NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 511

THE EFFECT OF BAFFLES ON THE TEMPERATURE DISTRIBUTION AND HEAT-TRANSFER COEFFICIENTS OF FINNED CYLINDERS

By OSCAR W. SCHEY and VERN G. ROLLIN

1934

For sale by the Superintendent of Documents, Washington, D. C. - Price 10 cents
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AERONAUTIC SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Abbreviation	Unit	Abbreviation
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	s	second (or hour)	sec. (or hr.)
Force	F	weight of 1 kilogram	kg	weight of 1 pound	lb.
Power	P	horsepower (metric)		horsepower	hp.
Speed	V	kilometers per hour	k.p.h.	miles per hour	m.p.h.
		meters per second	m.p.s.	feet per second	f.p.s.

2. GENERAL SYMBOLS

W, Weight $=mg$

g, Standard acceleration of gravity $=9.80665$ m/s^2 or 32.1740 ft./sec.^{2}

m, Mass $=\frac{W}{g}$

I, Moment of inertia $=mk^2$. (Indicate axis of radius of gyration k by proper subscript.)

μ, Coefficient of viscosity

ν, Kinematic viscosity

ρ, Density (mass per unit volume)

Standard density of dry air, 0.12497 kg-m^{-4}-s^2 at 15° C. and 760 mm; or 0.002378 lb.-ft.$^{-4}$ sec.^{2}

Specific weight of "standard" air, 1.2255 kg/m^3 or 0.07651 lb./cu.ft.

3. AERODYNAMIC SYMBOLS

S, Area

S_w, Area of wing

G, Gap

b, Span

c, Chord

$\frac{b^2}{S}$, Aspect ratio

V, True air speed

q, Dynamic pressure $=\frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L=\frac{L}{qS}$

D, Drag, absolute coefficient $C_D=\frac{D}{qS}$

D_o, Profile drag, absolute coefficient $C_{D_o}=\frac{D_o}{qS}$

D_i, Induced drag, absolute coefficient $C_{D_i}=\frac{D_i}{qS}$

D_p, Parasite drag, absolute coefficient $C_{D_p}=\frac{D_p}{qS}$

C, Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

R, Resultant force

i_w, Angle of setting of wings (relative to thrust line)

i_t, Angle of stabilizer setting (relative to thrust line)

Q, Resultant moment

Ω, Resultant angular velocity

$\rho\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension (e.g., for a model airfoil 3 in. chord, 100 m.p.h. normal pressure at 15° C., the corresponding number is 234,000; or for a model of 10 cm chord, 40 m.p.s. the corresponding number is 274,000)

C_p, Center-of-pressure coefficient (ratio of distance of *c.p.* from leading edge to chord length)

α, Angle of attack

ϵ, Angle of downwash

α_o, Angle of attack, infinite aspect ratio

α_i, Angle of attack, induced

α_a, Angle of attack, absolute (measured from zero-lift position)

γ, Flight-path angle

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THE EFFECT OF BAFFLES ON THE TEMPERATURE DISTRIBUTION AND HEAT-TRANSFER COEFFICIENTS OF FINNED CYLINDERS

By Oscar W. Schey and Vern G. Rollin

SUMMARY

An investigation was made to determine the effect of baffles on the temperature distribution and the heat-transfer coefficient of finned cylinders. The tests were conducted in a 30-inch wind tunnel on electrically heated cylinders with fins of 0.25- and 0.31-inch pitch.

Four types of baffles were tested: Plates mounted at various positions around the cylinder and at various angles with respect to the air stream; streamline baffles mounted near the rear of the cylinder; shell baffles with variously shaped openings mounted around the test cylinder symmetrically with respect to a plane through the axis of the cylinder and parallel to the air stream; and integral baffles composed of strips welded to the tips of the fins of the test cylinder to form passages between the fins through which the air could flow to the rear of the cylinder.

The results of these tests showed that the use of integral baffles gave a reduction of 31.9 percent in the rear-wall temperatures and an increase of 54.2 percent in the heat-transfer coefficient as compared with a cylinder without baffles.

Although the effects of the shell baffles were not equal to those of the integral baffles, they gave a large reduction in rear-wall temperatures and a large increase in the heat-transfer coefficient. The best results were obtained with the shell baffles mounted in contact with the fins, when the intake opening was equal to the arc subtended by an angle of approximately 145°, when the extensions were 3 inches or more in length, and when the ratio of exit area to area between fins was approximately 1.6. The heat-transfer coefficient with the best shell baffles varied as the air speed to the 0.85 power.

INTRODUCTION

The present trend in air-cooled engine design toward decreased frontal area and greater specific-power output requires that all possible means be investigated for increasing the efficiency with which waste heat is dissipated from the cylinder to the cooling air. A study of the air flow around a conventional air-cooled cylinder exposed to an air stream shows that the air follows the surface for about 100° from the front after which it breaks away from the cylinder, with the result that cooling at the rear is very poor. Considerable improvement in cooling can be obtained by the use of baffles around the cylinder to insure that the air follows the surface of the cylinder for a greater distance before the breakaway occurs.

In the first cowling tests conducted by the Committee the cooling was improved by using baffles (reference 1); recent tests have been conducted elsewhere on various types of baffles. (See reference 2.) The results of extensive flight tests by Higginbotham (reference 3) showed the shape and location of the baffles to be very important in obtaining the maximum cooling with the minimum drag. Baffles are generally used in conjunction with N. A. C. A. cowling; they improve the cooling by directing the air to the hot parts of the cylinder and they reduce the drag by limiting the quantity of air that flows through the cowling.

The purpose of the tests herein reported was to determine the shape and location in relation to the cylinder barrel of the baffles that give the largest improvement in cooling. Tests were made with plate baffles set at various angles with respect to the air stream and at various positions around the cylinder, with streamline baffles, with shell baffles having variously shaped entrance and exit passages, with baffles welded to the tips of the fins, and with combinations of shell baffles with strips inside the baffles. A few tests were also made of a cylinder having the fins bent at the front quarter to an angle of 50° with respect to the cylinder wall. Most of the tests were made at an air speed of 56 miles per hour; some data, however, were obtained at air speeds from 38 to 145 miles per hour.

The comparative cooling of electrically heated finned cylinders with and without baffles was obtained by testing each cylinder and baffle combination in a wind tunnel and measuring the temperature over the cooling surface for a given heat input and air speed. The comparison of the effectiveness of the various types of baffles was based mainly on the heat-transfer coefficient (the heat dissipated per unit of cooling surface per unit of time per unit of temperature difference between the cooling surface and the cooling air). The reduction in base temperature at the rear of the cylinder with the various types of baffles was also considered.

The tests were conducted at the Committee's laboratories at Langley Field, Va., between July 1932 and October 1933.

APPARATUS

Cooling tunnel.—The cooling tunnel used had a 30-inch throat and was designed to give air speeds up to 200 miles per hour. The air speed of the tunnel was measured with a pitot-static tube placed to one side and sufficiently far ahead of the test unit to assure the accuracy of the readings. A honeycomb grid was placed at the tunnel entrance to reduce air disturbances. Figure 1 shows the cooling tunnel with one of the test units mounted in place.

Test units.—The tests were conducted with three test units of the type shown in figure 2. The heating unit for the cylinders consisted essentially of a coil of nichrome wire embedded in alundum cement. The coils of the heating element were so distributed that the heating was uniform over the entire cylinder wall. In order to eliminate heat losses from the ends of the test cylinder, guard rings of the same construction as the test cylinder were placed on each end. The construction of such a unit is completely described in references 4 and 5.

A voltmeter and an ammeter were used for measuring the heat input to the test cylinder. Oil-cooled rheostats having a very sensitive adjustment over a wide range of electrical input were used to regulate the input to each of the guard rings and to the test cylinder.

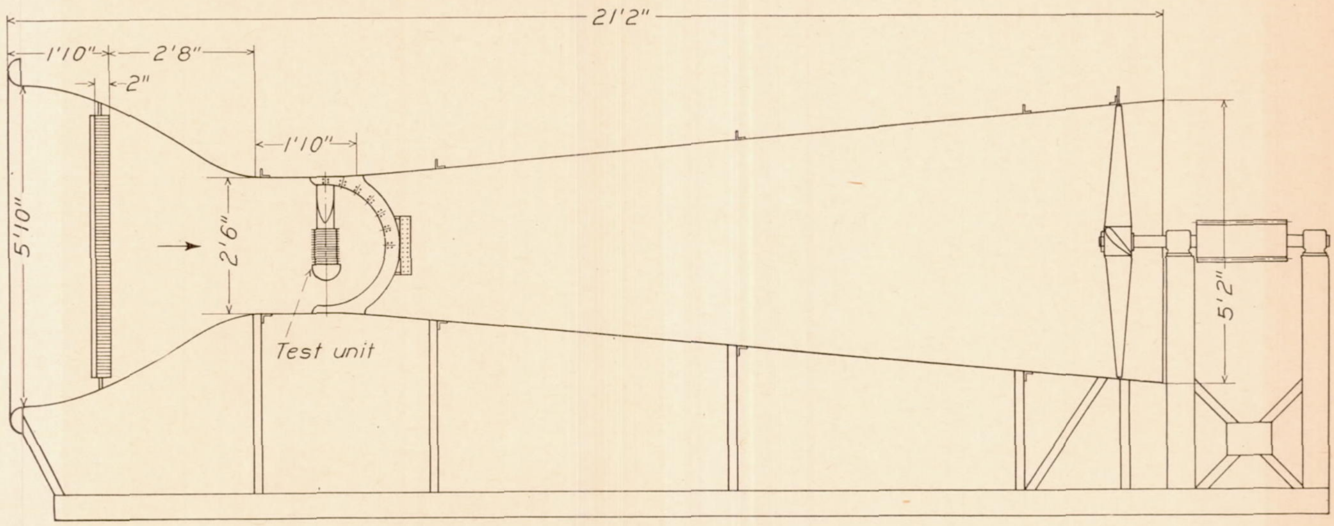


FIGURE 1.—Cooling tunnel with one of the test units mounted in place.

