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MEMORANDUM REPORT

for

Bureau of Aeronautics, Navy Department  
TANK TESTS OF THE EFFECT OF RIVET HEADS, ETC.,  
ON THE WATER PERFORMANCE OF A SEAPLANE FLOAT  
By J. B. Parkinson and J. B. Robertson, Jr.

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June 4, 1936

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~~CONFIDENTIAL~~ MEMORANDUM REPORT

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TANK TESTS OF THE EFFECT OF RIVET HEADS, ETC.,

ON THE WATER PERFORMANCE OF A SEAPLANE FLOAT

By J. B. Parkinson and J. B. Robertson, Jr.

SUMMARY

A 1/3.5 full-size model of the Mark V float of the Bureau of Aeronautics, Navy Department, was tested in the N.A.C.A. tank both with smooth painted bottom surfaces and with roundhead rivets, plate laps, and keel plates fitted to simulate the actual bottom of a metal float. The augmentation in water resistance due to the added roughness was found to be from 10 to 12 percent at the hump speed and from 12 to 14 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 which 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 undoubtedly 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 the projecting heads. Tests of small models in the wind tunnel and 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 sur-

face of a hull in contact with the water may be found by testing smooth and riveted plates in the towing tank. If full-size rivet heads are used and the plates are towed at the actual speeds attained in practice, the errors due to scale effect are eliminated. 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 flow conditions vary over the float or hull. A more quantitative investigation must therefore be made by tests 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 float model so large that a fairly accurate reproduction of the riveted surfaces becomes practical. At the request of the Bureau of Aeronautics, Navy Department, the rivet pattern, plate laps, and keel plate found on the float of the O2U seaplane were simulated to scale on a 1/3.5 full-size model and tested to determine the augmentation in model resistance. This paper presents the results of these tests, together with an analysis of the effect of the surface roughness on the take-off performance of the full-size seaplane.

#### DESCRIPTION OF MODEL

The basic model used was that described in reference 1. This model was built up of laminated mahogany and smoothly finished with gray enamel. In order to simulate the riveted plated surface, a rivet and plating plan similar to that of the full-size float was laid out on the model bottom. Roundhead brass escutcheon pins having a head diameter of approximately 0.075 inch and a height of 0.025 inch were used as rivets. These correspond to 1/8-inch roundhead rivets on the full-size float.

On the forebody were fitted two plate laps made of 0.012-inch brass, tapered forward and faired into the hull with pattern makers' wax, a keel made up of two 0.30-inch wide hull plates and a center bar of 0.08-inch by 0.034-inch brass. Both keel plate and chine rivets were spaced at 0.16-inch pitch, single rows on the forward portion of the forebody and double rows on the after portion. Between the keel and the chines there were four rows of stringer rivets each side, pitched 0.39 inch. Transversely there

were seven rows of stiffener rivets, single rows at 0.18-inch pitch, and the two plate laps, double rows at 0.18-inch pitch.

The afterbody was fitted with a single keel plate of 0.012-inch brass, total width 0.60 inch. The rivets in it and at the chines were at 0.16-inch pitch, partially single row. Between keel and chines there were four single rows of stringer rivets each side, pitched at 0.42 inch. There are also six rows of stiffener rivets at 0.19-inch pitch. Altogether in both forebody and afterbody there were about 7,500 rivets. Pictures of the riveting are given in figure 4.

#### TEST 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 2 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 general data for the smooth model had already been obtained in a previous test (see reference 1). The rough model was tested first with the rivets, laps, and keel plate on the forebody alone, and then on both forebody and afterbody in order to obtain the effect of roughness of the afterbody.

During a later test of the rough model to determine the effect of additional small excrescences, the wetted lengths of the forebody and afterbody at the keel and chine were read 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 model and the rough model without these additional excrescences.

#### RESULTS AND DISCUSSION

##### Model Data

The resistance and trimming moment obtained from the tests with rivets on the forebody alone, on the forebody

and afterbody, and the same data for the smooth model, reproduced from reference 1, 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 bottom roughness. 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 design center of gravity of the O2U seaplane. Moments that tend to raise the bow of the float are considered positive.

The percentage augmentation in resistance at a given trim caused by the presence of the rivet heads, etc., is from 10.5 to 12 percent at the hump speed and from 12 to 14 percent at 45 feet per second. This augmentation results, of course, in a decrease in maximum positive trimming moment and a general shift of the moment curves in a negative direction.

For  $7^{\circ}$  trim and below, the augmentation in resistance caused by rivets on the afterbody is negligible and a single mean curve for each load is sufficient for both the condition with rivets and the condition without rivets. At higher trims, this augmentation becomes appreciable at the hump speed and quite large at high speeds. Apparently rivets on the afterbody would 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 keel and chine are plotted against speed in figures 8 to 13. These wetted lengths are the distances from the intersection of the forebody keel and chine with the water to the main step and that 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  $9^{\circ}$  trim, 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, contributed additional frictional resistance as shown by the effect of rivets on the afterbody at high speeds in figure 5.

### Full-Size 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 blade, 9 ft. 4 in. diameter. 18° blade setting at 0.75 R

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 wheels and 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 in reference 4. The thrust for full power and three-fourths power at several speeds was calculated from the data of reference 5.

The full-size resistance of the smooth and riveted floats was first calculated from the model data by the usual assumption that the model and full-size forces are related according to Froude's law, i.e., at corresponding speeds, the resistance varies as the cube of the linear dimensions. The detailed procedure when general test data are available is given in reference 6. 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.

The above procedure does not properly take into account the variation in friction coefficient with Reynolds Number in going 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 outside of the boundary layer remains essentially the same, the resulting augmentation in resistance is frictional in nature. Hence, it is desirable to attempt a separation of frictional and wave-making resistance for a more accurate extrapolation of the model results. Although this is usually done for surface vessels, it is not generally attempted for sea planes because of the difficulties in estimating the continually changing wetted surface and speeds over the bottom during an actual take-off. The procedure followed in the present calculations is therefore described in detail.

Utilizing the trim and load schedule previously determined for the smooth model in extrapolating to full size according to Froude's law, corresponding values of wetted lengths were interpolated from the wetted length curves, figures 8 to 13, and plotted against speed. (See fig. 14.) The product of the mean of the chine and keel wetted lengths as given by these curves times the mean bottom girth of the portion of bottom included by these lengths times a fore-and-aft curvature correction factor, which varied from 1.1015 to 1.0000, was taken as the effective bottom wetted surface. This area was accordingly plotted against speed in figure 14. To this area was added a curve of the approximate area of the wetted sides estimated from observation. This wetting of the sides occurred only at low speeds, and the area was small as compared with the wetted area of the bottom.

The sum of the mean forebody and afterbody wetted lengths was taken as the effective wetted length, except when the wetted surface was approximately triangular in outline, when the sum of the lengths taken through the center of gravity of the area of the wetted surface was used. This procedure assumes that during planing the boundary-layer condition that applies just at the step does not change appreciably in the distance of the jump from the step to the afterbody surface. This assumption, while unquestionably open to doubt, is probably not a long way from the truth. Furthermore, whatever error may be involved is believed to be of little consequence, the afterbody becoming completely dry at about one-half take-off

speed. Too, in this range, friction coefficient changes fairly slowly with Reynolds Number.

Then, the mean effective speeds of advance in the planing range were computed according to the formula

$$V_a = \sqrt{V_o^2 - \frac{2 g W}{w A \cos \Gamma}} \quad (A)$$

where  $V_o$  is speed of model (or hull), f.p.s.

$W$ , load on model (or hull), lb.

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

$A$ , bottom wetted surface projected on base plane, sq. ft.

$\Gamma$ , trim, deg.

This 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  $V_o$ , and between this region and the full planing region a smooth transition curve was drawn. (See fig. 14.) With the mean effective speeds of advance and effective wetted lengths so evaluated and the kinematic viscosity of the test water known, the corresponding Reynolds Numbers were calculated. From these Reynolds Numbers, friction coefficients were obtained using the curve of figure 15. This curve is essentially Schoenherr's mean line as given in reference 7 down to a Reynolds Number of about  $10^6$  and a mean of Schoenherr's smooth plane results, as given in the same paper, below that Reynolds Number. It is believed to represent the most dependable information available on friction coefficients for smooth surfaces, and to apply reasonably well to smooth surfaced models and hulls, especially in the fully turbulent regime above a Reynolds Number of about  $10^6$ .

Once the friction coefficients are obtained the computation is quite similar to that usually performed in ship



work. The frictional resistance of the model is estimated and is subtracted from the total water resistance. Curves showing the frictional and total resistance of the model are given in figure 16. The residuary resistance is then converted to full size according to Froude's law. The full-size wetted surfaces are obtained by multiplying the figures for the model by the square of the scale, and full-size Reynolds Numbers by multiplying the values for the model by the  $3/2$  power of the scale and dividing by the ratio of the kinematic viscosities. Friction coefficients for the full size are then obtained from figure 15, Schoenherr's curve, and the full-size frictional resistance is computed for each speed. This resistance, added to the residuary resistance, makes up the total water resistance. The computations are performed in tabular form, an example of which is given.

For Model

Temperature of water,  $73^{\circ}$  F.

Kinematic viscosity,  $0.00001054$  ft.<sup>2</sup>/sec.

$$\lambda = 3.5$$

$$\lambda^{1/2} = 1.87$$

$$\lambda^{3/2} = 6.54$$

$$\lambda^2 = 12.24$$

$$\lambda^3 = 42.8$$

<u>Item</u>	<u>Source</u>	<u>Value</u>
V, f.p.s.		34.0
V <sub>a</sub> , f.p.s.	Equation A.	33.0
V <sub>a</sub> <sup>2</sup>		1090.0
Wetted length, in.	Figure 14,	10.6
Reynolds Number		$2.73 \times 10^6$
C <sub>f</sub>	Figure 15.	.00365
Wetted surface, sq. ft.	Figure 14./144	.923

$R_{\text{friction}}$ , lb.		3.6
$R_{\text{total}}$ , lb.	From model results	10.6
$R_{\text{residuary}}$ , lb.		7.0

## Corresponding Values for Full Size

Temperature of water, 70° F. (assumed)

Kinematic viscosity, 0.00001087 ft.<sup>2</sup>/sec.

<u>Item</u>	<u>Source</u>	<u>Value</u>
V, f.p.s.	$V_{\text{model}} \times 1.87$	63.6
$V_a$ , f.p.s.	$V_a \text{ model} \times 1.87$	61.7
$V_a^2$		3807
Reynolds Number	$R.N. \text{ model} \times 6.54 \times \frac{0.00001054}{0.00001087}$	$1.74 \times 10^7$
$C_f$	Figure 15.	.00265
Wetted surface, sq.ft.	$W.S. \text{ model} \times 12.24$	11.31
$R_{\text{friction}}$ , lb.		112.6
$R_{\text{residuary}}$ , lb.	$R_{\text{residuary}}(\text{model}) \times 42.8$	300
$R_{\text{total}}$ , lb.		413

For the model with projecting rivet heads no such direct method of separation was found to be possible. In this case, the surface may be considered neither as a smooth surface nor as 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 to penetrate considerably the laminar sub-boundary layer next the surface, the surface resistance will be essentially due to mass effects and the friction coefficient for any given length-roughness ratio will be essentially constant, i.e., Froude's law will hold in entirety. (See references 8 and 9.) However, the riveted surface represents a compromise between this condition

and that of a smooth surface. Its resistance is made up partly of dynamic pressure on the rivet heads and the remainder of shear in the boundary layer on the surface between the rivets. Both of these effects are interrelated and unfortunately neither theory nor experimental data seems to be far enough advanced at this time to permit of relative evaluation. In the hope that possibly a regular variation of friction coefficient with Reynolds Number at constant roughness ratios might be deduced from the test data, frictional resistance for the model with rivet heads was computed for different Reynolds Numbers at the same wetted lengths, the identical lengths being obtained at different loads and speeds. This computation was performed by subtracting the corresponding residuary resistances of the smooth model from the totals for the model with rivet heads, the remainder being assumed to be frictional resistance. The frictional resistances obtained seemed to show no regular variation whatsoever. There remained, therefore, nothing to do but to extrapolate the resistance of the model with rivet heads entirely according to Froude's law. However, it is to be emphasized that such an extrapolation is much more nearly the true extrapolation for a riveted model than it is for the usual smooth model because, as previously stated, hydrodynamically the riveted model represents a compromise between a smooth surface and a rough surface, and, for a rough surface of such magnitude, Froude's law would hold quite rigidly.

