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TECHNICAL NOTES

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 657

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SUMMARY

A 1/3.5 full-size model of a seaplane float constructed from lines supplied by the Bureau of Aeronautics, Navy Department, was tested in the N.A.C.A. tank, first with smooth painted bottom surfaces and then with round-head rivets, plate laps, and keel plates fitted to simulate the actual bottom of a metal float. The percentage increase in water resistance caused by the added roughness was found to be from 5 to 20 percent at the hump speed and from 15 to 40 percent at high speeds. The effect of the roughness of the afterbody was found to be negligible except at high trims.

The model data were extrapolated to full size by the usual method that assumes the forces to vary according to Froude's law and, in the case of the smooth model, by a method of separation that takes into account the effect of scale on the frictional resistance. It was concluded that the effect of rivet heads on the take-off performance of a relatively high-powered float seaplane is of little consequence but that it may be of greater importance in the case of more moderately powered flying boats.

INTRODUCTION

The resistance of a metal seaplane float or hull on the water and in the air is increased by rivet heads and other small excrescences on its surface. In order to justify the increased cost of flush riveting, it is desirable to know the improvement in performance to be expected from the elimination of projecting heads. Tests of small models in the wind tunnel and the towing tank to determine the extent of this improvement have, in general, been considered unsatisfactory because of the difficulty in reproducing the riveted surfaces to scale and the uncertainties due to scale effects in evaluating the results.

The increase in the friction coefficient of the surface of a hull in contact with the water has been investigated in the N.A.C.A. tank by testing smooth and riveted planing surfaces (reference 1). In these tests, full-size rivet heads were used and the surfaces were towed at the actual speeds attained in practice. The results, however, are only generally indicative of the improvement to be gained by flush riveting because only a part of the resistance during take-off is frictional and the rivet pattern and the flow conditions vary over the float or hull. A more quantitative investigation must therefore be made by tests of models of actual hull forms that are large enough to minimize difficulties due to scale.

The speed of the towing carriage of the N.A.C.A. tank permits tests over the entire speed range of a model float so large that a fairly accurate reproduction of the riveted surfaces becomes practical. The rivet pattern, the plate laps, and the keel plate on a float of the type used on a Navy seaplane were simulated to scale on a 1/3.5 full-size model, and the model was tested to determine the magnitude of the increase in resistance caused by the excrescences. This paper presents the results of these tests, together with an analysis of the effect of the excrescences on the take-off performance of the full-size seaplane.

DESCRIPTION OF MODEL

The basic model, described in reference 2, was built of laminated mahogany and finished with several coats of gray pigmented varnish. The surface was sanded between coats but not after the final coat. The form of the model and the reproduction of the riveted surfaces are shown in figure 1. Round-head brass escutcheon pins having heads with a diameter of approximately 0.075 inch and a height of 0.025 inch were used to simulate the rivets. The heads of these pins correspond to 1/8-inch round-head rivets on the full-size float. Each pin was driven into a hole drilled in the model until the bottom of the head was hard down on the surface.

On the forebody were fitted two plate laps made of sheet brass 0.012 inch thick, tapered forward and faired into the hull with pattern wax, a keel made of two 0.30-inch-wide brass plates 0.012 inch thick and a center bar of 0.08-inch by 0.034-inch brass. The rivets at both the keel and the chine were at 0.16-inch pitch with single rows on the forward portion of the forebody and double rows on the after portion. Between the keel and the chines were four rows of rivets on each side at 0.39-inch pitch, corresponding to the stringers. Transversely, there were seven single rows of rivets at 0.18-inch pitch, corresponding to frames or bulkheads, and two double rows at 0.18-inch pitch in the plate laps.

The afterbody was fitted with a single keel plate of 0.012-inch brass, whose total width was 0.60 inch. The rivets in it and at the chines were at 0.16-inch pitch, arranged in a partly double and partly single row. Between keel and chines were four single rows of rivets on each side pitched at 0.42 inch, corresponding to the stringers. There were also six transverse rows of rivets at 0.19-inch pitch corresponding to frames or bulkheads. Altogether, in both forebody and afterbody, there were about 7,500 rivets.

APPARATUS AND PROCEDURE

The N.A.C.A. tank and its equipment are described in reference 3. In the present tests, the towing gear described in reference 4 was used. The data were obtained over a wide range of loadings by the "general" method, in which simultaneous values of resistance, trimming moment, and draft are recorded for various combinations of the independent variables, speed, load, and trim. The model was tested first with the rivets, the laps, and the keel plate on the forebody alone, and then on both forebody and afterbody in order to obtain the effect of the excrescences on the afterbody.

The data for the smooth model had been obtained in a previous test (reference 2) several months before the present tests were made.

The wetted lengths of the forebody and the afterbody at the keel and the chine were read during the tests of the model with rivets by means of the numbered stripes shown in figure 1. It was assumed that these wetted lengths were the same within the limits of errors in observation for the smooth and the riveted models.

RESULTS AND DISCUSSION

Data from Tests

The resistance and trimming moment obtained from the tests with rivets on the forebody alone, on both the forebody and the afterbody, and the data for the smooth model, reproduced from reference 2, are plotted in figures 2 to 7. The resistance includes the small air drag of the model, which is assumed to be unaffected by the changes in roughness of the bottom. The moments are referred to a point 5.32 inches forward of the step and 14.14 inches above the deck on the model, corresponding to the designed center of gravity of the seaplane. Moments that tend to raise the bow of the float are considered positive.

The percentage increase in resistance at a given trim caused by the presence of the excrescences on the forebody alone varies widely with load. It ranges from 5 to 20 percent at the hump speed and from 15 to 40 percent at 45 feet per second. This increase results, of course, in a decrease in maximum positive trimming moment and in a general shift of the moment curves in a negative direction.

For 7° trim and below, the increase in resistance caused by the excrescences on the afterbody is negligible. At higher trims, this increase becomes appreciable at the hump speed and quite large at high speeds. Apparently rivets on the afterbody of this float have little or no effect on water resistance during most of the take-off but might have some effect if high trims are used near the get-away speed, as in a "pull-up."

The observed wetted lengths at the keel and the chine are plotted against speed in figures 8 to 13. These wetted lengths are the distances from the intersections of the forebody keel and chine with the water to the main step and from the intersections of the afterbody keel and chine to the second step. Where the wetted area of the forebody is triangular in shape and lies wholly inside the chine, the wetted lengths of the forebody chine are considered negative and represent the intersection of the water with the chine extended aft of the main step.

The wetted lengths of the afterbody become zero at speeds slightly above the hump speed, where the afterbody is clear of the water. At 9° trim (fig. 11), however, the afterbody is again wetted at higher speeds by spray from the main step but the wetted lengths are indeterminate and are not plotted. This wetting, nevertheless, contributes additional frictional resistance at high speeds, as shown in figure 5 by the effect on the resistance of rivets on the afterbody.

Effect of Rivet Heads on Performance

In order to find the effect of the riveted surfaces on the take-off performance of a full-size float, the results of the model tests were used in take-off calculations for a typical single-float seaplane having the following characteristics:

Gross load, lb.	4,000
Wing area, sq. ft.	346
Span, upper and lower wing, ft.	36
Angle of wing setting, deg. .	2
Horsepower.	450 at 2,100 r.p.m.
Propeller	2 blades, 9 ft. 4 in. diameter. 18° blade setting at 0.75 R
Linear ratio, full-size to model, λ	3.5

Lift and drag curves from tests in the full-scale tunnel of an airplane having similar characteristics were used to determine the load on the water and the air drag at various speeds throughout the take-off run. The drag curve for the seaplane, excluding the float but including the float struts and the tip floats, was assumed to be the same as that for the airplane with landing gear as tested in the wind tunnel. (The air drag of the float is included in the water resistance.) The curves were modified for ground effect by the method given in reference 5. The thrusts for full and three-fourths power, representing a higher power loading, were calculated from the data of reference 6.

