The case of a narrow strip supported on two knife-edges will be illustrated. Referring to Fig. 6, for the case of small deflections

$$1/\rho = 8D/L_2 \qquad [11]$$

Eliminating ρ from Equations [10] and [11], the deflection at the center of a freely supported strip will be

$$D = \frac{3}{16}\frac{(\alpha_2 - \alpha_1)(T_2 - T_1)L^2}{t} \qquad [12]$$

Usually the difference of the coefficients of expansion for a given pair of metals formed into a thermostatic bimetal is determined experimentally and combined with the $^3/_{16}$ factor into the deflection constant for the bimetal. The common form of the equation for the deflection of a bimetal strip is

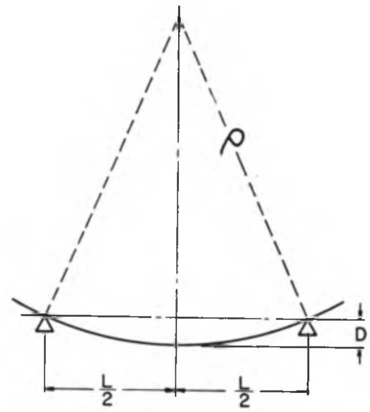

Fig. 6 Relation of Radius of Curvature to Deflection of Bimetal Strip

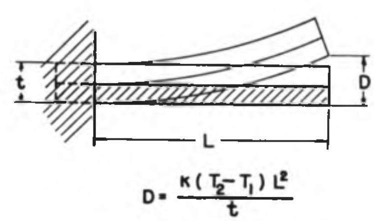

Fig. 7 Deflection of Bimetal Strip Mounted as Cantilever Beam

$$D = \frac{k(T_2 - T_1)L^2}{t} \qquad [13]$$

The deflection constant k is given in this paper for the case of a narrow strip clamped at one end and free to deflect at the other end, as shown in Fig. 7. The force necessary to return a heated bimetallic strip to its initial position is obtained by substituting in Equation [13] the value for the deflection of a cantilever beam with the force concentrated at the free end. The following relation gives the force in terms of the temperature change

$$F = \frac{c(T_2 - T_1)t^2w}{L} \qquad [14]$$

The constant c is known as the force constant and is usually determined experimentally.

Stresses in Thermostatic Bimetals

Bimetals free to deflect upon heating are subjected to internal stresses which at high temperatures may assume large values. If, in addition to internal stresses, bimetals are subjected to stresses due to restraints, loads, etc., their performance may be critically

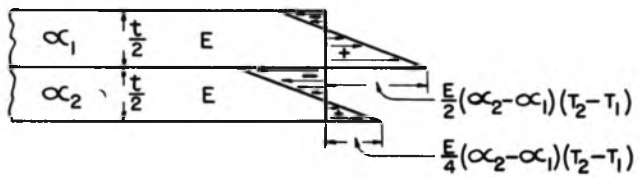

Fig. 8 Thermal Stresses in Bimetal Strip

affected if these stresses exceed the elastic limit of the material at high temperatures. Thus, it is necessary to evaluate stresses for applications involving high temperatures and loadings or restraints. In this paper, only the analysis of internal stresses due to heating will be presented. Stresses due to external forces will vary, depending upon the application. A few simple cases of combinations of internal stresses and stresses due to external forces have been analyzed by Timoshenko (5) and Rich (6).

The strains produced in a freely supported heated bimetallic strip are given in Equation [6] and consist of elongation due to thermal expansion, strains due to axial forces and strains due to bending. The stress at the joint will thus be given by the expression

$$p = P/a + Ea/2\rho \qquad [15]$$

assuming equal thicknesses of the two components of the bimetallic strip and equal moduli of elasticity. For this case the value of the force P from Equation [5] is

$$P = \frac{4EI}{t\rho} = \frac{Et^2}{24\rho} \qquad [16]$$

Substituting this value of the force into Equation [15] and remembering that $2a = t$ we obtain for the stress

$$p = \frac{Et}{12\rho} + \frac{Et}{4\rho} = \frac{1}{3}\frac{Et}{\rho} \qquad [17]$$

The value of the radius of curvature ρ is given by Equation [10] so that the stress in terms of the difference in coefficients of expansion of the two components of the bimetal and the temperature rise will be

$$p = E/2\,(\alpha_2 - \alpha_1)\,(T_2 - T_1) \qquad [18]$$

The maximum stress will occur at the joint of the two components as shown in Fig. 8. The stresses at the surfaces will be half as high as those at the joint.

For a temperature rise of 300 C (540 F) the maximum stress will have approximately the following value

$$p = \frac{20 \times 10^6}{2}\,(15 - 5) \times 10^{-6} \times 300$$

$$= 30000 \text{ lb per sq in}$$

Thermal Properties of Thermostatic Bimetals

The primary function of thermostatic bimetals is to produce a deflection or a rotation when heated. In many applications, such as thermometers, the bimetal does not have to do any work, except that of moving an indicating pointer, the bearings of which may have some friction. But there are other numerous applications where the bimetal is called on to do considerable work such as lifting parts of mechanisms or overcoming stored energy in spring or magnetic systems to obtain snap action. In the case of those applications where the function of the bimetal is mainly to provide an indication of the temperature change only, the deflection constant has to be considered. On the other hand, in cases where the bimetal is expected to do work, the force constant is of importance and has to be investigated to obtain a design of maximum efficiency.

The important thermal properties of a number of representative bimetals are illustrated in Fig. 9, the data for which were obtained from catalogs. The bimetals selected for this chart characterize the three general groups into which a number of commercially available bimetals may be classified, depending upon the temperature range of application, low, medium, and high. The bimetals selected for this illustration are made by the three largest manufacturers of this material in the United States: The H. A. Wilson Company, W. M. Chace Company, and General Plate Company. In the upper part of the figure, deflection and force constants are given as defined in Equations [13] and [14]. The lower part of the figure shows the temperature ranges for which the deflection is directly proportional to the temperature change and temperature ranges for which these bimetals have a maximum useful deflection. As will be noted, the three largest manufacturers of bimetals produced materials of practically identical thermal properties. Fig. 9 simplifies the problem of the selection of a bimetal for any specific purpose. By extending this chart to include all the available bimetals, it should be of value to designers of devices employing thermostatic bimetals. In addition to the bimetals selected for illustrative purposes in Fig. 9, a large number of other bimetals is available having widely different properties. Manufacturers' literature contains complete descriptions of these bimetals and gives their complete properties in full detail.

In order to prevent changes in calibration of devices using bimetals upon initial heating, it is necessary to anneal the bimetal carefully. During the rolling operation stresses are set up which can be relieved by a proper annealing procedure. High-temperature thermostatic metals should be annealed by, e.g., heating at 600 F for 1 hr if the material is to be used for any temperature below 550 F. In cases where the bimetals are to be used above 550 F, annealing should be done at a temperature 50 deg higher than the maximum temperature encountered in service. Low-temperature bimetals should be annealed at 350 F.

The deflection constant of bimetals changes as a result of annealing. Generally, different bimetals will show a varying degree of change in constant due to annealing. Manufacturers provide data on the effect of annealing on their respective products.

Physical Properties of Bimetals

In many applications the actuating bimetallic element is used

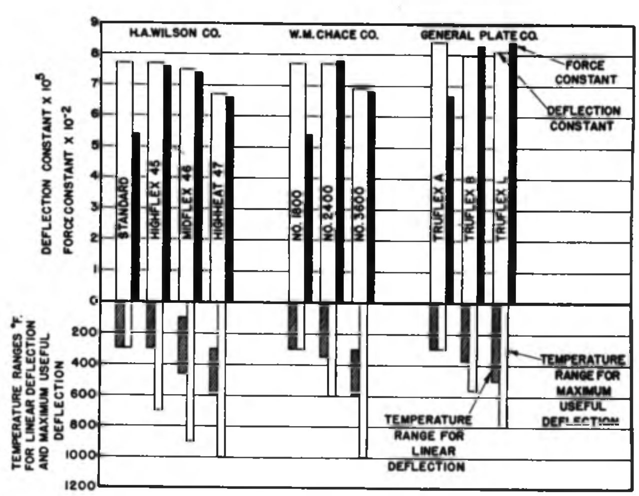

Fig. 9 Comparison of Thermal Properties and Temperature Ranges of Representative Bimetals of Different Manufacturers

in a manner similar to that of a spring; it is subjected to large deflections or rotations. It is expected, and usually it is essential, that the element should return to its initial position upon the removal of the restraining forces. The design of bimetallic elements, where large deflections occur and permanency of calibration is necessary, requires the knowledge of the stresses involved and the tensile properties of the material. The authors were unable to find in the available literature on thermostatic bimetals any data on the tensile properties of bimetals. Neither have the manufacturers of bimetals published information regarding the tensile properties of their products. In some cases, the moduli of elasticity have been determined and given in the catalogs.

One of the authors, S. G. Eskin, during a study of bimetals eleven years ago, measured the tensile strength and the elastic limit of metals available at that time. The results of these measurements are given in Table 2. Bimetals available at present probably have higher tensile properties than those shown in Table 2. With the exception of the brass-invar type of bimetals, all other bimetals can be subjected to tensile stresses of the order of 70,000 lb per sq in. at room temperatures. Manufacturers generally recommend stresses not exceeding 30,000 lb per sq in. which value allows a very wide margin of safety and makes necessary the use of more bimetal than with a less conservative value. Stresses must be considerably lowered due to "creep" which occurs at high temperatures. Table 3 shows the "safe working" stresses recommended by two manufacturers at various temperatures.

For the calculation of forces exerted by bimetallic strips when subjected to deflections, the moduli of elasticity must be known. The brass-invar type of bimetal has a modulus of elasticity approximately 17,500,000 lb per sq in. and most of the medium- and high-temperature bimetals have moduli of elasticity close to 25,000,000 lb per sq in. These values are for room temperatures; at elevated temperatures, the moduli of elasticity will be lowered from 10 to 20 per cent, depending upon the metal and the temperature rise.

Hardness of bimetals varies considerably with analysis, degree of rolling, and the heat-treatment. In general, bimetals are relatively hard, the low-expansion side varying from 205 to 260 Vickers or Brinell, while the high-expansion side of medium- and high-temperature metals varies from 230 to 300 Vickers or Brinell. These hardnesses are of the same order as those of medium-hard cold-rolled spring materials.

TABLE 2 TENSILE PROPERTIES OF THERMOSTATIC METALS

Metal and treatment	Tensile strength, lb per sq in.	Elastic limit, lb per sq in.	Elongation,[a] per cent
Wilson standard, as rolled	86000	75000	5.0
Wilson standard, annealed at 350 F	86000	75000	11.1
Wilson high-temp. constant, as rolled	110000	95000	3.5
Wilson high-temp constant, annealed at 750 F	110000	85500	12.9
Chace No. 2600, annealed	109000	94000	1.1
Chace No. 2800, annealed	105000	91000	4.0

[a] Gage length = 8 in.

TABLE 3 STRESSES FOR BIMETALS RECOMMENDED BY MANUFACTURERS

Temperature, F	W. M. Chace Co., lb per sq in.	General Plate Co., lb per sq in.
100	25000	...
150	...	30000
200	23000	...
300	...	25000
400	20000	...
500	...	20000
600	15000	...
700	...	15000
800	12000	...
900	...	10000
1000	7000	...
1200	3000	3000

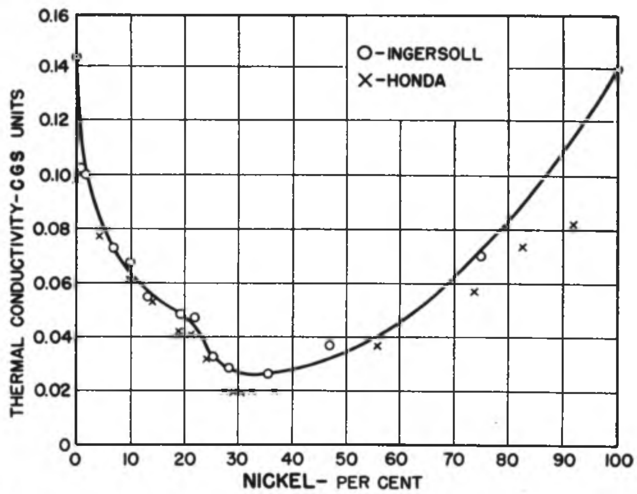

FIG. 10 THERMAL CONDUCTIVITY OF PURE NICKEL-IRON ALLOYS (INGERSOLL)

With the exception of brass-invar and some special bimetals, most high-temperature, high-deflection constant bimetals have very low thermal conductivity. This is due, first, to the fact that invar is a very poor conductor of heat. Fig. 10[7] shows that the lowest thermal conductivity of the nickel-iron alloys is in the range of their lowest thermal expansion, this latter property making them of particular value for bimetal applications. Second, heat-resisting chrome-nickel steels and similar alloys are also very poor conductors of heat. Since this is the case, most high-temperature bimetals are very poor conductors of heat. This should be borne in mind by designers of devices in which the bimetal is to be heated by conduction.

METHODS OF TEST

All manufacturers and most large users of bimetals have developed methods of test which meet their own particular requirements. While there may have been some similarity in the general test procedure, there usually existed enough difference to cause discrepancies in results when testing identical materials. To eliminate this difficulty, the American Society for Testing Materials formed Subcommittee VII, under Committee B4, to study bimetals and set up methods of test. Their latest work may be found in Tentative Standard B106–39T entitled "Tentative Method of Test for Flexivity of Thermostat Metals."

APPLICATION OF THERMOSTATIC BIMETALS

Space limitations do not allow enumeration of the very extensive uses of thermostatic bimetals. Suffice it to state that one manufacturer of bimetals lists in his catalog 60 general industrial applications and 15 automotive, aviation, and marine applications. A number of articles have been written on the application of bimetals, showing the more common forms in which bimetal elements are used; i.e., spirals, helices, disks, flat and U-shaped strips, and also special shapes to meet various unusual requirements. These references are listed in the Bibliography.

Recently two new bimetal units have been introduced which are shown in Figs. 11 and 12. The triple helix shown in Fig. 11 was developed by the Weston Electrical Instrument Corporation and is being used extensively in dial thermometers. By this method of winding a very sensitive element requiring small space is obtained. The double-helix coil shown in Fig. 12 is a product of the General Plate Company. This coil is made by winding a relatively long small diameter helix into a much shorter

[7] Bibliography (1) Fig. 40, p. 61.

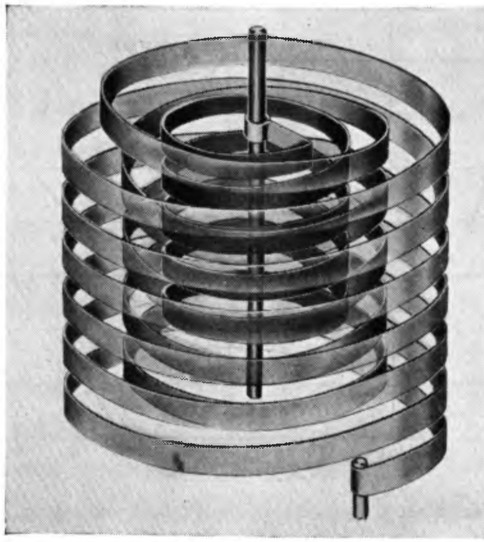

FIG. 11 TRIPLE HELIX FOR DIAL THERMOMETERS MADE BY WESTON ELECTRICAL INSTRUMENT CORPORATION

FIG. 12 DOUBLE-HELIX COIL PRODUCING LINEAR DEFLECTION WHEN HEATED; MADE BY GENERAL PLATE COMPANY

helix of a considerably larger diameter. Upon heating, this helix produces a linear deflection along the main axis of the coil.

ACKNOWLEDGMENTS

The authors are indebted to W. M. Chace Company, General Plate Company, and The H. A. Wilson Company, for valuable information in regard to bimetals and their properties. They are grateful to members of the A.S.M.E. Committee on Industrial Instruments and Regulators for many helpful suggestions in connection with this paper. Their appreciation also extends to J. C. Sharp, chief engineer of the Edison General Electric Appliance Company for making the preparation of this paper possible.

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Discussion

J. B. DOWDEN.[8] Fig. 11 of the paper shows a Weston triple-helix bimetal coil and a short discussion of its characteristics may be informative. These multiple-helix coils are made with 2 to 4 helices. A very stable self-supporting mechanical form is obtained by this construction so a long length of bimetal may be wound in a small space. A bimetal strip 25 in. long can be readily wound in a cylindrical form 0.32 in. diam $\times$ 0.50 in. long without any one convolution touching another. The combined axial expansion of the helices is practically canceled out since alternate helices are wound in opposite directions. Also there is no side thrust as in a spiral. The maximum accuracy in measuring the temperature of a small portion of a temperature gradient is secured by the most concentrated temperature-sensitive element as in this multiple-helix coil.

With regard to the expansion of invar, Benedicks (2) showed that invar has a two-phase structure and a hysteresis loop of thermal expansion if one considers the changes which occur during the heating cycle. His final experimental proof was obtained by taking many points rapidly on the temperature-expansion curve of an invar bar 80.05 mm (3.15 in.) long as it varied over the temperature range 16 to 58 C. He explained that the length of the invar changed as the material changed from one phase to the other and this counteracted the normal temperature-expansion change which was assumed to be that of iron or nickel, and thus produced an extremely small net linear-expansion coefficient.

Benedicks' experiment was chosen as being most suitable for demonstrating his theory regarding the expansion anomaly. To be directly applicable to the practical use of bimetals, similar tests would have to be made with a variety of heating rates and differently treated invar samples. It would seem from our experience with invar that the size and shape of the hysteresis loop are affected by the rate of heating (or cooling) and the condition of the invar as regards its treatment and composition. We find that bimetal coils with a small heat capacity, having one element of invar, when suddenly changed from one well-stirred liquid bath to another differing by 50 to 90 C, will register a maximum and final change in 10 to 25 sec, as nearly as can be read on a practical scale which is readable within 0.25 per cent. This would indicate that the effect of this hysteresis is negligible in practical applications with good material and design.

[8] Thermometer Division, Weston Electrical Instrument Corporation, Newark, N. J.

H. D. MATTHEWS.[9] Caution should be used in connection with the use of the word "annealing" in reference to thermal elements. Annealing is usually associated with softening the material by the use of elevated temperatures. The words "heat-treatment" are more commonly used rather than "annealing," and the heat-treating temperature should be a value slightly above the maximum value to which the material would be subjected while in service. An appreciable softening of the material will occur at temperatures above 1000 F and, if the material is annealed in the usual sense of the word, the strength would be considerably reduced. Heat-treatment relieves stresses which exist in the material, due to the fact that it is finished by cold-rolling. If the finished elements are formed in any way from strip material, additional stresses are set up, which are also relieved by the heat-treatment. Another very important factor in connection with heat-treatment is that the nickel steels which are used for the low expansive side of bimetals have an expansion value the magnitude of which is determined by the temperature history of the material. When bimetal is finished by cold-rolling, the nickel-steel components have minimum expansion values. During the first heating of the material, the expansion value is raised, depending upon the time of heat-treatment, as well as the exact temperature used. The increasing of the expansion coefficient of the nickel-steel side, due to heat-treatment, therefore, reduces the deflection constant of the finished bimetal element. This change in deflection constant can amount to a value as high as 10 per cent in some bimetals and, if an accuracy of 2 or 3 per cent is desired after being placed in service, it is necessary, of course, that the heat-treatment be given to the finished piece before determining the calibrated scale in the finished instrument. In ordinary air furnaces it requires from 2 to 3 hr properly to stabilize the elements rather than 1 hr, as mentioned in the paper.

In connection with the exact value of the deflection constant, this is affected to some extent by the relation of the thickness to the width of the strip. A change in deflection constant may be as great as 10 per cent if the width of the strip is out of proportion to the thickness. Therefore, for strips of unusual width or thickness, it is desirable to determine from tests the actual resulting deflection constant, as general formulas in use do not make any definite reference with respect to width.

The use of the phraseology "low- and high-temperature bimetals" is becoming more or less obsolete. Up to about 1915, the bimetal in general use was made of brass and invar, and was recommended for temperatures up to 300 F. Since that time a great advance has been made in heating elements, so that with the growth of the electrical heating industry, higher-temperature bimetals were necessary, resulting in the use of monel and nickel-chrome alloys for the high-expansive side. Due to our present wide range of temperature calibrations, the words "high- and low-temperature bimetals" are not particularly significant.

In connection with the theoretical deflection of bimetal as published in 1863 by Villarceau, it is interesting to note that, with present manufacturing methods of direct welding of bimetals, no slippage occurs. The actual deflection of the material agrees with the theoretical values.

The authors refer to other qualities of bimetals besides deflection as being of importance. These might be briefly enumerated as follows: Noncorrosive bimetal; bimetals with low and high resistivity; nonmagnetic bimetals; bimetals with small and large temperature coefficients.

Through several years of field experience, the writer has found that the stresses recommended by bimetal manufacturers referred to in Table 3 are extremely favorable for actual use in design work.

[9] Minneapolis-Honeywell Regulator Company, Minneapolis, Minn.

S. R. HOOD.[10] Based on Benedick's experiments (2) with invar, it is assumed that "where rapid rates of heating or cooling are encountered, the temperature-time hysteresis of invar may affect the deflection of bimetals by an appreciable amount." With all the progress in metallurgy since Benedick's work in 1926, there is still no adequate explanation of the thermal expansion anomaly of invar. The conclusions in Benedick's exposition are based on premises which cannot be verified by tests. In 1928, Howard Scott reported[11] no detectable hysteresis and suggested "the effect described by Benedick must therefore be attributed to temperature lag between his specimen and the temperature-measuring device."

An example of the rapidity with which thermostatic metal will respond to a change in temperature is best shown by the 5000 amperes test given by the Underwriters' Laboratory to domestic circuit breakers. One of the thermostatic metals for this application has a high expansive component with an expansion rate slightly lower than that of invar when the latter is heated to temperatures above 500 F. Therefore, at the beginning of a rapid temperature rise, if the initial expansion rate of the invar were high as stated, no deflection would result in the bimetal. However, in this device, the elapsed time between closing the circuit and its subsequent interruption is 0.010 sec. Only 0.003 to 0.004 sec is required for the deflection of the bimetal and the remaining 0.006 to 0.007 sec is consumed by the remainder of the mechanism. Therefore, it is safer to say that there is no temperature-time hysteresis in thermostatic metals but that their deflection depends upon the rate at which they are heated; they respond as any other metal to radiation, conduction, and convection.

It is also stated that, if the width is greater than 20 times the thickness, the resulting cross-buckling reduces the deflection of a strip. Actually, the deflection of a cantilever, either tapered or rectangular, increases with an increase of the width of the base when the length and thickness are held constant. This variation depends upon the width-length ratio, but for commercial sizes is of the order of 5 per cent.

R. G. WALTENBERG.[12] The writer believes this to be the first paper dealing with the properties of thermostatic bimetal presented before a technical society in this country, and the authors are to be commended for this initial step. There are some features of this paper which it appears desirable to amplify and there are some statements which may be questioned.

It should be noted that the data on the low-expanding nickel steels reviewed in this paper deal mainly with steels which have been cold-worked much less than the usual thermostatic bimetal.

It is surprising that anyone who has worked with thermostatic bimetal would accept without question Benedick's conclusions (2) on the expansion hysteresis of invar. There are millions of devices operated by thermostatic bimetal containing 36 per cent nickel steel which must respond to temperature changes in fractions of a second. The statement, that invar, when subjected to a temperature change from 16 to 58 C, expands for a few minutes at about the rate of iron or nickel, can be proved erroneous by plunging a thin strip of any of the common thermostatic bimetals into hot water. The bimetal will change form immediately. No time lag between change in form and change in temperature will be noticed.

We have attempted to determine expansion hysteresis by tests on thermometal composed of Armco iron against a 36 per cent nickel steel. The test specimen was a loosely wound spiral made

[10] Chief Engineer, W. M. Chace Company, Detroit, Mich.

[11] "Low Expansion Nickel Steels," by Howard Scott, Trans. American Society for Steel Treating, May, 1928, pp. 829–847.

[12] The H. A. Wilson Company, Newark, N. J.

from strip 0.006 in. thick, $^1/_8$ in. wide, and 12 in. long. This would rotate 1.12 angular deg per deg F. The spiral was held at 75 F (24 C), then immersed in oil at 200 F (93 C) and the amount of rotation was automatically recorded at intervals of 0.1 sec. The results were as follows:

Time after immersion, sec	Rotation, deg
0.1	22
0.2	32
1.0	90
4.0	140
No further rotation	

During the first 0.1 sec, the coil rotated 22 deg and in 1 sec it had rotated 90 deg. These rotations correspond to temperature changes of 75 to 95 F and 75 to 155 F with the coil in equilibrium at those temperatures. The temperature change of the bimetal could not have been much faster so there could have been little, if any, time lag between the temperature of the 36 per cent nickel steel and the corresponding expansion for equilibrium conditions. Certainly there was no hysteresis of the order discussed by the authors.

The properties of the components which affect the change in curvature of a bimetal may be combined into what is known as a deflection constant and a theoretical derivation of this constant K is given. The numerical value of K can be determined by temperature-deflection observations on a suitable specimen. Consistent results may be obtained on a beam specimen tested in the manner proposed by the American Society for Testing Materials (A.S.T.M.) and cited by the author in the section, "Methods of Test." The values of K observed by this method agree closely with the values derived from a theoretical analysis based upon the properties of the components.

A rate of deflection may also be determined by observations on the temperature rotation of spirals and helices, but the rate of rotation determined in this manner may vary 20 per cent on a given bimetal, because the amount of rotation depends upon the form and relative dimensions of the bimetal strip, particularly the ratio of width to thickness. The latter method of test is mentioned because some of the deflection constants shown in Fig. 9 were obtained in this manner. A comparison with deflection constants obtained by the A.S.T.M. method leads to erroneous conclusions.

This paper contains an analysis of stresses introduced by a change in temperature of a bimetal. This analysis gives changes in stress, not the actual value of the stress. The amount and distribution of the stresses in any part depend upon its previous history. It is possible to distribute stresses advantageously in a bimetal part by a suitable heat-treatment. Heat-treatment of a finished bimetal part is necessary if uniform and unchanging performance is desired. In most applications the treatment described by the authors is sufficient. In a few applications, it is necessary to carry the assembled thermostat over the entire temperature range, above the maximum and below the minimum temperature it will meet in service.

Authors' Closure

The authors regret that due to lack of space they were unable to give a detailed description of Benedicks' ingenious apparatus for determining the manner in which invar expands on fast heating and to describe more clearly and in greater detail results of Benedicks' experiments. Mr. Hood cited the work of Howard Scott[11] in which Scott was unable to check results previously obtained by Benedicks. The authors have studied the method used by Scott and it seems that, having been designed primarily for the purpose of obtaining easily and quickly a large amount of data, Scott's apparatus may not have been particularly suited to determine an effect which requires extremely high magnification. Scott's conclusion to the effect that Benedicks' results were incorrect was based on the premise that temperature lag existed between the test specimen and the platinum rod used as a thermometer. The authors do not agree with this conclusion but, even assuming that a temperature lag may have existed, it could not have changed the general aspects of Benedicks' investigation. The fact that at the start of heating the expansion of the invar rod was of the order of the expansion of either iron or nickel indicates that the temperature-measuring arrangement was satisfactory.

The authors were quite interested in data presented by Messrs. Hood and Waltenberg, showing that bimetals made with commercial invar give a deflection immediately upon heating. It was thus assumed that, since bimetals containing invar perform in a manner which does not follow from the results of Benedicks' researches, his work was wrong. While the authors have no definite data to explain the seemingly contradictory situation which exists, they feel that these discussers have erred in arriving at their conclusions, having approached the problem too much from the commercial point of view. A careful study of Benedicks' work would show the tremendous importance of the analysis of the invar rod on its hysteresis effect. Fig. 65 in Benedicks' book (2) shows that, while an alloy containing 36 per cent nickel will have a hysteresis effect, an alloy containing 36.9 per cent nickel will not show it. This in itself may be indicative that commercial invar may not have a hysteresis effect and, therefore, bimetals with it will respond instantaneously to temperature changes, as the results of Messrs. Hood and Waltenberg's tests indicate. The authors agree with Mr. Dowden that the entire subject requires further investigation, particularly from a purely scientific point of view.

