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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1292

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INVESTIGATION OF SUCTION-SLOT SHAPES FOR CONTROLLING A TURBULENT BOUNDARY LAYER

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SUMMARY

Tests of three types of boundary-layer-control suction slots have been made in a two-dimensional diffuser to investigate design criterions and to evaluate the practical minimum total-pressure losses. The tests were conducted at a velocity of about 100 feet per second with a boundary layer which had a displacement thickness of 0.85 inch and a shape parameter of about 1.8.

The shape of the boundary layer behind the slot was found to depend only on the quantity of air removed provided that the slot inlet had rounded edges. Near maximum effectiveness was obtained when the quantity rate of air flow through the slot was equal to that which would pass at free-stream velocity through an area equal to the displacement thickness per unit span.

The total-pressure losses through the slot were found to be appreciably reduced by rounding the inlet edges, inclining the slot, slightly diverging the slot walls, and, especially, providing adequate width. The optimum inlet-velocity ratio for a diffuser slot is of the order of 0.60 to 0.65. For the foregoing rate of air flow and with a round-edge diffuser slot inclined at 30° to the air stream, the total-pressure drop was 48 percent less than the value for a normal-opening sharp-edge slot. For this configuration only 55 percent of the measured total-pressure drop could be accounted for by the total-pressure deficiency in the part of the boundary layer removed.

INTRODUCTION

Boundary-layer control by suction, as a means of preventing flow separation on wings and in ducts, has been the subject of a great deal of experimental study; for example, see references 1 and 2. The power required for effective boundary-layer control was determined in many of these studies; however, most such power requirements must be considered unnecessarily high and hardly indicative of the power requirements

for optimum designs because of the excessive pressure losses through the usually arbitrarily designed suction slots. Obviously, if the losses through the suction slots can be minimized, the net difference between the free-stream total pressure and the total pressure in the suction duct need not greatly exceed the losses already present in the boundary layer that is being removed.

In the present work measurements were made of the additional losses through suction slots of various designs in order to develop design criterions for suction slots and to evaluate the practical minimum value of such additional pressure losses. Two-dimensional slots of various widths and entrance radii, flush and inclined at several angles to the surface and with various amounts of angular separation between the two walls, were tested. Only one boundary layer - one with a displacement thickness of about 0.85 and with a shape parameter of about 1.8 - was used for the tests.

SYMBOLS

U local velocity outside boundary layer, feet per second

q local dynamic pressure outside boundary layer, pounds per square inch

u local velocity inside boundary layer, feet per second

H_1, H_2 total pressure at stations 1 and 2 respectively, pounds per square foot

Q quantity rate of flow through suction slot, cubic feet per second

y distance normal to surface, inches

b span of suction slot, inches

w width of suction slot, inches

R_1 radius of front edge of suction slot, inches

R_2 radius of rear edge of suction slot, inches

δ^* boundary-layer displacement thickness, inches $\left(\int_0^\delta \left(1 - \frac{u}{U}\right) dy\right)$

θ boundary-layer momentum thickness, inches $\left(\int_0^\delta \left(1 - \frac{u}{U}\right)\frac{u}{U}\, dy\right)$

δ boundary-layer thickness, inches

H boundary-layer shape parameter $\left(\delta^*/\theta\right)$

c_Q flow coefficient $\left(Q/b\delta^*_1 U_1\right)$

C_{H_b} total-pressure-loss coefficient $\left(\frac{H_1 - H_b}{q_1}\right)$

φ angle of slot center line with respect to test surface, degrees

β diffuser angle, degrees

h distance normal to surface at station 1, which is determined by the amount of boundary layer removed; that is, when the part of the boundary layer between $y = 0$ and $y = h$ at station 1 is removed, inches

$\overline{H}$ mean total pressure of part of boundary layer to be removed, pounds per square foot

ΔH_s total-pressure loss through suction slot, pounds per square foot

Subscripts

b conditions in suction chamber

1 conditions at station 1, 5 inches ahead of center line of suction slot

2 conditions at station 2, 4 inches behind center line of suction slot

APPARATUS AND MODELS

The tests were conducted on a flat wall of a two-dimensional diffuser which was attached to the entrance cone of the $\frac{1}{15}$-scale model of the full-scale wind tunnel described in reference 3. Figure 1 is a diagrammatic sketch of the principal parts of the apparatus used in tests of boundary-layer-control suction slots.

The top and bottom of the diffuser and the side of the diffuser on which the slots were located were flat; the side opposite the test wall was adjustable. A vane (in the form of an airfoil) and three boundary-layer bleeds on the adjustable wall were used to maintain

nonseparated flow on the adjustable wall. Pressure to force air through the bleeds was obtained by placing a 16-mesh screen at the diffuser exit. The suction chamber (fig. 1) consisted of a large plywood box. A 100-mesh screen located 3 inches from the back of the box served to eliminate any local excesses of suction near the center of the box, where the suction duct was attached.