The full-size resistances of the smooth and riveted floats were first calculated from the model data by the usual assumption that the model and full-size forces and speeds are related according to Froude's law; i.e., the resistance varies as the cube of the linear dimensions when the speed varies as the square root of the linear dimensions. The detailed procedure to be used when general-test data are available is given in reference 7. In this case, the floats were assumed to be free-to-trim at low speeds, at best trim from 55 to 86 feet per second, and pulled up to take off from 86 to about 97 feet per second. There was assumed to be no wind.

This procedure does not take into account the variation in friction coefficient with Reynolds Number in the change from model to full size and therefore might be misleading in estimating the effect of surface roughness. If it be assumed that the addition of rivets, plate laps, etc., does not influence wave making, i.e., that the pressure distribution remains essentially the same, the resulting increase in resistance is frictional in nature. It is therefore desirable to attempt a separation of frictional and wave-making resistance for a more accurate extrapolation of the model results. Although this separation is usually made for surface vessels, it is generally not attempted for seaplanes. The procedure followed in the present calculations is therefore described in detail.

The trims and loads at the various speeds had been determined in extrapolating to full size according to Froude's law. The wetted lengths at keel and chine for these trims and loads were interpolated from figures 8 to 13. The area of the wetted surface was then calculated from the wetted lengths and the lines of the float.

The sum of the average wetted lengths of the forebody and the afterbody was taken as the effective wetted length. This procedure assumes that, during planing, the boundary-layer condition applying just at the step does not change appreciably in the distance of the jump from the step to the afterbody surface. This assumption is not strictly accurate but, with this model, the error involved is believed to be of little consequence, the afterbody becoming completely dry at about one-half take-off speed.

The mean speeds over the wetted surfaces in the planing range were computed according to the formula

$$V_a = \sqrt{V^2 - \frac{2g\,\Delta}{w\ S\ \cos\tau}} \qquad (1)$$

where

V is speed of model (or hull), f.p.s.

Δ, load on model (or hull), lb.

w, specific weight of water, lb. per cu. ft.

S, bottom wetted surface projected on base plane, sq. ft.

τ, trim, deg.

This formula is simply a form of Bernoulli's equation and states that there must be a reduction in velocity head equivalent to the static head necessary to carry the load.

Below the hump, V_a was assumed to equal V, and between this region and the full planing region a smooth transition curve was drawn. The values of mean speed V_a, wetted surface, and wetted length are plotted against model speed in figure 14.

The corresponding Reynolds Numbers were calculated from these wetted lengths, mean speeds, and the kinematic viscosity of the water at the time of the tests (ν = 0.00001054 ft.^{2}/sec. at water temperature T = 73° F.). From these Reynolds Numbers, friction coefficients C_f were obtained from figure 15. This curve is essentially Schoenherr's mean line (reference 8), down to a Reynolds Number of about 10^6, and a mean of Schoenherr's smooth-plane results (reference 8) below that Reynolds Number. The resulting values of Reynolds Number and friction coefficients are also plotted in figure 14. The friction coefficient C_f is based on the square of the speed and is defined as follows:

$$C_f = \frac{F}{\frac{\rho}{2} A V^2} \tag{2}$$

where

- F is the frictional force, lb.
- A, wetted area, sq. ft.
- ρ, density of water, slugs per cu. ft.
- V, speed, ft. per sec.

In this formula, V is the speed of the flat submerged plate from which the coefficients were determined. For the float, V_a is substituted for V and the expression becomes

$$C_f = \frac{F}{\frac{\rho}{2} A V_a^2} \tag{3}$$

Once the friction coefficients had been obtained, the computation was quite similar to that usually performed in ship work. The frictional resistance of the model was estimated and deducted from the total water resistance. Curves showing the resulting frictional and total water resistances of the smooth model are given in figure 16. The residuary resistance and the model speed were then converted to full size according to Froude's law. Friction coefficients for full size were obtained from figure 15, and from them the frictional resistance for full size was computed for each speed. This resistance, added to the full-size residuary resistance, gave the total water resistance of the seaplane. The computations were performed in tabular form, a sample of which follows:

Item	Source	Value
Model: $\nu = 0.00001054$ ft.^{2}/sec.; w = 63.3 lb./cu. ft.; T = 73° F.		
(1) V, f.p.s.	Given	34.0
(2) V_a, f.p.s.	Fig. 14	33.0
(3) V_a^2, ft.^{2}/sec.^{2}	$(2)^2$	1,090
(4) Reynolds Number	Fig. 14	2.72×10^6
(5) C_f	Fig. 14	0.00365
(6) Wetted surface, sq. ft.	Fig. 14	0.923
(7) Frictional resistance, lb.	Equation (3) $\frac{63.3}{2 \times 32.2} \times (3) \times (5) \times (6)$	3.6
(8) Total resistance, lb.	Fig. 16	10.6
(9) Residuary resistance, lb.	(8) - (7)	7.0
Full-size: $\nu = 0.00001087$ ft.^{2}/sec.; w = 63.3 lb./cu. ft.; $\lambda = 3.5$; T = 70° F.		
(10) V, f.p.s.	$(1) \times \lambda^{\frac{1}{2}}$	63.6
(11) V_a, f.p.s.	$(2) \times \lambda^{\frac{1}{2}}$	61.7
(12) V_a^2, ft.^{2}/sec.^{2}	$(11)^2$	3,807
(13) Reynolds Number	$(4) \times \lambda^{\frac{3}{2}} \times \frac{0.00001054}{0.00001087}$	1.72×10^7
(14) C_f	Fig. 15	0.00265
(15) Wetted surface, sq. ft.	$(6) \times \lambda^2$	11.31
(16) Frictional resistance, lb.	Equation (3) $\frac{63.3}{2 \times 32.2} \times (12) \times (14) \times (15)$	113
(17) Residuary resistance, lb.	$(9) \times \lambda^3$	300
(18) Total resistance, lb.	(16) + (17)	413

For the model with projecting rivet heads, no direct method of separation was possible because the variation of coefficients of friction of the surfaces with Reynolds Number was not known. In this case, the surface cannot be considered as either a smooth surface or a true rough surface. In a true rough surface there is a continuous irregularity. With such a surface, if the irregularities are of sufficient magnitude, the variation of the coefficient of friction for any given length with Reynolds Number disappears, i.e., Froude's law will hold. (See references 9 and 10.) The friction coefficients obtained in the tests reported in reference 1 did not have this characteristic variation of rough surfaces but, in this case, the "density" of the rivets was low as compared with that on the bottom of the float. For lack of more accurate information, the surface of the float was therefore assumed to be more nearly a true rough surface, and the resistance of the riveted model was extrapolated entirely according to Froude's law. It is to be emphasized that such an extrapolation is much more nearly the true extrapolation for a surface with a large number of rivets than it is for a smooth surface because, as previously stated, hydrodynamically the riveted surface represents a compromise between smooth and rough surfaces and, for a rough surface of such magnitude, Froude's law would hold quite rigidly.

