To: Mr. Chas. J. McCarthy

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ENERGY LOSS, VELOCITY DISTRIBUTION, AND TEMPERATURE DISTRIBUTION FOR A BAFFLED CYLINDER MODEL

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SUMMARY

A study has been made of the important principles involved in the operation of a baffle for an engine cylinder and shows that the cooling can be improved 20 percent by using a correctly designed baffle. Such a gain is as effective in cooling the cylinder with the improved baffle as a 65-percent increase in pressure drop across the standard baffle, which had a 1/4-inch clearance between baffle and fin tips.

INTRODUCTION

In the design of a cowling a certain pressure drop across the cylinders of a radial air-cooled engine is made available. Baffles are designed to make use of this available pressure drop for cooling.

The problem of cooling an air-cooled engine cylinder has been treated, for the most part, from considerations of a large heat-transfer coefficient. The knowledge of the precise cylinder characteristics that give a maximum heat-transfer coefficient should be the first consideration. The next problem is to distribute this ability to cool so that the cylinder cools uniformly.

Hartshorn (reference 1) has made some measurements on energy loss and effect of baffle arrangement. His interesting discussion of the problem might be read with profit by anyone interested in the problem of engine cooling. The tests made by Hartshorn suffer from incompleteness, no doubt owing to the fact that the effect of baffle arrangement was only one part of a large problem.

The subject report takes up the problem of the design of a baffle for a model cylinder. The variables that affect the design of the baffle and the several variables in the baffle itself are studied. The results also show, to some extent, the effect of fin spacing and cylinder diameter.

Each actual engine-cylinder type is a separate problem that must be solved by tests in flight or in a large wind tunnel. A simple case is studied to determine the function and importance of each element in the baffle-cylinder arrangement. The fundamental ideas brought out in tests of the model cylinder will be of assistance in intelligently planning these tests and adjustments.

This study is not concerned with the cooling performance of the front of the cylinder. It has been shown in reference 2 that the front of the cylinder is cooled by large-scale turbulence. It follows, then, that the cooling performance of the baffled part of the cylinder has no relation to the cooling performance of the front of the cylinder except for the fact that the air entering the baffle is slightly heated by the front of the cylinder.

ANALYSIS OF THE PROBLEM

The energy loss of the air flow around the baffled part of a finned cylinder as it is commonly installed on an engine is of the same character as the energy loss through a venturi tube.

Figure 1 shows a venturi tube with $P_0 V_0$ corresponding to the entrance of the baffle where the stream velocity is, for all practical purposes, zero. The value of $P_1 V_1$ corresponds to the finned part of the cylinder. It is apparent that V_1 is large in this region and that $P_1 + \frac{1}{2} \rho_1 V_1^2$ at the entrance to the fins is equal to P_0 assuming $V_0 = 0$. The value of P_1 varies throughout the restricted passage. Near the exit

$$P_0 = P_1' + \frac{1}{2} \rho_1 V_1^2 + l\, f(V)$$

where $f(V)$ is the loss in energy per unit length due to viscous friction and turbulence.

$$P_1 = P_1' + l\, f(V)$$

The expansion of the air to the condition $P_2 V_2$, where V_2 is approximately zero, is the most troublesome part of the operation, the object being to reconvert as much as possible of the kinetic energy to potential energy. It is well known that if the tube is ended at $P_1' V_1'$ the entire kinetic energy is lost. If a suitable expansion of the fluid is accomplished, an appreciable proportion of this energy may be reconverted to potential energy. In the case of the actual engine, the front and rear of the cylinder, corresponding to the regions $P_0 V_0$ and $P_2 V_2$, respectively, have a constant difference in pressure for a given cowling and at a given air speed. If by a suitable expansion V_2 is made small, the value of $\frac{1}{2} \rho_2 V_2^2$ will be small. In the ideal case, where $P_0 - P_2 = l\ f(V)$ and it is assumed that $V_2 = V_0 = 0$, the air passes through the restricted tube with a relatively large velocity and dissipates its entire available energy in friction. This condition would correspond to the maximum cooling possible on the baffled part of the cylinder.

In the arrangement shown in figure 1 the chance of approaching the ideal flow is considerably greater than in an actual case. Figure 2(a) shows the essential parts of the arrangement of the practical case. The entrance conditions of an actual cylinder can be made as good as in the simplified case, but from that point on the actual finned cylinder operates at a disadvantage. Two streams or jets come together from opposite directions at the point where a complete expansion of the jets would take place if there were no exit duct on the baffle. The two jets must come together from opposite directions because the rear of the cylinder must be cooled. It can be seen that, if the air were permitted to leave the cylinder farther to the front, the two jets could be brought together at a more acute angle. Thus a smaller loss in energy would result because the jets would not be required to make so sharp a turn and the mixing loss might be less. After the two jets have been united in a common exit, a considerable part of the kinetic energy may be salvaged by a suitable expansion, as in the ideal case. Any such improvement will serve either to dissipate more heat on an undercooled engine or to reduce the cost of cooling an adequately cooled engine. In the tests reported herein the optimum conditions for the combination of these two jets and their subsequent expansion were determined.

The foregoing analysis considers the problem from considerations of realizing the maximum heat transfer on a given cylinder by the optimum choice of baffles. The problem is really somewhat more complicated than the mere determination of maximum heat transfer. All temperatures on the cylinder must be held below some specified maximum. One can readily visualize a situation in which part of the cylinder is overcooled while another part is undercooled. Such a condition may be controlled by proper baffling and finning, neither of which takes the place of the other. Finning is used to give the cylinder a large heat-transfer coefficient; and baffling, to increase the air flow at a given Δp and to control the temperature distribution. This investigation is concerned only with baffling.

The study of correct baffle design is therefore made to determine not only the arrangements giving maximum average heat transfer with a given Δp but also those giving the lowest maximum temperature. The two conditions are not concomitant as it is necessary to sacrifice some of the over-all heat transfer to obtain a low maximum temperature.

APPARATUS AND TESTS

The pressure drop across the cylinder was made to simulate conditions in an actual engine. The cylinder was set in the side of a box so that the part corresponding to the front was open to the room and the baffled part was inside the box with the exit of the baffle discharging into the box (fig. 3(a)). The box was connected to the suction side of a blower. The pressure in the box could be varied from 0 to 24 inches of water below atmospheric pressure by the adjustment of a bypass and a cut-off valve. The pressure inside the box was measured by an open-end tube located in a position free from air currents. As only the baffled part of the cylinder was studied, all the essential characteristics of the practical case were reproduced by such a set-up. Small impact and static tubes (outside diameter, 0.015 inch) were placed in the fin spaces. All openings were along a given radius but not more than two tubes were in a given fin space. Five impact and five static tubes were thus located at suitable intervals along the center line between the fins. These tubes were used to give the velocity and total-head distribution in the fin spaces. As a check on these measurements a small platinum wire was located along the center line of the fin space on the same radius as the openings to the survey tubes. The current required to keep this wire at a given temperature above the air stream gave a measure of the cooling.

The cylinder was so arranged that it could be rotated with reference to the baffle. Another group of survey tubes made from 0.040-inch tubing was located in the exit-duct opening to measure the energy and velocity distribution at that point.

Two cylinder diameters were used with two fin spacings in the larger. For each arrangement the effect of the baffle-exit width and the exit-duct length and flare was studied. On one series the radius of curvature of the baffle at the point where it leaves the cylinder and joins the exit duct was varied. Figure 2(a) shows the general arrangement of cylinder and baffle.

A single test run was usually made at two air speeds for several angular positions around the cylinder. Such test data made it possible to plot curves of velocity, energy loss, and cooling of the hot wire against angular position around the cylinder. A series of tests in which exit-duct length, flare, and baffle-exit width were separately varied determined the optimum arrangement.

There are two sources of error: (1) an accidental change in the geometry of the arrangement, easily detected by the usual cross plots of the data; and (2) a systematic error due to making all the measurements on the center line of the fin space. A systematic error of this type causes no trouble for comparisons of a given fin spacing but, if the error were large, comparisons of various fin spacings would be difficult to make. Measurements of velocity and energy loss across the fin space show that the boundary layer is very thin and that the measurements made on the center line will be in error by only a few percent. In the range of fin spacings used this error is not important. Each measurement can be made with a high degree of accuracy.

As a check on the velocity and cooling tests of the unheated cylinder, tests were made of an electrically heated brass cylinder. This cylinder was 5-3/4 inches high with an inside diameter of 5 inches and a cylinder wall 1/4 inch thick. The fins were 1/32 inch thick and 3/4 inch wide with a 1/10-inch free-air space between fins.

The heater was wound on a soapstone core with about 1/4-inch gap between the cylinder wall and the coils. Heat transfer was principally by radiation from the heating coil to the cylinder wall.

The temperature of the cylinder wall was measured by ten thermocouples placed at 0°, 30°, 60°, 90°, 105°, 120° 135°, 150°, 165°, and 180° from the front of the cylinder on the six fins nearest the center of the cylinder. The hot junctions were made by peening constantan wires into the cylinder wall. A lead of the same material as that of the cylinder was joined at the cooling-air inlet to a constantan wire to form the cold junction. The thermocouples gave a practically straight-line calibration so that temperature differences measured in this manner are relatively free from error caused by small changes in cooling-air temperature.

A constant heat input of 3,100 watts was supplied to the heater. The cylinder was allowed to heat up for about 30 minutes under controlled conditions of heat input and pressure drop across the baffle. Readings on all thermocouples were taken at 5-minute intervals until approximate equilibrium had been attained. Equilibrium usually occurred about 50 minutes after the start of the test. The temperature distribution for the cylinder was then obtained.

It is obvious that, as only the rear of the cylinder was baffled in this study, the front of the cylinder would heat up excessively. On an actual cowled and baffled engine the front of the cylinder cools without difficulty owing to large-scale turbulence. In these tests it was found necessary to employ a baffle on the front of the cylinder, creating a flow toward the front (figs. 2(b) and 3(b)). The duct carrying the air away from this baffle discharged into the same box as the rear baffle and, accordingly, the same controlled Δp determined the flow through the front baffle. The front temperature remained fairly constant.

It was hoped that such an arrangement would give a front temperature independent of the rear baffle arrangement. This result was obtained when the rear baffle was tight-fitting, but loosely fitting rear baffles allowed a larger proportion of the cylinder to be cooled by the air flowing toward the front. As a result, the front temperature was higher with a loosely fitting rear baffle.

RESULTS

The detailed results, together with the method for evaluating them, are shown for a typical case; baffle arrangements used are shown in figure 2(a), and the basic results are plotted in figures 4 to 9. Figure 4 gives the distribution of total head along the center line of the fin space. Figure 5 gives the velocity distribution for the same position. The total head H_T and velocity V are presented in nondimensional forms in terms of Δp by using $H_T/\Delta p$ for total head and $\sqrt{\frac{H_T - H_S}{\Delta p}} = \frac{V}{V_H}$ for velocity where H_S is the static pressure and $H_T - H_S = q = \frac{1}{2} \rho V^2$.

The energy is given by $\int H_T V dA$ where A is the cross-sectional area. On a cylinder baffled as in figure 2(a), A is constant and V is approximately constant so that H_T can be considered a measure of the energy. Similarly $H_T/\Delta p$ can be considered as proportional to the energy.

The average values of $H_T/\Delta p$ and V/V_H are determined from an integration of the areas under the curves of figures 4 and 5. The value of V_A/V_H for a tightly fitting baffle must remain constant at all positions inside the baffle. The ratio $H_T/\Delta p$ becomes gradually smaller as the air moves around the cylinder and out the exit. Figure 6 shows a plot of the loss of $H_T/\Delta p$ against angle around the cylinder where 0° is the front and 180° the rear of the cylinder. Each curve is for a particular exit-duct opening. Figure 7 shows the effect of exit-duct opening on both the $H_T/\Delta p$ loss to the 150° position and the V_A/V_H. The peaks of the curves show the expansion that gives maximum cooling.

The curve for $I^2 - I_0^2$ also shown in figure 7 will now be discussed. The energy to heat the hot wire to a given temperature at any air speed is represented by KI^2 and, when the air speed is zero, the energy is KI_0^2. It follows, then, that the heat carried away by the air stream is proportional to $I^2 - I_0^2$. Figure 8 shows a plot of $I^2 - I_0^2$ against Δp. Separate curves are given for each angular position tested. The values of $I^2 - I_0^2$ are read from the curves at a value of $\Delta p = 10$ and plotted against angle in figure 9. The values of $I^2 - I_0^2$ are read at the 150° position from the curves and plotted against exit-duct opening (fig. 7). Figures 10, 11, and 12 show the loss in $H_T/\Delta p$ and the values of V_A/V_H and $I^2 - I_0^2$ at the 150° position all plotted against exit-duct length for several baffle openings. These results are for the 7-inch cylinder, 1/16-inch fin spacing, and 3/4-inch fin width. The peaks of the curves determined in this manner show the optimum conditions of the exit-duct expansion. The choice throughout of 150° for the point of comparison and of $\Delta p = 10$, at which all values of $I^2 - I_0^2$ were plotted, was entirely arbitrary.

Figures 13, 14, and 15 present similar results for the 1/4-inch spacing on the 7.00-inch cylinder; figures 16, 17, and 18 likewise present similar results for the 1/16-inch spacing on the 4.66-inch cylinder. Figures 10 to 18 give results that make it possible to choose the best baffle-exit width and to determine the gain obtainable for a particular exit-duct length. Figure 19 shows the optimum angle of expansion for each general arrangement tested.

All the results from the three cylinder arrangements tested have two characteristics in common: (1) A gain in cooling can be secured by lengthening the exit duct; and (2) the larger part of this gain can be secured by a relatively short exit duct. It is to be noted that these results all apply to the 150° position on the cylinder.

Figures 20 and 21, in which $I^2 - I_0^2$ is plotted against baffle-exit width for the various cylinder arrangements tested, show their relative cooling performance. The values appearing in figures 20 and 21 cannot be compared because the platinum wire was destroyed between the tests from which the two sets of data were taken. It is at once apparent that the narrower baffle exit gives the best cooling at the 180° position.

