REPORT No. 86

PROPERTIES OF SPECIAL TYPES OF RADIATORS

By S. R. PARSONS
Bureau of Standards

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This report describes an investigation of properties of special types of radiators, conducted for the National Advisory Committee for Aeronautics at the Bureau of Standards.

In respect to the relation of power absorbed to power dissipated, flat plate radiators have been found to be markedly superior to other types commonly employed, for use in unobstructed positions on airplanes flying at the higher speeds.

For such use, a pitch of one-half inch (1.27 cm.) between plates gives a higher figure of merit than the closer spacings.

The depth of such a radiator should be considerably greater than of the honeycomb types, and the most efficient depth is greater with the higher speeds. For a radiator of one-half-inch pitch to be used at a speed of 120 miles per hour (53.6 meters per second), the most efficient depth appears to be about 13 inches (33 cm.). A small increase above this depth has but a slight effect, however, on the figure of merit, and if it is desirable to reduce the frontal area of the radiator to a minimum, the depth may be increased to 20 inches (51 cm.) or more without serious reduction of efficiency.

Equations and plots are given, by means of which the properties of flat plate radiators may be obtained for various depths and spacings.

The plates should be continuous from front to rear of the radiator, in order to avoid excessive head resistance caused by eddies in the air stream.

Radiators of the "fin and tube" types are characterized by high head resistance, and low heat transfer at high speeds. With the possible exception of certain compact types for use in the wing they are unsuitable for airplane use.

The construction of some types of radiators is such that at certain air speeds they produce a whistling sound. These so-called "whistling radiators" are characterized by the following points:

(1) Unusual conditions of air flow, resulting in irregularities in the relations between different properties. For example, mass flow of air through the radiator is not proportional to free air speed as in ordinary radiators, and head resistance is not proportional to the square of free air speed.

(2) High heat transfer *for a given mass flow of air through the radiator.*

(3) Very low flow of air through the radiator *for a given free air speed.*

(4) In many cases, low heat transfer *for a given free air speed.*

(5) Very high head resistance and horsepower absorbed.

(6) Low figure of merit.

If water tubes made of smooth flat plates, continuous from front to rear of the radiator, are substituted for the rows of tubes of the whistling radiators described, the figure of merit will be greatly increased.

Different radiators show different effects on being inclined at an angle to the direction of the air stream, but in general the results (for angles up to 30°) are as follows:

(1) Decrease in air flow through the core;

(2) Increase in head resistance;

(3) In some cases slight increase in the heat transfer, for angles up to 20° or 25°.

PROPERTIES OF FLAT PLATE RADIATORS.

In the course of the investigation of airplane radiators, being carried on at the Bureau of Standards for the National Advisory Committee for Aeronautics, it has become evident that careful consideration should be given to a type of radiator consisting of water tubes which are thin flat hollow plates placed edgewise to the air stream. In regard to the relation of heat dissipation to absorption of power, radiators of this type have been shown to be markedly superior to other types now commonly employed for use in unobstructed positions on the faster planes. These conclusions are based on a comprehensive series of tests made to determine what properties such radiators may be expected to possess.

For the study of heat dissipation, three radiators were used, each $9\frac{3}{4}$ inches deep and with plates one-sixteenth inch thick, spaced one-fourth, three-eighths, and one-half inch respectively, center to center. They are the types E–6, E–7 and E–8 of Technical Report No. 63, Part I, and their properties and characteristics are given in that report. The curves are reproduced in plots 2, 3, and 4.

For the study of mass flow of air through the core and of head resistance, a set of 22 dummy radiators were constructed, using depths of 2, 4, 8, and 12 inches, spacings of one-fourth, three-eighths, one-half, three-fourths, and 1 inch, and thicknesses of one-sixteenth and one-eighth inch. The plates were made by covering cardboard of the proper thickness with thin sheet copper, and were used again and again. They were 12 inches long and with the exception of the one-fourth inch spacing, enough were used to make a radiator 12 inches wide, so that the frontal area was 1 square foot. Since the copper was thin it was not possible to obtain as smooth a surface as might have been desired, but the condition was fairly representative of what would be found in commercially manufactured radiators.

Extrapolations for depths of radiators not covered by the laboratory tests were made with the use of three empirical equations, given below. All quantities are based on one square foot frontal area of the radiator and an air density of 0.0750 pound per cubic foot.

Mass flow of air is given by the equation [1]

$$M=0.110\sqrt{\frac{p-t}{p}}(V)\left(1-e^{-10.95\sqrt{\frac{p-t}{x}}}\right) \tag{1}$$

where M = mass flow of air in pounds per second per square foot frontal area of radiator;
V = free air speed, in miles per hour;
t = thickness of plates, in inches;
p = pitch, in inches;
x = depth of radiator, in inches; and
e = base of natural logarithms.

Heat dissipation is given by the equation

$$Q=34.8\ M\left(1-e^{-\frac{Bx}{M^A}}\right) \tag{2}$$

where Q = heat dissipation in H. P. per 100° F. temperature difference and A and B are empirical constants.

The constants A and B were determined from actual heat tests on the three radiators mentioned above, and are as follows:

$\frac{1}{4}$-inch pitch: $A=0.24$; $B=0.0616$;
$\frac{3}{8}$-inch pitch: $A=0.11$; $B=0.0283$;
$\frac{1}{2}$-inch pitch: $A=0.23$; $B=0.0258$.

[1] The basis for the empirical equations (1) and (2) may be briefly stated as follows:
The air flow is found to be proportional to the free air speed, and may be expected to be proportional to the density of the air ρ, so that

$$M=b\,\rho\,V \tag{1'}$$

in which b may be expected to be a function of the "free area" a, the depth x, and the distance between plates $(p-t)$. From the form of the equa-

It was assumed that the head resistance might be regarded as made up of two parts: (1) that due to the impact of air on the edges of the plates, including suction on the rear; and (2) that due to skin friction of the air passing over the plates. The results were very well represented by the equation

$$R = n\ V^2\ (0.00016\ t + 0.0000025\ x) \tag{3}$$

where R = head resistance in pounds per square foot frontal area; and

n = number of plates per foot width of core.

The values given by this equation are a little low when n equals 48; i. e., with one-fourth-inch pitch, but for spacings greater than one-fourth inch, the resistance per plate was found to be constant for a given speed and air density.

The weights of the radiators and the water contained in them were estimated from geometrical considerations and the densities of copper and solder, assuming the Lepère type of construction. The weight is given by the equation

$$W = 0.0557\ n\ x \tag{4}$$

where W = weight of core and contained water, in pounds per square foot frontal area.

A lift-drift ratio of 5.4 was assumed for the airplane, and the horsepower absorbed (per square foot) is

$$H.\ P. = \left(R + \frac{W}{5.4}\right)\left(\frac{V}{375}\right) \tag{5}$$

Figure of merit is the ratio of the power dissipated to the power absorbed.

$$F.\ M. = \frac{Q}{H.\ P.} \tag{6}$$

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tion, b must be dimensionless, and therefore x and $(p-t)$, each having the dimension of a length, must enter as a ratio, and b may be of the form

$$b = f\left(a,\ \frac{p-t}{x}\right) \tag{2'}$$

Also $b=0$ when $a=0$ and $p-t=0$
and $b=1$ when $a=1$ and $x=0$.

These conditions are satisfied by the equation

$$b = \sqrt{a}\left(1 - e^{-k\sqrt{\frac{p-t}{x}}}\right) \tag{3'}$$

and tests on ab[illegible] a value of 10.95 for the units used. Substitution of equation (3') in (1'), with proper units and the assumed value [illegible].

For heat t[illegible]tation, let

T = [illegible] water in the radiator and air passing through it

c = specific heat of the air

y = total perimeter of the air tubes (in unit frontal area) around a section perpendicular to the direction of air flow

q = heat transfer per unit surface per unit time per unit temperature difference

T_0 = value of T at front face of the radiator.