It may be pointed out that the hysteresis effect theoretically would initially reduce the deflection of a piece of bimetal containing pure invar when heated at a fast rate by about 10 per cent. This amount of reduction in deflection requires careful measurements and may not be apparent when investigated by methods described by Messrs. Hood and Waltenberg.

The authors are grateful to Mr. Matthews for amplifying their comments in regard to annealing of thermostatic bimetals. They have used the term "annealing" in a broader sense, as defined by the American Society for Testing Materials (A.S.T.M.) in Standard A119–33.

Mr. Hood brings out the point that a bimetal strip when used as a cantilever will show a greater deflection with increases in the width. While this may be true in one particular case, generally, the contrary will occur. The authors would like to refer him to an A.S.T.M. treatise,[13] which gives the following formula as a correction for cross-curvature of a freely supported strip of thermostatic bimetal, where w is width of strip and L the distance between supports $1/\{1 + (w/L)^2\}$.

Mr. Waltenberg points out that, in Fig. 9 of the paper, the deflection constants of the General Plate Company's bimetals are higher than they should be, due to the fact that the company used coils rather than strips to determine deflection constants. It was pointed out in the paper that the data for this figure were taken from catalogs of manufacturers, assuming that the manufacturers were familiar with methods for determining accurately the deflection constants of their materials. At the request of the authors, the General Plate Company rechecked the deflection constants of their bimetals shown in Fig. 9 using strips, and found them to check very closely with constants given in the figure.

While it is quite possible that erroneous results may be obtained, when deflection constants of coils of certain geometry are recalculated to obtain deflection constants for strips, it is conceivable that a coil can be used of such a size that its deflection constant would be quite comparable with that obtained with a strip.

[13] "Tentative Method of Test for Flexivity of Thermostat Metals," B106–39T, A.S.T.M. Standards, 1939, Part 1—Metals, pp. 1164–1168.

The Significance of, and Suggested Limits for, the Stress in Pipe Lines Due to the Combined Effects of Pressure and Expansion

By D. B. ROSSHEIM[1] and A. R. C. MARKL,[2] NEW YORK, N. Y.

This paper has been prepared as a vehicle for discussion of the following basic problems, on which agreement must be reached to establish a satisfactory working-stress basis for pipe lines:

1 Proper allowable stresses for combined pressure and expansion effects.

2 Influence of and limits for localized stresses under static and repeated loading.

3 Capacity of bolted joints to withstand expansion effects without leakage or damage to flanges, bolts, or gaskets.

4 Effect of prespringing, self-springing, creep, and yielding on operating and off-stream stresses.

An attempt has been made to point out various aspects of each issue, rather than limit the presentation to the personal views of the authors.

During the last decade, economic considerations and new processes in the power, oil-refinery, and chemical industries have produced a trend toward large-scale units and high operating temperatures and pressures. With the attendant increase in line sizes and wall thicknesses, the subdivision of pipe lines into convenient runs with expansion bends soon proved entirely inadequate; instead, today most piping for severe service is carefully analyzed for forces and stresses, full advantage being taken of the inherent flexibility by minimizing the number of anchors, guides, or other restraints. A natural consequence of this improved accuracy in evaluating thermal effects is the necessity for a review of stress limitations as they apply to expansion stresses alone and in combination with internal pressure. This problem has received the active consideration of Subgroup No. 3 on Expansion and Flexibility, Subcommittee No. 8 on Fabrication Details of the Code for Pressure Piping (ASA B31) who, functioning with the Applied Mechanics and Power Divisions of the A.S.M.E., have sponsored this symposium.

This paper has been prepared as a vehicle for discussion of the following basic problems, on which agreement must be reached to establish a statisfactory working-stress basis for pipe lines. An attempt has been made to point out various aspects of each issue, rather than limit the presentation to the personal views of the authors.

1 Proper allowable stresses for combined pressure and expansion effects.

2 Influence of and limits for localized stresses under static and repeated loading, such as in bends and corrugated pipe.

3 Capacity of bolted joints to withstand expansion effects without leakage or damage to flanges, bolts, or gaskets.

4 Effect of prespringing, self-springing, creep, and yielding on operating and off-stream stresses

Résumé of Simple Stresses Involved

A pipe line is essentially a pressure vessel, the internal pressure causing a radial stress at the inner face which is converted into circumferential stress on the way through the wall, leaving zero radial stress at the outer surface; at any point the sum of the radial and hoop stresses varies inversely as the radius, while their difference is the same at all points. These are the laws of Lamé which involve the assumption of a uniform or zero longitudinal stress. Following this reasoning, in a closed cylinder the longitudinal stress equals the product of the pressure and the internal area divided by the cross-sectional area of the wall, and has the same value throughout the thickness. While most runs of piping are not technically closed cylinders, the same longitudinal stresses occur due to pressure effects on the projections of elbows, there being a complete absence of longitudinal pressure stress only in straight runs between vessels of infinite rigidity and such including frictionless expansion joints.

Under temperature changes with free expansion no stresses are introduced. However, ordinarily, expansion is restrained by the equipment to which the pipe line is attached, as well as anchors, guides, solid hangers, or supports.

In calculating a line for expansion, the ends are commonly assumed completely fixed; this connotes three forces and an equal number of moments in the case of a problem in space, resulting in longitudinal bending stresses in two mutually perpendicular planes and torsional stress about the pipe axis, as well as two shears and a normal stress. With the assumption of hinged ends, the end moments reduce to zero, a condition which would be fully realized only in a frictionless ball-and-socket joint. Actually, an intermediate condition of restraint will prevail in practice due to sympathetic deflections, rotations, or distortions of the equipment to which the pipe is attached. On the other hand, cases will arise where external movements of the ends tend to increase rather than decrease their degree of fixity; in addition, guides, solid or spring supports, and hangers often inhibit free deformation of the line and thus add to the stresses.

To avoid additional complexity, the weight of the piping is commonly neglected in flexibility calculations. In heavy pipe lines spring hangers are often used to balance the dead-load effects when the line is at working temperature, and solid hangers can be similarly employed in neutral locations. Properly designed supports appear to offer the most suitable means for handling this effect, which can then be disregarded in the stress analysis of most lines.

Where piping is not insulated, the heat loss from the exposed surface produces a temperature drop through the metal thickness causing longitudinal and circumferential stresses of equal magnitude which may be evaluated by the formulas of Lorenz (1).[3] Since the outside surface is relatively colder, tensile

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[2] Assistant Mechanical Engineer, M. W. Kellogg Co.

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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.

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

stresses will result there while the inner face will be in compression. Where conditions are not conducive to rapid heat loss, or where there is insulation even though of low thermal efficiency, these stresses are not of sufficient magnitude to warrant consideration.

Stress Intensifications and Their Significance

The Intensifications. A pipe being a simple cylinder is in itself free of secondary effects if we neglect nonuniform thickness and defects. However, most runs of piping contain joints, intersections, valves, and fittings, or may be fabricated into special shapes, which introduce differences in rigidity or changes in contour with the result that localized bending stresses must occur to preserve the continuity of the structure. It is our purpose to list the sources of stress intensifications, and briefly discuss their significance under static and repeated loadings.

When a pipe is curved into a torus shape (pipe bend or welding fitting), increased flexibility results under moments in the plane of the bend as a result of flattening of the cross section toward the neutral axis. The effect of this ovalization is to produce circumferential bending stresses of a relatively high order, and also to increase the longitudinal stress in proportion to the added flexibility. This distortion of the cross section is resisted at the ends of the bend by the straight tangents which probably tends to reduce the out-of-round shape of the curved pipe in this area and introduce corresponding ovalization into the straight pipe for a short distance.

All intersections or outlets involve bending effects due to the transmission of direct stress around the opening and the directional change at the sharp corners. In consideration of these effects, standard fittings are increased in thickness and branch connections are provided with reinforcement which, while lessening these influences, on the other hand introduces discontinuity effects due to the additional stiffness. The distortion produced by circumferential welds is in effect a corrugation which, with present-day welding technique, is probably so slight as to be of little or no consequence. In welds, as well as hot or cold bends or other fabricated parts, residual stresses are encountered which are probably reduced to an insignificant level where stress-relieved.

To secure added flexibility, straight and curved pipe are sometimes provided with circumferential corrugations which behave analogous to a bellows; as a result, appreciable longitudinal bending stresses are introduced while the circumferential stresses are probably reduced. Creased bends involve somewhat the same consideration. No adequate analytical analysis has been made of the stresses in corrugated or creased-bend pipe although some experimental research on full-size specimens has been done at the Navy Experimental Station at Annapolis by Dennison (2) and this has been supplemented with tests on 2-in. pipe by the M. W. Kellogg Company. Details of these tests and a summary and discussion of the Dennison and Kellogg data are given in Appendix 1.

In recent years the trend in high-pressure piping has been toward the omission of bolted joints wherever possible. On the other hand, such connections have performed satisfactorily in oil-refinery service under extremely high pressures and temperatures, involving in some cases frequent cycles of application. Pressure discontinuity stresses occur at the back of the flange due to the differences in stiffness, and bending stresses result from the bolt load. Due to the importance of this subject on pipe design, a detailed discussion is considered advisable and this has been incorporated into Appendix 2 to avoid overburdening the body of this paper.

Static Loading. It is generally conceded that, under static loading, localized stresses do not necessarily influence the failure load of a structure. Tests on plain pipe with stiffening rings usually result in failure in the pipe beyond the influence of the ring and not in the zones of high bending stress adjacent thereto. Similarly, many tests on unreinforced openings in pressure vessels and pipe have shown a bursting strength comparable with that of the plate or piping material, unless the openings were large with comparable distortion, in which case the effect is no longer local and failure is a function of the shape involved. Many structures have to undergo a considerable degree of cold working in manufacture and nevertheless give satisfactory service in the absence of stress relieving; for instance, the A.S.M.E. Pressure Vessel Code permits unstress-relieved vessels up to $1^1/_2$ in. plate thickness in proportion to the diameter for U-69 construction, and in most of these vessels the plates are formed cold. With the present safety factor of 5, many stress intensifications are countenanced such as the knuckle stress on relatively shallow dished heads and unreinforced openings of limited size. In the API-ASME Unfired Pressure Vessel Code, an attempt has been made to reduce the degree of possible stress intensification in view of the safety factor of 4, but here also these stress raisers are present although to a lesser degree.

In connection with transverse stresses in curved pipe, Hovgaard (3) has made tests which show that for low-carbon steel, with a minimum yield point of 25,000 lb per sq in., appreciable permanent set would not be obtained until the longitudinal stress as calculated with the longitudinal-stress coefficient derived by him exceeded 20,000 lb per sq in.

With bolted joints, the trend at the present time is to permit an overstress in the hub in proportion to the capacity of the flange proper to take additional stress, that is, the mean of the hub and flange stress must not exceed the allowable. Flanges calculated in this manner have given satisfactory performance on hundreds of heat exchangers. There is the additional consideration of maintaining a tight joint and this aspect is covered in Appendix 2. It would appear that with ASA standard flanges the bolt area is sufficient to transmit bending stress of ordinary magnitude without lowering the gasket pressure beyond the minimum required so that the flange and bolting would not be unduly overstressed. At the same time the bolt load is able to withstand appreciable torsion by means of friction between the gasket and flange faces although to a lesser degree than applies to bending, and caution must be exercised where high torsional moments are involved.

In so far as static stresses are concerned, it would, accordingly, seem to be entirely proper to permit principal stresses in the line which closely approach the yield point and to neglect the influence of highly localized stresses. This is particularly true where the distortion due to the stress intensification tends to result in a shape of increased stiffness.

Cyclic Loading. The problem of establishing suitable allowable stresses becomes more involved in the case of cyclic loading. There are some cases where load variations will definitely exist such as in discharge lines from reciprocating pumps or compressors; more often the number of stress cycles and magnitude of alternating stresses during operation are a matter of conjecture. There is little doubt as to the limitations that should be imposed for applications definitely involving alternating stresses. On the other hand, for average applications, the question whether and to what extent fatigue should be considered is not readily answered. This is particularly true in view of the large amount of pipe which is giving satisfactory service in proportion to the relatively few fatigue failures which have been encountered. Alternating stresses may also be a source of fatigue corrosion which might occur as accelerated attack in corrosive media, and is sometimes present in applications where the type of material is considered fully resistant.

In Appendix 1 the available data on fatigue tests of different types of tubular members are given and an attempt has been made to correlate this information to determine what relation, if any, exists between the flexibility factor and the useful life as indicated by the number of cycles to failure. It is regrettable that the information available is exceedingly meager; on the other hand, we believe that the data indicate a definite trend between the resistance to fatigue and the stress intensification, of which the flexibility factor is a rough measure.

Subzero Temperatures. Occasional failures are sometimes encountered in subzero piping and usually occur in an area of localized stress which might be only a flaw or scratch in the pipe. At the present time, resistance of materials to impact is compared on the basis of a Charpy keyhole-notch bar test, although attempts have been made to correlate similar information from tension impact tests. In the absence of a better explanation, failures of this nature are connected entirely with the composition and the physical properties of the material involved, although stress may be found to be a factor when our knowledge of this subject is further expanded.

Stresses Beyond Yield Point and Creep

The stresses in a pipe wall, being the superposition of individual effects following individual laws of distribution, are by no means uniform over the cross section or length of the pipe. The longitudinal pressure stress is assumed constant; the radial and circumferential pressure stresses are a function of the radius; the bending stresses due to expansion are a maximum at the extreme fibers and zero at the neutral axis of the section, and vary over the length of the pipe; the discontinuity stresses at flanges and intersections, the effects of ovalization in plain bends, and the stress concentrations at corrugations follow laws peculiar to themselves.

Local yielding, where overstress occurs in restricted areas, will effect a redistribution to adjacent fibers. When the load is removed, the fibers which have undergone permanent set will assume a negative strain, the net result being an increase in the elastic range of the structure. As an extreme condition, and neglecting increase in yield point due to strain hardening, it is possible to visualize a line carrying a negative strain with stresses at the yield point in the unloaded condition, and deflecting under expansion forces and moments until an equal positive strain is produced, with stresses again at the yield point but of opposite sign. This increase in elastic range could also be obtained by prespringing the bend to the yield point cold and then applying displacements twice as great as the prespringing but of reversed sign. The difference between "self-springing" and "prespringing" structures would be that in the first case the bend has undergone permanent deformation of certain areas which combine to shorten the distance between the ends while in the latter case no yielding was necessary.

Normally the entire elastic range (from the positive to the negative yield point) will not be utilized, and self-sprung bends will operate hot with a yield-point stress, with an off-stream stress somewhat lower. The same bend with 50 per cent prespringing will have equal stresses hot and cold, both below the yield point, or the prespringing could be up to the yield point with a low operating stress.

We next consider the effect of time and temperature on this problem. Prolonged loading at room temperature will result in a slightly lowered yield point, and as we go to higher temperatures the short- and long-time yield points decrease. This fact is the basis of stress relief at temperatures below the critical range, the procedure being to select a corresponding temperature and soaking time to yield all locked-up stresses to a desired level. This effect probably extends down to room temperature although the soaking time rapidly increases below 1000 F for carbon steel. The first effect of time and temperature therefore is to lower the stress-carrying capacity to the level of the long-time yield point at that temperature.

The yielding which occurs during the period just described is so-called "first-stage creep." Subsequently, the structure will continue to elongate under second-stage creep analogous to decreasing load or relaxation tests, the rate diminishing rapidly as the load is reduced. In time, a stress is reached which will be practically maintained over an extended period, which is commonly called the "relaxation strength." Consequently the second effect of time and temperature is to reduce the stress level to the relaxation strength of the material at that temperature.

With this reasoning, an unstressed bend will first assume elastic stresses proportional to the deflections; subsequently, first-stage and second-stage creep will reduce the stress to the relaxation strength, given sufficient time, so that there appears no possibility of more than initial overstress. The strain corresponding to the overstress will reappear when the off-stream condition obtains, so that the bend has self-sprung itself to this degree. For high-temperature piping, the self-springing therefore provides an expansion range from the desired stress cold to a relaxation stress of reversed sign. If the bend is presprung the period of adjustment is reduced and, if the prespringing is 100 per cent, there would be no bending stress due to expansion under operation.

It is evident that prespringing is beneficial and should be encouraged since it maintains the stresses in a more favorable range. Calculations should have for object the determination of the elastic range rather than the maximum stress. The average stress level should be considered in setting the allowable range, a procedure which also recommends itself from fatigue considerations. For high-temperature piping in the creep range the off-stream stress becomes significant as relaxation occurs, so that calculation for stresses at room temperature is indicated, since the maximum high-temperature stress is regulated at the relaxation-stress level for that temperature. This involves the room-temperature elasticity modulus.

Where the thrusts and moments in operation are significant, calculations at operating temperatures are also necessary. For these the reduced moduli of elasticity at the line temperature are used. In all calculations Poisson's ratio is assumed constant since few data are available on its variation.

Methods of Combining Stresses

A pipe line in space is subjected to flexural and torsional stresses as well as normal stresses and shears due to expansion effects in addition to circumferential, longitudinal, and radial stresses caused by pressure. This connotes a triaxial state of stress which must be correlated with the uniaxial failure stress obtained from the standard tensile test. The three better-known criterions of elastic failure (4) are the principal-stress theory (Rankine), the maximum-shear theory (Coulomb-Guest), and the maximum-shear strain-energy theory (Hencky-Mises). The first, although still widely applied on account of its simplicity, has been discredited, while the results of the other two theories agree fairly well among themselves and with elastic-limit tests. The Hencky-Mises theory is generally conceded to predict yielding most closely.

Neglecting, as of little practical significance, such effects as the radial pressure stress, the circumferential bending stress (which occurs in curved members), the normal stresses, and shears due to expansion, and assuming that stress-intensification factors, where applicable, have already been included in the stress components, the simple stresses which remain to be considered as acting in the extreme tensile fiber are

f_L = longitudinal pressure stress plus extreme-fiber stress due to bending
f_C = circumferential pressure stress
f_S = shear stress due to torsion

The resulting principal stresses can be written as

$$f_1 = {}^1/_2\,\{f_L + f_C + \sqrt{[4f_S^2 + (f_L - f_C)^2]}\} \qquad [1]$$

$$f_2 = {}^1/_2\,\{f_L + f_C - \sqrt{[4f_S^2 + (f_L - f_C)^2]}\} \qquad [2]$$

$$f_3 = 0$$

In the principal-stress theory, f_1 is directly compared with the allowable stress f and the two lower principal stresses are disregarded.

In the maximum-shear theory, the difference between the two extreme stresses governs. If both f_1 and f_2 are positive, the same criterion is obtained as for the principal-stress theory. If, on the other hand f_2 is negative, a different limitation applies. Accordingly, the following two conditions must be satisfied

$$f_1 = {}^1/_2\{f_L + f_C + \sqrt{[4f_S^2 + (f_L - f_C)^2]}\} \leqq f \qquad [3]$$

$$f_1 - f_2 = \sqrt{[4f_S^2 + (f_L - f_C)^2]} \leqq f \qquad [4]$$

Finally, for the Hencky-Mises theory, the failure criterion can be written as

$$\sqrt{(f_1^2 + f_2^2 - f_1 f_2)} = \sqrt{(3f_S^2 + f_L^2 + f_L^2 - f_L f_C)} \leqq f \qquad [5]$$

With steady creep, the maximum-shear theory is commonly used, although Bailey (5) has developed a formula analogous to the Hencky-Mises equation, involving constants which are evaluated from the creep properties of each material. Before selecting a theory, some thought should be directed at criteria of failure. The stress theories mentioned predict yielding, not ultimate failure and, with extensive yielding or creep, the elastic theory no longer controls the stress distribution. In thick cylinders under pressure the Hencky-Mises theory would no doubt predict the yielding of the innermost fibers, but general yielding of the cylinder is probably best predicted by the simple stress theory, and the same is true of failure.

Where stresses are calculated according to the elastic theory, the principal-stress theory could be safely continued in use since it is in step with present-day safety factors. In fact, the Barlow or outside-radius formula of the Piping Code already penalizes piping in comparison with pressure vessels (A.S.M.E. and API-ASME Codes both use the Lamé formula which follows Rankine's theory for thick pressure vessels). If stresses are to be evaluated more closely and the safety factor reduced, then the Hencky-Mises theory should be used if yielding destroys the utility of the structure. If some yielding is not considered detrimental, then further studies and tests are desirable to correlate combined loads with rupture.

Where stresses are calculated under conditions of steady creep rather than the elastic theory, the maximum-shear theory is used by Bailey (5). Norton (6) has made creep tests on tubular specimens which indicate similar rates where the average simple circumferential stress and the simple tensile stress in the cylinder and test bar were the same, although with longer tests there are indications that the creep rate would decrease for the cylinders. Bailey establishes the ratio of diametral to tensile creep as $^1/_2$ to $^1/_3$ for the same shear stress. Norton's tests would indicate that the use of the average stress from the elastic theory (Lamé—mean of outside- and inside-diameter stresses) is entirely safe for a hydrostatic loading.

Suggested Design Assumptions and Stress Limits

Up to this point, we have been concerned with the presentation and discussion of the data and reasoning affecting the problem of pipe lines under expansion, and have found the subject to be quite complex. It is obvious that considerable simplification must be introduced in order to establish rules suitable for general application. In the following, basic assumptions for use in evaluating stresses and limits for the combined stresses resulting from pressure and expansion are suggested.

Evaluation of Expansion Thrusts and Stresses. The calculation of the line for expansion serves the purpose of establishing the range over which stresses vary rather than the maxima and minima, and accordingly consideration of the amount of prespringing required can be deferred to the end of the calculation. Since the relaxation stress definitely limits the hot condition, both prespringing and self-springing will tend to make the room-temperature condition the controlling one and calculations accordingly should be based on the elasticity modulus at room temperature rather than the reduced value in operation. Where a close estimate of thrusts in operation is desired, a simple conversion by the ratio of the hot to the cold elasticity moduli will give the required results.

Simple Stresses to Be Considered. In arriving at the principal stresses, consideration should be given in all problems to the following stresses at the outside surface of the pipe:

Simple pressure stresses:
- Circumferential
- Longitudinal

Line expansion stresses:
- Longitudinal bending (two planes)
- Torsion

Local stresses:
- Longitudinal bending in curved pipe and creased ells
- Longitudinal bending in corrugated pipe.

The radial pressure stress is zero at the outer surface. Direct stress due to the axial force and shearing stresses caused by transverse forces need be calculated only where the piping is extremely stiff. The local stresses are evaluated conveniently by using stress-intensification factors which are discussed later. We believe that the circumferential bending stress (curved pipe) and other local stress raisers, such as branch connections, may be neglected, although the use of unreinforced openings and reinforcement details should be more closely controlled where service conditions involve distinctly cyclic loading. Bolted joints will be commented on as a separate item.

Stress-Intensification Factors—Fatigue. In addition to the ordinary usage of the term "stress-intensification factor," it will here also be applied as a correction factor reducing the elastic range of stresses where cyclic loading is involved. The values given in Table 1 are suggested as a working basis.

TABLE 1 SUGGESTED STRESS-INTENSIFICATION FACTORS

Shape	Stress-intensification factor: Noncyclic stresses	Cyclic stresses
Plain tangents	1	1
Plain bends and creased bends	β	k
Corrugated tangents and bends	$2^1/_2$	5

Cyclic stresses would be considered to obtain where service conditions involve pulsating loads or frequent temperature changes; a closer definition should be considered, although it cannot be expected to relieve the designer entirely from a consideration of the merits of each case.

The factors for plain tangents require no comment. In the case of plain bends, k and β are the familiar Kármán or flexibility factor and Hovgaard's longitudinal stress-intensification factor, respectively. Both are applied to flexure stresses in the plane of the bend only. Creased bends are realized to occupy an intermediate position between plain and corrugated pipe, but are usually classed with the former in the absence of reliable data. For straight and curved corrugated pipe, stress calculations are based on the moment of inertia of the pipe before corrugating, and the stress-intensification factor is applied to the longitudinal pressure stresses and the bending stresses. It is realized that the moment transverse to the plane of a bend can vary from pure bending to pure torsion with an intermediate range where both effects are present, and that the true state of stress is not evaluated by present methods; however, this and kindred problems can be considered as future refinements as our knowledge of this subject expands.

For noncyclic stresses, we have used the endurance properties of 20,000 cycles as a rough guide; for cyclic loading, 500,000 stress reversals have been considered as fairly representative of the endurance limit.

Combination of Stresses. If failure is associated with yielding, the Hencky-Mises theory most accurately predicts the elastic limit under combined stresses, although the maximum-shear theory offers sufficiently close results and is somewhat easier to apply. On the other hand, present-day safety factors for pressure stress are predicated on the principal stresses (Rankine theory) and, since local yielding results in stress redistribution with no loss in utility, average stresses are probably the best over-all strength index. For the present, we would favor the continued use of the principal-stress theory for calculations based on the elastic theory. Where stresses under steady creep (5) are being considered, the maximum-shear theory may be used.

Suggested Stress Limits. Using as a basis the allowable stress established in the codes for calculating pipe wall thickness under pressure, the following limits for the available stress range (already discussed as a suitable criterion for stress-deflection comparison) are suggested:

For no prespring

$$\text{Allowable stress range} = 0.75\ (S_A + S_A') \qquad [6]$$

and for 50 per cent prespring

$$\text{Allowable stress range} = S_A + S_A' \qquad [7]$$

where S_A = allowable stress at room temperature and S_A' = allowable stress at operating temperature, both to be introduced with their algebraic values so that they will always be numerically additive.

As an illustration, the allowable ranges obtained for combined pressure and expansion stresses in accordance with formulas [6] and [7] are listed in Table 2 for various line temperatures, as based on allowable pressure stresses for seamless low-carbon steel piping (48,000 lb per sq in. minimum ultimate tensile strength) to A.S.T.M. Specification A-106 as proposed for the ASA Code for Pressure Piping.

TABLE 2 ALLOWABLE RANGES FOR COMBINED PRESSURE AND EXPANSION STRESSES AS OBTAINED WITH FORMULAS [6] AND [7]

Line temperature, F	Allowable stress range — Oil-piping section, No prespring	Oil-piping section, 50% prespring	Power-piping section, No prespring	Power-piping section, 50% prespring
Up to 650	18000	24000	14400	19200
700	17663	23550	14100	18840
750	17175	22900	13740	18300
800	16500	22000	13200	17600
850	15413	20550	12330	16440
900	14250	19000	11400	15200
950	12563	16750	10050	13400
1000	10875	14500	8700	11600

For any intermediate degree of prespring, the relation

$$\text{Allowable stress range} = (0.75 + s/2)\ (S_A + S_A') \qquad [8]$$

could be used, where s = percentage of prespring expressed as a fraction = 0.50 maximum.

A higher degree of prespring than 50 per cent is beneficial at elevated temperatures in establishing the piping nearer equilibrium, thus reducing the amount of permanent set necessary to bring stresses to the relaxation limit. However, this will not affect long-time conditions, and is therefore not considered to warrant an increase in the elastic range.

Investigation of Bolted Flanged Joints. The rules suggested in the preceding apply to pipe and do not consider joints. These are properly investigated after the completion of the line calculations. We have already indicated that ASA standard flanges are probably sufficiently strong to take any bending stresses ordinarily encountered and, at least with ring-type gaskets, possess a fair resistance to torsion. In doubtful cases, and where special flanges are used, an investigation analogous to that carried out for two flanges in Appendix 2 will be indicated. The ASA flanges should probably be explored more thoroughly, and definite limits imposed on the allowable bending and torsion moments depending on the type of gasket used.