All the results presented except those in figure 21 were obtained from the tests made of a baffle having a 1/4-inch radius at the point of junction of the skirt and baffle. A few tests were made to determine the effect of a larger radius. Figures 22 and 23 show the results for 1.0-inch and 1.5-inch baffle exits where the radius was varied and the exit-duct length remained at 7.00 inches for all the tests. The cooling was improved 8 to 10 percent by increasing the radius from 1/4 to 3/4 inch. Such an improvement in cooling is to be expected in any design.

It has been pointed out that the cooling problem is not merely a problem of securing a high heat-transfer coefficient but also a problem of obtaining a reasonable uniformity of cylinder temperature. Some 40 baffle arrangements were studied in these cooling tests. Table I presents the results of 15 of these tests and gives a description of each of the 15 baffles. The results show the change in temperature distribution and temperature when the baffle entrance, baffle exit, baffle-exit radius, and exit-duct length are varied.

Baffle 1 is representative of the most usual baffle now in use. It was made with an entrance similar to that shown in figure 2(a). The baffle exit was given a 1/4-inch radius and was placed concentric with the test model allowing a 1/4-inch clearance between fin tips and baffle.

Baffles 7 and 8 show the effect of the entrance opening. Baffle 9 shows the effect of speeding up the air very rapidly at the rear of the cylinder. Baffle 10 represents an attempt to accentuate the effect developed in baffle 9. It was thought that the sharp break in the baffle tended to set up turbulence. The addition of a tightly fitting inner baffle, 1-1/2 inches wide, leaving a 1/2-inch opening ahead of the point where the baffle comes in contact with the fins was expected further to increase this turbulence. Such a baffle acts as an injector, increasing the velocity between the fins and forcing cold air between the fins at the point where the greatest cooling is needed. Baffle 11 was developed to increase this turbulence. This baffle had a 1/8-inch strip at the baffle entrance and a 3/8-inch gap for the entrance of the air. The 1-1/2 inch inner baffle was located the same as in baffle 10. This arrangement failed to cool satisfactorily without the inner baffles. Later tests made it relatively certain that a gap wider than 3/8 inch would have further improved the cooling. Baffle 12 was tested in an attempt to show that a greater improvement in cooling would result from a very high flow velocity than from an entrance designed to cause turbulence. The results are indecisive. The addition of a 1-1/2 inch strip, with a 3/4-inch opening between the strip and the baffle, similar to that in baffle 10, gave very satisfactory cooling. Baffle 13 shows the effect of an injector in the exit duct near the baffle exit. There is a definite improvement in cooling, which undoubtedlycould be further extended. The benefit derived from such an arrangement, however, is small in comparison with the complications of construction that such a baffle would involve. Baffle 14 is a type which might be used for blower cooling. Baffle 15 was designed to show the improvement to be obtained by separating the two jets and should be compared with baffle 14. Each cools the entire cylinder, in contrast to the previously discussed baffles. Note that both maximum and average temperatures are 20° lower with baffle 15 than with baffle 14. As indicated in figure 2(j), baffle 15 has an interfin baffle. This interfin baffle was so fitted that sufficient air was allowed to pass to give relatively good cooling at the point of passage of the air. An adaptation of this idea might prove advantageous for cooling half the cylinder.

DISCUSSION

The loss in total head for the baffle arrangement is conveniently divided into three parts: (1) the drop in the fins; (2) the loss at the baffle exit, caused chiefly by the necessarily sharp turn that the air flow must make; and (3) the expansion loss that takes place in the exit duct and the exit-duct opening. The first loss is the only loss that can be utilized for cooling the cylinder. Figure 16 shows the useful loss at the 150° position to be 0.275 $H_T/\Delta p$. The baffle extends to the 165° position. Thus the loss in the baffle will be 0.36 $H_T/\Delta p$. Figure 17 shows that V_A/V_H for the same arrangement is 0.725. From the thickness of the fins and the exit-duct expansion angle it can be shown that the air is expanded to approximately half velocity before it reaches the exit-duct opening where V_A/V_H will be about 0.36. The air at this point has only $(V_A/V_H)^2$ energy remaining, or 0.13 $H_T/\Delta p$. This value leaves approximately half of the original energy to be accounted for at the turn and entrance to the exit duct. Tests of an actual engine gave losses of $H_T/\Delta p$ of 0.2 in the baffle and 0.8 at the baffle exit.

These values are only approximate but are sufficiently accurate to indicate how very inefficient the mechanism of cooling really is, either on an actual engine or with the more elaborate arrangement tested. There is still a great opportunity for improvement. The present test results, however, seem to indicate that further improvement in this direction will be slight and difficult unless the general system of cooling now employed is changed to make the useless losses small and the useful drop in total head large. Closer fin spacing will increase this useful drop in total head.

The results indicate that as long an exit duct as possible should be used, with a radius of approximately 3/4 inch at the baffle exit. A smaller radius causes a large loss in energy with a consequent lower velocity, whereas a larger radius allows the flow to miss the rear of the cylinder. The baffle should fit closely near the baffle exit, gradually expanding toward the front.

The results for either the energy-loss study or the tests of the heated cylinder show that, compared with the standard baffle, an improvement of 20 to 25 percent of the heat transfer of the cylinder can be realized by the use of the best arrangement tested. Such an improvement might not appear to be particularly interesting. It would be necessary to increase the Δp 65 percent to obtain such a gain when the standard baffle is used. The standard baffle, however, would require about double the power.

The best baffle tested, because it fits closely, requires no more power at a given Δp than the poorest arrangement. The good baffle has a higher velocity between the fins, thus using more air where actual cooling is accomplished; the standard baffle has a lower velocity between the fins. The standard baffle, moreover, allows air to escape without cooling. Thus the quantity of air used is about the same for each baffle.

The function of the baffle in controlling temperature distribution will now be considered. The temperature on the cylinder would be expected to increase from the baffle entrance to the baffle exit. Table I confirms this result for every baffle tested. The effect is more pronounced if the baffle has the same shape as the cylinder and less pronounced when the baffle has some clearance at the entrance, gradually approaching the cylinder and coming in contact with it just in advance of the exit. This perfectly logical result is due to the heating of the air. It could be predicted that the gradual increase in the velocity and the continued addition of cool air between the fins would counteract to some extent the effect of the heating of the air.

The problem, then, is to determine to what extent it is possible to overcome this heating effect. The results show several arrangements that are particularly effective in leveling off the temperature distribution. Such arrangements, of course, are only effective to the extent that the maximum temperature is reduced. Table I shows several devices that appear to be equally good. A more exhaustive series of tests would probably further improve the cooling. Such tests should be incidental to the fitting of a baffle to an actual engine cylinder.

CONCLUSIONS

1. A 20-percent increase in cooling was obtained by the use of a properly designed baffle exit. This gain corresponds to that which can be obtained when the standard baffle with a 65-percent increase in pressure drop across the baffle is used.

2. The baffle-exit radius and width and exit-duct length were found to be the most critical parts of the baffle.

3. A baffle was developed that gave a relatively uniform temperature distribution around the cylinder.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., January 14, 1937.

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2. Theodorsen, Theodore, Brevoort, M. J., and Stickle, George W.: Full-Scale Tests of N.A.C.A. Cowlings. T.R. No. 592, N.A.C.A., 1937.

TABLE I

Temperature Difference, ^{o}F.

Baffle	Figure	Baffle-exit radius in.	Baffle-exit width in.	Baffle-entrance width in.	Exit-duct length in.	Exit-duct width in.	Flared to	Angular position around cylinder, degrees									
								0	30	60	90	105	120	135	150	165	180
1	-	0.25	1.875	0.25	0	-	-	126.5	113.5	109.5	116.5	124	132	139	150	160	167.5
2	2(a)	.25	1.875	close	0	-	-	126	114	110	113	119	125	132	140	148	155
3	2(a)	.75	1.875	do.	7	2.625	-	120	106	100	99	103	107.5	114	124	136	144
4	2(a)	.75	1.375	do.	7	2.25	-	119.5	106	100.5	101	106	109.5	115	122	128.5	136
5	2(a)	.25	1.875	do.	7	2.625	-	126.5	114	109	111.5	115	119.5	125	133	141	151
6	2(a)	.25	1.250	do.	7	2.625	-	127	113.5	109.5	112	117.5	122.5	128	135	140.5	147
7	2(c)	.75	1.250	0.50	7	2.25	-	121	109.5	103	107.5	110.5	114	115.5	120.5	126.5	133.5
8	2(c)	.75	1.250	1.00	7	2.25	-	122	107.5	103	108.5	112	114	117	119.5	125	132.5
9	2(d)	.75	1.250	1.00	7	2.625	-	132	120	117	124	127	129	130	133	135	142
10	2(e)	.75	1.250	2.00	7	2.25	-	130	117	116	124	126.5	126.5	126	126.5	130	134
11	2(f)	.75	1.250	2.625	7	2.25	-	141	128	125	127	128.5	126	125	125.5	127	131.5
12	2(g)	.75	1.250	1.25	7	2.25	1.00	131.5	116	113.5	120.5	125	125.5	126.5	128.5	130	134.5
13	2(h)	.75	1.375	.75	7	2.625	-	119	105	100	106	111	114	116.5	120	124	129.5
14	2(i)	.75	1.250	1.25	7	2.25	-	157.5	149.5	150	150.5	150	147	146	149	154.5	160
15	2(j)	-	.750	.625	7	1.25	-	132	132	132	135.5	140	136.5	127	128	128	132

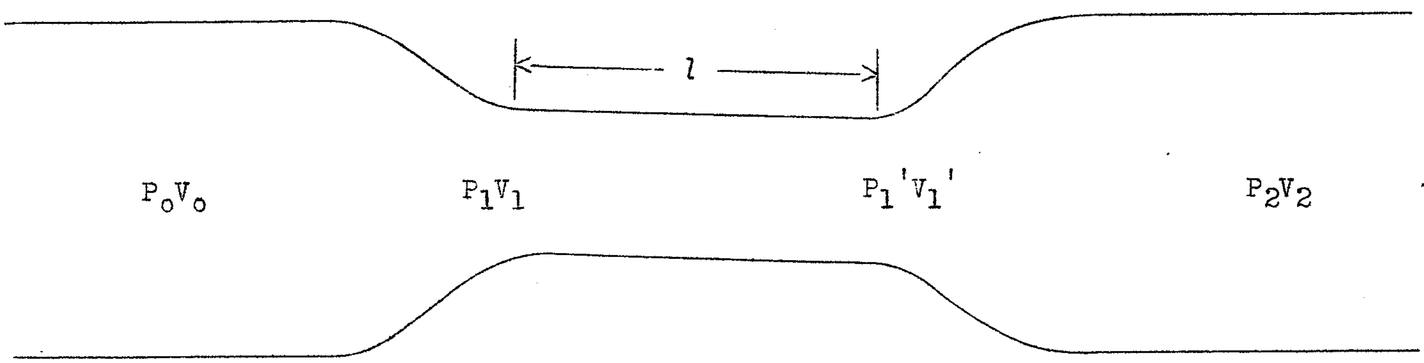

Figure 1.- Ideal case.

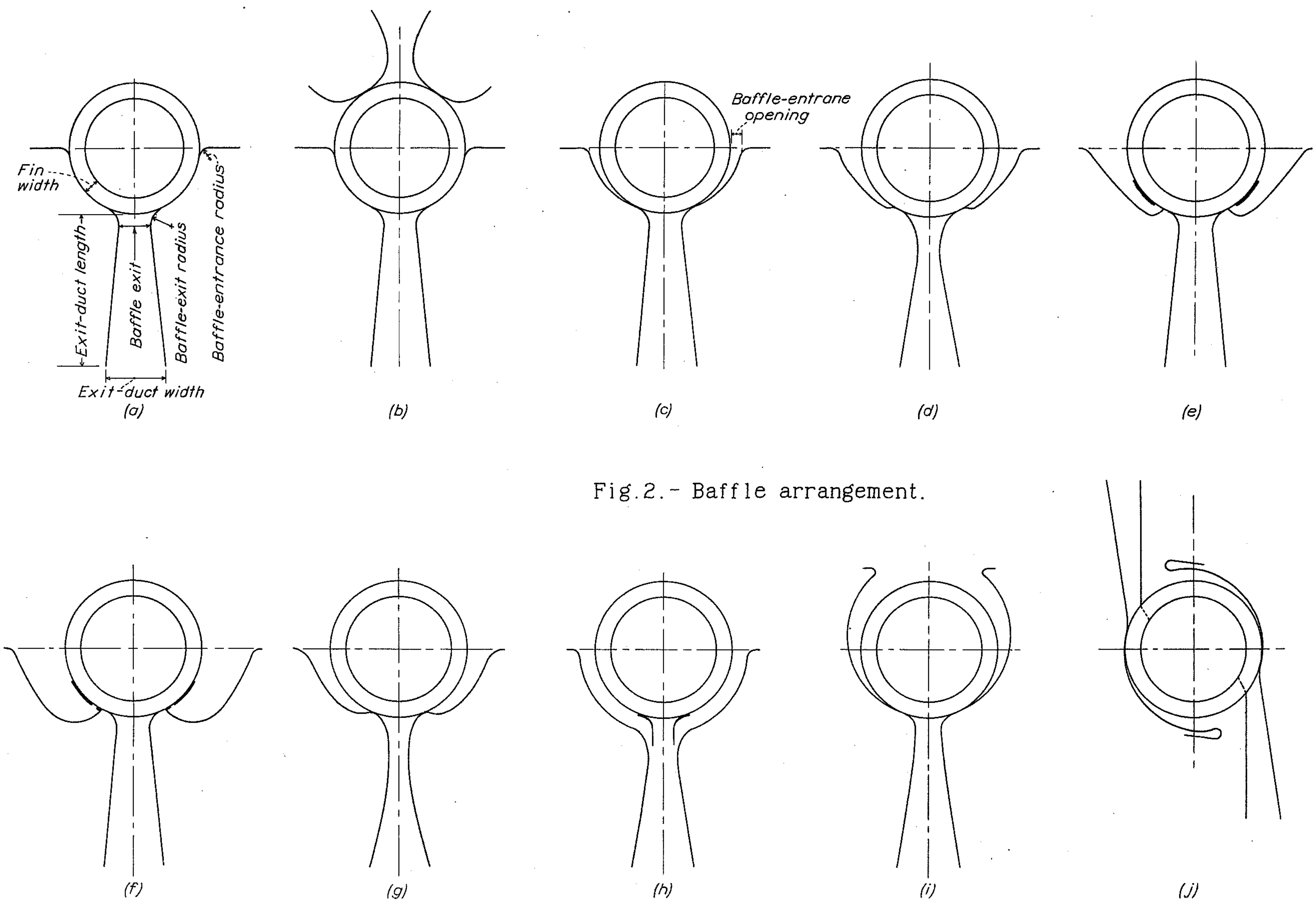

Fig.2.- Baffle arrangement.

