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No. 621

PRESSURE DROP ACROSS FINNED CYLINDERS

ENCLOSED IN A JACKET

By Vern G. Rollin and Herman H. Ellerbrock, Jr.
Langley Memorial Aeronautical Laboratory

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SUMMARY

The pressure drop across finned cylinders enclosed in a jacket for a range of air speeds from approximately 13 to 230 miles per hour has been investigated. Tests were made to determine the effect on the pressure drop of changes in fin space, fin width, fin shape, jacket entrance and exit areas, skirt-approach radius, and the use of fillets and a separator plate at the rear of the cylinder.

The pressure drop across the cylinder increased as the fin space decreased, the increase being very rapid at fin spaces smaller than approximately 0.20 inch. Fin width had little effect on the pressure drop for the range of widths tested. The pressure drop across the cylinder was nearly halved by increasing the skirt-approach radius from $3/4$ inch to $1-1/4$ inches, but fillets and a separator plate at the rear of the cylinder had little effect on pressure drop. The pressure drop across a cylinder with tapered fins was greater than that for a cylinder having rectangular fins with the same effective fin spacing.

INTRODUCTION

A considerable amount of work has been done in the past few years to determine the effect of various engine cowlings and cylinder baffles on pressure drop, cylinder temperatures, and temperature distribution over air-cooled cylinders. The N.A.C.A. tests have been conducted on single-cylinder and multicylinder engines (references 1 and 2) and on electrically heated cylinders (references 3 to 6). The results, in general, showed that cooling was dependent on the pressure drop across the cylinder except for the front of the cylinders of a radial air-cooled en-

gine enclosed in an N.A.C.A. cowling, for which condition the cooling was dependent to a great extent on air turbulence.

The tests of the electrically heated finned cylinders were made to determine the cooling obtained with fins of various spaces and widths. Although some measurements had been obtained of the pressure drop across the cylinders in the heat-transfer tests, additional tests were necessary to investigate the entire range of fin space and fin width. Furthermore, measurements of pressure drop would be of value in studying the use of blowers for cooling aircraft engines and in determining the fin design that could be used for cooling with the pressure head produced by the forward motion of the airplane. In the latter case, the available pressure drop is limited. The present tests were conducted to extend the range of the pressure-drop data obtained in the heat-transfer tests of electrically heated cylinders and to determine what losses occurred across the cylinder to cause such pressure drops.

The pressure drop in straight pipes and channels can be divided into several possible losses, all of which may or may not exist in a particular set-up. One loss present in all arrangements is the drop caused by friction of the air over the surfaces. Prandtl and Tietjens have shown (reference 7) that at the entrance of a pipe there is a loss caused by changing the almost constant velocity distribution which exists at the entrance to the distribution for either laminar or turbulent flow. In addition, they show a loss due to contraction if the pipe has a sharp-edge entrance. In bent pipes and in some bent channels, secondary vortices cause a pressure drop. Also where two fluid streams having different directions of flow or different velocities meet, there is a loss due to the turbulence created. Finally, there is a loss due to expansion of the air as it leaves the pipe or channel. If the air expands into a large chamber, this loss is equal to the velocity head in the pipe.

The fins of a cylinder and the enclosing jacket form a bent channel and, as the air splits into two streams, enters the fins, passes around the cylinder, and finally reunites at the rear of the cylinder, some or all of the foregoing losses may occur. Löhner, in his tests of finned cylinders enclosed in a jacket, differentiated the pressure drop across the cylinder into the drop required to overcome friction and the drop required for the vortex formation. (See reference 8.)

As a part of a program on the study of these possible losses, the present investigation was undertaken to determine the effect, on pressure drop around a cylinder, of: jacket entrance and exit width, fin space, fin width, fin shape, skirt-approach radius, and a fillet at the rear of a cylinder over a range of interfin velocities from approximately 13 to 230 miles per hour.

The N.A.C.A. tests covered in the present report were conducted during 1936 and 1937.

APPARATUS

Test Cylinders

Each of the finned cylinders used in these tests was made of two sets of disks clamped together with a 1/2-inch rod through their common central axis. (See fig. 1.) One set of large-diameter disks served as fins and the other set of 4.66-inch diameter disks was placed between the fins to serve as interfin spacers. The fin disks, with the exception of some of 1/16-inch stock used in a few tests, were made of 1/32-inch flat steel stock. Sufficient disks were assembled to form a cylinder approximately 3-1/8 inches long. For convenience in referring to these finned cylinders, designations consisting of fin space and fin width are used in this report. A cylinder having a fin space of 1/2 inch and a fin width of 3/4 inch is designated a "1/2-3/4 cylinder." The following cylinders were tested:

Fin space	Fin width
1/2, 3/8, 1/4, 1/8, 1/32	1-1/2
1/2, 1/4, 1/8, 1/16, 1/32	3/4
1/2, 1/4, 1/8, 1/16, 1/32	3/8

Jackets

The air was directed around the cylinders by wooden jackets placed in contact with the fin tips. The jackets were highly polished on the surface exposed to the air flow and were mounted around the cylinder symmetrically

with respect to its central axis (fig. 1). The jacket entrance shape (fig. 2) was chosen to reduce entrance losses to a minimum. The width of the entrance and exit passages was varied, depending on the ratio of the entrance or exit area to the free-flow area between the fins of each test specimen. The rear portion of two of the jackets was cut out, as shown by the dotted lines in figure 2, to provide space for inserting blocks to change the skirt-approach radius. (See fig. 4.)

Air System

A diagrammatic sketch of the air system used in these tests is shown in figure 3. The orifice tank was used for measuring air flow through the jacket, which was induced by means of a Roots blower connected to a variable-speed electric motor. A large tank was installed between the test unit and the blower to absorb pressure pulsations created by the blower. The air-bleed valve on the blower intake line was used as an auxiliary means of regulating the air flow through the cylinder jacket.

Instruments

The air temperature at the entrance of the orifice tank was measured with an alcohol thermometer. The static pressure in the orifice tank was measured with an inclined water manometer and in the depression tank with an inclined water manometer for pressures up to 7 inches of water, a U-tube water manometer for pressures between 7 and 18 inches of water, and a U-tube mercury manometer for pressures above 18 inches of water.

Pressures in the jacket entrance and exit were measured with pitot and static tubes, respectively, made of 0.040-inch seamless steel tubing.

TESTS

Tests were conducted to determine the effect of fin space and fin width on the pressure drop across the cylinder over a range of spaces from $1/32$ to $1/2$ inch and of fin widths from $3/8$ inch to $1-1/2$ inches through a range of air speeds from approximately 13 to 230 miles per hour,

corresponding to values of weight velocity of the air from 1.4 to 25 pounds per second per square foot, respectively. The air velocity was changed by varying the speed of the blower, small adjustments being obtained with the air-bleed valve. In these tests, the ratio of the entrance and exit areas of the jacket to the free-flow area between the fins was maintained at 1.6. This ratio was chosen as the result of tests of electrically heated cylinders (reference 4).

The 1/4-3/4 cylinder was used for tests to determine the effect of changes in jacket entrance and exit ratios on the pressure drop across the cylinder and the energy loss out of the exit passage for a range of air speeds between 13 and 93 miles per hour. ($V_2 \rho_{av} g$ ranged from 1.4 to 10 pounds per second per square foot). In the first series of tests the entrance ratio was held constant at 1.0 and the exit ratio varied (0.6, 1.0, 1.6, and 2.0). With the exit ratio of 2.0, the tests were repeated for entrance ratios of 1.0, 1.6, 2.0, 3.0, and 4.24.

A few tests were conducted of the 3/8 - 1-1/2 cylinder enclosed in a jacket resembling a conventional baffle, contacting only the rear half of the cylinder, with an exit ratio of 1.6.

The range of the tests was extended to include the effect of a change in skirt-approach radius. Tests were made with radii of the skirt approach from 0 (sharp edge) to 1-1/2 inches in 1/4-inch increments on both the 1/32 - 1-1/2 and 3/8 - 1-1/2 cylinders.

An analysis of the results of these tests showed that further tests were necessary to study the losses in the jacket exit. These tests were made of the 3/8 - 1-1/2 cylinder with a fillet placed at the rear of the cylinder (see fig. 12) and on the 1/4-3/4 cylinder with a separator plate at the rear of the cylinder (see fig. 13) when using jackets having skirt-approach radii of 1/4 inch and 1-1/2 inches, respectively.

Tests were made of the 1/32 - 1-1/2 and 3/8 - 1-1/2 cylinders to determine the effect of tapering the fins on the pressure drops across the cylinders. The fins were 1/16-inch thick and were tapered by grinding to a sharp edge from the base to the tip, giving average spaces of 0.062 inch and 0.406 inch for the two cylinders.

The tests of the skirt-approach radius, fillet separator, and tapered fins were made with jacket entrance and exit ratios of 1.6 over a range of air speeds from approximately 13 to 230 miles per hour.

A few total-head surveys were made in the entrance and static-head surveys in the exit of the jacket to compare with those in the orifice and depression tanks, respectively.

The temperature of the air entering the orifice tank and the static pressures in the orifice and depression tanks were recorded for all tests.