The results of the various take-off calculations are plotted against speed in figure 17, together with the computed thrust at full power and three-fourths power. Generally, the riveted surfaces cause 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 in the total resistance of the smooth model obtained by the two methods of calculation 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 in this figure 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	

Comparing the full power values, using the performance of the smooth hull according to the separation method as a standard, the riveted hull requires 13 percent more take-off time and 16 percent longer take-off run. Using the performance of the smooth hull according to Froude's law 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 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. Now, it is to be remembered that these values are for roundhead rivets. For the brazier type of head more commonly used, lower augmentations probably on the order of  $\frac{2}{3}$  might be used. Therefore, one could reasonably conclude that with the usual riveted hull of about the same size and with comparable propeller thrust, one might expect about 7 to 8 percent more take-off time and 9 to 10 percent longer take-off run than for a smooth hull. But, if we assume take-off with the same propeller but at three-fourths power, we find that the relative effect of the roughness of the hull has been 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, computed 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 augmentation in take-off time would be 14 percent and the augmentation 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 moderate reserve thrusts such as are typical of most flying boats.

## CONCLUSIONS

1. The percentage augmentation in the water resistance of the model caused by the projecting rivet heads, laps, and keel bar is fairly constant over a wide range of loads. It varies from 10 to 12 percent at the hump speed and from 12 to 14 percent at 45 feet per second.

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

3. The increase, caused by the excrescences with 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., June 4, 1936.

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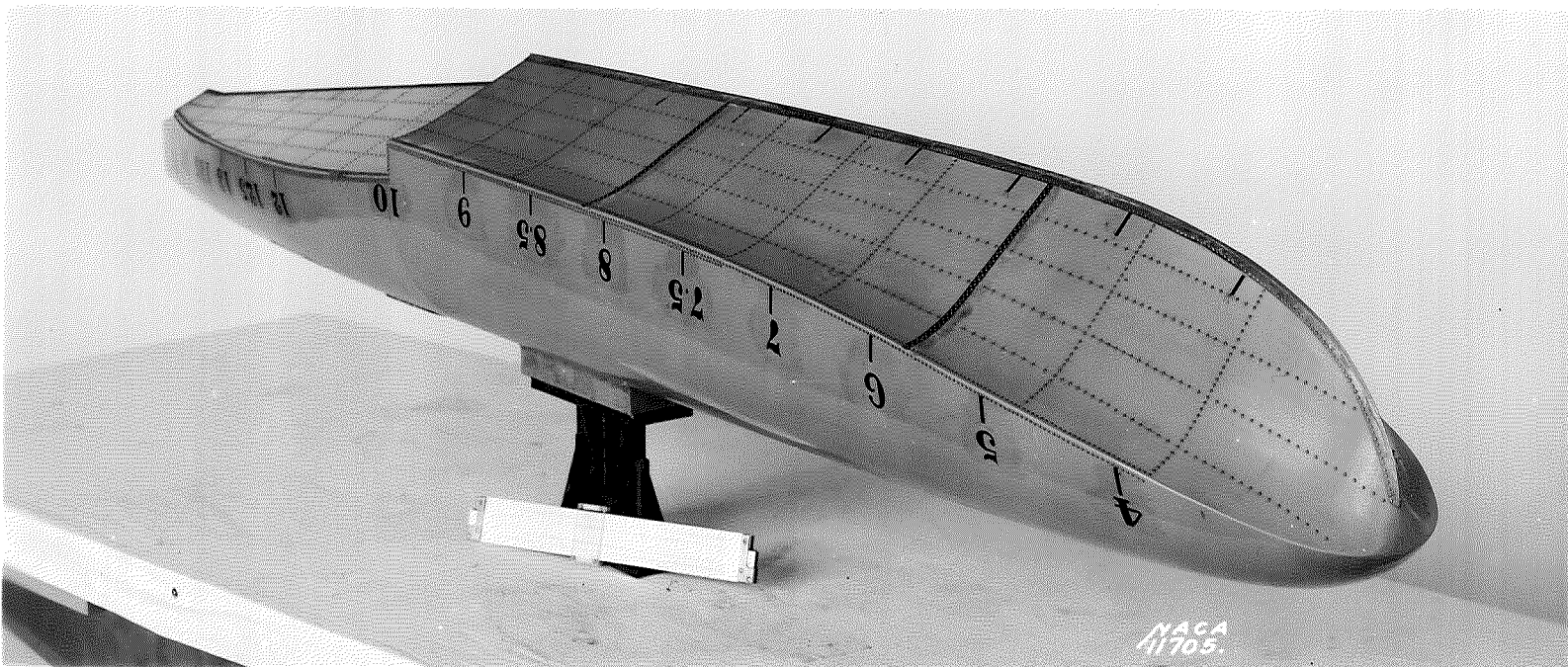
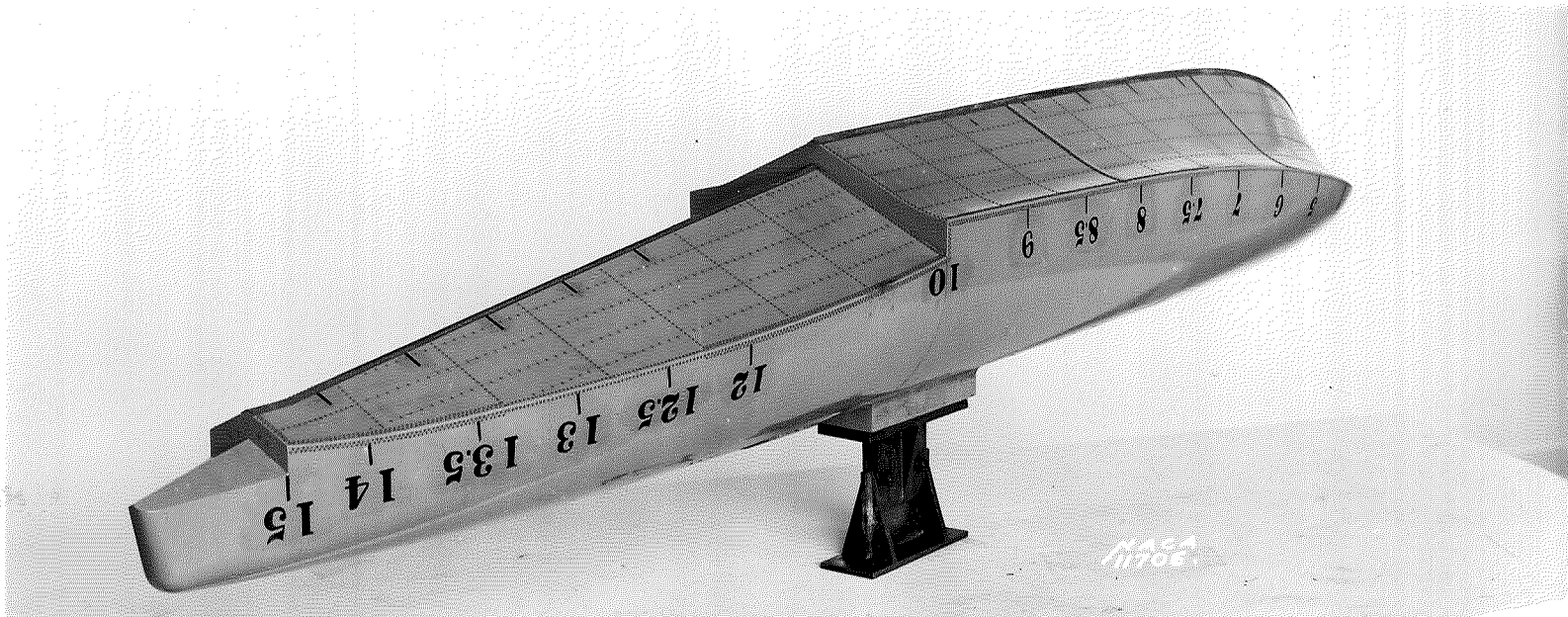
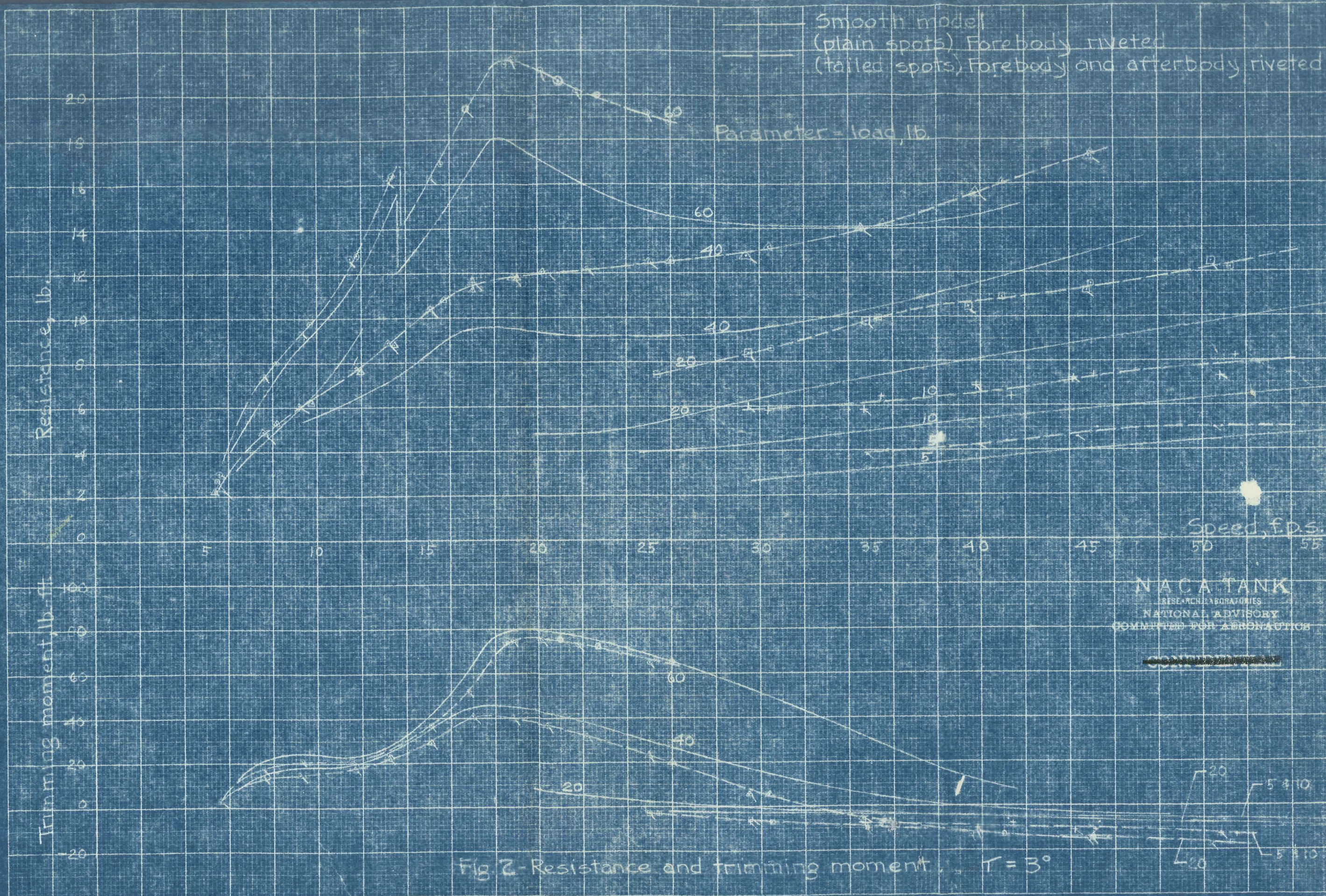


Figure 1. Photographs of model airfoil for reproduction of riveted surfaces.