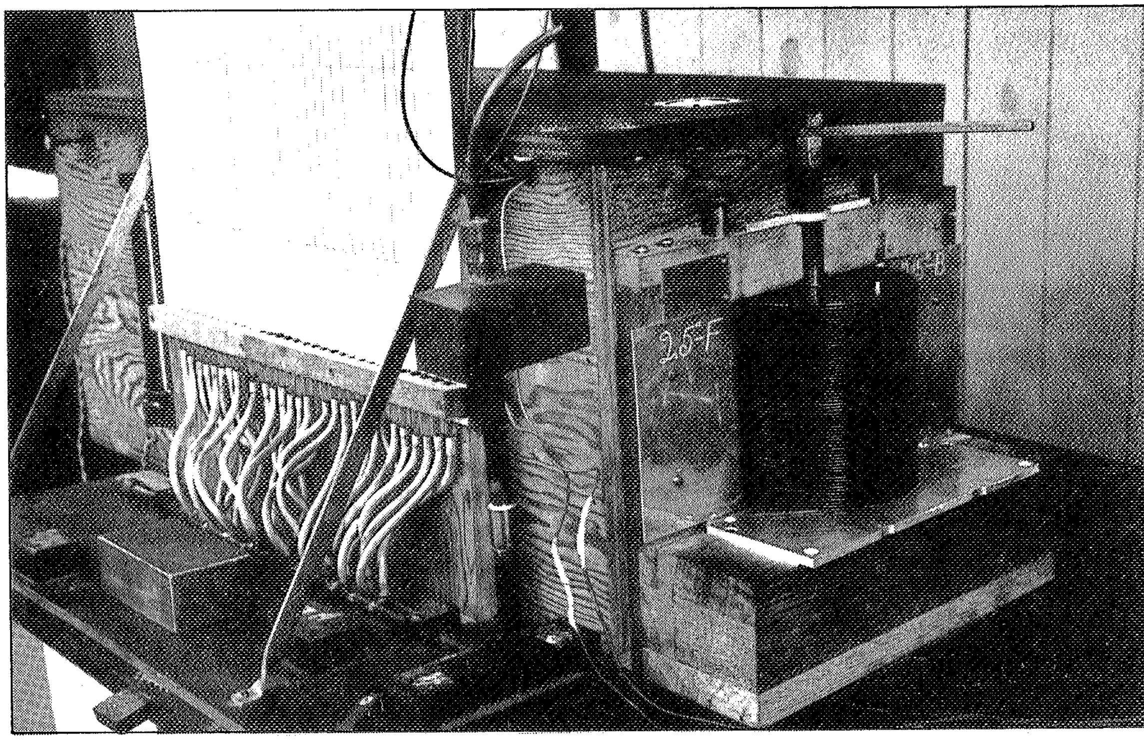

Figure 3(a).- Test arrangement for velocity study.

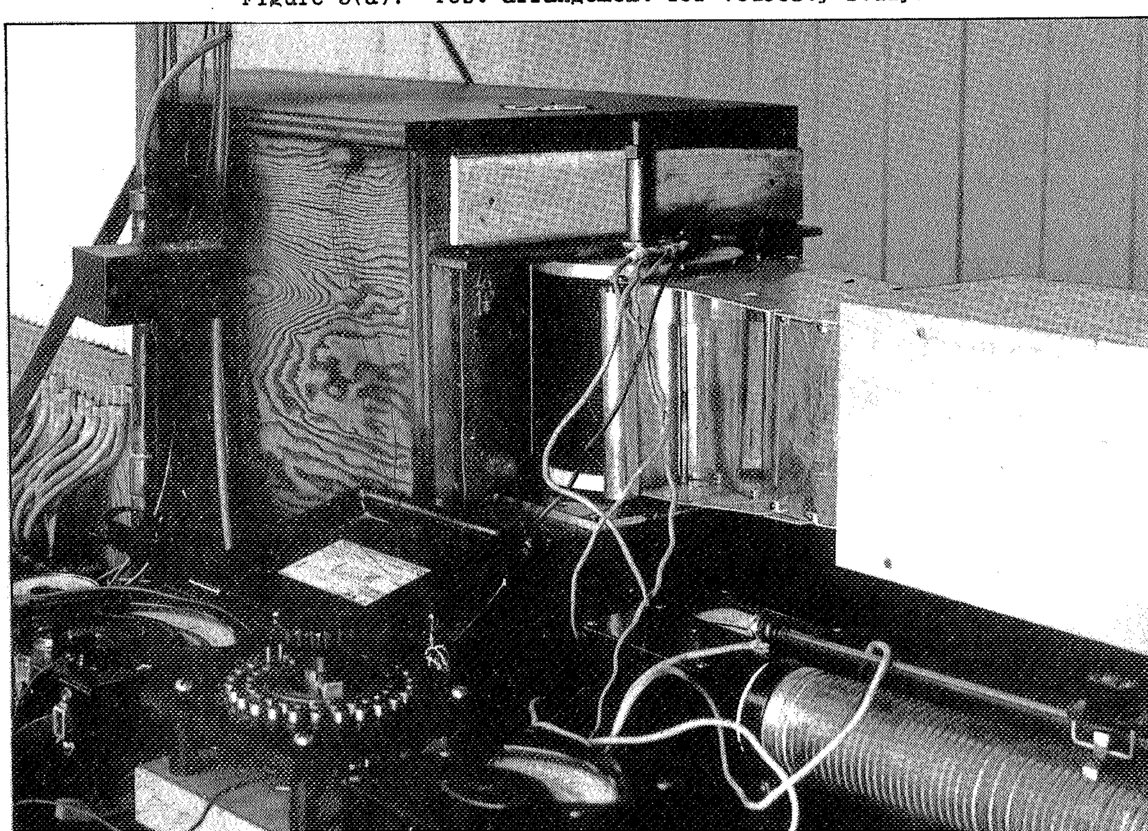

Figure 3(b).- Test arrangement for cooling with heated cylinder.

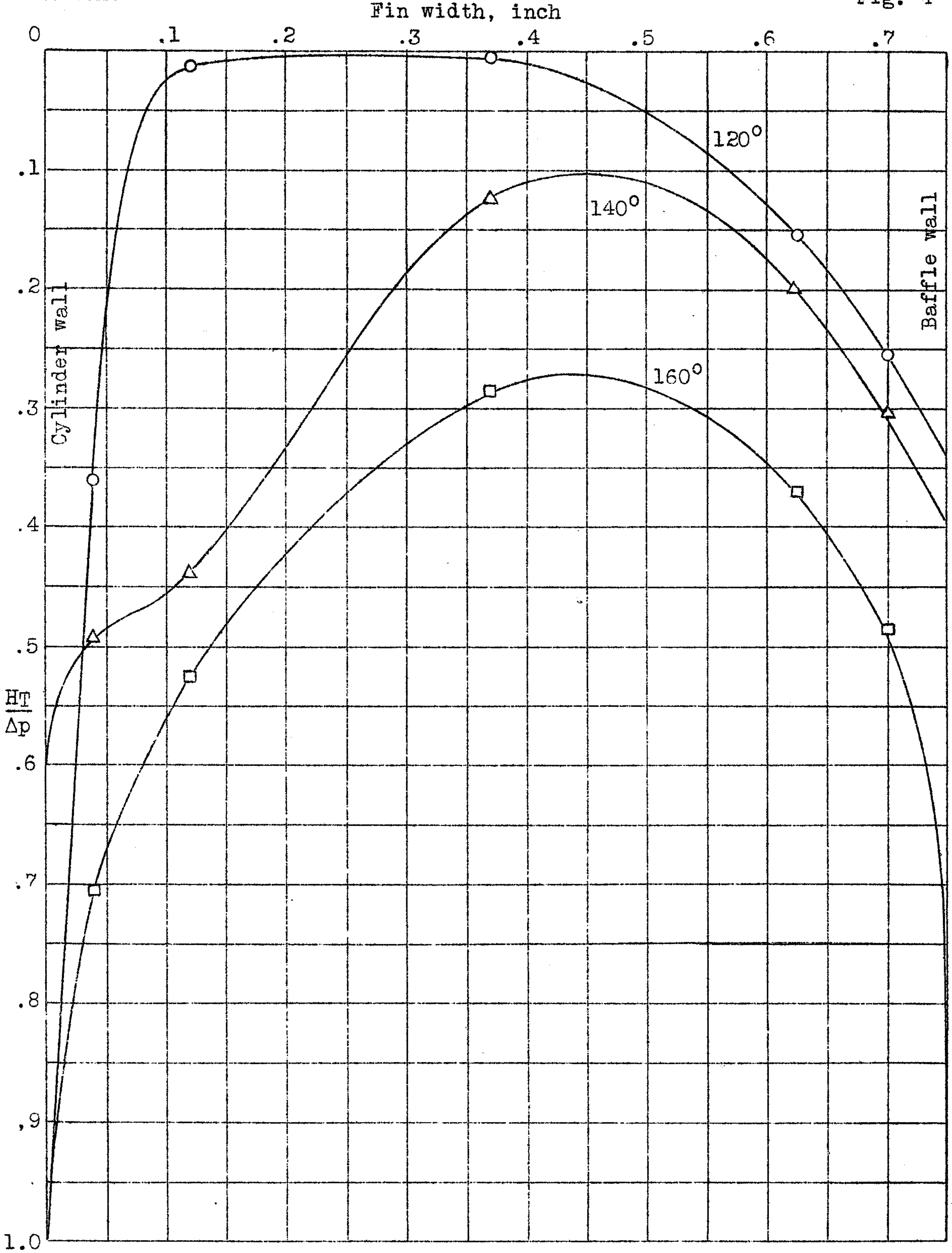

Figure 4.- Loss in total head between fins for several positions around cylinder. Exit-duct length, 3.75 inches; exit-duct opening, 1.50 inches; baffle opening, 1.00 inch.
Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

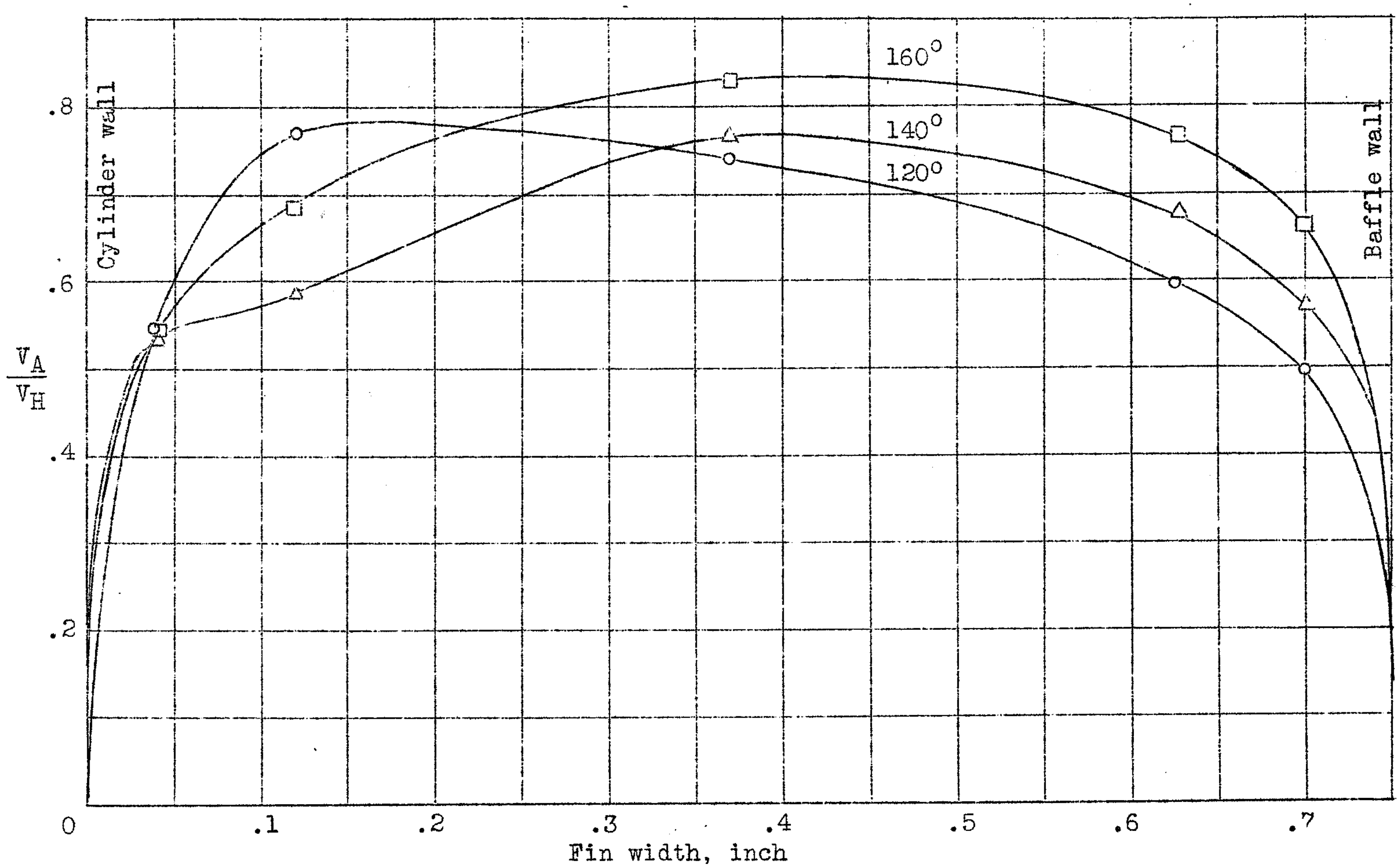

Figure 5.- Velocity distribution on center line of fin space. Exit-duct length, 3.75 inches; exit-duct opening, 1.50 inches; baffle opening, 1.00 inch. Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

