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EFFECT OF ENCLOSURES ON DIRECT STEAM RADIATOR PERFORMANCE

BY

MAURICE K. FAHNESTOCK

BULLETIN NO. 169

ENGINEERING EXPERIMENT STATION

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ENGINEERING EXPERIMENT STATION

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EFFECT OF ENCLOSURES ON DIRECT STEAM RADIATOR PERFORMANCE

BY

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I. Introduction

1. *Preliminary Statement.*—This bulletin is a report of the results of the first year's work under the terms of a coöperative agreement between the National Boiler and Radiator Manufacturers' Association, the Illinois Master Plumbers' Association, and the University of Illinois, providing for an investigation of steam and hot-water heating systems. The agreement was formally approved March 9, 1926, and became operative on April 10, 1926. The results presented in this bulletin are based upon the work done between April 1926, and April 1927. The two coöperating associations have been represented during the first year by an advisory committee, the membership of which is as follows:

C. D. Brownell, *Chairman,* representing the Illinois Master Plumbers' Association, Champaign, Illinois.

C. K. Foster, representing the National Boiler and Radiator Manufacturers' Association, Chicago, Illinois.

O. J. Prentice, representing the Steam Specialties Manufacturers, Chicago, Illinois.

F. W. Herendeen, representing the National Boiler and Radiator Manufacturers' Association, Geneva, New York.

H. S. Ashenhurst, representing the Insulation Manufacturers, Chicago, Illinois.

Seward Best, representing the Heating Contractors, Quincy, Illinois.

J. M. Robb, representing the Heating Contractors, Moline, Illinois.

I. H. Cogley, representing the Heating and Piping Contractors' Association of Chicago, Chicago, Illinois.

It is the function of this committee to propose such problems for investigation as are of the greatest interest to the installer of small direct steam and hot-water heating systems, operating on gravity circulation. Of these problems, the Engineering Experiment Station staff selects for study those which can best be investigated with the facilities and equipment available at the University. The coöperating associations provide funds for defraying a major part of the expense of this research work. For the first year the Chicago Master Steam Fitters' Association also assisted by contributing materially to the funds for the prosecution of the work.

2. *Object of Investigation.*—The immediate object of the series of tests reported in this bulletin was to determine the effect of various types of present-day commercial radiator enclosures, shields, and covers on the steam condensing capacity of a direct cast-iron radiator.

3. *Scope of Investigation.*—The effect of an enclosure, shield, or cover upon the steam condensing capacity of a radiator depends upon many factors. The tests made in connection with the present investigation were planned to determine the influence of a number of these factors in the case of radiator enclosures, and those studied were air inlets, air outlets, heights, and grilles.

In conjunction with the work on enclosures, tests were run on an unenclosed radiator, a commercial type of shield, and a cloth cover.

4. *Acknowledgments.*—This investigation has been carried on under the personal supervision of A. C. WILLARD, Professor of Heating and Ventilation and Head of the Department of Mechanical Engineering, and A. P. KRATZ, Research Professor in the Department of Mechanical Engineering. Particular credit is due Professor A. C. Willard for his original layout of the test program and his support and advice during the investigation. Credit is due Professor A. P. Kratz for his personal supervision of the work, and his aid in presenting and interpreting the test results.

The investigation has been carried on as a part of the work of the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the director, and of the Department of Mechanical Engineering, of which Professor A. C. Willard is the head.

II. DESCRIPTION OF APPARATUS

5. *Description of Unenclosed Radiator.*—The radiator used for these tests was a standard 20-section, 38-inch, 3-column, cast-iron water type radiator having a nominal area of 100 sq. ft. The surface was brushed and painted (not dipped) with two coats of flat black paint.

6. *Arrangement of Plant.*—The general arrangement of the plant is shown in Fig. 2. The radiator stood on the main floor of the Mechanical Engineering Laboratory and was partly surrounded by the test booth shown in Figs. 1, 2, 3, and 4. The back of the booth was placed 2½ inches from the radiator, and, together with the sides, served to shield the latter from transverse air currents. The overall dimensions of the booth are given in Fig. 2. The top of the booth

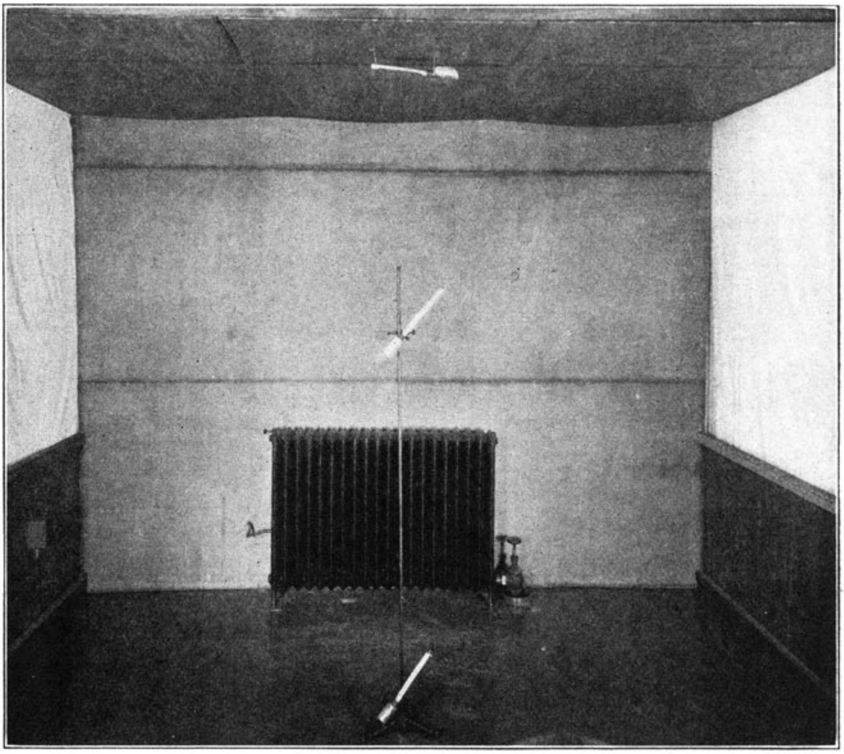

FIG. 1. UNENCLOSED RADIATOR IN TEST BOOTH

served to deflect the vertical air currents in approximately the same manner as the ceiling of a room. The whole inside of the booth presented surfaces similar to papered walls for receiving radiation. These conditions, while not exactly duplicating those of actual service, where the radiator is often set under a window which is in an exposed wall, did duplicate standard conditions under which practically all radiator tests have been conducted, and certainly afforded means of obtaining valid comparative data.

As shown in Fig. 2, the piping, separator, receiver, and weighing tank were placed in the basement of the laboratory, directly beneath the radiator. A separator was used to remove all entrained moisture from the steam, and a mercury manometer and thermometer indicated the steam pressure and temperature. A glass section was installed in the 2-inch vertical riser to the lower tapping of the radiator. The condensate left the radiator through this same connection, and was

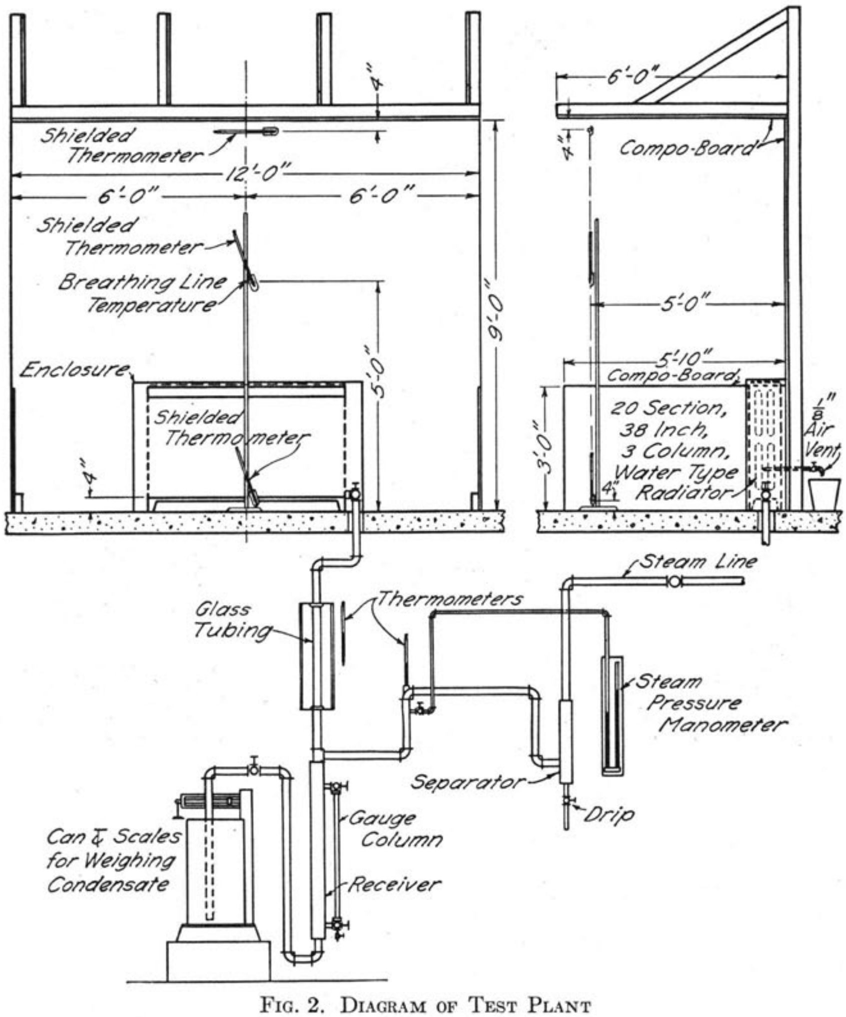

Fig. 2. Diagram of Test Plant

collected in a receiver having a gage glass. The weighing tank was connected through a water seal to the receiver, and the minimum subdivision of the scales used was 0.01 pound. The separator, receiver, and piping were heavily lagged, and the glass section in the vertical riser was enclosed in a triangular glass observation box for the purpose of protection and for the prevention of heat loss. The ⅛-inch air vent tapping on the last radiator section was fitted with a short length of piping and a hand controlled globe valve. All thermometers in the test booth, Fig. 2, were shielded to protect them against the