Thermocouples.—The temperatures of the surfaces of the finned cylinders were measured with iron-constantan thermocouples connected through a selector switch to a medium-resistance pyrometer of the portable type. The thermocouples were made of 0.013-inch diameter silk-covered enameled wire.

The 24 thermocouples were electrically welded to the barrel and the fins at the positions shown in figure 3. The temperature distribution being symmetrical with respect to the air stream, the thermocouples were located on only one-half of the cylinder. Tests have shown (reference 5) that at any given position on the fins the temperature is practically the same for all the fins and that the results for one fin can be taken as representative of all.

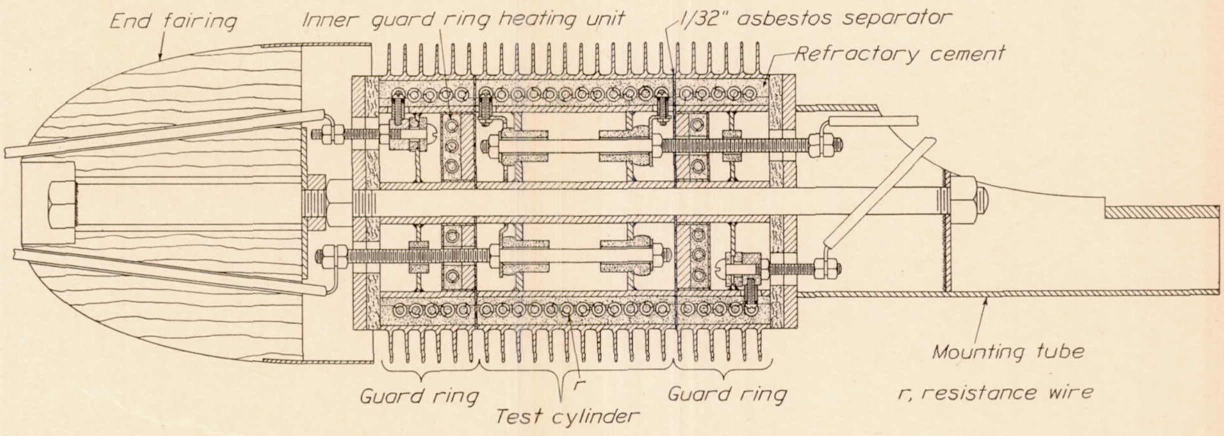


FIGURE 2.—Details of construction of the test unit.

Apparatus used for measuring air-flow speed.—The air speed between the fins was determined from measurements made with impact and static tubes connected to water manometers. The tubes had an outside diameter of 0.040 inch and an inside diameter of 0.035 inch.

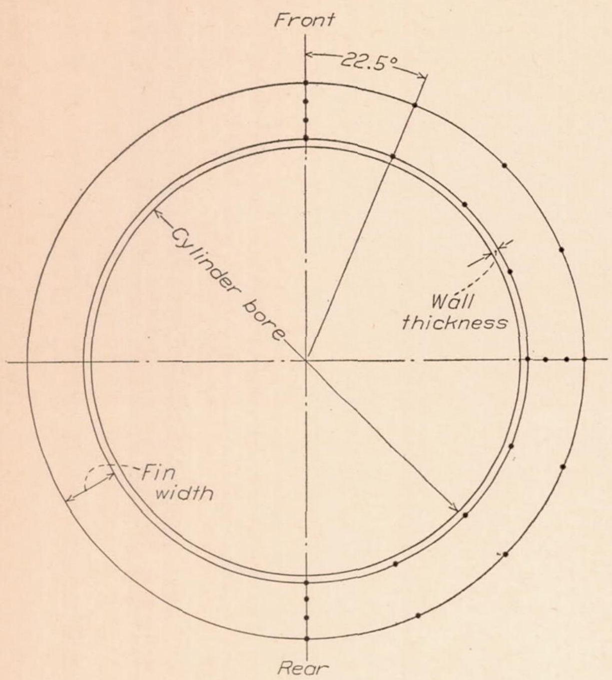


FIGURE 3.—Location of thermocouples on test cylinder.

Cylinders and baffles.—Figure 4 shows cross sections of the fins used in this investigation. The 0.25-inch-pitch fins on cylinders 1 and 2 were of 1.22-inch and 0.67-inch widths, respectively. Cylinder 3, which was cut from a Wright J–5 cylinder barrel, had fins of 0.31-inch pitch and 0.6-inch width. The cylinders were constructed from steel corresponding to specifications for S. A. E. 1050 steel. Figure 5 shows one of the cylinders equipped with shell baffles and attached to the mounting bracket ready for testing.

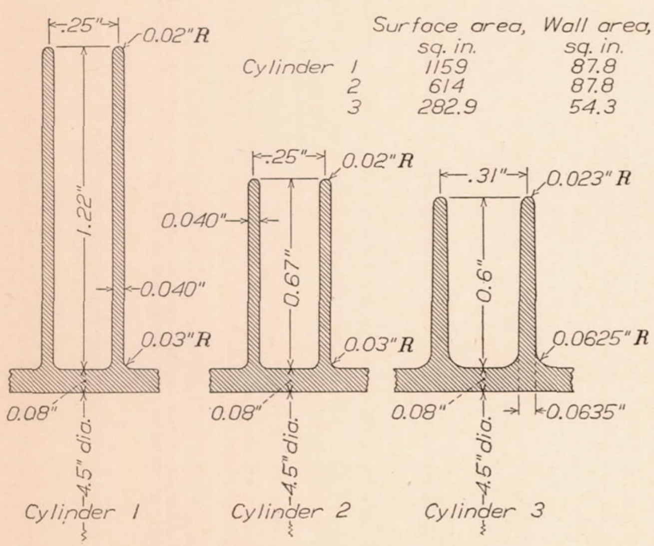


FIGURE 4.—Shape of fins and outside-wall and total-surface areas on test cylinders.

The plate baffle (fig. 6) consisted of two plates mounted symmetrically on each side of the test cylinder with respect to the plane through the axis of the cylinder and parallel to the air stream. Two sizes of these plates were tested, the narrow one being 1¼ inches wide and the wide one 4½ inches. The plate baffles and all other baffles tested were constructed from 20-gage sheet steel.

The streamline baffles (fig. 7) consisted of two pieces of streamline tubing symmetrically mounted parallel to the axis of the cylinder and 135° from the front.

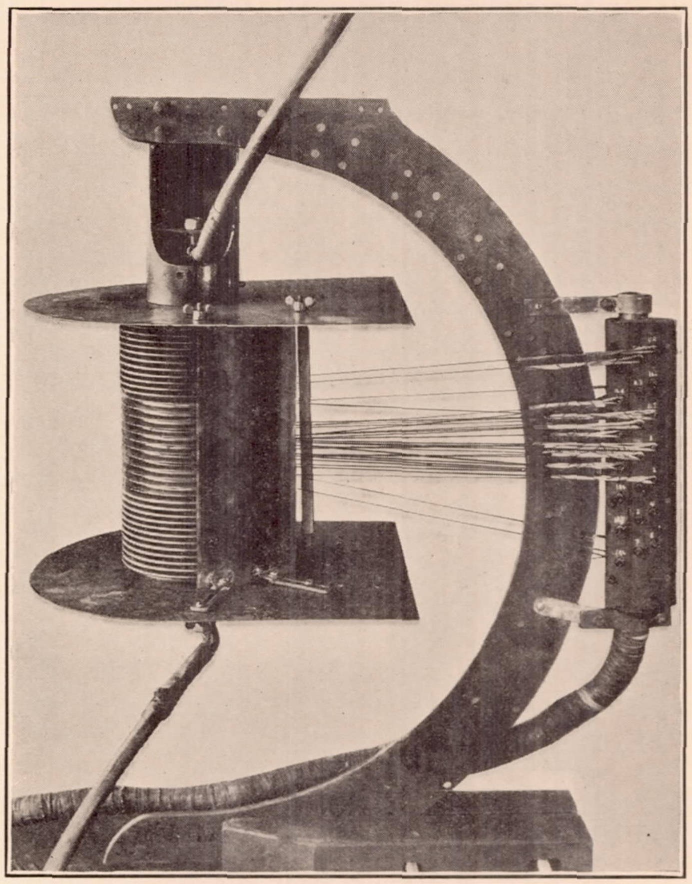


FIGURE 5.—Assembly of finned test cylinder and guard rings with mounting bracket, baffles, and thermocouples in place.

The shell baffles (see figs. 8, 9, 11, 13, and 15) consisted of two pieces of sheet steel of single curvature mounted concentrically with the surface of the test cylinder and symmetrically with respect to the cylinder diameter parallel to the air stream. Some of these baffles were provided with extensions forming entrance and exit passages.