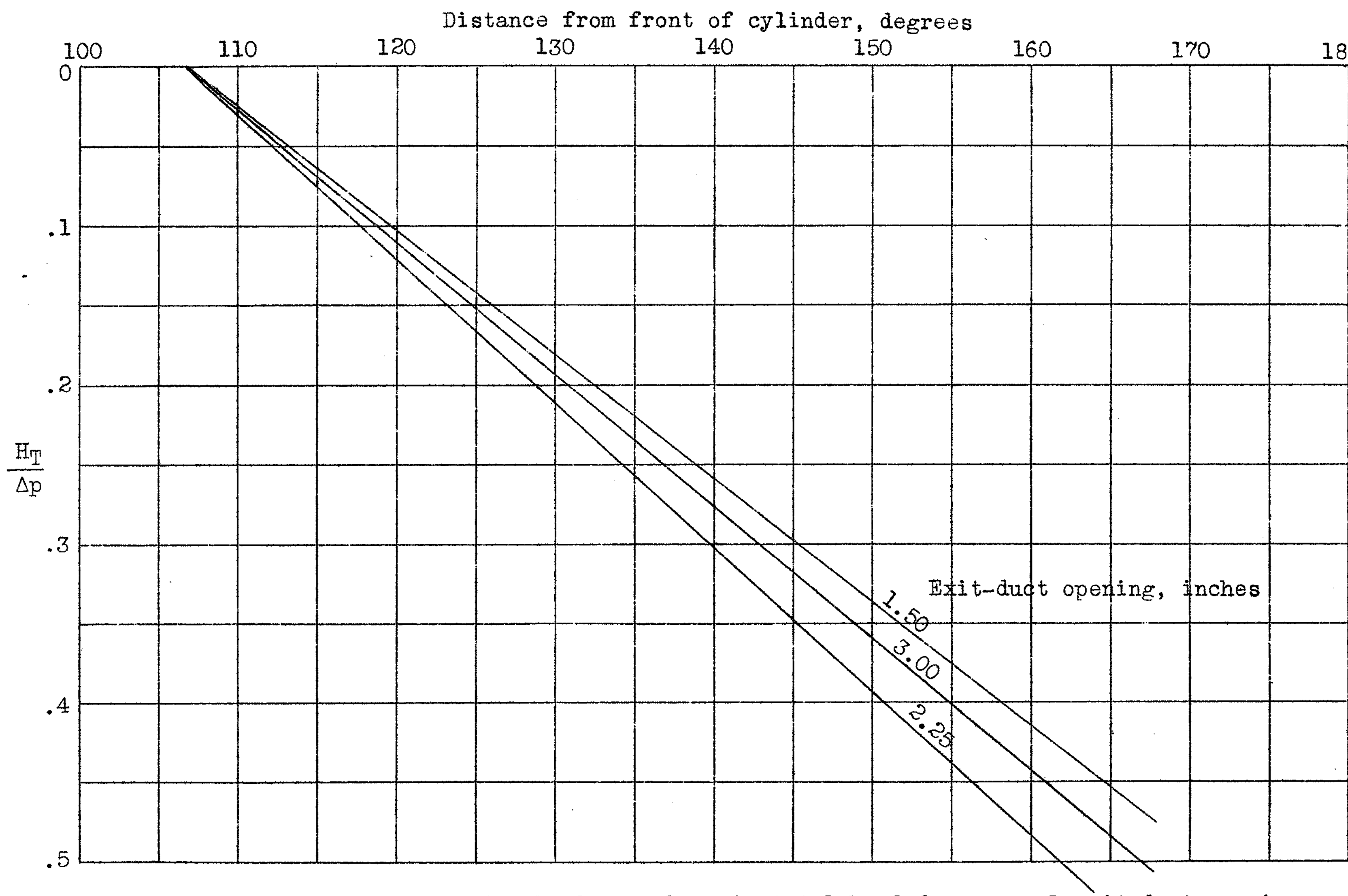

Figure 6.- Effect of position around cylinder on loss in total head for several exit-duct openings. Exit-duct length, 3.75 inches; baffle opening, 1.00 inch; Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

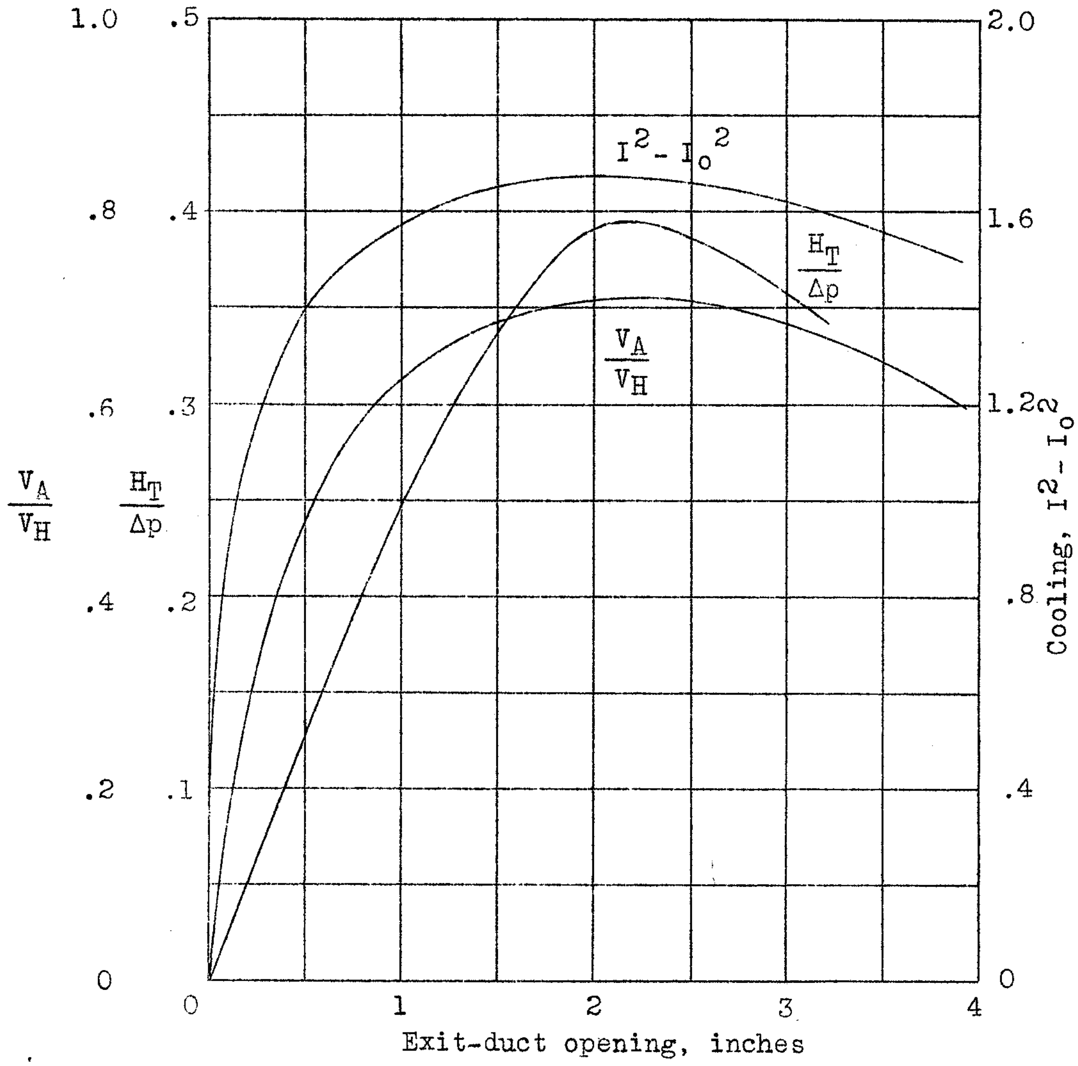

Figure 7.- Effect of exit-duct opening on $H_T/\Delta p$, V_A/V_H, and $I^2 - I_o^2$ at 150° from front of cylinder. Exit-duct length, 3.75 inches; baffle opening, 1.00 inch;
Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

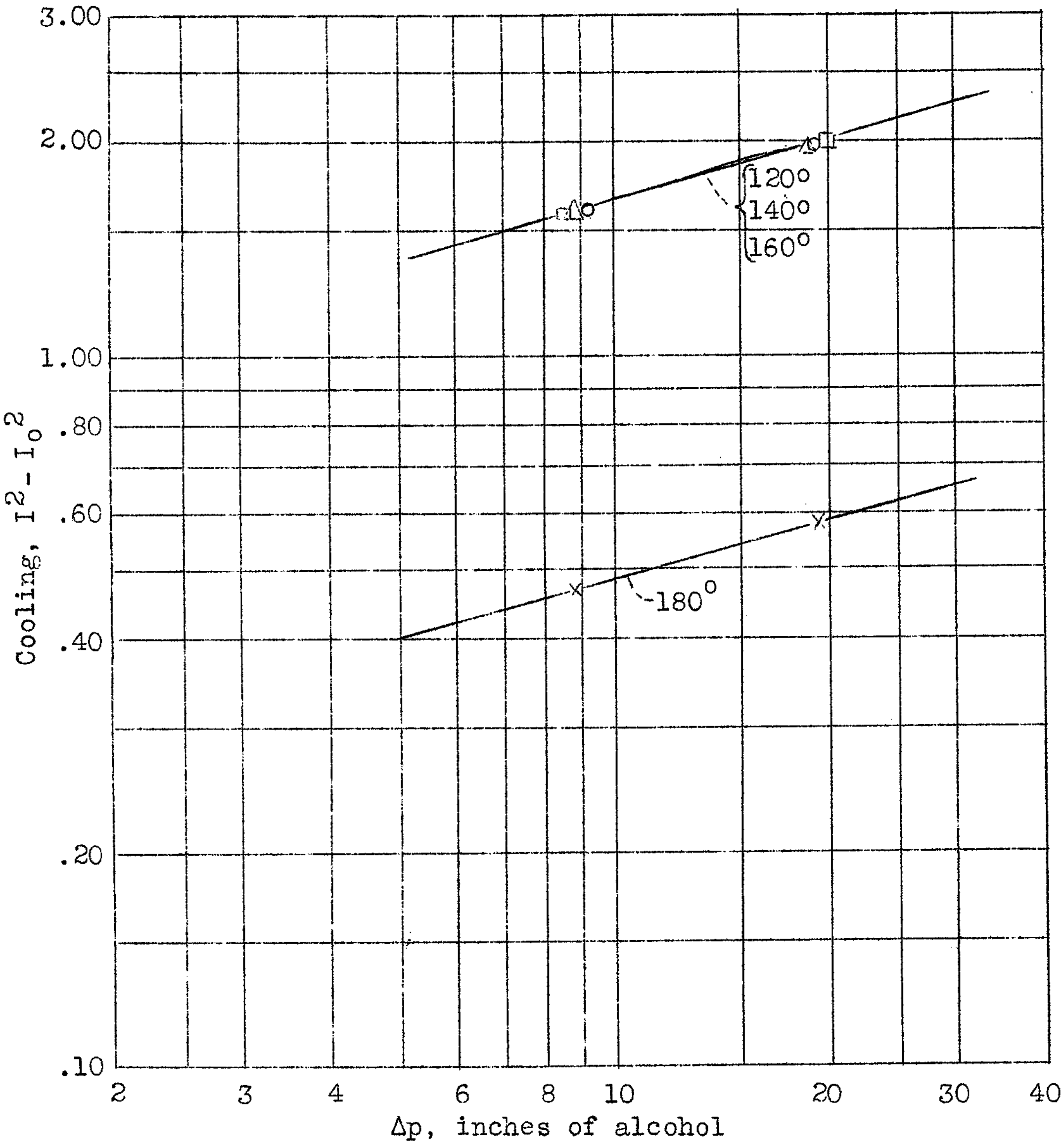

Figure 8.- Effect of Δp on cooling for several positions around the cylinder. Exit-duct length, 3.75 inches; exit-duct opening, 1.50 inches; baffle opening, 1.00 inch. Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

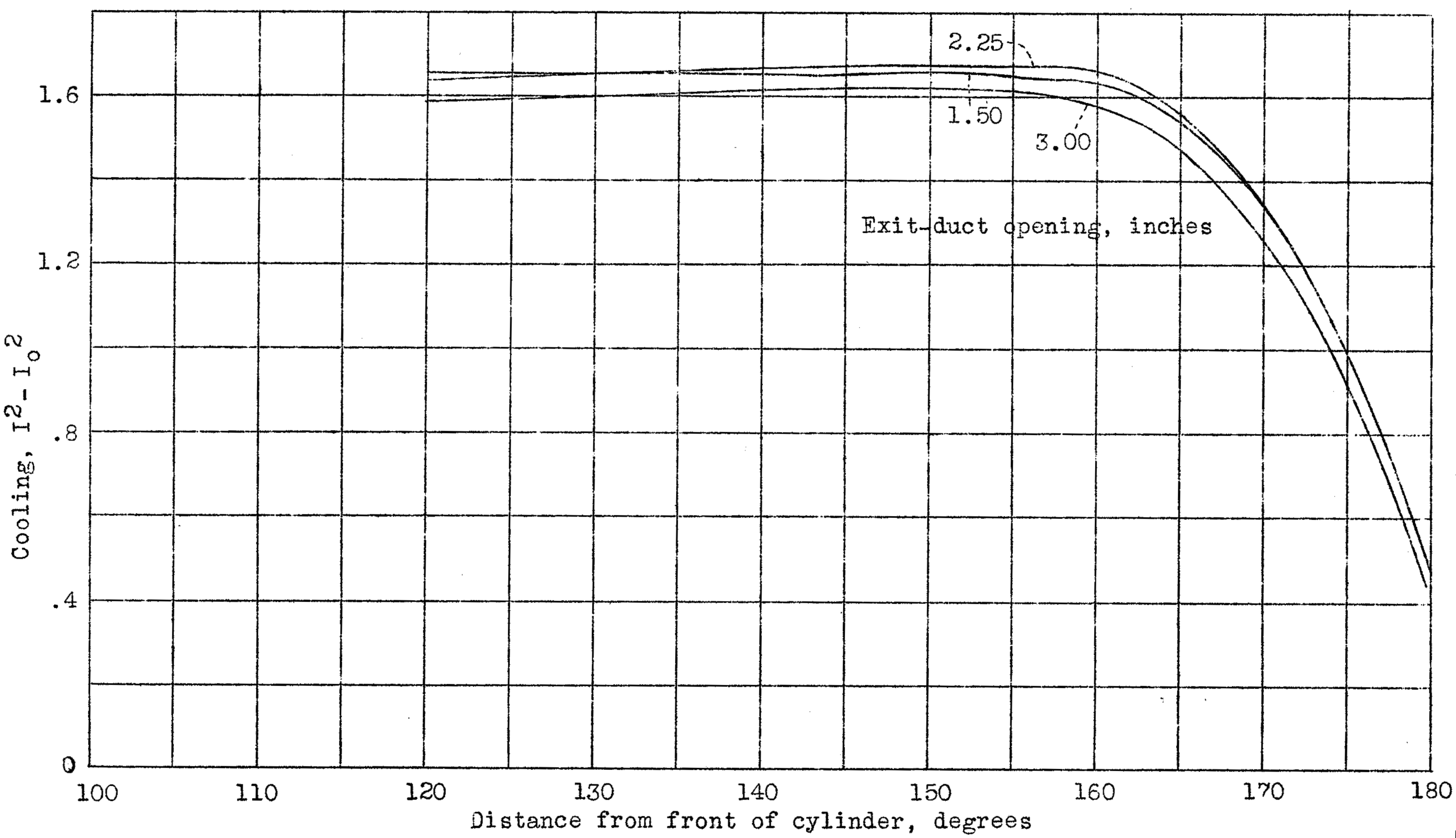

Figure 9.- Effect of position around cylinder on cooling for several exit-duct openings. Exit-duct length, 3.75 inches; baffle opening, 1.00 inch.
Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