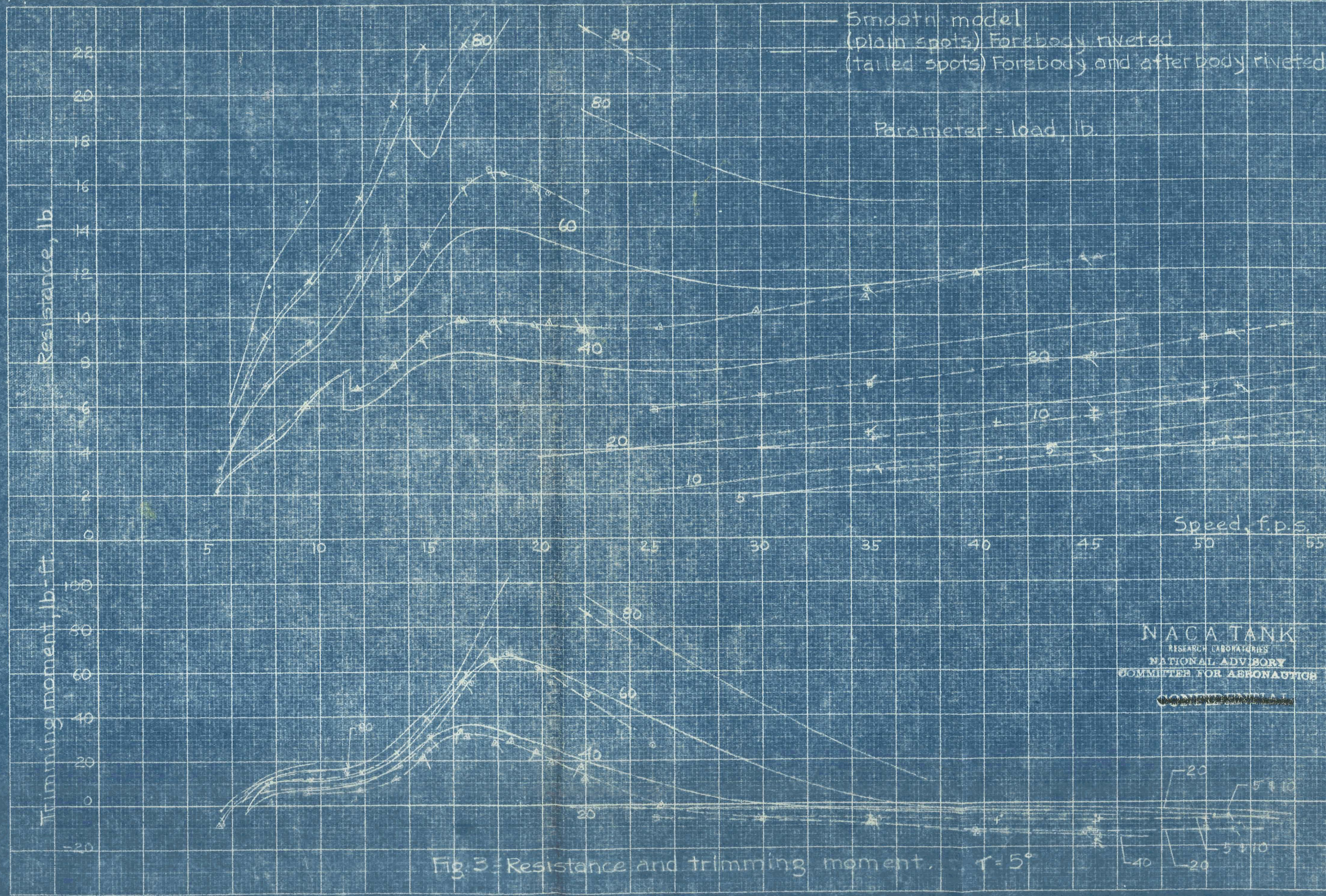
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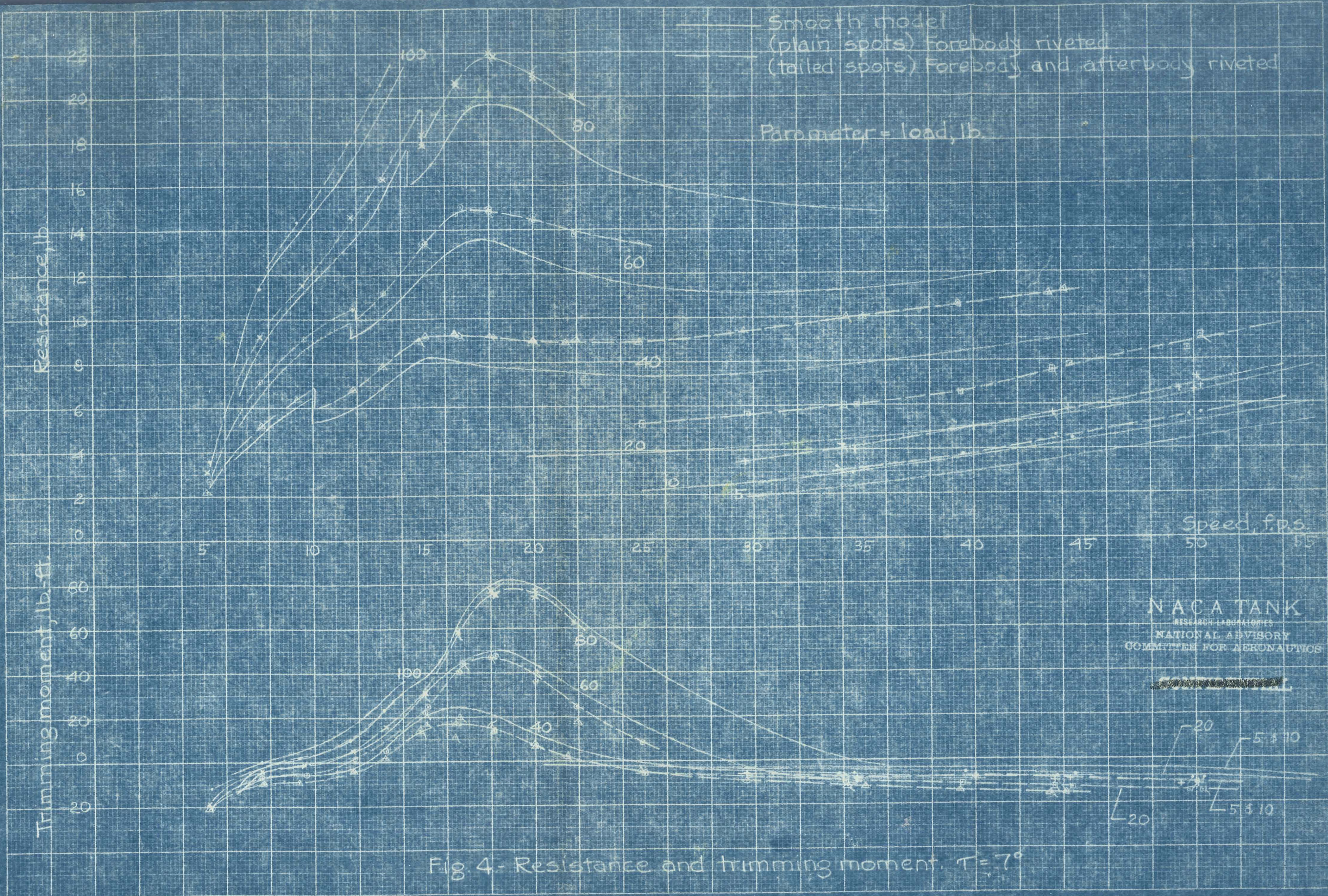


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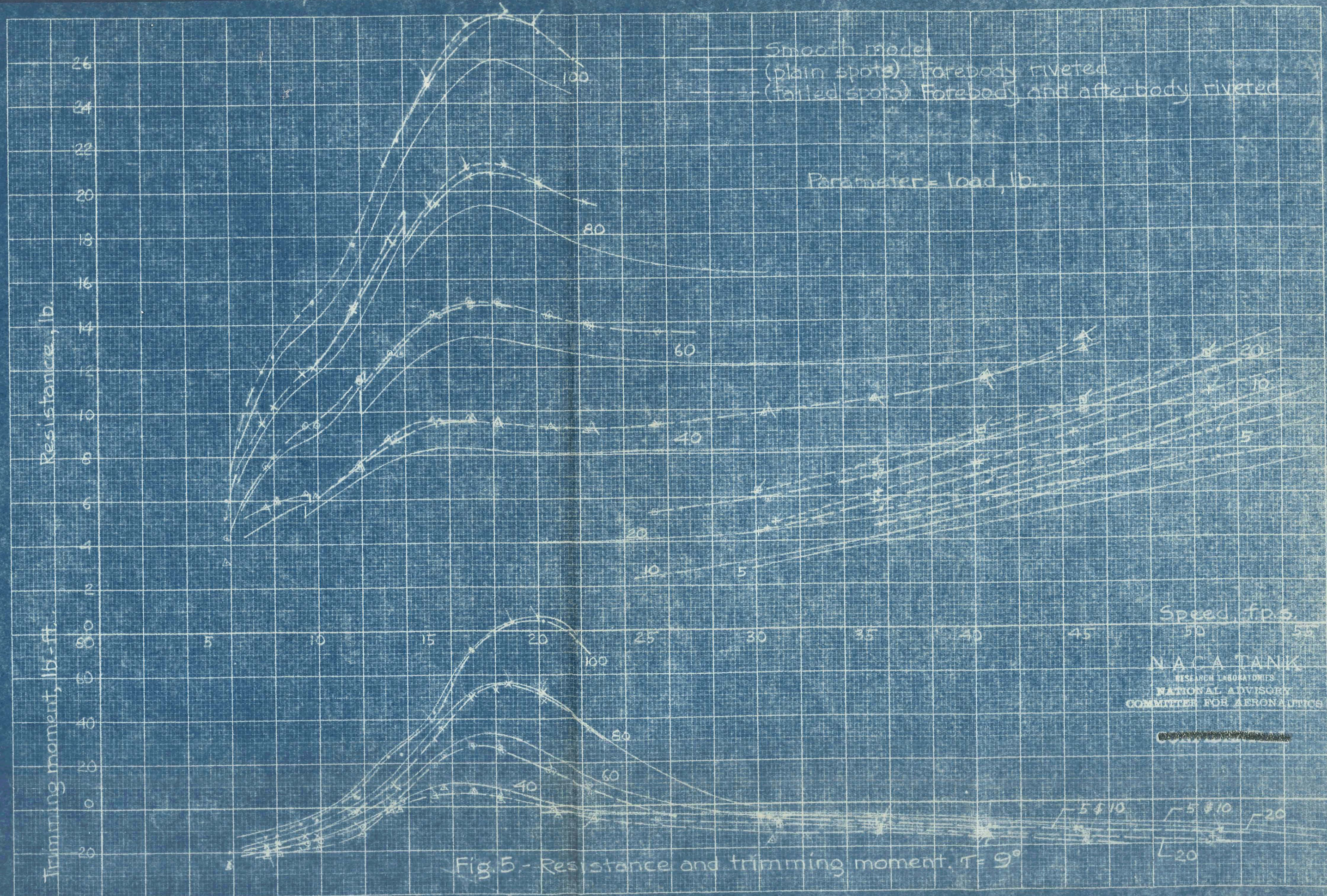