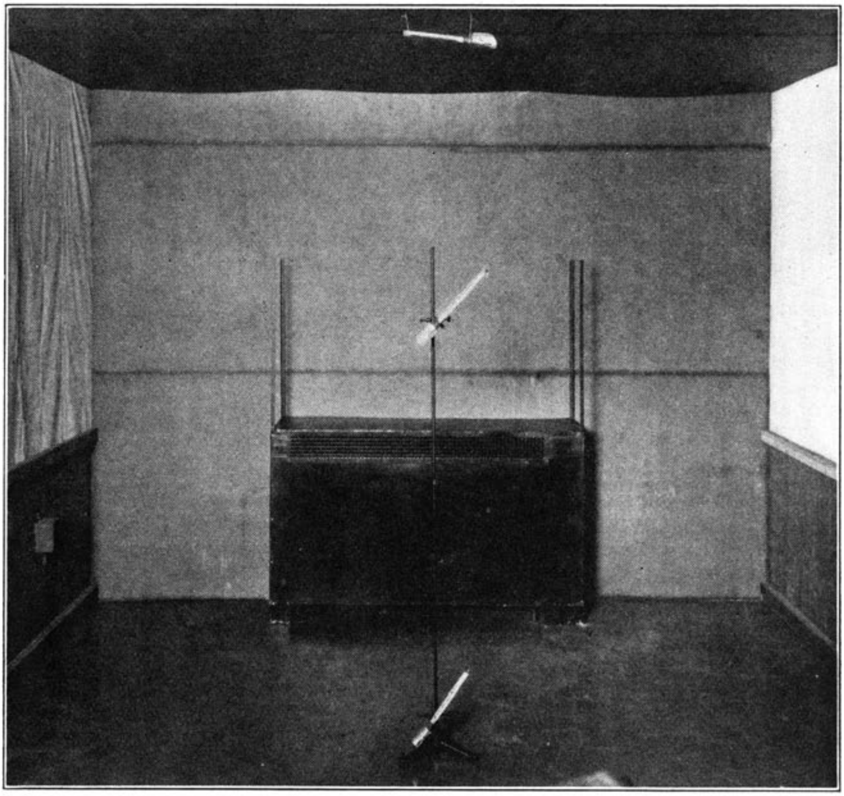

FIG. 3. MINIMUM HEIGHT SHEET IRON ENCLOSURE

effect of direct radiation. A thermometer was placed near the piping in the basement, and one was also placed at breathing line level on the main floor of the laboratory.

7. *Description of Enclosures.*—No commercial enclosures were used, but all enclosures were constructed so as to be adapted to limiting the effect to the particular factor being studied, and at the same time to avoid presenting features that were commercially impossible. All enclosures were painted inside and outside with two coats of flat black paint. In all cases there was a clearance of 2½ inches between the back of the radiator and the back of the enclosure, and of one inch between the radiator and the enclosure at the front. The end clearances were 8 inches and 4 inches, the larger clearance being at the steam inlet end. The effects of height, and of sizes and types of openings were determined with enclosures having the sides and ends

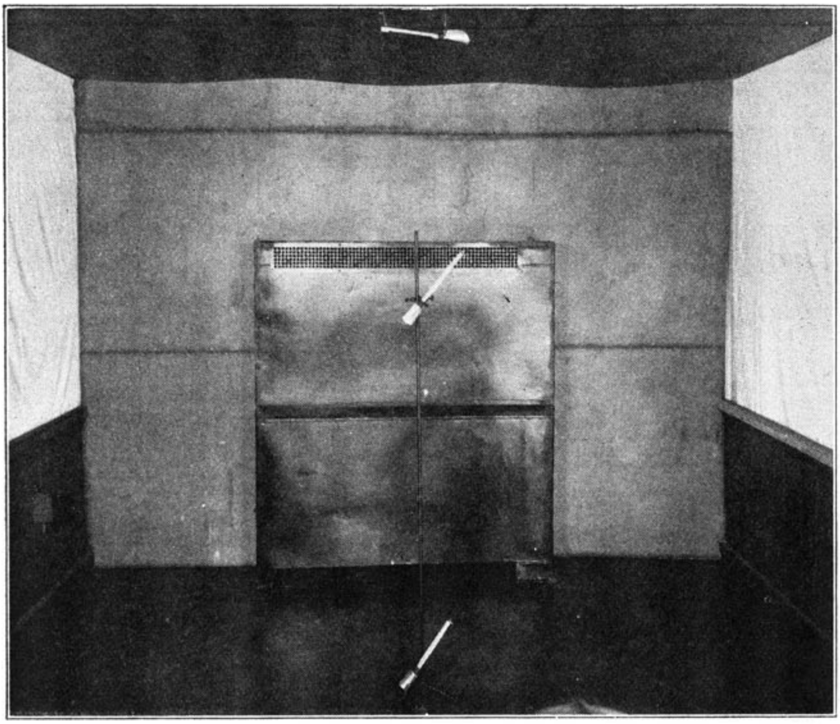

FIG. 4. MAXIMUM HEIGHT SHEET IRON ENCLOSURE

made of solid sheet iron. The minimum height of enclosure for this type is shown in Fig. 3, and the one of maximum height is shown in Fig. 4. The various arrangements of these enclosures are indicated in Fig. 5. They are also shown corresponding to the curves in Figs. 6, 7, and 8. The type and dimensions of the enclosures used to study the effect of different grille constructions are shown in Figs. 14 and 15. The three types of grilles tested are shown to actual scale in Fig. 16.

8. *Description of Shield and Cover.*—The shield tested is shown in Fig. 18 and in Fig. 20 corresponding to the result curve. It was a metal shield of a common commercial type without a humidifier. The top was insulated and extended 2 inches in front of the radiator. The clearance between the shield and the top of the radiator was approximately ½ inch, and the back, which extended to within 3½ inches of the floor, was in contact with the radiator.

The cloth cover shown in Fig. 19 was 6½ inches deep, the same width and length as the radiator, and was made of a closely woven crash cloth.

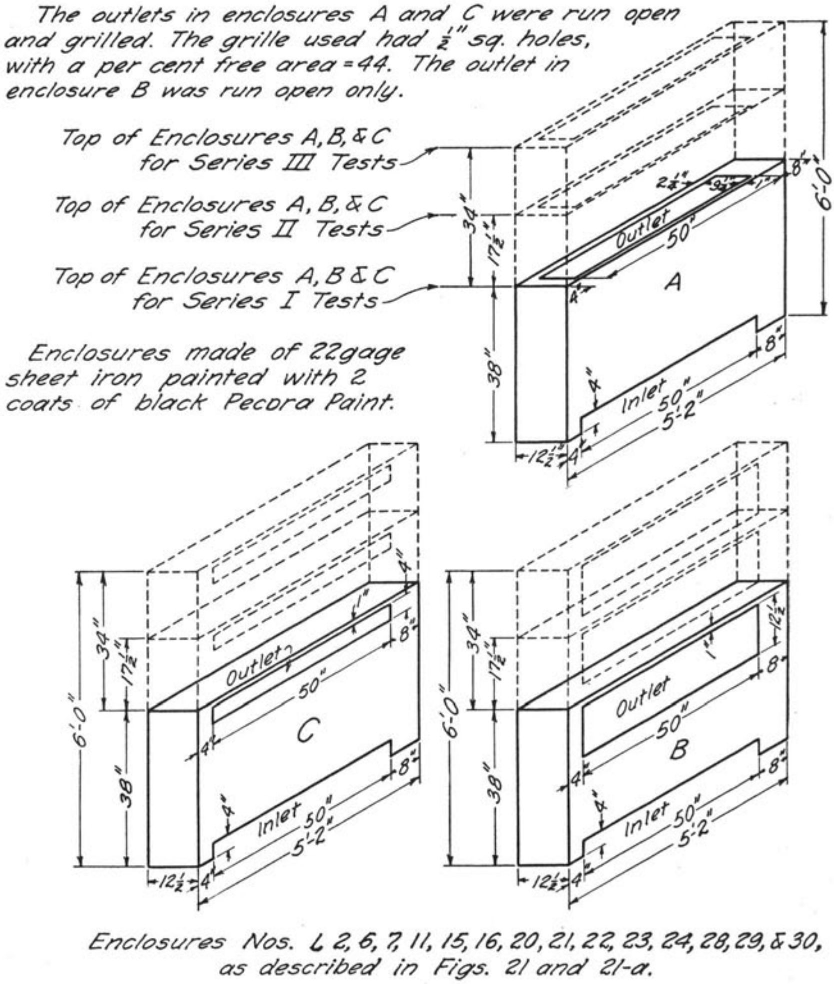

FIG. 5. ENCLOSURES USED IN SERIES I, II, and III TESTS

III. TEST PROCEDURE

9. *Test Procedure.*—As the tests were made in a large laboratory where it was impossible to maintain a constant temperature from day to day it was necessary to run a number of tests on each set-up and determine a performance curve for the different temperature ranges. Since the laboratory was large, the heat given off by the radiator at no time materially affected the breathing line temperature in the test booth, and this temperature was used as the air temperature base in connection with all performance curves. The large number of tests

made on the different enclosures, the shield, and the cover extended over a long period of time, and as the tests were conducted in a laboratory where other work was in progress simultaneously, it was necessary to make check tests from time to time in order to make certain that no factors aside from those being studied were affecting the results. These check tests were usually made on the unenclosed radiator, and although the investigation was carried on during summer and winter months it was always possible to duplicate results.

During every test the steam pressure was maintained constant by a combination of an automatic pressure reducing valve and a manually controlled throttle valve. A constant water level was maintained in the receiver. In all cases, the temperature of the steam in the radiator was 216 deg. F. The separator through which the steam passed just before entering the riser to the radiator was allowed to blow continuously, and similarly the air vent on the radiator was partly open during each test. The condensate left the radiator through the same connection by which the steam entered, was collected in the receiver, passed into the weighing tank, and was weighed every 10 minutes. No test was accepted that showed a variation of more than 2½ per cent in these successive increments of weight. With such constant conditions, an hour was considered as sufficient duration for a test. During this time all temperature and pressure readings were taken every ten minutes and the average values used in the computations of results.

By means of the glass section installed in the vertical riser to the radiator the flow of condensate could be observed, thus insuring that the critical velocity was never exceeded, and that no condensate was carried back into the radiator. This arrangement had the further advantage, that, since the condensate leaving the radiator had no chance to accumulate at one end and was always in intimate contact with live steam, the temperature of the condensate was unquestionably the same as that of the steam. This method has proven unusually satisfactory and no record has been found in which it has been previously reported in connection with radiator tests. The heat loss from the piping at various basement temperatures was determined by running a series of tests on the piping alone, with the radiator disconnected. Correction of all condensation weights was made to allow for the steam equivalent of this loss.

IV. Results

10. *General Statement.*—The results of all the tests are presented in the form of curves shown in Figs. 6-9, 11-13, 17, and 20, using the

temperatures at the breathing line in deg. F. as abscissas and net pounds of steam condensed in the radiator per hour as ordinates. For direct comparative purposes the result of each test for a breathing line temperature of 80 deg. F. is given in the graphical table, Figs. 21 and 21-a. The heavy black bars present a graphical representation of the net pounds of steam condensed in the radiator per hour, using the particular enclosure, shield or cover indicated, with the steam temperature at 216 deg. F. The columns on the left page of this table give data covering the controlling factors of each enclosure, shield, and cover tested. The number assigned to each enclosure is given in the extreme left column, and for a more detailed description of any particular enclosure reference may be made to Figs. 5, 10, or 14. The relative steam condensing capacity of each arrangement, based upon the performance of the unenclosed radiator, is given in the third from the last column on the right-hand page. The last two columns give the amount of radiation, in per cent, to be added or subtracted when using the particular enclosure, shield, or cover indicated, in order to obtain the same amount of steam condensed as would be obtained with the bare radiator.