Appendix 1

TESTS OF PLAIN, CORRUGATED, AND CREASED-BEND PIPE UNDER CYCLIC VARIATION OF STRESS

Few, if any, pipes break in service due to the application of a steady load or but a few repetitions. Since the common pipe materials are quite ductile, under steady stress a small amount of yielding ordinarily will relieve highly localized overstress and effect a redistribution without hazard of failure, provided the average stress does not already exceed the yield point. Where, on the other hand, repeated stress variations are incurred due to temperature or pressure changes, or vibration, an entirely different situation arises. The application of a large number of stress cycles of sufficient amplitude will cause fatigue failure, which is accelerated by flaws in the material or surface irregularities.

The fatigue properties of a metal are characterized by the endurance limit which is the stress below which no fracture will occur even after an indefinite number of stress reversals, and is usually derived from cyclic flexure tests in which the extreme fibers are alternately subjected to maximum tensile and compressive stresses of equal magnitude. It has been found that the ratio of endurance limit to ultimate strength, or so-called endurance ratio, ranges between 40 and 50 per cent. Where stresses are not completely reversed, but vary about a definite mean, the limiting stress is raised. For a variation from zero to a maximum, an approximately 50 per cent higher limit is found comparing to between 60 and 75 per cent of the ultimate strength. A general formula covering any arbitrary stress range has been suggested by Moore and Kommers (7), as

$$S_{\max} = \frac{3S_L}{2 \pm (S_{\min}/S_{\max})} \qquad [9]$$

where S_L is the endurance limit for complete reversal, and the sign of the ratio of minimum to maximum stress is negative where both stresses act in the same sense, and positive where one is tension and the other compression. In the absence of corrosion, cycle frequency appears to have no influence on fatigue-resisting properties, the number of cycles alone controlling.

Tests of Corrugated and Creased-Bend Piping by Dennison (2)

Realizing the advantages offered by corrugated and creased-bend piping, particularly in confined spaces, and at the same time noting the lack of concise information on the characteristics of such piping, the Bureau of Engineering of the Navy Department authorized large-scale flexibility and fatigue tests on which Dennison (2) reports. The information contained in this valuable paper is summarized briefly below.

Standard commercial products were used, the specimens including tangents, square and offset quarter bends, and expansion U-bends of 6-in. standard-pipe-size seamless tubing of carbon steel to A.S.T.M. Specification A106–33T, with different heat-treatments. The ultimate tensile strength of the various pieces ranged from 51,500 to 72,600 lb per sq in.; the yield point, from 33,400 to 46,000 lb per sq in.; the elongation in 2 in. from 25.5

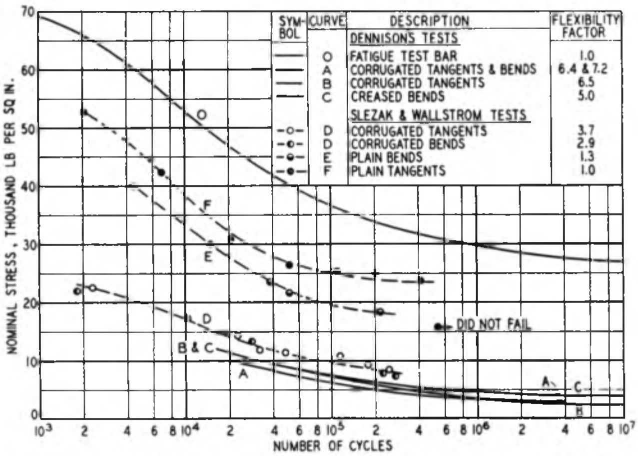

FIG. 1 ENDURANCE LIMITS FOR PIPE AND PIPE MATERIAL FROM DENNISON'S AND M. W. KELLOGG COMPANY'S DATA

to 48.0 per cent; and the reduction in area from 33.5 to 75.1 per cent.

The corrugations showed a $2^1/_2$-in. pitch and $^3/_4$-in. height. The creased bends were made on a 15-in. radius, the creases being on approximately 11 deg 15 min pitch and extending around two thirds of the circumference. For the corrugated tangents, a flexibility factor of 6.4 was established, and for the corrugated arcs, a value of 7.2 (as compared with 2.8 for plain pipe bent to the same radius); for the creased bends the flexibility factor was 5.0 (as compared with a theoretical value of 3.8 for plain pipe of the same radius).

Two types of fatigue-testing equipment were used—the one, a rotating-beam machine, producing a specific stress in the specimen while the other applied a definite deflection through a crank mounted on an eccentric. Tests made with complete stress reversal were correlated with tests where the stress ranged from zero to a maximum by application of formula [9] Moore and Kommers (7); this simply requires reducing the stresses for the second form of loading in the ratio of 2 to 3 before plotting.

Dennison's results are reproduced in the three lower curves on the fatigue graph shown in Fig. 1. Curve *A* includes corrugated arcs and tangents. Curve *B* applies to corrugated tangents of slightly different thickness and shape, and curve *C* to creased bends. Curve *O* gives the results of a standard fatigue test on the material itself which had an ultimate strength of 63,000 lb per sq in. and an endurance limit of 27,000 lb per sq in. or an endurance ratio of 0.43.

The presentation used is the customary semilogarithmic graph in which the logarithmic scale of abscissas measures the number of cycles to failure; it makes little difference whether this is defined as the ultimate fracture or the point at which slightly eccentric rotation heralds the end of a test, since the intervening interval is extremely brief. The vertical scale reads the "nominal" stress calculated for the point of fracture in accordance with the common or Rankine theory based on the pipe dimensions before the corrugations or creases are formed; in other words, the "nominal" stress equals the bending moment divided by the section modulus of the original pipe.

Space does not permit a detailed discussion of the interesting analysis and conclusions offered in Dennison's paper (2). Instead, we will conclude with a few notes on the type of failure experienced. With a single exception the corrugated pipe always failed at the crest of a corrugation, the cracking progressing from the inside out. On the other hand, the creased pipe invariably failed through the trough between two creases, with the cracks starting at the outer surface and proceeding inward; the same was true of the corrugated pipe cited previously as an exception, which was bent to a short radius so that it assumed an intermediate shape between corrugation and creasing. Dennison points out that the failure in all instances starts at the surface of sharpest curvature which is the inside of the crest for corrugated pipe and the outside of the trough for creased pipe.

TESTS OF PLAIN AND CORRUGATED PIPING BY E. J. SLEZAK AND H. WALLSTROM OF THE M. W. KELLOGG COMPANY

While Dennison has admirably achieved his immediate objective of developing information on the endurance of corrugated and creased-bend piping and correlating this with the endurance limit of the parent metal as obtained from a standard fatigue-test specimen, the authors have been reluctant to accept fully the conclusions he derives therefrom in the absence of a parallel investigation on plain piping with which a wealth of service experience is associated. It is felt that comparison with plain pipe would furnish a better yardstick for the appraisal of the limits within which corrugated or creased pipe should be stressed in actual installation. In order to supply such a basis, a simple series of tests were projected which have recently been completed by E. J. Slezak and H. Wallstrom under the authors' supervision. A description of the tests follows:

A 24-in. lathe was transformed into a rotating fatigue-testing machine of the specific-strain variety. The size of the model bends was restricted by available clearance; also, by the force applied, due to the construction of the lathe, in particular the feedscrew. Since the tests were intended to give results representative of commercial pipe and the corrugating processes used for the larger pipe cannot be properly extended to too small a diameter, 2-in. nominal-size standard-weight seamless pipe was used throughout. In setting up for a test, the model was rigidly attached to the chuck of the lathe at one end after the outstanding leg had been carefully aligned to revolve on the dead center of the tailstock. A self-aligning roller bearing attached to the cross-feed mechanism was then advanced over the end of the pipe which, owing to the difference between the size of the bearing and the pipe, was fitted with a reducing collar. Next, a predetermined amount of deflection was applied by means of the transverse feedscrew; as an additional precaution, this was checked against a dial gage mounted on the lathe bed. In order to be able to dispense with an accurate determination of the moment arm and obviate the necessity of complicated deflection calculations (made uncertain by variations in stiffness introduced by the reducing collar as well as the varying flexibility of the corrugated sections), the stress introduced by the applied deflection was directly gaged by means of an accurately calibrated Huggenberger extensometer attached to the plain pipe tangent provided at the chuck end. Both the dial and extensometer were, of course, removed before the lathe was started up. The number of revolutions was recorded by a mechanical counter attached to the shaft. The test arrangement is shown in Figs. 2, 3, and 4.

The material for the 2-in. seamless pipe was taken from stock and was manufactured to A.S.T.M. Specification A53–36. After fabrication, the pipe received a stress relief at 1150 F with

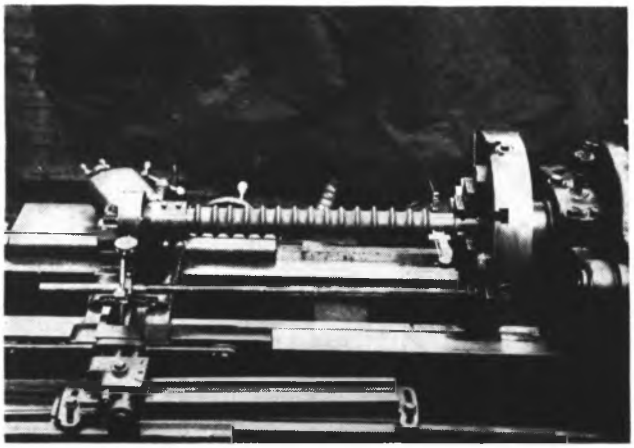

FIG. 2 CORRUGATED TANGENT MOUNTED IN THE LATHE FOR GAGING STRESS AND DEFLECTION PRIOR TO FATIGUE TEST

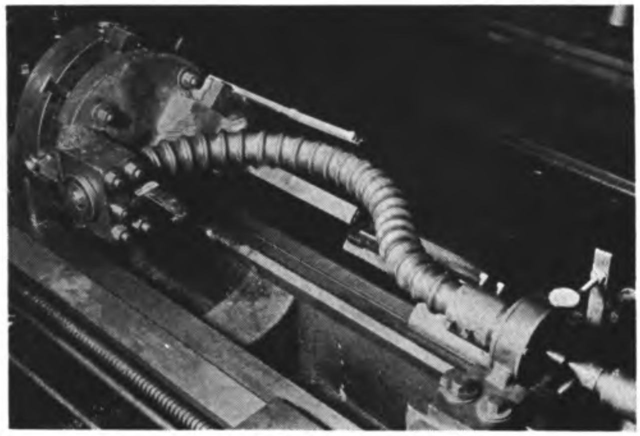

FIG. 4 CORRUGATED OFFSET QUARTER BEND MOUNTED IN A LATHE FOR GAGING STRESS AND DEFLECTION PRIOR TO FATIGUE TEST

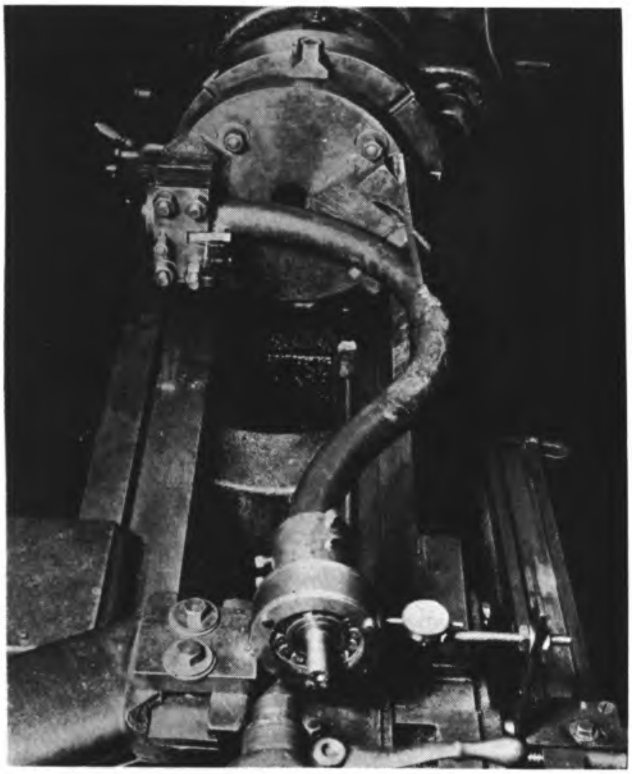

FIG. 3 PLAIN OFFSET QUARTER BEND MOUNTED IN THE LATHE FOR GAGING STRESS AND DEFLECTION PRIOR TO FATIGUE TEST

FIG. 5 MICROGRAPH OF PIPE MATERIAL USED IN M. W. KELLOGG COMPANY'S TESTS

subsequent furnace cooling to 600 F. A tensile test gave an ultimate strength of 59,400 lb per sq in., a yield point of 36,700 lb per sq in. (as obtained by the drop of the beam), and an elongation of 45.6 per cent in $1^1/_4$ in. gage length. Satisfactory bend and flattening tests were made. A micrograph is shown in Fig. 5 which reveals the pearlite nearly completely dispersed, with cementite particles well spheroidized.

A number of tests were run with each of the four types of models shown on the right-hand margin of Fig. 6; these include tangents and offset quarter bends of plain and corrugated pipe. The corrugations used had an average pitch of $1^9/_{16}$ in. and a depth of slightly over $^3/_8$ in. As was demonstrated by a series of load-deflection tests, this contour does not offer the degree of flexibility common to the larger pipe sizes, averaging 3.7 for the tangents and 2.9 for the arcs. This was the first 2-in. pipe corrugated by our shop; the flexibility probably could be improved by changing the corrugations. For the plain offset quarter bend, calculations give a flexibility factor of 1.3 and a (static) stress-intensification factor of 0.9; following the established theory, the bend actually would be stressed lower than an equivalent tangent.

The results of the fatigue tests are plotted in Fig. 1 on the same basis as Dennison's data. In addition to the average curves marked D, E, and F, the values of each individual test are shown by a circle, different symbols being used for each of the four test

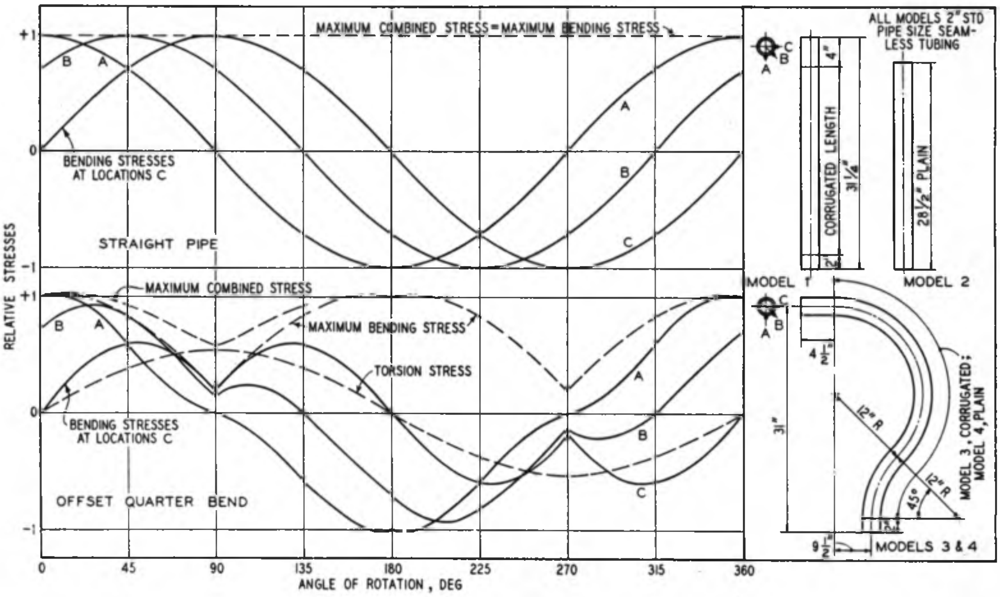

FIG. 6 STRESS CYCLES OF M. W. KELLOGG COMPANY'S MODELS

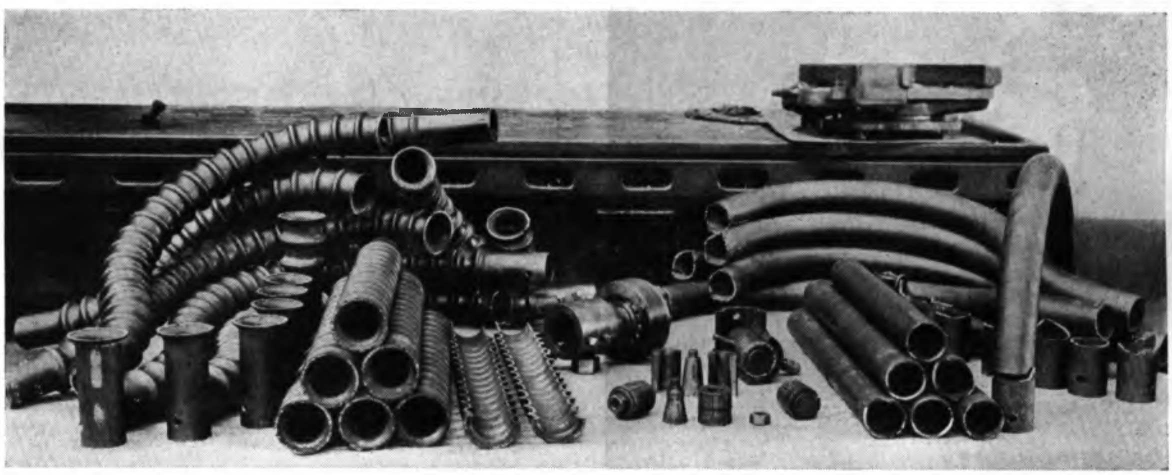

FIG. 7 FAILED SPECIMENS, PARTS OF TESTING APPARATUS, AND CORRUGATED TANGENTS CUT TO ILLUSTRATE THE SHAPE OF THE CORRUGATIONS

series; the arrow attached to the point furthermost to the right denotes a test which was discontinued without failure. Since the results for the corrugated tangents and bends were in close accord, a single curve, marked *D*, was drawn for both series. It should be noted that there is quite a reduction in endurance for the plain tangents (curve *F*) as compared with the metal itself. (Since the ultimate strength of the pipe material in the present tests is close to that obtained for the material in Dennison's tests, it is assumed that curve *O* can serve as a parameter for both investigations.) A further reduction in fatigue properties is found for the plain arcs (curve *E*) despite the reduced theoretical stress.

A short note may be in order on the type of cyclic variation of the stresses applied. While the stress curve for each point on the circumference of the tangents follows a sine wave, this is not true for the offset quarter bends shown in Fig. 6. The curves of both flexural and torsional stress are given at three points on the circumference for the location of maximum stress near the fixed end; point *A* is on the inside of the bend in the plane of its center line, and points *B* and *C*, respectively, are 45 and 90 deg distant therefrom. Only point *A* and a point diametrically opposite are subjected to stress reversal of the maximum amplitude. Despite this fact, and the added complication of a cyclic torsional stress, the corrugated arcs behaved similarly to the tangents, and the conclusion, also implied in Dennison's presentation (2), can be drawn that the highest stressed point controls failure provided average conditions are not too dissimilar.

Examination of the failed specimens of corrugated pipe confirms the observations of Dennison (2). In the case of plain pipe it was natural to expect a more jagged break since there is no sharp concentration of stresses; here surface markings and weak ligaments dictate the path of least resistance. Fig. 7 shows all the failed specimens assembled, with corrugated bends and tangents at the left, and plain bends and tangents at the right; in the center, the roller bearing, reducing collar, and various parts of the clamping devices are shown, together with the two halves of a corrugated tangent cut through longitudinally to illustrate the shape of the corrugations.

DIGEST OF RESULTS OF FATIGUE TESTS

In order to review the results of both the Navy (2) and the Kellogg tests for the purpose of formulating rules suitable for general application to the practical problem of pipe-line design,

TABLE 3 FATIGUE DATA FROM DENNISON'S (2) AND KELLOGG'S TESTS

Test series[a]	Description of models tested	Average flexibility factor, k	Avg ult tensile strength, lb per sq in.	Number of cycles	Limiting stress S, lb per sq in.	S/UTS	S/S_O	S/S_F
O	Standard fatigue test bar, polished	1.0	63000	20000	46900	0.740	1.00	1.46
				100000	36700	0.580	1.00	1.47
				500000	31100	0.500	1.00	1.32
				2500000	28100	0.450	1.00	..
F	2-In. standard - pipe - size seamless steel tubing, plain tangents	1.0	59400	20000	32000	0.540	0.68	1.00
				100000	25000	0.420	0.68	1.00
				500000	23500	0.400	0.76	1.00
				2500000	...	...	..	..
E	2-In. standard - pipe - size seamless steel tubing, plain offset bends	1.3	59400	20000	27800	0.470	0.59	0.87
				100000	19700	0.330	0.54	0.79
				500000	17600	0.300	0.57	0.75
				2500000	...	...	...	...
D	2-In. standard - pipe - size seamless steel tubing, corrugated tangents and offset quarter bends	3.3	59400	20000	14100	0.240	0.30	0.44
				100000	9800	0.160	0.27	0.39
				500000	7500	0.130	0.24	0.32
				2500000	...	...	..	..
C	6-In. standard - pipe - size seamless steel tubing, creased quarter bends and U-bends	5.0	59000	20000	11400	0.190	0.24	0.36
				100000	7600	0.130	0.21	0.30
				500000	5300	0.090	0.17	0.23
				2500000	4300	0.073	0.15	..
B	6-In. standard - pipe - size seamless steel tubing, corrugated tangents (one creased radius)	6.5	57300	20000	11400	0.200	0.24	0.36
				100000	7600	0.130	0.21	0.30
				500000	4400	0.077	0.14	0.19
				2500000	2800	0.049	0.10	..
A	6-In. standard - pipe - size seamless steel tubing, corrugated tangents and various shapes of bends	6.8	58400	20000	9900	0.170	0.21	0.31
				100000	6600	0.110	0.18	0.26
				500000	4000	0.069	0.13	0.17
				2500000	3000	0.051	0.11	..

[a] Series *A*, *B*, *C*, and *O* are Dennison's tests. Series *D*, *E*, and *F* are Kellogg's tests made by Slezak and Wallstrom.

S_O = Limiting stress using standard fatigue-test bar, test series *O*.

S_F = Limiting stress using plain straight pipe, test series *F*.

UTS = Ultimate tensile strength.

all the pertinent information on each separate series has been assembled in Table 3. The listing follows the order of increasing flexibility of the significant part of each bend model. Limiting stresses have been read from the original fatigue curves for 20,000, 100,000, 500,000, and 2,500,000 cycles and these are correlated to the ultimate tensile strength in column 7. In column 8, the limiting stresses obtained for the pipe for each number of cycles are compared with the corresponding stresses for the standard fatigue-test bar. In column 9, the limiting stresses for the plain bends, creased bends, and corrugated tangents and corrugated bends are correlated with those applying to the plain pipe tangents.

A relationship can be established between the flexibility factor and the endurance of the material, which appears to be independent of the source of increased flexibility (whether flattening of the cross section, as in the case of plain bends, or partial or complete corrugation, as in the case of creased or corrugated pipe). This will be illustrated by reference to the 20,000-cycle and 500,000-cycle stresses, for which the following formulas permit a reasonably close estimate of the limiting stresses for pipe of varying flexibility. For 20,000 cycles

$$S_{k_{20}} = S_{20}/\sqrt[3]{k^2} \qquad [10]$$

and for 500,000 cycles

$$S_{k_{500}} = S_{500}/k \qquad [11]$$

where k is the flexibility factor of the pipe; S_{20} and S_{500} are the "virtual" endurance limits, or limiting stresses for pipe of unit flexibility at 20,000 and 500,000 cycles, respectively; and $S_{k_{20}}$ and $S_{k_{500}}$ are the "virtual" endurance limits for pipe of a flexibility factor k at 20,000 and 500,000 cycles, respectively. Values computed in accordance with formulas [10] and [11] are compared in Table 4 with those obtained by test.

TABLE 4 ENDURANCE LIMITS COMPUTED FROM FORMULAS [10] AND [11] COMPARED WITH TEST RESULTS

	Virtual endurance limit $S_{k_{20}}$			Virtual endurance limit $S_{k_{500}}$		
Flexibility factor, k	Computed	From test	Difference, % of computed stress	Computed	From test	Difference, % of computed stress
1.0	32000	32000	±0	23500	23500	±0
1.3	26900	27800	+ 3.3	18100	17600	− 2.8
3.3	14400	14100	− 2.1	7100	7500	+ 5.6
5.0	11000	11400	+ 3.6	4700	4300	− 8.5
6.5	9200	11400	+23.9	3600	2800	−22.2
6.8	8900	9900	+11.2	3500	3000	−14.3

If the exponents of k were changed to 0.65 and 1.04, respectively, for the 20,000- and 500,000-cycle stresses, the difference would be reduced considerably. However, formulas [10] and [11] should be viewed as rough guides in the interpretation and extrapolation of these tests only, and additional refinement appears unwarranted in view of the meager basis on which they are founded.

In the tests, the endurance of the plain tangents was between two thirds and three quarters that of the test bar and, by introducing a factor approximately 0.7, an estimate of the virtual endurance limit for pipe of any flexibility can be derived from the fatigue characteristics of the material.

Appendix 2

THE PROBLEM OF BOLTED FLANGED CONNECTIONS

A study of the effects on bolted flanged joints of eccentric loading introduced by the expansion of a pipe line cannot be properly undertaken without a reasonably correct understanding of the fundamental problem of the action of flange and gasket under the bolt load required to produce a tight seal under internal pressure. While thousands of bolted flanged connections are in continuous successful use in an ever-widening variety of services, surprisingly little precise knowledge of the stresses and deformations of the component parts is available. It is only the recent trend toward higher temperatures and pressures in the steam power plants and chemical industries which has forced a reconsideration of the entire basis of design and lent impetus to research which is already yielding valuable returns. We refer in particular to the work of the British Pipe Flanges Research Committee (8) which operates on a broad basis, to the improved approaches to flange design published by Waters, Wesstrom, Rossheim, and Williams (9), as well as Bailey's (10) contribution to our understanding of the behavior of flanges under conditions of creep, and the many individual investigations made in recent years here and abroad with reference to the performance of gaskets.

The complexity of the problem involved will be apparent from a consideration of the variables. In general, the shape of a flange does not readily lend itself to accurate mathematical analysis, particularly since the method of its attachment to the pipe, by welding, rolling, screwing, or otherwise, affects the stress distribution. The effects of bolt holes and the gasket groove are difficult to include. Serious uncertainties are introduced by the necessity of making assumptions for the locations of the bolt and gasket reactions which depend on the stiffness of the flange. While a formidable number of test results are available on a wide variety of gasket shapes and materials, no entirely satisfactory correlation of the data has yet been presented and opinions regarding the gasket contact pressures required to seal a joint under varying conditions show considerable divergence. The share of the elastic deformation of the compound system assumed by the bolting depends on the material, size, length, and spacing of the bolts, on the development of the shank and type of thread used, on the material and height of the nut and, to a small extent, even on the compressibility of the washer. Allowable design stresses have to consider the yield strength as well as the fatigue properties and are affected by the manufacture and heat-treatment of the steel.

Conditions are further aggravated in the case of hot lines where the problems of creep and relaxation make it extremely difficult, if at all possible, to develop an accurate analysis of the behavior of a bolted joint. In addition, the different expansions of the component parts must receive consideration and this, in turn, presupposes at least an approximate knowledge of the temperatures assumed by the flange, bolting, and gasket in operation and during starting up and shutting down. The question arises whether flanges should be insulated. If lagging is provided, the metal temperatures can be maintained more nearly uniform and sudden failure due to change in operating or atmospheric conditions or thermal shock is less likely to occur; on the other hand, the life of an uninsulated connection will probably be greater due to the reduced temperatures.

In view of the multiplicity of considerations entering into the design of an ordinary bolted flanged connection it is comforting to be able to state that, in general, a few relatively simple rules supported by experience and good judgment will enable a designer to proportion the parts in such a way that the joint will perform satisfactorily in operation, at any rate under conditions which do not involve excessive temperatures or rapid fluctuations.

Both the A.S.M.E. Code and the API-ASME Code for Unfired Pressure Vessels have adopted the flange formulas developed by Waters and Taylor. An extension of the method to include shear and to cover tapered hubs has been published recently (9); this new derivation also permits the evaluation of the effects of internal pressure by a secondary investigation while the earlier analysis restricted itself entirely to the bolt moment. While the present paper is not concerned with flange design proper, a brief exposition of certain phases will be necessary in order to show how the effects of expansion of a pipe line on the bolted flanged connection can be taken into account.