If it is assumed (1) that the heat transfer varies as the temperature difference, (2) that the transfer coefficient q is independent of the depth x, and (3) that the temperature change in the water with depth of core is negligible in comparison with the temperature difference T, so that the change in T may be regarded as due entirely to the rise in air temperature; then equating the heat gained by the air to that dissipated from the surface gives

$$M\,c\,d\,T = -q\ T\,y\,dx \tag{4'}$$

which integrates to the form

$$T = T_0\,e^{-\alpha x} \tag{5'}$$

where

$$\alpha = \frac{qy}{c\,M}$$

The heat transfer is given by the equation

$$Q = \int^{x} q\ T\,y\,dx \tag{6'}$$

which gives the form

$$Q = c\ M\ T_0\left(1 - e^{-\alpha x}\right) \tag{7'}$$

Experiment shows that q varies as M^{1-A}, where A is about 0.2, but varies with different types of radiators.

If T_0 is taken as 100° F., and proper units are used, equation (7') reduces to equation (2).

The following example illustrates the use of the equations. Let it be required to obtain the figure of merit of a radiator with plates one-sixteenth inch thick, one-half inch pitch, 16 inches deep, and at 120 miles per hour.

In equation (1) for mass flow of air,

$$M=0.110\sqrt{\frac{p-t}{p}}(V)\left(1-e^{-10.95\sqrt{\frac{p-t}{x}}}\right)$$

p = pitch = 0.5

t = thickness = $\frac{1}{16}$ = 0.0625

x = depth = 16

V = speed = 120

$$M=0.110\sqrt{\frac{0.5-0.0625}{0.5}}(120)\left(1-e^{-10.95\sqrt{\frac{0.5-0.0625}{16}}}\right)$$

$$M=0.110\sqrt{0.875}\ (120)\ (1-e^{-1.81})$$

M = 12.35 (0.836) = 10.33 lb. per sq. ft. per sec.

In equation (2) for energy dissipated,

$$Q=34.8\ M\left(1-e^{-\frac{Bx}{M^A}}\right)$$

$A=0.23$

$B=0.0258$

$$A=34.8(10.33)\left[1-e^{-\frac{(0.0258)\ (16)}{10.33^{.23}}}\right]$$

$$Q=359\ (1-e^{-0.241})=359\ (0.215)$$

Q = 77.3 *H. P.* per sq. ft. per 100° F.

In equation (3) for head resistance,

$R=n\ V^2\ (0.00016\ t+0.0000025\ x)$,

$n=24$

$$R=24\ (120)^2\left(\frac{0.00016}{16}+0.0000025\ (16)\right)$$

$$R=24\ (120)^2\ (0.00001+0.00004)$$

R = 17.27 pounds per square foot.

From equation (4),

$W=0.0557\ (16)\ (24)=21.4$ lb. per sq. ft.

From equation (5),

$$H.\ P.=\left(R+\frac{W}{5.4}\right)\left(\frac{V}{375}\right)$$

$$H.\ P.=\left(17.27+\frac{21.4}{5.4}\right)\left(\frac{120}{375}\right)=6.80\ H.\ P.\text{ per sq. ft.}$$

From equation (6),

$$F.\ M.=\frac{Q}{H.\ P.}=\frac{77.3}{6.80}=11.4$$

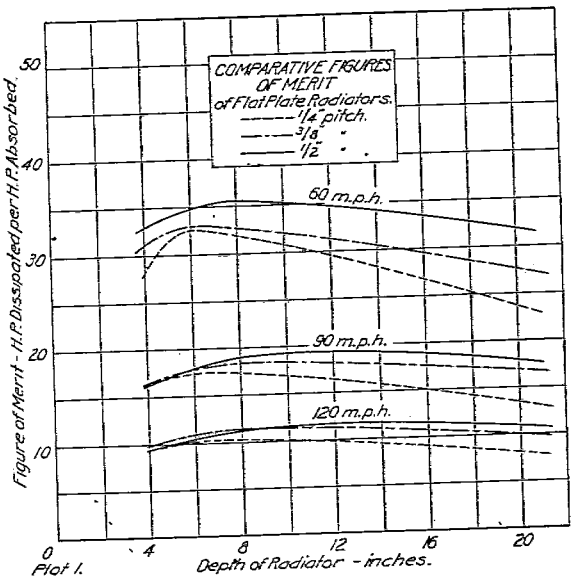
DESCRIPTION OF CURVES.

Plot 1 shows figure of merit computed with the aid of the above equations for various depths and for speeds of 60, 90, and 120 miles per hour, and for spacings of one-fourth, three-eighths, and one-half inch between plates. The curves illustrate the following points:

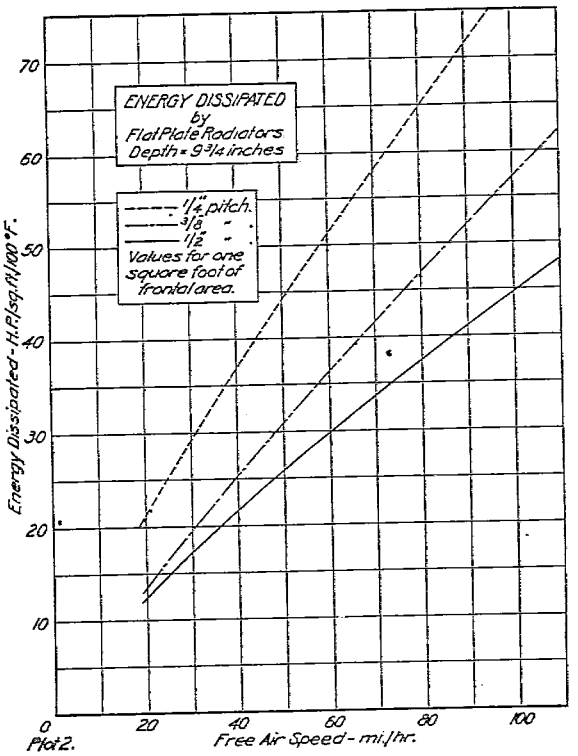
(1) The one-half inch pitch gives, in general, a higher figure of merit than those of closer spacings.

(2) For high speeds the radiator may be somewhat deeper than for low speeds.

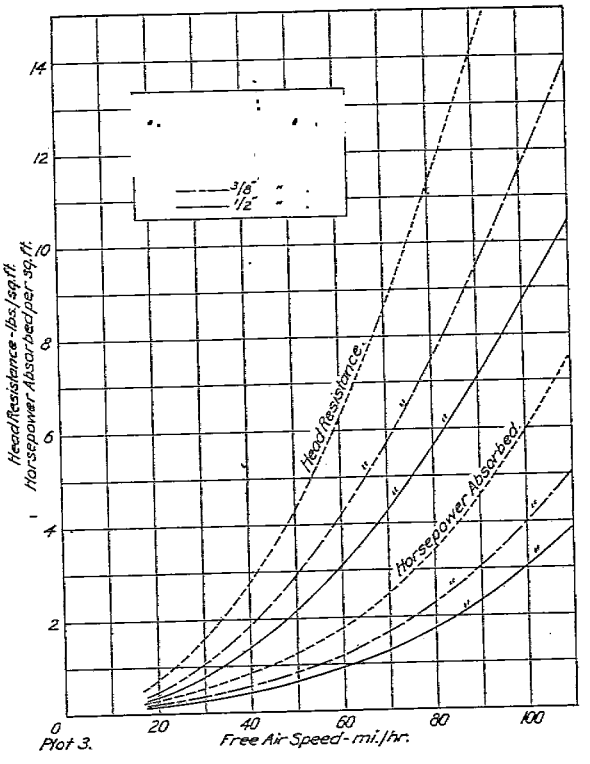
(3) For the higher speeds the most efficient depth is considerably greater than those in common use with the cellular types of core.