As previous tests (reference 4) showed that the cooling was greatly improved by directing the cooling air at an angle with respect to the fins, it was believed that guiding the air to the rear of the cylinder and, at the same time, directing it at an angle with respect to the fins, would reduce the rear-wall temperatures.

Narrow strips were accordingly placed at an angle between the baffles and the fins so as to direct the air spirally toward the rear of the cylinder as shown in figure 19. A section of the fins on cylinder 3 was bent at an angle of 50° to the cylinder wall in order to investigate further the cooling with the air directed at an angle with respect to the fins (fig. 20).

The integral baffle (fig. 21) was a special shell baffle constructed by welding strips of metal between and at the fin tips to form channels for the cooling air being tried at various points around the circumference. Only wall-temperature measurements were obtained when testing this arrangement. (See fig. 20.)

TESTS

Tests on cylinders 1 and 2 were made with a heat input of 85 B. t. u. per hour per square inch of cooling surface and those of cylinder 3 with a heat input of 100. The air speed for all the tests was approximately 56 miles per hour except in the tests made to determine the effect of speed.

When conducting this investigation preliminary tests were made with the plate, streamline, and shell baffles; as a result of these preliminary tests the shell baffles were selected as the most promising for further investigation. The wide and narrow plate baffles were tested on cylinder 1. The narrow baffles were tried at the front, side, and rear of the cylinder and for each position the plates were tested at three different angles with respect to the air stream. The wide baffles were mounted at the rear of the cylinder and tested at three angles to the air stream. The arrangement of the plates and the results of the tests are shown together in figures 6 (a) to 6 (f).

The streamline baffles were tested with cylinder 1 in the single condition shown with the results in figure 7.

Although most of the tests with the shell baffles were made on cylinder 3 (figs. 9 to 18), which had fins of 0.6-inch width, a sufficient number of tests were made on cylinder 1 (fig. 8), which had fins of 1.22-inch width, so that the effect of fin width on the heat-transfer coefficient and the temperature distribution could be established. The radial distances from the shell baffles to the fins were varied from 0.07 to 2 inches. The lengths of the exit extensions tested on the shell baffles of cylinder 3 were varied from 0.375 to 5 inches; the size of the intake opening, equal to the arc subtended by the angle, was varied from a minimum of 40° to a maximum of 210°. Determinations were made of the effect of varying the ratio of the area of the flow passage between the deflector and the cylinder to the discharge area from 1 to slightly more than 3.

The results of the tests made on cylinder 2 with shell baffles and with combinations of shell baffles and strips to direct the air at an angle with respect to the fins are shown in figure 19. Tests were also made with and without shell baffles on cylinder 3 having a section of the fins bent so that the cooling air would impinge upon the fins at an angle instead of parallel. The section varied in size from 90° to 360°, the small section being tried at various points around the circumference. Only wall-temperature measurements were obtained when testing this arrangement. (See fig. 20.)

A few tests were made on cylinder 3 to determine the effect on base temperature of using baffles consisting of metal strips welded to the tips of the fins. A sketch of this baffle and the results of the tests are shown in figure 21.

Tests were also conducted at several air speeds between 38 and 145 miles per hour to determine whether the improvements obtained at one air speed with the bent shell baffles could be obtained at other air speeds. A few tests were also made, for the same range of air speeds, in which the air speed between the fins at several points approximately 90° from the front of the cylinder was determined with and without baffles.

COMPUTATIONS

The calculations for this report are substantially the same as those of reference 5 from which more detailed description and the derivation of the formula may be obtained.

The air speeds were corrected to a standard density corresponding to a pressure of 29.92 inches of mercury and a temperature of 80° F. according to the relation: corrected air speed=

$$\frac{\text{observed air speed} \times \text{density in tunnel test section}}{\text{standard density}}$$

The cold junction of the thermocouple that measured the temperature difference between the cylinder and the air was located outside the tunnel. A correction was applied to take care of the adiabatic cooling of the air due to the drop in pressure before reaching the test section and the frictional heating of the air by the grid and tunnel wall ahead of the test section.

The average cylinder-wall temperature was found by arithmetically averaging the readings of the nine thermocouples located on the cylinder wall.

The temperatures at any point on the surface of the cylinder wall and fins were determined by cross-plotting the measured temperatures. The average cooling-surface temperature was obtained by graphically integrating the temperature with respect to the area over the entire cooling surface and dividing by the area covered.

The average surface heat-transfer coefficient was found by dividing the total heat dissipated per hour by the product of the total exposed area of the cylinder wall and fins and the average surface-temperature difference. The experimental cylinder-wall heat-transfer coefficient was obtained by dividing the total heat dissipated per hour by the product of the outside cylinder-wall area and the average cylinder-wall temperature difference.

For convenience in this report whenever "temperature" is used it will be understood to mean the corrected "temperature difference" between a point on the cylinder and the cooling air.

The theoretical cylinder-wall heat-transfer coefficient was calculated from the following formula (reference 5):

$$U=\frac{q}{p}\left[\frac{2}{a}\left(1+\frac{w}{2R_b}\right)\tanh aw'+s_b\right]$$

where $a=\sqrt{\frac{2q}{kt}}$

U, over-all heat-transfer coefficient, B. t. u. per square inch base area per hour, per °F. temperature difference between the cylinder wall and cooling air.

q, surface heat-transfer coefficient, B. t. u. per square inch total surface area per hour, per °F. temperature difference between the surface and the cooling air.

k, thermal conductivity of metal, B. t. u. per square inch, per °F. through 1 inch per hour (2.17 for steel).

t, average fin thickness, inches.

p, pitch of fins, inches.

w, fin width, inches.

$w'=w+\frac{t_t}{2}$, effective fin width.

t_t, fin-tip thickness, inches.

R_b, radius from center of cylinder to fin root, inches.

s_b, distance between adjacent fin surfaces at the fin root, inches.

The following additional symbols are used in this report:

θ_b, average temperature difference between the root of the fin and the cooling air, °F.

θ_a, average temperature difference between the cooling surface and the cooling air, °F.

RESULTS AND DISCUSSION

PLATE BAFFLES

The effect on the temperatures of the cylinder base of using plate baffles is shown by the temperature-distribution curves of figure 6. The single curve of figure 6 (a) is for cylinder 1 without baffles. The temperature-distribution curve and the heat-transfer coefficients obtained for cylinders 1 and 3 (fig. 9 (a)) without baffles are used throughout this report as standards for comparison. The condition of shell baffles alone on cylinder 2 is used as a standard to compare with the cylinder when strips are attached to the baffles. (See fig. 19.)

When plate baffles 1.25 inches wide are placed 45° from the front, the second group of curves (fig. 6 (b)) and the heat-transfer coefficient show that the cooling is impaired for all conditions except when the baffles are perpendicular to the general direction of the air stream, in which case the heat-transfer coefficient was increased only 4.4 percent. The use of the baffle plates on the sides of the cylinder at various angles with respect to the air stream caused a large variation in the base temperatures as shown by the third group of curves (fig. 6 (c)). The heat-transfer coefficient varied from a reduction of 10.9 percent with the plates parallel to the air stream to an increase of 22.6 percent with the plates perpendicular to the air stream. The reduction in rear-base temperature with the plates perpendicular to the air stream was small. When the plates are mounted 60° from the rear of the cylinder (fig. 6 (d)) and perpendicular to the direction of the air stream, the heat-transfer coefficient is increased 27.2 percent. Practically no reduction in rear-base temperature was obtained for the latter condition but the temperatures at points between the sides and the rear show a large reduction.

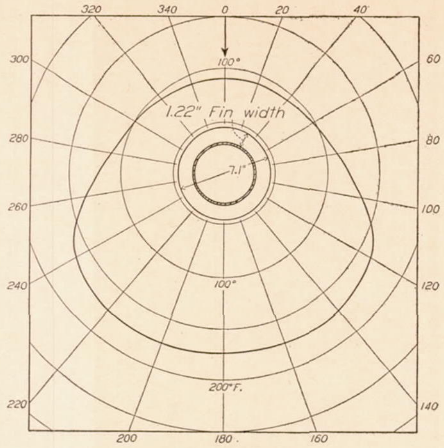
Figure 6 (e) shows a comparison of the base temperatures obtained for the best condition at each of the three positions tried. The solid curve represents the results obtained with two plates mounted on each side as shown. With the latter plate arrangement the heat-transfer coefficient is increased 21.5 percent. The rear-wall temperature is increased slightly but the wall temperatures at points between the sides and the rear are greatly reduced. The wall temperatures for the condition with the baffles mounted 60° from the rear are higher than with one or two baffles on the side even though the heat transfer is higher. This apparent discrepancy is explained by the fact that the heat-transfer coefficient is based on the average fin and barrel temperature.

In general, it may be said that the narrow plate baffles give a large improvement in cooling when mounted perpendicular to the air stream and between the side and rear of the cylinder but when mounted parallel to the air stream they may seriously impair the cooling. At best only a slight reduction in rear-wall temperature can be obtained with the narrow baffles. As this type of baffle is very sensitive to the direction of air flow, great care must be exercised in its use or the cooling may actually be impaired.

The curves in figure 6 (f) show that the plate baffles 4½ inches wide mounted 60° from the rear of the cylinder do not improve the cooling as much as do the narrow plates (fig. 6 (e)). For the best condition with the wide plates the heat-transfer coefficient was increased 8.7 percent as compared with an increase of 27.2 percent with narrow plates mounted near the rear of the cylinder. When the wide plates were mounted parallel to the air stream the cooling was seriously impaired, as indicated by an increase of 40° F. in the rear-wall temperature and a decrease of 18.6 percent in the heat-transfer coefficient.