The following forms of loading must be considered in the design of bolting for a flange:

1 The internal pressure load, which is usually taken over the circular area bounded by the center line of the gasket.

2 The minimum load required to set the gasket in the cold condition. It has been found that, up to certain pressures, the force required to maintain a tight seal is independent of the internal fluid or gas pressure; this limit varies for different materials and shapes of gasket and also depends on the flange facing. The minimum contact load can be interpreted as the load required to establish line contact around the circumference for the harder gaskets; for the softer packings, it probably signals a sufficiently intimate contact over a larger area so that the gap between the gasket and flange faces is reduced to such minute dimensions that crossflow cannot occur.

3 The minimum load required to maintain the joint tight under operating pressures exceeding the limit mentioned in 2. For such pressures the required gasket force appears to be a direct function of the internal pressure. The value of the gasket-contact ratio is dependent on the type of gasket and facing.

4 A sufficient additional compressive load to allow for the reduction in gasket pressure caused at the joint by direct tensile stresses and tensile bending stresses due to thermal expansion of the line, the fundamental idea being that this pressure under no circumstances can be allowed to drop below the minimum required to preserve a tight joint.

5 A sufficient total load to prevent rotation in the joints due to torsional stresses induced by the line expansion. These are taken in friction which is a direct function of the bolt load.

6 An adequate initial load to compensate for yielding of any part of the assembly due to suddenly applied pressure or temperature loads.

7 For service at elevated temperatures a design load which does not stress the component parts beyond their relaxation strength, so that the required gasket compression will be maintained for the desired period.

While the conservation of a sufficiently high gasket load under all conditions of operation is the primary concern in the design of the bolting, care should be taken to proportion the gasket so that overpulling of the bolts or the application of expansion thrusts or compressive bending stresses will not lead to its failure by crushing. Furthermore, the effect of internal pressure on the gasket in producing excessive hoop stresses and the danger of blowouts must be considered. The flanges of course, must be able to withstand the maximum bolt moment exerted, and, while ordinary allowable stresses need not apply, the proportions must be such as to prevent excessive distortion.

Disregarding the effects of line expansion, cold design stresses between 20,000 and 25,000 lb per sq in. have been used rather widely in this country for alloy-steel bolting; these are stresses maintained in operation, although the initial stresses required to set the gasket are very often much higher. For elevated temperatures where creep controls, stresses producing 1 per cent strain in 100,000 hours are applied; as determined by investigators in this country, such stresses will remain below the relaxation strength. In England, Bailey (10) from a test on carbon-molybdenum steel arrives at the conclusion that the relaxation limit for 100,000 hours corresponds to the stress causing 0.1 per cent creep in that period (11). For the flanges, the applicable code stresses are used except that it is not uncommon to allow a 50 per cent higher limit for the hub stresses which are of a highly local nature and easily relieved.

With the introduction of a more accurate analysis of a stress problem, it is common practice to reconsider safety factors. In the past, flanges have withstood stresses in excess of those calculated considering the bolt load required for tightness only, and accordingly a precedent is established for permitting higher design stresses where the effects of line expansion are included. Before we suggest any new stress limits, it will be in order to form an estimate of the magnitude of the hitherto uncalculated expansion stresses in a bolted flanged connection.

A review of a large number of stress calculations involving

TABLE 5 RESULTS OF INVESTIGATION ON TWO ASA WELDING NECK FLANGES WITH STANDARD RING JOINTS FOR THE CONDITION WHERE THE BENDING MOMENT DUE TO EXPANSION IS HIGH ENOUGH TO PRODUCE A STRESS OF 15000 LB PER SQ IN. IN THE ADJOINING PIPING

Pipe data:			
1	Line size, in.	4	24
2	Flange series, lb	300	1500
3	Pipe schedule	40	160
4	Bending moment producing a stress of 15000 lb per sq in. in the pipe, in-lb	48200	11730000
Limits for unit gasket compression based on gross width of ring:			
5	Minimum pressure required to set the gasket with no internal pressure (1/8 ring width × 18000 lb per sq in. yield point), lb per sq in.	2250	2250
6	Minimum pressure required to maintain tightness under operation (1/4 ring width × 5.5 × internal pressure), lb per sq in.	413	2063
7	Maximum pressure to prevent crushing gasket, lb per sq in.	30000	30000
Bolt stresses for various conditions:			
8	Minimum stress required to set gasket with no internal pressure, lb per sq in.	7500	1900
9	Minimum stress to hold internal pressure plus gasket pressure under operation, lb per sq in.	4750	7950
10	Stress required to withstand moment causing a stress of 15000 lb per sq in. in the pipe, lb per sq in.	10100	8600
11	Minimum stress required for tightness (item 8 or 9, whichever is higher), lb per sq in.	7500	7950
12	Minimum stress required for tightness and bending moment (item 10 + item 11), lb per sq in.	17600	16550
13	Probable limit to which bolt may be pulled without control, lb per sq in.	31800	20000
Flange stresses for various conditions:			
For minimum bolt load required for tightness (item 11):			
14	Longitudinal hub stress, lb per sq in.	3300	5300
15	Radial flange stress, lb per sq in.	3700	6500
16	Tangential flange stress, lb per sq in.	2800	6200
For minimum bolt load for tightness and bending moment (item 12):			
17	Longitudinal hub stress, lb per sq in.	6900	10000
18	Radial flange stress, lb per sq in.	7900	12200
19	Tangential flange stress, lb per sq in.	5900	11700
For condition where bolts are pulled up to limit (item 13):			
20	Longitudinal hub stress, lb per sq in.	12000	11800
21	Radial flange stress, lb per sq in.	13600	14400
22	Tangential flange stress, lb per sq in.	10300	13900
Maximum gasket compression on gross width:			
23	For minimum bolt load required for tightness (item 11), lb per sq in.	2250	9500
24	For minimum bolt load for tightness and bending moment (item 12), lb per sq in.	5300	19750
25	For condition where bolts are pulled up to limit (item 13), lb per sq in.	9550	23800

pipe lines connecting to pumps gave the following averages for the end reactions:

Vertical thrust plus weight of pump riser including valves and fittings, lb	$= 3.25\ D^3$
Lateral thrust in any direction, lb	$= 1.50\ D^3$
Bending or torsional moment, in-lb	$= 60\ D^3$

where D equals the outside pipe diameter increased by 3 in. The corresponding bending or torsional stresses in the pipe due to expansion ranged from 1000 to 6000 lb per sq in. It has been our practice to limit the total stresses in pipe lines to 1.5 times the allowable code stresses, one third of which is absorbed by the longitudinal pressure stress; accordingly, some carbon-steel installations below 750 F based on the API-ASME Code with material of 60,000 lb per sq in. ultimate strength must have supported up to 15,000 lb per sq in. bending stress due to expansion. Since the ends of a line usually are locations of maximum stress and normally will be flanged for attachment to a nozzle provided on the vessel, there will probably always be one or two flanges which are subject to the maximum moment.

In order to appraise conditions at a joint under such circumstances, two ASA welding neck flanges with standard ring joints will be investigated for the conditions where the bending moment due to expansion is high enough to produce 15,000 pounds per square inch stress in the adjoining piping; the data are given in Table 5. It will be noted by referring to Tables 6 and 7 that the examples selected are fairly representative of the range of conditions met with in ASA flanges.

TABLE 6 PIPE DATA

Pipe schedule	Nominal size, in.	Area, sq in.	Section modulus, in.³
40	4	3.17	3.21
	8	8.40	16.81
	12	15.74	47.09
	16	24.35	91.49
	20	36.15	170.40
	24	50.30	285.10
80	4	4.41	4.27
	8	12.76	24.51
	12	26.04	74.53
	16	40.14	144.50
	20	61.44	277.20
	24	87.17	467.90
160	4	6.62	5.90
	8	21.97	38.47
	12	47.14	122.50
	16	70.85	233.50
	20	109.92	453.50
	24	157.51	780.60

TABLE 7 FLANGE DATA

		Bolts[a]		Ring gasket[b]	
Flange series, lb	Nominal size, in.	Root area, sq in.	Section modulus, in.³	Gross area, sq in.	Section modulus, in.³
150	4	1.616	3.03	5.77	8.45
	8	2.416	7.10	9.57	23.33
	12	5.028	21.37	14.73	55.24
	16	8.816	46.84	17.55	78.43
	20	14.560	91.00	21.60	118.80
	24	18.580	137.00	26.02	172.38
300	4	2.416	4.76	8.07	11.87
	8	5.028	16.34	14.60	38.78
	12	11.650	51.69	20.62	77.33
	16	18.580	104.51	25.43	117.61
	20	22.300	150.50	36.13	207.75
	24	33.720	269.80	53.51	364.54
400	4	3.350	6.60	8.07	11.87
	8	6.610	21.49	14.60	38.78
	12	14.860	65.94	20.62	77.33
	16	23.100	129.94	25.43	117.61
	20	33.720	227.61	36.13	207.75
	24	47.520	380.16	53.51	364.54
600	4	3.350	7.12	8.07	11.87
	8	8.736	30.03	14.60	38.78
	12	18.580	89.42	20.62	77.33
	16	28.100	166.84	25.43	117.61
	20	40.320	287.28	36.13	207.75
	24	55.300	456.23	53.51	364.54
900	4	5.820	13.46	8.07	11.87
	8	13.860	53.71	14.60	38.78
	12	23.100	121.27	20.62	77.31
	16	33.600	203.70	36.32	167.98
	20	53.040	391.17	54.19	311.59
	24	85.840	761.83	85.61	583.22
1500	4	7.430	17.65	8.76	13.96
	8	20.160	78.12	20.86	55.41
	12	43.430	244.29	41.23	154.61
	16	68.670	476.40	65.38	302.38
	20	101.18	828.41	90.32	519.34
	24	140.000	1365.00	117.71	801.90
4000[c]	4	7.430	17.65	11.49	14.00
	8	23.760	94.30	28.27	56.54
	12	51.500	291.30	48.60	136.69

[a] Root areas of bolts are computed for V-threads with 8 threads per inch, minimum. In calculating the section modulus of the bolting, it has been considered permissible to assume the bolt area uniformly distributed over the bolt circle.

[b] The approximate formulas $A = 2\pi rw$ and $SM = \pi r^2 w$ have been used in calculating the ring-gasket areas and section moduli, respectively, where r = radius of the pitch circle and w = gross width of the ring.

[c] Proposed.

TABLE 8 INVESTIGATION OF ASA FLANGES (TABLE 5) EXTENDED TO COVER TORSION EQUAL IN MAGNITUDE TO THE BENDING MOMENTS INVESTIGATED[a]

Line size, in.	4	24
Flange series, lb	300	1500
Pipe schedule	40	160
Torsion moment, in-lb	48200	11730000
Vertical thrust required $P = M/(f \times$ gasket radius), lb	82500	4290000
Corresponding bolt stress = (P/bolt area), lb per sq in.	34000	30600
Required minimum gasket compression = P/gasket area, lb per sq in.	10200	36500

[a] A friction factor of $f = 0.20$ is assumed.

This investigation would tend to show that the ASA flanges, at least where used with ring joints, have a sufficient margin of strength in all parts to take considerable bending moments without danger of overstress. In both examples, the check against overpulling of bolts, to which the authors subject each flange as a routine precaution, gives higher values than the computation including an unusually high bending moment. It is a well-known fact that ASA flanges are overbolted in order to allow for high gasket contact ratios required on some forms of gaskets. This is no longer true of the proposed 4000-lb standard, which is intended for ring joints only or alternate gaskets requiring a low sealing force. The 4-in. 4000-lb size actually has the same bolting as the 4-in. 1500-lb size, and correspondingly the margin for bending stresses is considerably reduced; also, the gross ring area is by no means in proportion to the rating, and accordingly caution must be exercised in the use of such flanges where line stresses are high.

Where torsion moments are applied to a joint and no mechanical means for absorbing them are provided, these must be transmitted through friction between the gasket and the flange faces. Our study of representative ASA flanges was extended to cover torsion equal in magnitude to the bending moments investigated; the data are given in Table 8. A friction factor of $f = 0.20$ is assumed.

Neither flange can be considered adequate to withstand the torsional moments applied. While it is true that the assumptions used are extremes, the investigation clearly indicates the need for caution where torsion is concerned. This is even more true for gaskets other than metal rings with a lower degree of rigidity particularly such that suffer from abrasion.

Similar lines of reasoning can also be applied to shear effects and pipe thrusts, although it is believed that neither ordinarily will be significant.

Concerning the calculation of flanges under the conditions of creep and relaxation obtaining at elevated temperatures, the reader is referred to an interesting article by Bailey (10). Unfortunately, a large degree of uncertainty attaches to the creep constants and temperature assumptions which must be made, and present experience at high temperatures is insufficient to permit of correlating theory and practice. As far as superimposed stresses due to line expansion are concerned, Bailey suggests that these tend to diminish rapidly and need not be considered in connection with the ultimate steam tightness of the joint.

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7 "The Fatigue of Metals," by H. F. Moore and J. B. Kommers, McGraw-Hill Book Company, Inc., New York, N. Y., 1927.

8 "First Report of the Pipe Flanges Research Committee," by H. J. Gough, Proceedings of The Institution of Mechanical Engineers, vol. 132, February, 1936, p. 201.

9 "Formulas for Stresses in Bolted Flanged Connections," by E. O. Waters, D. B. Wesstrom, D. B. Rossheim, and F. S. G. Williams, Trans. A.S.M.E., vol. 59, April, 1937, paper FSP-59-4, p. 161.

10 "Flanged Pipe Joints for High Pressures and Temperatures," by R. W. Bailey, *Engineering*, vol. 144, July–December, 1937, pp. 364, 419, 490, 538, 615, and 674.

11 For further discussion of working stresses in bolting see: "Piping Handbook," by J. H. Walker and S. Crocker, McGraw-Hill Book Company, Inc., New York, N. Y., third edition, 1939, pp. 259–263 and 376–379. Also: "Flanged Pipe Joints—Methods of Applying and Determining Bolt Stresses," by E. C. Petrie, *Heating, Piping and Air Conditioning*, vol. 8, June, 1936, p. 303. Also: "First Report of the Pipe Flanges Research Committee of the British Standards Institution," *Engineering*, vol. 141, 1936, pp. 243–245 and 271–273.

Discussion

William Hovgaard.[4] This paper is very suggestive and affords a good basis for a discussion of the many difficult questions which arise in the design of pipe lines. The problem presented may be summarized as that of the determination of factors of safety for pipe stresses.

It is necessary to distinguish between the states of stress which exist locally and those which exist generally throughout the pipe. In the former class belong the longitudinal stresses denoted by f_L in the paper and stresses localized near the ends of the pipe. These stresses combine with hoop stresses and torsional stresses, creating a state of stress which should not be allowed to approach the condition of plasticity. If flow does occur locally, a permanent set is liable to occur, although the safety of the pipe may not be endangered. It is believed that this state of stress is best measured by the Hencky-Mises formula, given as Equation [5] in the paper. The expression on the left side of the equation does not represent an actually existing stress, although it has the dimensions of a stress. It is simply a function of the component stresses, so constructed that when it reaches the value of the yield point, flow of the material will occur. This function the writer has called the "equivalent stress" in various papers, and has denoted it by f_{eq}. It is suggested that the equivalent stress should not exceed one half of the stress at the yield point; that is, the factor of safety, in case of the stresses here under consideration, should be at least 2 relative to the yield point. It is found that then with all ordinary values of the circumferential or hoop stress, f_C, the longitudinal stress, f_L, can be from 1000 to 2000 lb per sq in. higher than the equivalent stress.

In the latter class of stresses, which are general over the entire pipe, belongs the hoop stress, f_C, caused by the steam pressures. It has a special significance, because it cannot be relieved by a change of form of the pipe or by a readjustment of the strains. When the pressure is associated with high temperatures, creep has to be considered and, due to our incomplete knowledge of this phenomenon, it is advisable to use a relatively high factor of safety. Hitherto, it appears that, in the case of the hoop stress, the factor of safety has been ordinarily related to the ultimate strength. This was quite satisfactory so long as the ratio between the ultimate and the yield strength was fairly constant, but in modern alloy steels, where this ratio may vary widely, it seems safer and more rational to relate this factor to the yield point, which determines incipient breakdown. It is of interest to note that Lloyd's Rules, for instance, prescribe a thickness of the pipe wall, which, including the additional increment of 0.12 in., gives a factor of safety of about 7 relative to a yield point at 500 F of 27,000 lb per sq in. This is obtained, assuming a steam pressure of 400 lb per sq in., a wall thickness of 0.319 in., and a hoop stress of 3750 lb per sq in. An analysis of other rules representing ordinary practice seems to indicate that the factor of safety of the hoop or "bursting" stress relative to the yield point should not fall below 6.

A. E. R. de Jonge.[5] The authors should be highly commended

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[5] Mechanical Engineer, The Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

for having undertaken the arduous task of outlining the problems encountered in determining the stresses in pipe lines. As their paper is intended to be a "vehicle for discussion" of these problems to supply material for a new piping code, it offers an occasion to stress the difficulties of this undertaking.

As the total stress can only be arrived at by a combination of the individual stresses, the latter have first to be considered separately before they can be combined.

With regard to internal pressure, the Lamé formula for thick-walled cylinders is probably the best to use as it gives results which agree fairly well with practical measurements. It might be said that an attempt should be made to obtain uniformity in all codes concerning pressure vessels with respect to the use of this formula.

Expansion stresses, on the other hand, provide a very different problem. First, the fundamental assumption made in their determination is that the ends of the pipe line are rigidly fixed. This assumption, however, is rarely if ever correct. It is therefore necessary to begin with the investigation of the rigidity of the various apparatus connected by pipe lines and to classify them with regard to the "degree of fixedness" assigning definite percentage values or percentage limits to them. For instance, a pressure vessel supported on a high gantry will offer but a slight degree of fixedness in the horizontal plane while it must be considered more or less rigid in the vertical direction. This investigation and classification has not only to be extended to linear displacements, but also to rotations.

However, even when disregarding all considerations of partly fixed ends, the problem of calculating stresses in pipe lines in space is by no means solved as yet with anything approaching accuracy. The theoretical values for the deflections and rotations in pipe bends, subjected to torsion, transverse bending, and transverse forces, do not check with the test results, differing in some cases by more than 50 per cent, the error being systematic, i.e., always in the same direction. Even straight pipe lines do not seem to follow the laws of ordinary beams any better. Thus, compound pipe lines in space exhibit quite different flexibilities from those usually assumed in the calculations, although it should be stated that these values could be allowed for in the calculations if they were known with any degree of certainty. Dead weight and local stresses at hangers, anchors, etc., are usually neglected and so are temperature stresses. In heavy pipe lines, however, deadweight should not be neglected. With all this uncertainty, it does not seem to be of much use to lay down fixed rules for maximum stresses.

Fortunately, pipe lines are generally more flexible than assumed, and this factor works in the direction of lower stresses. In addition, should overstressing occur, the usual materials for pipe lines exhibit the useful property of ductility and thus automatically lower the stresses in the pipe line by local yielding. The same holds with respect to creep. Prespringing is another uncertain factor, and so is the fact that load variations or load alternations occur. In combining the stresses obtained for these individual effects, it is clear that the end result will be even more uncertain.

Taking all these factors into consideration, the fixing of stress limits can only be regarded as a very rough approximation which of necessity must prove to be highly controversial. The actual approach to the subject obviously lies in investigating all of the different factors separately and more carefully than has hitherto been done, to fix for each one the limits within which theory and measured results agree, and then to compound these limit factors into a common limit factor. It is apparent that, for this purpose, experimental investigations on the flexibility of actual large-size pipe lines are required, and while such investigations have probably been made by individual firms, very little has been published regarding the results of such experiments. Only by coordinating theoretical calculations and experimental investigations is it possible to arrive at any data which may be regarded as approaching reality and thus be of value. By using such an approach to this complicated problem, the usual safety factor will no longer be a "factor of ignorance" in so far as pipe lines are concerned.

Joseph Marin.[6] Considering the condition of static loading, the authors' Equation [5] includes the stresses due to bending, torsion, and the longitudinal and circumferential stresses due to pressure. It is of interest to study, in addition, the radial stress and to determine the influence of the various forces in producing failure. If the radial stress f_3 is included as shown in Fig. 8 of this

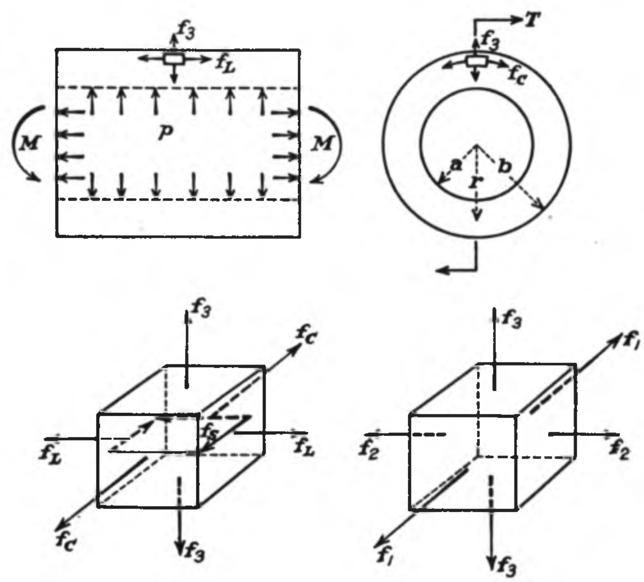

Fig. 8 Three-Dimensional Aspect of Stress Problem With Radial Stress Included in Addition to Stresses Due to Bending, Torsion, and Pressure

discussion, the problem becomes a three-dimensional one and the distortion-energy theory of failure becomes

$$f_1^2 + f_2^2 + f_3^2 - f_1f_2 - f_2f_3 - f_1f_3 = f^2 \ldots\ldots [1$$

where f_1, f_2, and f_3 are the principal stresses.

Designating f_L, f_C, and f_S as the longitudinal stress, circumferential-pressure stress, and shear stress, respectively, the principal stresses f_1 and f_2 are

$$f_1 = {}^1/_2[f_L + f_C + \sqrt{(f_L - f_C)^2 + 4f_S^2}] \ldots\ldots [13]$$

and

$$f_2 = {}^1/_2[f_L + f_C - \sqrt{(f_L - f_C)^2 + 4f_S^2}] \ldots\ldots [14]$$

Placing values of f_1 and f_2 from Equations [13] and [14] in [12]

$$f_L^2 + f_C^2 - f_Lf_C + 3f_S^2 + f_3(f_3 - f_L - f_C) = f^2 \ldots [15]$$

Referring to Fig. 8 the components of stress are

$$f_S = k_1Tr, \quad f_L = k_2p + 2k_1Mr, \quad f_C = k_2p\left(1 + \frac{b^2}{r^2}\right)$$

$$f_3 = k_2p\left(1 - \frac{b^2}{r^2}\right) \ldots\ldots\ldots\ldots [16]$$

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where $$k_1 = \frac{2}{\pi\,(b^4 - a^4)} \quad \text{and} \quad k_2 = \left(\frac{a^2}{b^2 - a^2}\right) \ldots\ldots [17]$$

Substituting values of the stress components from Equation [16] in Equation [15]

$$3k_2^2p^2\,\frac{b^4}{a^4} + 4k_1^2M^2r^2 + 3k_1^2r^2T^2 = f^2 \ldots\ldots [18]$$

The left side of Equation [18] is proportional to the distortion energy for an element at a distance r from the center. The critical element at which failure occurs will be for the maximum value

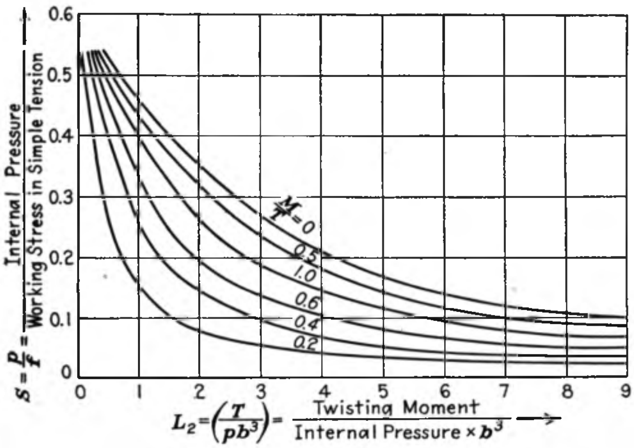

FIG. 9 VALUES OF LOAD RATIO S PLOTTED VERSUS LOAD RATIO L_2 FOR VARIOUS VALUES OF TWISTING MOMENT TO BENDING MOMENT AS A PARAMETER

of the distortion energy or for $r = a$. Then Equation [18] becomes

$$3k_2^2p^2\,\frac{b^4}{a^4} + 4k_1^2M^2b^2 + 3k_1^2b^2T^2 = f^2 \ldots\ldots [19]$$

Substituting values of k_1 and k_2 from Equation [17] in Equation [19]

$$\frac{3p^2}{(1 - R^2)^2} + \frac{4(4M^2 + 3T^2)}{\pi^2b^6(1 - R^4)^2} = f^2 \ldots\ldots\ldots [20]$$

where $R = \dfrac{a}{b}$.

Equation [20] gives a relation between the loads, dimensions, and the stress f in simple tension at failure. If the stress f is considered as the allowable stress, Equation [20] can be used for purposes of design. A design chart can be constructed by rewriting Equation [20] thus

$$\frac{3}{(1 - R^2)^2} + L_2^2\,\frac{(4L_1^2 + 3)}{(1 - R^4)^2} \times \frac{4}{\pi^2} = \frac{1}{S^2} \ldots\ldots [21]$$

where $$L_2 = \frac{T}{pb^3}, \quad L_1 = \frac{M}{T}, \quad \text{and } S = \frac{p}{f}$$

A plot of Equation [21] is shown in Fig. 9, in which values of the load ratio S are plotted versus the load ratio L_2 for various possible values of the twisting moment to the bending moment as a parameter. Fig. 9 is plotted for a value of $R = \frac{a}{b} = 0.25$. In this way the allowable loads can be determined for a given pipe or, by plotting Equation [21] for various values of R, the required size of piping can be selected.

ARTHUR MCCUTCHAN.[7] The authors of this paper have furnished a comprehensive analysis of stress problems involved in the design of pipe lines with a view to finding satisfactory bases for establishing allowable combined bending-plus-pressure stresses. Their lucid account of the factors involved should stimulate discussion of controversial issues and thus aid in reaching a consensus of opinion on how far a safety code such as the Code for Pressure Piping need go in setting limits on stresses other than girth stresses due to internal pressure.

The section on effect of "cold springing," or "prespringing" as the authors have termed it to distinguish from the similar effect resulting from "self-springing," is particularly valuable. Although those who have worked intimately with the subject of flexibility of piping fully recognize that a hot line will relieve itself of excessive bending stress through creep or yielding of the material, there appears to be a rather widespread misconception that creep tends to reduce rather than increase the initial cold spring usually provided in piping for high-temperature service.

The fact that an overstressed line at high temperature tends to relieve itself of any overburden of bending stress through creep or yielding of the component parts is responsible for the statement sometimes heard that: "The Lord takes care of flexibility experts." It is only when excessive bending moments result in leakage of bolted joints, or excessive thrusts or moments displace anchors or cause misalignment of equipment, that there is any untoward manifestation of lack of adequate flexibility. In recognition of the greater significance of stress related to high bending moments at bolted joints, the practice of the writer's company for the last 10 years has been to assign lower values of permissible stress at bolted joints than in curved pipe or straight pipe remote from joints.

In the case of creased bends, and corrugated bends and corrugated straight pipe, lower total combined pressure-plus-bending stresses also have been assigned, but for a different reason. The tests made by Lieutenant Robert Dennison (2), to which the authors refer, indicated rather high stress concentrations in such structures. Under cyclic stress, cracks eventually developed in the creases and corrugations at stresses corresponding to relatively small bending moments.