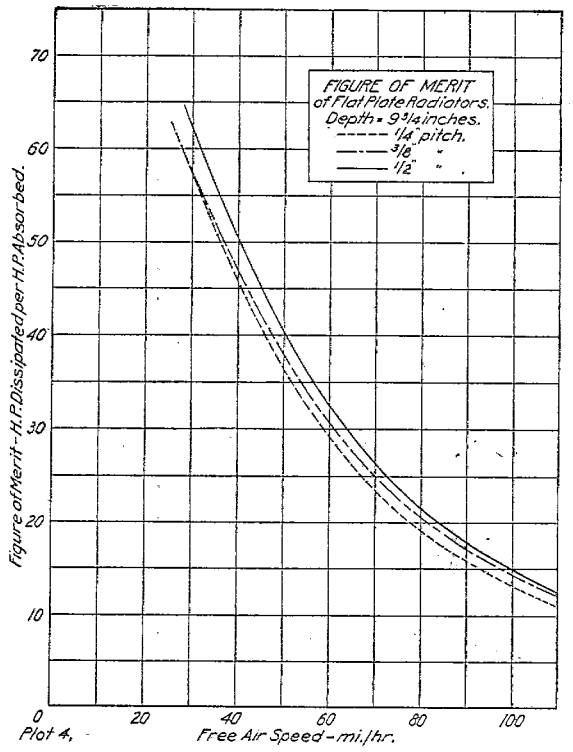
Plot 1.



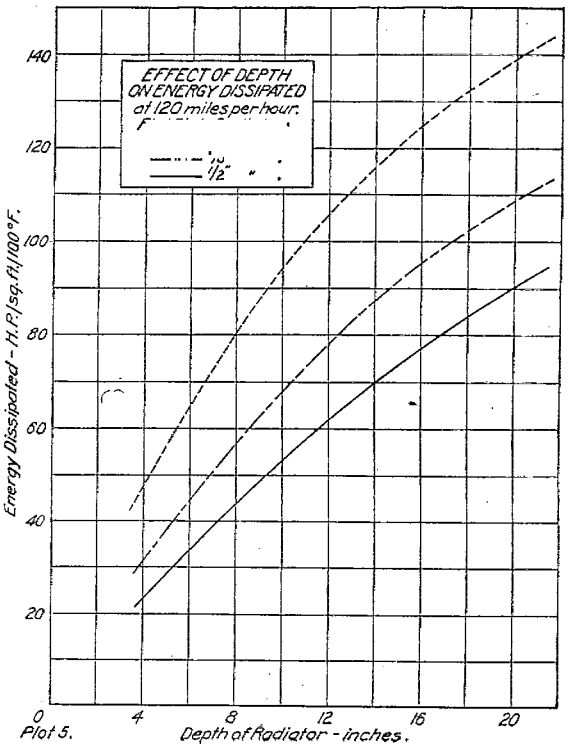
Plot 2.



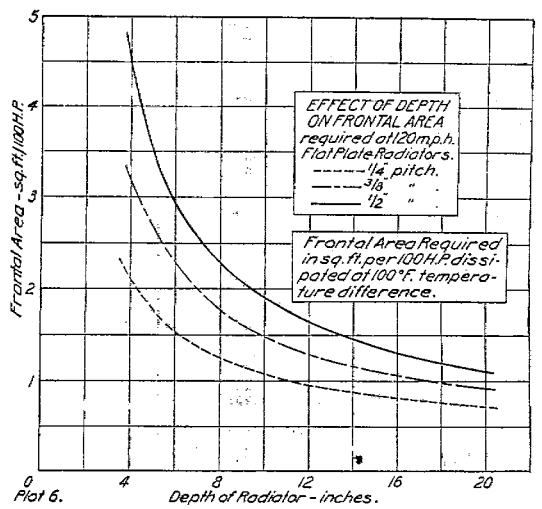
Plot 3.



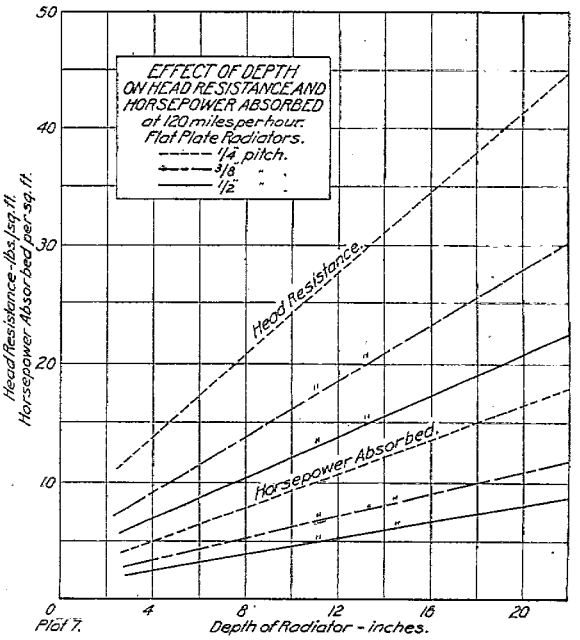
Plot 4.



Plot 5.



Plot 6.



Plot 7.

(4) For the higher speeds, the figure of merit is practically at its maximum value over a considerable range of depth, so that if considerations of compactness make it desirable to reduce the frontal area to a minimum, a reasonable increase in depth beyond the optimum will have but a small effect on the figure of merit.

The properties of the three radiators tested for heat transfer are reproduced from Technical Report No. 63, Part I, in plots 2, 3, and 4.

The effect of depth on the properties of the radiator is shown in plots 5, 6, and 7. The values of "area required per 100 H. P." in plot 6 are 100 times the reciprocals of the corresponding values of energy dissipated (in H. P./sq. ft.) of plot 5.

The following example illustrates one way in which the curves may be used. For a radiator with one-half inch pitch, to be used at 120 miles per hour, the maximum figure of merit is given on plot 1 as 11.7 at 13 inches depth. From plot 6, the frontal area of radiator required with 13 inches depth is 1.55 square feet per 100 horsepower. If, in order to reduce the frontal area, a depth of 20 inches should be used, the area required would be 1.10 square feet per 100 horsepower (from plot 6), and the figure of merit would be 11.0 (from plot 1). The horsepower absorbed would be increased from 5.6 to 8.1 per square foot (plot 7). Since, however, the frontal area may be reduced in the ratio $\frac{1.10}{1.55}$, the actual power absorbed that should be compared with the value for 13 inches depth would be $\frac{1.10}{1.55}$ (8.1), or 5.75. These results may be summarized as follows:

	13 inches.	20 inches.	Change.
			Per cent.
Area required, sq. ft./100 H. P.	1.55	1.10	−29
Horsepower absorbed, per square foot required at 13-inches	5.6	5.75	+ 3
Figure of merit	11.7	11.00	− 6

A careful distinction should be made between radiators whose water tubes are smooth, flat plates and other types using perforated plates, or deep and narrow tubes placed in rows, one behind the other. Holes in the water tubes, or spaces between them in the direction of the air flow, cause very great increase in head resistance and decrease in mass flow of air; and although the heat transfer per square foot of cooling surface may be increased by the great turbulence caused, it is at a very heavy cost in head resistance, and with a decrease in figure of merit.

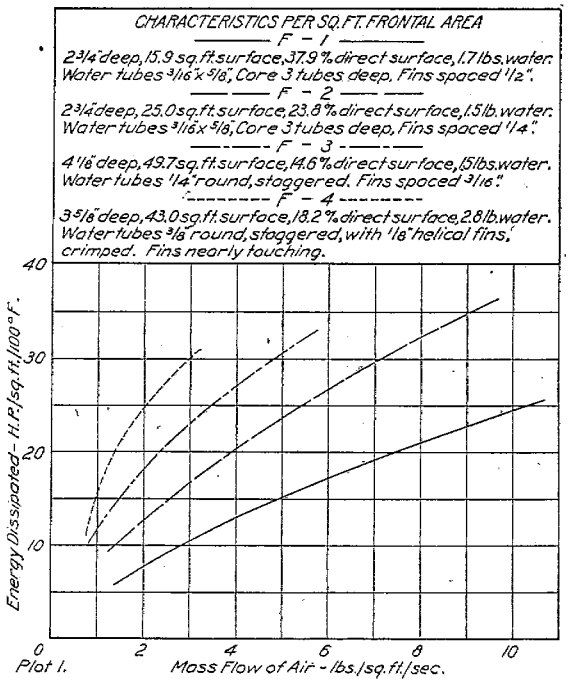
The effect of holes in the water tubes, and of spaces between them in the direction of the air flow is taken up later in this report under "Properties of whistling radiators," in which the properties of six such types are given.