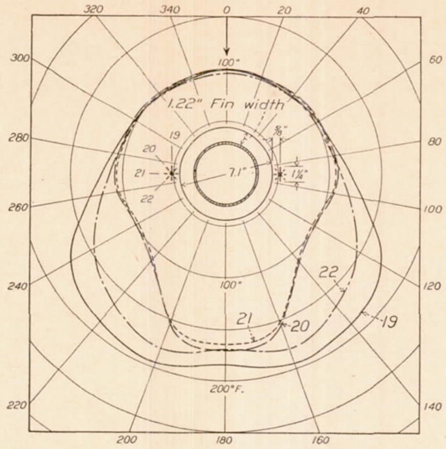
STREAMLINE BAFFLES

The curves and data of figure 7 show that streamline baffles on cylinder 1 increased the heat-transfer coefficient from 0.0866 to 0.1024, an increase of 18.3



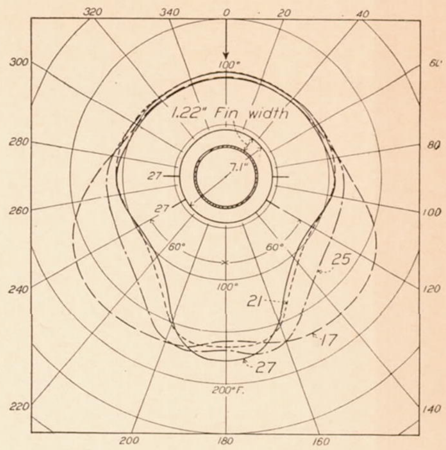
(a) Cylinder 1 without baffles:

q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
0.0866	0.625	131.3	72.3



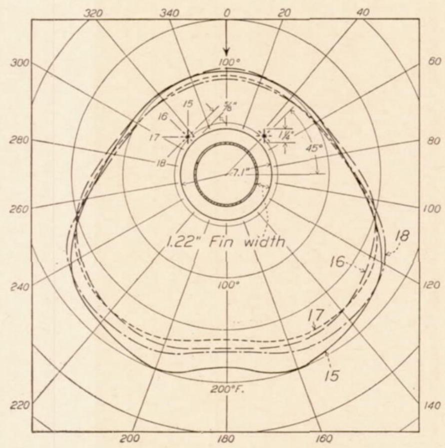
(c)

Position	q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
19	0.0772	0.604	138.1	81.9
20	.1048	.740	112.8	60.3
21	.1062	.746	111.8	59.5
22	.0873	.654	127.6	72.4



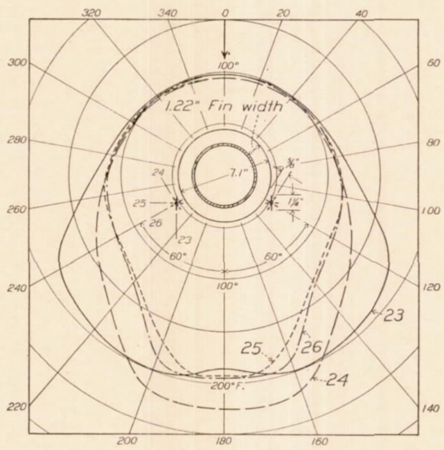
(e)

Position	q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
17	0.0904	0.636	131.3	70.0
21	.1062	.746	111.8	59.5
25	.1101	.684	122.0	57.4
27	.1052	.767	108.8	60.1



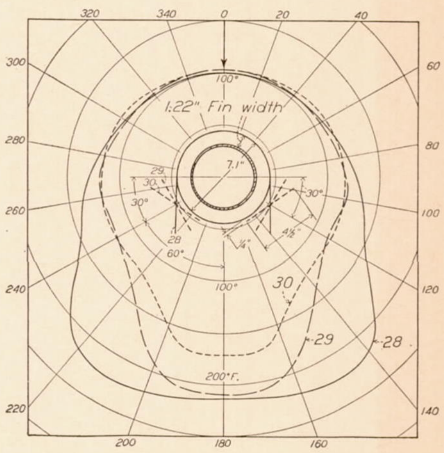
(b)

Position	q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
15	0.0754	0.594	140.6	83.9
16	.0834	.620	134.7	75.8
17	.0904	.636	131.3	70.0
18	.0848	.619	134.9	74.5



(d)

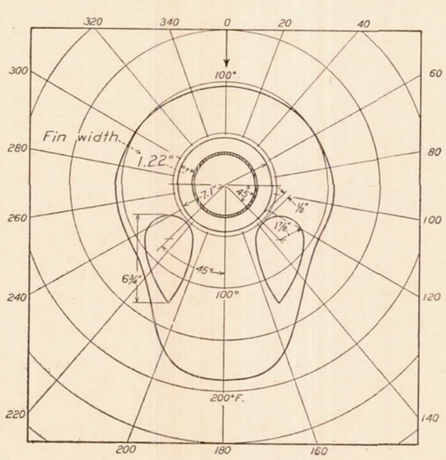
Position	q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
23	0.0768	0.586	142.4	82.3
24	.0822	.619	134.8	76.9
25	.1101	.684	122.0	57.4
26	.0948	.675	123.6	66.7



(f)

Position	q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
28	0.0705	0.652	150.8	89.7
29	.0824	.689	127.9	76.7
30	.0937	.724	121.1	67.5

FIGURE 6.—Cylinder-base temperature differences and heat-transfer coefficients obtained with plate baffles in different positions. Cylinder 1.



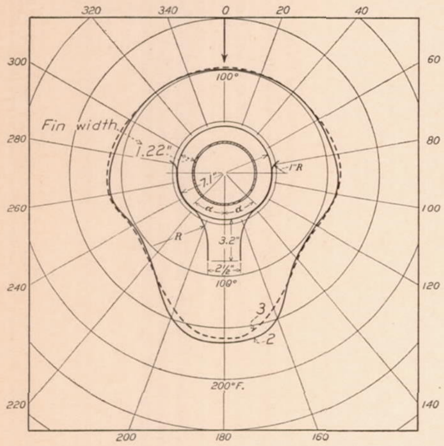
q	U	$\frac{\theta_b}{°F.}$	$\frac{\theta_a}{°F.}$
0.1024	0.724	115.4	61.7

FIGURE 7.—Cylinder-base temperature differences and heat-transfer coefficients obtained with streamline baffles. Cylinder 1.

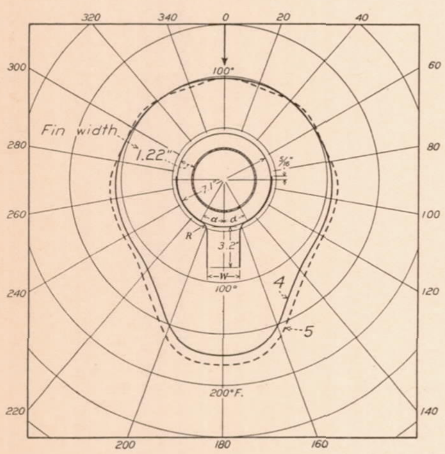
percent, while the rear-wall temperature was increased 15° F., or 8.6 percent. Apparently with these baffles there is, in the rear of the cylinder, a low-pressure area that causes the air to flow to the rear through the openings between the baffles and the cylinder.

SHELL BAFFLES

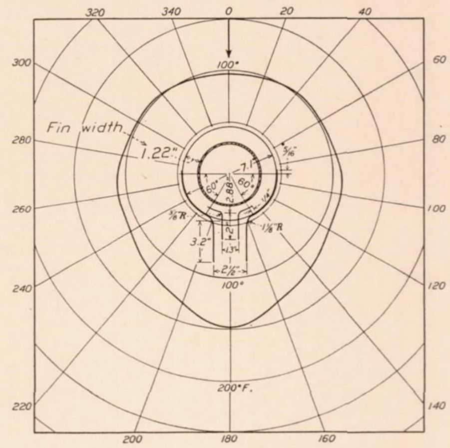
The curves in figure 8 (a) show that although, within limits, increasing the extent of surface contact of the shell baffles with the cylinder so as to restrict the exit opening decreases the rear-wall temperatures, it also, for the two conditions shown, reduces the heat-transfer coefficient from 0.1087 to 0.0972. A further restriction of the air flow effected by reducing the size of the exit opening (fig. 8 (b)) causes an increase in rear-wall temperatures and an additional reduction in the heat-transfer coefficient. The results, however, show that little difficulty will be experienced in effecting a large reduction in temperature at points between 20° and 70° from the rear of the cylinder although it will be considerably more difficult to reduce the temperature directly in the rear.