SYMBOLS

- A_1 , area of inlet of jacket around test cylinder, sq.in.
- A_2 , total area of spaces between fins of test cylinder, sq. in.
- A_3 , area of outlet of jacket around test cylinder, sq. in.
- A_4 , area of depression tank, sq. in.
- D , cylinder diameter at fin root, ft.
- g , acceleration of gravity, ft. per sec. per sec.
- L , cylinder length, in.
- l , equivalent length for straight tube, ft., ($l = \phi R_a$).
- p_1 , absolute static pressure of air at inlet of jacket, in. Hg.
- p_2 , absolute static pressure of air between the fins, in. Hg.
- p_3 , absolute static pressure of air at outlet of jacket, in. Hg.
- p_4 , absolute static pressure of air in depression tank, in. Hg.
- p_D , absolute total pressure of air in orifice tank, in. Hg.

- P_1 , power required to force air across cylinder per inch of cylinder length, hp.
- R_a , average radius from center of cylinder to finned surface, ft., $\left(R_a = R_b + \frac{w}{2 \times 12}\right)$.
- R_b , radius from center of cylinder to fin root, ft., $\left(R_b = \frac{D}{2}\right)$
- s , space between adjacent fin surfaces, in.
- T_1 , temperature of air at inlet of jacket, °F.
- t , thickness of fins, in.
- V_1 , average velocity of air at inlet of jacket, ft. per sec.
- V_2 , average velocity of air across the fins, ft. per sec.
- V_3 , average velocity of air at outlet of jacket, ft. per sec.
- V_4 , average velocity of air in depression tank, ft. per sec.
- W_t , weight of air flowing across cylinder, lb. per sec.
- w , fin width, in.
- $\rho_1 \text{ g}$, specific weight of air at inlet of jacket, lb. per cu. ft.
- $\rho_2 \text{ g}$, specific weight of air across fins, lb. per cu. ft.
- $\rho_3 \text{ g}$, specific weight of air at outlet of jacket, lb. per cu. ft.
- $\rho_4 \text{ g}$, specific weight of air in depression tank, lb. per cu. ft.
- $\rho_{av} \text{ g}$, average specific weight of air, lb. per cu. ft.,

$$\left(\frac{\rho_1 \text{ g} + \rho_3 \text{ g}}{2}\right)$$

- $\rho_0 g$, specific weight of air, lb. per cu. ft.,
(0.0734 lb. per cu. ft. at 80° F. and 29.92 in. Hg).
- ϕ , equivalent angle of curvature, radians.
- α , see figure 4, radians.
- α' , see figure 4, radians.
- Δp_1 , pressure drop across cylinder, in. water (from section AA to BB, fig. 4).
- Δp_2 , pressure drop due to expansion of air from exit of skirt, in. water (section BB to tank).

CALCULATIONS

The results were obtained by the following formulas.
Weight of air flowing across the test cylinder, W_t :

The method of calculating W_t from the pressure drop across the sharp-edge orifice is given in reference 9.
Specific weight of the air, ρg :

$$\rho g = \frac{1.325 p}{460 + T_1} \quad (1)$$

The pressure and temperature of the air at the section considered are used. Velocity of the air, V :

$$V = \frac{W_t}{\rho g} \frac{144}{A} \quad (2)$$

The specific weight of the air and the area at the section considered are used. Pressure drop across the cylinder, Δp_1 :

$$\Delta p_1 = (p_D - p_4) 13.52 - \Delta p_2 \quad (3)$$

The static head in the orifice tank was used instead of the total head at section AA (fig. 4) and the static head in the depression tank instead of the static head at section BB (fig. 4) as would be theoretically correct. The velocity heads in the orifice and depression tanks were

negligible. The static heads were used because they were easier to obtain than were head surveys in the entrance and exit. The use of these values leads to very little error, however, as can be seen from table I. Pressure drop out of exit, Δp_2 :

$$\Delta p_2 = \frac{1}{2} \rho_3 v_3^2 \cdot 0.1925 \quad (4)$$

Power required across the cylinder per inch of cylinder length, P_1 :

$$P_1 = \frac{0.1286 W_t}{\rho_{av} g L} \left\{ p_D - \left(p_4 + \frac{\rho_3 g v_3^2}{4553} \right) \right\} \quad (5)$$

RESULTS AND DISCUSSION

Effect of Fin Space and Fin Width

Figure 5 presents faired curves of pressure drop across the jacketed cylinder against weight velocity of the air for the range of fin spaces and fin widths tested, in jackets having entrance and exit ratios of 1.6. Throughout this report all pressure drops have been corrected to a specific weight ($\rho_0 g$) of 0.0734 pound per cubic foot. The curves hold for fin widths from 3/8 inch to 1-1/2 inches as no definite effect of fin width could be noted. The pressure drop varied approximately as the 1.93 power of the weight velocity of the air at high values of weight velocity and from the 1.48 to the 1.73 power, depending on the fin space, at low values. The break in the curves possibly indicates a change in flow conditions from viscous to turbulent or from the transition region to turbulent flow.

The first part of the report deals with the general theory of the problem. It is shown that the problem can be reduced to a set of ordinary differential equations. The solution of these equations is given in the next section.

$$\frac{d^2 y}{dx^2} + p(x) \frac{dy}{dx} + q(x)y = r(x)$$

The second part of the report deals with the numerical solution of the problem. It is shown that the problem can be solved by using the Runge-Kutta method.

$$y(x) = \int_0^x \frac{e^{-\int_0^t p(s) ds} r(t) dt}{e^{-\int_0^x p(s) ds}} dx + C_1 e^{-\int_0^x p(s) ds} + C_2$$

References

1. J. D. Murray, *Mathematical Biology*, Wiley, 1988.

The third part of the report deals with the stability analysis of the problem. It is shown that the problem is stable under certain conditions. The fourth part of the report deals with the sensitivity analysis of the problem. It is shown that the problem is sensitive to certain parameters. The fifth part of the report deals with the conclusion. It is shown that the problem can be solved by using the Runge-Kutta method.

TABLE I. COMPARISON OF PRESSURE HEADS

(1/4-3/4 cylinder; entrance ratio, 1.0; exit ratio, 0.6)

Run	$V_2 \rho_{av} g$ (lb./sec./sq.ft.)	Static head in orifice tank (in. H ₂ O)	Total head in jacket entrance (in. H ₂ O)	Static head in jacket exit (in. H ₂ O)	Static head in depression tank (in. H ₂ O)	Remarks
26	1.40	0.37	0.40	0.80	0.76	
27	1.80	.62	.65	1.25	1.20	
28	2.14	.88	.88	1.73	1.69	
29	2.86	1.57	1.57	2.99	3.06	
30	4.29	3.54	3.55	6.75	6.80	
31	8.10	3.14	3.28	14.57	14.41	
32	10.02	4.91	5.05	22.75	22.59	
2	1.38	.37	.38	.70	.78	
3	1.78	.62	.60	1.25	1.25	
4	2.15	.89	.89	1.77	1.78	
5	2.83	1.57	1.56	3.03	3.05	
6	4.26	3.55	3.54	6.64	6.82	
8	10.03	4.91	4.94	22.28	23.27	
34	1.39	.37	.34	.78	.78	
35	4.29	3.54	3.58	7.04	6.99	

One total-head tube in entrance and one static-head tube in exit, both located in center of area.

Entrance total heads obtained with bank of five tubes. Static head in exit obtained with one static tube. Surveys made in planes parallel to cylinder axis.

One total-head tube in center of entrance area. Ten static-head tubes in horizontal and vertical center lines of exit area.

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As a check on the effect of fin width, figure 6 was plotted showing the effect of weight velocity of the air on the power required per inch of cylinder length per inch of fin width across the cylinder. The power is corrected to a standard specific weight. The experimental points are plotted on the curves for a given fin space; the power varies directly as the fin width. It can be shown by equation (5) that, if the power varies directly as the fin width for a given fin space, weight velocity, and specific weight of the air, the pressure drop is unaffected by the fin width. Theoretically, fin width should have little effect on pressure drop. For a given jacket entrance ratio, jacket exit ratio, and weight velocity between the fins, the only loss mentioned in the introduction that would be affected by fin dimensions would be the drop caused by friction. The friction drop is affected by the fin dimensions as they affect the equivalent diameter $4ws/2(w+s)$. For small spaces the equivalent diameter is almost entirely unaffected by fin width and for large spaces the friction drop would be only a small part of the total drop.

Figure 7 is a cross plot of figure 5 and a similar figure taken from unpublished tests of heated cylinders to show the effect of fin space on pressure drop across the cylinders. The jacket entrance and exit ratios varied for the heated cylinders but this variation should have little effect on the pressure drops. (See figs. 8 and 9.) Although the pressure drops of the heated cylinders were slightly greater than those for the cold cylinders under the same conditions, the curves in figure 7 are representatives of both hot and cold cylinders for all practical purposes. Figure 7 shows a rapid increase in pressure drop at small fin spaces.