11. *Performance of Unenclosed Radiator.*—In all cases, the performance was defined by a curve with room temperatures at the breathing line used as abscissas and net pounds of steam condensed in radiator per hour as ordinates. On each of the curve sheets shown in Figs. 6-8, 11-13, 17, and 20, the performance of the unenclosed radiator is shown as a base curve for the purpose of making comparisons, and all direct comparisons are made with a room temperature of 80 deg. F. at the breathing line. The curves for the unenclosed radiator in Figs. 11-13, 17, and 20 were transferred directly from the ones in Figs. 6, 7, and 8 on which the experimental points are shown.

It may be noted from Fig. 8, curve A, that at a room temperature of 70 deg. F. at the breathing line, and with steam at 216 deg. F., the unenclosed radiator condensed 24.25 lb. of steam per hour. Using a value of 969.3 B.t.u., taken from Goodenough's Steam Tables, as the latent heat of steam, this weight represents a total heat transmission of 23 500 B.t.u. per hour. Since the temperature difference from steam to air is 146 deg. F., a value of $K_{70} = 1.610$ is obtained, where K_{70} is the coefficient of heat transmission, or the heat transmitted per square foot of radiation per degree temperature difference per hour for room temperature at the breathing line of 70 deg. F.

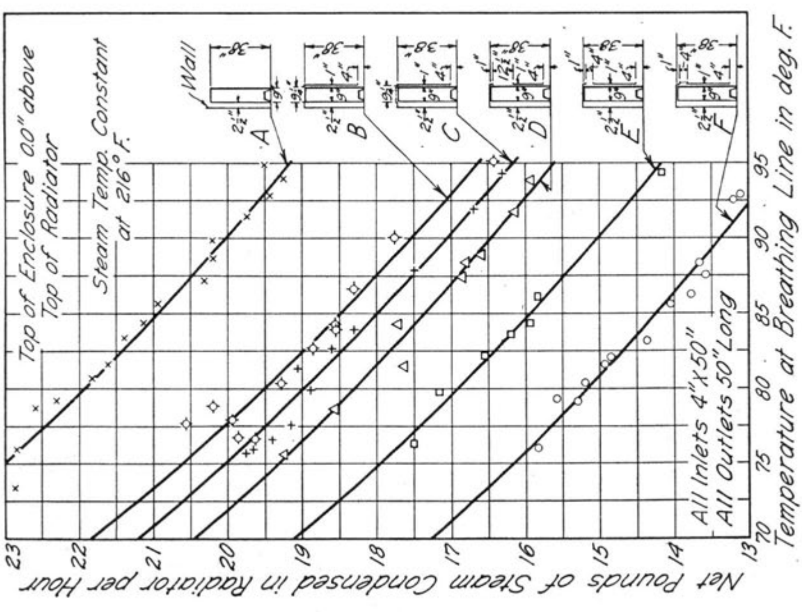

FIG. 6. PERFORMANCE CURVES FOR VARIOUS OUTLETS WITH MINIMUM HEIGHT ENCLOSURE

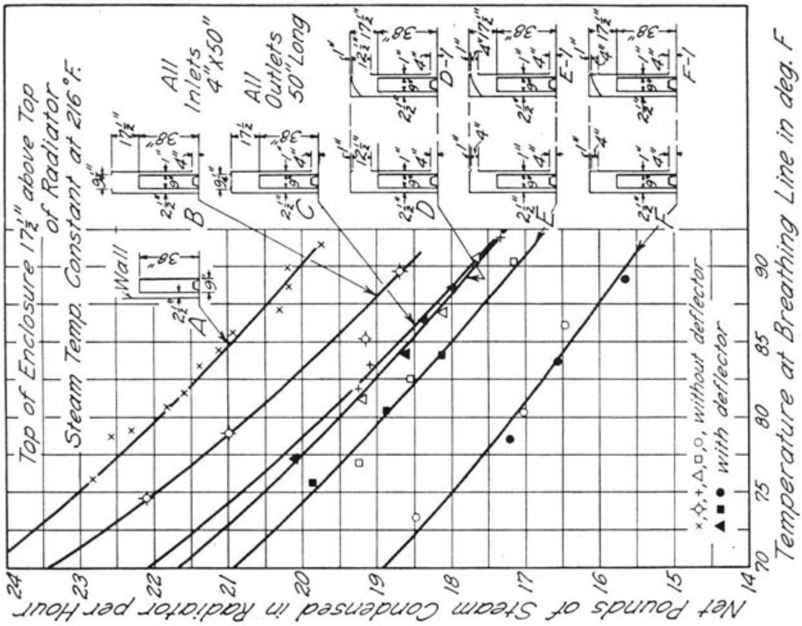

FIG. 7. PERFORMANCE CURVES FOR VARIOUS OUTLETS WITH MEDIUM HEIGHT ENCLOSURE

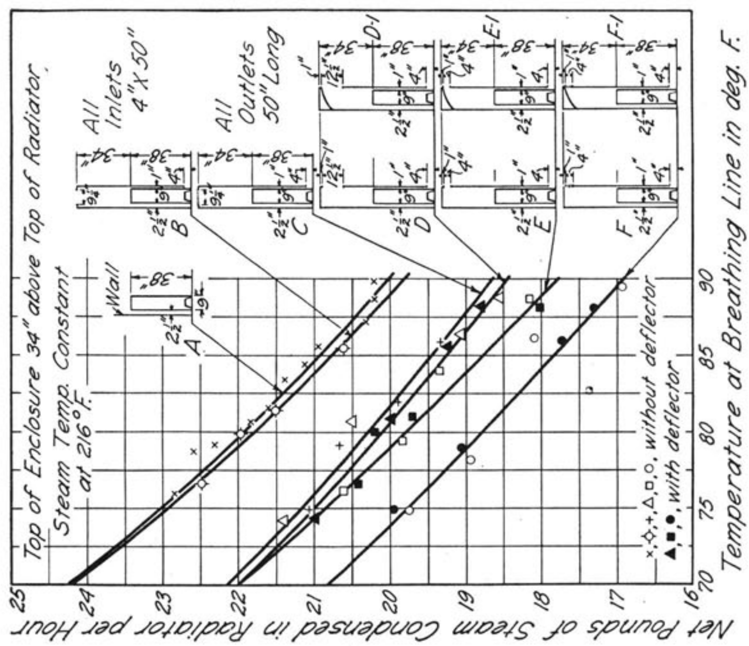

FIG. 8. PERFORMANCE CURVES FOR VARIOUS OUTLETS WITH MAXIMUM HEIGHT ENCLOSURE

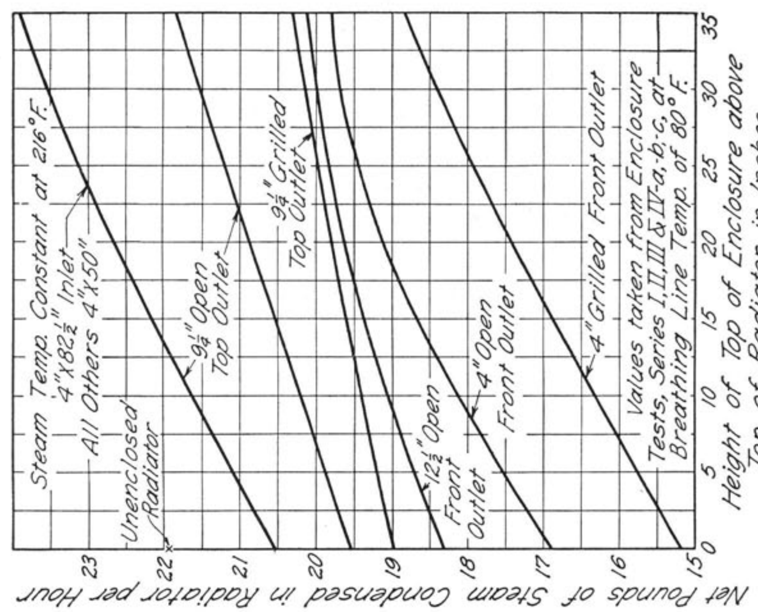

FIG. 9. EFFECT OF HEIGHT OF ENCLOSURE ON STEAM CONDENSED

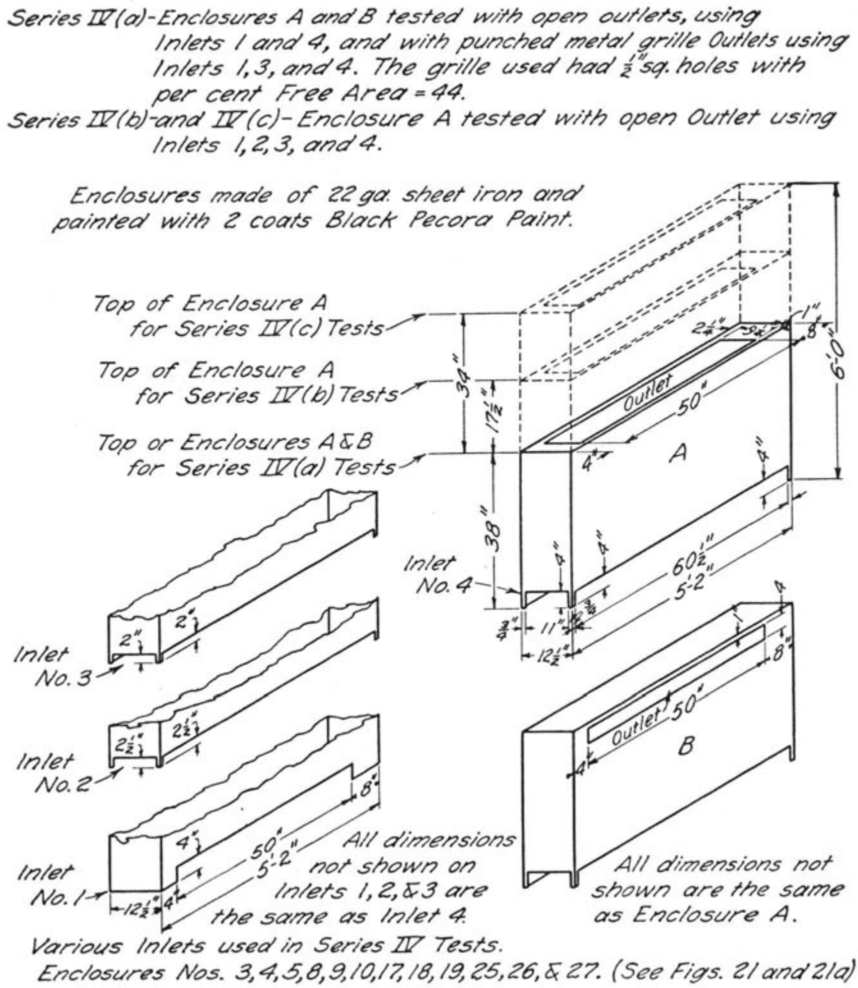

FIG. 10. ENCLOSURES USED IN SERIES IV-a, IV-b, AND IV-c TESTS

Reference to the table on page 37 of the Heating and Ventilating Guide for 1926-27 indicates that the rating of a 20-section, 38-inch, 3-column radiator with steam at 215 deg. F. and air at 70 deg. F. is 21 990 B.t.u. per hour. From the table on page 42 of the Guide, a correction factor of 0.993 may be obtained by interpolation. Making use of this factor, the 21 990 B.t.u. per hour may be corrected to terms of steam at 216 deg. F. and air at 70 deg. F. If this is done, a value of 22 140 B.t.u. per hour is obtained, and for the temperature differ-