The results of the different take-off calculations are plotted against speed in figure 17, together with the computed thrusts at full power and three-fourths power. Generally, the presence of the rivets on the surfaces causes a small increase in total resistance at the hump speed and a considerable increase at planing speeds. The increase in resistance acting below the center of gravity causes a slight decrease in the free-to-trim angle at low speeds, no appreciable change in the best trim, but causes a shift in the trimming moment at best trim in a negative or nose-heavy direction. The difference between the total resistance of the smooth float obtained by the method of separation and that obtained by applying Froude's law to the total resistance of the model is very small at the hump speed but is as much as 8 percent at higher speeds. The take-off performances calculated from the thrusts available for acceleration (fig. 17) are as follows:

Float	Time, sec.		Distance, ft.	
	Froude's law	Separation method	Froude's law	Separation method
Smooth, full power . .	14.5	14.0	752	724
Riveted, full power . .	15.8	-	841	-
Smooth, three-fourths power	19.4	18.3	1,019	953
Riveted, three-fourths power	22.8	-	1,228	-

In a comparison of the full-power values, when the performance of the smooth hull according to the separation method is used as a standard, the riveted hull requires 13 percent more take-off time and 16 percent longer take-off run. When the performance of the smooth hull according to Froude's law is used as a standard, the riveted hull requires 9 percent more take-off time and 12 percent longer take-off run. If it were possible to extrapolate the data for the riveted model satisfactorily, the values would probably lie between those just given, say 11 percent more take-off time and 14 percent longer take-off run. It should be remembered, however, that these values are for round-head rivets. For the brazier type of head more commonly used, smaller increases - probably on the order of two-thirds of those for the round heads (see fig. 16, reference 1) - might be used. It may therefore be reasonably concluded that the usual float of about the same size with projecting rivet heads and with comparable propeller thrust would require about 7 or 8 percent more take-off time and 9 to 10 percent longer take-off run than would a smooth hull.

If, however, take-off with the same propeller but at three-fourths power is assumed, the power loading is greater and the effect of the roughness of the hull is appreciably increased. At this power, the riveted hull requires 24 percent more take-off time and 29 percent more take-off distance than the smooth hull, according to the separation method, and 17 percent more time and 20 to 21 percent more distance than the smooth hull, according to Froude's law. About two-thirds of a mean between these values probably represents the correct increase for brazier-head rivets. Accordingly, the increase in take-off time would be 14 percent and the increase in run would be 16 percent. Thus it seems that the hydrodynamic advantage of the smooth hull may be a matter of some importance at low reserve thrusts such as are typical of most flying boats. It should also be noted that the afterbody of most flying-boat hulls is wetted at planing speeds because the depth of step is relatively lower and, when such wetting occurs, rivets on the afterbody might result in a longer take-off run.

CONCLUSIONS

1. The percentage increase in the water resistance of the model caused by the projecting rivet heads, laps, and keel bar varies widely with load. It ranges from 5 to 20 percent at the hump speed and from 15 to 40 percent at 45 feet per second.

2. The increase in resistance caused by rivet heads on the afterbody is negligible except at high speeds and high trims.

3. The increase, caused by round-head rivets, in the total resistance of the single-float seaplane investigated is estimated to be less than 5 percent at the hump speed but as much as 25 percent at planing speeds. The resulting effect on take-off performance is small with the low wing and power loadings found in this class of seaplane.

4. With the size of model used (1/3.5 full size), the total resistance of the smooth float calculated by Froude's law was found to be 2 percent higher at the hump speed and 8 percent higher at planing speeds than that calculated by taking into account the effect of scale on the frictional resistance.

5. The prevailing practice of converting the total water resistance by Froude's law gives a margin of safety in practice and may be considered as satisfactory except where the ratio of full size to model is considerable.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., April 26, 1938.

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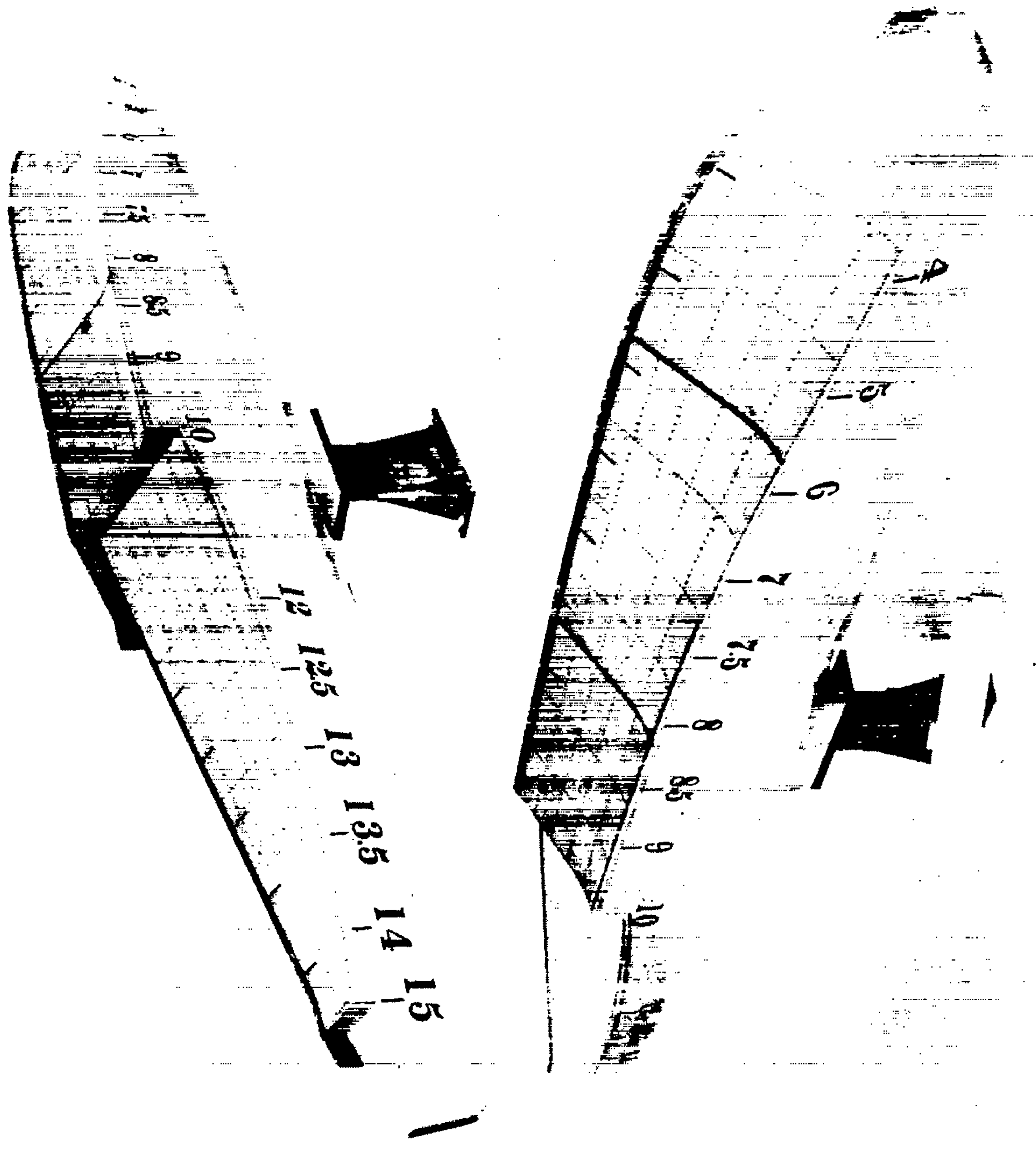

Figure 1.- Photographs of model showing reproduction of riveted surfaces.