Effect of Jacket-Exit Area

Figure 8 shows the effect of varying the exit ratio of the jacket from 0.6 to 2.0 on the pressure drop (Δp_1) across the $1/4$ - $3/4$ cylinder for an entrance ratio of 1.0. When the exit ratio was changed the width of the rectangular passage was varied but the skirt-approach radius was maintained at $1-1/2$ inches. The change in exit area had no definite effect on the pressure drop (Δp_1) across the cylinder, as shown in the figure, but it did affect the energy loss from the skirt exit. It can be shown from equation (4) that, for a given weight velocity of air

across the fins and a constant specific weight of the air, the pressure drop from the skirt exit (Δp_2) varies inversely as the square of the exit area. Small errors in manometer readings may be responsible for the scattering of points at the lower weight velocities in the figure. The pressure drops shown in figure 8 were divided by the length of the flow path of the air around the cylinder equal to $(\alpha + \alpha') R_a$ (see fig. 4) to determine whether the length of the flow path reduced the dispersion of the points. Since the length of the flow path showed no effect, a mean line was drawn through the data.

Effect of Jacket-Entrance Area

A decrease in pressure drop Δp_1 across the 1/4-3/4 cylinder was noticed when the jacket entrance ratio was increased (fig. 9(a)). Increasing the entrance ratio decreased the angle of contact and possibly the flow path. Therefore, the pressure-drop data were divided by the length of the flow path, $l = (\pi/2 + \alpha) R_a$, to determine whether any relationship existed between a change in angle of contact of the jacket with the cylinder and the pressure drop. The results are plotted in figure 9(b). An angle of $\pi/2 + \alpha$ radians was used instead of $\alpha + \alpha'$ because, in the tests where the exit ratio was changed, α' had little effect on the pressure drop Δp_1 . With the results plotted in this manner, the points fell very nearly on the same curve, except at low weight velocities, showing that the increase in pressure drop with decrease in entrance ratio is approximately proportional to the increase in length of the flow path around the cylinder.

A test was made of a 3/8 - 1-1/2 cylinder enclosed in a jacket for which $\alpha = 48^\circ$ and in a baffle for which $\alpha = 0^\circ$, as shown in figure 10. These tests were made to determine whether the relationship between the length of the flow path and the pressure drop as determined held for the conventional baffle. Figure 10(a) shows the pressure drop across the cylinder with the baffle and jacket and figure 10(b) shows the same pressure drops divided by the length of the flow path. For the jacket, l was based on 138° and for the baffle, on 90° . The pressure drop per unit length of flow path was greater with the baffle than with the jacket.

Calculations were made of the drop due to friction and vortex loss for the 48° difference between the jacket and the baffle, but the calculated pressure differences were less than the difference between the jacket and the baffle. Some loss that cannot be isolated at present is probably present at the front of the jacketed cylinder. Further tests will be made to isolate these losses.

Effect of Radii of Skirt Approach, Fillet, and Separator

When two fluid streams having different directions of flow or velocities meet, there is a loss due to the turbulence created. The air in passing around the cylinder splits into two streams and in reuniting at the rear such a loss may occur. With this fact in mind, several tests were made to study the losses at the rear of the cylinder.

Figure 11 shows the effect on the pressure drop of changing the radii of the skirt approach from 0 (sharp edge) to $1-1/2$ inches across the $1/32 - 1-1/2$ and $3/8 - 1-1/2$ cylinders. For both fin spaces, the pressure drop decreases for each increase in radii up to the $1-1/4$ inch radius. Radii larger than $1-1/4$ inches did not decrease the pressure drop. With the $3/8$ -inch fin space, the pressure drop with a sharp corner was more than twice that with the $1-1/4$ inch radius. With the $1/32$ -inch fin space, the percentage difference in pressure drops was almost as great as for the $3/8$ -inch space, showing that, with a small radius, about 50 percent of the drop across the cylinder occurs in the skirt. This result would explain some of the large losses occurring at the rear of some baffles designed and used by various investigators. Schey and Rollin (reference 4) made tests of baffles in which $3/4$ -inch, $1-1/8$ inch, and $4-1/4$ inch radii were used; the $1-1/8$ inch radius was found to be best from heat-transfer considerations. These results indicate that this value would also be satisfactory on the basis of pressure drop.

The tests with variable radii did not satisfactorily determine the exact cause of loss in the skirt. Two possibilities were considered. The large radii may have turned the two fluid streams so that they met tangent to each other. Even with the small radii, the two fluid streams may have met tangentially but a loss may have occurred owing to the break-away of the air from the walls of the jacket at the rear, which the large radii eliminat-

ed. Tests were therefore made with a fillet placed at the rear of the $3/8 - 1-1/2$ jacketed cylinder using a $1/4$ -inch skirt-approach radius, as shown in figure 12. The wooden fillet was slotted to fit tightly against the cylinder wall and to guide the air into the exit so as to eliminate the impinging of the two streams. The curves show that the fillet had little effect on the pressure drop across the cylinder. Similar tests were made with a fillet in the exit of a jacket having a $1-1/2$ inch skirt-approach radius with the same result. With small radii of skirt approach, the loss in the skirt exit is probably due to break-away of the air from the wall of the jacket.

Figure 13 shows the effect on the pressure drop of using a separator at the rear of the $1/4-3/8$ cylinder. A sketch of the separator is included in the figure. The thickness of the separator was considered in calculating the pressure drop across the cylinder. No decrease in pressure drop was noted when the separator was used.

Effect of Tapered Fins

The air in entering the fins may undergo a contraction with a resulting drop in pressure. Likewise, as the air leaves the fins at the rear, an expansion of the air may ensue owing to the sudden increase in area (due to the fin thickness), causing a loss. A few tests were made with fins tapered from the base to the tip to try to eliminate such losses if they existed by allowing a gradual increase in expansion of the air. Figure 14 shows the results of tapering the fins of the $1/32 - 1-1/2$ and $3/8 - 1-1/2$ cylinders. The area of the trapezoid formed by the fins, base, and jacket was used in calculating the weight velocity of the air for the tapered fins. The curves for the rectangular fins shown in figure 14, for the $1/32 - 1-1/2$ and $3/8 - 1-1/2$ cylinders, were taken from the curves in figure 11 for the $1-1/2$ inch skirt-approach radius. When these curves are compared, it is seen that the pressure drop is reduced for the tapered fins. The average spaces for the tapered fins, however, are 0.062 and 0.406 inch and, in the determination of their effect on pressure drop, they should be compared with cylinders having rectangular fins and the same spacing because the surface heat-transfer coefficient depends principally on the average fin space (reference 3). Pressure-drop data taken from figure 7 for rectangular fins having fin spacing equal to the effective fin spacing for the tapered fins were

plotted in figure 14. On the basis of equal effective fin spacing, the lower pressure drop is obtained with the rectangular fins. Although the foregoing losses may be reduced or eliminated, it is possible that additional turbulence may be created by the trapezoidal shape of the flow passage, which might cause a pressure drop that exceeds the reduction in pressure drop due to tapering the fins.

General Considerations

From the results of the present tests it is difficult to isolate the losses occurring from the front to the rear of the cylinder. With small radii of skirt approach, approximately 50 percent of the pressure drop across the cylinder can be accounted for in the skirt. This large loss may be caused by the break-away of the air from the wall of the jacket as it passes into the skirt. The exit area had little effect on the pressure drop Δp_1 when using large skirt-approach radii, and the entrance area affected the pressure drop only in proportion to the change in length of flow path (except for the baffled cylinder). This fact seems to indicate that, whatever losses are occurring, a large part of them occur around the cylinder proper.

A study of total heads between the fins of baffled cylinders showed no sharp increase in pressure drop at the baffle entrance, so the velocity profile and the jet contraction losses occurring at the entrance of sharp-edge pipes may not be present as the air enters the fins. Calculations of the pressure drop due to friction and the vortex formation from the 90° to 135° position on these baffled cylinders showed that the experimental pressure drop was a little greater than the calculated drop. Formulas for friction drop and vortex formation for pipes were used in these calculations. The length of the flow path used was equal to ϕR_a where ϕ was $\pi/4$ radians and the equivalent diameter was used in place of the pipe diameter. The friction-drop formula was for a straight pipe but the flow was turbulent and it has been shown that, in such a case, the pressure drop is approximately the same in bent as in straight pipes. The existence of secondary vortices between fins would have to be established from smoke-flow investigations.

Although from 90° to 135° the calculated pressure drop was a little lower than the observed total-head tube

drop, the difference was not appreciable. From 135° to 150° , however, the difference is greater, showing a sharp rise in the total heads. An expansion loss between the fins is thought to occur in this region. A study of some smoke-flow pictures seemed to indicate the presence of such an expansion. From the 150° position to the exit there is a small pressure drop, which can be accounted for almost entirely by friction and vortex formation. It is possible that the air expands between the fins at the rear to an area greater than any exit area tested during the present investigation, resulting in no expansion loss as the air leaves the fins and enters the exit passage. In such a case, the exit area would show no effect on pressure drop.

The foregoing discussion has been based on the pressure drop across the cylinder and does not include the loss at the exit of the skirt. A properly designed skirt will appreciably reduce this loss. Further tests with pitot-static tubes are required to study the losses around the cylinder, especially in the region from 135° to the exit. These tubes should be placed close together in this region. The few total-head curves available seemed to bear out the results of this investigation in that the pressure-drop losses were mainly confined to the cylinder proper when jackets having large skirt-approach radii were used.