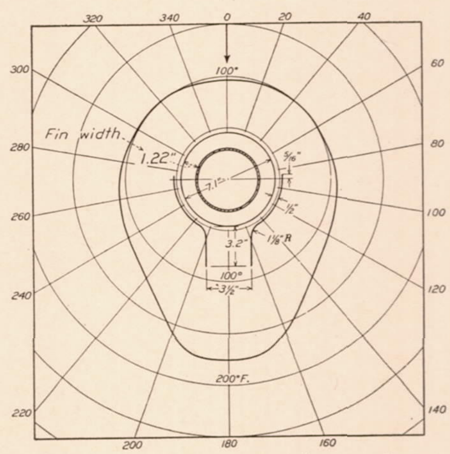
(a) Test	R in.	a deg.	q	U	θ_b °F.	θ_a °F.
2	4¼	45	0.1087	0.737	113.2	58.2
3	1⅛	30	.0972	.736	113.3	65.0



(b) Test	R in.	a deg.	W in.	q	U	θ_b °F.	θ_a °F.
4	1⅛	30	2½	0.1093	0.741	112.6	57.8
5	¾	23	1$\frac{11}{16}$	.1055	.717	116.4	59.9

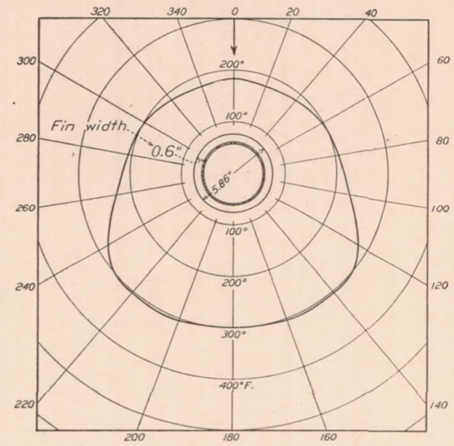


(c)	q	U	θ_b °F.	θ_a °F.
	0.1038	0.855	97.6	55.5

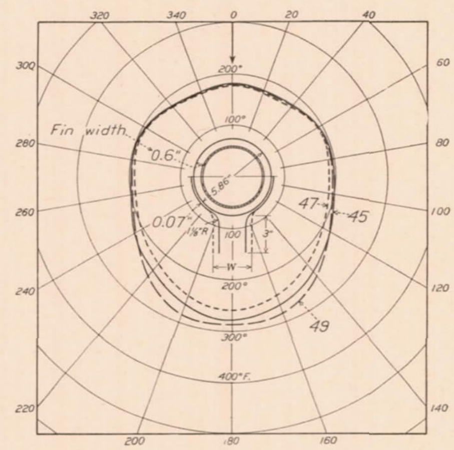


(d)	q	U	θ_b °F.	θ_a °F.
	0.1058	0.728	114.7	59.7

FIGURE 8.—Cylinder-base temperature differences and heat-transfer coefficients obtained with shell baffles. Cylinder 1.



(a)	q	U	θ_b °F.	θ_a °F.
	0.0965	0.424	236.1	198.8



(b) Test	W in.	q	U	θ_b °F.	θ_a °F.
45	1	0.1131	0.474	211.4	169.9
47	2	.1188	.498	200.9	161.7
49	3½	.1122	.474	213.1	172.6

FIGURE 9.—Effect of the width of exit opening of shell baffles on the temperature distribution and heat-transfer coefficients. Cylinder 3.

A comparison of the results for test 4 in figure 8 (b) with those for test 3 in figure 8 (a) shows that without flares on the intake opening the heat-transfer coefficient is 12.5 percent higher and the rear-wall temperatures 10° higher than with flares.

The curve in figure 8 (c) shows that an appreciable reduction in rear-wall temperatures may be obtained by using small baffles between the fins in conjunction with the shell baffles. As compared with conditions without baffles (fig. 6 (a)) this arrangement reduced the rear-wall temperatures 15.4 percent; whereas the shell baffles without the small baffles (fig. 8 (b)) gave a reduction of only 2.9 percent. For the foregoing conditions the heat-transfer coefficient increased 19.9 percent with the small baffles and 26.2 percent without the small baffles. Apparently these small baffles are very effective in guiding the air toward the rear of cylinders having long fins, but they impair the cooling at other points on the cylinder because they slow up the air and cause a reduction in the heat-transfer coefficient.

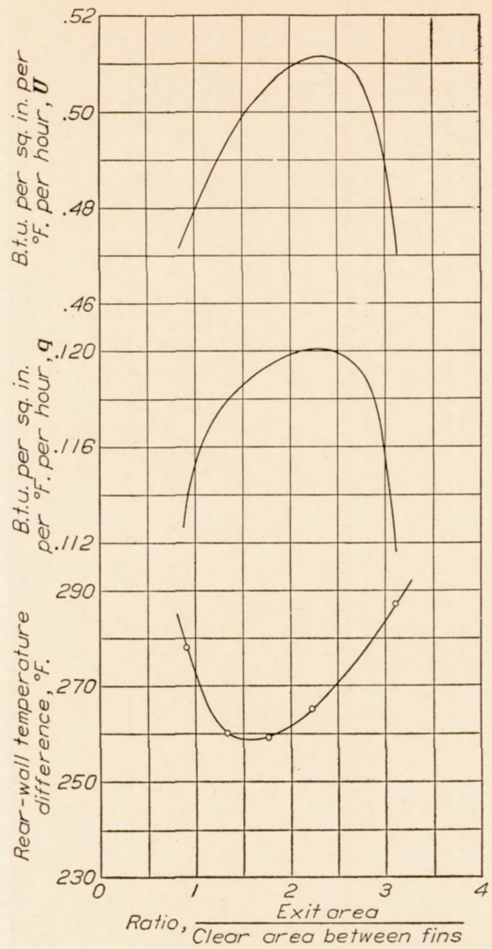


FIGURE 10.—Variation of rear-wall temperatures and heat-transfer coefficients with ratio of exit area to clear area between fins for a cylinder with shell baffles. Cylinder 3.

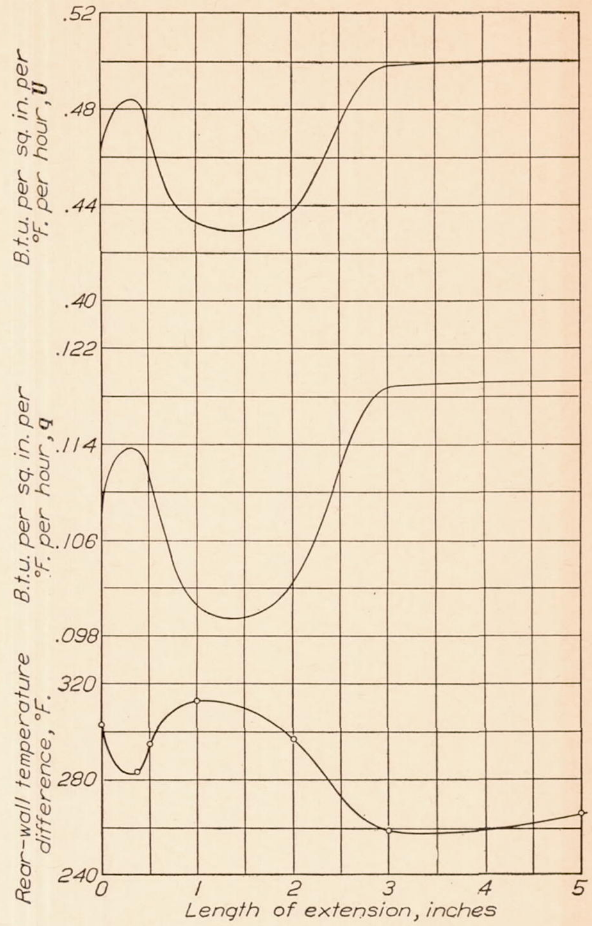
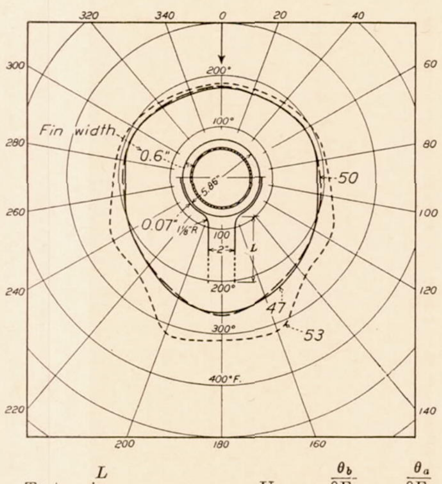
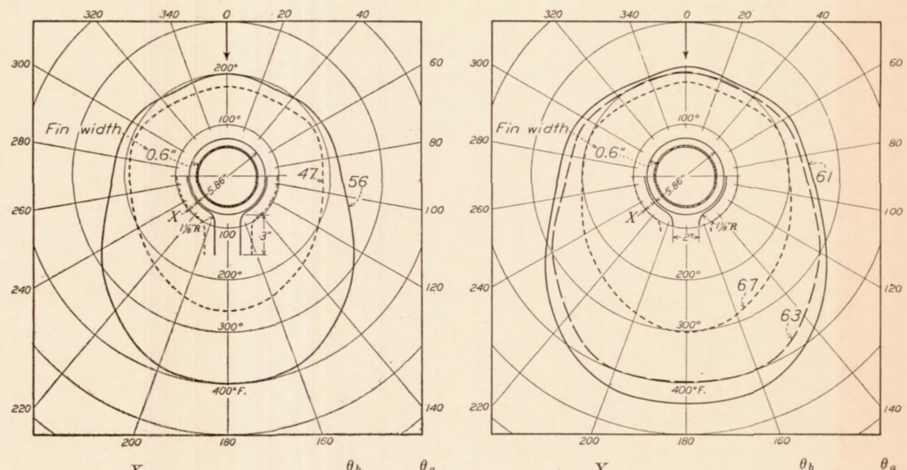


FIGURE 12.—Variation of rear-wall temperatures and heat-transfer coefficients with length of exit extensions of shell baffles. Cylinder 3.