PROPERTIES OF FIN AND TUBE RADIATORS.

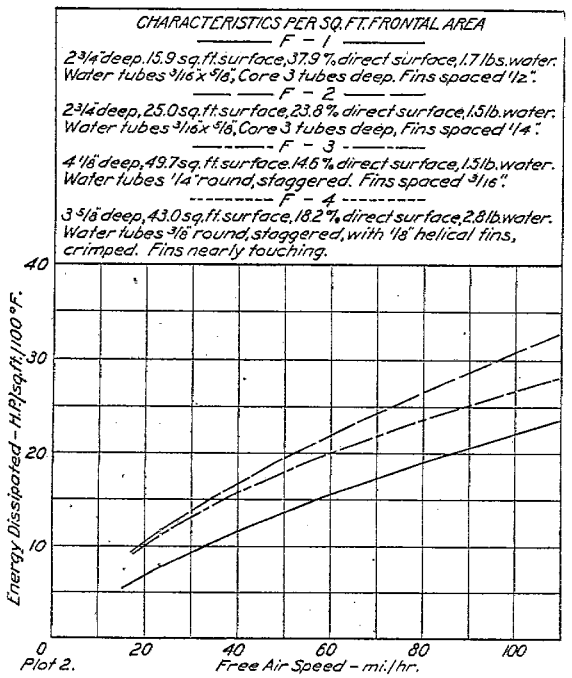
The properties of "fin and tube" types of cooling radiators are distinctly different from those of the better cellular types and warrant special mention.

In general, the fin and tube types are characterized by high head resistance and by low heat transfer at high speeds. The low heat transfer is accounted for by the small flow of air through the radiator and by the large amount of indirect cooling surface. For radiators as ordinarily made, with depths of 5 inches or less, head resistance has been shown to be due principally to the impact of the air on the front face and suction on the rear, and only to a small degree to skin friction on the walls of the air passages. With the fin and tube construction the effects of impact and suction are exaggerated; for each separate water tube is subjected to impact on one side and suction on the other, with the result that the total (projected) area subjected to impact and suction—on all tubes—is much greater than that necessitated in a radiator of the same size but of cellular construction. To the effect of this impact and suction must be added the effect of skin friction on the fins.

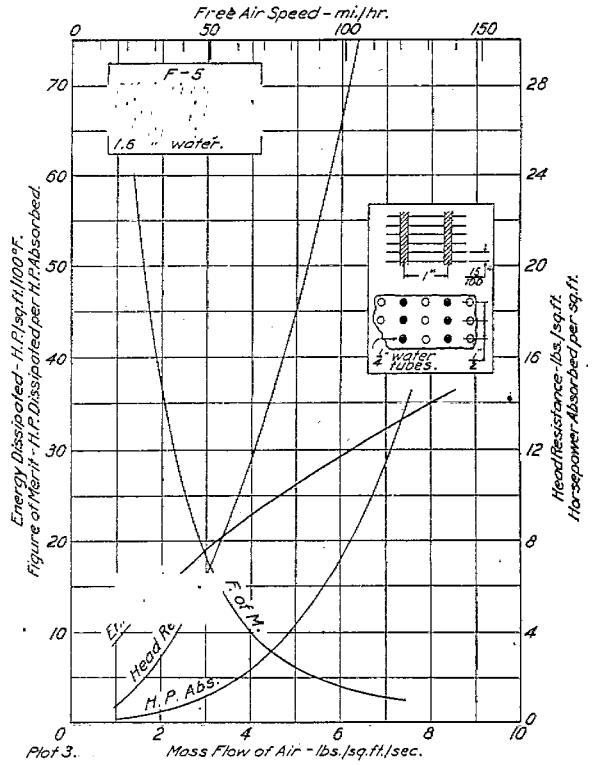
Excessive head resistance, accompanied by low heat transfer, make the fin and tube types unsuitable for use on an airplane where they would be exposed to a current of air at a high speed.



Plot 1.



Plot 2.



Plot 3.
Mass Flow of Air - lbs./sq. ft./sec.

The more compact types, however—notably F–4, which has large water tubes with crimped spiral fins nearly touching each other—show a relatively high rate of heat transfer at very low speeds, which justifies their extensive use on trucks and lower speed automobiles. Indeed the type F–4 would doubtless dissipate a considerable amount of heat with convection currents only.

The accompanying curves show the heat transfer (energy dissipated) for five types of core, in terms of the mass of air flowing through the core. The heat transfer is expressed in horsepower per square foot of frontal area, for a difference of 100° F. between the mean temperature of the water and the temperature of the entering air. Mass flow of air is expressed in pounds per second per square foot of frontal area. Energy dissipated is also shown in terms of free air speed; that is, the speed of the machine on which the radiator is mounted. Plot 3, reproduced from Technical Report No. 63, Part I, shows also the figure of merit of the radiator, which is the ratio of the horsepower dissipated to the horsepower absorbed.

The attempt to determine the relation between the mass flow of air through the core and free air speed was unsuccessful in the case of the type F–4 (with spiral fins); for the air flow was too small to be measured with the instrument used on the other radiators.

In general, it may be stated that fin and tube radiators are unsuitable for airplane use, with the possible exception of a type similar to F–4, placed in the wing, where the mass flow of air must be very small (even less than for the nose position), and consequently head resistance is not necessarily a detriment.

PROPERTIES OF WHISTLING RADIATORS.

Certain types of cooling radiators that whistle in an air stream show peculiar properties, and while radiators of this construction are not recommended, they are being used to some extent, and appear to be worthy of special mention.

TYPES OF CORE TESTED.

The whistling radiators tested at the Bureau of Standards fall into two general classes:

(1) Plain water tubes, and

(2) Perforated water tubes.

The photographs show the forms of construction. In each case the radiator is made up of separate water tubes about 2 inches deep (in the direction of air flow), arranged in rows. In addition, the plain tube type has fins spaced 2 inches apart.

The test sections included one of the plain tube type, designated as E–9, and five perforated tube types, made up in different depths, and with different spacings between rows of tubes. These sections are designated as follows:

E–1, $\frac{5}{16}$-inch pitch, 2 tubes deep.
E–2, $\frac{5}{16}$-inch pitch, 3 tubes deep.
E–3, $\frac{5}{16}$-inch pitch, 4 tubes deep.
E–4, $\frac{1}{2}$-inch pitch, 4 tubes deep,
E–5, $\frac{5}{8}$-inch pitch, 4 tubes deep.

CAUSE AND EFFECT OF THE WHISTLE.

The form of construction leaves continuous air passages *across* the radiator; that is, perpendicular to the direction of the air stream. In the plain tube type these air passages occur between the water tubes, and in the perforated tube types not only between the tubes but at each perforation. All through the radiator there are short columns of air, across the ends of which air is blowing, with the result that vibrations are set up in the short columns and perpendicular to the air stream. The resulting whistle will of course vary widely in intensity and in pitch as the speed of the air stream varies, and conditions of resonance have very marked effects, not only upon the sounds, but upon the properties of the radiator.

By the vibrations of the cross columns, air is alternately being forced into and withdrawn from the fast moving stream. Air drawn out of the stream will be retarded and air forced into it will be accelerated, thus acting as a drag on the stream. These two effects cause a great decrease in the flow of air through the radiator and a great increase in head resistance.