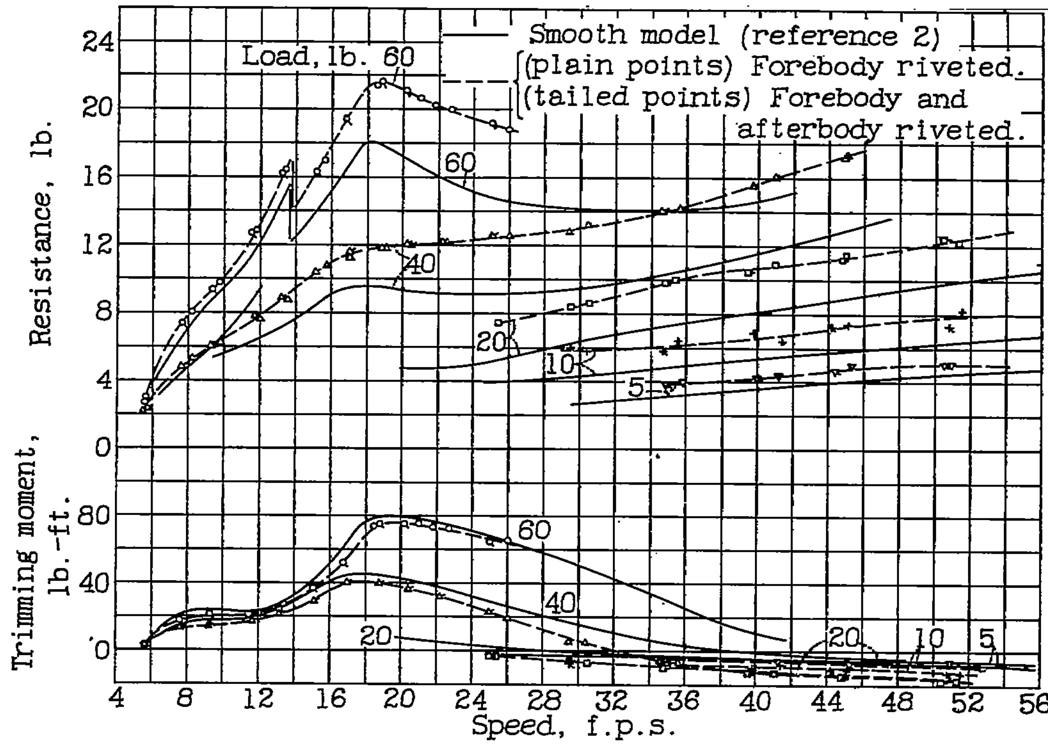

Figure 2.- Resistance and trimming moment. τ=3°

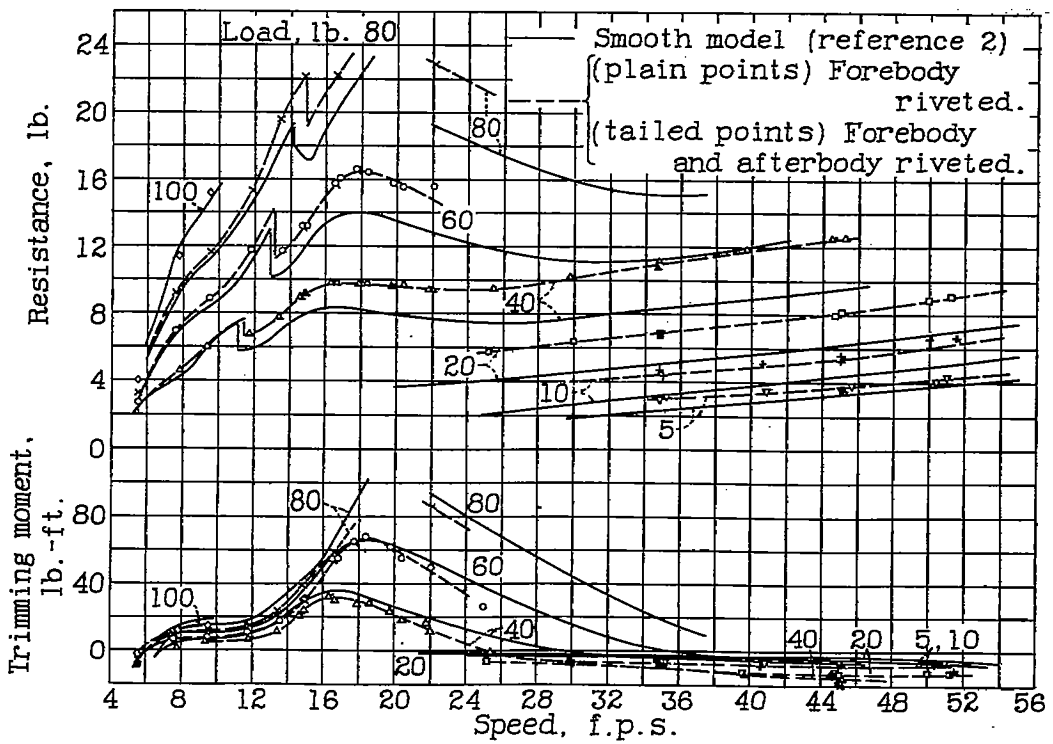

Figure 3.- Resistance and trimming moment. τ=5°

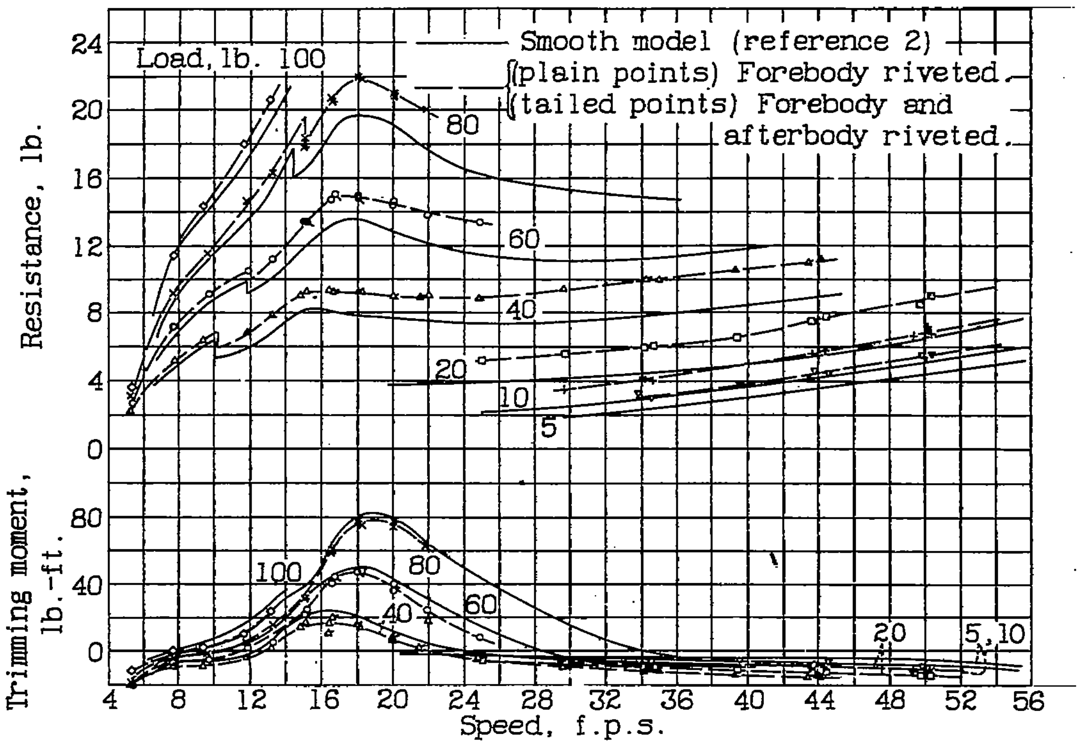

Figure 4.- Resistance and trimming moment. τ=7°.