Test	L in.	q	U	θ_b °F.	θ_a °F.
47	3	0.1188	0.498	200.9	161.7
50	5	.1192	.500	201.9	162.6
53	1	.1003	.433	231.8	191.9

FIGURE 11.—Effect of the length of the exit extensions of shell baffles on the temperature distribution and heat-transfer coefficients. Cylinder 3.



(a)

Test	X in.	q	U	θ_b °F.	θ_a °F.
47	0.07	0.1188	0.498	200.9	161.7
56	2.00	.0860	.383	263.3	225.1

(b)

Test	X in.	q	U	θ_b °F.	θ_a °F.
61	2.00	0.0781	0.349	289.3	248.0
63	.75	.0850	.376	271.3	227.8
67	.07	.1078	.463	218.3	179.5

FIGURE 13.—Effect of clearance between the shell baffles and the fin tips on the temperature distribution and heat-transfer coefficients. Cylinder 3.

The results of the tests on shell baffles have shown that with proper installation of baffles an appreciable reduction in rear-wall temperatures and a large in-

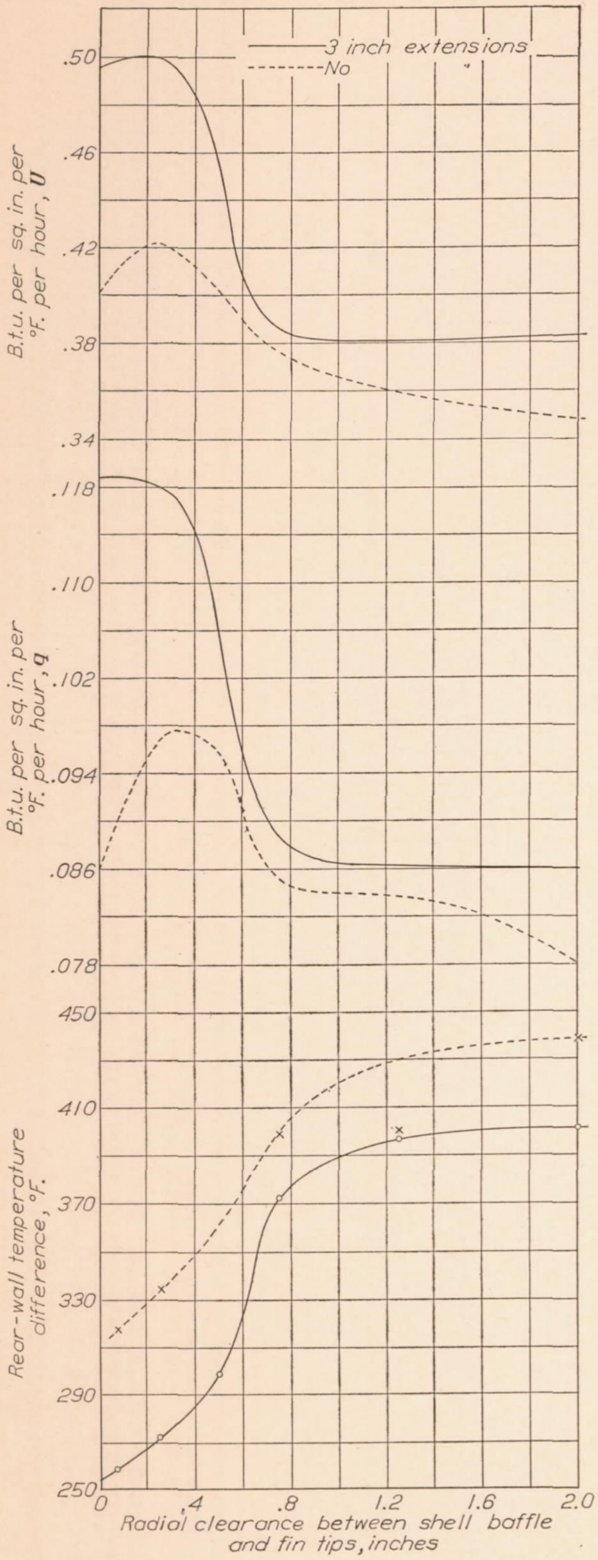
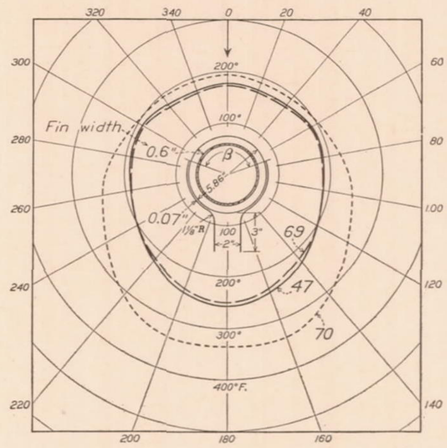


FIGURE 14.—Variation of rear-wall temperatures and heat-transfer coefficients with clearance between shell baffles and fin tips. Cylinder 3.

crease in the heat-transfer coefficient may be obtained. These results were sufficiently encouraging to warrant further work to determine the effect of size of exit opening, length of extension, frontal area exposed to air stream, and the distance between baffle and cylinder.

Effect of exit opening.—The curves in figures 9 and 10 show the importance of the ratio of exit area of baffles to the area between the cooling surface and the baffles in reducing rear-wall temperatures and increasing the heat-transfer coefficient. These tests were conducted on cylinder 3 (fig. 9 (a)). With baffles having a ratio of exit area to clear area between fins of 1.75 (figs. 9 (b) and 10) the rear-wall temperature was reduced to less than 260° F., or 13.3 percent, and the heat-transfer coefficient was increased 23.1 percent. In figure 10 the curve for the heat-transfer coefficient shows an optimum ratio of areas of 2.3, which is higher than the value of 1.6 based on the curve for rear-wall temperatures. The optimum ratio of areas is a ratio that will give a large reduction in rear-wall temperatures by bringing the air as far as possible to the rear without appreciably decreasing its velocity. These



Test	β deg.	q	U	θ_b °F.	θ_a °F.
47	180	0.1188	0.498	200.9	161.7
69	140	.1238	.515	196.0	156.4
70	220	.0968	.407	248.0	200.1

FIGURE 15.—Effect of the entrance angle of a shell baffle on the temperature distribution and heat-transfer coefficients. Cylinder 3.

ratios would be meaningless if the area of the flow passage between the cylinder and the baffle and the exit opening were greatly increased, unless there was a corresponding increase in the cylinder diameter.

Effect of extension length.—The curves in figures 11 and 12 show that baffles having extensions 3 inches or more in length give the largest reduction in rear-wall temperatures and the highest heat-transfer coefficients. That the optimum extension length is short is fortunate, because long extensions would increase installation difficulties; whereas 3-inch extensions can probably be conveniently used on all installations. The irregularities in the curves of rear-wall temperatures and heat-transfer coefficient when extensions shorter than 3 inches are used indicate that the length of the extension has considerable effect on the air flow in the rear of the cylinder and that the cooling can be regulated by slight changes in the extension length. Increasing the length of the extension to more than 3 inches results in a gradual increase in rear-wall temperatures and a small increase in the heat-transfer coefficient.

Effect of radial clearance between baffles and fin tips.—The test results shown in figures 8 (b), 8 (d), 13, and 14 indicate that the distance between the baffle and the cylinder influences the rear-wall temperatures and the heat-transfer coefficient q and that the amount of the change depends upon the fin width. Placing the baffles in contact with the fins on cylinders having fins of 1.22- and 0.6-inch widths resulted in increases in the heat-transfer coefficient of 26.2 and 23.1 percent, respectively, as compared with conditions without baffles. Increasing the distance between the baffles and the cylinders to one-half inch resulted in the reduction of these percentages to 22.2 and 9.84 for cylinders having fins of 1.22- and 0.6-inch widths, respectively. The rear-wall temperatures of the cylinder with the wide fins are not appreciably affected by the use of baffles regardless of the distance between the baffles and the cylinder; whereas, for the cylinder having fins 0.6 inch in width the rear-wall temperatures are reduced from 298° F. without baffles to 259° F. with baffles in contact with the fins.