At the same time the very great turbulence caused in the air stream results in a high heat transfer per square foot of cooling surface *for a given mass flow of air through the radiator*, and this increase in heat transfer may be so great as to counterbalance the decrease in air flow, but is not great enough in any case observed to overcome the disadvantage of the increased head resistance.

DESCRIPTION OF CURVES.

The accompanying curves show the properties of the six types of radiator, expressed as follows (all values have been reduced to an air density of 0.0750 pound per cubic foot):

Free air speed, in miles per hour.

Mass flow of air through the radiator, in pounds per second per square foot frontal area.

Energy dissipated, in horsepower per square foot per 100° F., difference between the temperature of the entering air and the mean of the temperatures of the entering and leaving water.

Head resistance, in pounds per square foot frontal area.

Horsepower absorbed, in horsepower per square foot frontal area.

Figure of merit is the ratio of the horsepower dissipated to the horsepower absorbed.

Plot 1, reproduced from Technical Report No. 63, Part I, shows the properties of the plain tube type, E–9. Its heat transfer is high, but its head resistance is also high, and the figure of merit is low. The great weight of the radiator accounts for its very low figure of merit at low speeds.

The relation between the mass flow of air through the radiators and free air speed is shown for three of the perforated tube types in plot 2. For ordinary radiators this relation is linear, and it is practically so for the plain tube whistling type. The irregular form of these curves shows clearly the effect of the peculiar conditions of air flow in the radiator, the sudden changes in slope of the curve corresponding to sudden changes in tone of the whistle.

Plot 3 shows energy dissipated (heat transfer) in terms of mass flow of air. The curve for the type E–4 was determined by interpolation,[2] since the water boxes had been removed in order to measure its head resistance. Too much importance should not be assigned to the fact that some of these curves show a high heat transfer with a given mass flow of air, for Plot 4 shows that the highest curve of Plot 3—that for E–3, four tubes deep, five-sixteenth inch pitch—becomes low when heat transfer is plotted against free air speed. About 70 radiators tested at this bureau have been graded from "A" for very high heat transfer for a given free air speed to "E" for very low heat transfer, and these perforated tube types are graded "D" on such a scale.

Plots 5 and 6, reproduced from Technical Report No. 63, Part I, show the complete properties of the type E–4, in terms of mass flow of air, and in terms of free air speed, respectively. The curves are reliable within the ranges covered, but the irregular relation between mass flow of air and free air speed makes extrapolations extremely doubtful.

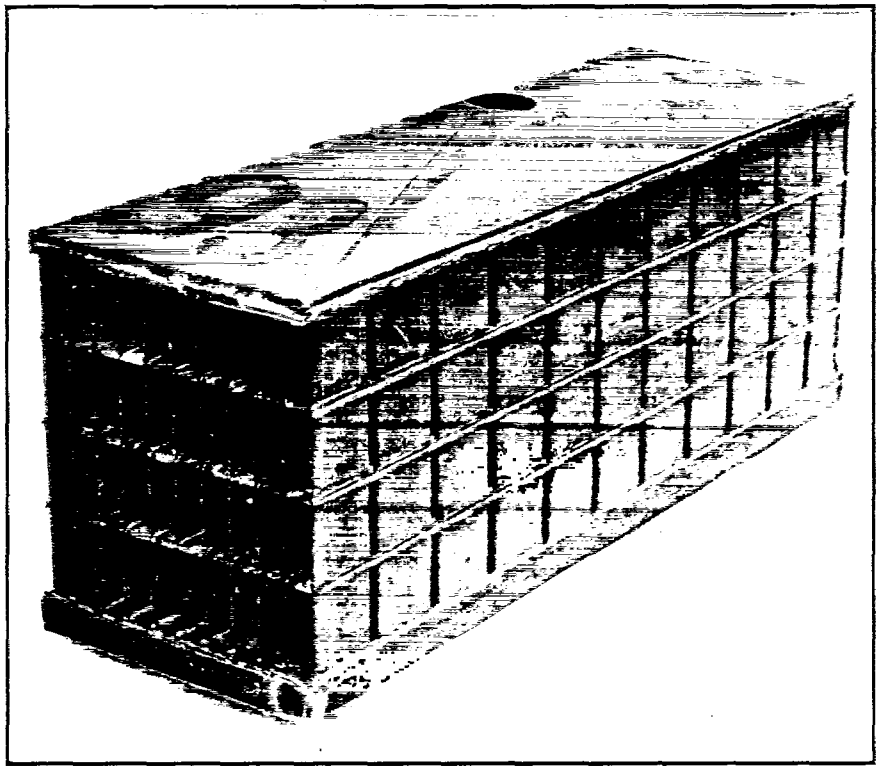
Head resistance of ordinary radiators is nearly proportional to the square of free air speed, but the irregularities in the head resistance curves of Plots 1 and 6 show that this is not the case with the whistling types.

COMPARISON BETWEEN WHISTLING RADIATORS AND FLAT PLATE RADIATORS.

The foregoing statements should not be interpreted as applying in any degree to radiators whose water tubes are flat plates with smooth surfaces, and continuous from front to rear of the radiator.

The whistling types are characterized by low air flow and often low heat transfer (for a given free air speed), by high head resistance, and by a low figure of merit; but the flat plate types are characterized by high air flow and heat transfer (for a given free air speed), by low head resistance, and by a high figure of merit. The following comparative tables show the superiority of the flat plate types over the whistling types.

[2] Types E–3, E–4, and E–5 are each four tubes deep, and are five-sixteenths, one-half, and five-eighths inch pitch, respectively. The heat transfer for E–3 and E–5 was found to be proportional to the number of tubes per foot width of front, and this proportionality was used in interpolating for E–4.



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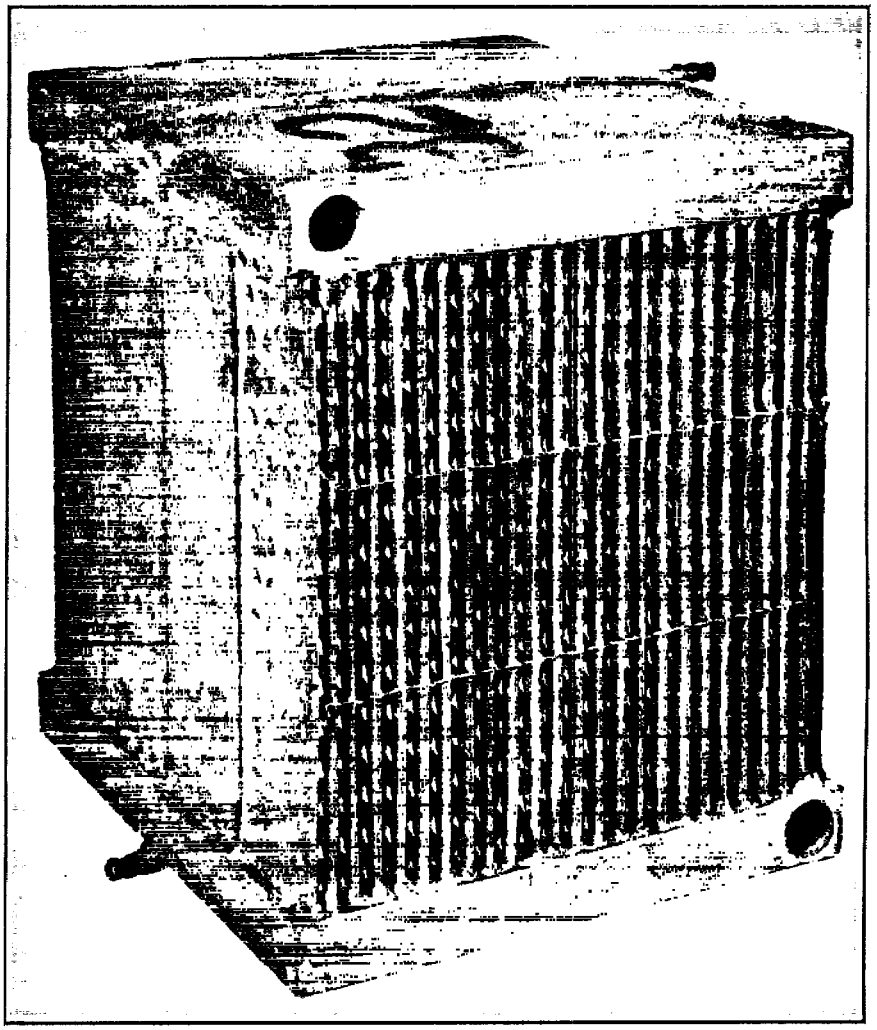