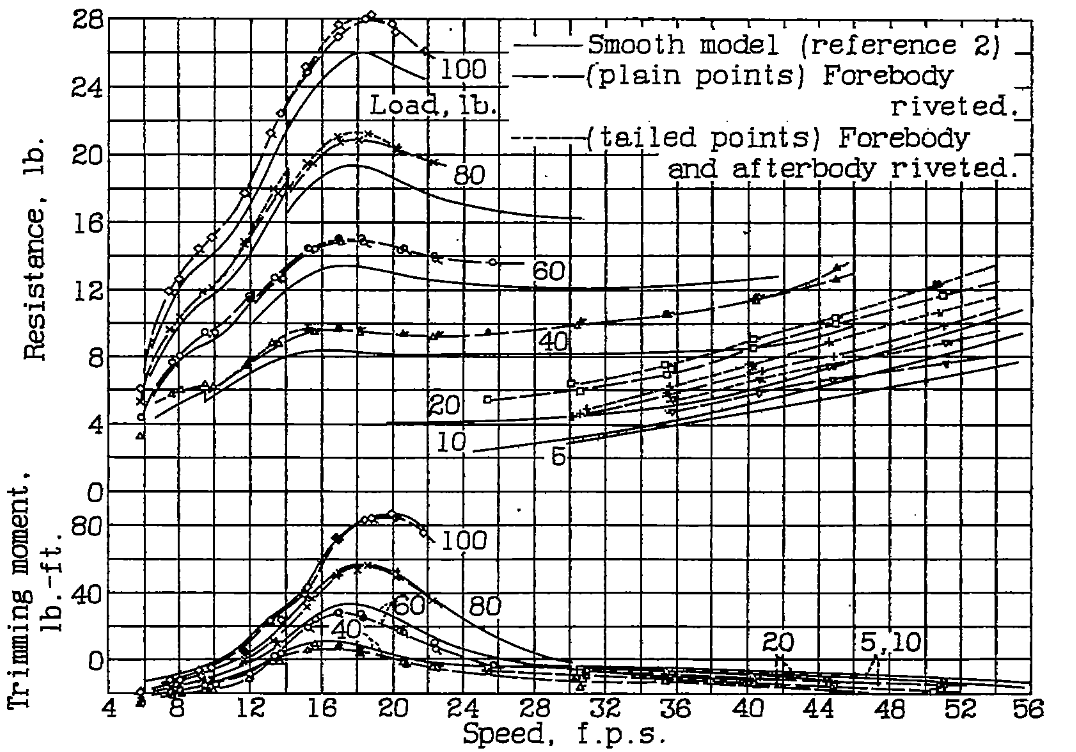

Figure 5.- Resistance and trimming moment. τ=9°.

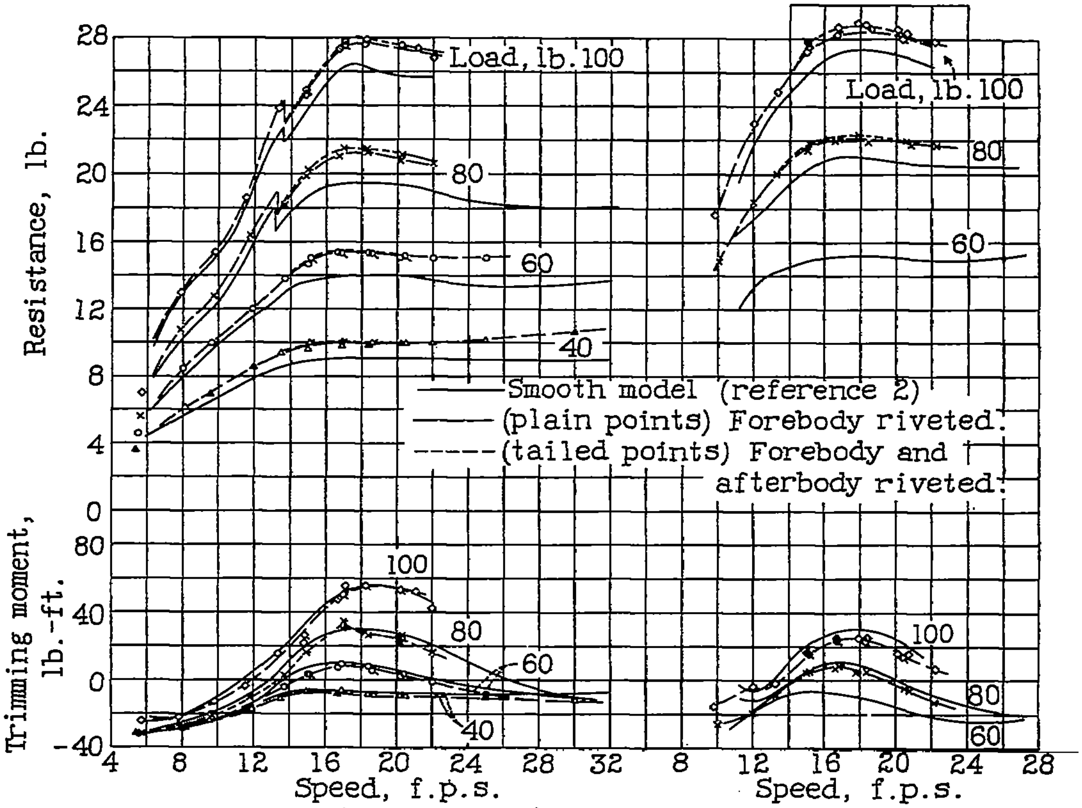

Figure 6.- Resistance and trimming moment. $\tau = 11°$.

Figure 7.- Resistance and trimming moment. $\tau = 13°$.