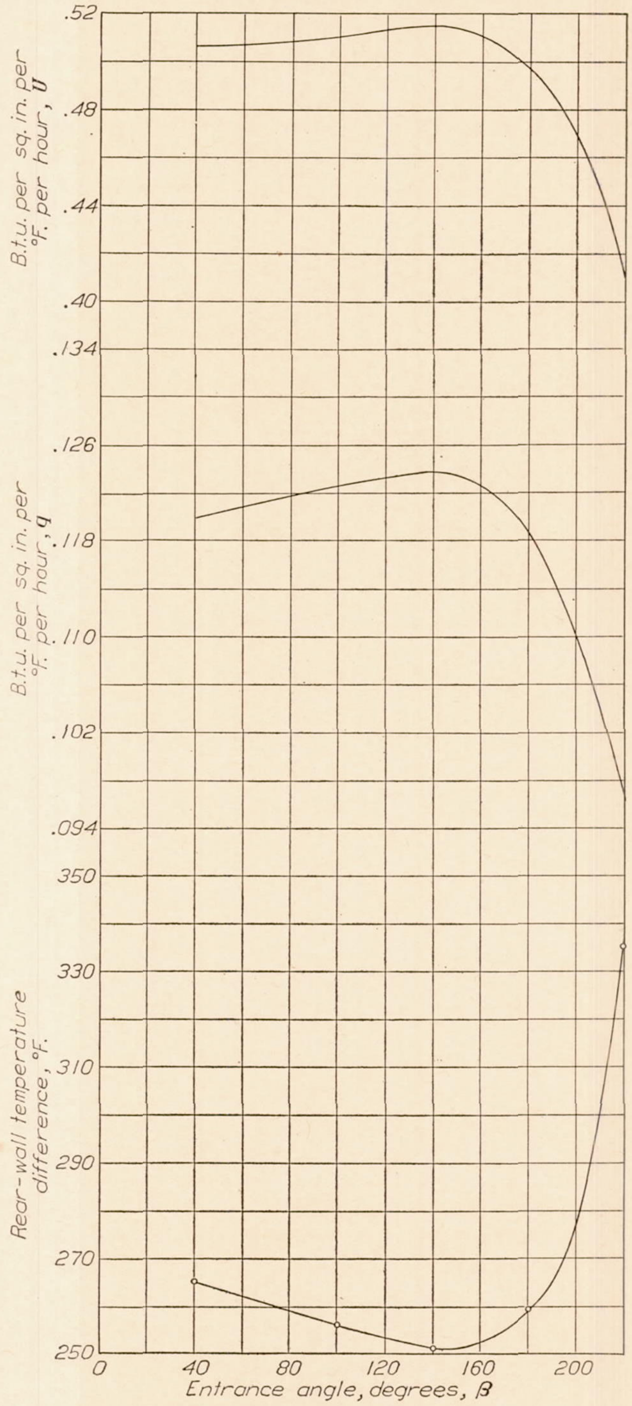


FIGURE 16.—Variation of rear-wall temperatures and heat-transfer coefficients with entrance angle of shell baffles. Cylinder 3.

A shell baffle with 3-inch extensions may seriously impair the cooling if it is mounted too far from the cylinder (fig. 13 (a)). For example, when the baffles are mounted 2 inches from the cylinder the rear-wall temperature is 104° F. higher than for conditions with no baffles and the heat-transfer coefficient is reduced from 0.0965 for a cylinder without baffles to 0.0860 for a cylinder with baffles having extensions. The curves (fig. 14) also show that considerably more is to be gained in reducing rear-wall temperatures and increasing the heat-transfer coefficient by placing baffles with extensions in contact with the fins than can be gained by placing plain baffles in contact with the fins.

Effect of size of intake opening.—A comparison of the curves in figures 9 (a), 15, and 16 shows that the entrance angle has a large effect on the value of the heat-transfer coefficient. For example, with an entrance angle of 145° (fig. 16), the heat-transfer coefficient was 0.1240 as compared with 0.0965 without baffles (fig. 9 (a)), an increase of 28.5 percent. For the same entrance angle the rear-wall temperature was 250° F. as compared with 298° F. without baffles. Increasing the entrance angle to more than 160° results in a sharp rise in temperature because the air breaks away from the cylinder before it reaches the baffle; whereas reducing the angle to less than 120° results in only a gradual increase in temperature, the speed being reduced because of increased length of passage and because the intake opening normal to the air flow is smaller.

The curves (fig. 16) for heat-transfer coefficient show the same trend as the curves for base temperature. When the entrance angle is increased to 210° the temperatures in the rear of the cylinder are the same as without a baffle, while the heat-transfer coefficient shows an increase of 9.1 percent, indicating the beneficial effect of baffles on the temperatures at other points.

Although these tests indicate (fig. 16) that the heat-transfer coefficient decreases when the entrance angle is less than 120°, it is believed that on installations having a large part of the area between the cylinders blocked or on engines having cowling and baffle arrangement such that only sufficient air is admitted to cool the engine, angles less than 120° will give better cooling. Improved cooling with small entrance angle is obtained for the above-mentioned condition because the air speed will be higher over a large part of the front area of a cylinder. For tests herein reported the air speed over the front of a cylinder with baffles was probably about the same as for a cylinder without baffles because the air could flow freely over the outside of the baffle as well as on the inside.

Effect of air-stream speed.—The test results submitted in figure 17 show that the heat-transfer coefficient varied as the speed to the 0.85 power for conditions without baffles and with baffles of the type shown in figure 15. The tests were made on cylinder 3 at several air velocities from 38 to 145 miles per hour.

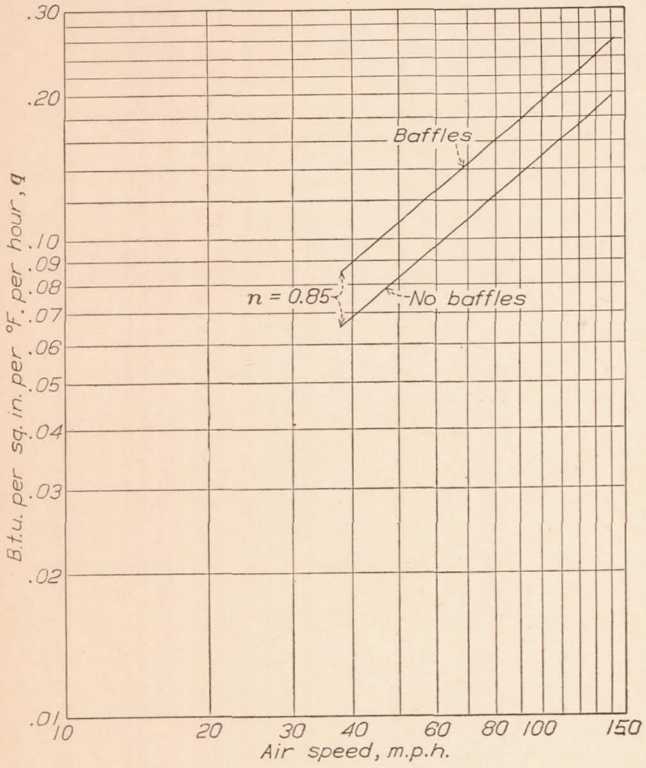


FIGURE 17.—Effect of air speed on the heat-transfer coefficient of cylinder 3 with and without shell baffles.

The use of the best shell baffles resulted in a large improvement in cooling at all air velocities investigated. The test results in figure 18 show the effect of velocity on the cylinder-wall heat-transfer coefficient U for cylinder 3 with and without baffles. The calculated values of U (equation (1)) are also shown. As these calculated values check the experimental values, the method proposed in reference 5 for the design of cylinders without baffles can be extended to the design of cylinders with baffles by using the experimental values of q as determined for cylinders with baffles.

The measurement of the air speeds between the fins at 90° from the front of cylinder 3 without baffles showed that the average air speed between the fins was 35 percent higher than the tunnel air speed. With the best shell baffles having a radial clearance between the baffle and the fin tip of 0.07 inch, the average air speed between the fins was 30, 92, 118, and 172 miles per hour for tunnel air speeds of 30, 80, 100, and 140 miles per hour, respectively. At a tunnel air speed of 100 miles per hour the highest air speed between the fins without the baffles was approximately 0.12 inch from the root of the fin and with shell baffles it was approximately 0.09 inch from the root of the fin. This difference would indicate that part of the improvement in cooling obtained through the use of baffles may be attributed to reduced boundary layer.

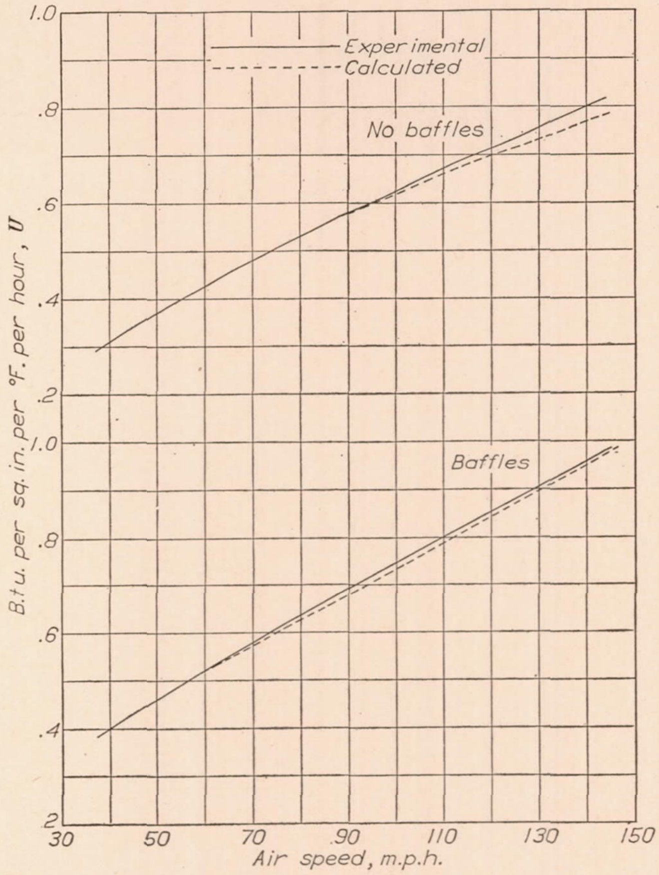


FIGURE 18.—Comparison of experimental with calculated over-all heat-transfer coefficient for cylinder 3 with and without shell baffles at various air speeds.