Sketches of the three basic boundary-layer control slots (designated types I, II, and III), which completely spanned the test wall, are shown in figure 2. The interchangeable inserts (fig. 1) which formed the slots were constructed of mahogany and were lacquer-finished to within 0.01 inch of the specified dimensions (fig. 2). Sheetmetal end plates were provided to close the ends of the slots and to assist in the adjustment and alinement of the slots. All surface breaks were sealed after the slot was installed on the test wall.

Measurements of the pressures in the boundary layer were made with the rake shown in figure 3. The tubes of the rake were connected to a multiple-tube manometer, and the pressures were recorded by means of a camera. A total-pressure and a static-pressure tube outside the boundary layer were used to measure the free-stream total and dynamic pressures ahead of the slot. The average total pressure in the suction chamber was determined from four static orifices on the walls of the chamber, connected in parallel to a micromanometer. The rate of air flow through the slot was determined from a calibrated total-static-pressure tube located in the suction duct and connected differentially to a second micromanometer. The calibration was made with an eleven-tube rake located in the duct between the suction chamber and the blowers. Quantity rate of air flow was regulated by two butterfly valves, one in the main duct and the second in a by-pass duct.

TESTS

Preliminary tests were made, by use of tufts, to adjust the inclined wall and its three boundary-layer bleeds and the auxiliary vane in order to prevent flow separation on the inclined wall. Separation of the flow from the top wall or the bottom wall did not occur when the air flow adhered to the inclined wall. Several spoiler rods were then placed upstream of the suction slots in the region of maximum velocity; careful adjustment of these rods resulted in the formation of a thick, turbulent boundary layer at the suction-slot location. Further minor adjustments of the spoiler rods were necessary to obtain spanwise uniformity of the boundary layer.

With several different slots in position, tests were made for a range of rate of air flow up to 20 cubic feet per second to verify the uniformity of the total pressure in the suction chamber. Since the main tests were run with the rake removed at station 1, preliminary tests were also made to determine the relation between the dynamic pressures at stations 1 and 2 (fig. 1) as a function of the quantity of air removed through the slot and to verify the fact that the relation was not a function of the slot design.

For the main tests simultaneous measurements were made of the boundary-layer total and static pressures at station 2, the average total pressure in the suction chamber, and the quantity rate of air flow through the slot. The following slot configurations were tested:

Type I. Sharp-edge slots with straight parallel sides inclined at angles φ with respect to the test wall of 90°, 60°, 45°, and 30° and with slot widths w of 0.38, 0.63, and 0.75 inch. One test was made for $\varphi = 90^\circ$ and $w = 1.50$ inches.

Type II. Slots similar to type I but with rounded edges and with $R_1 = R_2 = 0.06$, 0.13, 0.25, and 0.38 inch, $\varphi = 90^\circ$, 60°, 45°, and 30°, and $w = 0.75$ inch. An additional test was made for $\varphi = 90^\circ$, $w = 0.75$, $R_1 = 1.50$, and $R_2 = 0.38$ inches.

Type III. Slots with rounded edges and diverging walls (ratio of exit area to entrance area constant and equal to 2) with $R_1 = 1.50$ inches and $R_2 = 0.25$ inch, $\varphi = 90^\circ$, 60°, 45°, and 30°, and $w = 0.75$, 1.25, and 1.75 inches (except $\varphi = 30^\circ$ for which $w = 0.75$, 1.13, and 1.50 inches). The larger values of w were included in these tests after it became clear that the smaller values could result in very large losses at the higher flow coefficients; it must be admitted, however, that such large slots in a wing surface may present difficult design problems. Diffuser angles β of 12°, 18°, and 24° were tested for each combination of slot angle and slot width. One test was also made for $\varphi = 45^\circ$, $w = 0.75$ inch, and $\beta = 6^\circ$.

The tests were made at a velocity outside the boundary layer of about 100 feet per second with quantity rates of air flow through the slots up to about 20 cubic feet per second. The turbulent boundary layer at the slot was approximately 3 inches thick and had values of displacement thickness δ^* and shape parameter H of about 0.85 inch and 1.8, respectively. The Reynolds number based on the momentum thickness R_θ was approximately 25,000.

RESULTS AND DISCUSSION

Preliminary tests showed that the displacement thickness δ^* and the shape parameter H of the initial boundary layer at station 1 ahead of the slot remained constant within 5 percent for the entire range of air-flow rate tested. With the slot sealed the dynamic pressure outside the boundary layer was essentially the same at station 2 as at station 1 and, although friction between stations 1 and 2 should cause an increase of about 3 percent in the momentum thickness, the measurements showed no appreciable change in either momentum thickness or displacement thickness between the two stations. The flow coefficient $\left(c_Q = \frac{Q}{b\delta^*_1 U_1}\right)$ and the total-pressure-loss coefficient $\left(C_{H_b} = \frac{H_1 - H_b}{q_1}\right)$ were referred to the stream velocity and dynamic pressure at station 1 ahead of the slot.