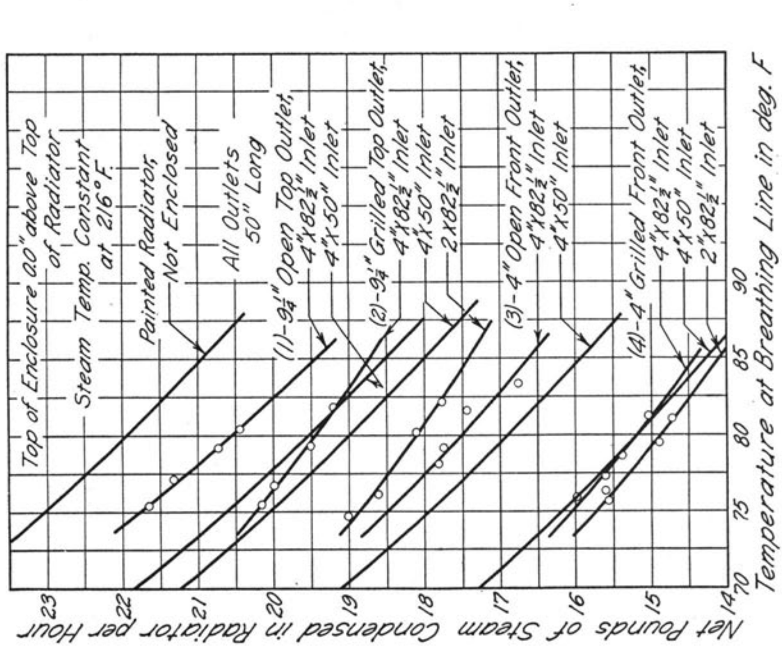

FIG. 11. PERFORMANCE CURVES FOR VARIOUS INLETS WITH MINIMUM HEIGHT ENCLOSURE

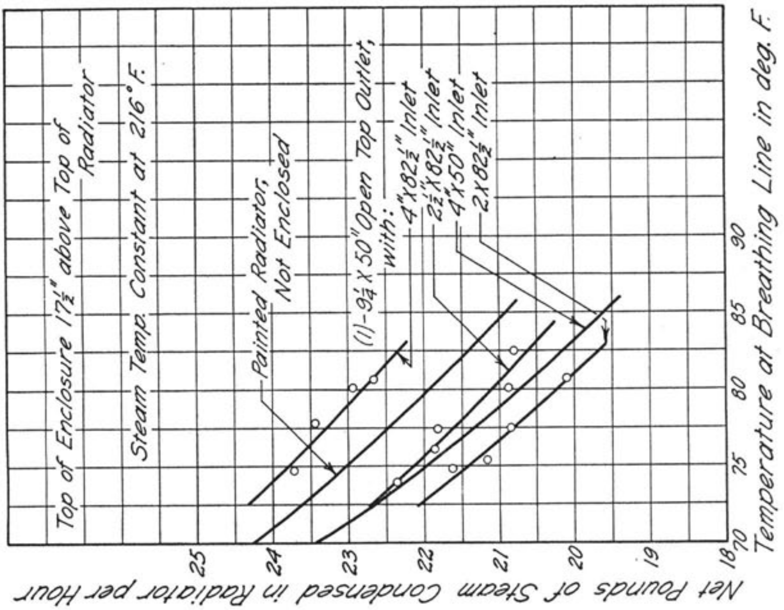

FIG. 12. PERFORMANCE CURVES FOR VARIOUS INLETS WITH MEDIUM HEIGHT ENCLOSURE

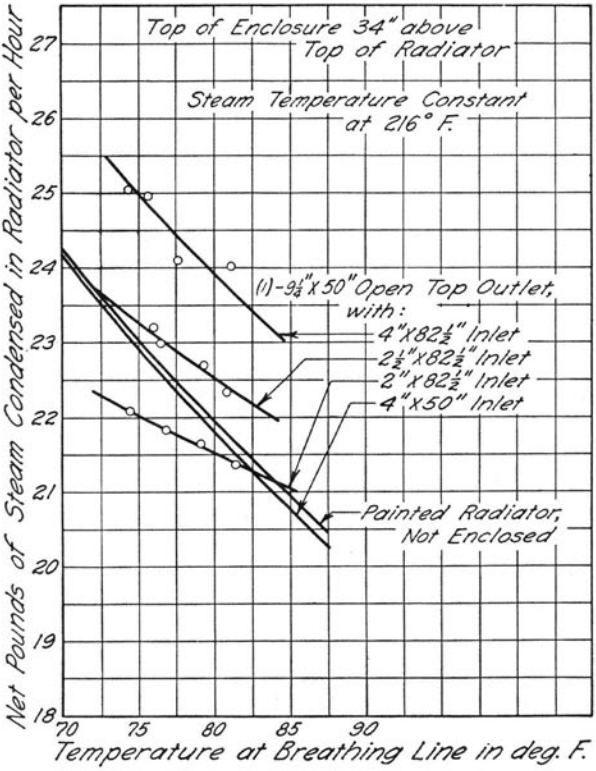

FIG. 13. PERFORMANCE CURVES FOR VARIOUS INLETS WITH MAXIMUM HEIGHT ENCLOSURE

ence of 146 deg. F., $K_{70} = 1.517$. Comparing this with the 1.610 obtained from curve A in Fig. 8, a difference of 6.1 per cent is indicated. This is considered a satisfactory agreement, particularly since the radiator tested was painted flat black, which would have a tendency to increase the heat transmission over that for an enameled or bare radiator.

Several rules have been proposed for correcting the heat transmission or the coefficient K to terms of a temperature range deviating from the standard range of 145 deg. F. A rule proposed by Charles A. Fuller* is that K, the coefficient of heat transmission, varies by 0.2 per cent per degree above or below the standard range. In order to examine the application of this rule to the data obtained on the tests reported in this bulletin, the value $K_{70} = 1.610$, with a temperature

*Harding, L. A., and Willard, A. C., "Mechanical Equipment of Buildings," Vol. I, First Edition, p. 78.

TABLE 1
HEAT TRANSMISSION COEFFICIENTS FOR VARIOUS TEMPERATURE RANGES

Room Temp. deg. F.	Steam Temp. deg. F.	Temp. Range deg. F.	Coefficient K		Difference	Per cent Difference
			Calculated from Curve A, Fig. 8	Corrected from Base Value 1.610		
70	216	146	1.610	1.610	0.000	0.000
75	216	141	1.584	1.584	0.000	0.000
80	216	136	1.563	1.578	+0.015	0.959
85	216	131	1.546	1.562	+0.016	1.035
90	216	126	1.542	1.546	+0.004	0.259

range of from 216 deg. F. to 70 deg. F., was chosen as a base. This was then corrected to several temperature ranges by applying the 0.2 per cent rule, and the results compared with those obtained by direct calculation from the curve for the unenclosed radiator shown in Fig. 8. The results are given in Table 1.

From the table it may be noted that the corrected values of the coefficient K agree within one per cent with those obtained by direct calculation from the curve in Fig. 8. Over the temperature range and at the actual temperatures used, the 0.2 per cent rule, therefore, seems to be well adapted to the data obtained. A second rule, proposed by Dr. Dietz,* is that the total heat transmission varies as the 1.3 power of the temperature range. That is

$$H = W_s r_s \left(\frac{t_{1a} - t_{2a}}{t_{1s} - t_{2s}}\right)^{1.3}$$

where

H = total heat transmission, B.t.u. per hour
W_s = weight of steam condensed at standard range, lb.
r_s = latent heat of steam at standard steam temperature, deg. F.
t_{1a} = actual steam temperature, deg. F.
t_{2a} = actual room temperature, deg. F.
t_{1s} = standard steam temperature, deg. F.
t_{2s} = standard room temperature, deg. F.

In order to compare the results of the application of these two rules to the actual test data, the following values have been calculated

*Brabbée, C. W., "Heating Effect of Radiation," Jour. A. S. H. and V. E., Vol. 31, No. 11, November, 1925, page 502.

for the heat transmission at a room temperature of 80 deg. F. with steam at 216 deg. F.:

(1) From Curve A, Fig. 8

$$H_{80} = 21.93 \times 969.3 = 21\ 250 \text{ B.t.u. per hour.}$$

(2) From the coefficient, K, corrected by the 0.2 per cent rule

$$H_{80} = 100 \times 1.578\ (216 - 80) = 21\ 460 \text{ B.t.u. per hour.}$$

(3) From the exponential rule

$$H_{80} = 24.25 \times 969.3\left(\frac{216-80}{216-70}\right)^{1.3} = 21\ 450 \text{ B.t.u. per hour.}$$

From these values it may be seen that the agreement between the two rules is very close and that both rules are applicable to the test data within one per cent, over the temperature range actually used in these tests.

12. *Effect of Character of Air Outlet.*—The group of tests designated as Series I were made for the purpose of determining the effect of the location, size, and type of the air outlet, using an enclosure of the same height as that of the radiator. The general details of the solid sheet iron enclosures used in these tests are given in Figs. 3 and 5, and also in Fig. 6, together with the corresponding result curves. The comparative data are based upon the performance of the enclosures with the temperature of the steam in the radiator at 216 deg. F. and the temperature at the breathing line at 80 deg. F. These data are tabulated graphically in Fig. 21 under enclosure Nos. 1, 2, 6, 7, and 11. In all cases the top of the enclosure practically touched the top of the radiator, and the air inlet consisted of a 4-inch by 50-inch open slot at the bottom of the front of the enclosure. The best combination in this series was obtained with a 9¼-inch by 50-inch horizontal opening in the top of the enclosure. The area of the outlet was 462.5 sq. in. compared with 200 sq. in. for the inlet. The performance of this combination is shown in curve B, Fig. 6, and the relative capacity was 89.1 per cent, using the steam condensing capacity of the unenclosed radiator at the 80-deg. F. room temperature at the breathing line as 100 per cent. A grille having a ratio of free to gross area of 44 per cent, or a total free area of 209.4 sq. in., was then placed in the outlet, and the performance is shown by curve C. The relative capacity was reduced to 86.5 per cent. Hence adding the grille reduced the performance of the same enclosure 2.9 per cent.