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TABLE I.—*Perforated tube type and flat plate type of the same pitch and practically the same depth.*

Type.	E–4, perforated tube.	E–8, flat plate.	Type.	E–4, perforated tube.	E–8, flat plate.
Pitch, inches	0.5	0.5	Energy dissipated, horsepower per square foot per 100° F	30.8	29.8
Depth, inches	9.50	9.75	Head resistance, pounds per square foot	6.73	3.12
Cooling surface, square feet, per square foot front	29.5	39.2	Horsepower absorbed per square foot	1.75	.92
Mass flow of air, pounds per second per square foot	3.70	5.48	Figure of merit	17.6	32.4

Test data are not available for a direct comparison of plain tube and flat plate types of radiator of the same pitch, but the plain tube whistling type E–9 of three-fourths inch pitch may be compared with flat plate types of one-half inch pitch. It was previously shown that for flat plate types a pitch of one-half inch is better at high speeds than the closer spacings, and there seems to be no reason to suppose that a pitch of three-fourths inch would be less efficient. A comparison of the two types would appear to give an advantage to the one of three-fourths inch pitch. Two depths of the flat plate type are considered, giving approximately the same direct cooling surface (not including fins), and the same actual depth of tubes, respectively, as the whistling types. A plain tube whistling type H–3, described in a French report, is also included.

TABLE II.—*Comparison of flat plate and plain tube radiators.*

FREE AIR SPEED, 120 MILES PER HOUR.

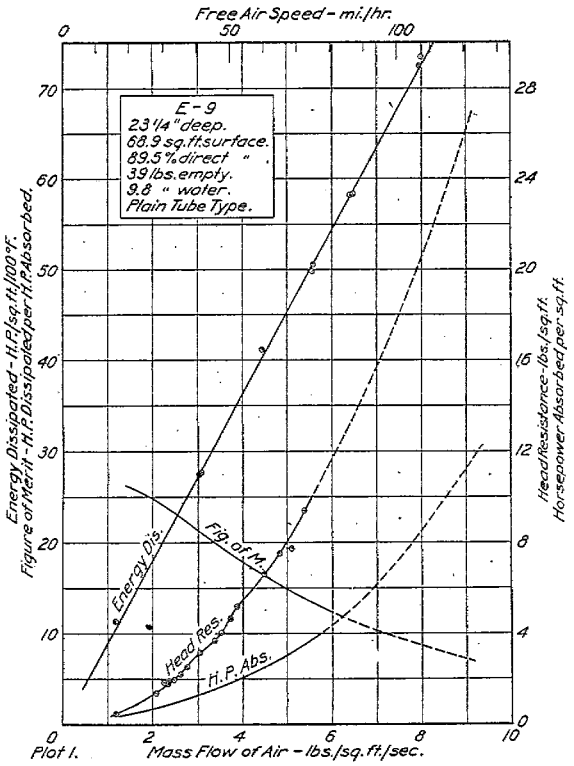
Type.	Flat plate.	Flat plate.	E–9, plain tube.
Pitch, inches	0.5	0.5	0.75
Depth, inches	16.	20.	23.25
Direct cooling surface	61.9	80.5	61.7
Mass flow of air	10.33	9.91	8.96
Energy dissipated	77.4	90.5	81.5
Head resistance	17.1	20.8	25.9
Horsepower absorbed	6.7	8.2	11.2
Figure of merit	11.5	11.0	7.3

FREE AIR SPEED, 90 MILES PER HOUR.

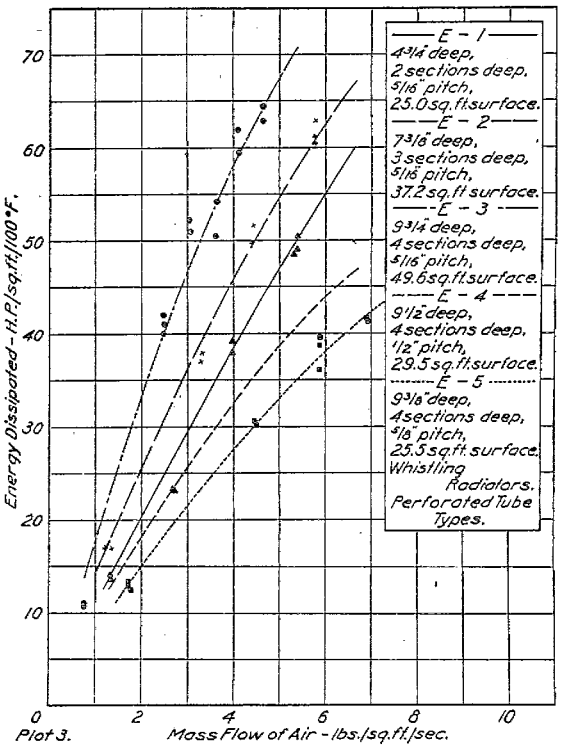
Type.	Flat plate.	H–?, plain tube.	Flat plate.	E–9, plain tube.
Pitch, inches	0.5	0.57	0.5	0.75
Depth, inches	16.	15.8	20.	23.25
Figure of merit	18.9	12.5	18.0	10.8

CONCLUSIONS.

The above tables seem to show clearly that any increase in heat transfer caused by perforations in the water tubes or spaces between them in the direction of air flow is at the expense of a great increase in head resistance and is accompanied by a decrease in the figure of merit. The same result has been found in the case of turbulence vanes in cellular radiators, and indeed no type of radiator is known to this bureau in which an artificial increase of turbulence is not accompanied by a decrease in figure of merit. For use in obstructed positions, such as the nose of the fuselage, it may be necessary to sacrifice figure of merit of the radiator core for the benefit of heat transfer, but for unobstructed positions it appears that smooth straight air passages through the radiator should be provided.



Plot 1.



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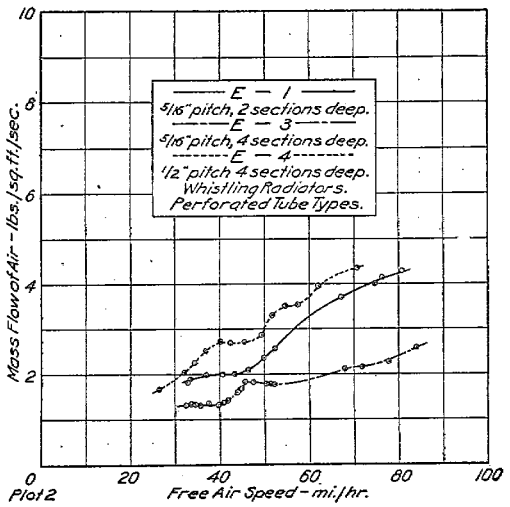