CONCLUSIONS

Over the range of the tests:

1. The pressure drop across the cylinder increased as the fin space decreased, the increase being very rapid at fin spaces below approximately 0.20 inch.
2. Changes in fin width had little effect on the pressure drop across the cylinder.
3. A change in the ratio of the jacket-exit area to the free-flow area between the fins had little effect on the pressure drop across the cylinder.
4. The effect of a change in the ratio of the jacket-entrance area to the free-flow area between the fins on the pressure drop across the cylinder was approximately directly proportional to the length of the flow path around

the cylinder. The pressure drop divided by the flow-path length for the baffle with an opening of 180° in front was a little greater than for the jackets.

5. Increasing the radii of skirt approach of the jacket reduced the pressure drop across the cylinder appreciably up to a $1\text{-}1/4$ inch radius. Radii greater than $1\text{-}1/4$ inch did not reduce the pressure drop.

6. Fillets or a separator plate placed at the rear of the cylinder had little effect on the pressure drop across the cylinder.

7. The pressure drop across a cylinder with tapered fins was greater than that across a cylinder with rectangular fins having the same effective space.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., October 11, 1937.

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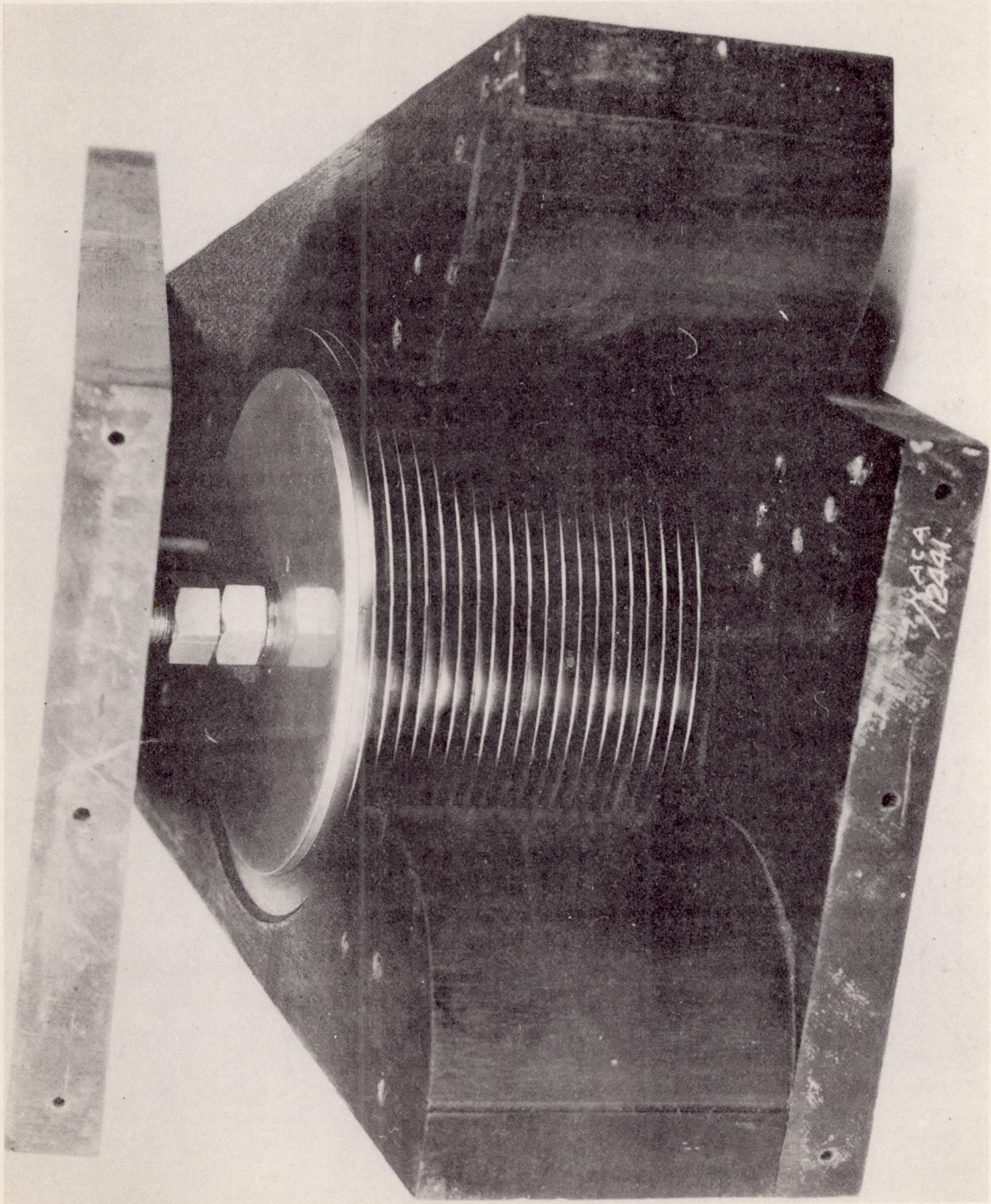


Figure 1.- Finned cylinder and a jacket.

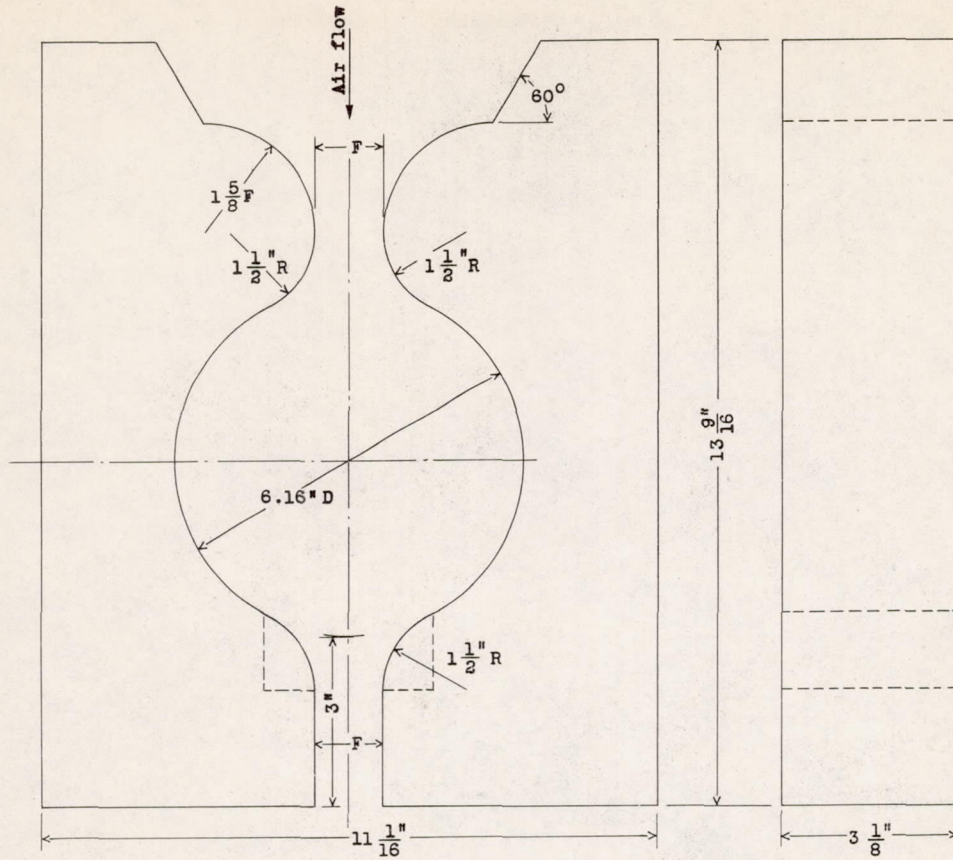


Figure 2.- Sketch of cylinder jacket.