Curve D, Fig. 6, shows the performance when a 12½-inch by 50-inch vertical outlet having an area of 625.0 sq. in. was used at the top of the front of the enclosure. The relative capacity of this combination was 83.5 per cent. When the area of this outlet was reduced to 4 inches by 50 inches, or 200 sq. in., as shown by curve E, the relative capacity was reduced to 77.0 per cent. The addition of a grille, see curve F, reduced the relative capacity to 69.2 per cent. In this case the effect of the grille on the same combination was to reduce the performance 10.1 per cent.

From these results it is apparent that the use of an enclosure materially affects the steam condensing capacity of a radiator, and that the use of a front vertical outlet, even with a comparatively large area, is less advantageous than the use of a top outlet. A grille causes more serious reduction when used with a front outlet than with a top outlet. Referring to Fig. 6 it may be noted that the performance curves are all practically parallel to each other and to the base curve of the unenclosed radiator. Therefore, although all of the values used in the discussion were based upon the performance with a breathing line temperature of 80 deg. F., the relative percentages quoted are approximately the same over the entire range of temperatures included in the tests, and may be used for all practical purposes. This is not true for any other series of tests.

13. *Effect of Height of Enclosure.*—The two groups of tests made in addition to those of Series I, for the purpose of determining the effect of the height of the enclosure, were known as Series II and III. This work was also continued under Series IV-a, IV-b, and IV-c tests, in connection with the tests made for studying the effect of size of air inlet. The general arrangements of the enclosures used are shown in Figs. 5 and 10 and in Figs. 6, 7, and 8 corresponding to the performance curves. In every case the enclosures had solid ends and front, and in the tests of Series I, II, and III a 4-inch by 50-inch open inlet at the bottom of the front. In the tests made under Series IV-a, IV-b, and IV-c the inlet was 4 inches by 82½ inches, extending across the entire bottom of the front and ends.

The results obtained in Series I are shown in Fig. 6 and were discussed in Section 12. Figure 7 shows the results of the tests included in Series II made with enclosures 55½ inches high, and having a clearance of 17½ inches between the top of the radiator and the top of the enclosure. Figure 8 shows the results of the third series of tests made with enclosures 72 inches high, and having a clearance of 34 inches between the top of the radiator and the top of the enclosure. With the exception of the increase in height, these enclosures were exactly similar in every respect to those used in Series I. The difference in relative capacity between the enclosures having similar inlet and outlet characteristics was therefore due directly to the variation in height. In order to facilitate comparison, the results of the tests involving different heights have been grouped and are shown in Table 2. The relative capacities are based on the performance of both the enclosed and the unenclosed radiator at a breathing line temperature of 80 deg. F., and for other performances not at the given temperature range just stated direct reference may be made to the curves shown in Figs. 6, 7, 8, 11, 12, and 13. The results of the six groups of tests given in Table 2 show the consistent increase in capacity with height.

The results of all of these tests are shown in slightly different form in Fig. 9 in order to permit interpolation for heights other than the ones tested. These curves indicate that the steam condensing capacity of an enclosed radiator is appreciably increased by increasing the height of the enclosure, but that the height increases at a greater rate than the performance. The last group of tests given in Table 2 shows that with the best combination of inlets and outlets the capacity of the enclosed radiator was greater than that of the unenclosed radiator for both the 55½-inch and the 72-inch heights. It may be noted by comparing the three groups of curves in Figs. 6, 7, and 8 that as the height of enclosure is increased the total spread of the curves becomes less, thus indicating that disadvantageous outlets have less effect when used with high enclosures than when used with lower ones.

In conjunction with the tests in Series II and III, six groups of tests were made to determine whether a deflector in the top of the enclosure had any effect on the steam condensing capacity of the radiator. The same metal deflector, extending the full length of the enclosure, was used with each of the three front outlets shown corresponding to the performance curves D-1, E-1, and F-1, Figs. 7 and 8. The results obtained when the deflector was used are shown together with those obtained when it was not used, and it may be seen that the deflector did not have any appreciable effect. It was possible, therefore, in every case to represent the performance of the enclosure with and without the deflector by means of a single curve.

14. *Effect of Size of Air Inlet.*—The greater portion of the work done for the purpose of determining the effect of size of air inlet was included in the tests of Series IV-a with enclosures 38 inches high, but some additional tests were also made in Series IV-b and IV-c in connection with the study of effect of height of enclosure. The enclosures used in Series IV-b were 55½ inches high and those used in Series IV-c were 72 inches high. In all cases the enclosures with solid ends and fronts were used. By referring to Fig. 11, which shows the results of

TABLE 2
RELATIVE STEAM CONDENSING CAPACITIES OF ENCLOSURES OF DIFFERENT HEIGHTS

Inlet	Outlet	Enclosure Number*	Tests	Height of Enclosure in.	Relative Capacity per cent
4 in. x 50 in. at the bottom of the front	9¼ in. x 50 in. Open Top	1	Series I	38	89.1
		15	Series 11	55½	94.4
		23	Series III	72	99.3
	9¼ in. x 50 in. Grilled Top	2	Series I	38	86.5
		16	Series II	55½	89.9
		24	Series III	72	92.5
	12½ in. x 50 in. Open Front	11	Series I	38	83.5
		22	Series II	55½	89.0
		30	Series III	72	91.7
	4 in. x 50 in. Open Front	6	Series I	38	77.0
		20	Series II	55½	86.2
		28	Series III	72	90.2
	4 in. x 50 in. Grilled Front	7	Series I	38	69.2
		21	Series II	55½	78.3
		29	Series III	72	85.6
4 in. x 82½ in. across front and ends	9¼ in. x 50 in. Open Top	3	Series IV-a	38	93.6
		17	Series IV-b	55½	104.0
		25	Series IV-c	72	109.1

*These numbers refer to the enclosure numbers given in Figs. 21 and 21-a, and the performances are shown there diagrammatically.

Series IV-a tests, it may be noted that several different inlets were used in conjunction with four types of outlets. Two different inlets were used with the 9¼-inch by 50-inch open top outlet and the 4-inch by 50-inch open front outlet. Three inlets were used with the 9¼-inch by 50-inch grilled top outlet and the 4-inch by 50-inch grilled front outlet. The performances of these enclosures with the various combinations of inlets and outlets for a room temperature at the breathing line of 80 deg. F. are shown in Fig. 21 under enclosures Nos. 1 to 10. Figures 12 and 13 show the results obtained with enclosures 55½ inches and 72 inches high, using four different inlets with a 9¼-inch by 50-inch open top outlet, and the capacities for a breathing line temperature of 80 deg. F. are given in Fig. 21-a, under enclosures Nos. 15, 17, 18, 19, 23, 25, 26, and 27. Since the slopes of the performance curves in Series IV-a, IV-b, and IV-c vary considerably, the relative capacities given in Figs. 21 and 21-a are applicable only to the temperature ranges indicated, and in order to determine the capacities at other ranges, direct reference must be made to the original curves given in Figs. 11, 12, and 13.

Again referring to Fig. 11, and considering any group of curves for an enclosure with a given outlet, it may be noted that the relative positions and the slopes of the curves vary for different inlets. To point out and explain this fact the second group of curves may be used as an example. In all of the three cases, the enclosure had a grilled top outlet 9¼ inches by 50 inches. This outlet had a total free area of 209.4 sq. in. Three types of inlets were used. The first consisted of a 4-inch by 60½-inch slot across the bottom of the front and a 4-inch by 11-inch slot across the bottom of each end of the enclosure. This inlet had a total free area of 330 sq. in. and a total perimeter of 189 inches. The second was 4 inches by 50 inches, with an area of 200 sq. in., and a perimeter of 108 inches. The third was similar to the first except that the slots were two inches wide instead of four inches. This inlet had a total free area of 165 sq. in. and a total perimeter of 177 inches. Since the outlet was always the same, the inlet was the determining factor and the relative capacities obtained with a breathing line temperature of 80 deg. F., for enclosures 2, 4, and 5, as shown in Fig. 21, were 86.5 per cent, 88.6 per cent, and 82.5 per cent, respectively. It may be noted that these relative capacities were in the order of the areas of inlet, with the maximum relative capacity corresponding to the maximum area. It may also be noted that the slope of the curve for the enclosure with the 4-inch by 50-inch inlet was greater than that for those with the 4-inch by 82½-inch and the 2-inch by 82½-inch inlets, which had practically the same slope. The following may serve as a possible explanation for this difference in performance.

The difference in performance for the three enclosures is undoubtedly caused by a difference in the frictional resistances and free areas of the inlets. The expression for head lost due to friction may be assumed to be of the form:

$$h = \frac{f\,V^2\,L\,P}{2\,g\,A}$$

where

$h =$ the head lost in feet of fluid flowing
$f =$ the coefficient of friction
$V =$ the velocity in ft. per sec.
$L =$ the length of duct (or thickness of plate in this case) in feet
$P =$ the perimeter in feet
$A =$ the area in sq. ft.
$g =$ the acceleration due to gravity $= 32.2$ feet per sec. per sec.

Assuming the same motive head, or temperature difference between air and steam, the area of the inlet orifice would determine the actual velocity through the inlet. As the area increased, the velocity would decrease. Since the head lost varies as $\frac{V^2 P}{A}$ and the fraction $\frac{V^2}{A}$ would decrease at a greater rate than P increased, it is reasonable to presume that $\frac{V^2 P}{A}$, and hence the friction loss, h, would decrease as A is increased. Therefore A would be the factor that determines the relative positions of the three curves under discussion. This reasoning is substantiated by the fact that the maximum relative capacity was obtained with the maximum area of inlet, and the other relative capacities were also in the same order as the inlet areas.

In considering any one curve alone, it is evident that since A is constant, the head lost varies as V^2P. As the room temperature decreases, the available motive head, and hence the velocity, increases, and the condensation becomes greater. But the product V^2P also increases with the velocity, thus tending to increase the head lost and to reduce the condensation below what would be obtained if friction did not exist. Therefore the product V^2P determines the slope of the curves, and the smaller slopes correspond to the greater values of V^2P. It is evident that if P is large, the slope will be small. This reasoning is substantiated also by the curves, since the 4-inch by 50-inch inlet with a perimeter of 108 inches gave a curve having the greatest slope, while the two others with perimeters of 189 inches and 177 inches, respectively, gave curves with practically the same slope, and less than that for the 4-inch by 50-inch inlet.

The results of these tests indicate that the air inlet is an important factor in the performance of an enclosure, and that it should extend across the entire front and ends. This is shown by comparing the performance of any one of the enclosures, using first a 4-inch by 50-inch inlet, and then a 4-inch by 82½-inch inlet, all other factors remaining the same. The ratio of the perimeter to the area is also of importance, and long narrow inlets are not desirable.