Fig. 5 - Resistance and trimming moment,  $T = 9^\circ$

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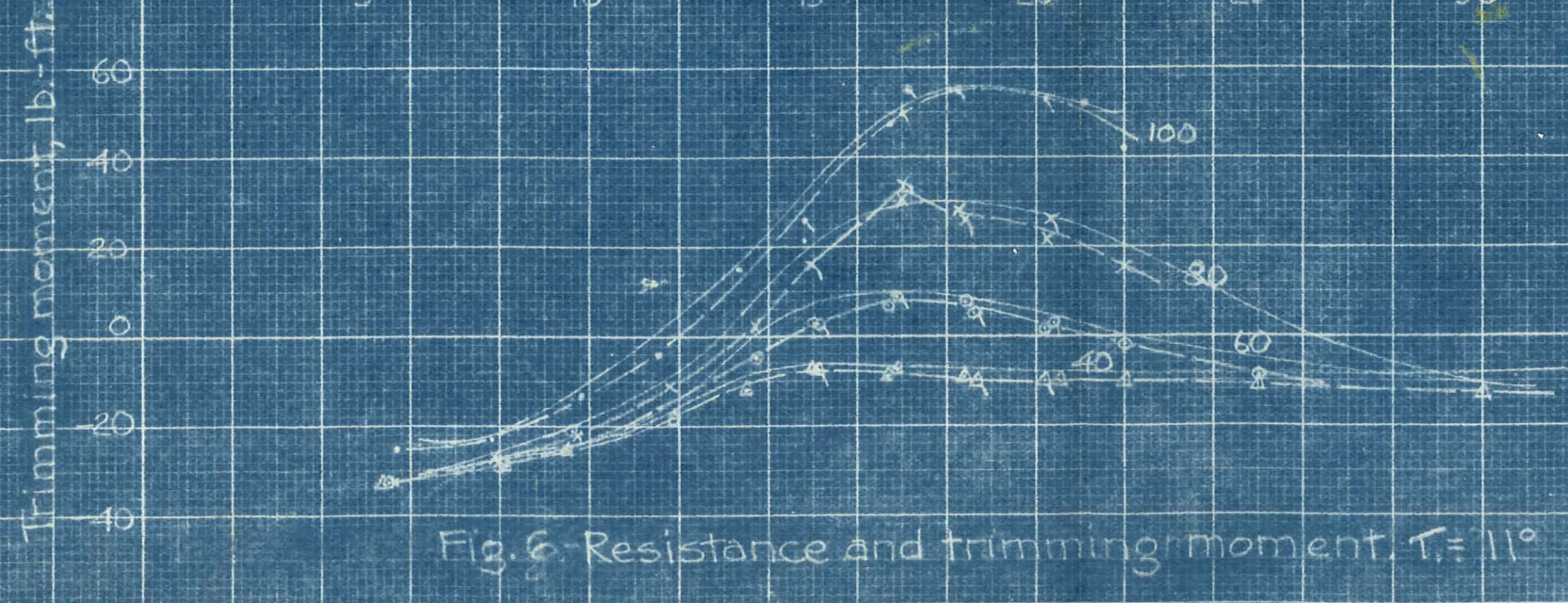
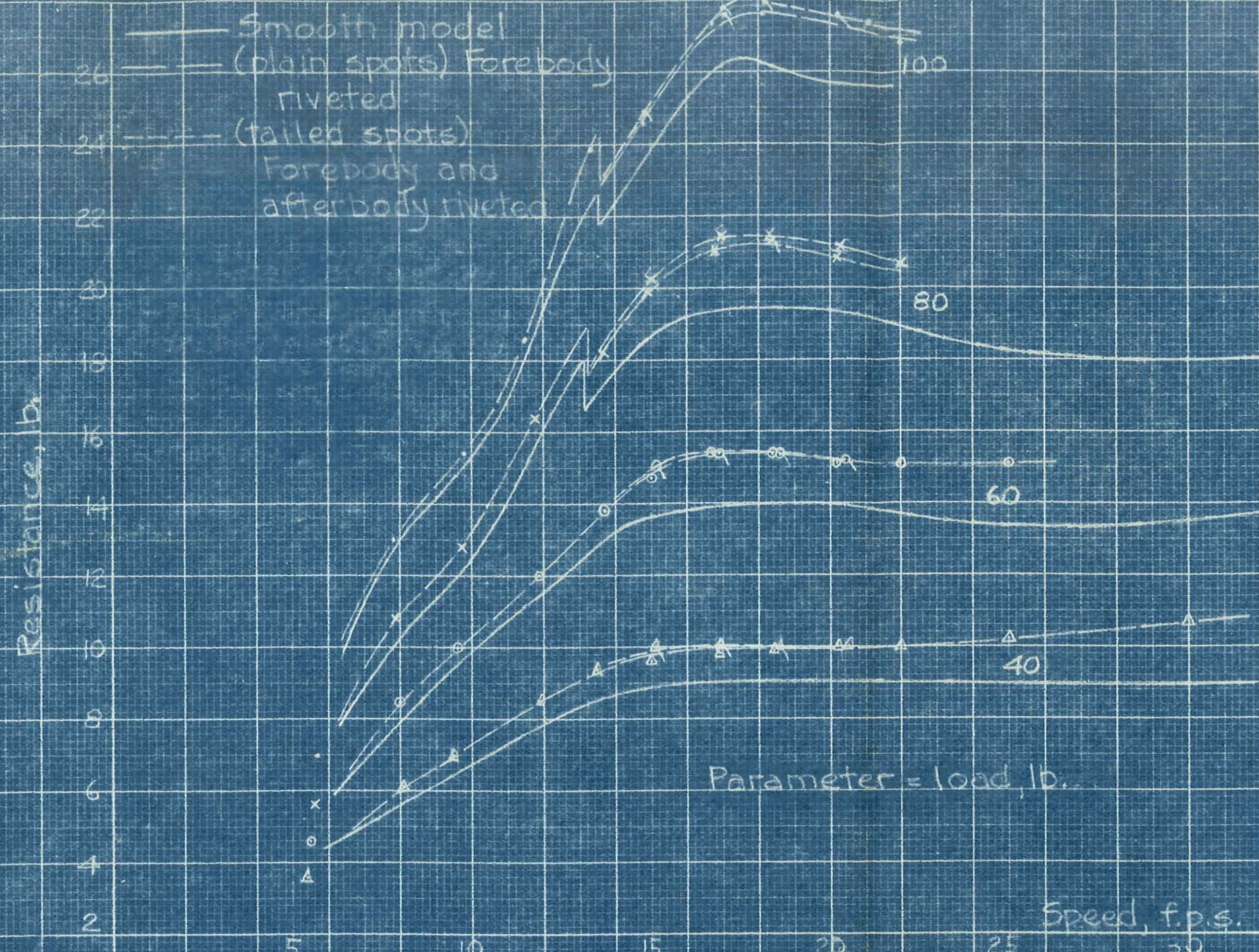


Fig. 6 - Resistance and trimming moment,  $T=11^\circ$

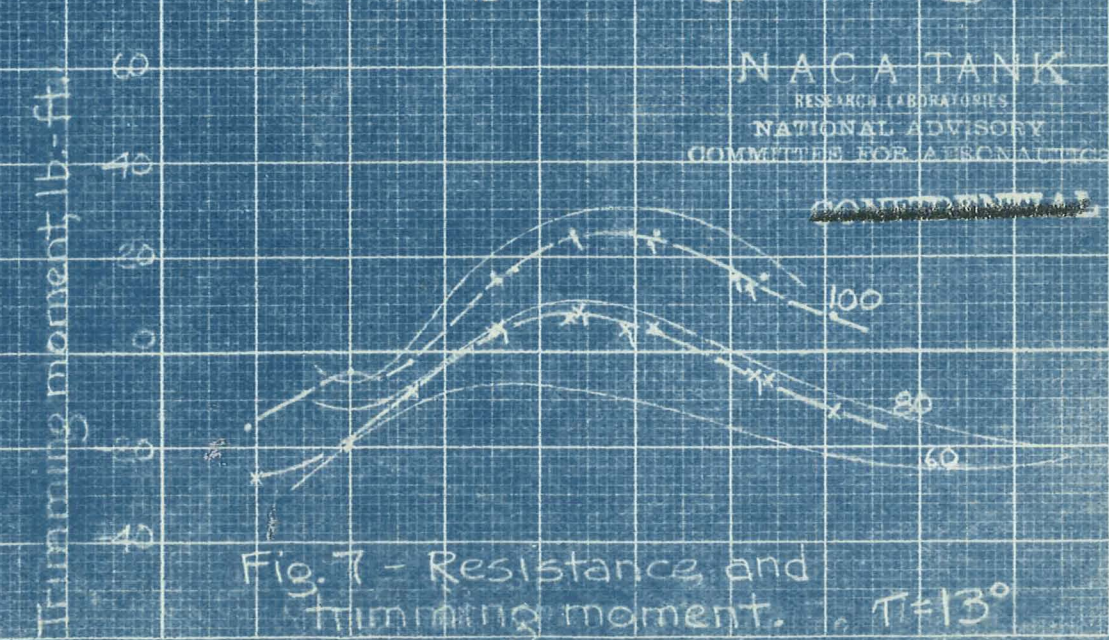
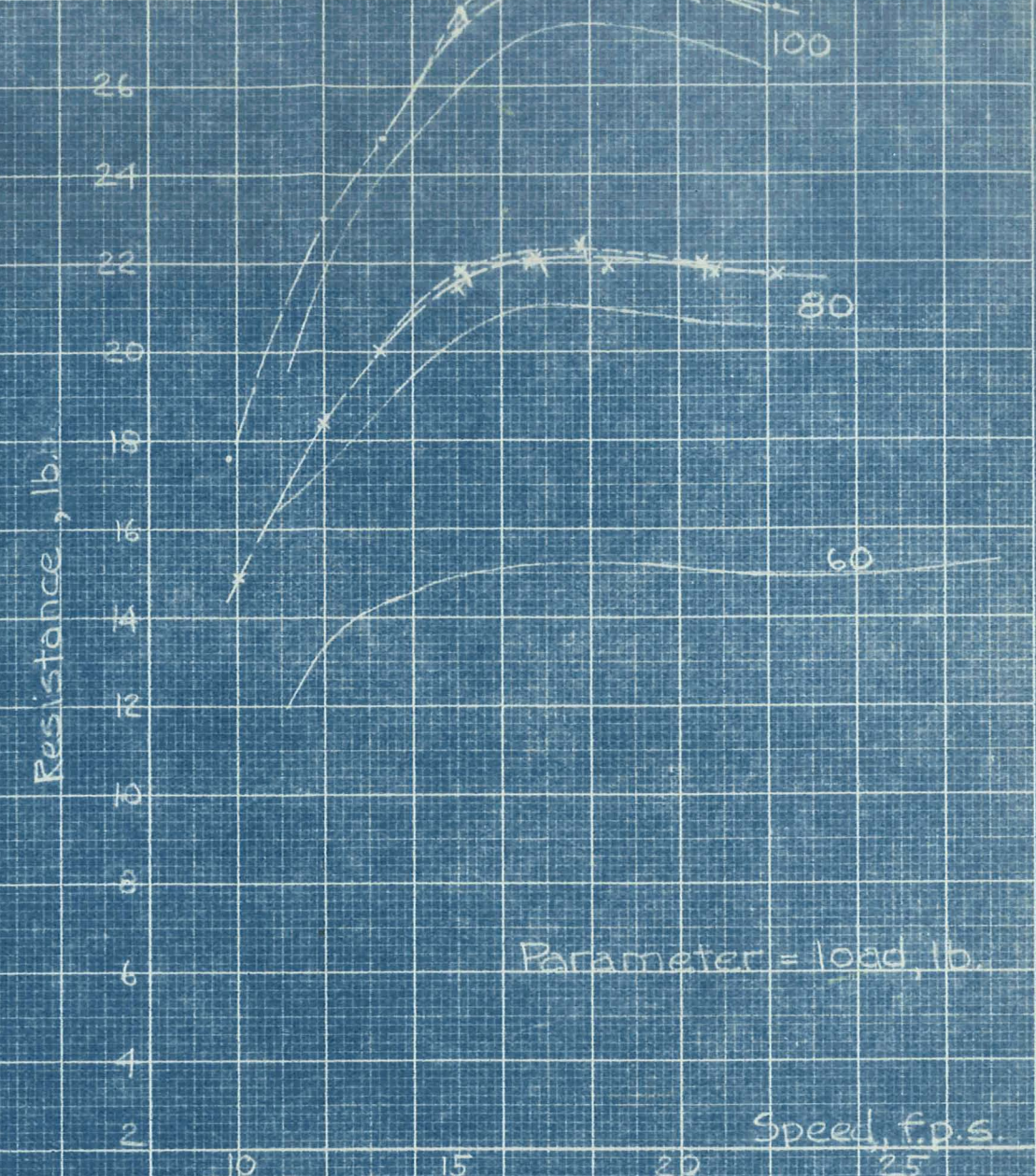