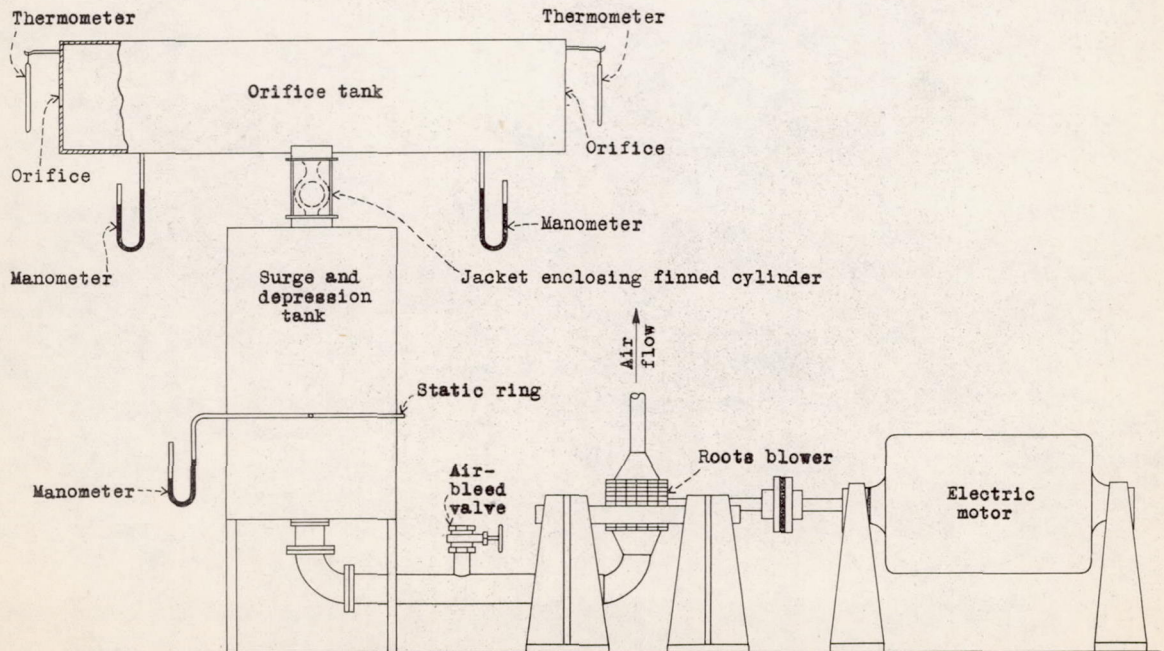


Figure 3.- Diagrammatic set-up of test equipment.

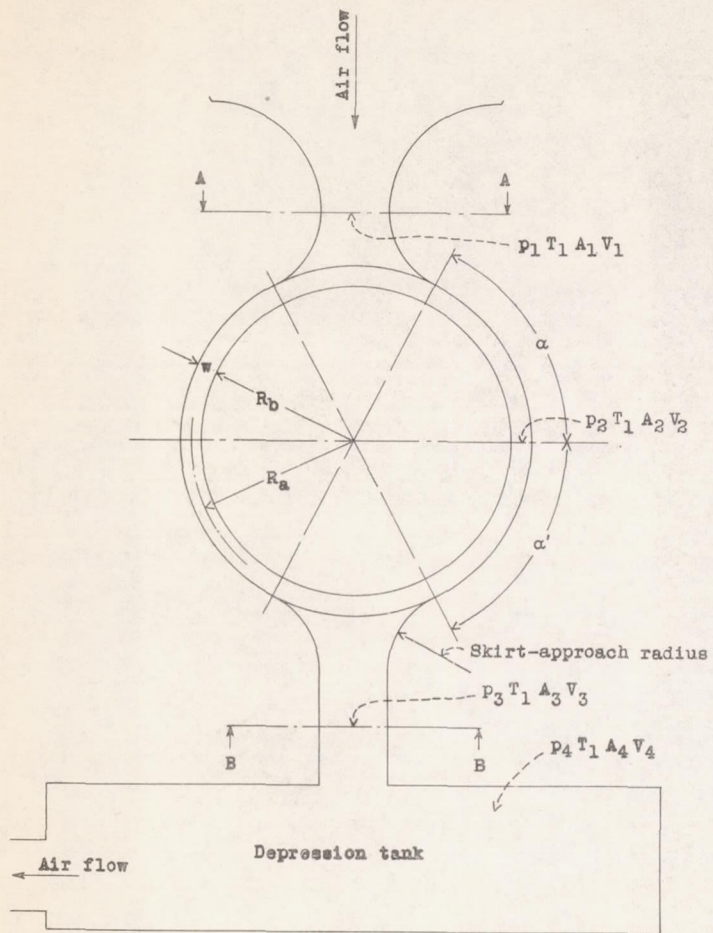


Figure 4.- Diagram of finned cylinder enclosed in a jacket.

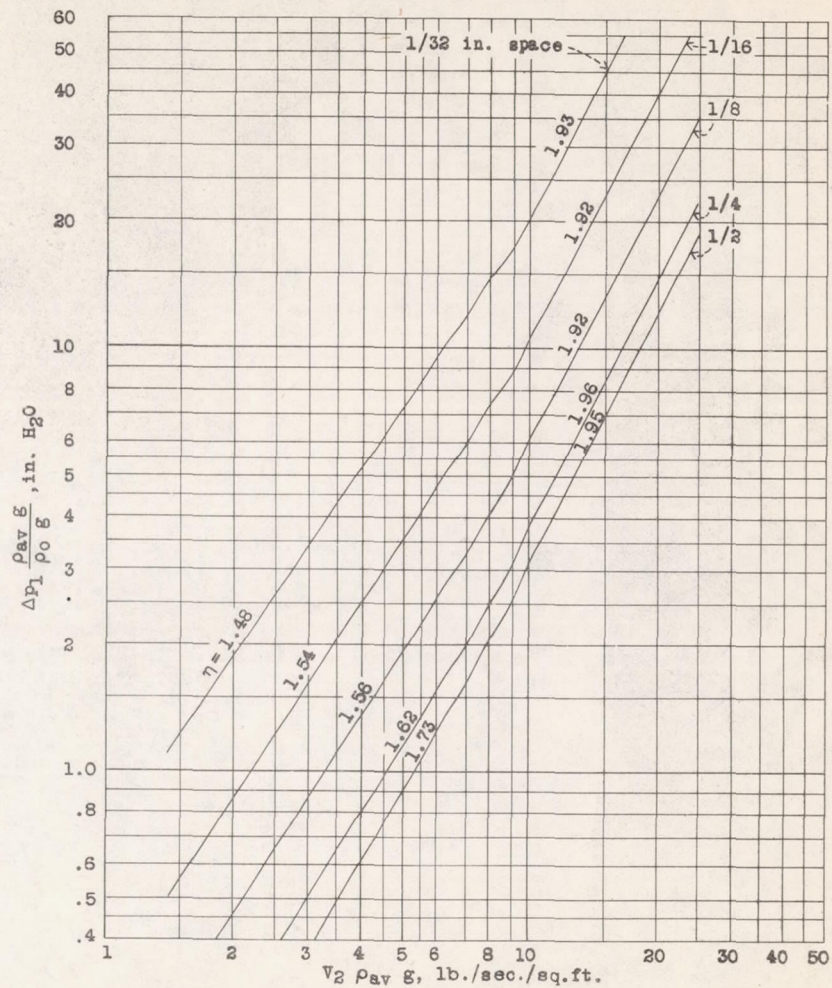
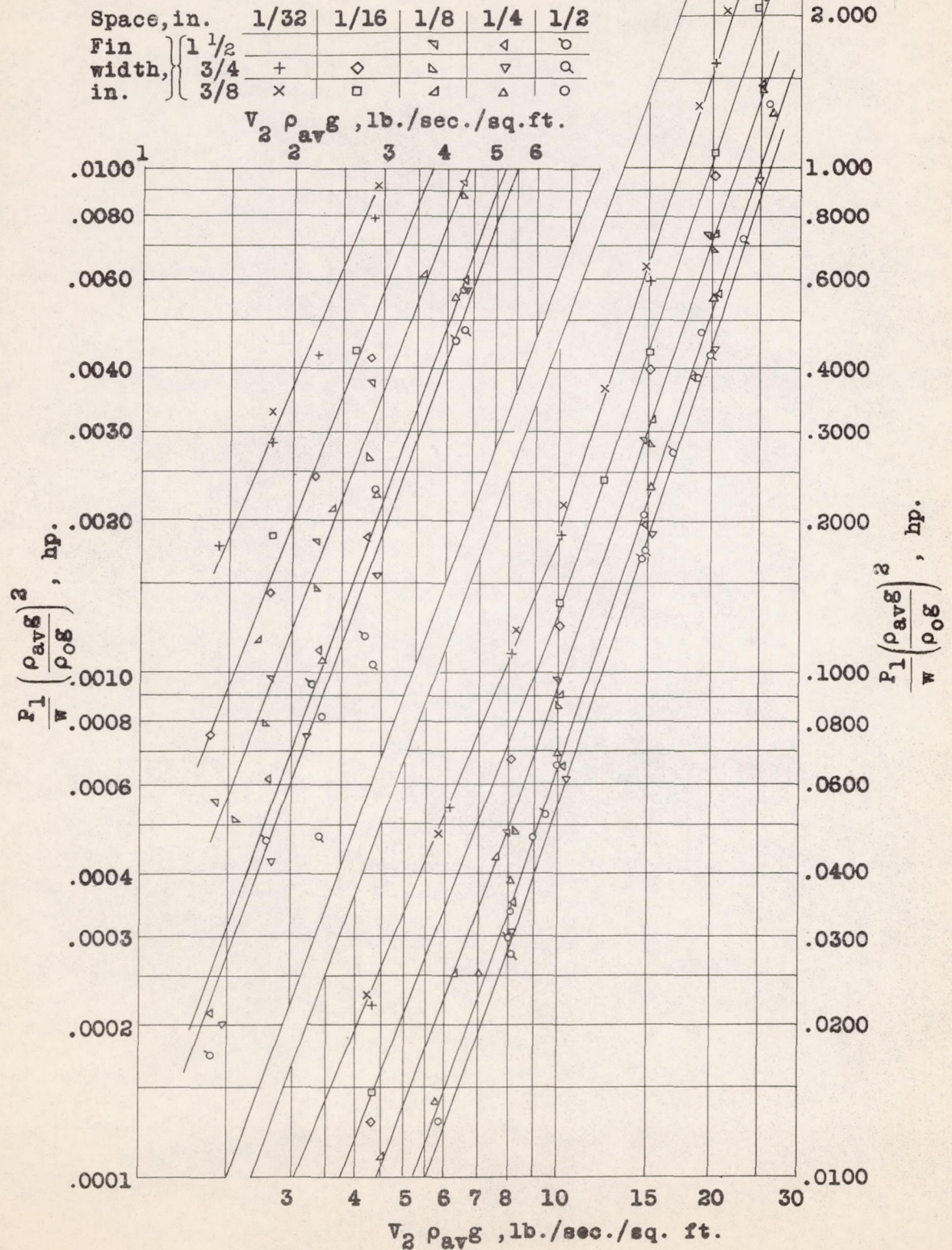


Figure 5.- Effect of weight velocity of the air on pressure drop across the cylinder for various spaces. Fin width, 3/8 to 1 1/2 inches; entrance and exit ratios, 1.6

Figure 6.- Effect of weight velocity of the air on power required across the cylinder per inch of fin width for various spaces. Entrance and exit ratios, 1.6.