A study of the variable stress conditions existing in steam piping in central-station power plants, however, indicates that only in the case of a combination of a large constant bending stress and severe vibration of the line would it be possible to produce fatigue failure. Moreover, because of the self-unloading character of bending stress, arising from constraint of thermal elongation of a line, it appears doubtful if even this condition need be feared in practice.

For conditions encountered in power plants, the writer deduced the following stress-intensification factors from Lieutenant Dennison's tests (2) and Professor Moore's repeated pressure tests on boiler drums,[8] as applicable to creased bends and to corrugated pipe and bends:

Creased bends....................	2 to 2.5
Corrugated pipe and bends.........	2.5 to 3

These values for corrugated pipe and bends agree surprisingly well with the factor of 2.5 suggested by the authors for noncyclic loading. In the case of creased bends, the authors, in their effort to link such factors with flexibility factors, appear to have overlooked the fact that almost as great stress intensification was found in Lieutenant Dennison's tests of creased bends as in tests of corrugated bends.

[7] Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[8] "Tests of the Resistance to Repeated Pressure of Forged, Riveted, and Welded Boiler Shells," by H. F. Moore, Trans. A.S.M.E., vol. 53, 1931, paper FSP-53-6, pp. 55–60.

The satisfactory behavior of creased and corrugated pipe under actual conditions of use may be explained on the basis of the self-relieving characteristic of bending stress induced by constraint of thermal elongation. This unloading automatically reduces the bending stresses in the convolutions to those which the material will support at the operating temperature. Since the endurance limit holds up better than the yield strength and the creep strength as the temperature is raised, it is doubtful if a high-temperature line can support enough steady stress for the superposed vibratory stresses to cause fatigue failure.

Since the amount of the unloading will appear as an addition to the cold spring originally provided, rather severe stresses may exist in the convolutions of creased and corrugated pipe in the cold condition. These bending stresses emphasize the desirability of eliminating locked-up stresses by annealing such pieces after fabrication. The number of these major stress cycles, caused by change from a cold to a hot condition in a main steam line of a central station, ordinarily does not exceed a few hundred in the life of a plant. Consequently, failure under such conditions is related to a property of material termed by Professor H. F. Moore as "crackless plasticity."[9] Material used in fabricating creased and corrugated pipe usually is capable of considerable plastic deformation without developing cracks. It can be concluded therefore that the stress-intensification factors given by the authors, with the exception of those for creased bends, will insure proper application of such structures. The writer would suggest a factor of 2 for creased bends under noncyclic stresses and 4 for cyclic stresses.

In the writer's opinion, the maximum-shear theory (4) should be used in combining stresses resulting from effects of bending and pressure rather than the principal-stress theory advocated by the authors, since yielding, not eventual fracture, is the condition of so-called failure which can be anticipated in a structure loaded by constraint of linear expansion. The maximum-shear theory possesses the further advantage of extreme simplicity.

In the absence of torsion, the combination of bending and pressure stresses to obtain the tensile stress equivalent to the maximum-shear stress reduces to the following simple expressions:

1 For side of pipe in tension

$$Seq = S_b + S_l$$

2 For side of pipe in compression

$$Seq = S_b - S_l + S_t$$

in which

Seq = tensile stress equivalent to maximum shear stress
S_b = stress caused by bending moment, tension on one side of pipe, compression on the other
S_l = longitudinal stress due to internal pressure
S_t = transverse or hoop stress due to internal pressure.

Fine distinctions as to which theory to use in combining stresses are not justified, however, in analyzing structures in more than one plane, the flexural characteristics of which can hardly be determined within an assured accuracy of 10 per cent. Furthermore, commercial variations in pipe-wall thickness within the permissible limit of $12^1/_2$ per cent from nominal dimensions may affect the flexibility of the pipe line a corresponding amount. The writer would point out that the paper does not attempt to cover the method of computing expansion stresses, moments, and reactions, since there are a number of available methods which give comparable results, at least for simple structures. Some structures, however, are extremely difficult or impossible to compute by methods approaching rigorous accuracy, and square-corner approximations or other simplified assumptions often have to be resorted to. If too specific flexibility requirements are written into the code, the writer begs leave to ask who can determine whether they have been met in complicated structures? In the case of branch lines, three-ended structures, and the like, the most that can be determined is that certain stresses will not be exceeded.

The idea of establishing a permissible range of total combined stress to limit the stresses in both the hot and cold conditions has the advantage that the numerical value of such stresses can be made the same as the values of allowable stress given in the code for determining pipe-wall thickness and yet be high enough not to interfere with good design from a flexibility standpoint. The definitions of S_A and S_A' given in the paper seem inadequate, however, in that they fail to specify that these terms relate to bursting stress as distinguished from bending stress, or that the so-called allowable stress refers to total combined stress.

The proposed setup allows a conservative designer to ignore, if he so desires, the ameliorating effect of cold spring, whether actually present or not, but still permits him to use a total combined stress of not to exceed 75 per cent of the sum of the allowable bursting stresses at room and at operating temperatures. On the other hand, it allows more latitude in case of need or for those who want it. For instance, at 900 F using carbon-molybdenum pipe, the 75 per cent rule for no cold spring would allow a total combined stress of 15,750 lb per sq in., whereas the 100 per cent rule for 50 per cent cold spring would allow 21,000 lb per sq in.

It is possible that in lieu of these detailed stresses the simple requirement that a line should be designed so that it could be cold sprung 100 per cent of the computed linear expansion without developing stresses exceeding $^2/_3$, or some such fraction, of the minimum specified room-temperature yield strength of the pipe material would be a satisfactory code statement as far as any actual hazard is involved.

The writer's observation of the disturbing effect of bending moments on bolted-flanged joints is at variance with the reassuring conclusions reached by the authors which are to the effect that ASA standard flanges, at least where used with ring joints, can withstand any bending moments commonly encountered. Actual experience with 1000 F experimental flanged joints has indicated that even small bending moments can cause leakage of an otherwise satisfactory ring joint. Calculations also showed the desirability of using 1500-lb alloy-steel flanges on 865-lb per sq in. 910-F service, assuming a $^3/_8$-in-wide metal gasket, even where the bending moment was limited to that corresponding to a total combined stress of only 10,000 lb per sq in. Hence, it would seem desirable that the authors support their conclusions with adequate explanation of how they are arrived at.

Although there are a number of assumptions, regarding initial bolt loadings, necessary residual gasket compressions, and probable residual bending moments existing in steam lines after years of service, to which exception might be taken, the writer feels that a discussion of these points might tend to obscure the real purpose of this symposium, which, as he understands it, concerns how far a safety code need go in prescribing limits for total combined stress in piping systems, together with methods for combining such stresses.

P. E. Pendleton.[10] The paper is valuable in that it points out the controlling design features for high-pressure and high-temperature pipe lines. Among these are relaxation stress, a proper theory of failure, and the importance of high localized

[9] "An Investigation of the Fatigue of Metals—Series of 1925," by H. F. Moore and T. M. Jasper, University of Illinois, Engineering Experiment Station, Bulletin No. 152, 1925.

[10] Lieutenant-Commander, U. S. N., Bureau of Engineering, Navy Department, Washington, D. C.

stresses in systems subject to repeated loading, as well as their lack of importance under conditions of static loading.

However, no quantitative data are presented in regard to relaxation stress, i.e., at what temperature does it become the controlling factor for a specific material and what are the values of this stress at various temperatures. Accurate data on this question are difficult to obtain, although several laboratories in the United States are actively engaged in this problem.

The importance of keeping the range of stress of a piping system midway between the positive and negative yield strengths is timely. In this connection, it would be interesting to see a development made of the influence of locked-in stresses of a welded-pipe system and their effect in extending the stress range between cold and hot conditions.

At present, the Bureau of Engineering of the Navy Department calculates total combined or equivalent stress by the Hencky-Mises maximum-shear strain-energy theory, specifying a maximum of 15,000 lb per sq in. for this stress. A 50 per cent cold pull-up is required on all main steam lines, but no credit is given for this in calculating the stress in the pipe line when hot. It is thus unlikely that many regions exist in which any localized stresses greatly exceed the yield point of the pipe material. Consequently, there will be but little self-springing in pipe lines designed in this manner. On the other hand, the 50 per cent prespringing will subject the piping to a maximum negative, or cold stress of 8500 lb per sq in. Since the combined working stress of 15,000 lb per sq in. is less than the relaxation stress, the former will control the upper limit of the stress range. This range will then be 23,500 lb per sq in. (giving no credit for reduction in hot stresses) for Navy main steam piping, which is carbon steel for temperatures up to 650 F, and carbon-molybdenum, for temperatures ranging from 650 to 850 F. However, if credit be given for 50 per cent cold pull-up, the stress range will be 16,000 lb per sq in., which compares with a 19,200-lb per sq in. combined-stress range for carbon steel at 650 F as given in the paper. It would be interesting to know what upper limits of combined stress were used in compiling the allowable stress ranges given in Table 2.

Attention drawn by the authors to the limitations of the various theories of failure when applied to piping design suggests that further development of this subject is desirable, especially in the plastic range.

The use of simplified formulas for the determination of thrust and bending moment for particular pipe systems suggests that much could be done in this connection to facilitate design calculations.

The writer wishes to endorse the suggestion often made that the various research problems involved in high-pressure-temperature piping design be allocated by a combination of interested groups to laboratories and technical organizations best-fitted for their solution, so that duplication of effort may be avoided.

In the preparation of this discussion, the writer is indebted to R. Michel, marine engineer, Bureau of Engineering, Navy Department, for his able assistance.

E. L. Robinson.[11] During the last 10 years new installations of high-temperature steam piping have definitely gone beyond the temperature range throughout which elasticity is preserved, until at the present time there are approximately 2,000,000 kw of steam turbines in operation at temperatures where plastic deformation under stress definitely takes place. Furthermore, most new installations are being made at temperatures above that at which elasticity may be expected to persist without plastic effects.

Ten years ago the determination of the elastic reactions and the corresponding stresses due to thermal expansion constituted one of the principal problems in connection with the design of high-temperature piping installations. Most modern installations are being designed for operation at temperatures at which relaxation causes a decrease in elastic stresses as time goes on, rapidly at first and then more slowly. On this account, industry is confronted with problems of design for which there is little substantial background. The authors have made a valuable contribution toward the solution of these problems. They are careful to distinguish between yielding and rupture and, for the most part, their conclusions are so well qualified as to make it difficult to disagree with them.

In a few respects it is possible that the writer can contribute some useful observations.

The authors favor more than 50 per cent prespringing, but it does not seem to the writer that they emphasize the benefits of the greater prespringing sufficiently. The attitude is taken that, since relaxation results in self-springing, the long-time conditions will be the same. However, as they point out, this requires a greater amount of permanent set or creep in order to effect the relaxation.

The results of high-temperature rupture tests conducted during the last 2 years raise a very real question as to whether or not it is wise to ask too much of readjustment by means of creep. Creep, in itself, is beneficial, but when it leads to a brittle break, it does not seem wise to base design upon the expectation that it will occur without any deterioration of the material.

The authors do not discuss the invalidation of elastic analysis by the occurrence of plastic deformation. It is, perhaps, unwelcome to those who have devoted much high-grade analytical work to the determination of elastic stresses in high-temperature piping to realize that the reactions fall off and the stress distribution alters. On the other hand, it might be pointed out that, if the practice of prespringing to the extent of 100 per cent of the thermal expansion should be generally adopted, the elastic analysis would have reality when the piping is cold and strong and when the stresses are not combined with pressure. While at high temperature when the strength is reduced, existing stresses will be due to pressure only and not due to the reactions brought into play by the prevention of thermal expansion. The writer believes that prespringing should certainly be sufficient to assure extremely low rates of creep from the beginning.

Equation [9] of the paper is quoted from Moore and Kommers (7) to represent the fatigue strength where the stresses are not completely reversed. The tests given in the reference were sponsored by the writer's company and were first reported in bulletin form.[12] Originally, in presenting this formula Professor Moore made a number of qualifying remarks. In particular the formula fails as the stress ratio in the denominator approaches unity because the tensile strength is almost never 3 times the endurance limit for complete reversal, and a maximum stress beyond the tensile strength should not be implied as satisfactory. Where test results are not available, the writer would prefer a modified Goodman diagram based upon $^1/_2$ the tensile strength for complete reversal. However, an actual diagram of test results for the conditions of application would be better.

The authors repeatedly refer to the "relaxation limit." As a matter of fact there is no true relaxation limit any more than there is a true creep limit. It all depends upon the time. Thus at high temperature within a few hours all high stresses decrease rapidly to a level at which further decrease is slow. This is what is accomplished during a stress-relief anneal. On the other hand, if the same process is continued for 1000 hr, the gradual reduction of stress proceeds still further, and the residual stress is progres-

[11] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[12] "An Investigation of the Fatigue of Metals—Series of 1922," by H. F. Moore and T. M. Jasper, University of Illinois, Engineering Experiment Station, Bulletin No. 136, 1923.

sively lower after 10,000 hours and again after 100,000 hours.

The authors quote Bailey's conclusion that the relaxation limit for 100,000 hr corresponds to the stress causing 0.1 per cent creep in that period. The writer doubts the wisdom of such a generalization. It is much better to have a relaxation test than to try to determine bolt performance from a constant-stress test. Several simple methods of conducting relaxation tests, as reported by four different companies, were described in a report of a Subcommittee of the A.S.M.E.-A.S.T.M. Joint Research Committee on the Effect of Temperature on the Properties of Metals made in 1938.[13]

The writer's company has run hundreds of relaxation tests on all varieties of high-temperature steels and prefers such tests for the prediction of bolt performance and uses them instead of constant-stress tests for preliminary evaluations of material for high-temperature service. In many such tests, the residual stress at 10,000 hr corresponds closely to the constant stress causing 0.1 per cent per 100,000 hr, although in so many cases the residual stress is far below this level that it is unwise to take the correspondence for granted. Almost without exception, the residual stress for 100,000 hr is far below the stress corresponding to a creep rate of 0.1 per cent per 100,000 hr.

The authors refer to the "applied stress" for alloy-steel bolting. It is always difficult for the writer to visualize the applied stress in a bolt. It is easier to think of the bolt as having a measured extension when initially assembled, determined by the length of the bolt, the number of threads per inch, and the number of flats the nut is turned in tightening up. Technically, a "strain" is applied rather than a stress, and even this is not easily measured with soft gaskets. The authors suggest applying a stress to produce 1 per cent creep in 100,000 hr. The writer believes that an initial strain, corresponding to a considerably higher elastic stress, is desirable to assure tightness even though relaxation soon reduces it. Bolts may well be taken up 0.1 or 0.15 per cent. However, the effective stress remaining in the bolt to keep the joint tight after a period of time must be determined from the relaxation properties of the material. This will be far less than the elastic stress corresponding to the inital setup, although it is not quite independent of it. A bolt tightened 2 mils per in. initially will be measurably tighter after 10,000 hr than a similar bolt first tightened 1 mil per in.

N. O. Smith-Peterson.[14] The problems involved in such calculations as are carried out in this paper are not simple tension or bending-stress problems to which there can be only a right or a wrong answer. They are complicated and involve many assumptions as well as a choice of different formulas for their solution. In presenting the solutions of these problems in complete detail, with formulas, assumptions, and calculated results, it becomes possible for others to compare their methods with those of the authors. The writer has done so and has obtained results differing somewhat from those listed in Table 5, pertaining to flange stresses. The difference in amount is not nominal and is due to difference in choice of formulas and assumptions made. It is the stated aim of the authors in publishing this very thorough paper to develop just such difference in methods of calculation, and the writer is hereby presenting his methods as they apply to some of the problems.

The writer uses the same formula for flange stresses as the authors, namely, the Waters, Rossheim, Wesstrom, and Williams formula, presented to this society in 1937. The writer's reason for using this formula, among the many flange formulas in existence, is the fact that it is the only formula which permits calculating stresses in the three axial directions, viz., hub, radial, and tangential flange stresses. However, before being able to use these formulas, it is necessary to make assumptions concerning the end-force area, ring contact area (or gasket area), and residual-contact-pressure ratio.

The authors consider the end-force area to extend to the ring pitch diameter, the ring contact area to be equal to $^1/_4$ the annular-ring area, and the residual contact pressure to be $5^1/_2$ to 1.

The writer follows the A.S.M.E. Boiler Code regulation in considering the end-force area to extend to the outside diameter of the ring. The contact area is accurately calculated according to the sloping surfaces of the ring, and the residual contact pressure is taken to be 6 to 1.

This difference in calculations gives approximately 10 per cent higher flange stresses for a 24-in. 1500-lb flange than those listed by the authors in Table 5, items 14, 15, and 16.

These differences are largely in assumptions in that the same flange formulas are used. However, a formula difference also exists between the authors and the writer when it comes to calculating the transmission of pipe-line bending moments by a bolted flanged pipe joint.

The authors consider such bending moments to be transmitted directly to the flange bolts and, hence, use the section modulus of the bolt arrangement in the joint to obtain the total preloading bolt stress required for maintenance of joint tightness.

The writer believes that no bending-moment action can be transmitted to the flange bolts until the contact pressure on the ring has been completely absorbed (leak condition) and that, therefore, for a tight joint, where the existence of a residual contact pressure on the ring is required, the bending moment will affect directly the ring-contact pressure and not the bolt stress; hence, the ring section modulus is involved in obtaining the total preloading bolt stress.

In explanation of this latter theory, which does not take into account elastic effects, consider the universally accepted manner of calculating the required preloading of a ring in order to insure joint tightness when an end force is exerted upon it. The universally accepted way is to assume the end force, a tensile force, to be transmitted directly to the ring contact surfaces and there to reduce the existing contact pressure. The end force is not considered as being transmitted to the flange bolts and, therefore, does not cause any change in the bolt stress. Similarly, a bending moment, which on one side exerts a tensile force, should not be conceived of on that side as affecting the bolt stress, but should be conceived of as here reducing the contact pressure. On the other side, the compression side of the bending moment, the contact pressure is, of course, increased. Therefore, to calculate the amount of such decrease and increase in the contact pressure, the section modulus of the pressure area is required to be used and not the section modulus of the bolt arrangement. After calculating the contact-pressure changes due to a bending moment, the calculation of the required bolt stress is then undertaken.

The writer's method leads to an 18 per cent higher flange stress than those listed under items 17, 18, and 19 of Table 5, for a 24-in. 1500-lb flange.

These percentage differences in calculated stress results are not nominal, and it is to be noted further that the basic flange formulas used by the authors and the writer are the same. Were there any differences in the choice of these formulas, the percentage differences in calculated results would be greater yet. The situation, therefore, seems to call for the adoption, by code-making authorities, of flange formulas and calculating methods in general, as well as allowable stresses. The authors' paper makes this also evident throughout and is an excellent exposition of the whole pipe-line problem.

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[13] "The Resistance to Relaxation of Materials at High Temperatures," by E. L. Robinson, Trans. A.S.M.E., August, 1939, pp. 543–550.

[14] Walworth Company, New York, N. Y.

Authors' Closure

The authors acknowledge their debt of gratitude to the many discussers who, by their interesting comments, have materially added to the value of this paper. Considering the relative novelty of the subject dealt with, it is gratifying to note fair agreement on the basic conclusions and, consequently, the closure can be limited to the clarification of certain points brought out in the discussions.

While the discussion of the theory of calculating a pipe line for flexibility has purposely been excluded from this paper, Mr. de Jonge's reminder that much is still to be learned on this subject is not untimely. However, the assumptions usually made are on the side of safety and the available methods of calculation, if applied by competent engineers, are entirely satisfactory for engineering use.

The main issue, that of allowable stresses for pipe-bend calculations, has received wide comment. Professor Marin has contributed a study of the triaxial state of stress and its implications. The Hencky-Mises theory is advocated as a stress criterion by Professor Hovgaard, Lieut-Com. Pendleton, and, in the verbal discussion, by Mr. Soderberg, whereas Mr. McCutchan favors the maximum-shear theory. While it is conceded that a matter of principle is involved, either failure criterion appears adequate considering the present undeveloped state of our knowledge. As pointed out by Mr. Robinson, the authors have avoided more than passing comment on the invalidation of the elastic analysis by the occurrence of plastic deformation. They felt justified in slighting this aspect by the fear of confusing the issue which is to define workable rules for evaluating the flexibility of piping. Again, the redistribution of stresses due to creep and the phenomenon of relaxation are in favor of the structure, and an elastic analysis gives the limits of stress through which the material goes in each cycle of change from the elastic to the semi-plastic and ultimately, in extreme cases only, the fully plastic state.

On the other hand, the authors gladly comply with Mr. Robinson's request to lend added emphasis to the injunction to use prespringing in excess of 50 per cent. Appreciation of the benefits obtained from cutting a line short is growing steadily, and we would assume that this will soon be general practice. The authors have used the following rough rule as a guide in setting the desirable amount of prespring s:

$$\frac{S_A/E}{S_A/E + S'_A/E'} > s < \frac{2S_A/E}{2S_A/E + S'_A/E'}$$

In order to understand the significance of this formula, it should be remembered that S_A, the bursting stress specified in the codes, corresponds to 40 to 50 per cent of the yield point cold; and S'_A either corresponds to the same percentage of the hot yield point or to from 80 to 100 per cent of the stress producing 1 per cent creep in 100,000 hr, which can be considered the virtual relaxation limit (in the same sense as Lieut. Dennison defines the "virtual" endurance limit). E and E' are the cold- and hot-elasticity moduli, respectively. Since the allowable stress range for a prespring s in excess of 0.5 is $S_A + S'_A$, the lower limit of s in the formula gives a stress of S_A cold and S'_A hot, or a range between, roughly, one half the cold yield point and one half the hot yield point or, for elevated temperatures, the relaxation limit; this state of stress will also be produced in an unsprung bend (at temperatures involving creep) after a period of operation producing complete relaxation. The upper limit of s, which should be approached in particular for high-temperature applications, places the average stress midway between the yield point at room temperature and the virtual relaxation limit, thus reducing the operating stress. The formula gives presprings between 50 and close to 100 per cent, depending upon the material strengths at room and operating temperatures.

Mr. McCutchan favors reduced stresses at bolted joints. While the authors agree that bolted joints should receive special attention, their experience is that a reduction in allowable stress is not generally necessary with ring-type-gasket joints. Actually, the stress specified at the bolted joint would govern the entire piping design, since maximum moments usually occur at the connections to equipment where bolted joints are normally located. Mr. Smith-Peterson's comments on the detail calculations of the flanges have been found interesting, and the authors agree that it would have been more proper to base their investigations of the effect of bending moments on the ring section modulus in place of the section modulus of the bolting.

Properties and Performance of Plastic Bearing Materials

By L. M. TICHVINSKY,[1] EAST PITTSBURGH, PA.

Bearings made of plastic materials can be used successfully not only for the case of perfect fluid lubrication but also for that of semifluid lubrication. Certain additions, such as graphite, will sometimes permit the application of these materials under conditions of dry friction.

By virtue of good physical properties these bearings find a wide application. Heavy-duty plastic bearings are used in the steel-mill industry. Lubricated and cooled with water, they carry heavy loads at pressures of several thousand pounds per square inch. As guide bearings their performance ranges from small, high-speed spindles, to large ship-propeller shafts. Oil-, water-, and grease-lubricated plastic bearings are used extensively in industrial, marine, and farm machinery.

There are many differences in the behavior of plastic and metal bearing materials by virtue of which the performance is also different. This article intends to point out the most important physical properties, as well as some of the characteristic performances of plastic bearing materials.

THERE are many reasons for the increase in the use and application of various kinds of bearing materials bonded with synthetic resin. The industry in this country became interested in these materials mainly because of some qualities superior for certain applications to those found in bearing metals and alloys. An extensive development and application of plastic bearing materials abroad was and is due primarily to the lack of basic bearing metals.

This article describes the bearing materials bonded with synthetic resin. Rather complete data are given on the laminated materials, including physical properties as well as the results of various bearing performance tests. This information was gathered from various domestic and foreign technical and scientific publications, as well as from testing in the author's laboratory.

Types of Synthetic Resin-Bonded Bearing Materials

All bearing materials bonded with synthetic resin can be divided into three types in accordance with their internal structure.

1 The bearings of the first type are made from various plain or graphitic molding powders. Numerous kinds of bearings for light and medium load applications are made from these materials. Guide bushings for high-speed spindles, or gland materials for sealing purposes, are typical examples of the use of this material; these bearings perform under all types of friction, namely, dry, boundary, and fluid. The application of these materials is rather limited because, as they are very brittle and fragile, impacts of even small magnitude may cause failure. Those made of graphite powder have a rather high heat-transfer coefficient.

2 The second type comprises the most important and widely used bearings. Such bearing material is obtained by molding at high pressure and temperature sheets of woven textile fabric treated with an organic binder. Synthetic resin of the phenol-formaldehyde type is usually used in the manufacture of laminated bearing materials. The molding is done in hydraulic presses (pressure varies from 1200 up to 2200 psi) at temperatures ranging from 120 to 180 C (1).[2] Under the combined action of heat and pressure the resin softens or melts and undergoes a further chemical reaction, resulting in a material which no longer is fusible. By this action the filler (cuttings, laminations, and the like) is permanently bound. The desired thickness for the final product determines the duration of this operation. It may last from 10 minutes to 30 hours.

Laminated bearing material is strong and tough. Bearings made of such materials perform efficiently when lubricated with water in which case the latter serves as lubricant and as a cooling agent. Almost any liquid (except strong alkalies) can be used, and although it might be recommended sometimes to add a lubricant, e.g., tallow or suitable emulsion, to the water, these bearings do not require oil or grease (2).

3 Bearing material of the third type is based on an internal felt-like structure. It is obtained by impregnating a felt of fibrous fillers consisting of cellulose fluff, linters, and similar materials. The resin is generally precipitated from a thin aqueous solution (sodium hydroxide) on and in the felt-forming fibers.

These materials range in their physical properties between the brittle powder moldings and the extremely tough and strong laminated materials.

Properties of Synthetic Resin-Bonded Materials

Laminated materials are the most important in bearing applications as already mentioned, therefore most of the information pertaining to the physical properties will be given in connection with these materials.

(*A*) *Modulus of Elasticity.* The modulus of elasticity, or Young's modulus, gives the relationship between stress and strain in the elastic region. It defines the amount of strain under load and, therefore, enables one to determine the deflection.

In connection with the materials in question, for a long time the value of the modulus of elasticity was found to be equal to 1.0×10^6 psi (2). However, methods were devised through which the use of artificial silk, cotton, and many other fibers resulted in the increase of the value of the modulus of elasticity up to 1.8×10^6 psi (3).

Many of the synthetic bonded materials were developed in connection with their extensive use by the electrical industry. Through a change in the molding pressure alone it is possible to attain a considerable increase in the modulus of elasticity. This is clearly shown in Table 1 (4).

TABLE 1

Molding pressure, psi	2300	4600	9200
Modulus in tension, psi	0.72×10^6	0.85×10^6	1.03×10^6
Modulus in bending, psi	0.78×10^6	0.91×10^6	1.14×10^6

According to Riechers (4) the values of the modulus of elasticity

[1] Research Laboratories, Westinghouse Electric & Manufacturing Co.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

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

for molded resinoid materials with various fillers are given in Table 2.

TABLE 2

Filler	Modulus, psi
Wood flour.................	$(0.8-1.17)\times 10^6$
Organic textile..............	$(1.32-2.35)\times 10^6$
Organic filler...............	$(1.32-2.2\ )\times 10^6$

In the case of Micarta[3] tubing where the fabric during rolling is maintained in tension, the modulus of elasticity reaches a value of about 3.0×10^6 psi.

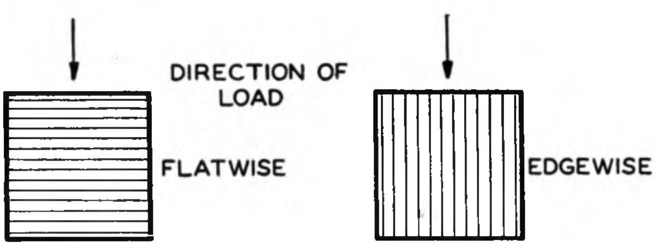
(*B*) *Compressive Strength.* For bearing application the compressive strength of these materials is important.