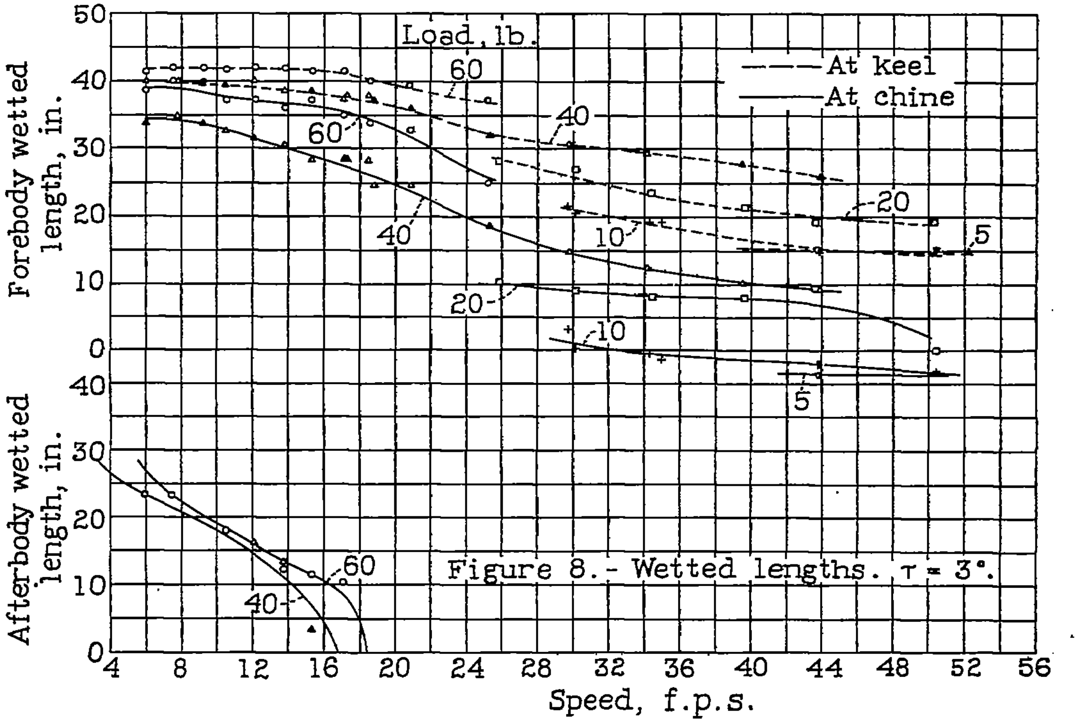

Figure 8.- Wetted lengths. $\tau = 3°$.

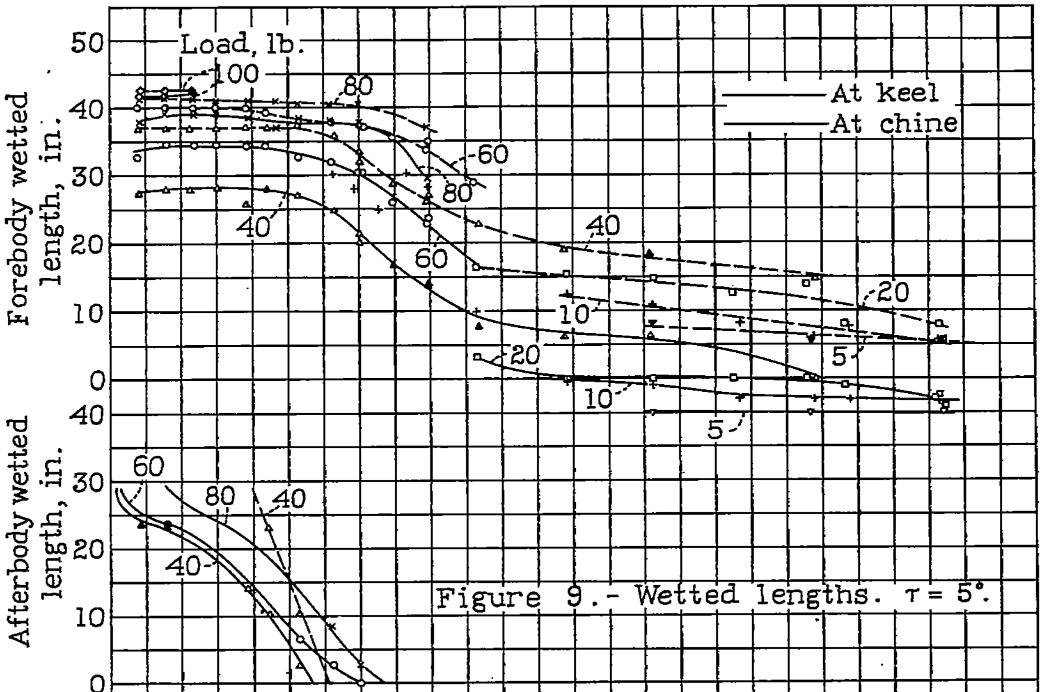

Figure 9.- Wetted lengths. $\tau = 5°$.

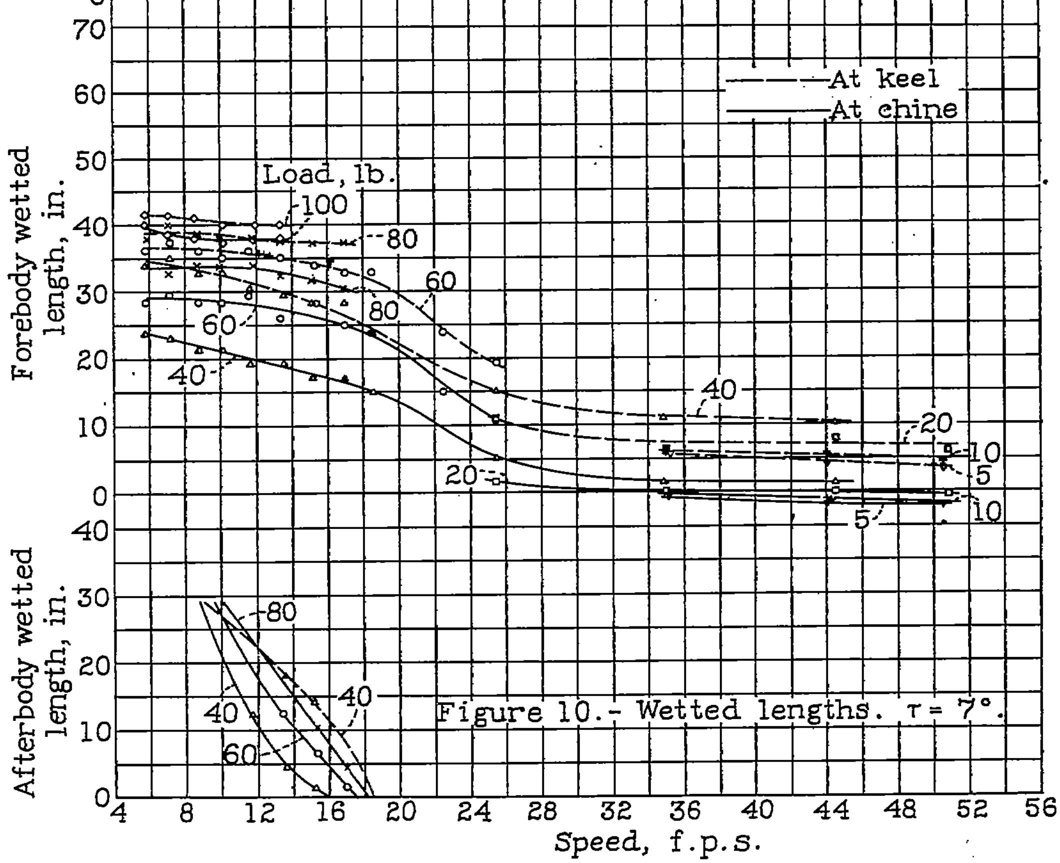

Figure 10.- Wetted lengths. $\tau = 7°$.