Fig. 7 - Resistance and trimming moment,  $T=13^\circ$

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Forebody wetted length, in

Aft body wetted length, in

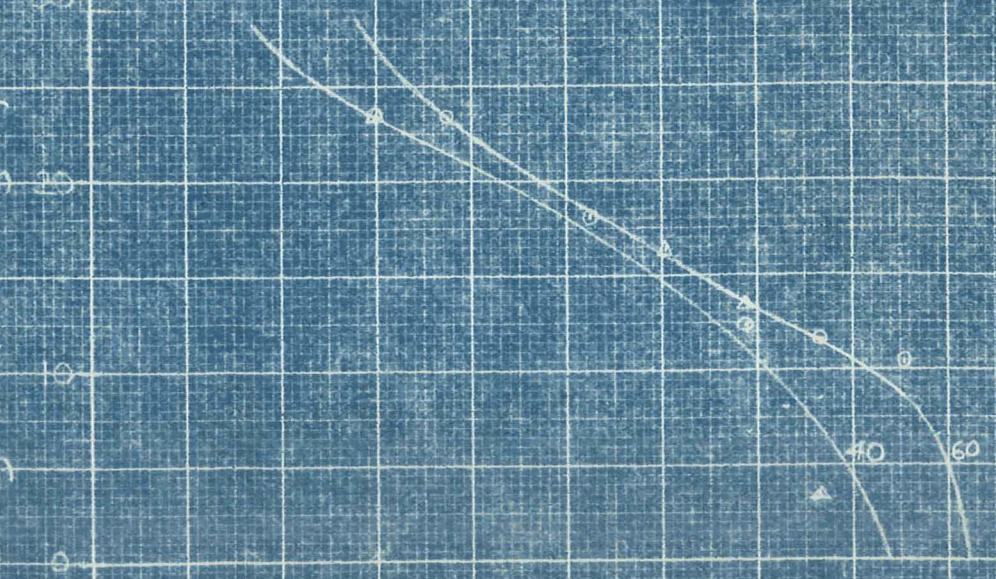
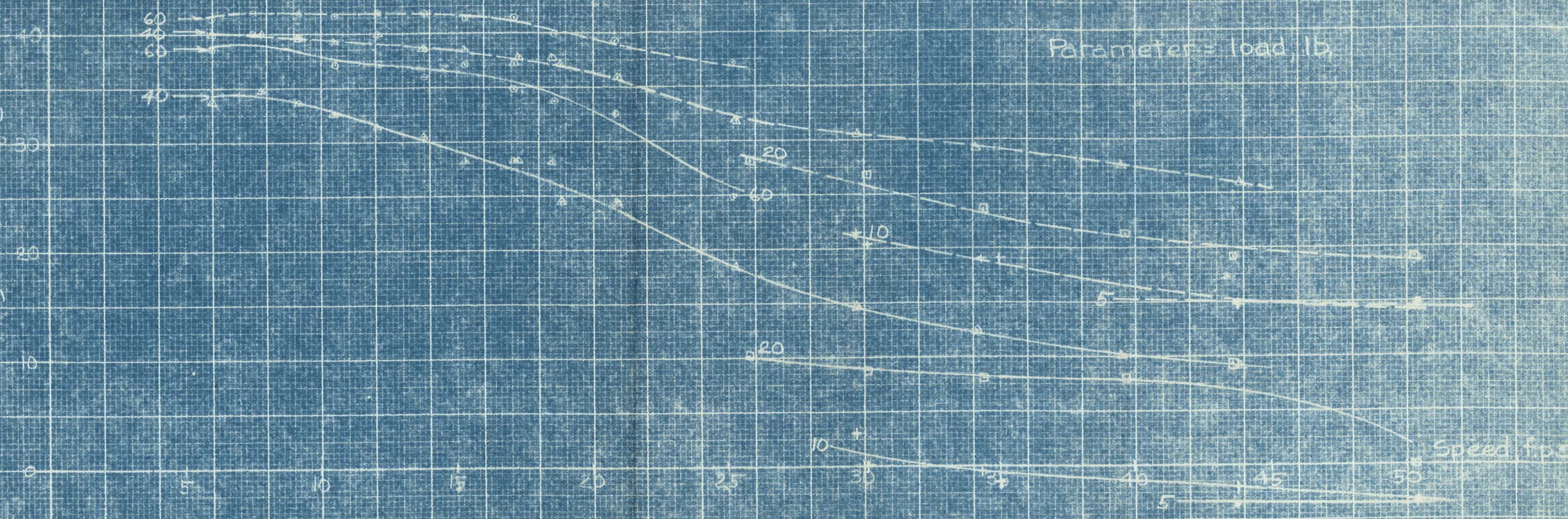


Fig 8 - Wetted lengths,  $T=3^\circ$

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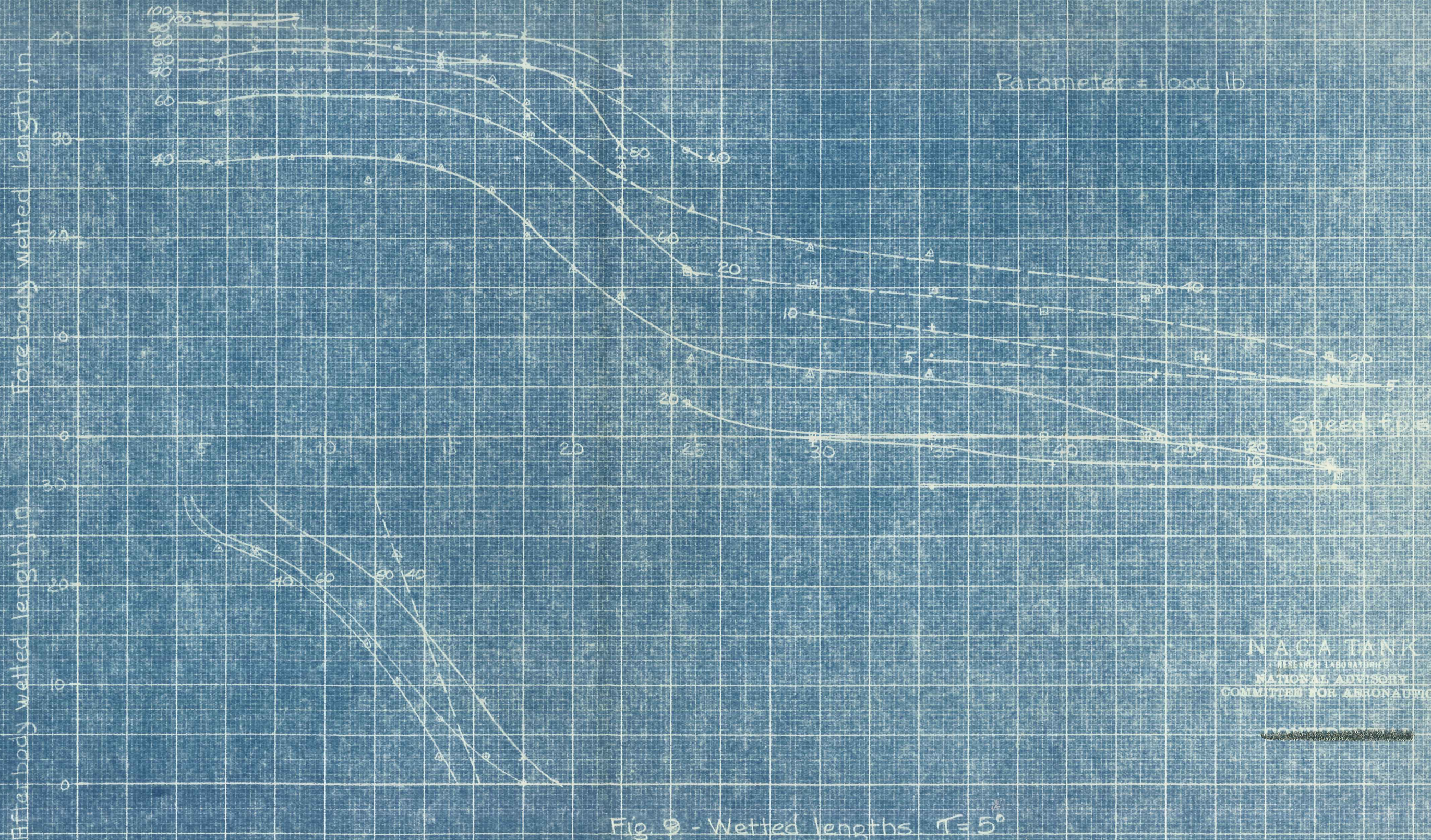


Fig. 9 - Wetted lengths  $T=5^\circ$

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Forebody wetted length, in.

Afterbody wetted length, in.