Compressive strength, as well as such properties as ultimate tensile strength and shear strength, can be increased by improved fabric filler as can be seen from Table 3 (3).

TABLE 3

Lb per sq in.	Ordinary fabric-filled phenol resin	Improved fabric filler
Compressive strength, psi....	25,000	27,000
Ultimate tensile strength, psi.	11,000	27,000
Shear strength, psi..........	4000–7000	6000–7000

[The compressive strength is usually given for laminated materials in two directions: right angle to laminae and parallel to laminae, as illustrated by the accompanying sketch.]



In Table 4 data on the compressive strength are given.

TABLE 4 COMPRESSIVE STRENGTH OF LAMINATED MATERIALS

Flatwise, psi	Edgewise, psi	According to
40–45,000	20–24,000	Eyssen (2)
45–50,000	30–35,000	Rochester (5)
30–45,000	......	Smyth (6)

(*C*) *Heat Conductivity.* In this property all synthetic resin-bonded materials are inferior to white bearing metals. However, some improvements in this property have to be mentioned, especially those achieved during the last year. About a year ago one could say roughly that bearing plastic materials showed a heat-conductivity coefficient which is 100 times smaller than that of a tin-base white metal (0.845 w per sq in. per deg C per in. for babbitt). By reason of improvements made it can now be said that this ratio has decreased about one half.

The thermal conductivity of ordinary Micarta is taken as equal to 0.007 w per sq in. per deg C per in. Micarta for bearing application (I and II) has higher thermal conductivity as shown in Table 5.

TABLE 5

Bearing Micarta I, w per sq in. per deg C per in...	0.0099 –0.01
Bearing Micarta II, w per sq in. per deg C per in..	0.00925–0.01

Eyssen's (2) published data vary between 0.0045 and 0.0059 w per sq in. per deg C per in. for laminated products, and between 0.0077 and 0.0089 w per sq in. per deg C per in. for moldable materials based on felts of impregnated nonoriented fibers.

According to Richardson (7) the thermal conductivity of phenolic-plastic bearings is approximately 0.0072 w per sq in. per deg C per in.

Erk (8) gives a table of the thermal conductivity in a number of the "Kunstharz." In the phenol group there are some with thermal conductivity as high as 0.020, 0.021, and 0.023 w per sq in. per deg C per in.

In an endeavor to increase the thermal conductivity of Micarta and at the same time to improve friction qualities, a new method was developed for the manufacture of graphited Micarta. The thermal conductivity of this material was found to be equal to 0.015 w per sq in. per deg C per in. The heat conductivity of synthetic resin-bonded materials will never be of the same order as that for white bearing alloys. Certain developments in the methods of manufacture may still improve this property without impeding the others.

(*D*) *Other Properties.* The coefficient of expansion for the laminated product consisting of sheets of heavy square-woven duck bonded together by means of a synthetic resin of the phenol-formaldehyde type is 0.00002 per deg C per in. (6).

The specific gravity of the synthetic resin-bonded materials varies between 1.25–1.40. For general calculation it can be assumed that 1 cu in. weighs 0.05 lb. Oil absorption is negligible for practical purposes while water absorption was found to vary from 0.5–0.8 per cent when a test piece ($2 \times 2 \times {}^1/_2$ in.) was submerged in water for 24 hr (2).

The Brinell hardness of synthetic resin-bonded materials was determined by Rochester (5) as 30–42, by Smyth (6) as 35–42.

Performance of Synthetic Resin-Bonded Bearings

The performance of all synthetic resin-bonded bearings is limited, as in the case of bearing metals and alloys, by the maximum allowable temperature of the bearing materials. The heat conductivity of the synthetic resin-bonded materials is lower than that of bearing metals. Therefore, such efficient cooling is required by which maximum amount of heat generated by friction can be removed by the circulating lubricant. An example of this is shown in Fig. 1. Such a bearing being lubricated and cooled with water performs at low speeds and medium loads. The sketch shows the arrangement of cooling and also the location of the high-pressure water inlet. This bearing is not operating in the boundary region at the starting and stopping periods because the high-pressure pump forces water into the bearing clearance.

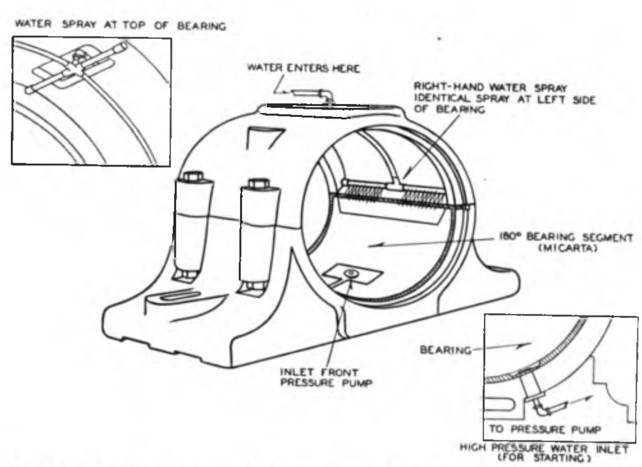


Fig. 1 Typical Water-Lubricated Micarta Bearing for Ball and Rod Mills

[3] "Micarta" is a trade name for a plastic material. Other materials of the same general class are Textolite, Formica, etc.

TABLE 6 FINAL BEARING TEMPERATURE C AND COEFFICIENT OF FRICTION MEASURED AFTER 2-HR-20-MIN RUNS

(According to Heidebroek)

Clearance ratio, in. per in.	Bearing pressure = 22 psi —Peripheral velocity, fps—				Bearing pressure = 155 psi —Peripheral velocity, fps—			
	19.1	32.8	41.3	57.1	19.1	32.8	41.3	57.1
0.00153	55.0	65.0	71.0	....	55.0	67.0	75.0	....
	0.0183	0.0197	0.0206	....	0.0049	0.0049	0.0050	....
0.00175	51.5	60.5	66.5	78.0	55.0	65.5	74.5	85.0
	0.0206	0.0221	0.023	0.0226	0.0051	0.0051	0.0051	0.0051
0.0020	49.5	60.5	66.5	80.5	52.0	63.0	72.0	83.0
	0.0247	0.0267	0.0263	0.0278	0.0058	0.0060	0.0061	0.0062

The softening of the synthetic resin-bonded materials varies with type and grade of the bond. In this respect it is of interest to note some remarks by British authors pertaining to working temperatures of bearings. Smyth (6) indicated a safe working temperature of 100 C and an actual temperature of 150 C at which the material is unaffected. Rochester (5) recommends a maximum working temperature of 60–80 C. He mentions that the maximum allowable running temperature for short periods might reach 250 C. It is self-evident that high, localized temperatures might reach rather high values, not only in the region of dry or boundary friction, but even during perfect fluid lubrication (9). In Table 6 data are given for a series of tests (10) during which the final bearing temperature and the values of the coefficient of friction were measured as a function of peripheral velocity, clearance ratio, and pressures of 22 and 155 psi. The

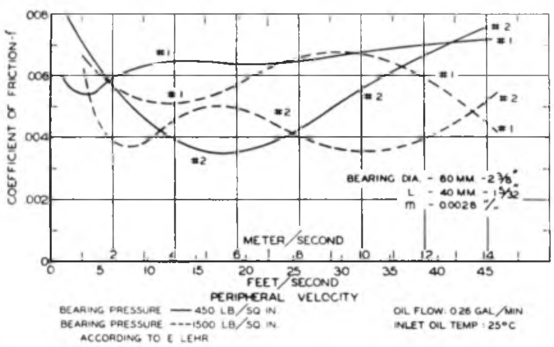


FIG. 2 COEFFICIENT OF FRICTION AS FUNCTION OF PERIPHERAL VELOCITY AT 450 AND 1500 PSI PRESSURE

bearing material was made of resin-impregnated cotton-cloth cuttings. A flow of light turbine oil varied between 0.238–0.715 gpm. The amount of circulating oil showed a tendency to increase with increasing clearance. Results of these tests indicate that the lower the bearing clearance ratio the higher is the running temperature, and the lower is the value of the coefficient of friction. For this reason it is advisable to provide the bearings with chamfered grooves, so that a large amount of lubricant may be admitted not only for lubricating purposes, but also for more efficient cooling. Another investigation was carried on during which the performance of various bearing materials was studied on a specially built test machine (11). The values of the coefficient of friction of two materials are plotted on Fig. 2 as a function of peripheral velocity for pressures of 450 and 1500 psi. The bearing bushings were $3^3/_8$ in. diam, $1^5/_{32}$ in. long, and were machined with a 0.0028 in. per in. clearance ratio. A light oil having 80 Saybolt sec at 100 F and 49 Saybolt sec at 150 F was fed into the bearing housing at 25 C and a rate of 0.26 gpm. The

TABLE 7

Material	Kind of resin	Resin, per cent	Structure	Molding time, min	Molding temp, C	Additional treatment
No. 1	Phenol	50	Cloth cuttings	45	165	10 hr at 80 C
No. 2	Phenol	45	Cloth cuttings	20	165	8 hr at 125 C

two materials tested showed approximately the same performance, the values of the coefficient of friction being slightly higher for material No. 1 than for material No. 2. In Table 7 some of the characteristics of these materials are given.

The performance of wound bushings impregnated by cresol is given in Figs. 3 and 4. The variation of the coefficient of friction with bearing

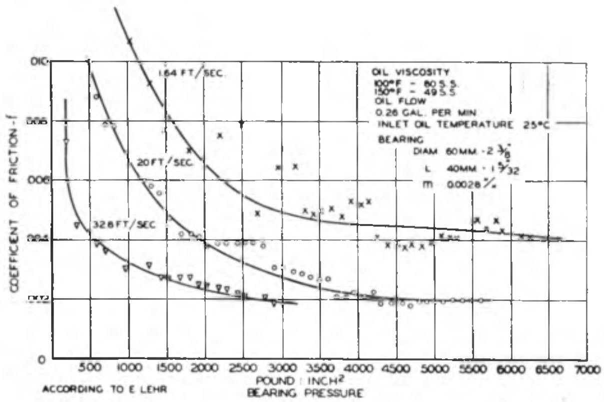


FIG. 3 VARIATION OF THE COEFFICIENT OF FRICTION UPON THE BEARING PRESSURE AND PERIPHERAL VELOCITY

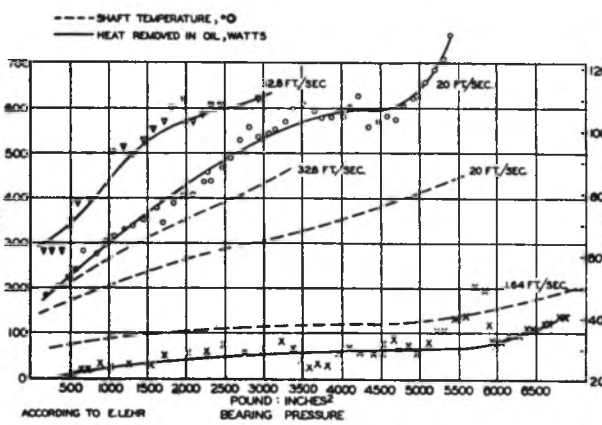


FIG. 4 VARIATION OF HEAT LOSSES AND SHAFT TEMPERATURE UPON THE BEARING PRESSURE AND PERIPHERAL VELOCITY

pressure and peripheral velocity is given in Fig. 3, while the variation of the heat losses and shaft temperature with pressure and velocity is given in Fig. 4.

These bearing tests were made at the same conditions as the tests of bearing materials given in Table 2. From Fig. 3 it is seen that the value of the coefficient of friction shows less variation at high loads at which it approaches some constant value. By diminishing the load, the values of the coefficient of friction rise rapidly. In the region of maximum pressures around 6000 psi the values of the friction coefficient lie between 0.002 and 0.005. More heat is being removed in the oil at higher peripheral velocities than at the lower, as can be seen in Fig. 4. It can also be noted that the slope of the shaft temperature is about the same as that of the heat losses. These shaft temperatures were recorded at steady-state conditions by means of a thermometer inserted in the journal hole specially drilled for this measurement.

All bearing surfaces must be well run in before optimum friction conditions are established. This is especially true for the case of bearings made of plastic material, where the running in must be done carefully and skillfully. It is found (1) that the best scheme for running in plastic bearings consists in loading the bearing gradually at low speeds until full load is reached and then bringing the journal by steps to full speed. If the load be applied too quickly the excessive heat generated on the sliding surfaces might cause seizure. By frequent loading and unloading at various speeds the plastic bearings finally become insensitive to pressure increase as they become better fitted to the journal.

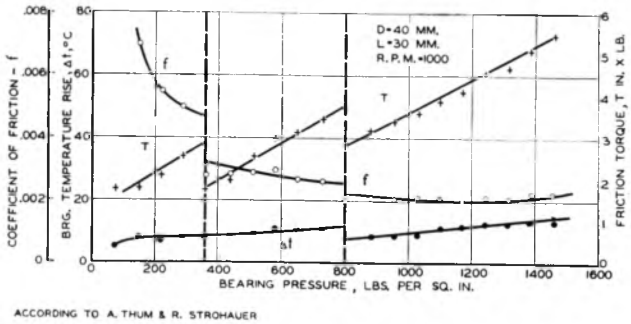


Fig. 5 Running-In Performance of a Laminated Bearing

The bearing can be considered well run in when the originally light-colored and finished areas disappear and form a light-colored, mirror-like, uniform surface. The bearing friction and temperature, which at the beginning were high, decrease considerably at the end of the running-in period. This period usually requires several days and can be substantially reduced by good machining of the sliding surfaces. Such running-in performance is very well represented graphically in Fig. 5. A bearing with textile laminae perpendicular to the axis, impregnated with cresol, was run in on a testing machine with forced-feed oil lubrication during 110 hr. The bearing pressure was increased in steps of 75 psi after steady-state conditions were reached. The friction torque and temperature increased gradually with the loading until at a pressure of 370 psi they rose suddenly and the journal speed dropped rapidly. After a few minutes, however, the torque and temperature readings decreased and their values slowly reached lower readings than previously recorded. By a further similar loading procedure the torque and temperature rose again as may be noticed in Fig. 5. At a pressure of 800 psi the torque and temperature reading went again off scale and the journal slowed down. After a few minutes the torque and temperature decreased again and rose slowly with increased load. The variation of the coefficient of friction may be seen also on the curve.

An article published recently on the performance of railway bearings (12) also points to the importance of running-in performance. This is done in connection with laminated textile material impregnated with cresol resin (65 per cent resin content). This running-in period is artificially accelerated by removing the upper layer of the bearing surface and polishing it by means of a special bushing having the same diameter as the bearing journal. As shown in Fig. 6, such bearing before the surface layer is removed performs with a value of the coefficient of friction which is in the boundary region. At a pressure of slightly over 400 psi the bearing seized. However, the same bearing after the upper bearing surface was removed and polished performed satisfactorily at much higher loads in the region of fluid film lubrication. These tests were carried on several bearings at peripheral velocities of 1.6 and 3.2 fps. Bearing temperatures are also plotted in Fig. 6. Optical measurements of bearing surface finish were made (13) on new and worn bearings; this, it is hoped, will help in further study and improvement of the performance of synthetic resin-bonded materials made for bearings. This is especially so since finish measurements of journal surfaces are now made (14, 15) and the performance of bearings can be elucidated also from this important point of view.

Fig. 7 indicates that the performance of a grease-lubricated Micarta bearing is approximately identical with that of an oil-lubricated babbitt bearing, both in the boundary region. These tests were made at various pressures and peripheral speeds while using a $2^1/_2$-in-diam bearing bushing 2 in. long. Fig. 8 shows the results when another type of Micarta bearing material lubricated with grease was tested at various loads and speeds. The friction properties of synthetic resin-bonded materials can be improved by various additions. The variation of the coef-

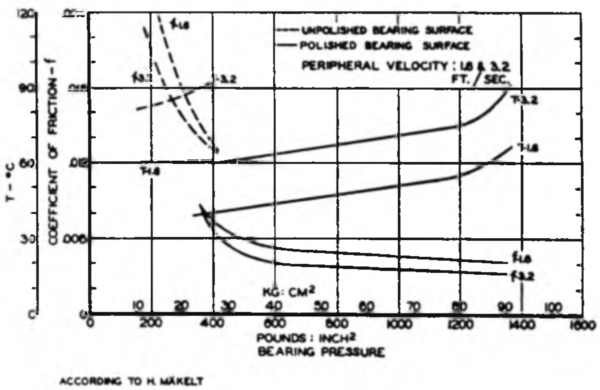


Fig. 6 Bearing Performance of Railway Bearings
(Material: laminated textile sheets impregnated with cresol 65 per cent resin.)

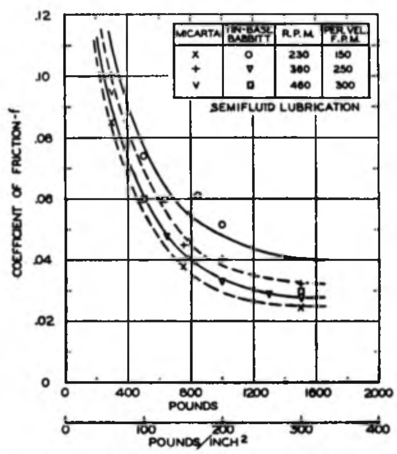


Fig. 7 Comparison of Friction of Grease-Lubricated Micarta and Oil-Lubricated Tin-Base Babbitt Bearings

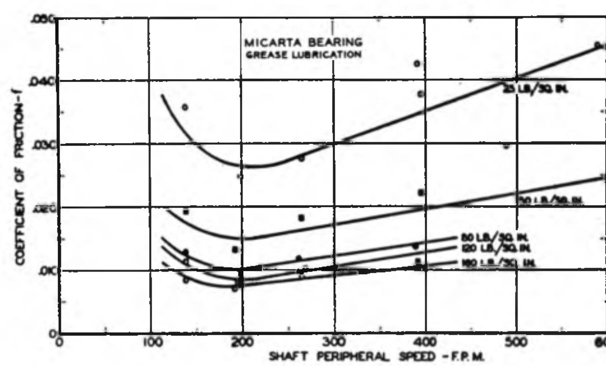


Fig. 8 Variation of the Coefficient of Friction With Peripheral Velocity and Pressure

ficient of friction depending upon such additions is shown in Fig. 9. These experiments were made on the machine previously described by the author (14). All bearings tested were run in for a few hours with a small amount of lubricant. Its supply was then shut off and the values of the coefficient of friction and of the temperature were recorded for different pressures. The standard Micarta, as can be seen, performs with a higher friction and temperature. Additions of mica and talc decrease both the friction and the temperature. A marked improvement is obtained when hard graphite is added to Micarta; the values of the coefficient of friction are low and so is the temperature, due to the improved heat conductivity as mentioned previously.

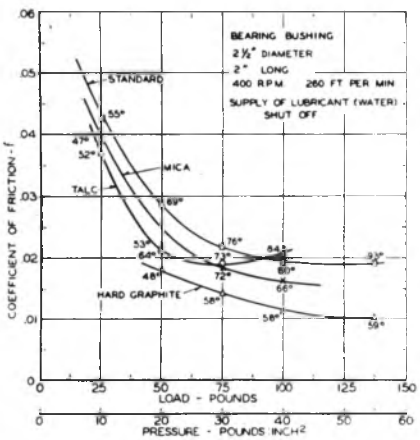


FIG. 9 INFLUENCE OF ADDITIONS TO SYNTHETIC-RESIN BEARING MATERIALS UPON THE VALUE OF THE COEFFICIENT OF FRICTION

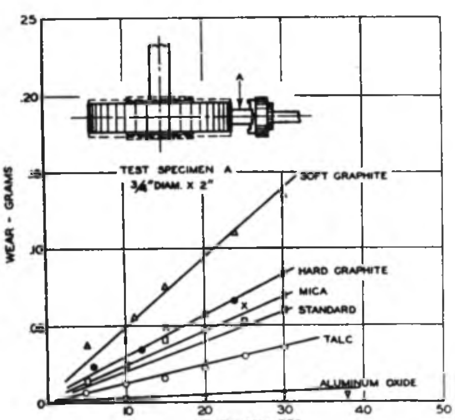


FIG. 10 INFLUENCE OF ADDITIONS UPON THE WEAR

The wear properties of the same materials were also investigated on the wear-testing machine previously described by the author (16). The principle of the machine is sketched in Fig. 10. The rotating test specimen *A* is pressed on the rotating and oscillating outside surface of a hard disk. A small jet of water is played upon the disk for removing the heat. The loss in weight of the specimen gives a comparative measure of wear. The addition of aluminum oxide greatly improves the wear qualities of Micarta. The rate of wear of the standard Micarta is approximately the same when hard graphite and mica are added to it.

The foregoing data and several performance examples, it is hoped, will assist a designer in his work when selecting and specifying synthetic resin-bonded bearings. Being made of a comparatively new material, the plastic bearings now find more and more application. The continuous development of the material itself and studies of its performance which are carried on by a combined force of chemists, physicists, and engineers will result undoubtedly in many more improvements and in wider application.

ACKNOWLEDGMENT

The writer thanks the Westinghouse Electric and Manufacturing Co. for permission to publish this paper. All the data of experiments performed by the author were made with the assistance of E. G. Fischer, of the Westinghouse Research Laboratories, and C. Jahing, graduate student.

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Discussion

CARL CLAUS.[4] The author mentions that appreciation of plastic bearing materials is not caused by lack of basic bearing materials but by advantages in certain applications. These seem to be confined to installations where it is possible to carry off frictional heat by water or oil which act as coolants and lubricants. There is little objection to this method in rolling mills where operators have long been accustomed to these conditions.

The author states that bearings made from graphitic molding powders show a better heat conductivity, but that they are ex-

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tremely brittle. A familiar bearing used for several decades consists of a metal shell with graphite inserts. This combination of the conventional metallic-sleeve bearing with the plastic graphite dissipates the heat satisfactorily without a cooling liquid. The metallic structure gives proper support to the graphite. A well-known application is the trolley bushing of streetcars for which no better bushing has as yet been found. The constant jars of the trolley pole do not loosen a well-bonded plastic graphite in this and other applications.

Another bearing somewhat related to plastic bearings is the porous-metal bearing. It is manufactured by the cold pressing of metal powders. Instead of using a resinous binder it is sintered and obtains a strength and ductility not approached by the resin-bonded bearing. Its heat conductivity is excellent and its porosity is used for storing lubricant in its structure.

By using hard graphite in the author's tests with Micarta a "marked improvement" was found in reducing the coefficient of friction and the operating temperature. It would be interesting if the terms "hard graphite" and "soft graphite" were explained, as well as the effect of the different kinds of graphite. Fig. 10 of the paper shows more wear for soft graphite than it does for hard graphite.

The condition of the shaft before and after the test is also of interest. Graphite has a tendency to fill up depressions of the shaft surface and make it smoother. Unless a pure graphite is used it may also act as a polishing agent which will rub off high spots.

Aluminum oxide is reported to have improved the wear qualities of Micarta in a test shown in the testing machine, Fig. 10. This material is an abrasive which may have smoothed the contacting surfaces. It was apparently firmly imbedded in the molded test specimen and may have acted as a lapping compound, which wears down the shaft or disk rather than being worn down by them.

An interesting observation can occasionally be made on oil-impregnated wooden bearings which may wear down a steel shaft when abrasive dust embeds itself into the wood. As a result of such a condition the soft-wood bearing shows no wear but the shaft does.

In conducting the tests the author has found it necessary to run in the bearing surfaces carefully for hours. The designing engineer quite naturally wonders what may happen to a bearing which he specifies and which will assuredly not receive such tender care. Since two bearings are mostly used to support a shaft, the question of misalignment will also have to be considered. Misalignment can easily result in failure at the start when excessive pressure prevails over a small area.

The difficulty with testing machines for bearings seems to be that they do not duplicate operating conditions. Results obtained from them cannot be applied to cases which vary from those of the individual testing machine itself. In the ideal sleeve bearing the load is supposed to be carried by an oil film, and a wearing down of bearing material and shaft should be impossible without contact between them.

As a result of the limitations of the bearing testing machines, most engineers reluctantly turn to comparative tests of bearing materials in commercial applications such as electric motors, automobile engines and their accessories, etc.

Performance tests under operating conditions rather than accelerated breakdown tests give us the only reliable answers to our bearing problems. Even then results are greatly influenced by numerous conditions outside of the bearing material itself.

The hardness and finish of the shaft, the degree of misalignment, and particularly the lubricant between the shaft and the bearing are often of even greater importance than the bearing material itself.

H. A. Everett.[5] It is an axiom in research work that wherever possible the actual points for observed data be shown when curves are presented. The author has been meticulous in this respect in all of the illustrations in his paper except Fig. 2. Will he be kind enough to let us have the observed data points for that figure, as there seems to be a surprising and gregarious change in the coefficient of friction with increasing speed.

E. F. Heer.[6] Limited investigations have been conducted on the graphites used in bearings. While experience thus far has not been sufficiently comprehensive to arrive at definite conclusions, the fact is definitely indicated that for each class of bearing there probably is at least one type out of the several varieties of graphite which is superior to others.

Following the author's division of the subject we shall first discuss bearings made from various plain or graphitic molding powders. Since it is the purpose of the graphite to supply a lubrication factor, it is important to have a complete distribution of the graphite throughout the molding powder in the ratio desired in order to supply a constant coefficient of friction. At the same time, it is equally important to make sure that the structural strength of the bearing is not weakened. This immediately brings up the consideration of—What effect, if any, does the particle size of the graphite have on the performance of the bearing in service and does the particle size influence the internal strength of the bearing? Further consideration must also be given the question as to whether or not the particle size of the graphite influences the molding characteristics of the bearing. What effect does loading have on the finished product?

It is conceivable that the internal strength of a plastic will be affected by the relation of binder to inert material. If such is the case, the embrittlement mentioned by the author may be attributed to overloading with graphite or other inerts (by inerts is meant any nonplastic ingredient).

It would seem to be equally important to study the performance of bearings formed from resin in which dry flake graphite has been used as opposed to bearings formed from resins which have been treated with colloidal graphite in either an aqueous or organic carrier.

As mentioned, some experience along these lines has already been had, sufficient to indicate that there is a relation between the particle size of the molding powder and the particle size of the graphite to be used, if admixture of the graphite and resulting friction properties are to be constant.

A study of the particle size of the molding powder examined under a low-power microscope indicated that the small particles are rod-like crystals about 20 mu in diam and from 80 to 100 mu in length. These are held together in agglomerates with a binder making up particles which range in size up to $^1/_8$ in. diam. It is the writer's opinion that, with molding-powder particles of this nature, it is almost imperative to have the graphite particles of suitable size to maintain proportionate distribution of graphite to produce a homogeneous finished product. If large particles of dry powdered graphite surround the finer particles of molding powder, the proportion of graphite to molding powder will probably be greater than is desired and a weakness in structural strength would probably result. On the other hand, if the largest of the particles of molding powder are to bear their complement of graphite in the form of a coating, it becomes necessary to use a form of graphite other than a dry powder, for obviously it is most difficult to maintain the proper proportion of dry powdered graph-

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ite adjacent to and in contact with these large particles of molding powder. Experience thus far has indicated that graphite, the range of particle size of which lies well below the ordinary fine-mesh powders, when dispersed in a suitable organic fluid, will give much better distribution of the graphite throughout the mass of molding powder and will enable the graphite to be attached to all resin particles regardless of size of the latter. The organic solvent can be disposed of by evaporation which leaves the molding powder in discrete particles not cemented together but at the same time properly coated with graphite.

With respect to the second and third types of bearings mentioned by the author, namely, the laminated type, we would again submit for consideration the selection of the graphite to be used for impregnating the textiles and the methods which should be adopted. A study should be made of the effect of overloading the textile fabrics with graphite as well as a study of impregnating the fabrics with a colloidal dispersion of graphite as compared with fabric which has been treated with powdered graphite. These studies, of course, would not be complete without a further study concerning the variations produced by treating the fabrics with aqueous and organic liquid dispersions of graphite.