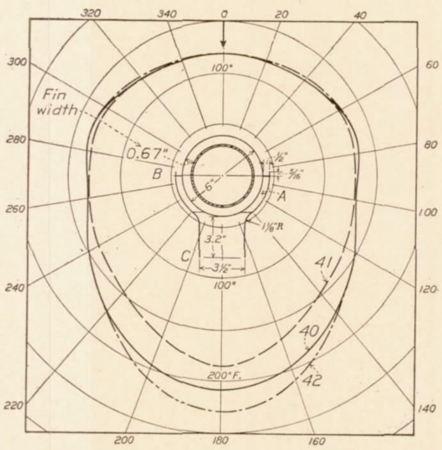
The curves in figures 9 (a) and 15 show that the temperatures are reduced at a position 90° from the front when using baffles as compared with conditions without baffles even though the average air speed between the fins is less, indicating that the boundary layer at this point is less on the cylinder with baffles.

COMBINATION OF SHELL BAFFLES AND STRIPS

The results in figure 19 for shell baffles show that the addition of the strips increased the wall temperatures around the rear of the cylinder, as compared with the cylinder with only the shell baffle, and that the heat-transfer coefficient was decreased. As the strips were placed on the fin tips, the air passing over the fins probably did not change its direction of flow and, furthermore, the speed of the air between the cylinder and the baffle may have been reduced by a restricting action of the strips.

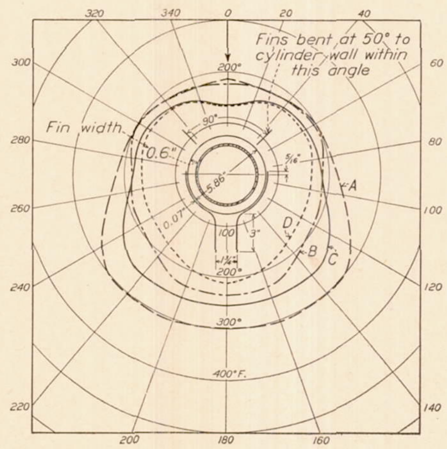
COMBINATION OF CYLINDER WITH BENT FINS AND SHELL BAFFLES

Next to the integral baffle, the combination of bent fins and shell baffles reduced the rear-wall temperatures more than any other baffle. Complete data were not obtained for these tests so that only the wall temperatures are given in figure 20. The rear-wall temperature was reduced from 298° F. for a cylinder with straight fins and no baffles to 210° F., or 29.6 percent, as compared with the 31.9 percent reduction obtained with integral baffles and 16.1 percent reduction obtained with baffle 69 (fig. 15). It should be noted that if the pitch of the fins were less than those used in the test cylinder it might be detrimental to the cooling to bend the fins as much as 50° at the front, because the smaller the pitch and the greater the angle the smaller will be the opening between fins for the air to enter and to flow across the fins.



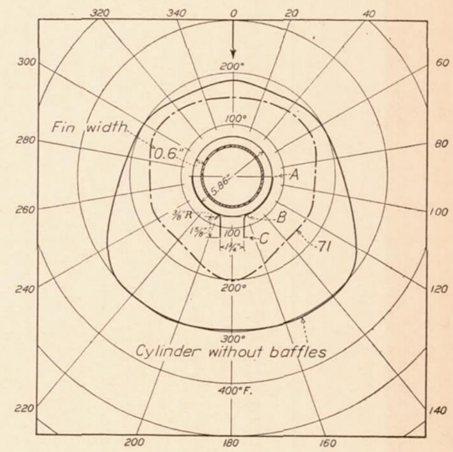
Test	Combination	q	U	θ_b °F.	θ_a °F.
40	$BC+A$ at 45°	0.1063	0.569	146.2	111.8
41	BC alone	.1139	.604	136.9	103.4
42	$BC+A$ at 30°	.1057	.554	149.4	112.1

FIGURE 19.—Effect of strips between shell baffles and fin tips on the temperature distribution and heat-transfer coefficient. Cylinder 2.



Test	Description	θ_b °F.
A.	Straight fins, no baffles	236.1
B.	Straight fins, with baffles	197.6
C.	Bent fins, no baffles	200.9
D.	Bent fins, with baffles	170.0

FIGURE 20.—Effect on the base temperatures of bending a section of the fins at a 50° angle to the cylinder wall. Cylinder 3 with and without baffles.



Test	q	U	θ_b °F.	θ_a °F.
No baffles	0.0965	0.424	236.1	198.8
71	.1488	.598	167.0	129.0

FIGURE 21.—Cylinder-wall temperatures and heat transfer coefficients obtained with integral baffles Cylinder 3.

INTEGRAL BAFFLES

The base temperatures were reduced at all points around the cylinder with integral baffles as compared with the cylinder without baffles, the reduction being the greatest that has been attained in any of the baffle tests. The rear-wall temperature was reduced from 298° F. to 203° F., or 31.9 percent, and the heat-transfer coefficient was increased from 0.0965 to 0.1488, or 54.2 percent. The largest reduction in rear-wall temperature obtained on any of the other baffles was approximately 15 percent. The great improvement with this integral baffle can no doubt be partly attributed to the increased cooling area. The difference in cooling obtained with this type of baffle with and without extensions was small.

CONCLUSIONS

The results of these tests show that:

1. Properly installed shell baffles reduced the cylinder-wall temperatures and increased the heat-transfer coefficient to a greater extent than either plate baffles or streamline baffles.

2. Optimum cooling was obtained with shell baffles when they were mounted as closely to the cylinder as possible, when the entrance was equal to the arc subtended by an angle of approximately 145°, when the rearward extensions were 3 or more inches long, and when the ratio of exit area to free-flow area between the fins was between 1.6 and 2.3.

3. The surface heat-transfer coefficient with and without baffles varied as the 0.85 power of the air speed for a range of speeds from 38 to 145 miles per hour.

4. The average air speed 90° from the front and between the fins of a cylinder with the best shell baffles was less than for a cylinder without baffles, but it was greater than the tunnel air speed when the tunnel air speed was over 30 miles per hour. The highest air speed between the fins at a tunnel air speed of 100 miles per hour was measured at a point approximately 0.09 and 0.12 inch from the root for the conditions with and without baffles, respectively.

5. The theoretical formula for calculating the heat dissipated from finned cylinders fitted with baffles checked closely the heat dissipation determined experimentally.

6. Baffles welded to the tips of the fins gave the largest reduction in rear-wall temperature and the greatest increase in the heat-transfer coefficient; the

reduction in rear-wall temperature was 31.9 percent and the increase in heat-transfer coefficient was 54.2 percent as compared with the same cylinder without baffles.

7. Bending the fins in the front quarter of the cylinder to an angle of 50° resulted in a large improvement in cooling for conditions with and without baffles.

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY FIELD, VA., *September 26, 1934.*

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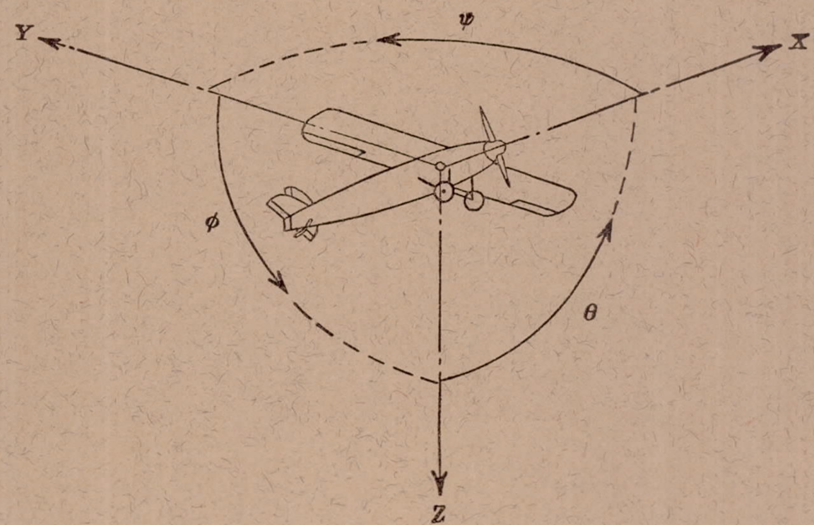

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	Rolling	L	$Y \longrightarrow Z$	Roll	ϕ	u	p
Lateral	Y	Y	Pitching	M	$Z \longrightarrow X$	Pitch	θ	v	q
Normal	Z	Z	Yawing	N	$X \longrightarrow Y$	Yaw	ψ	w	r

Absolute coefficients of moment

$C_l = \frac{L}{qbS}$ (rolling) $\quad C_m = \frac{M}{qcS}$ (pitching) $\quad C_n = \frac{N}{qbS}$ (yawing)

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter
p, Geometric pitch
p/D, Pitch ratio
V', Inflow velocity
V_s, Slipstream velocity
T, Thrust, absolute coefficient $C_T = \frac{T}{\rho n^2 D^4}$
Q, Torque, absolute coefficient $C_Q = \frac{Q}{\rho n^2 D^5}$
P, Power, absolute coefficient $C_P = \frac{P}{\rho n^3 D^5}$
C_s, Speed-power coefficient $= \sqrt[5]{\frac{\rho V^5}{P n^2}}$
η, Efficiency
n, Revolutions per second, r.p.s.
Φ, Effective helix angle $= \tan^{-1}\left(\frac{V}{2\pi r n}\right)$

5. NUMERICAL RELATIONS

1 hp. = 76.04 kg-m/s = 550 ft-lb./sec.
1 metric horsepower = 1.0132 hp.
1 m.p.h. = 0.4470 m.p.s.
1 m.p.s. = 2.2369 m.p.h
1 lb. = 0.4536 kg.
1 kg = 2.2046 lb.
1 mi. = 1,609.35 m = 5,280 ft.
1 m = 3.2808 ft.