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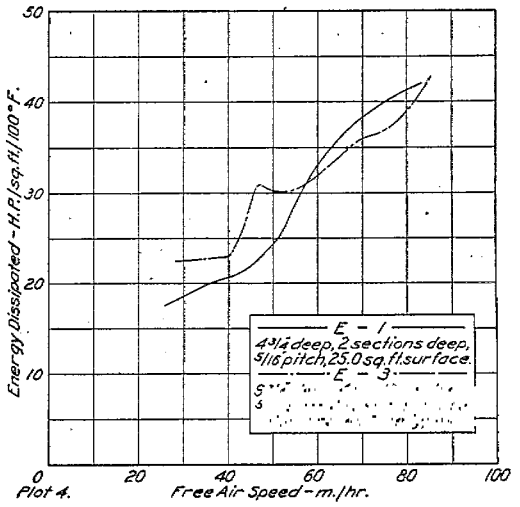
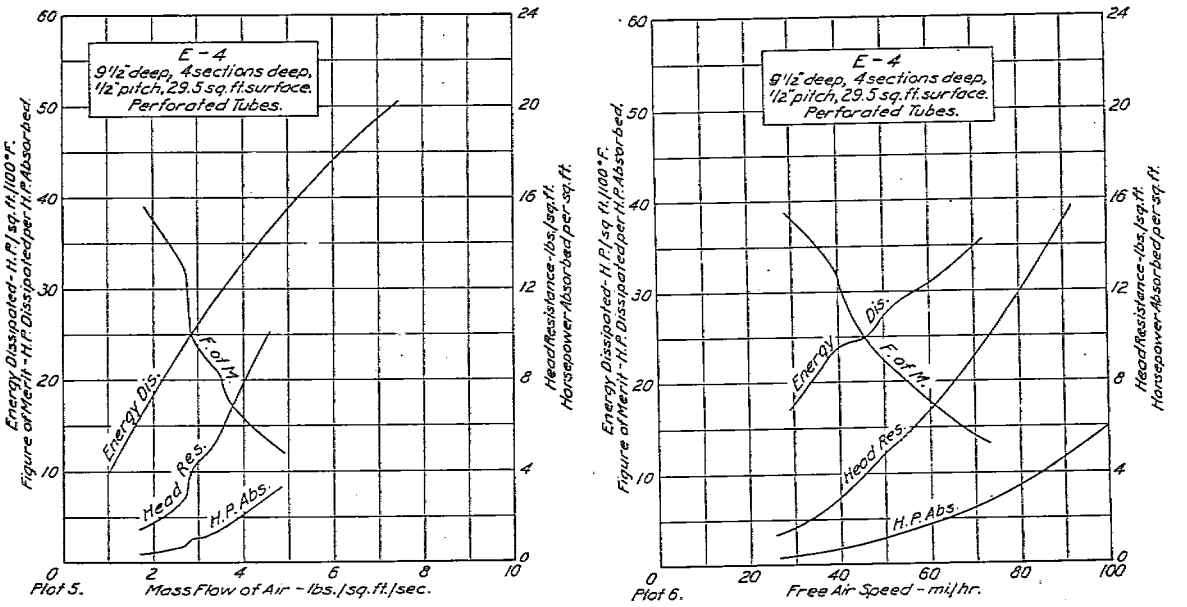


Plot 4.



EFFECTS OF YAWING AIRPLANE RADIATORS.

Wind-tunnel tests on radiators for aeronautic engines have usually been made with the face of the radiator normal to the direction of the wind, so that although local eddies might be set up, the general direction of the air stream was straight as it passed through the radiator. If the axes of the air passages are inclined at an angle with the general direction of the air stream; or, in other words, if the face of the radiator is not normal to the air stream, the properties of the radiator will be somewhat changed, and it is the purpose of this report to show the general form of these changes.

The effect of inclining the radiator at an angle with the air stream, or yawing the radiator, is of interest in connection with the following conditions:

(1) Radiator mounted in the propeller slip stream, where the air strikes the radiator at other angles than normal to its face.

(2) Radiator mounted in any position (such as the wing) where the axes of its passages for the air are not parallel to the direction of motion of the plane.

(3) Radiator pivoted about an axis perpendicular to the direction of motion of the airplane. This construction represents one method that has been suggested for the regulation of cooling capacity.

The effects of yawing a radiator through angles from 0° to 45° fall into three classes, namely, the effects on (1) air flow through the core, (2) heat transfer, and (3) head resistance.

(1) EFFECT ON AIR FLOW THROUGH THE CORE.

For ordinary types of radiator the mass flow of air through the core is directly proportional to the free air speed (that is, to the speed of the air stream in which the radiator is placed) when the face of the radiator is normal to the air stream. Measurements of air flow have been made on two sections in yawed positions and the results obtained with one of the sections are shown in Plot 1, which indicates that the relation between mass flow of air through the core and free air speed is still linear between 30 and 90 miles per hour, but the line points below the origin. The two types tested are of square cell construction, about one-fourth inch in diameter.

A similar effect, but much more pronounced, was observed in the case of a single type of core when in its normal (not yawed) position. The result of this test is also shown in Plot 1. This type, which is the type G–4, of Technical Report No. 63, Part I, is made up with irregular shaped cells inclined at an angle of 4½° to the normal to the face, alternate rows of cells being inclined on opposite sides of the normal It is illustrated in the accompanying photograph.

The type G–4 and types that whistle in the air stream are the only radiators noted in which the air flow through the core is not directly proportional to the free air speed.

(2) EFFECT ON HEAT TRANSFER.

No tests have been made at this bureau to determine the effect of yawing the radiator on heat transfer, but a French report mentions a section (type not specified, but probably cellular) which showed an increase in heat transfer as the angle of yaw increased from 0° to 20° and then a decrease, bringing the heat transfer back to its normal value with an angle of about 40°. The maximum increase over the normal value was about 7 per cent. The French report further states that "with radiators of water-plate type the increase in heat transfer is not so clearly shown, but the decrease in effectiveness is hardly perceptible below 40°."

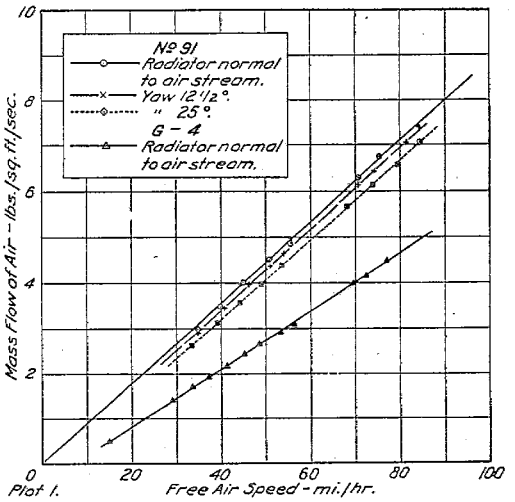
A British report states that three honeycomb types, each 12 inches square, showed a maximum increase in heat transfer of somewhat over 20 per cent with angles of yaw of about 45°. A large honeycomb section with 4.8 square feet frontal area gave a maximum of about the same per cent at 25°.

(3) EFFECT ON HEAD RESISTANCE.