Much that has been said with respect to molding powders of course applies to impregnating fabrics, inasmuch as particle size is of prime importance; also distribution of the graphite throughout the fabric to supply properties which will be constant. In dealing with fabrics, as is the case with molding powders, the wetting properties of the graphite dispersions are also of great importance. Some experience has already been had in connection with fabrics. It has been found that dispersions of graphite in aqueous as well as organic liquid dispersions are quite useful. Furthermore, a selection of particle size is important because there must be the proper relation between the particle size and the fibers used in the fabric, also the coarseness of the mesh of the woven fabric. The limited experience thus far indicates that, among the methods which can be used to impregnate the textiles with graphite, the one which appears to hold greatest promise is the immersion of the fabric in a dispersion of graphite in water or suitable organic solvent. The content can be closely controlled by using colloidal dispersions of graphite in these fluids for submerging the fabric for a given length of time. The graphite content together with the time involved in keeping the fabric submerged contributes to a homogeneous result and the proportion of graphite to the fabric can thereby be controlled and, of course, varied according to requirements.

From the paper it may be concluded that the initial run-in period is a predominating factor in the life of the bearing. If that is the case, it is apparent that a study should be made of the behavior of such bearings where water is used as the lubricant. Such study should determine the effect or what can be accomplished by introducing an aqueous lubricant, consisting of a colloidal graphite dispersed in water, into a circulating system for delivery to the bearing during this run-in period. This technique, in the opinion of the writer, should be valuable particularly in locations where the water used to lubricate the bearing is very high in mineral content.

In conclusion the writer should like to emphasize by repeating that experience thus far definitely points to the desirability of using graphite of fine particle size dispersed in suitable fluid carriers in order to produce homogeneous distribution of graphite throughout the bearings, thereby giving the bearings constant friction properties.

W. Trinks.[7] The writer has often wondered why composition bearings are called "plastics," when they are as hard and brittle as a piece of coal. Even less success has attended the determination of the origin of the word "Micarta." It seems to indicate that the bearing material contains, or at some time did contain, mica. Can the author supply any information on these questions?

A great deal of information is given in the paper on strength of Micarta in different directions. This brings to mind the question: Why is Micarta made in layers? The writer is familiar with at least one material in which the shredded cotton fibers are mixed with the uncured phenolformaldehyde and the mass cured as a solid block. If the block of composition is not sufficiently strong, a steel framework is placed in the raw composition and is cooked with it. This design has been successful not only for bearings but also for slippers of universal joints in rolling mills. These slippers act, in a way, like bearings.

Author's Closure

Regarding Mr. Claus's remarks, it was indicated in the paper that it is necessary to differentiate between plastic materials of the first and the other two types. The materials of the first type are made from various plain or graphitic molding powders, while the second and third types are obtained by molding laminated textiles and felt fillers. By virtue of their different internal structures, finished products made according to the three types mentioned have different physical properties, as shown in the paper. Material of type one might be brittle, but laminated bearings are very strong and tough. They are not brittle, and have high bond strength.

Hard and soft graphite Micarta simply designates the way in which graphite is put into the material. In the case of hard Micarta, the graphite is put on the cloth before the resin is applied, while in the case of soft graphite the graphite is put into the resin.

Mr. Heer's discussion is interesting since it indicates a few fine points pertaining to the properties of graphite. The amount of 325-mesh lubricating graphite powder used was evenly distributed in the cloth in such a quantity as not to decrease the bond strength.

Professor Everett correctly indicated the surprising change in the coefficient of friction at increasing speed as shown in Fig. 2 of the paper, taken without alterations from Mr. Lehr's paper.

As far as the writer was able to find a correct answer to Professor Trinks's question, the name "Micarta" (a trade name, used for laminated products only) derives its name from mica, which has a laminar structure and high dielectric strength, and which was originally added in a small quantity to Micarta.

[7] Professor of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa. Mem. A.S.M.E.

Neoprene as a Spring Material

By FELIX L. YERZLEY,[1] WILMINGTON, DEL.

In this paper the mechanical properties of neoprene which are important for vibration isolation and damping are evaluated and compared with the analogous properties of rubber. Static and dynamic moduli and damping action are specifically treated.

ADVANCES in engineering design proceed from the development of new materials, from increasing knowledge of available materials, and from the exercise of imagination in the creation of new applications. It must be evident that every increase in the variety of available materials diminishes the number of restrictions on mechanical invention. It must also be true that, as the variety of physical characteristics available in different materials increases, fewer compromises are necessary with imperfectly suited materials. Neoprene has taken its place at the side of rubber as an elastic material for springs. As an addition to the variety of materials available to engineers it has already increased the variety of structures that can be made from them.

The Field for Neoprene Springs

There can be no question of the superiority of metallic springs over those made of the rubber-like materials (1),[2] including rubber for certain applications, and there can be no doubt of the superiority of rubber-like materials in other cases. One needs only to survey the present applications of rubber-like materials in automobiles to verify the latter statement. Many improvements in riding comfort and performance became possible only as the availability and characteristics of rubber were appreciated. Further affirming evidence of the unique advantages of rubber may readily be found in the literature (2, 3) and in the prevailing trend toward further uses of rubber.

Analysis of spring applications reveals at least five outstanding reasons for the use of rubber springs. These are:

1 To reduce noise. The function of rubber in this respect is to avoid transmission of audiofrequencies through metallic supports (4, 5). It is interesting to note, however, that other steps must sometimes be taken simultaneously to eliminate sounds which are air-borne.

2 To impart flexibility. In addition to the technically more interesting applications, involving periodic vibrations which are subject to more or less precise mathematical treatment, there are numerous applications arising from the need for a mild cushioning effect, for example, as supplied by the numerous rubber pads and cushions used in automobile bodies (6). These provide the automobile structure with a necessary degree of over-all flexibility and elastic yield in an economical and simple way.

3 To reduce production cost. The trend in modern machine production is toward lighter construction and higher speeds. In general, this means also that precision in manufacture must be increased, usually at an increase in the cost of machining operations. Rubber in compression will conform readily to irregular surfaces and can, in some cases, be employed to eliminate a machining operation as well as maintenance expense. This is illustrated in Fig. 1. A self-aligning bearing is made by replacing a lubricated ball-and-socket construction with a less expensive assembly involving a rubber-like material. A spherically formed band of neoprene between the bearing and the block not only provides the necessary self-aligning feature, but does so with an appreciable saving in cost.

4 To reduce maintenance costs. Rubber springs frequently eliminate guides and other points of metal-to-metal contact, and thereby reduce maintenance costs (7).

5 To obtain definite damping action. When damping action is required with metallic springs, auxiliary frictional equipment is usually required. Leaf springs provide damping, but the extent of the damping action varies with the lubrication between the leaves and is not subject to practical control. The damping action of a rubber compound is definite and reasonably constant and is obtained without mechanical wear and without need of attention.

The foregoing considerations apply generally, but in varying degrees, to all of the rubber-like materials. In addition, the following distinctive advantages of neoprene over rubber are characteristic:

1 Relative immunity from deterioration in petroleum products. All petroleum solvents cause disintegration of rubber. Although the rate of deterioration is dependent upon the nature of the petroleum product, the end results are similar. Penetration into the rubber first causes softening and swelling, followed by gradual loss of strength and eventual failure. Neoprene differs from natural rubber, first, by showing much greater resistance to the penetration of petroleum products and, second, by retaining its original physical properties to a much greater extent (8, 9).

Fig. 1 A Self-Aligning Bearing Constructed With Neoprene Cushion

2 Greater chemical stability at high temperatures. Rubber and neoprene, which are organic materials, cannot be exposed to the moderately high temperatures at which metallic springs frequently operate without undergoing chemical changes. The changes of rubber and neoprene on continued exposure to high temperatures are quite different. Continuous exposure to elevated temperatures of the best modern heat-resisting rubber compositions, which have low sulphur content, tends to cause reversion. Reverted rubber compositions are characterized by loss of resilience, elastic modulus, and tensile strength. Neoprene compositions do not revert at high temperatures, but gradually become harder with increased modulus and with relatively slow

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

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loss of tensile strength. The residual plastic effect, drift, of the vulcanizate is decreased by continued heating, and the modulus increases rather than decreases.

3 Resistance to the action of sunlight and ozone. Sunlight and ozone are grouped together because formation of ozone at the surface of rubber in sunlight is considered the prime cause of the surface deterioration which occurs (10). In many industrial situations, the presence of ozone precludes the use of rubber. Neoprene, on the other hand, is used extensively as a jacketing material for high-voltage cables, where ozone is generated by corona discharge. It is also used in direct sunlight, where rubber does not possess the necessary degree of permanence.

4 Greater damping action. The hysteresis loss and hence the damping action of both rubber and neoprene may be varied considerably by compounding. In general, however, hysteresis loss, with stocks that are also practical in other respects, is greater for neoprene than for rubber. The thermal conductivity of neoprene is also higher than that of rubber and permits more rapid dissipation of heat.

The foregoing statements relate to the preliminary selection of a spring material. It is equally important to have definite knowledge of the various moduli of elasticity in order to determine the size and shape of the required spring. For that reason the balance of the paper will deal almost exclusively with the elastic properties of neoprene.

Elastic Properties of Neoprene

The author has recently described a compact mechanical oscillograph (11), Fig. 2, particularly for the evaluation of rubber-

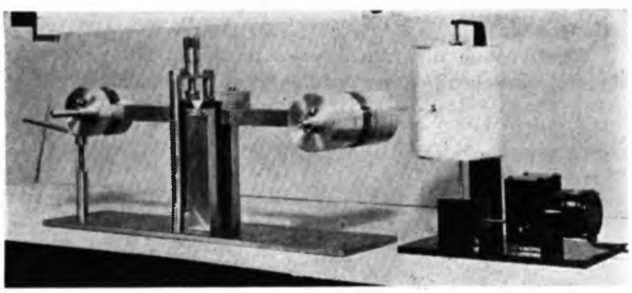

Fig. 2 The Oscillograph; Oscillating Bar and Loading Platform Are Mounted on Knife-Edges

(Machine is shown in static equilibrium position with test piece under compression.)

like materials in compression, both statically and dynamically. This apparatus has been used for the tests reported herein. Small cylindrical test pieces nominally $^3/_4$ in. diam $\times$ $^1/_2$ in. high are used. For static-load-compression relationships, deflections under various dead loads up to 280 lb per sq in. are recorded. For dynamic characteristics, the bar is allowed to oscillate at its natural frequency of vibration under the combined action of gravity and the elastic force of the test piece. The dynamic modulus is then calculated from the frequency and the inertia of the bar by means of the equation, $K = 210\,If^2$. Test frequencies are usually of the order of 2 to 5 cycles per sec. Data are recorded autographically. A typical oscillogram from this series of tests is shown in Fig. 3.

It is well known that rubber and neoprene do not show the same stiffness on the first cycle of loading and unloading as they do on successive cycles. Each of the test pieces was subjected to several conditioning cycles of loading and unloading under an average load of 280 lb per sq in. Static-load-compression data were then obtained, followed by further conditioning and a dynamic test under the average load corresponding to approximately 30 per cent compression. Data obtained in this way were plotted as shown for a rubber and a neoprene compound in Fig. 4. It will be observed, as Haushalter (12) has said, that the curves are approximately straight up to 20 per cent compression but that for greater deformations there is pronounced curvature. The slopes of the dotted lines, shown in Fig. 4, represent the dynamic moduli and were calculated from the dynamic tests under average loads of 120 and 100 lb per sq in. for the neoprene and rubber compounds, respectively. The crosses shown on the dotted lines indicate the points of equilibrium reached at the end of the dynamic test. The difference between the slopes of these lines and the tangents to the static load-compression curve illustrates the difference between the dynamic and static moduli.

It will be noted in Fig. 4, that the general shape of the curves for rubber and neoprene is the same. The neoprene loop is slightly wider, however, indicating greater hysteresis loss or damping action. The increase in modulus of neoprene in the dynamic test is greater than that of rubber. These differences are characteristic of the two materials today. Subsequent chemical developments may change this comparison.

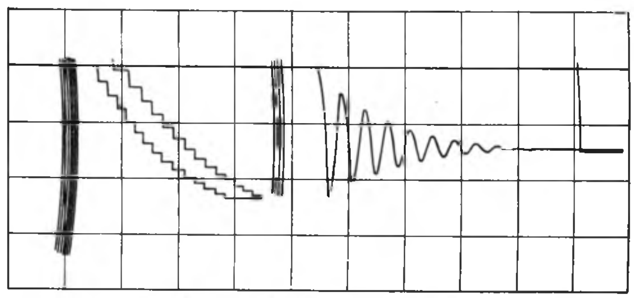

Fig. 3 Typical Oscillogram

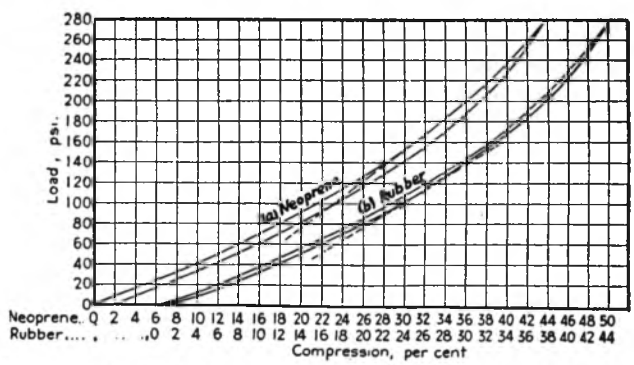

Fig. 4 Static Hysteresis Loops and Dynamic Moduli of Neoprene Test Cylinder and Rubber Test Cylinder in Compression

For the purposes of this paper, it is considered satisfactory to apply the term "compression modulus" to the slope at any point of the load-compression curve. It will be noted, however, that the values of the static modulus are obtained as slopes on the loading branch of the hysteresis loop. Although this is an arbitrary procedure, it appears to be common practice. For certain problems, the slope at points of the unloading curve might be more important. In other cases, an over-all average for successive cycles of loading and unloading is a more important value. Indeed the dynamic modulus reported in this paper is such a value.

During the last few years, correlations have been developed between the elastic moduli of rubber and values of the Shore durometer type A hardness. These correlations, though only approximate, provide a convenient way of determining the elastic moduli near zero load without resort to more extensive and sometimes expensive tests.[3] Conversely it provides the convenient method of specifying moduli by stating the hardness desired, particularly for springs in small quantities. The author intends in the following discussion to establish a similar relationship for neoprene with the hope, however, that some evaluation more precise than hardness data may win preference in the future.

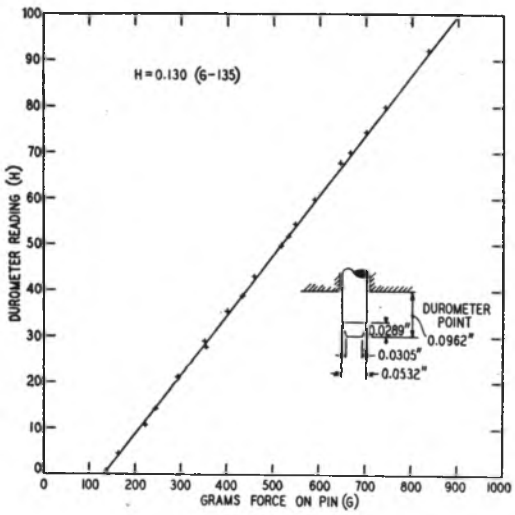

FIG. 5 CALIBRATION OF DUROMETER USED IN TESTS

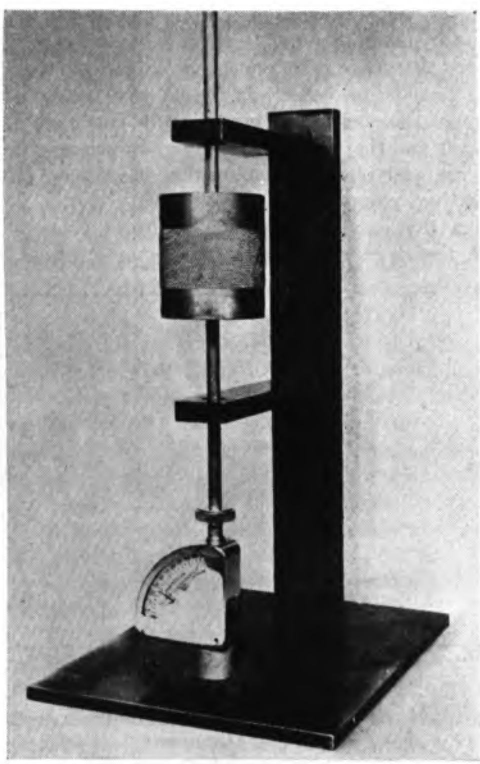

FIG. 6 SPECIAL DUROMETER STAND

There are three apparent reasons for unavoidable errors in correlations between durometer hardness and moduli. In the first place, the calibration of the durometer has not been adequately standardized by the manufacturer. As a consequence, different instruments will lead to different evaluations. In the second place, rubber technologists will readily agree that, unless operators are schooled with extreme care, different operators will obtain quite different durometer readings on the same compound and with the same durometer. In the third place, one cannot expect the same correlation between shear modulus and durometer readings for all types of rubber compositions. In appreciation of the fine work that has been done in establishing this correlation, expressed by Haushalter (12) and others, this criticism is offered only as constructive caution.

To remove doubt regarding the calibration of the durometer employed in this work, durometer readings were correlated with force in grams on the pin. These data are given in Fig. 5. The dimensions of the pin are also shown. Other dimensions are assumed to be standard. To eliminate the personal element as far as possible a stand, Fig. 6, was built. The instrument and

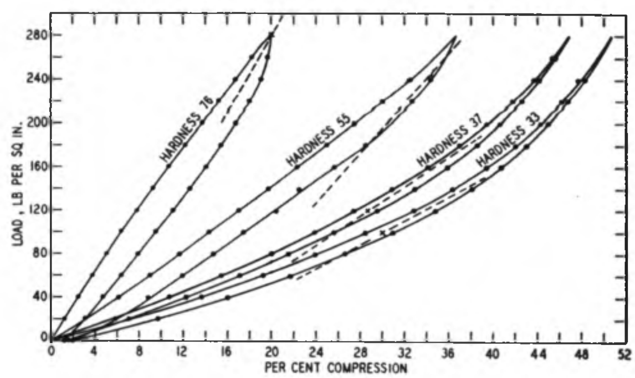

FIG. 7 LOAD-COMPRESSION DATA FOR GROUP OF NEOPRENE TYPE-G VULCANIZATES

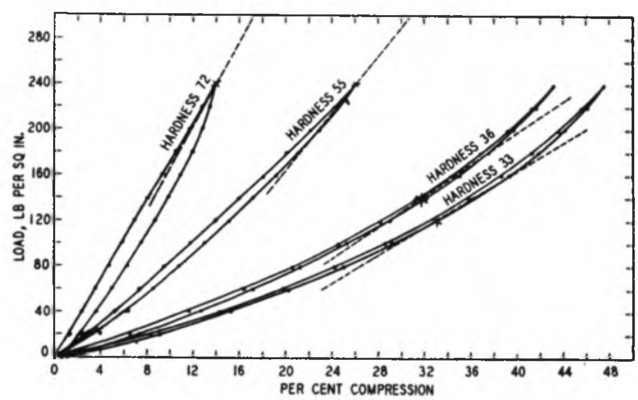

FIG. 8 LOAD-COMPRESSION DATA FOR GROUP OF RUBBER VULCANIZATES

the sliding rod together weigh 2 lb which then becomes the total actuating force (13).

In Fig. 7 are shown load-compression data of a group of neoprene type G compounds having different hardness numbers and, in Fig. 8, a similar group of rubber compounds. It is not feasible to present extensive data in curve form; condensation for the purposes of this paper as well as for practical use is required.

Data for neoprene and rubber compounds in compression are given in Table 1. It will be noted in the table that the static modulus at zero load is in most cases considerably lower than the static modulus under the load used for the dynamic test. As stated previously, the compression curve for rubber, up to 20 per cent, is practically a straight line. This is convenient, since

[3] The oscillograph is intended to provide a direct and inexpensive test for compression moduli and will soon be adapted to direct tests in shear.

TABLE 1 DATA FOR NEOPRENE AND RUBBER COMPOUNDS UNDER COMPRESSION

Compound	Durometer A hardness	Slope of static load-compression curve at zero load, psi	Average compression load in dynamic test, psi	Compression at equilibrium under stated average load, per cent	Slope of static load-compression curve at stated average load, psi	Dynamic modulus calculated from frequency and inertia, psi	Ratio of dynamic to static modulus	Resilience at 30 per cent compression, per cent	Frequency, cycles per min
Neoprene									
302N-42	30	320	100	31.0	440	590	1.34	77	145
						606	1.38	75	212
-41	31	266	100	29.6	440	585	1.33	77	145
						579	1.32	75	206
-40	33	366	100	30.0	440	580	1.32	78	144
						553	1.26	77	202
1522N- 1	33	316	100	28.4	480	542	1.13	78	139
- 3	37	380	120	28.6	550	675	1.23	80	151
-19	49	508	160	30.4	630	1017	1.61	55	185
- 4	52	620	180	30.8	680	982	1.44	65	187
- 5	53	720	200	30.4	740	1069	1.44	71	190
- 6	55	720	220	32.0	820	1142	1.39	68	202
- 7	57	692	200	31.8	770	984	1.28	60	182
-17	61	908	280	31.6	1120	1509	1.35	70	225
-20	62	800	220	31.0	760	1328	1.75	42	218
- 9	62	984	280	31.6	1120	1549	1.38	71	228
-10	62	944	280	32.2	960	1505	1.57	68	225
- 8	63	852	260	31.4	1060	1680	1.58	60	200
-12	73	1264	280	25.4	1360	2110	1.55	51	266
-11	76	1468	280	21.8	1360	1853	1.36	52	250
Rubber									
1333-30	25	150	80	31.4	376	413	1.10	91	126
-31	29	230	80	27.5	403	421	1.04	92	127
-32	33	260	120	33.0	556	614	1.10	92	154
-33	36	340	140	31.6	655	689	1.05	92	169
-34	46	530	200	29.8	872	963	1.10	89	193
-35	55	720	240	26.0	1100	1253	1.14	82	220
-36	64	1100	240	20.6	1352	1580	1.17	74	246
-37	72	1700	240	14.0	1750	1850	1.06	66	266

it may be used in many cases to simplify calculations appreciably. It is true, however, that as a general rule the tangent to the load-compression curve at zero deformation is lower than the tangent at 20 per cent compression. This trend continues and, at 30 per cent compression, a still greater slope exists. The values given in Table 1 for the compression modulus at zero load are the closest values that could be obtained graphically, disregarding any irregular curvatures that were shown by the data for small deformations. In some cases, they correspond to the constant slope of the curve up to 10 per cent compression and, in other cases, they correspond more closely to the particular slope at 10 per cent compression. It will be recognized that errors in obtaining a load-deformation curve for rubber may readily occur at zero because of the difficulty sometimes experienced in determining the zero deformation. This is particularly true of soft stocks which may undergo appreciable deformations with very light loads and it is also true for compositions which have excessive creep or pronounced damping characteristics.

The dynamic tests reported were conducted at 30 per cent average compression, which in terms of usual practice is somewhat abnormal. Various tests conducted by the author have indicated that the relation between static and dynamic moduli is practically independent of the load up to 30 per cent compression. It is also independent of the frequency obtainable by this technique as indicated by the uniformity of the ratio obtained at two frequencies for compounds 302N-40, -41, and -42 in Table 1.

It has frequently been claimed that rubber is considerably stiffer dynamically than it is in static loading. Circulation has even been given to the statement that the dynamic modulus of rubber is 2 or 3 times greater than the static modulus. As the result of his recent work, Kosten (14) has concluded that the ratio of the dynamic modulus to the static modulus lies, as a rule, between 1 and 2 and that the ratio is independent of frequency. The present work, employing a different technique of measurement and a somewhat different frequency range, confirms these conclusions drawn by Kosten. Naunton and Waring (15) on the other hand, have found in tests up to 60,000 cycles per min that the increase in modulus is proportional to frequency. They confirm the finding, however, that the ratio of dynamic to static modulus is between 1 and 2.

The ratio of the dynamic modulus to the static modulus in Table 1 is given in the eighth column and it will be seen that these are in the range 1.04 to 1.75. In the next to the last column, the resilience in per cent, or in other words the elastic efficiency of the compound, is given. These data were calculated from the dynamic records by obtaining the ratio of successive heights of rebound on the oscillograph. If resilience is plotted against the ratio of dynamic modulus to static modulus, as in Fig. 9, it will be noted that there is an increase in the ratio as the resilience of the compounds decreases. It should also be kept in mind, however, that the compounds given in this paper are of a commercial grade and that many complicated factors involved in the state of cure and physical composition have been introduced other than the obvious change in resilience.

Assuming that, under operating conditions likely to be encountered, the dynamic modulus can be as much as 2 times

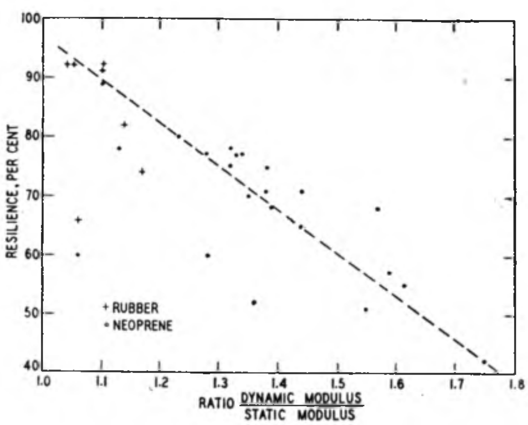

FIG. 9 RESILIENCE PLOTTED AGAINST RATIO OF DYNAMIC MODULUS TO STATIC MODULUS
(The dash line is drawn to emphasize the major trend.)

greater than the static modulus at a given load, one might naturally inquire as to the importance of the change. According to the usual theories of vibration, the frequency at resonance is given by the equation, $f = \text{constant} \sqrt{\text{spring constant}}$. In other words, a doubling of the modulus of rubber or neoprene will cause a 41 per cent increase in resonant frequency. This is not a negligible factor, but as this work and that of Kosten (14) indicate, a doubling of the modulus is not apt to occur. The actual change in resonant frequency will be moderate and in most cases negligible. This is especially true, in view of the usual design practice of allowing a factor of 3 or 4, if possible, between the natural frequency of the supports and the probable frequency of impressed forces.

In his useful summary of the theory of vibration, Rosenzweig (16) has presented a discussion of the importance of damping at resonance. When critical speeds are encountered, some damping action or restraining force is absolutely essential. The data in the next to the last column of Table 1 give values for the resilience of the compounds. For the neoprene compounds selected, resilience ranges from 42 to 80 per cent and the corresponding energy absorption would consequently range from 58 to 20 per cent. The values for rubber are different, ranging from 66 to 92 per cent for resilience and from 34 to 8 per cent for energy absorption in a given cycle of loading and unloading. The art of compounding has not yet progressed to the point where resilience can be varied independently from all other properties, but there are ways of varying the resilience in a limited range for a given modulus.

In Fig. 10 durometer hardness is plotted against compression modulus at zero load as taken from Table 1. Other data of this same form have been published by Haushalter and are shown by the dotted line. This illustration confirms the relationship found by Haushalter (12) between compression modulus and durometer hardness as closely as one might expect. It will be noted, however, that there are points which do not lie on the curve.

Haushalter (12) and Kosten (14) both have emphasized the importance of the ratio between the height of a cylindrical test piece and its diameter. Haushalter has indicated that, if the ratio is approximately 1:1, the compression modulus is 6.5 times the shear modulus. In the present case, the ratio of the height to the diameter is 0.67. Consequently, it is of interest to determine how nearly the test pieces $^1/_2$ in. high and $^3/_4$ in. diam approach the characteristics of test pieces fulfilling the ratio requirement. Fig. 11 shows the changes caused in the load-compression curve by changes in the height of the cylinder from 0.135 in. to 0.995 in., i.e., by varying the ratio from 0.18 to 1.33. It can be seen that the difference in compression between the 0.502-in. and 0.995-in. cylinders is considerably less than 10 per cent of the percentage of compression at all loads.