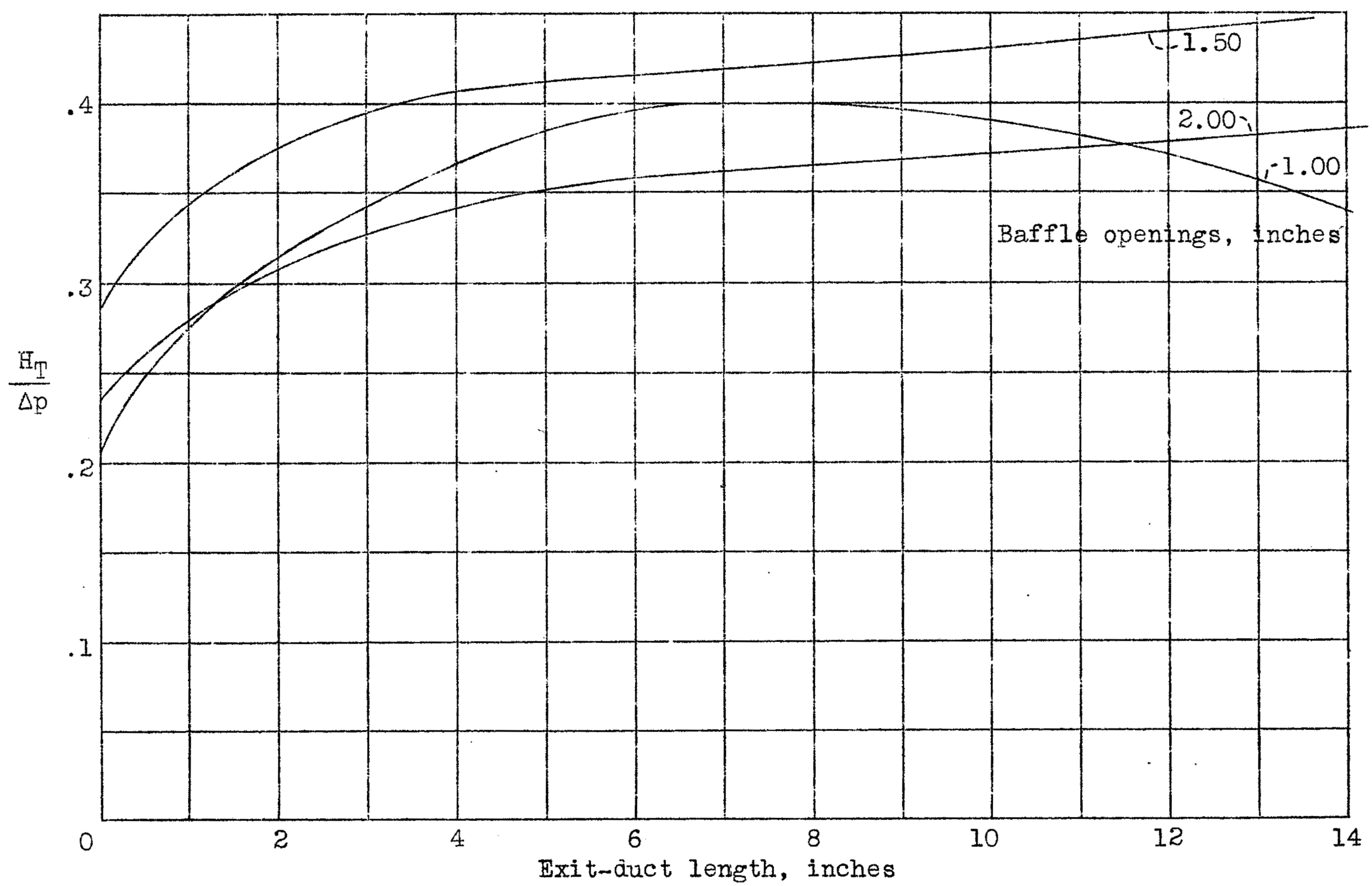

Figure 10.- Effect of exit-duct length on loss in total head at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space, 1/16 inch

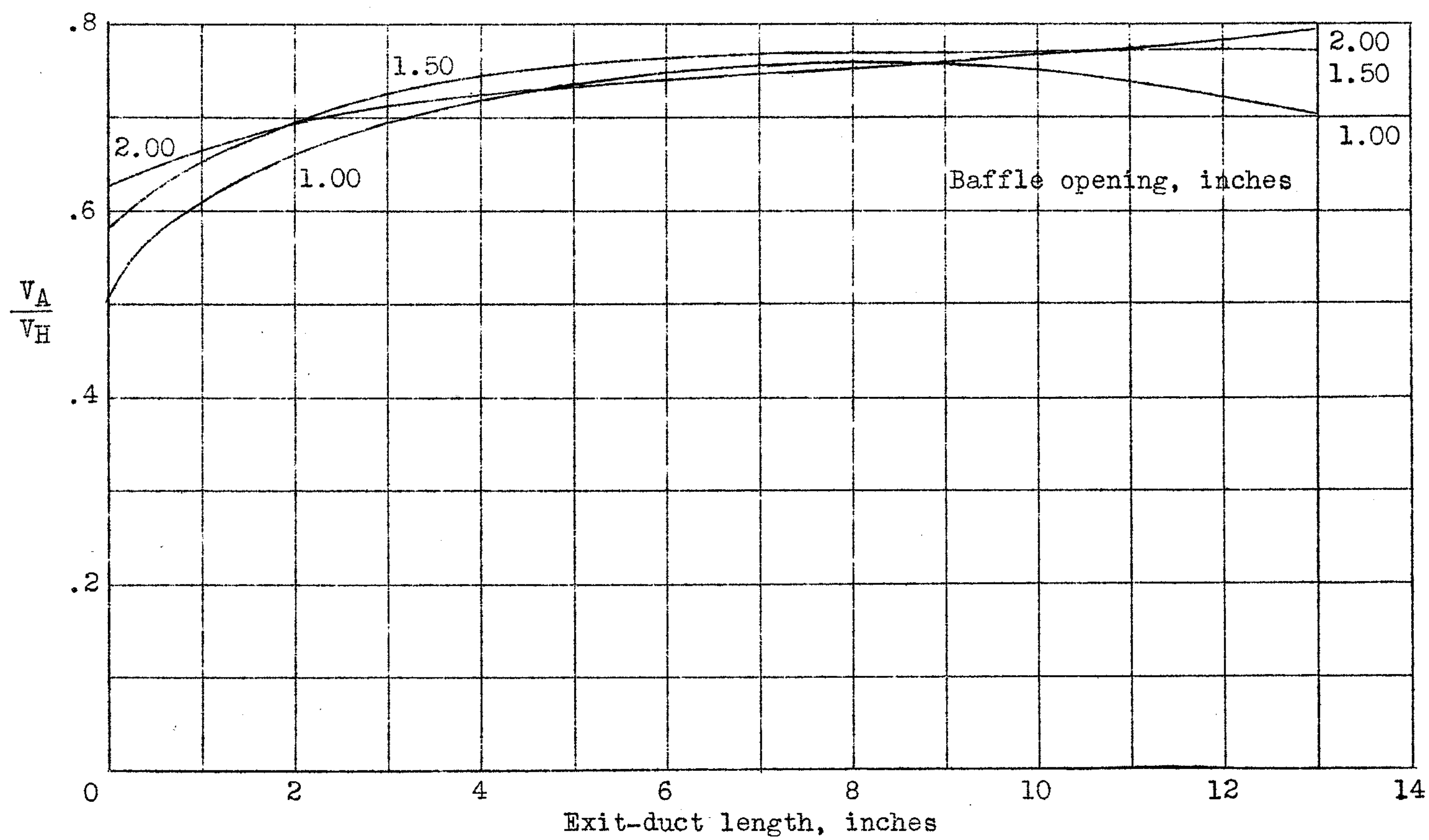

Figure 11.- Effect of exit-duct length on V_A/V_H at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

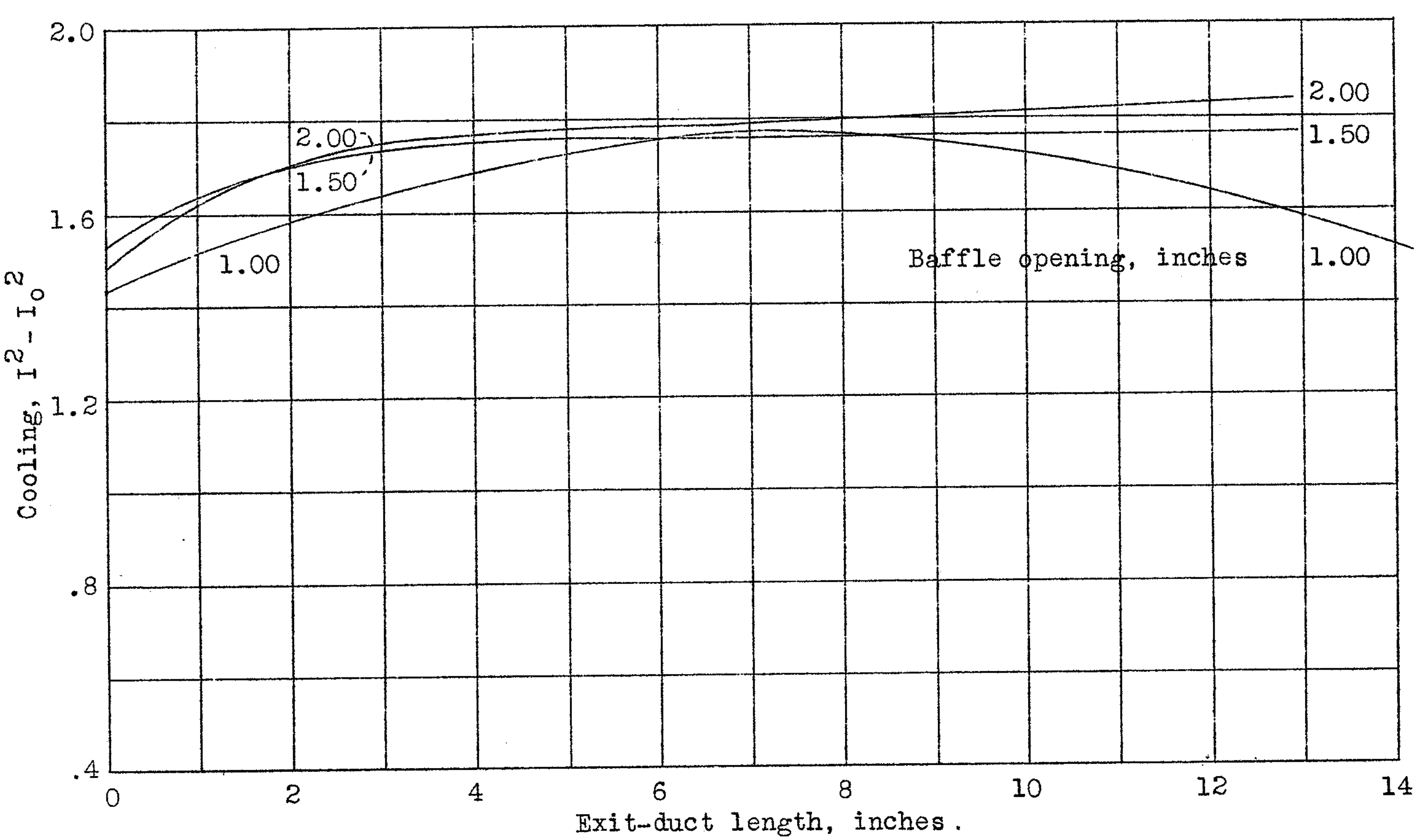

Figure 12.- Effect of exit-duct length on cooling at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space, 1/16 inch.