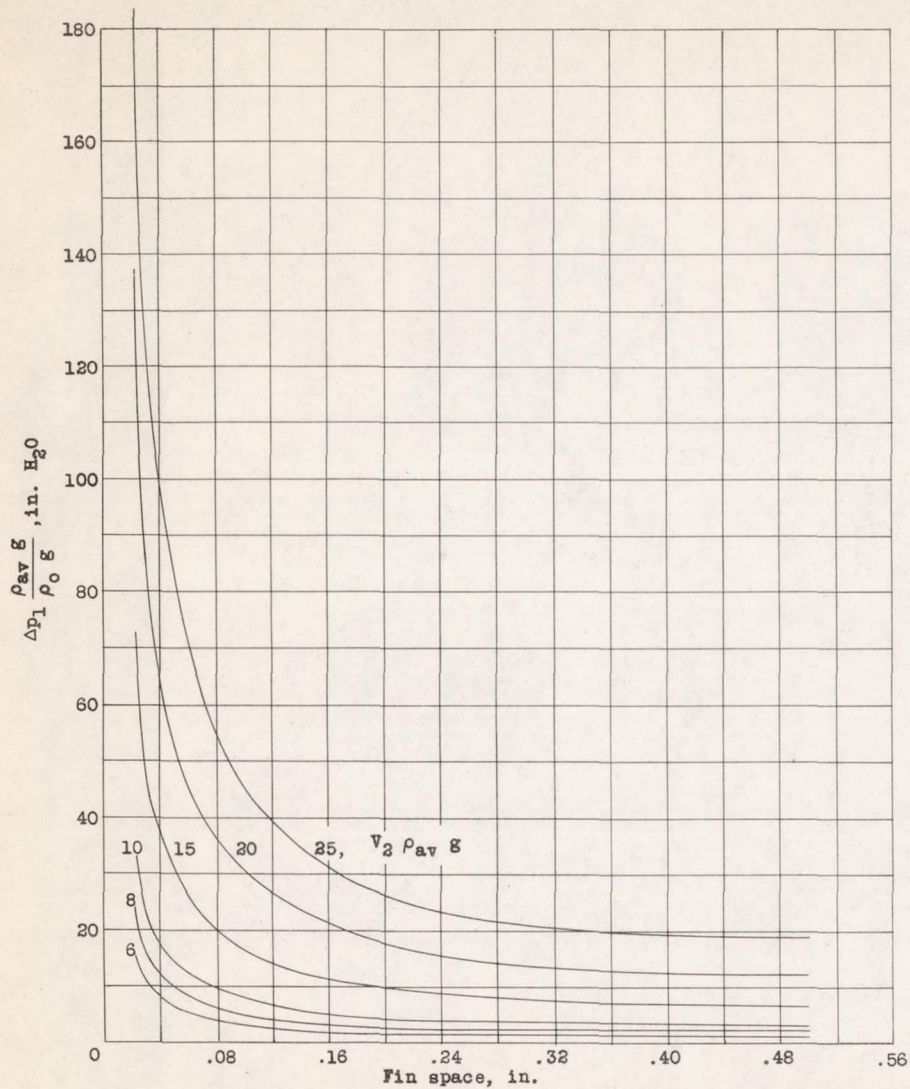


Figure 7.- Effect of fin space on pressure drop across finned cylinders at several weight velocities for fin widths from 3/8 to 1 1/2 in.

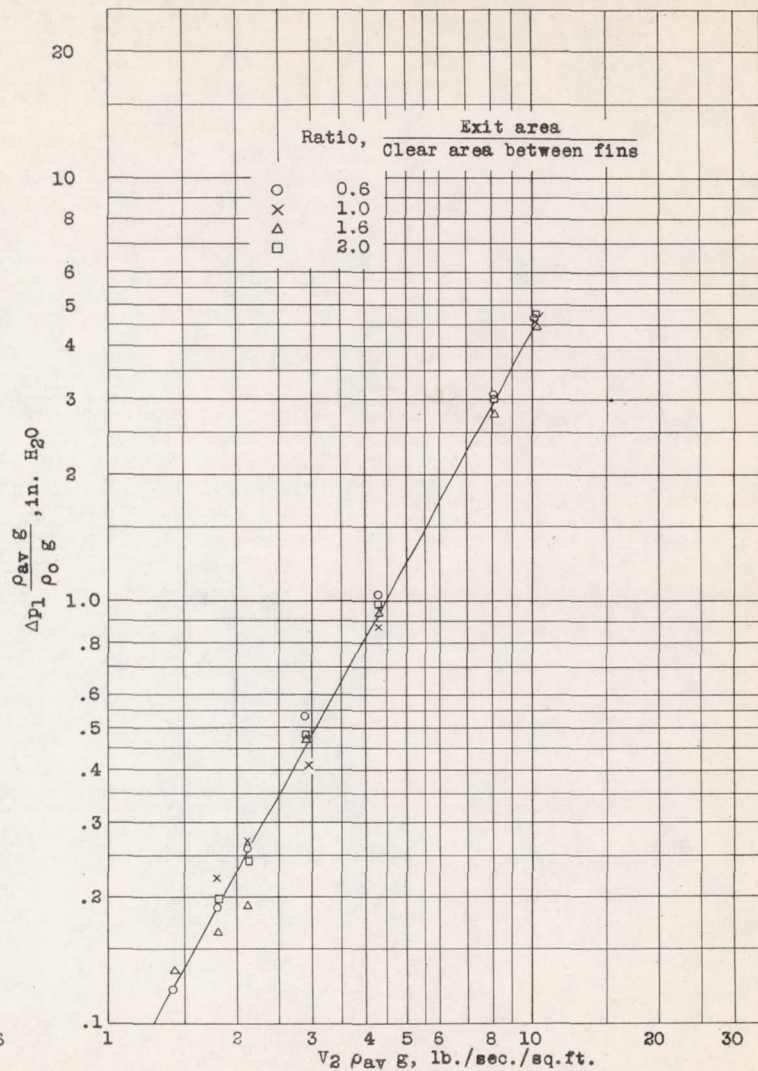


Figure 8.- Effect of exit area of jacket on pressure drop across the 1/4 - 3/4 cylinder. Entrance ratio, 1.0