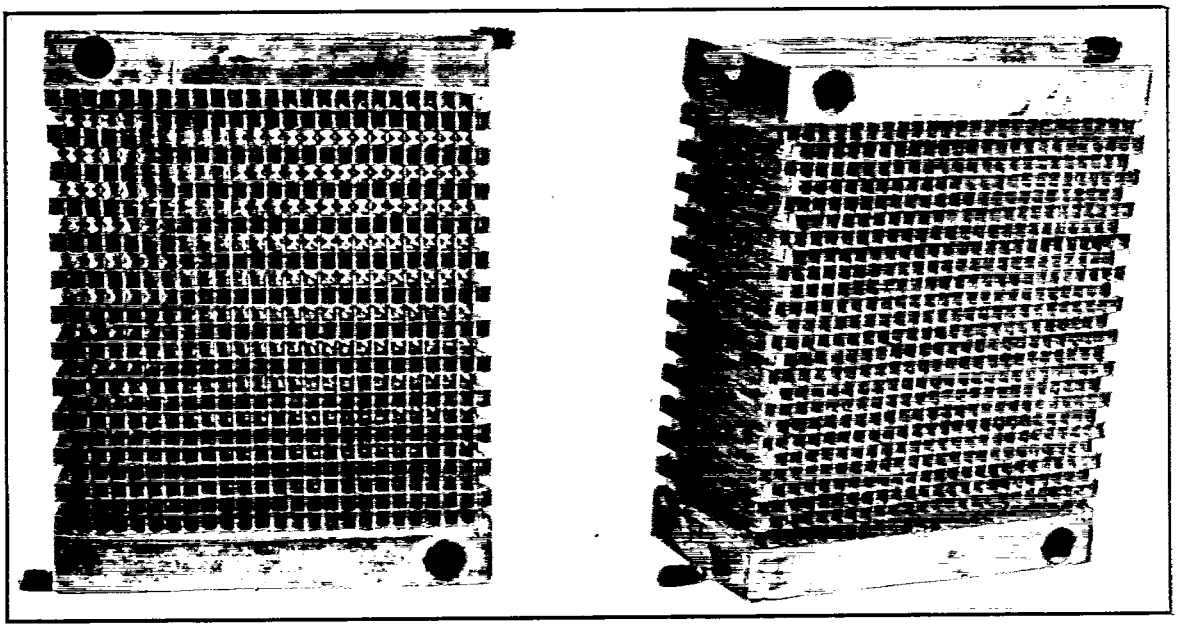
This effect has been treated in Technical Report No. 61, Part I, "Head resistance of radiator cores," and the accompanying curves are taken from that report.

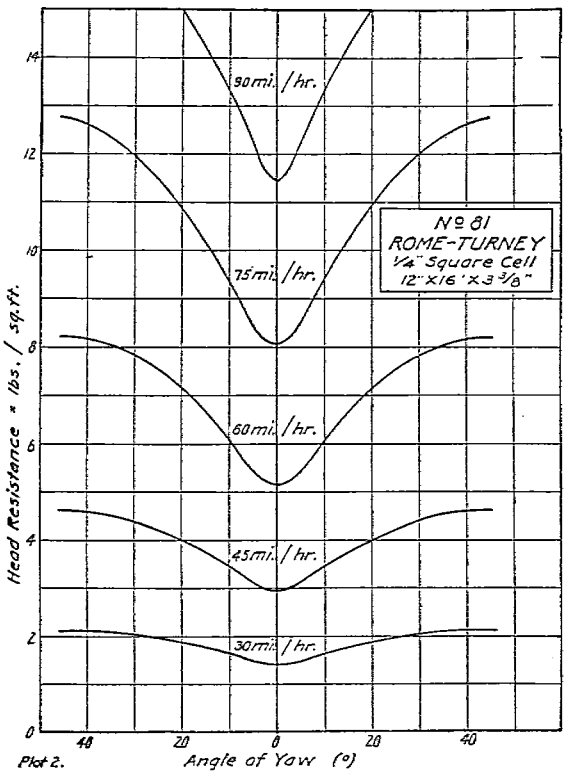
Plots 2 and 3, for one-fourth inch square cell (No. 81) and three-eighths-inch elliptical cell (No. 73), respectively, are given as typical of ordinary cellular types. The head resistance increases rapidly with the angle of yaw up to at least 30°.

Plots 4 and 5 are for special types which show the variations in the form of the curves. The type No. 80 has spiral air passages, and its head resistance shows very little change for angles up to about 30°. The type No. 79, shown in the photographs, is the type whose peculiar air flow is mentioned above. This shows a large increase in head resistance with increase in angle of yaw, and in particular it shows a very sudden increase with small angles.

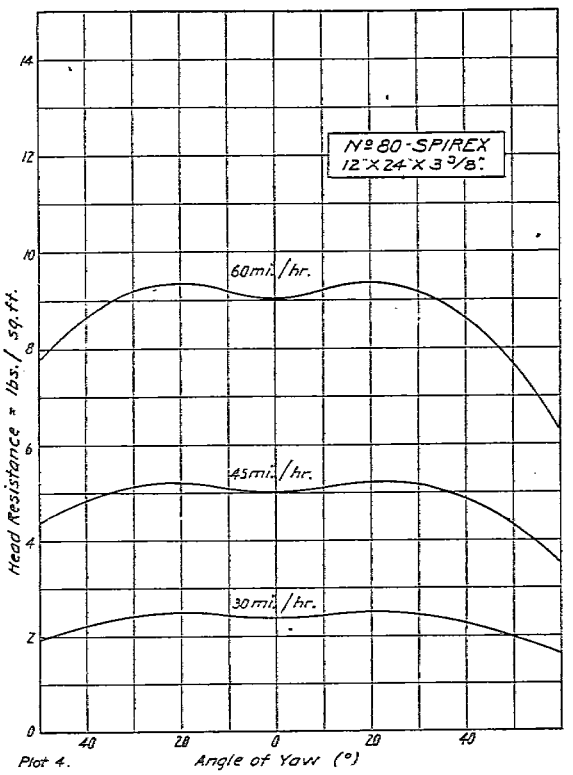


Plot 1.

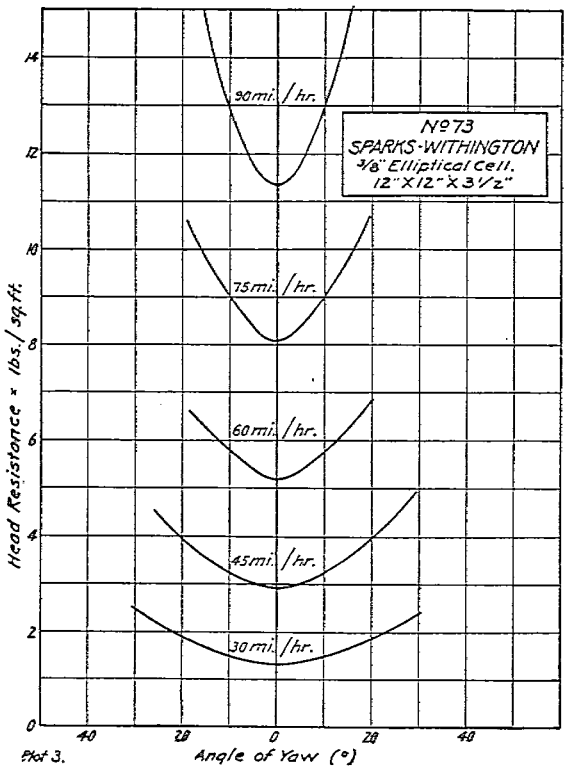




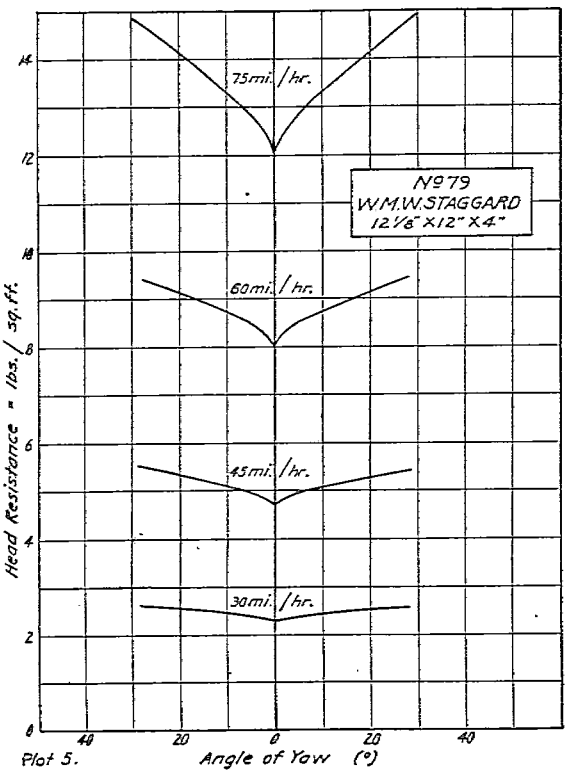
Plot 2.



Plot 4.



Plot 3.



Plot 5.