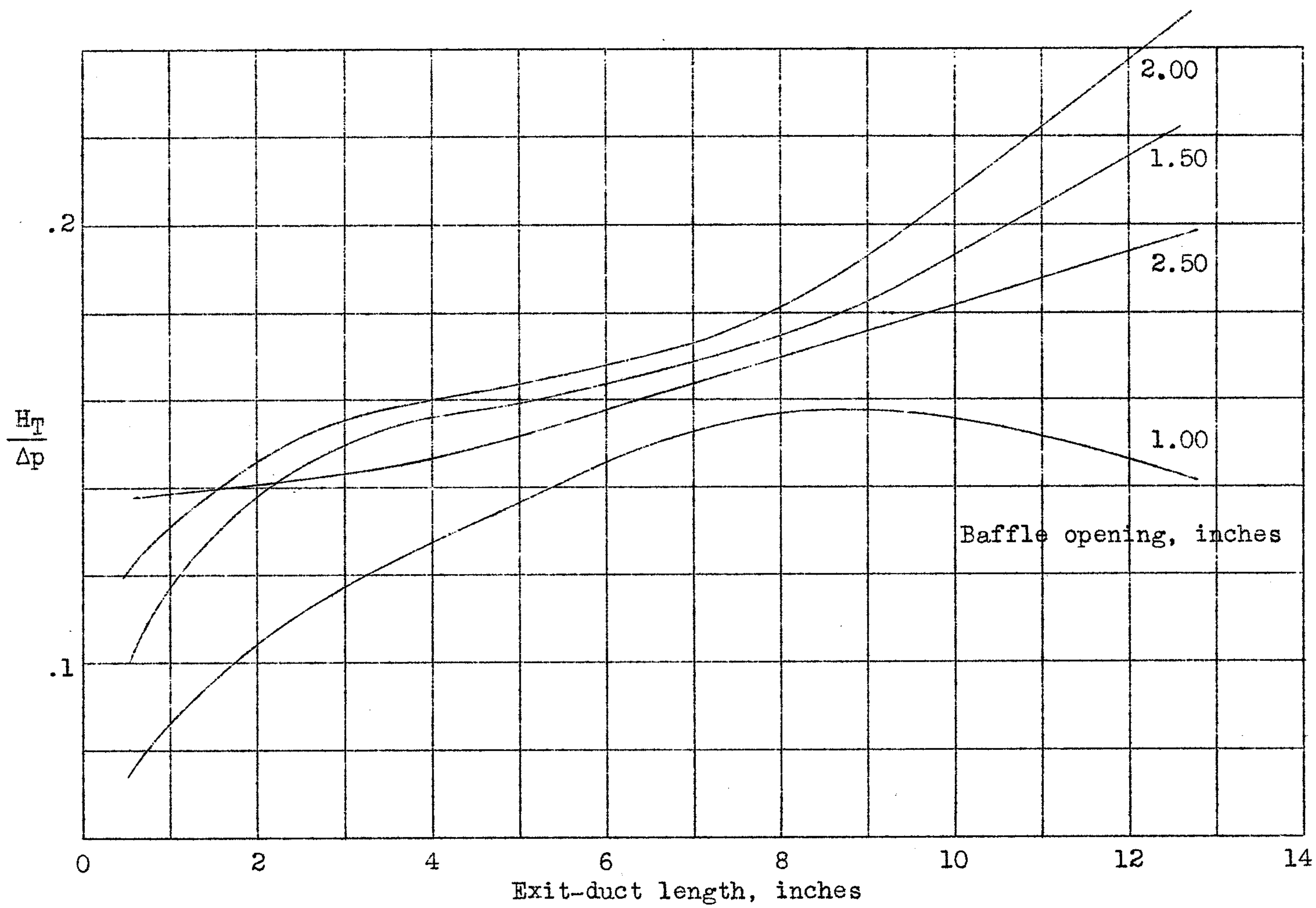

Figure 13.- Effect of exit-duct length on loss in total head at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space, 1/4 inch.

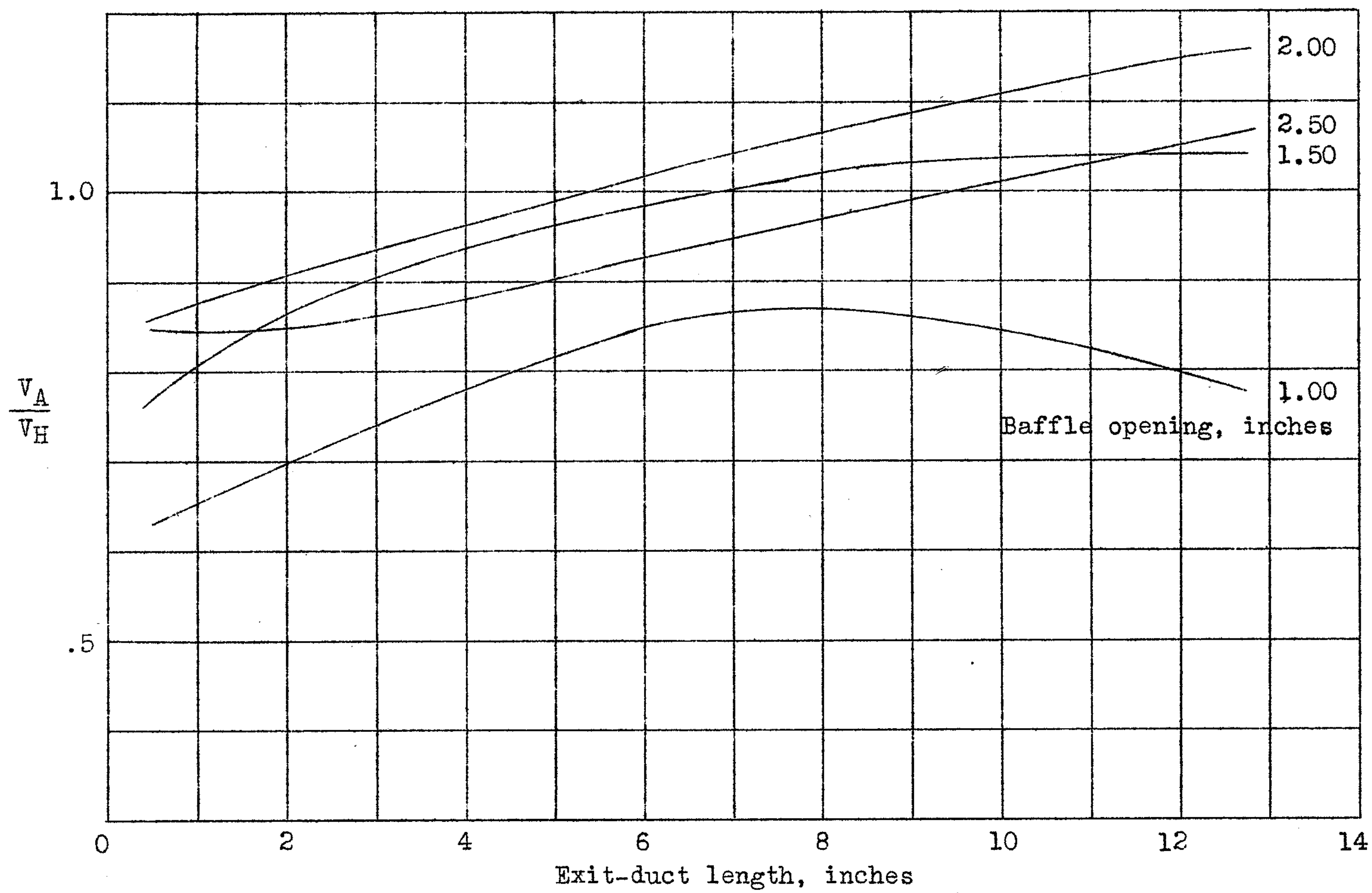

Figure 14.- Effect of exit-duct length on V_A/V_H at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space, 1/4 inch.

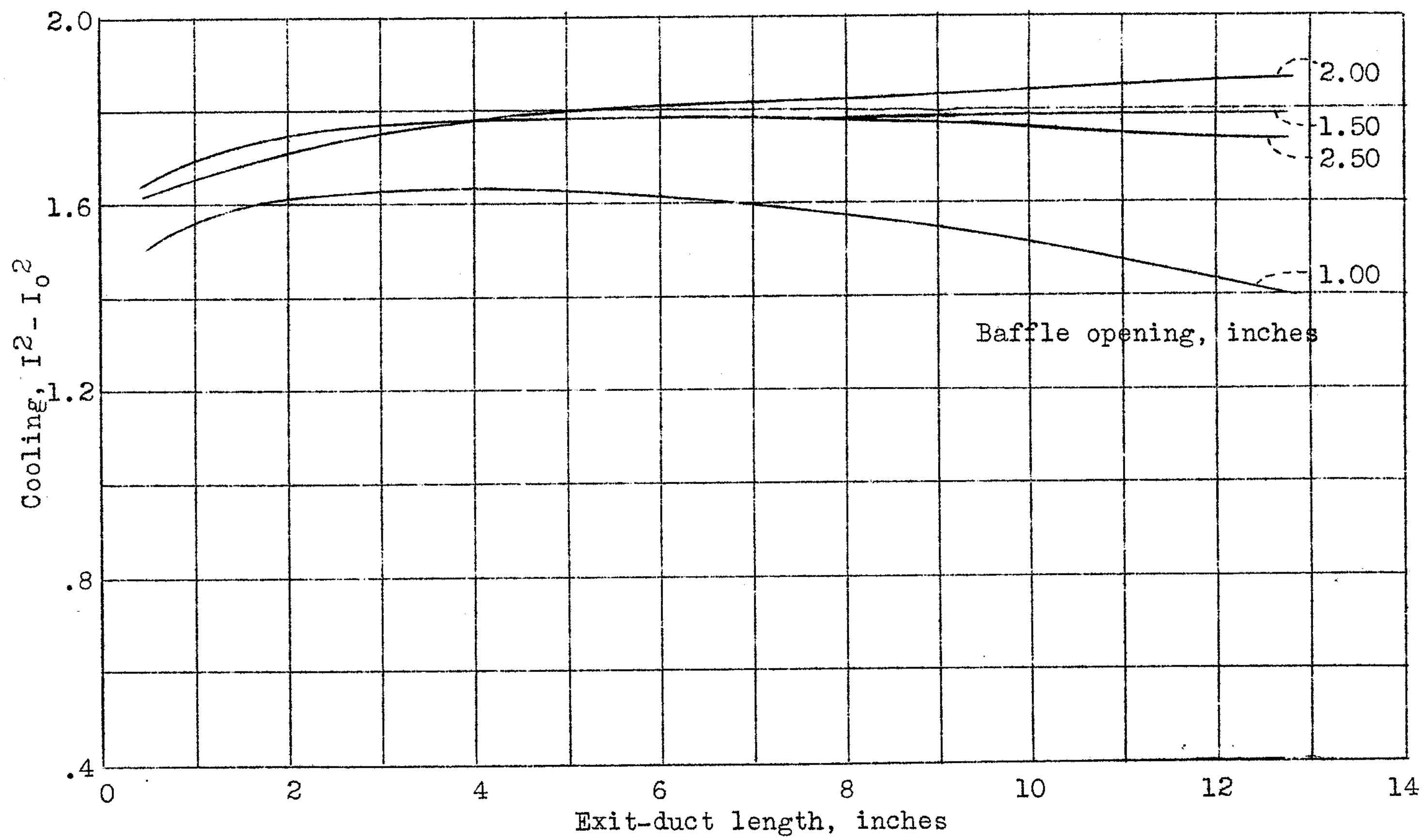

Figure 15.- Effect of exit-duct length on cooling at 150° from front of cylinder for several baffle openings at optimum exit-duct opening.
Cylinder diameter, 7.00 inches; fin space. 1/4 inch.

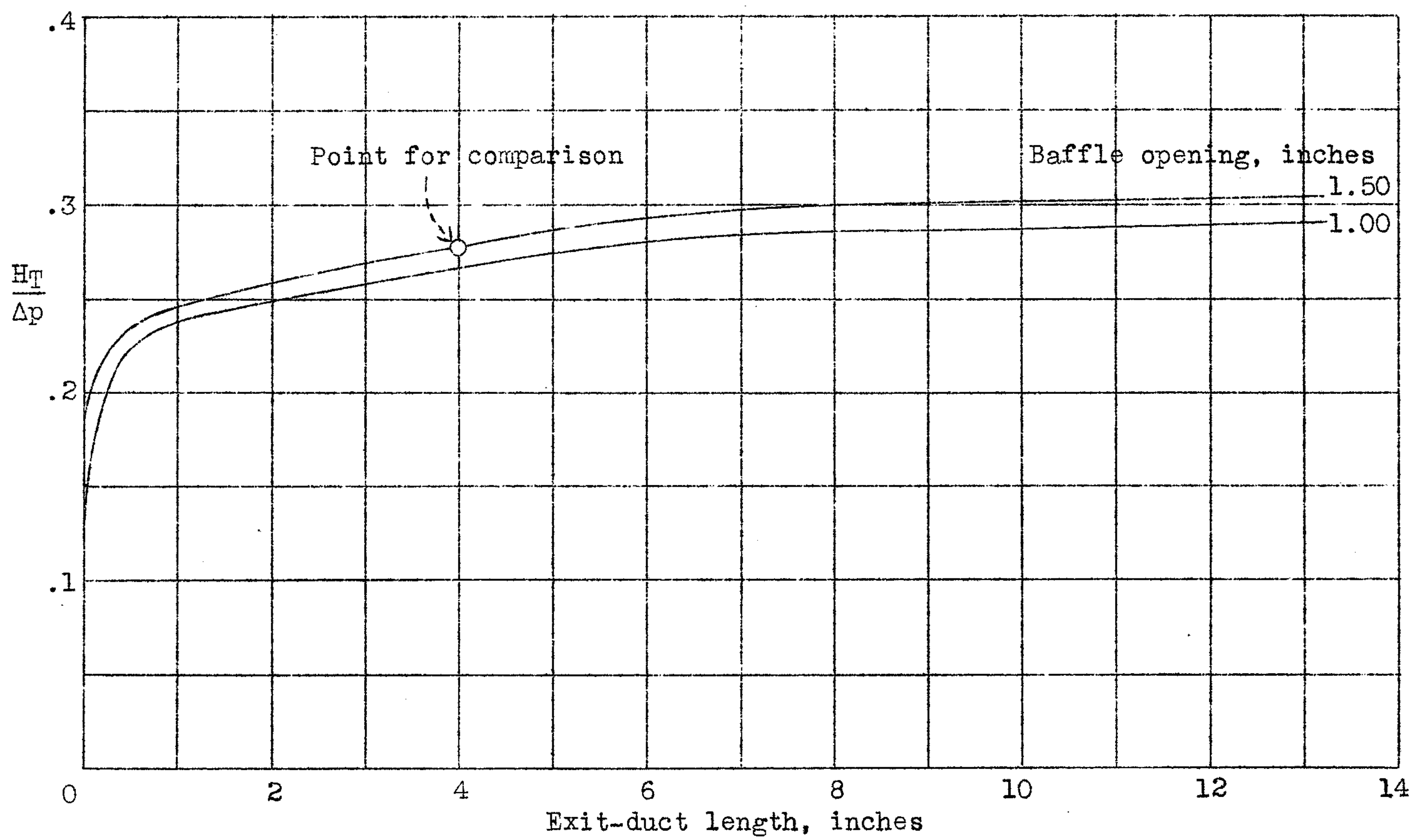

Figure 16.- Effect of exit-duct length on loss in total head at 150° from front of cylinder for different baffle openings at optimum exit-duct opening. Cylinder diameter, 4.66 inches; fin space, 1/16 inch.