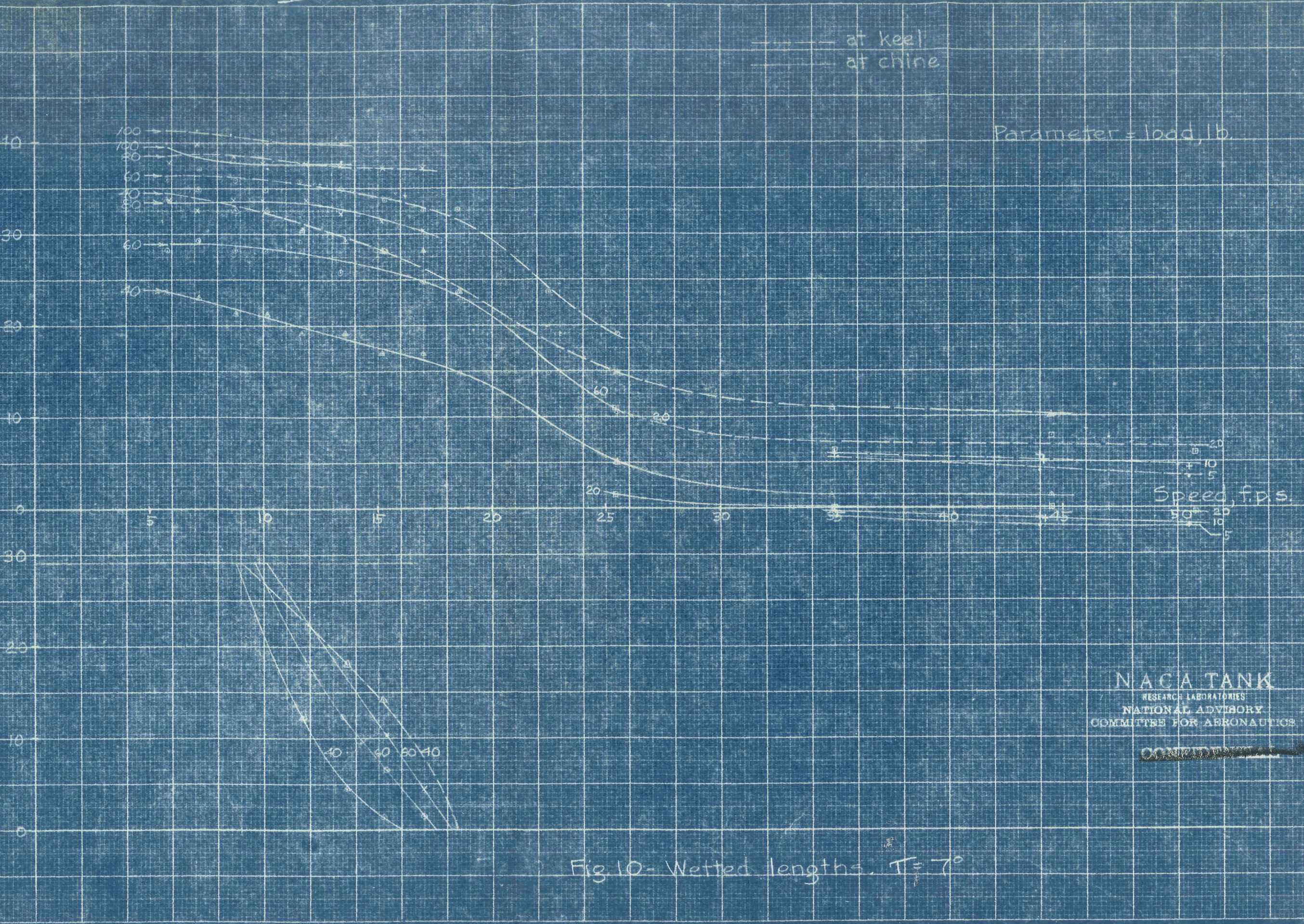


Fig 10- Wetted lengths.  $T = 7^\circ$

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Forebody wetted length, in

Afterbody wetted length, in

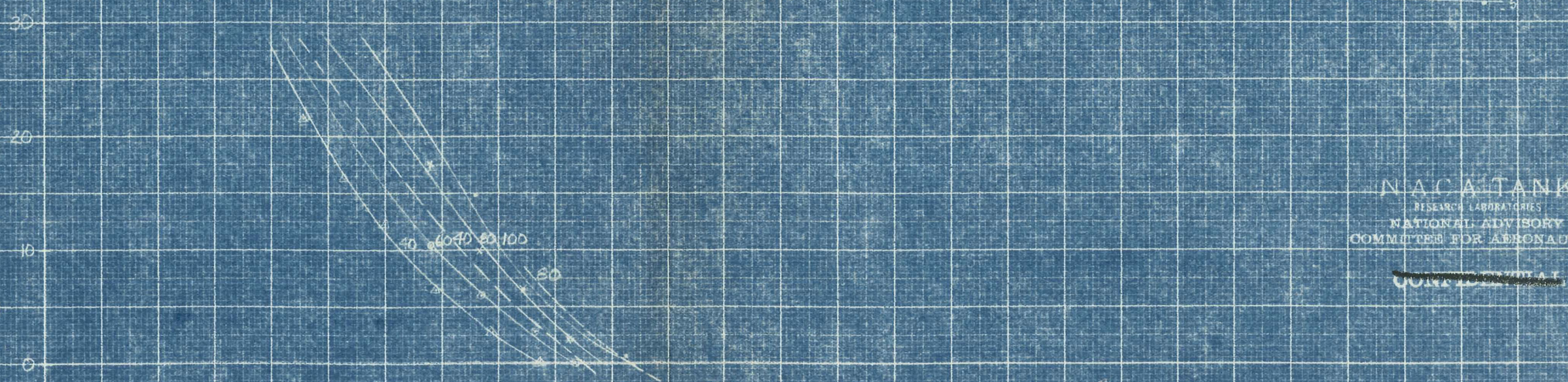
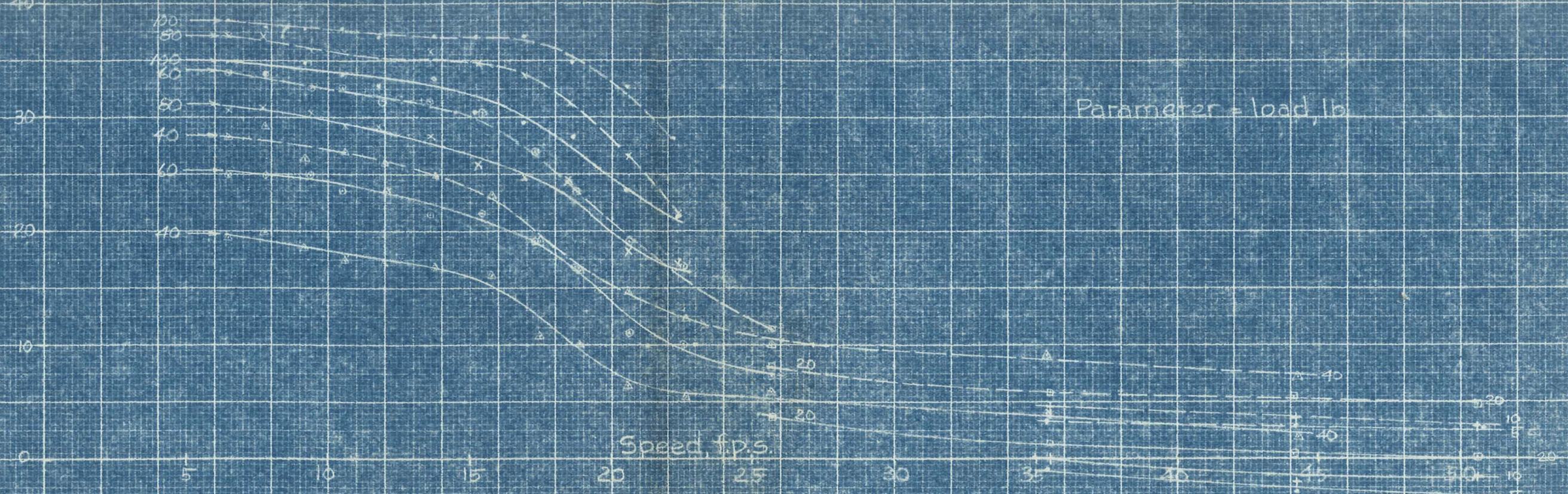


Fig 11 - Wetted lengths  $\tau = 9^\circ$

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ENGRAVING 334-3, 10 X 10 TO THE HALF INCH.  
WHEN ORDERING STATE COLOR GRAMING OR TRACING PAPER OR CLOTH  
PRINTED IN U. S. A.  
100% PAU PAPER

Forebody wetted length, in.

After-body wetted length, in.

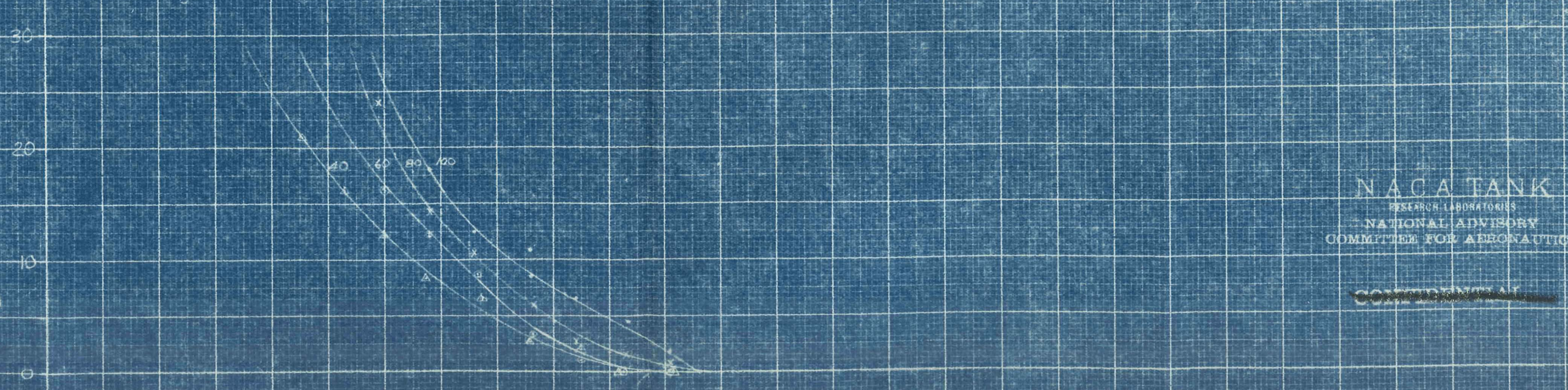
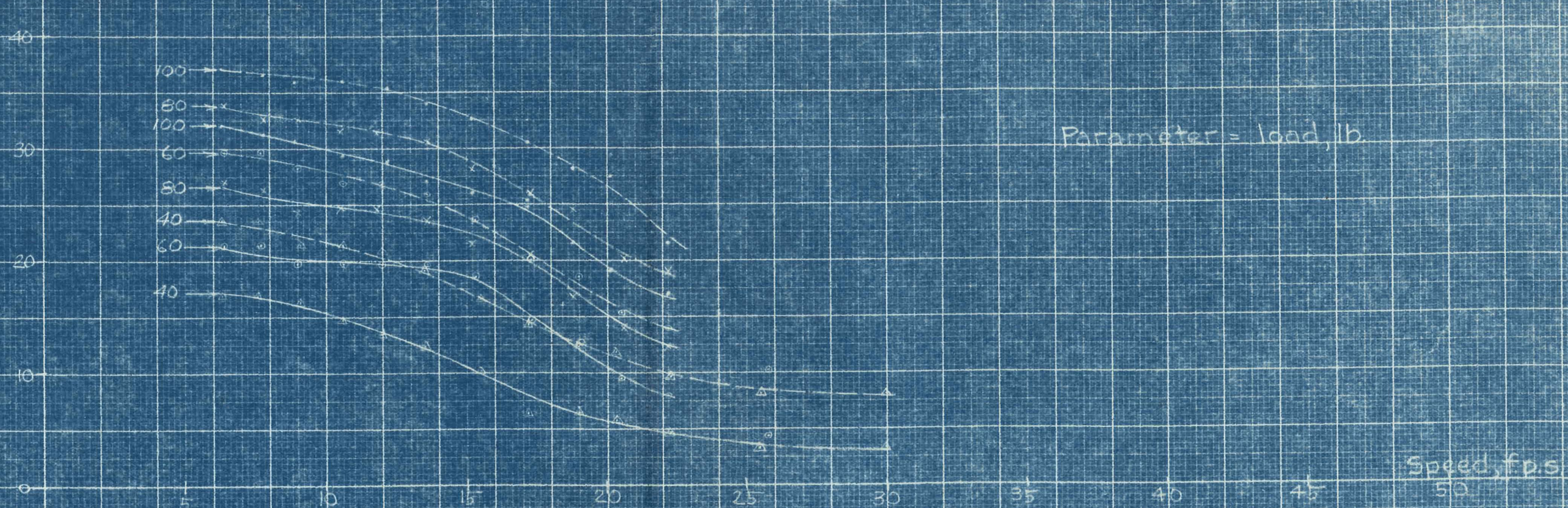