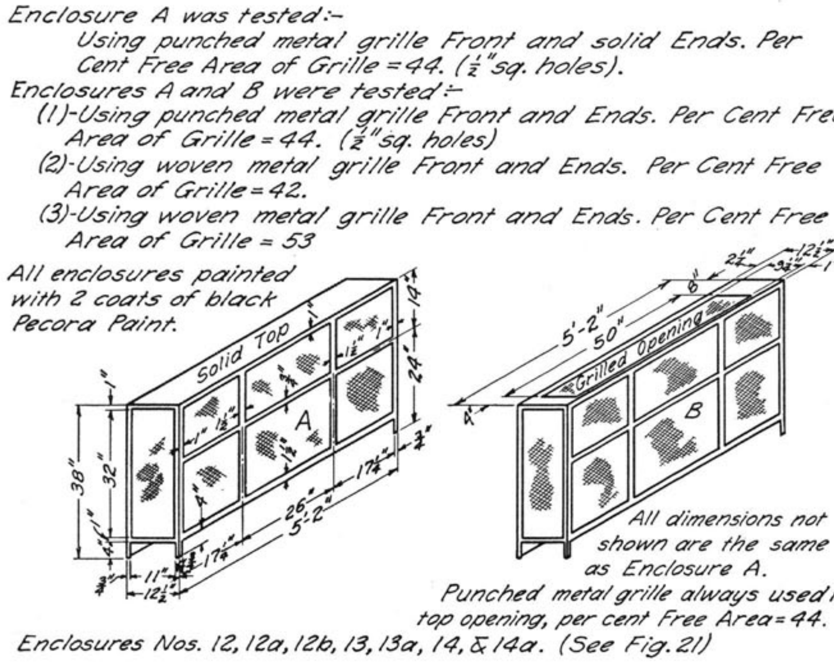

FIG. 14. ENCLOSURES USED IN SERIES V TESTS

15. *Performance of Enclosures With Grilled Construction.*—The group of tests made to show the effect of enclosing the radiator with enclosures having full grilled construction is designated as Series V, and the general arrangements of the enclosures may be seen in Figs. 14 and 15. The grilles used were: a woven metal grille having a ratio of free area to gross area of 53 per cent, a woven metal grille having a free area ratio of 42 per cent, and a punched metal grille having a free area ratio of 44 per cent. In computing the free areas of the various grilles, all the openings, both large and small, were included in the total free areas. Figure 16 shows the actual size and construction of the different grilles.

Each grille was tested on two types of enclosures. One type had a 9¼-inch by 50-inch grilled opening in the top, while the other type had a solid top. One enclosure was constructed with solid ends and top, with the punched metal grille covering the front. Both the sheet iron and the grille work extended to within 4 inches of the floor. On all other enclosures the grille work began 4 inches above the floor and extended over the entire front and ends.

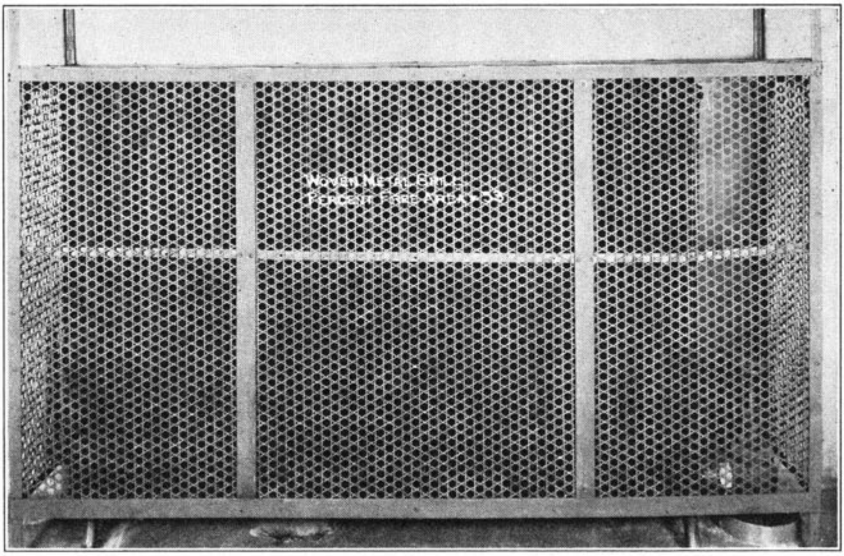

FIG. 15. ENCLOSURE WITH WOVEN METAL GRILLE CONSTRUCTION

The results of the various tests are given by the curves in Fig. 17, and the tabulated values for enclosure Nos. 12, 12-a, 12-b, 13, 13-a, 14, and 14-a, in Fig. 21. The enclosure with the outlet in the top, and covered with the woven metal grille having a free area ratio of 53 per cent, gave a relative capacity of 93.1 per cent for a room temperature at the breathing line of 80 deg. F. This same enclosure with a solid top gave a relative capacity of 85.5 per cent, the solid top reducing the capacity approximately 8.2 per cent. Retaining the same general construction, a woven metal grille with 42 per cent free area ratio was substituted in place of the woven metal grille with 53 per cent free area. In the case of the enclosure with the outlet in the top, a relative capacity of 91.8 per cent was obtained, and with the solid top the relative capacity was 84.2 per cent. In this case the substitution of the solid top in place of the top with the grilled opening caused a reduction in capacity of 8.3 per cent, which is approximately the same reduction as obtained when the enclosures were covered with the woven metal grille having a free area ratio of 53 per cent. The third construction tested was similar to the first two with the exception that a punched metal grille having a free area ratio of 44 per cent was used in place of the woven metal grilles. Referring to Fig. 17, it may be noted that there was no appreciable difference between the perform-

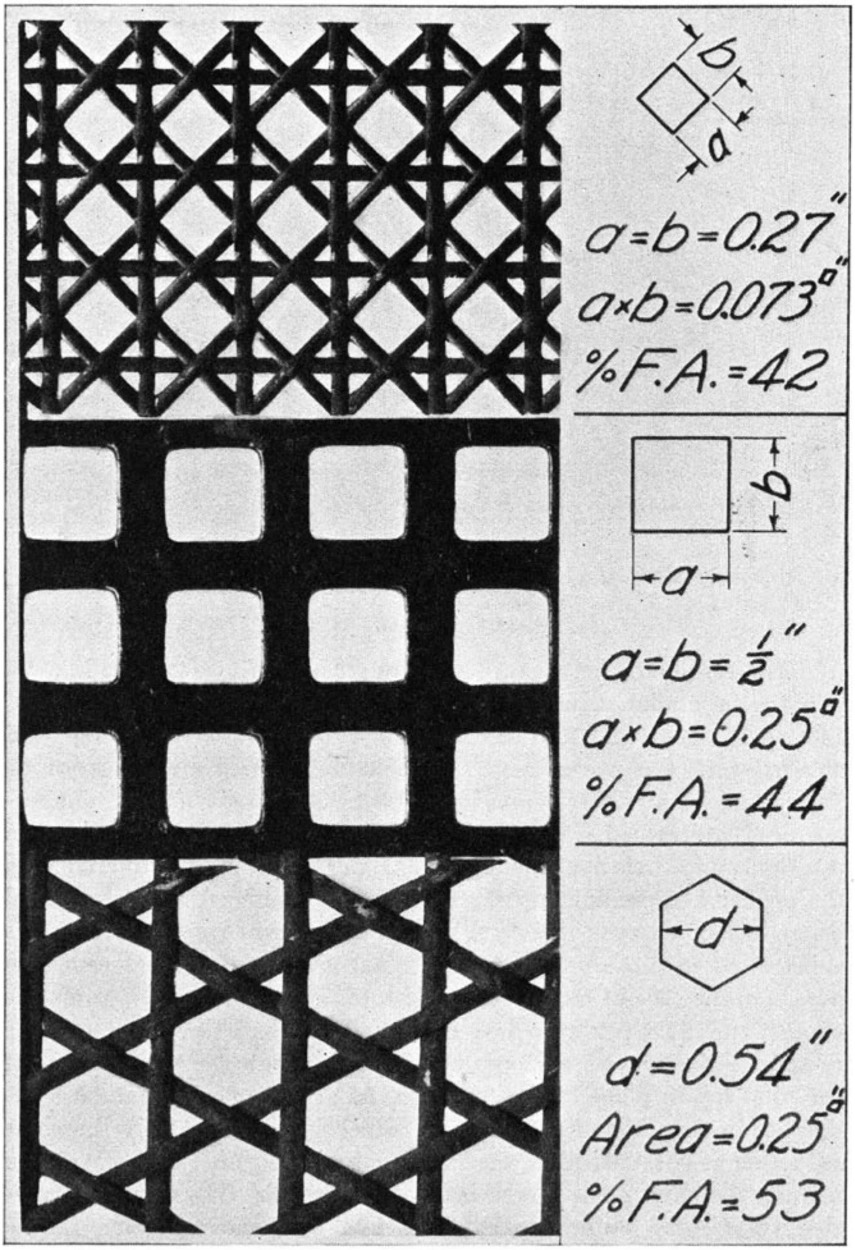

FIG. 16. SECTIONS OF METAL GRILLES, ACTUAL SIZE

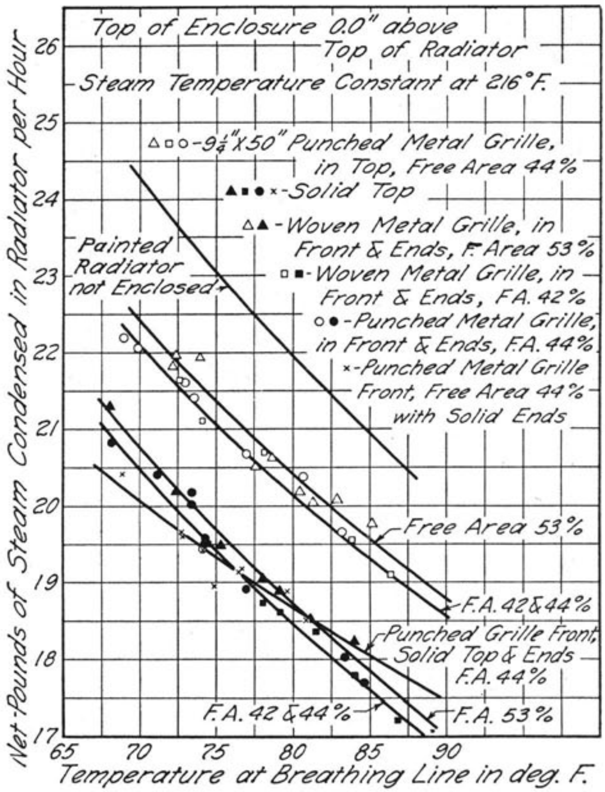

FIG. 17. PERFORMANCE CURVES FOR ENCLOSURES WITH VARIOUS GRILLED CONSTRUCTIONS

ance of this enclosure and the one covered with the woven grille having a 42 per cent free area ratio. This was true of both the enclosure having an outlet in the top and the one with a solid top, and in each case the same curve served to represent the performance of the enclosure covered with either grille. The fourth construction tested was the one in which the enclosure had solid ends, with a punched grille front having a free area ratio of 44 per cent. This combination of front and ends was used with the solid top only and the results obtained are shown in Fig. 17 by the curve having the least slope. This curve indicates that for low motive heads, or small temperature differences between the radiator and the surrounding air, the enclosure with solid ends has a greater capacity than similar enclosures with ends of grilled construction. However, since the slope of this curve is much less than the slope of the performance curves for similar enclosures