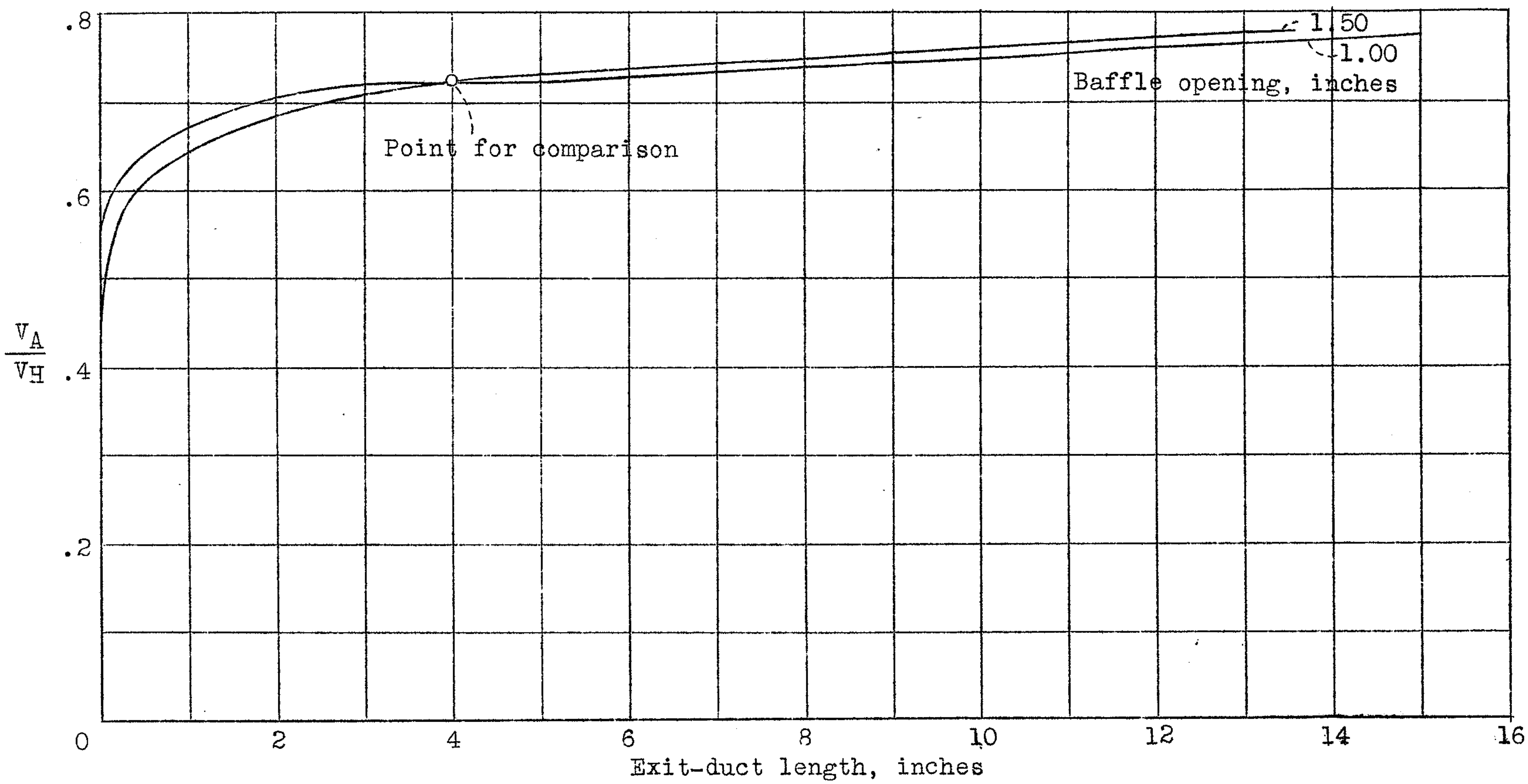

Figure 17.- Effect of exit-duct length on V_A/V_H at 150° from front of cylinder for different baffle openings at optimum exit-duct opening.
Cylinder diameter, 4.66 inches; fin space, 1/16 inch.

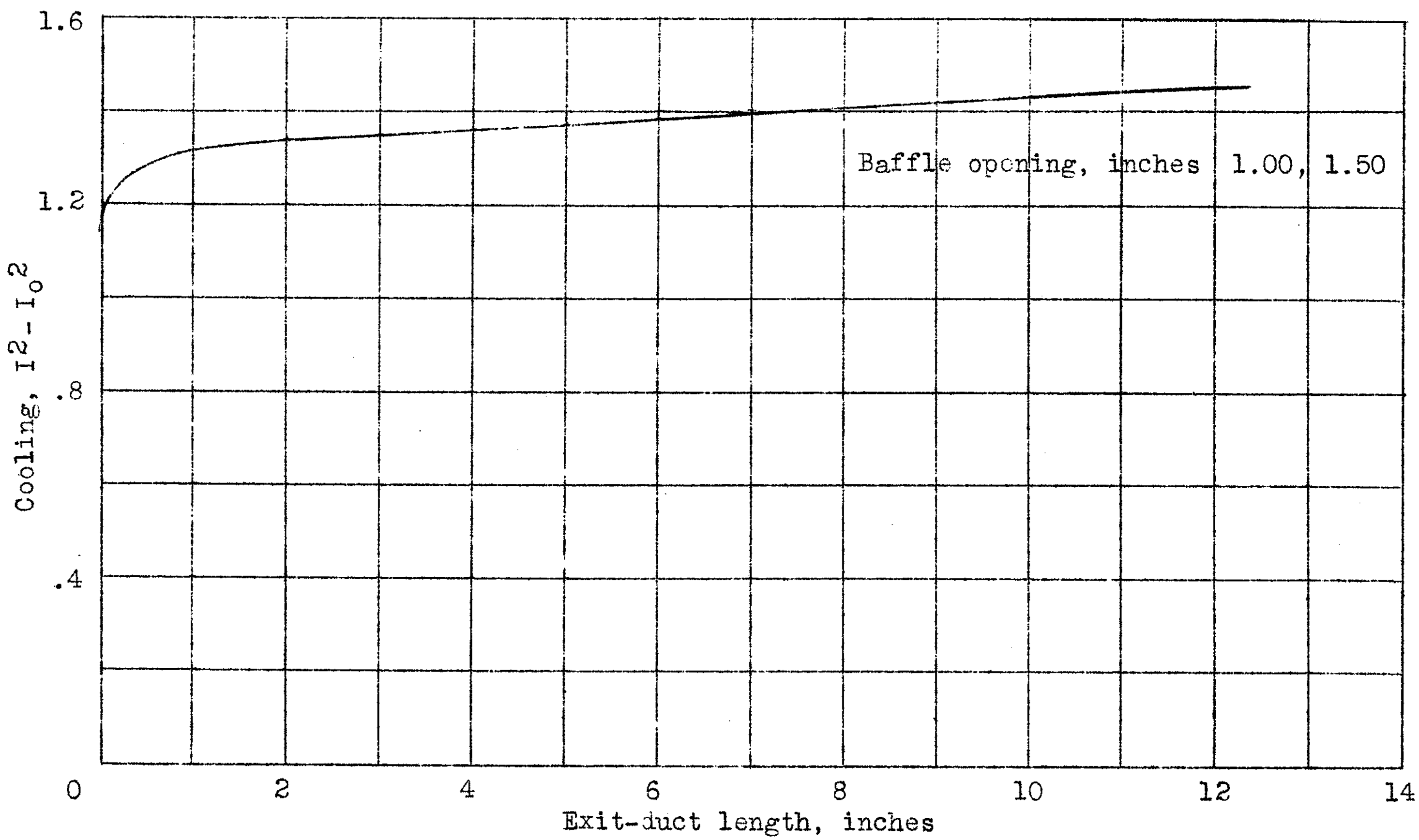

Figure 18.- Effect of exit-duct length on cooling at 150° from front of cylinder at optimum exit-duct opening.
Cylinder diameter, 4.66 inches; fin space, 1/16 inch.

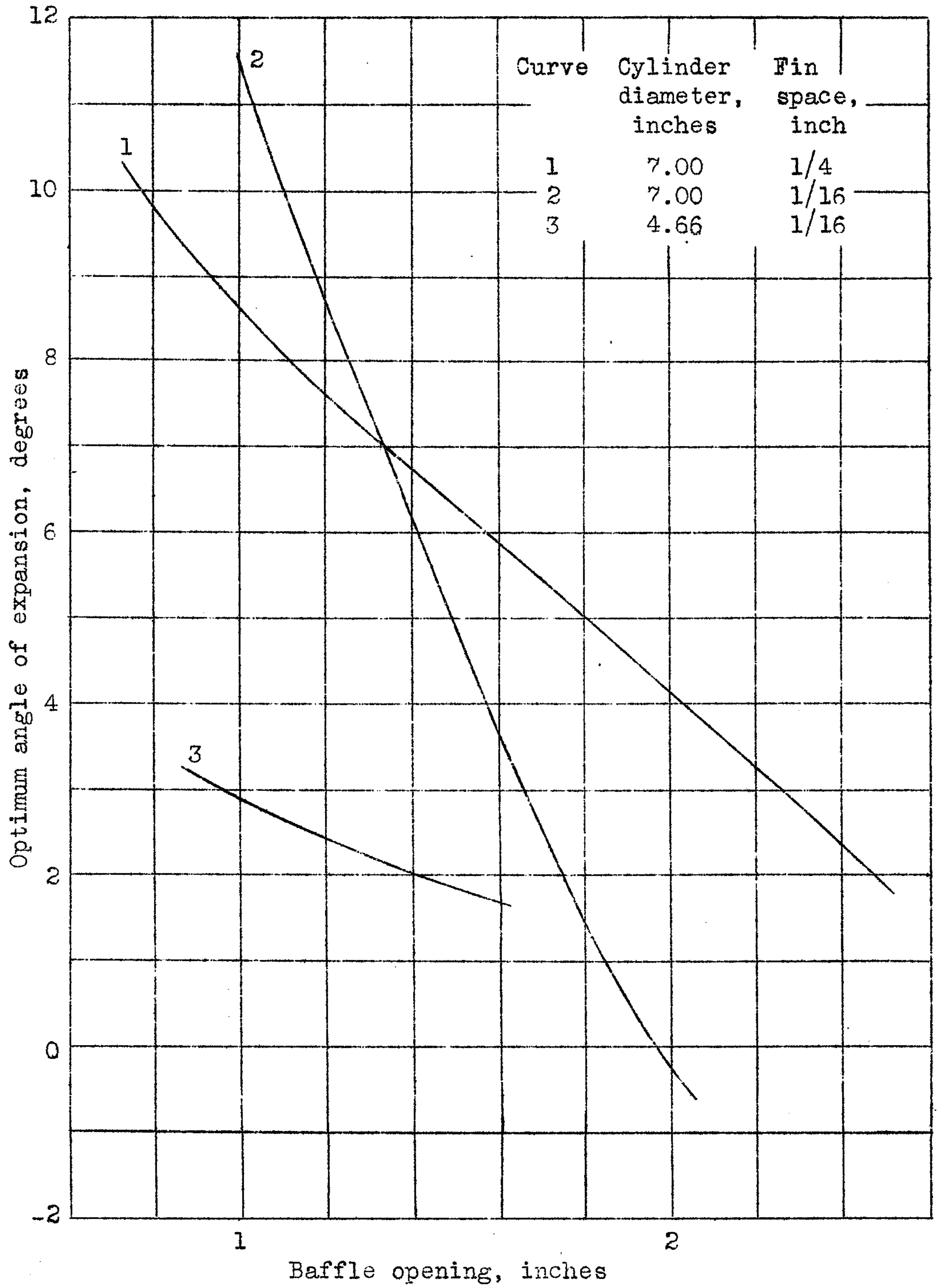

Figure 19.- Effect of baffle opening on optimum angle of expansion.