In Fig. 12, the diameter of the cylinder is varied from 0.433 in. to 0.950 in., i.e., varying the ratio from 1.15 to 0.53. The discrepancy between the curves for the 0.742-in. and 0.433-in. cylinders is less than 10 per cent of the percentage of compression. If the ratio of 6.5 : 1 is valid for rubber, one might reasonably expect approximately the same ratio to apply for neoprene. Smith, on the other hand, has found the theoretically derived ratio of 3 : 1 (17) between compression modulus and shear modulus to be useful. Although the discrepancy between 6.5 and 3 appears offhand to be disturbingly large, further investigation will probably unify the methods of treatment and contribute better understanding. The measurements in shear so far made by the author have been limited and inadequate to establish a clear relationship between the shear and compression moduli. It is known from several comparisons, however, that the shear modulus of neoprene can be varied over the same range as that of rubber; for example, for practical purposes from 50 to 350 lb per sq in.

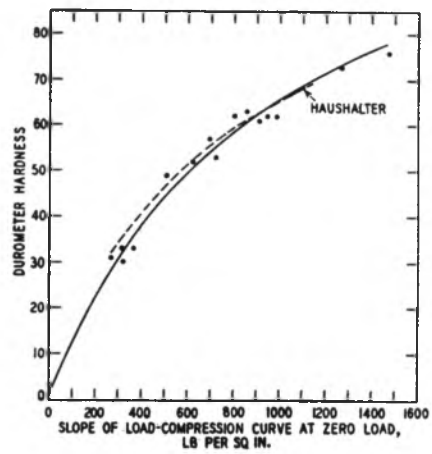

FIG. 10 DUROMETER A HARDNESS PLOTTED AGAINST SLOPE OF LOAD-COMPRESSION CURVE AT ZERO LOAD
(The average obtained by Haushalter for similar data is shown by the dash line.)

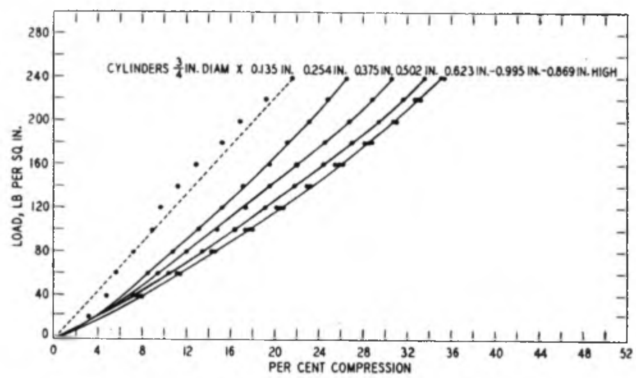

FIG. 11 EFFECT ON LOAD-COMPRESSION CURVE OF TEST CYLINDERS $^3/_4$ IN. DIAM, CAUSED BY VARYING HEIGHT

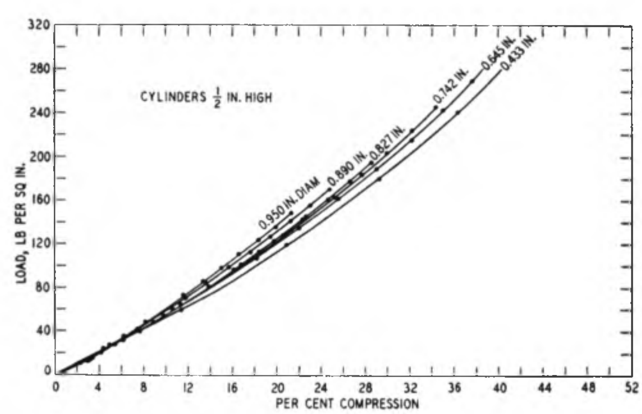

FIG. 12 EFFECT ON LOAD-COMPRESSION CURVE OF TEST CYLINDERS $^1/_2$ IN. HIGH, CAUSED BY VARYING DIAMETER

Whenever damping action is important, due to the unavoidable occurrence of critical speeds, it will be recognized that mountings will be called upon to absorb substantial quantities of energy. The direct result of this absorption of energy will be an increase in the temperature of the mounting. The specific heat of uncompounded and unvulcanized neoprene is 0.52 cal per g and its thermal conductivity 0.00046 cal per sec per cu cm per deg C. Vulcanization seems to have little, if any, effect on the thermal properties of neoprene, although loading ingredients will alter thermal values of vulcanizates in amounts proportional to the volume content and properties of the added ingredients. Since the specific gravity of neoprene compounds is relatively low, being as a rule between 1 and 2, the thermal capacity of neoprene on the volume basis is low. The temperature rise on absorption of energy is, therefore, rapid. In comparison with rubber, however, the thermal conductivity is about 30 per cent higher. Neoprene is, therefore, a better damping material than rubber from the standpoint of its damping action, its chemical stability when heated, and its higher thermal conductivity, permitting more rapid dissipation of heat.

It has been shown that the elastic characteristics of neoprene in which design engineers are interested are similar to those of rubber. Greater damping action and a somewhat higher value for the ratio of dynamic to static moduli are, however, characteristic. The similarity permits the design and application of neoprene springs by the same methods developed for rubber by many investigators (2, 3, 7, 12, 16, 17).

Electrical Properties of Neoprene

Although not related to the main subject of this paper, brief mention of the electrical properties of neoprene is warranted by the occasional demand for combined electrical and mechanical service. The values given in Table 2 are roughly representative of neoprene compounds which contain no fillers and only enough added ingredients to insure proper vulcanization.

TABLE 2 ELECTRICAL PROPERTIES OF NEOPRENE AT 28 C

D-c resistivity	1×10^{12} ohm-cm
Dielectric constant at 1000 cycles per sec	7.5
Power factor at 1000 cycles per sec	3.0 per cent
Dielectric strength, rms at 60 cycles per sec	800 v per mil

Individual compositions will, of course, have different electrical properties. The d-c resistivity, the dielectric constant, and the power factor can be improved somewhat over the figures given in Table 2 by compounding. Compositions containing high proportions of carbon black have lower values of d-c resistivity and dielectric strength than those given in Table 2, and higher values of the dielectric constant and power factor.

Adhesion of Neoprene to Metal

Neoprene compounds can be adhered firmly to brass and brass-plated metals by vulcanization in direct contact with the brass surface. Other commercial methods employ cements to obtain equally strong adhesion. Bond strengths above 1300 lb per sq in. in tension have been obtained under the most favorable conditions.

Performance

Under normal operating conditions, the permanence of neoprene mountings is of a high order. They are not damaged by millions of cycles of loading and unloading and they resist deterioration from oils, chemicals, heat, sunlight, and oxidation. Abnormal conditions may be encountered, however, if mountings are installed or operated incorrectly so that excessive temperature rise occurs in the neoprene as the result of overstressing or excessive frequency of operation. Although no general statement will apply to all conditions, a temperature of 65 C is a reasonable maximum. If operation at higher temperatures is unavoidable, a commensurate reduction should be made in the unit load on the mounting to compensate for loss in spring strength resulting from the abnormal temperature condition.

The functional purpose of neoprene mountings is to isolate and dampen vibrations. The structural requirement is that the mounting shall safely support a given load. If the mounting is operated continuously at excessively elevated temperatures, it is inevitable that the modulus of the composition will gradually increase. The effect of this change will be to alter the characteristics of the mounting, as a vibration isolator, without impairing the ability of the mounting to carry the required load. Hence, although the functional purpose of the mounting may be impaired, the structural safety of the installation ordinarily remains unchanged.

At low temperatures, neoprene temporarily acquires an abnormal hardness. The rate of hardening depends primarily upon the temperature. Preliminary tests, at a temperature of 40 F, have indicated that the modulus of neoprene type-E compositions may increase by a factor of 2 or 3 over a period of 5 days. Furthermore, the change will not occur if the mounting is in normal operation, since the mechanical working of the compound will prevent freezing. If a neoprene mounting has been stiffened by freezing, tests have proved that the mounting will recover its original characteristics quickly upon resumption of operation. It is believed that the rate of freezing increases slightly as the temperature decreases to a value of about 20 F, and that below this temperature the rate of freezing decreases. A study of these points is in progress and more information will be available in the future.

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Discussion

B. S. Cain.[4] It would be helpful if the author could furnish some of the characteristics of neoprene at low temperatures. In particular, is material available for use where some flexibility is required over a wide range of temperatures, including a sub-zero range?

Stuart H. Hahn.[5] This is another excellent presentation of reasons why neoprene can be used for the construction of rubber

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springs. This material, as well as several other modern synthetic rubber-like materials, has certain desirable properties for this service.

The value of the oil resistance and good aging properties of neoprene is well known and appreciated by rubber technologists. However, the admitted tendency of the material to harden continuously upon exposure to elevated temperatures must be balanced against its age resistance at elevated temperatures when spring service is to be considered.

Many designers of rubber springs will disagree with the author's thesis that damping characteristics greater than those of normal rubber compounds are desirable, particularly since high damping usually is accompanied by increased creep. It is, unfortunately, true that low creep is rarely, if ever, a characteristic of compounds of rubber and other rubber-like substances which have low resilience (high damping). Available information on the creep-resilience relationship for neoprene is even less complete than that for rubber compounds. It is hoped that the author will soon be able to fill the need for such information.

The addition to the literature of data on the dynamic modulus of neoprene and other compounds is welcome. It is unfortunate that the author has not been able to extend his investigations above a frequency of about 4 cycles per sec, which is below the ordinary range of resonance frequencies for the more common rubber-spring designs. The modulus effective in determining the actual working transmissibility of such springs is the modulus at the impressed or forcing frequency, which is usually from 2.5 to 10 times the resonance frequency.

It is interesting to note that the data points, in the author's Fig. 9 which shows the resilience versus dynamic/static-modulus ratio for neoprene and rubber compounds, appear to be grouped distinctly into two categories which seem to distinguish the two materials. Although the rubber data are too limited to be conclusive, a curve giving a good fit to the available points would appear to intercept the 40 per cent resilience axis at a value much lower than the intercept for the line shown, which fits the neoprene data rather well.

One most important property of rubber-like materials for spring service is the safe and economical stress to which they may be subjected in service. The working stresses for compressive use are nearly always dictated by creep considerations so that deflections are ordinarily limited to values of about 20 per cent or less. Failures due to actual fracture or cracking of the rubber are quite rare in such service. The determination of working stress for service in shear is not nearly so simple. Some applications which have proved practical in service have made use of deformations somewhat more than the thickness of the rubber (shear strain > 100 per cent thickness) and static or mean dynamic stresses in the region of 100 lb per sq in. are known to be practical in certain limited classes of use, especially where compressive strain perpendicular to shear adhesion areas is imposed. However, in no case does the actual breaking shear stress in continuous service approach the breaking stress as determined by short-time loading tests. It is important to the designer, who contemplates the use of a material such as neoprene in spring service, to know what safe working loads may be applied to the material if service over a given period at approximately known temperatures is to be rendered without failure. The effect of temperature increase in reducing the strength of neoprene and rubber cannot be ignored.

The author's experience with poor standardization of the durometer (type A) agrees with that of many others. His description of the calibration of his instrument adds to the value of his hardness-versus-modulus curves, even though there are theoretical objections to the method of calibration which he used. It also emphasizes the precautions which must be taken to avoid or compensate for the well-known shortcomings of the instrument. Tests and theoretical studies by another member of this laboratory have indicated very clearly that the most reliable results may be obtained only when the durometer is held in such a way that the lightest possible pressure is obtained between the rubber surface and the foot or disk surrounding the indenter point.

The author seems to be guilty of overemphasis in saying that the slightly higher thermal conductivity of neoprene compounds is sufficiently great to constitute an appreciable advantage of this synthetic over natural rubber. The thermal conductivities of neoprene and rubber compounds suitable for spring service are all so low that any appreciable energy absorption produces sizable temperature rises in both classes of materials. The conductivity of the softer rubber-spring stocks is about 0.95 Btu/(hr) (sq ft) (F/in.) or roughly $^1/_{300}$ that of steel.

An estimate of the temperature which might be produced in mountings made of a neoprene compound and used under conditions of moderately severe vibration has been made for the following hypothetical case:

Machine weight, 2000 lb
Shear stress on mountings, 40 lb per sq in.
Static deflection of mountings, 1 in.
Resilience of neoprene compound used in mountings, 60 per cent
Magnitude of simple harmonic disturbing force, ± 20 lb
Frequency of disturbing force, resonant (approximately 190 cycles per min)

For this particular case, the mean rate of temperature rise works out to something like 3 F per min and, since the mountings would necessarily have a thickness of 1 in. or more, the maximum temperature in each mounting would probably attain a value of about 200 F after starting at 80 F and running for about 1 hr, if failure of the material did not occur before that, through the combination of the weakening effects of temperature and the gradually increasing amplitude which would result as the damping coefficient decreased with increasing temperature. Hence, intentional operation at resonant frequency would be impractical, since the comparatively higher conductivity of neoprene stocks is still too low to permit their extensive use as shock absorbers.

In some cases there has been a real need for a cool-running rubber or synthetic compound for use in services where occasional or even rather frequent shocks or dynamic-stress cycles are encountered. However, the solution of the problem in most cases, and probably in every case, has been the use of a compound which possesses relatively high resilience with ordinary values of thermal conductivity, rather than low resilience accompanied by the maximum possible conductivity. It should be emphasized that thermal conductivities of all rubber-like materials are so low that they are unable to dissipate very much energy continuously except in the case of relatively small pieces.

F. L. Haushalter.[6] The writer expected to learn from this paper how high neoprene might be stressed as compared with rubber, without experiencing trouble from creep or drift. If neoprene springs are used to support a machine, what would be a reasonable estimate as to how much lower the machine would be setting in a year's time? Also, what would be the modulus or rate of the spring in a year and, possibly, in 3 or 10 years hence? If the springs are to be used in an atmosphere of elevated temperature, data on creep would be essential before choosing the stress at which the spring will function. Values of dynamic modulus versus static modulus for this material are useful enough, but what are these moduli after creep has taken place at elevated temperatures? We know that for rubber the values are appreci-

[6] Development Engineer, The B. F. Goodrich Company, Akron, Ohio.

ably higher, and it is reasonable to suppose that for neoprene the values would be still higher because it is the writer's experience that neoprene compounds have greater creep than equivalent rubber compounds.

The more work the writer does with springs made from rubber, or rubber-like materials, the more he is convinced that damping is an unwanted characteristic in the material. About the only place one would require a spring to operate at resonance would be to counteract and neutralize the vibration of a body or member, the critical period of which was at that same frequency. Such applications are rare. About the only instance of this kind which the writer can recall occurred several years ago. In this case, the front wheels of an automobile developed a critical period causing bad wheel shimmy at a certain operating speed when traveling on cobblestone pavement. This vibration, which was in the horizontal plane, was neutralized by attaching a mass on the end of a spring, which in turn was fastened to the tie rod. The spring was designed to give the suspended mass the same resonant frequency as the wheels in the horizontal plane.

In the vast majority of cases, it would certainly be ridiculous to want to isolate a vibrating body with springs and then have the springs function at their point of resonance. If we wish to do any damping in passing through critical speeds, it can be done much more cheaply by using hydraulic shock absorbers. If we do not operate springs in the resonant range, which Hull pointed out should not be less than a frequency ratio of 1.4, damping is a distinct disadvantage, for the energy consumed in damping increases the transmissibility through the mounting.

Furthermore, the more damping there is in the material, the more creep will take place in the spring. The author has a good opportunity to disprove this statement, but if Isaac Newton was right, it is very probable that, as compounds made from neoprene G are developed, which take on less and less creep, we shall find that the hysteresis loss, or damping effect, will diminish accordingly.

As for the argument on durometer and hardness, which seems to be an endless one, as unscientific as this instrument appears to be, the writer dislikes the thought of condemning it to the scrap heap, for when properly used it certainly can pick out small differences in hardnesses of stocks in a comparative way. For the detection of these small changes in hardness, the writer would not use the author's method, but would depend upon the human hand. The writer was once confronted with the task of making a rubber spring slightly stiffer than a previous spring, which meant an increase of about $^1/_2$ point in hardness on the durometer. After using the Olsen hardness tester and the Pusey & Jones plastometer with conflicting results, the only way discovered for accomplishing this was to lay on a table disks of the possible compounds of the same thickness, and go from one to another with the durometer in hand. The stock was selected in this way and, when the spring was made up, the shear-modulus value was found to be exactly that required. It is truly a difficult matter to standardize durometers, but there is no reason why we cannot calibrate one against the other. Undoubtedly, one who works constantly on the compounding of rubber and rubber-like materials soon acquires facility in the use of his own durometer.

J. F. Downie Smith.[7] The material given in this paper will, in the writer's opinion, be welcomed by designers who are interested in the application of rubber and rubber-like materials in industry. It is hoped that more information of this type on other rubber-like materials will be made available in the near future. At present the mechanical engineer feels rather at a loss when he is trying to determine the proper material to use for a specific application.

A short time ago the author gave the writer two samples of neoprene and one of rubber each bonded to metal plates in the form shown in Fig. 13 of this discussion. In order to make a

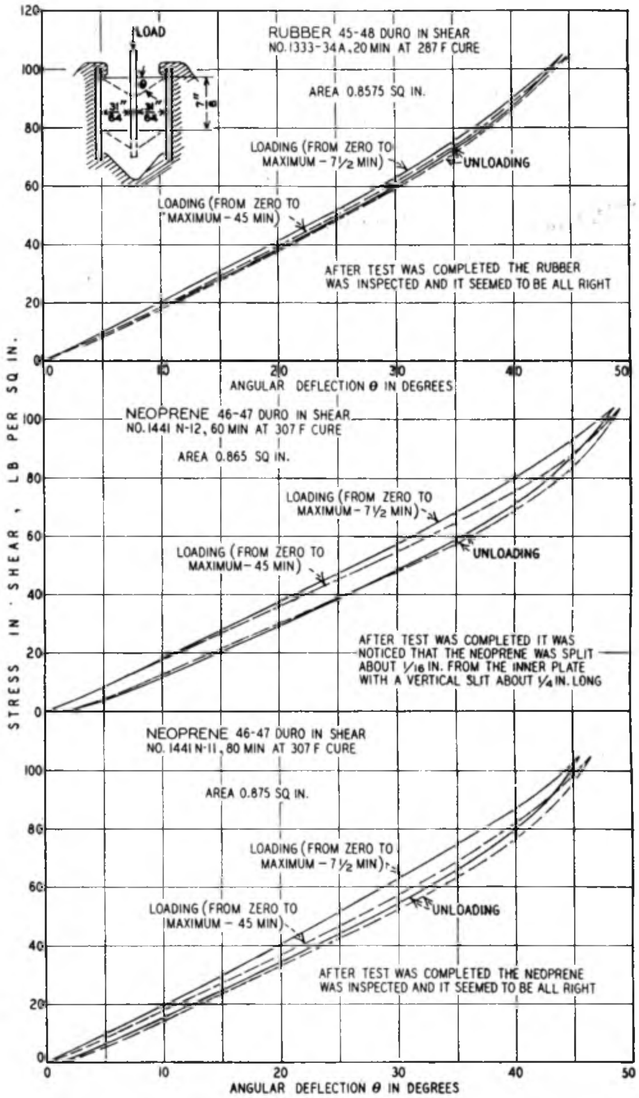

Fig. 13 Loading and Unloading Curves of Neoprene and Rubber Samples Bonded to Metal Plates

cursory examination of each sample to see if it would be applicable in springs as used in experimental work, each sample was loaded and unloaded quickly a few times. Then a loading curve was taken in $7^1/_2$ min with the unloading curve taken in about the same time. A second set of curves was then made with loading and unloading periods 6 times as great. The results are shown in Fig. 13.

One of the neoprene samples, No. 1441 N-12, after the tests were completed, was found to be split, and could no longer be recommended as a spring. The other neoprene sample and the rubber one were apparently satisfactory.

It may be observed that one of the neoprene samples gave a much greater variation between quick and slow loading than did the rubber. At two shear stresses the percentage variations between the two loading curves are as follows:

[7] Executive Engineer of Research, Edward G. Budd Manufacturing Company, Philadelphia, Pa. Mem. A.S.M.E.

		At 40 lb per sq in.	At 60 lb per sq in.
1	For rubber No. 1333-34A	4.4	2.6
2	For neoprene No. 1441 N-12	4.5	4.8
3	For neoprene No. 1441 N-11	9.9	8.9

The writer would like to ask the author if such results would always be expected from typical rubber and neoprene springs, or if these samples are unusual in some way. If results of this type are expected, would he consider them detrimental to the use of neoprene as a spring?

An additional point of concern is that none of the curves comes anywhere near the theoretical shear loading curve which might be expected from previously published data on rubber. This theoretical curve is based on the calibration of definite Shore durometers against known moduli of elasticity in shear for rubber. Now all three of the curves shown in Fig. 13 check one another reasonably well, and one of the samples is of rubber and two are of neoprene so that the change from natural rubber to a synthetic rubber-like material is not the answer. One thought which arises is: "Does the size of the sample have an influence on the rough relationship of durometer hardness and modulus of elasticity in shear?" These particular samples are much smaller than those ordinarily used by the writer, and it occurred to him that this may explain the discrepancy to some extent.

Author's Closure

In this paper, an attempt was made to summarize such portions of the present knowledge of neoprene as might contribute to the design of springs. It is a paper on a spring material, however, and not a detailed paper on springs or on vibration in engineering. The author attempted to cite adequate references, without giving a complete bibliography, on these topics as collateral reading for anyone sufficiently interested. The fact is appreciated by the author that the present knowledge of neoprene has limitations. In some respects, additional research and further years of experience will permit a more definite account in the future than is possible today.

Mr. Cain has asked a question particularly pertinent to outdoor service. A specific answer would require definition of the expression "some flexibility." At — 40 F neoprene compositions of a high-grade mechanical type properly cured approximate the stiffness of hard leather but they are not brittle after exposure for as long a period as 14 days. If this condition is considered to fall within the intended meaning of the question, neoprene is flexible at all low temperatures down to — 40 F for any length of exposure. The change of modulus under these conditions would, of course, drastically alter the characteristics of a neoprene spring and would be against its application in some cases. Stiffness acquired by exposure to low temperatures can be overcome by heating from external sources, or by mechanical working.

A detailed discussion of each of the points raised by Mr. Hahn and, in some cases, again by Mr. Haushalter would be inordinately long and involved. There seems to be peculiar disagreement in the literature and among engineers as to what properties in rubber are desirable for some types of service. Every individual taking part in such technical controversies as this has some real basis for his views but, when the situation is viewed broadly, the arguments in good faith seem to arise from deductions based upon distinctly different types of service. The author has culled from the literature the outstanding reasons given for the use of rubber as a spring material under particular conditions. These reasons are, therefore, advantages possessed by rubber springs over metallic springs and are listed as such in the introduction to the paper. The same literature has pointed out further properties to be desired. Such properties, when possessed by neoprene, thus become advantages of neoprene over natural rubber for particular conditions which have been adequately defined in the early part of the paper. It seems trite to point out that, where these conditions do not exist for either rubber or neoprene, other selections would be made and that the selection of neoprene or rubber would sometimes be based upon one or two outstanding characteristics, in spite of attendant disadvantages.

The relatively high damping rate obtainable with neoprene is an advantage when relatively rapid damping is desirable. In other cases a high rate of damping would be a distinct handicap. The author does not believe that any engineer could specify the properties of an ideal spring system without including in the specification a desirable range of damping rates. Although Rosenzweig (16) has discussed the advantage of damping at critical frequencies, the bulk of his paper is devoted to proof that, outside of the critical range, damping is undesirable in the isolation of vibration. Numerous other references could also be given which cover the same point. The author has merely attempted to show that neoprene increases the range of damping rates available with rubber-like materials beyond the limits feasible with rubber. It is the job of designers to determine how much damping is required in a given case.

Mr. Haushalter's statement that spring systems are not designed to operate at resonance, apparently, is made without considering the present importance of dynamic balancers. The example of front-wheel shimmy on automobiles has been supplanted in industrial importance by a host of harmonic balancers in other applications. Those used on lightweight crankshafts to reduce torsional vibration are particularly interesting. Combinations of frictional and dynamic absorption offer the possibility of effective refinements in design.

Various authorities have suggested the desirability of damping action by rubber springs. Among them Haushalter, in a recent discussion,[8] expressed himself as follows: "It would be very desirable to have a rubber spring with a high damping coefficient but, with a compound giving this, the drift greatly increases. A combination of high damping and low drift in a rubber stock is yet to be found. The two conditions, apparently, are diametrically opposed. Work is still going on, however, and there are some possibilities in this direction with synthetic rubber."

In his experience with neoprene, the author has noted frequent exceptions to the statement that there is a general correspondence between creep and resilience. In a current series of tests, for example, two neoprene compounds, cured for the same length of time and having the same compression moduli, have 78 and 76 per cent resilience, respectively. However, in a shear-drift test, the former compound shows roughly twice the drift rate indicated by the second. This is not an isolated case, but is a repetition of a rather frequent occurrence. If the correlation were strictly true for rubber, as it certainly is not for neoprene, long and expensive tests of creep could be dispensed with in favor of fast and relatively inexpensive tests of resilience. Unfortunately, this is not the case.

Very properly, Mr. Hahn has pointed out that the thermal conductivity of rubber and of neoprene is relatively low. Many mechanical designs of rubber parts have failed because excessive heat was developed in operation. Other things being equal, any increase in thermal conductivity will assist the dissipation of energy, but a 30 per cent increase can only offer help to a minor extent. The relative values of thermal conductivity are quite incidental and, if the author is guilty of overemphasis, the point is well taken.

The use of neoprene commercially as a spring material is a relatively recent development and it is not possible, at the present

[8] "Rubber Cushioning Devices," by C. F. Hirshfeld and E. H. Piron; discussion by F. L. Haushalter, Trans. A.S.M.E., vol. 60, 1938, p. 204.

time, to predict what will happen to a spring in 3 to 10 years. Yet, it is believed to be true that the best rubber compositions have lower drift than the best neoprene compositions. However, the author knows that it is possible to produce neoprene compositions which compare very favorably with some of the rubber compositions commercially used. Unfortunately, tests which are now in progress at the du Pont rubber laboratories have not yet progressed to the point where detailed presentation is permissible.

With reference to the discussion by Dr. Smith, the author considers creep to be a type of dimensional instability and any type of instability in a structural material is undesirable. It is indicated by the tests that neoprene will creep more under dead load during the first hour than a comparable high-quality rubber under similar conditions. Since the neoprene samples under discussion are the best obtainable at the present time from the creep standpoint, it is apparent that neoprene will creep more during the first hour than rubber. The fact that a defective sample appeared to be better than an undamaged specimen is somewhat surprising and is probably an indication that a more reliable comparison between the compositions might be obtained from a more extended test. It has been the author's experience that the maximum discrepancy in rate of creep between properly vulcanized neoprene and rubber compositions occurs during the first 24 hours.

Inasmuch as the discrepancy in theory mentioned by Dr. Smith concerns a correlation of shear modulus with durometer hardness, it does not seem to be a matter of the greatest importance. If we remember that the correlation between elastic moduli and durometer readings is at best an approximation, occasional departures should not be too disturbing. On the other hand, data in the author's personal files indicate that the published values of shear modulus referred to by Dr. Smith are about 30 per cent low and this observation is confirmed by Dr. Smith in this case. It should be more disturbing if the size of the samples has a large effect upon the value of the shear modulus, since this would vastly complicate the design of shear springs by requiring a shape or size factor. The author's experience has been more extensive with small samples than with large ones, but there is inclination to doubt that the size of shear mountings has any great influence upon the modulus of a spring composition, provided each specimen is uniformly cured throughout its entire volume.

It is quite apparent that the introduction of rubber springs to industry, particularly to the automotive industry, preceded the development of an adequate literature and engineering background for general purposes. Even today, data on rubber are extremely meager and without a satisfactory network of correlating theory. It is to be hoped that information on rubber and similar materials will develop more rapidly in the future and that the present mystery surrounding the mechanical properties of these materials will soon no longer exist.