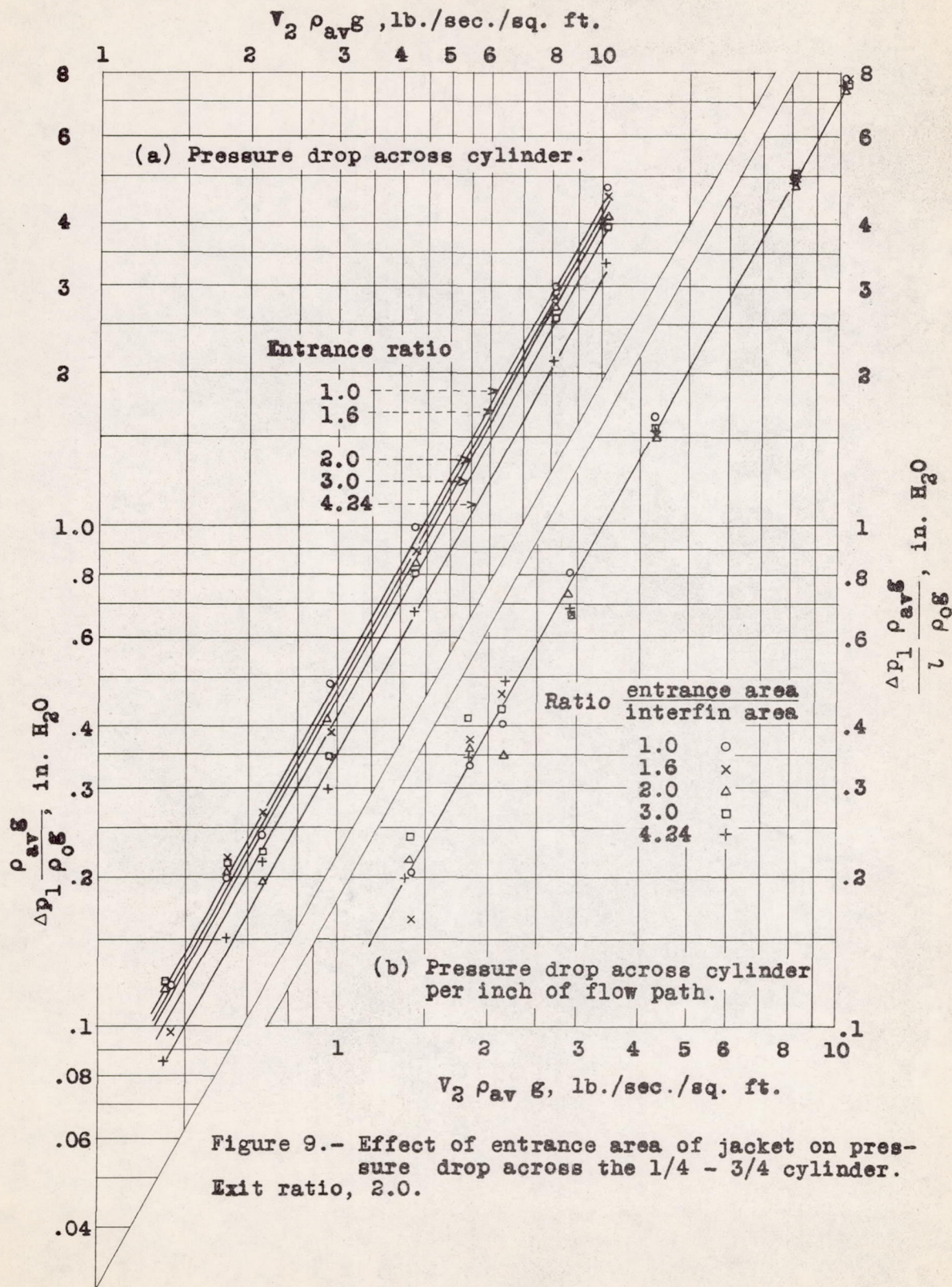


Figure 9.- Effect of entrance area of jacket on pressure drop across the 1/4 - 3/4 cylinder. Exit ratio, 2.0.

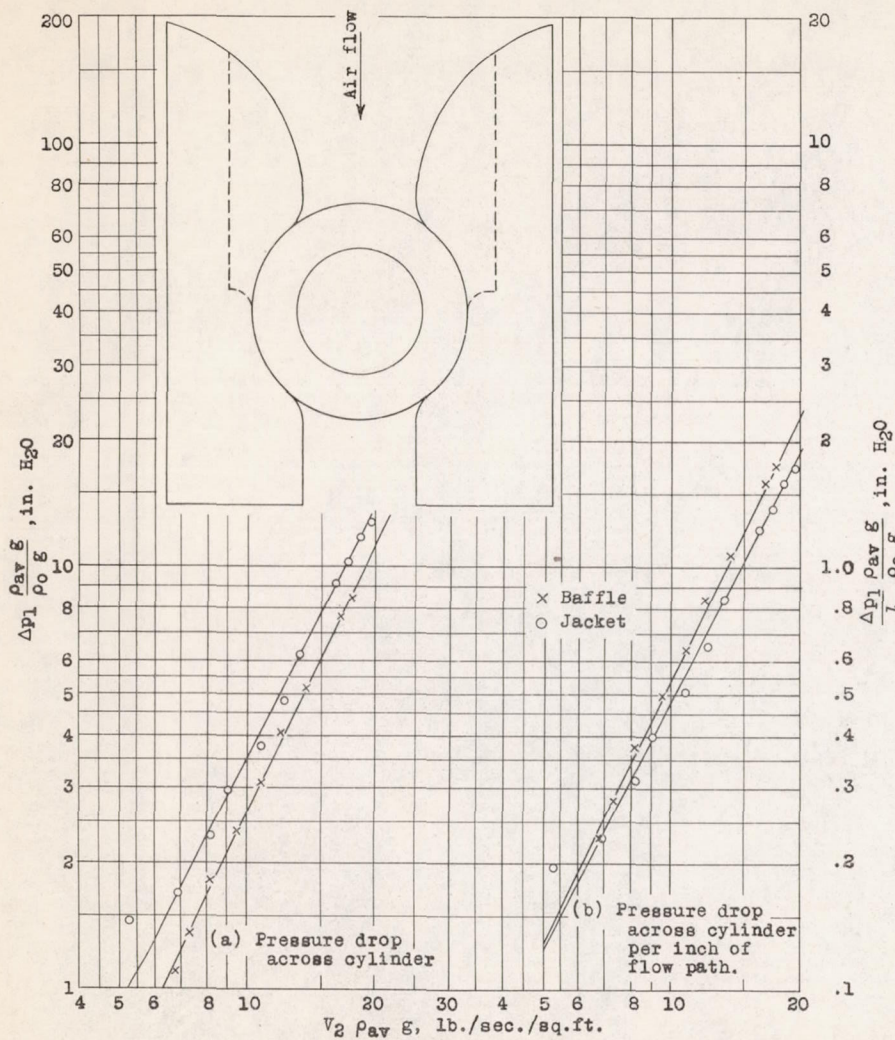


Figure 10.- Comparison of pressure drop across the 3/8-1 1/2 cylinder with a baffle and in a jacket. Entrance and exit ratios, 1.6. Dotted lines show modified entrance for baffle tests.

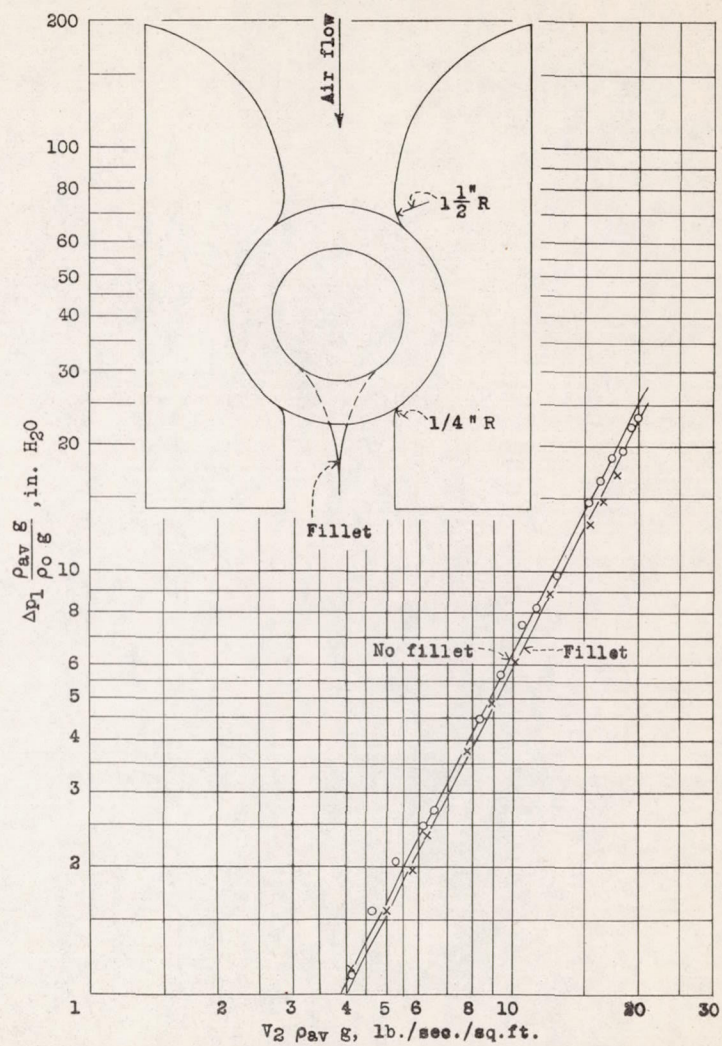


Figure 12.- Effect of a fillet on the pressure drop across a 3/8-1 1/2 cylinder. Entrance and exit ratios, 1.6