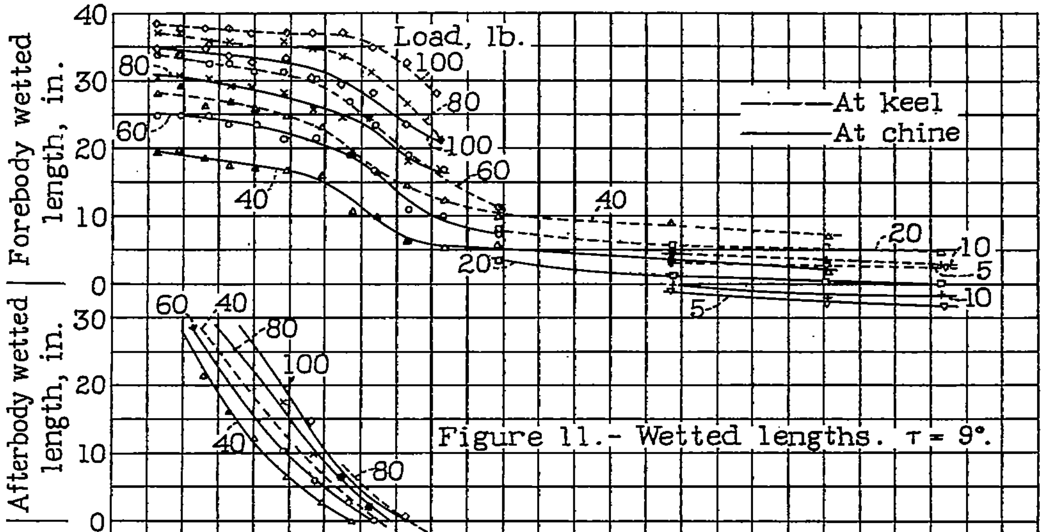

Figure 11.- Wetted lengths. $\tau = 9°$.

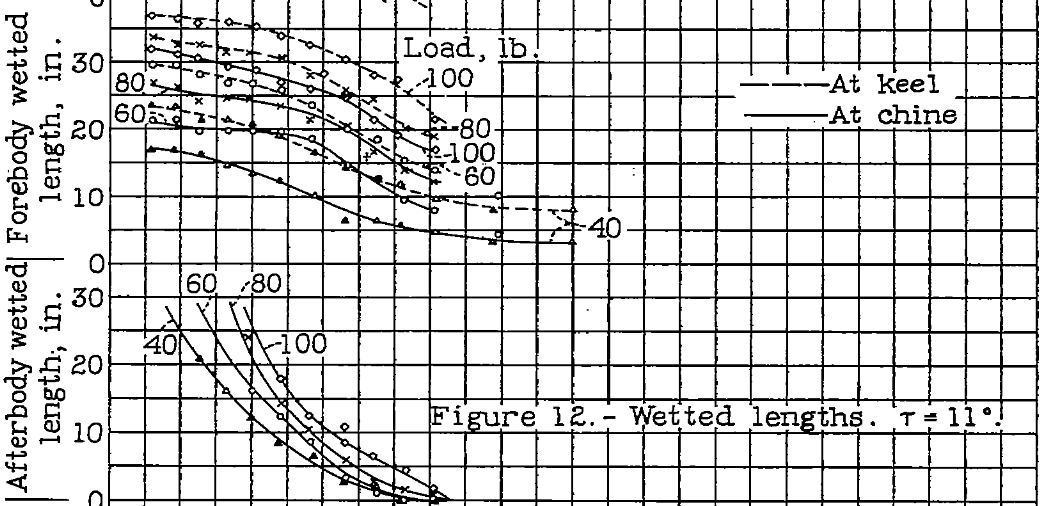

Figure 12.- Wetted lengths. $\tau = 11°$.

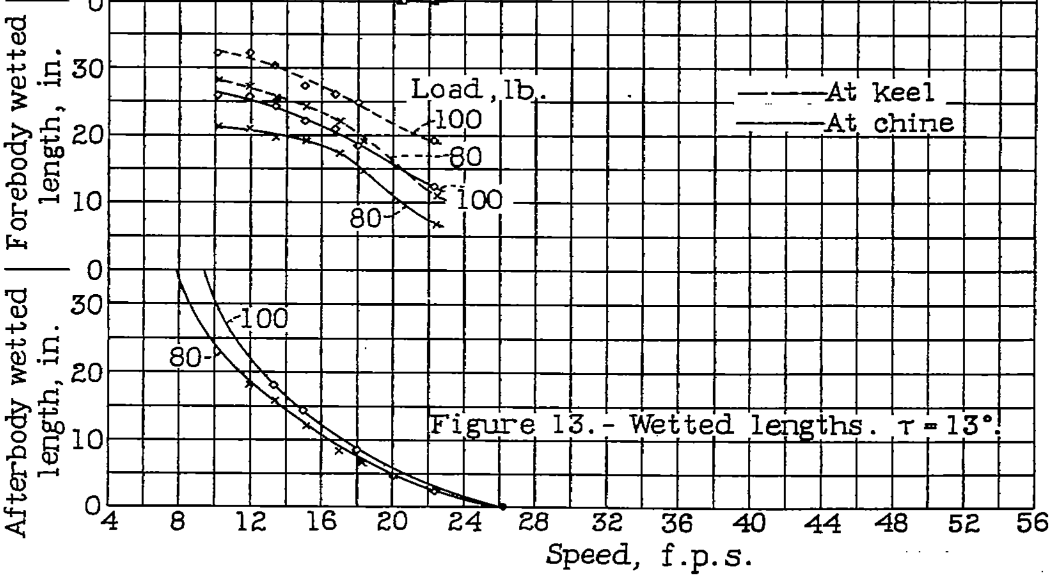

Figure 13.- Wetted lengths. $\tau = 13°$.

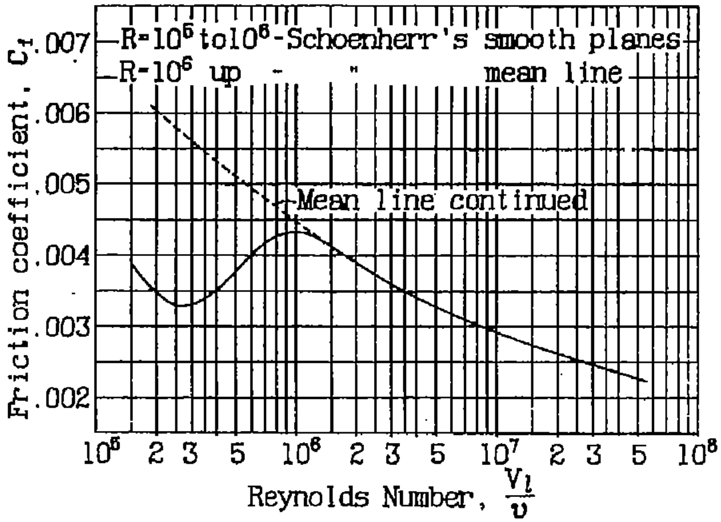

Figure 15.- Friction coefficient plotted against Reynolds Number. (See reference 8).