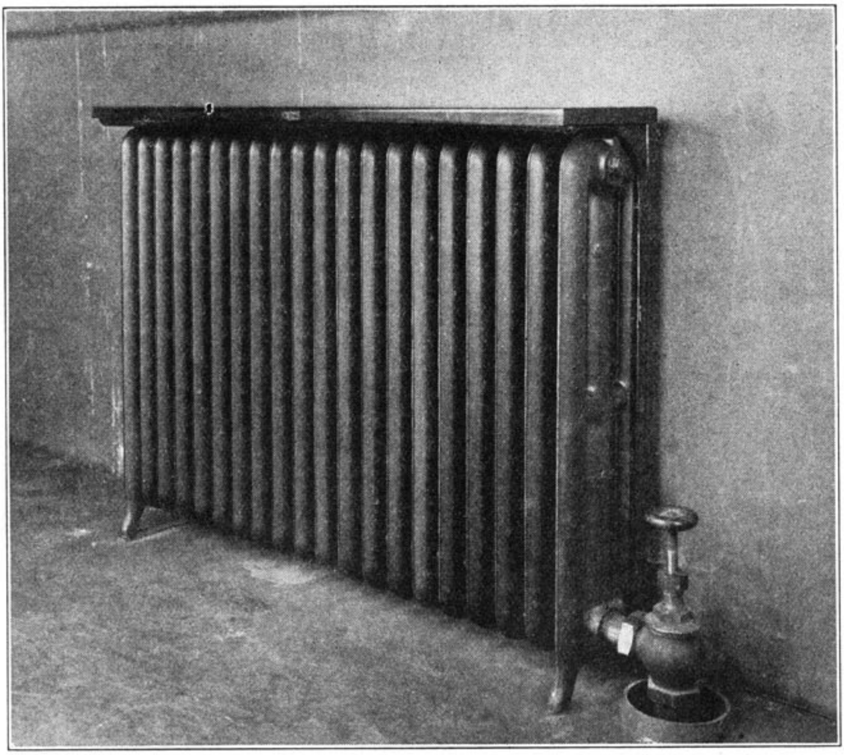

FIG. 18. METAL RADIATOR SHIELD

having grilled ends, the results obtained over the whole temperature range at which the tests were made show that for the motive heads found in common practice the grilled ends have a small advantage.

For a room temperature at the breathing line of 80 deg. F. the relative capacity of the enclosure with the grilled construction having a 53 per cent free area ratio was only 1.3 per cent more than it was with the grilles having either 42 or 44 per cent free area ratio. This was true for both types of enclosures, the one with the outlet in the top and the one with the solid top. Thus it may be seen that, although the difference in the free area ratios of the grilles used was as great as 11 per cent, the difference in the relative capacities was only 1.3 per cent. This is very small in comparison with the effects of other factors, such as height, air outlets, and air inlets.

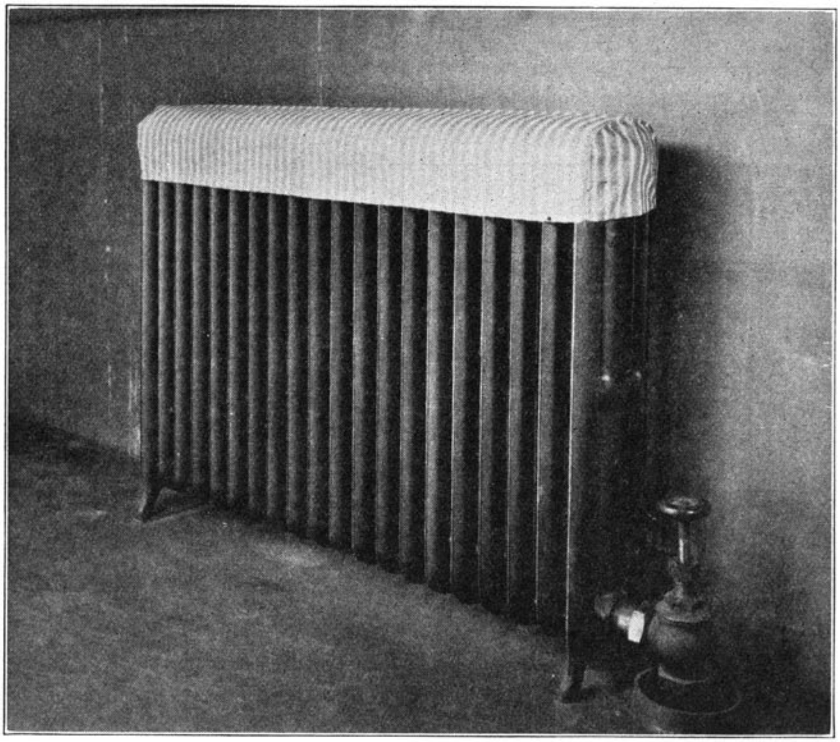

FIG. 19. RADIATOR WITH CLOTH COVER

The enclosures used in this series of tests were very close approximations to the usual type of commercial enclosures, and while some improvement over the types having solid front and sides with slotted inlets and outlets was indicated, it is evident that any enclosure, unless extended to a considerable height above the top of the radiator, will reduce the condensation that would normally be expected from the bare radiator.

16. *Performance of Metal Shield and Cloth Cover.*—Tests of Series VI were made for the purpose of determining the effect of a common commercial type of metal radiator shield and a cloth radiator cover. The shield and cover tested are shown in Figs. 18 and 19, and again in Fig. 20 corresponding to the result curves. A general description of both shield and cover was given in Section 8.

Curve B, Fig. 20, shows the results of the tests made on the shield, and curve C the results of those made on the cloth cover. With a room temperature at the breathing line of 80 deg. F., the relative capacity

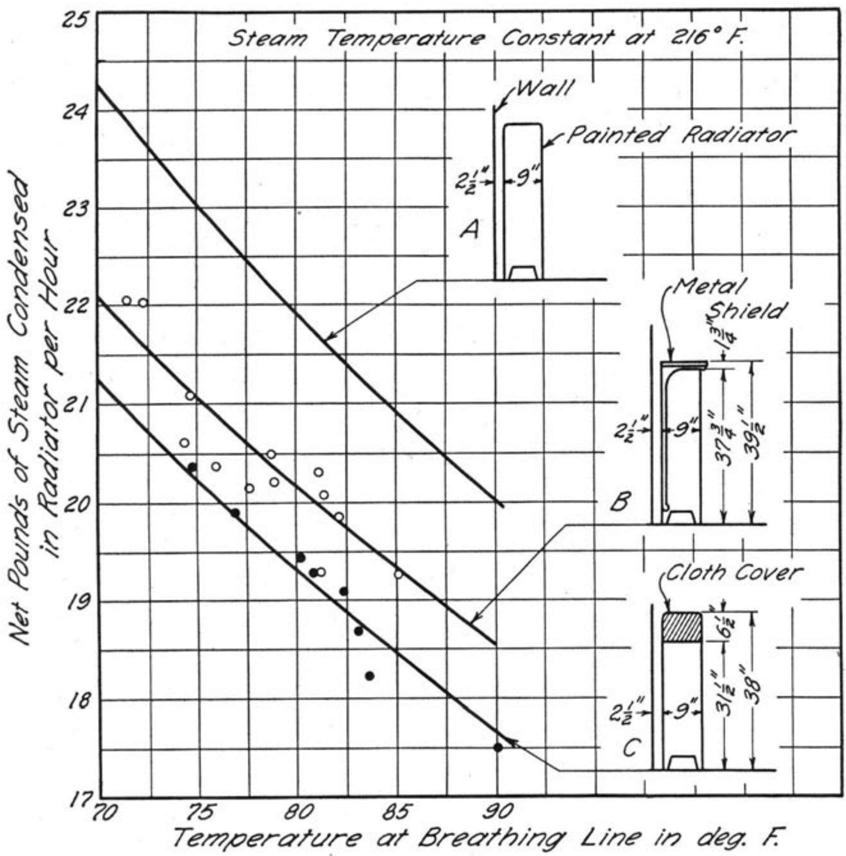

FIG. 20. PERFORMANCE CURVES FOR METAL SHIELD AND CLOTH COVER

of the radiator with the shield was 91.9 per cent, comparing favorably with the relative capacities of 91.8, 91.8, and 93.1 per cent for the enclosures with grilled construction having grilled openings in the top. However, when this is compared with the relative capacities of 84.2, 84.2, and 85.5 per cent for the enclosures with grilled construction having solid tops, the effect of the shield is seen to be less disadvantageous than the effects of enclosures which are very close approximations to the commercial types. The relative capacity of the radiator with a cloth cover for a room temperature at the breathing line of 80 deg. F. was 88.0 per cent. These values are tabulated in Fig. 21-a, under shield No. 1, and cover No. 1.

V. Conclusions

17. *Conclusions.*—As a result of this investigation the following conclusions were drawn:

(1) The one-pipe connection to the radiator, in which the steam enters and the condensate leaves through the same connection, gives very consistent and satisfactory results when used for the determination of the steam condensing capacity of radiators.

(2) The coefficient of heat transmission K, or the heat transmitted per sq. ft. of radiation per degree temperature difference per hour, may be corrected to terms of a temperature range deviating (within reasonable limits) from the standard range of 145 deg. F., either by the 0.2 per cent rule proposed by Charles A. Fuller* or by the exponential rule proposed by Dr. Dietz.†

(3) The type of air outlet used materially affects the performance of a radiator enclosure, and a front outlet, even with a comparatively large area, is less advantageous than a top outlet.

(4) The capacity of an enclosed radiator is appreciably increased by increasing the height of the enclosure, but, for a given radiator, an increase in height with low enclosures results in a greater gain in steam condensing capacity than a corresponding increase in height with high enclosures.

(5) By using the proper combination of inlets and outlets and increasing the height of the enclosure a sufficient amount, it is possible to increase the steam condensing capacity of the enclosed radiator above that of the unenclosed radiator.

(6) Disadvantageous outlets have less effect when used with high enclosures than when used with lower ones.

(7) The frictional resistance in the air inlet is an important factor in the design of an enclosure, and long narrow inlets having large ratios of perimeter to area are undesirable.

(8) Enclosures which were very close approximations of common commercial types reduced the relative capacity of the radiator approximately 15 per cent.

(9) The common commercial shield tested reduced the relative capacity of the radiator about 8 per cent.

(10) The cloth cover tested reduced the relative capacity about 12 per cent.