Fig 17 - Wetted lengths.  $\tau = 11^\circ$

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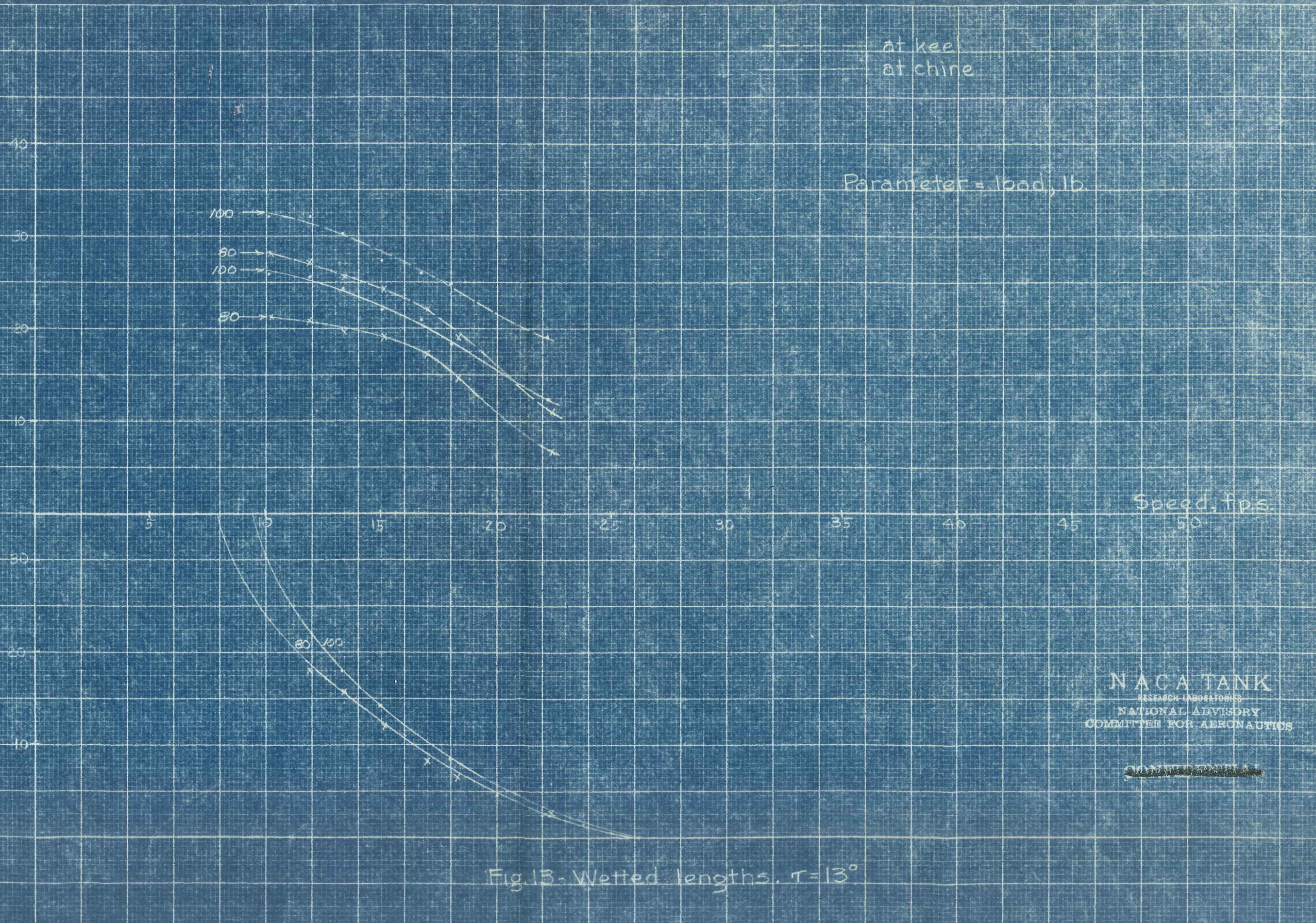
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ENGRAVING 334-3. 10 X 10 TO THE HALF INCH.  
WHEN ORDERING STATE COLOR DRAWING OR TRACING PAPER OR CLOTH,  
PRINTED IN U. S. A.,  
100X GRA PAPER

Forebody wetted length, in

Afterbody wetted length, in

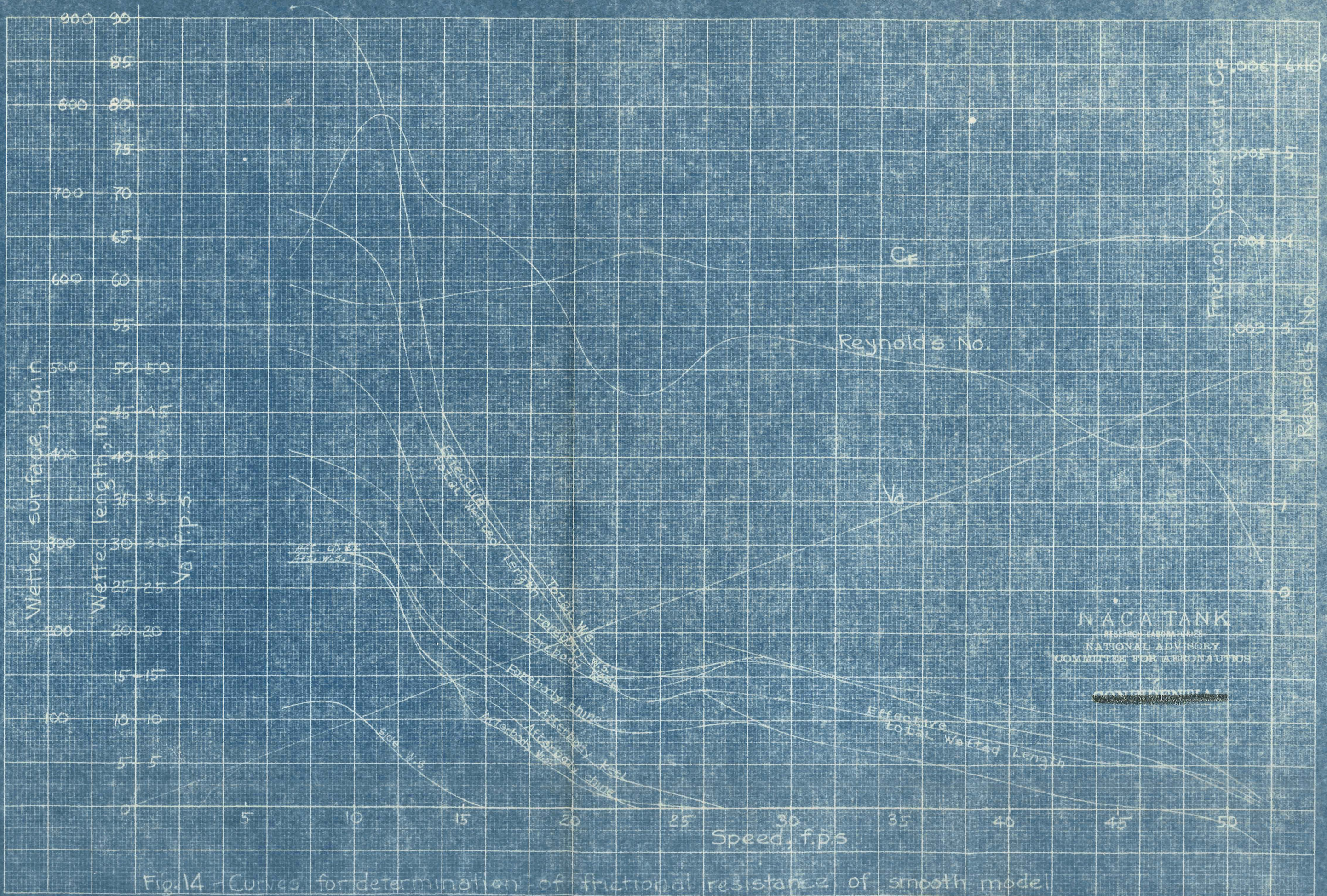


at keel  
at chine  
Parameter = load, lb

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Fig. 13 - Wetted lengths,  $\tau = 13^\circ$

ENGRAVING 334-3 10 X 10 TO THE HALF INCH.  
 WHEN ORDERING STATE COLOR DRAWING OR TRACING PAPER OR CLAM  
 PRINTED IN U. S. A.  
 100% MIL WARE