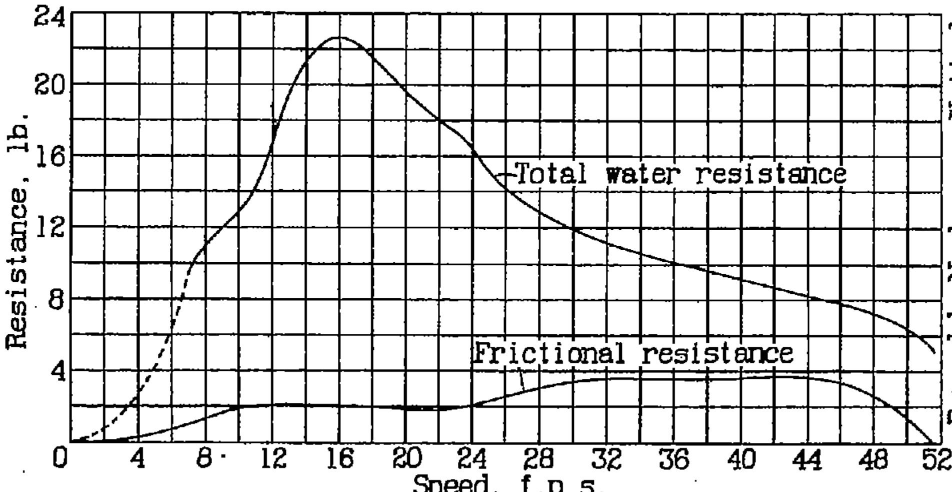

Figure 16.-Variation of frictional and total water resistance with speed for smooth model.

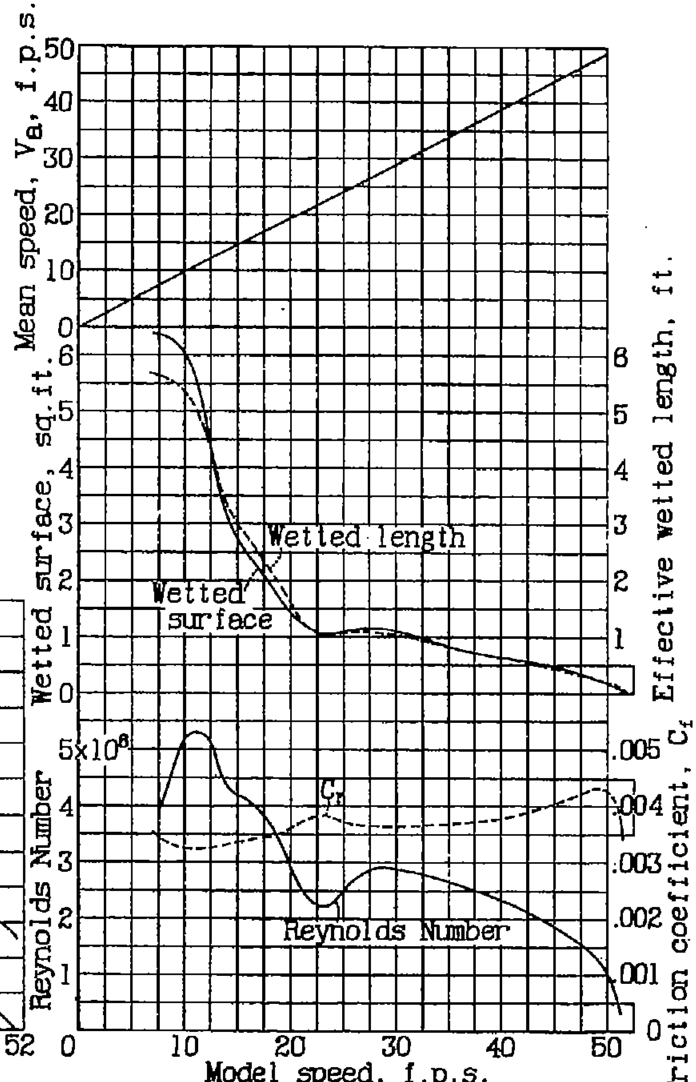

Figure 14.- Curves for determining frictional resistance of smooth model.

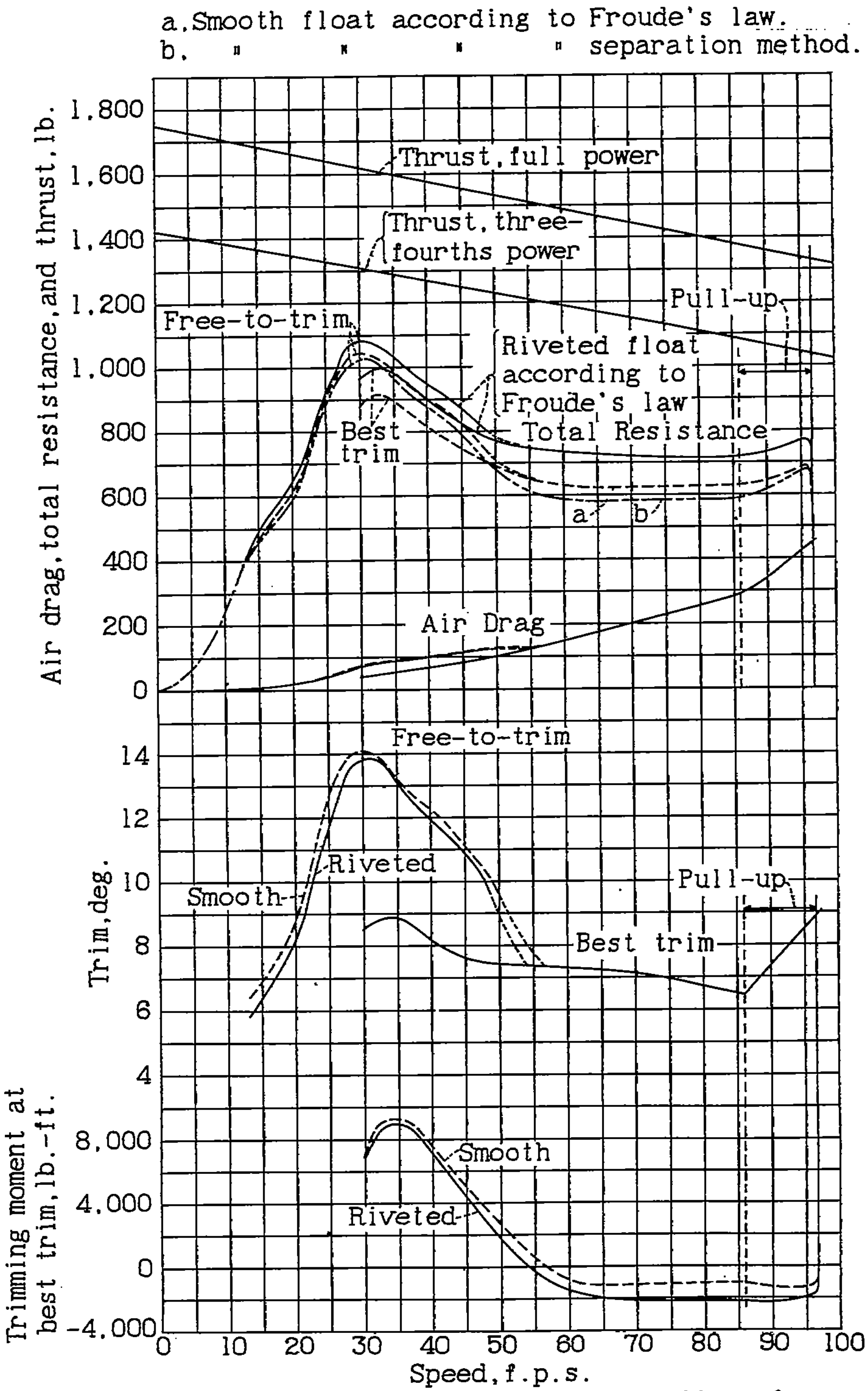

Figure 17.-Effect of rivets on take-off performance.