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*Harding, L. A., and Willard, A. C., "Mechanical Equipment of Buildings," Vol. I, First Edition, page 78.

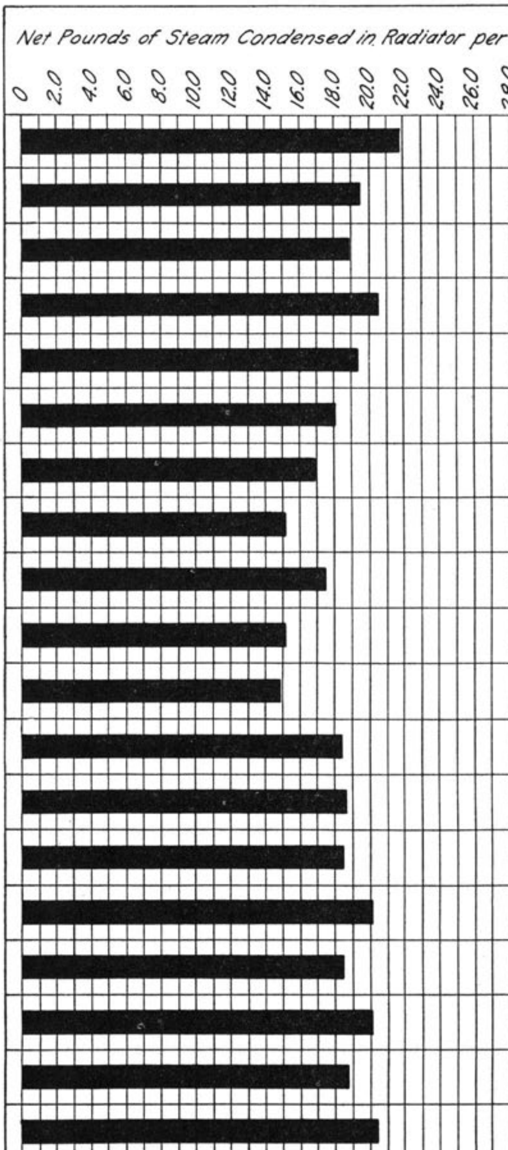
†Brabbée, C. W., "Heating Effect of Radiation," Journal of the A. S. H. and V. E., Vol. 31, No. 11, November, 1925, page 502.

Enclosure Number	Description of Enclosure: Height of Top of Enclosure Above Top of Radiator	Inlet	Outlet	Free Area of Inlet sq. in.	Free Area of Outlet sq. in.
	Radiator not enclosed—Painted with two coats black Pecora paint (Not dipped)			——	——
1	0.0"	4"x50" Across front	$9\frac{1}{4}$"x50" Open top	200.0	462.5
2	0.0"	4"x50" Across front	$9\frac{1}{4}$"x50" Grilled top	200.0	209.4
3	0.0"	4"x$82\frac{1}{2}$" Across front and ends	$9\frac{1}{4}$"x50 Open top	330.0	462.5
4	0.0"	4"x$82\frac{1}{2}$" Across front and ends	$9\frac{1}{4}$"x50" Grilled top	330.0	209.4
5	0.0"	2"x$82\frac{1}{2}$" Across front and ends	$9\frac{1}{4}$"x50" Grilled top	165.0	209.4
6	0.0"	4"x50" Across front	4"x50" Open front	200.0	200.0
7	0.0"	4"x50" Across front	4"x50" Grilled front	200.0	92.1
8	0.0"	4"x$82\frac{1}{2}$" Across front and ends	4"x50" Open front	330.0	200.0
9	0.0"	4"x$82\frac{1}{2}$" Across front and ends	4"x50" Grilled front	330.0	92.1
10	0.0"	2"x$82\frac{1}{2}$" Across front and ends	4"x50" Grilled front	165.0	92.1
11	0.0"	4"x50" Across front	$12\frac{1}{2}$"x50" Open front	200.0	625.0
12	0.0"	Punched grill front, solid ends and top. Grill-work beginning 4" above floor. Free area grill, 44 %. $\frac{1}{2}$"sq. holes.			
12a	0.0"	Same as 12 with punched grill ends			
12b	0.0"	Same as 12a with $9\frac{1}{4}$"x50" grilled top outlet ($\frac{1}{2}$"sq. holes)			
13	0.0"	Woven metal grill front and ends, solid top. Grill-work beginning 4" above floor. Free area grill 42 %.			
13a	0.0"	Same as 13 with $9\frac{1}{4}$"x50" grilled ($\frac{1}{2}$"sq. holes) top outlet			
14	0.0"	Woven metal grill front and ends, solid top. Grill work beginning 4" above floor. Free area grill, 53 %			
14a	0.0"	Same as 14 with $9\frac{1}{4}$"x50" grilled ($\frac{1}{2}$"sq. holes) top outlet			

Note: 20-Section, 3 Col., C.I., Water Type Radiator (Direct) Used in all Tests.

21-1

FIG. 21. GRAPHICAL TABLE OF RESULTS



Net Pounds of Steam Condensed in Radiator per Hour	Relative Condensing Capacity Per Cent	When Using Enclosure, % Radiation To Add	When Using Enclosure, % Radiation To Subtract
21.93	100.0		
19.53	89.1	12	
18.98	86.5	16	
20.53	93.6	7	
19.43	88.6	13	
18.09	82.5	21	
16.89	77.0	30	
15.17	69.2	45	
17.50	79.8	25	
15.17	69.2	45	
14.87	67.8	47	
18.31	83.5	20	
18.67	85.1	17	
18.47	84.2	19	
20.14	91.8	9	
18.47	84.2	19	
20.14	91.8	9	
18.74	85.5	17	
20.42	93.1	7	

Note: Steam Temperature Constant at 216°F. Breathing Line Temp. = 80°F.

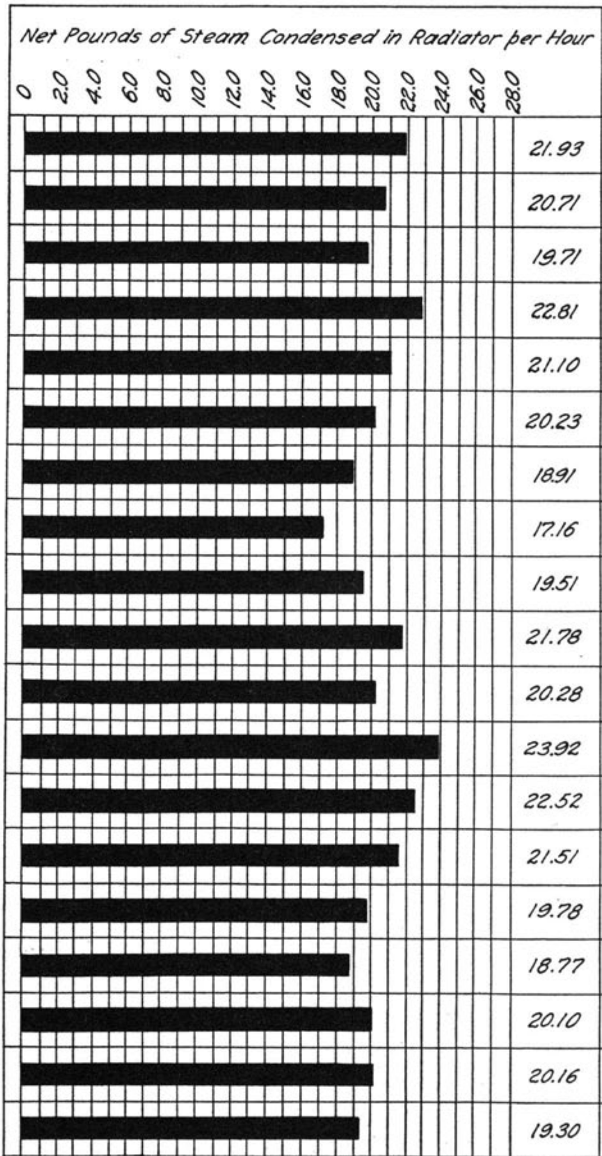
21-2

Enclosure Number	Description of Enclosure				
	Height of Top of Enclosure Above Top of Radiator	Inlet	Outlet	Free Area of Inlet, sq. in.	Free Area of Outlet, sq. in.
	Radiator not enclosed—Painted with two coats black Pecora paint. (Not dipped)				
15	17½"	4"x50" Across front	9¼"x50" Open top	200.0	462.5
16	17½"	4"x50" Across front	9¼"x50" Grilled top	200.0	209.4
17	17½"	4"x82½" Across front and ends	9¼"x50" Open top	330.0	462.5
18	17½"	2½"x82½" Across front and ends	9¼"x50" Open top	206.3	462.5
19	17½"	2"x82½" Across front and ends	9¼"x50" Open top	165.0	462.5
20	17½"	4"x50" Across front	4"x50" Open front, with and without deflector	200.0	200.0
21	17½"	4"x50" Across Front	4"x50" Grilled front, with and without deflector	200.0	92.1
22	17½"	4"x50" Across front	12½"x50" Open front, with and without deflector	200.0	625.0
23	34"	4"x50" Across front	9¼"x50" Open top	200.0	462.5
24	34"	4"x50" Across front	9¼"x50" Grilled top	200.0	209.4
25	34"	4"x82½" Across front and ends	9¼"x50" Open top	330.0	462.5
26	34"	2½"x82½" Across front and ends	9¼"x50" Open top	206.3	462.5
27	34"	2"x82½" Across front and ends	9¼"x50" Open top	165.0	462.5
28	34"	4"x50" Across front	4"x50" Open front, with and without deflector	200.0	200.0
29	34"	4"x50 Across front	4"x50" Grilled front, with and without deflector	200.0	92.1
30	34"	4"x50" Across front	12½"x50" Open front, with and without deflector	200.0	625.0
Shield No. 1	Metal Shield with Insulated Top; No Humidifier.				
Cover No. 1	Cover, 6½"x50", made of close woven crash cloth.				

Note: 20-Section, 3 Col., C.I., Water Type Radiator (Direct) Used in all Tests.

21-a-1

Fig. 21-a. Continuation of Graphical Table of Results



Net Pounds of Steam Condensed in Radiator per Hour	Relative Condensing Capacity, Per Cent	When Using Enclosure, % Radiation To Add	When Using Enclosure, % Radiation To Subtract
21.93	100.0		
20.71	94.4	6	
19.71	89.9	11	
22.81	104.0		4
21.10	96.2	4	
20.23	92.3	8	
18.91	86.2	16	
17.16	78.3	28	
19.51	89.0	12	
21.78	99.3	1	
20.28	92.5	8	
23.92	109.1		8
22.52	102.7		3
21.51	98.1	2	
19.78	90.2	11	
18.77	85.6	17	
20.10	91.7	9	
20.16	91.9	9	
19.30	88.0	14	

Note: Steam Temperature Constant at 216° F. Breathing Line Temp. = 80° F.

21-a-2

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