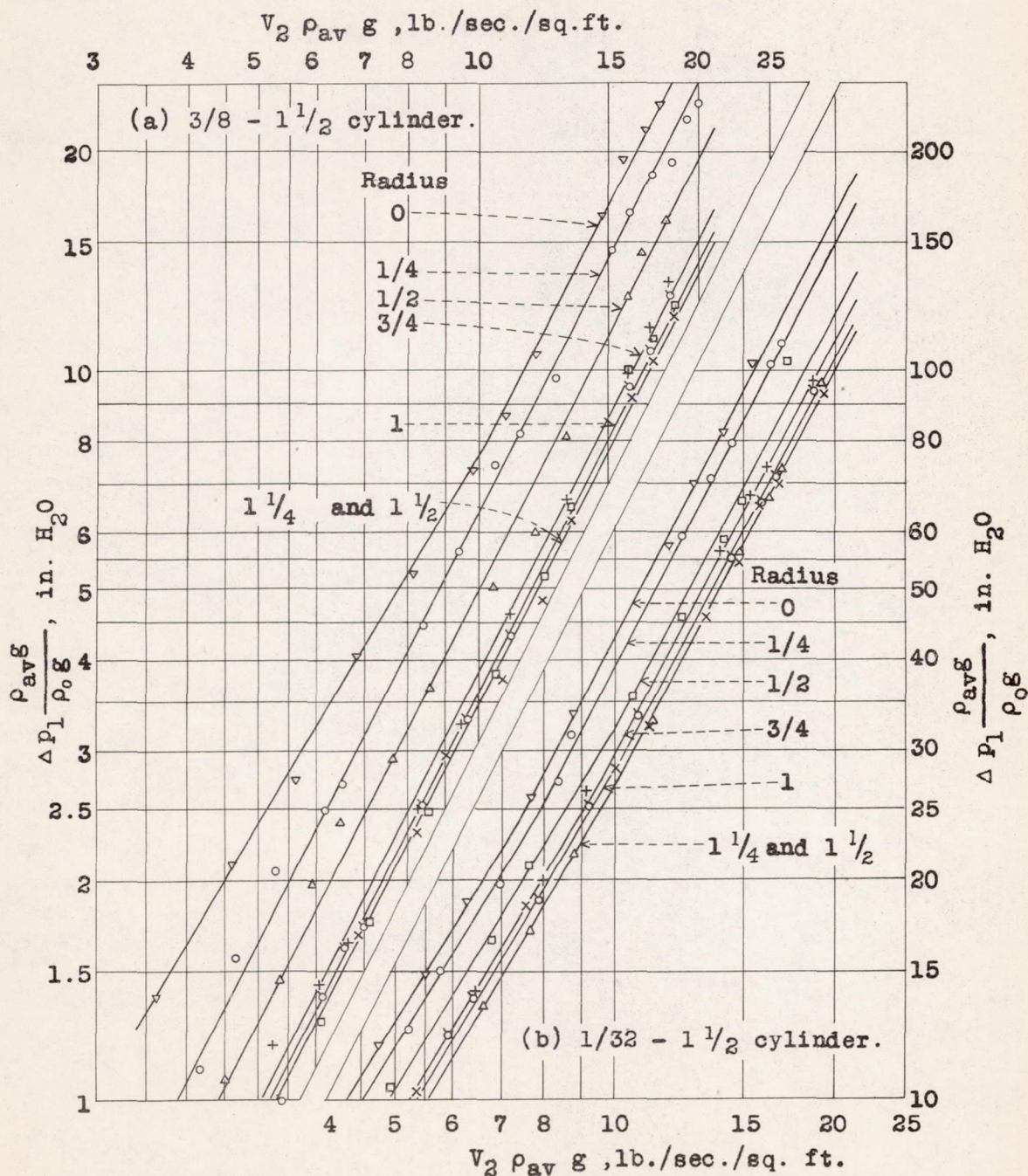


Figure 11.- Effect of radii of skirt approach on pressure drop across the $\frac{3}{8} - 1\frac{1}{2}$ and $\frac{1}{32} - 1\frac{1}{2}$ cylinders. Entrance and exit ratios 1.6

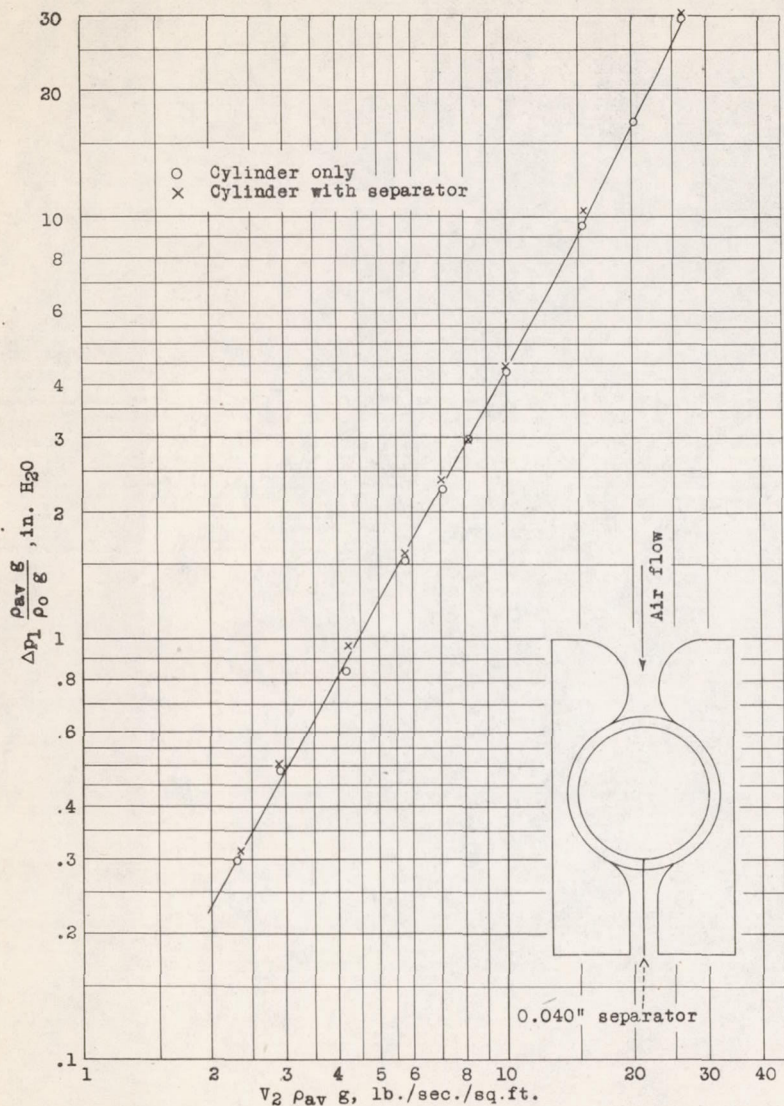


Figure 13.- Effect of a separator on the pressure drop across a 1/4-3/8 cylinder. Entrance and exit ratios, 1.6

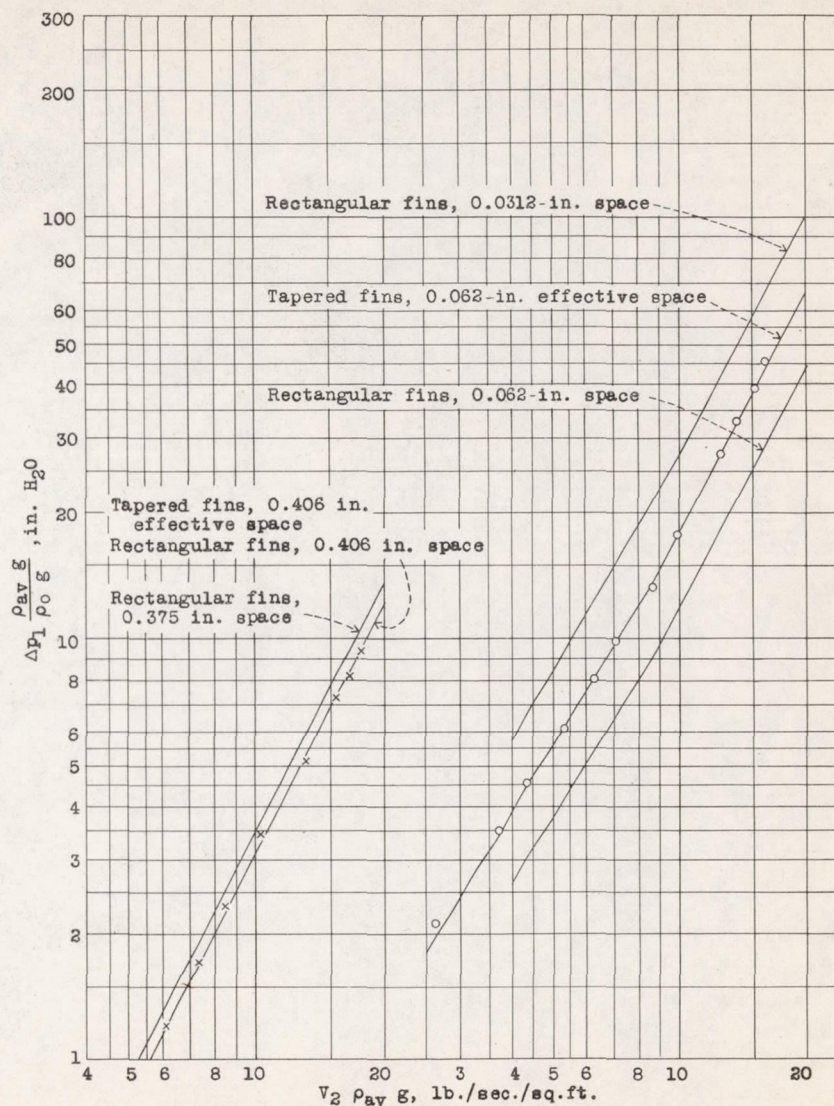


Figure 14.- Effect of tapered fins on pressure drop across a jacketed cylinder.