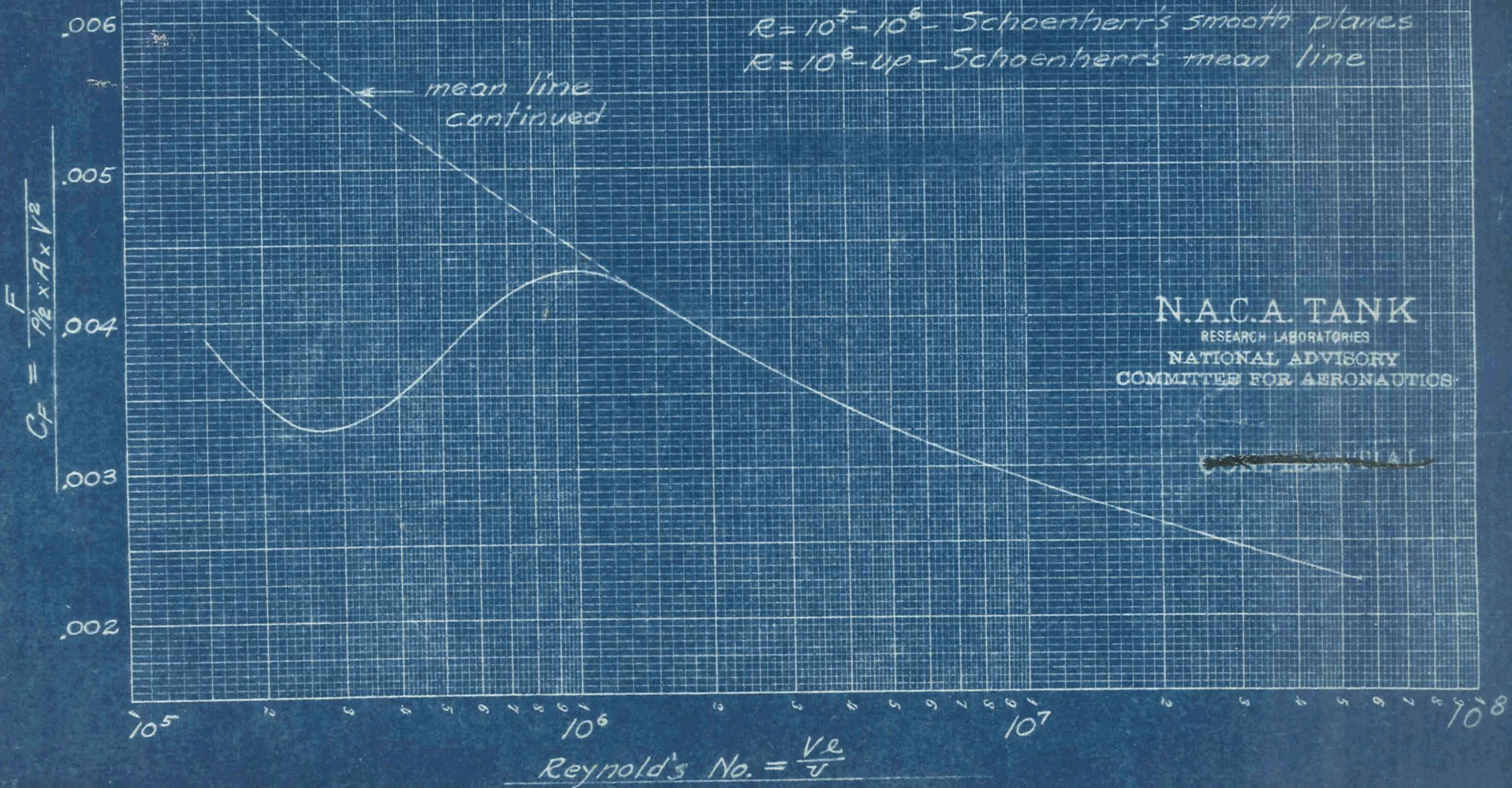


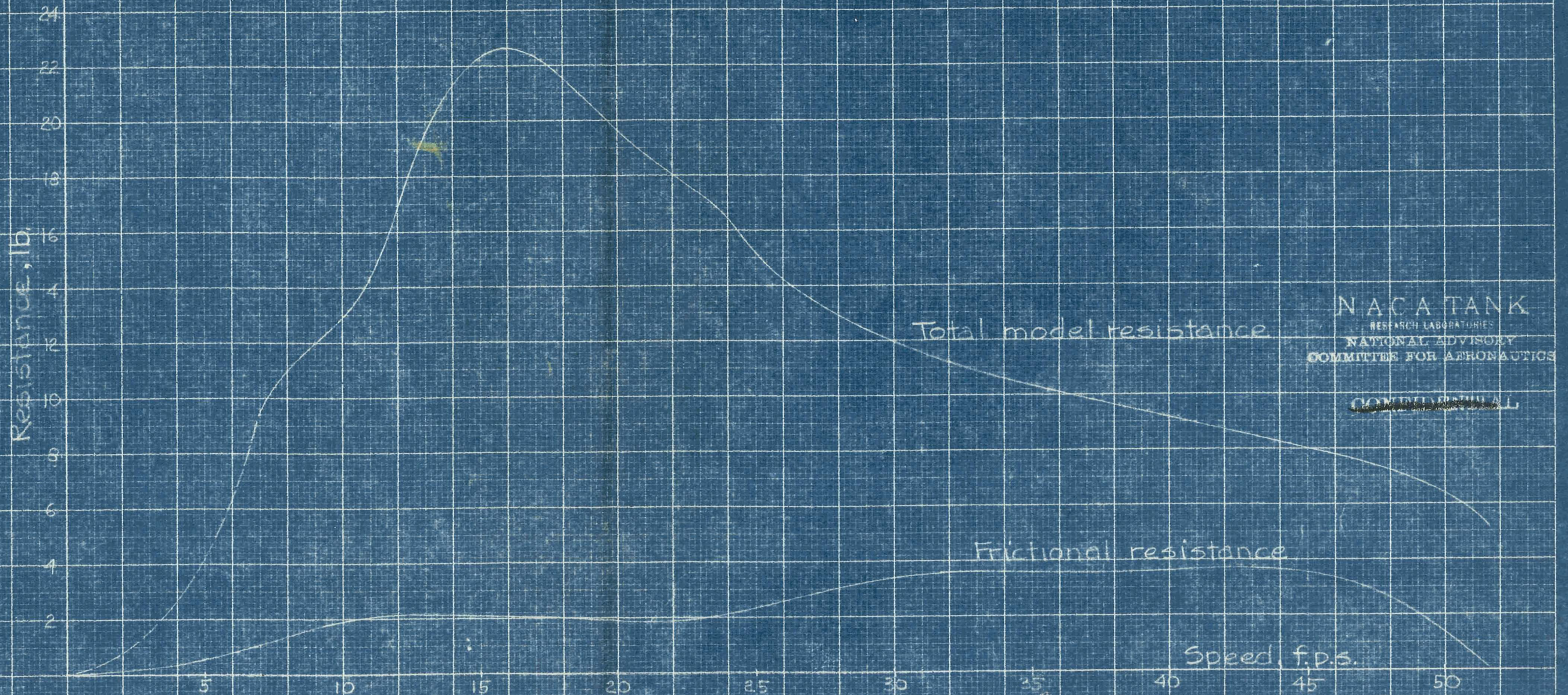
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Fig. 14 - Curves for determination of frictional resistance of smooth model

Fig. 15 - Frictional Coefficient plotted against Reynold's number.

R = 10<sup>5</sup> - 10<sup>6</sup> - Schoenherr's smooth planes  
R = 10<sup>6</sup> - up - Schoenherr's mean line





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

ENGRAVING 334-3 10 X 10 TO THE HALF INCH  
REPRODUCING STATE COLOR DRAWING ON TRACING PAPER OR CLOTH  
PRINTED IN U. S. A.  
100X 745-1000

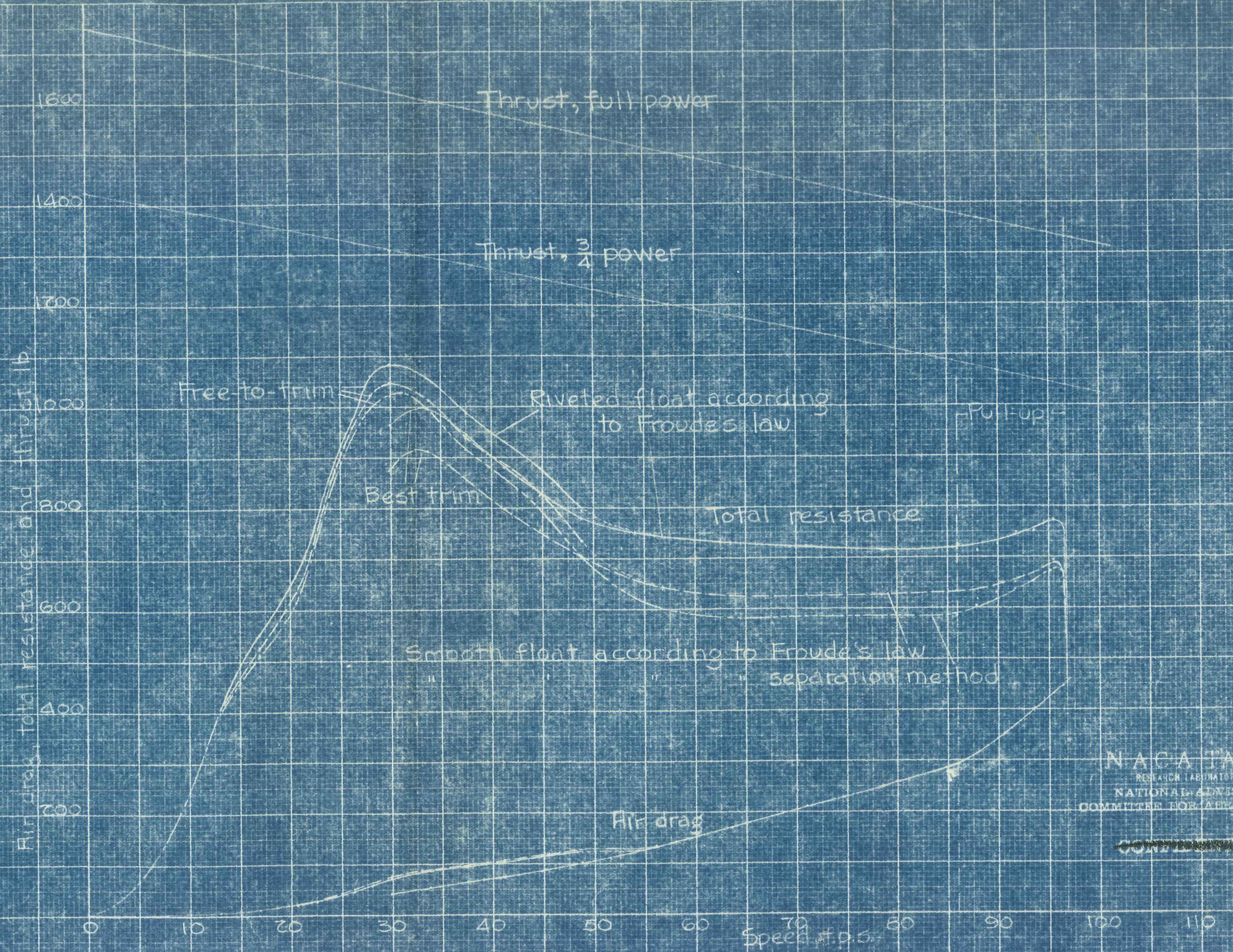


Fig 17- Effect of rivets on take-off performance.

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ENGRAVING 334-3, 10 X 10 TO THE HALF INCH  
WHEN ORDERING STATE COLOR DRAWING CONTRACT NO. MAKE OR CLOTH  
PRINTED IN U. S. A.  
100% MAG. HAYER

Trimming moment at best trim, lb-ft.

10000  
8000  
6000  
4000  
2000  
0  
-2000  
-4000

Trim, deg.

14  
13  
12  
11  
10  
9  
8  
7  
6  
5

10 20 30 40 50 60 70 80 90 100

Speed, f.p.s.

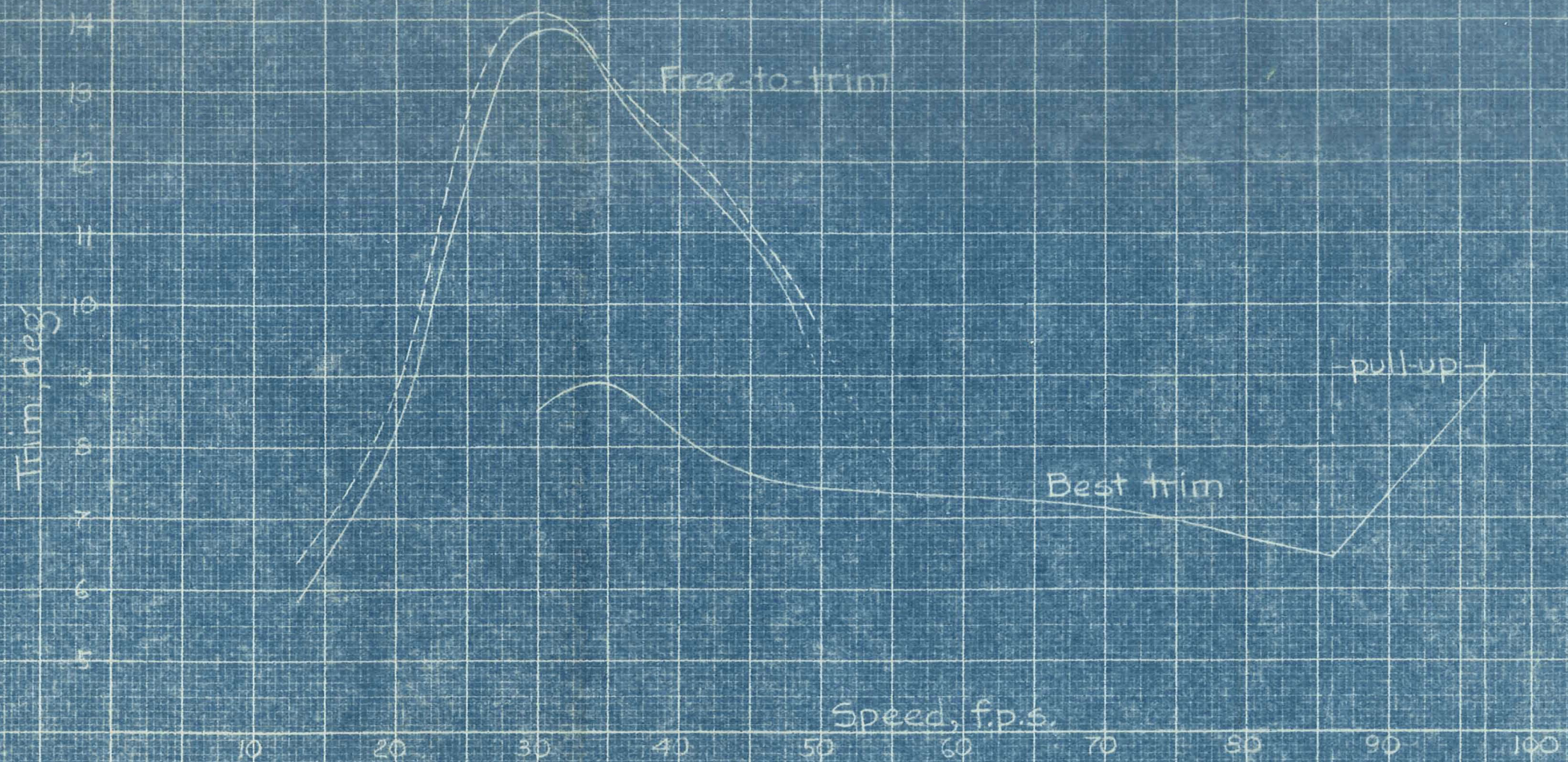


Fig. 17 - Cont'd

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