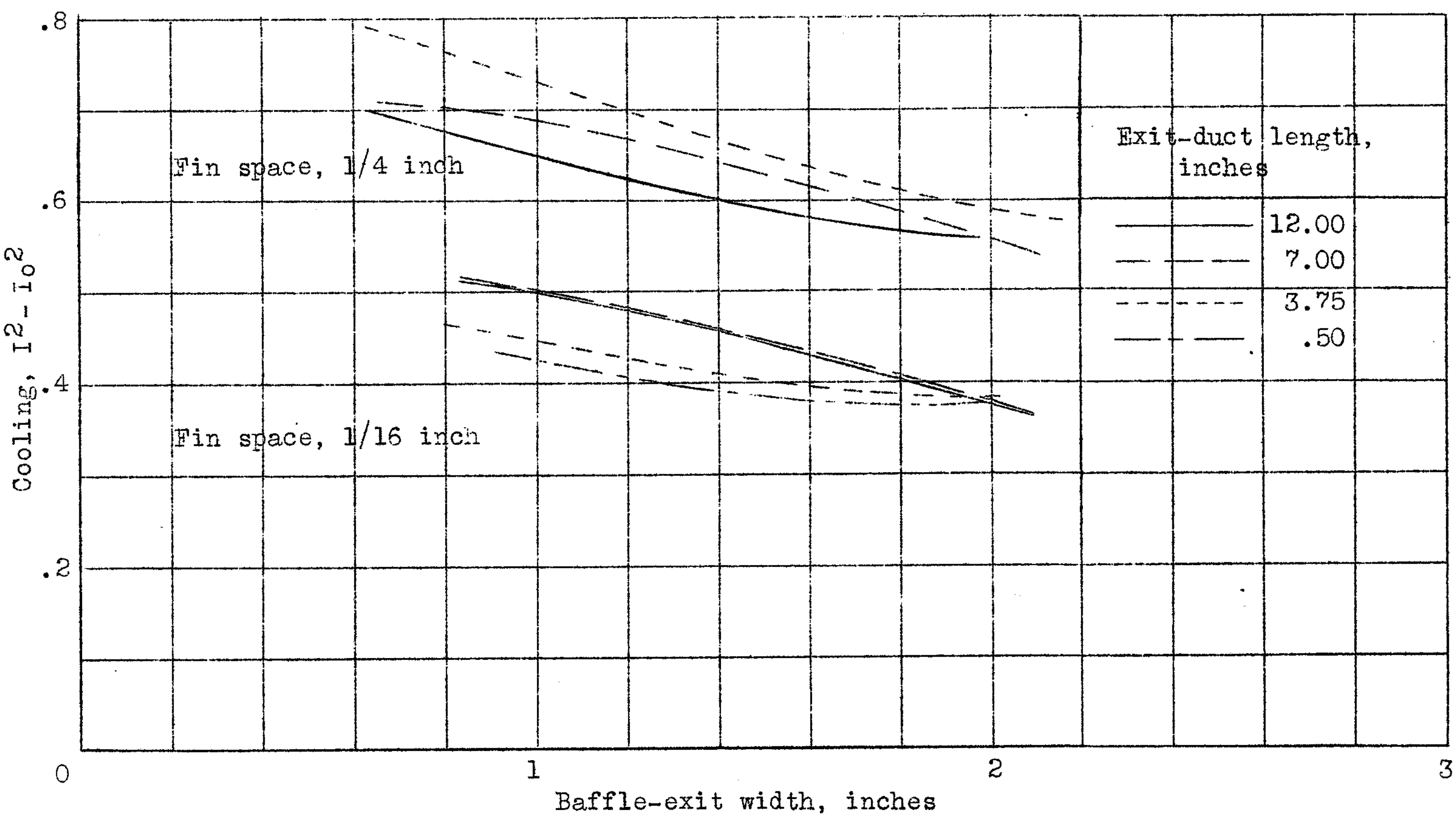

Figure 20.- Effect of baffle-exit width on cooling at rear of cylinder for two fin spacings and several exit-duct lengths at optimum exit-duct opening.
Cylinder diameter, 7.00 inches.

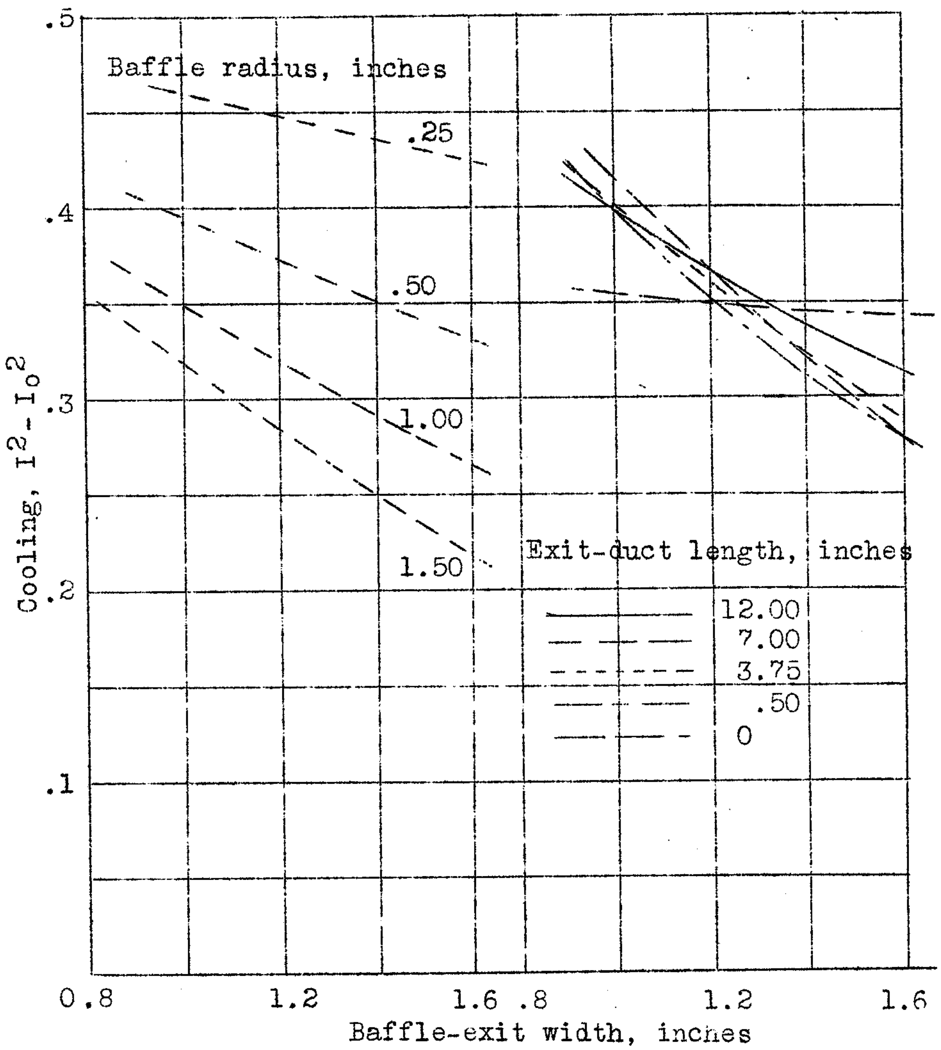

Figure 21.- Effect of baffle-exit width on cooling at rear of cylinder for several exit-duct lengths and baffle radii at optimum-exit-duct opening.
Cylinder diameter, 4.66 inches; fin space, 1/16 inch.

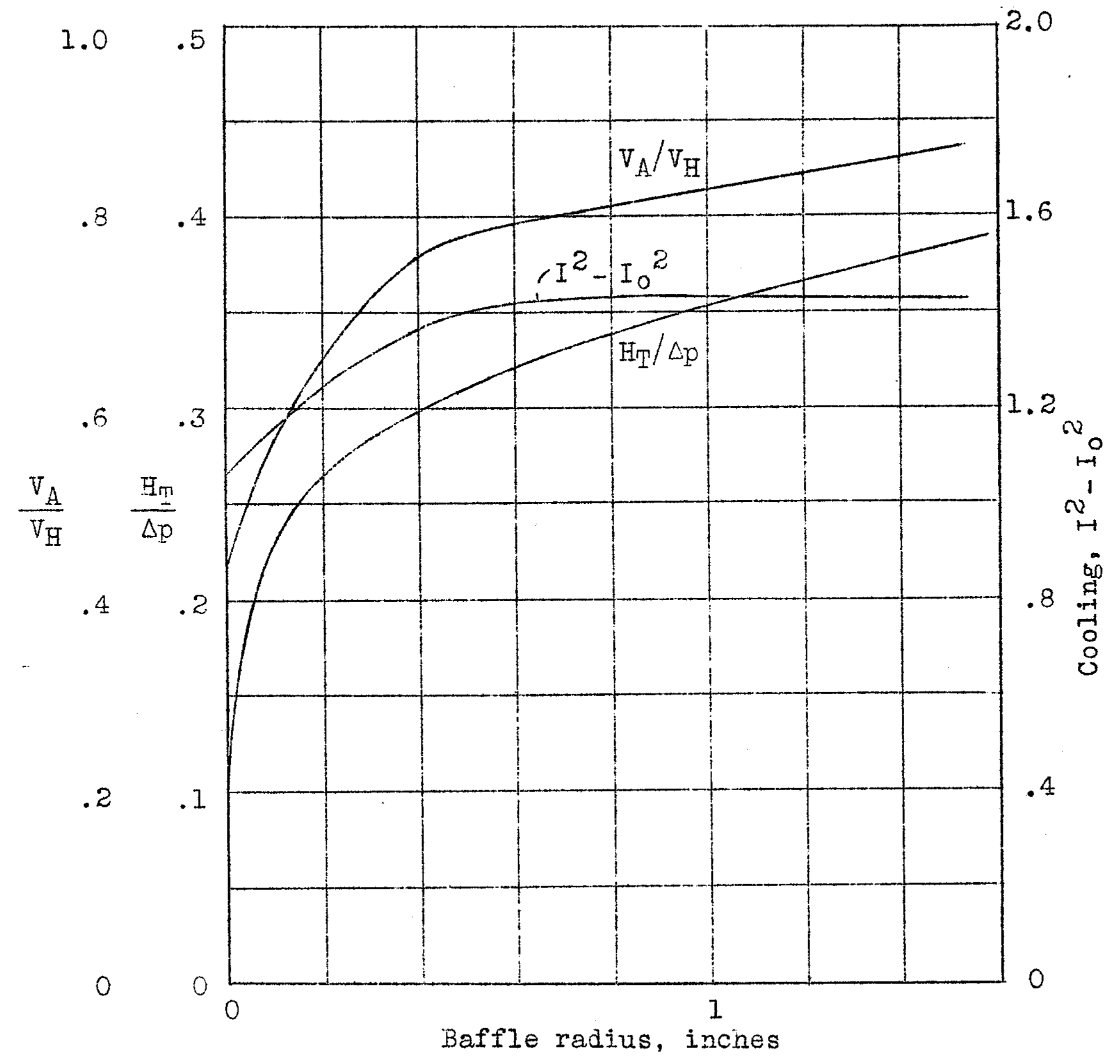

Figure 22.- Effect of baffle radius on $H_T/\Delta p$, V_A/V_H, and $I^2 - I_o^2$ at 150° from front of cylinder at optimum exit-duct opening. Baffle exit, 1.00 inch; exit-duct length, 7.00 inches. Cylinder diameter, 4.66 inches; fin space, 1/16 inch.

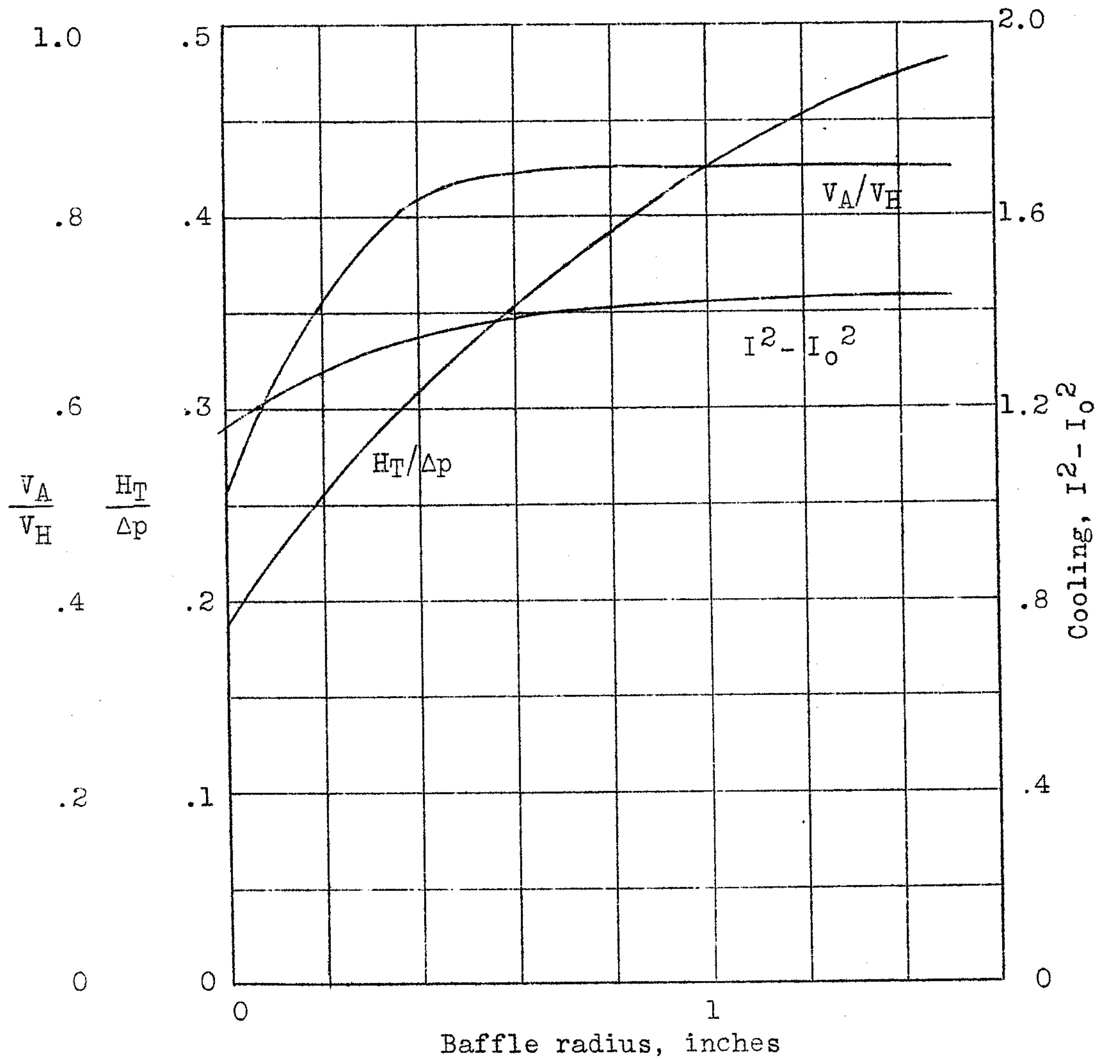

Figure 23.- Effect of baffle radius on $H_T/\Delta p$, V_A/V_H, and $I^2 - I_o^2$ at 150° from front of cylinder at optimum exit-duct opening. Baffle exit, 1.5 inches; exit-duct length, 7.00 inches. Cylinder diameter, 4.66 inches; fin spaces. 1/16 inch.