Type I slots (straight sharp-edge).- Typical boundary-layer velocity profiles at station 2 are shown in figure 4 for several rates of air flow through a type I slot ($\varphi = 90^\circ$, $w = 1.50$ in.) The no-flow curve was obtained with the slot sealed. Mean curves of the boundary-layer shape parameter H and the displacement-thickness ratio $\frac{\delta^*_2}{\delta^*_1}$ for all the type I slots are shown in figure 5. No systematic variations of H and $\frac{\delta^*_2}{\delta^*_1}$ were observed for the different slot angles or slot widths, and the maximum deviation of the displacement thickness from the mean value was less than 5 percent for most conditions. Nearly maximum effectiveness appears to have been obtained when $c_Q = 1$ since the shape parameter is approximately equal to the value for a $\frac{1}{7}$-power velocity profile, and the displacement thickness has been reduced to about 0.20 of its initial value.

The magnitude of the total-pressure-loss coefficient C_{H_b} plotted against flow coefficient is shown in figures 6(a), 6(b), 6(c), and 6(d) for slot angles of $\varphi = 90^\circ$, 60°, 45°, and 30°, respectively. The total-pressure-loss coefficient appears to drop rapidly as the slot width increases. No very consistent effect of slot angle can be seen. The high losses shown in the uppermost curve of figure 6(d) may be due to particularly violent flow separation from the rear edge and may thus indicate that, for high inlet-velocity ratios, slot angles as small as 30° may be harmful for sharp-edge slots.

Type II slots (straight with rounded edge).- Results of a few tests to determine the effect at station 2 of slightly rounding both front and rear edges of the slot simultaneously are shown in figures 7 and 8, from which the variation with flow coefficient of the profiles and of the mean values of the shape parameter and the displacement-thickness ratio can be seen. A small improvement in the external flow is observed for the type II slots by a comparison of the curves in figure 8 with those of figure 5 for type I slots. For the flow coefficient $c_Q = 1$ the displacement thickness has been reduced to 0.14 of its initial value.

Curves for total-pressure-loss coefficient against flow coefficient for the four slot angles are shown in figure 9. Reductions in excess of 30 percent from the corresponding type I slots were obtained by slightly rounding the slot edges. Since the reduction in total-pressure-loss coefficient which resulted from an increase in the front radius from $R_1 = 0.38$ to $R_1 = 1.50$ inches was small, further reductions did not appear feasible; therefore subsequent tests with a diffuser slot employed a front radius of $R_1 = 1.50$ inches.

Type III slots (round-edge diffuser of area ratio 2).- Curves of the mean values of shape parameter and displacement-thickness ratio for all the type III slots are shown in figure 10. Comparison of the curves of this figure with the curves for the two previous types (figs. 5 and 8) indicates that, once the slot edges have been rounded, the effectiveness of boundary-layer control by suction is primarily dependent on the quantity of air removed.

Total-pressure-loss coefficients are plotted against flow coefficient for the type III slots in figure 11. The effect of a change of slot width, slot angle, or diffuser angle can be seen by comparing the corresponding curves of these figures. The diffuser appears to offer a powerful means for reducing slot losses as can be seen by comparing the curves of figure 9 and figure 11 for $w = 0.75$ inch (although the larger value of R_1 for the diffuser slots probably also contributed somewhat to the improvement). The 12° diffuser gave lower total-pressure-loss coefficients than the 18° or 24° diffusers for all slot widths and slot angles through the entire range of flow coefficient tested. In order to determine what further improvement might be obtained, one test was made for a slot with the same area ratio, but with a smaller diffuser angle ($\varphi = 45^\circ$, $w = 0.75$ in., $\beta = 6^\circ$). No appreciable improvement was observed. Reducing the slot angle showed appreciable improvement, especially for the narrower slot ($\varphi = 30^\circ$) at flow coefficients less than 1.0; the 0.75-inch slot was almost as efficient as the 1.50-inch slot.

Comparison of the values of total-pressure-loss coefficient for a normal-opening type I slot with the best diffuser slot of the same width indicated a reduction of about 48 percent for a flow coefficient $c_Q = 1$. For this flow coefficient the total-pressure-loss coefficient for the best slot was $C_{H_b} = 1.22$.

Two tests were made with modifications to the best diffuser slot ($\varphi = 30^\circ$, $w = 1.50$ in., and $\beta = 12^\circ$) in an effort to obtain further improvements in the flow through the slot. Because splitter vanes have been used effectively to reduce large losses associated with unstable and irregular flow in some airplane inlet installations, the inlet opening was divided into several low-aspect-ratio openings by placing first three and later five splitter vanes in the slot. Neither of these modifications, however, altered the results.

Estimation of losses through the suction slot.- The total-pressure loss may be broken down into two parts: the total-pressure deficiency in that part of the boundary layer which is removed and the total-pressure loss attending the flow through the slot. Thus, if there is no appreciable mixing between station 1 and the slot inlet

$$C_{H_b} = \frac{H_1 - \overline{H}}{q_1} + \frac{\Delta H_s}{q_1} \tag{1}$$

where

$\overline{H}$ mean total pressure in the boundary layer to be removed, measured at station 1

ΔH_s total-pressure loss through the slot

The total-pressure deficiency in the removed boundary layer is

$$\frac{H_1 - \overline{H}}{q_1} = 1 - \frac{\int_0^h \left(\frac{u}{U}\right)^3 dy}{\int_0^h \left(\frac{u}{U}\right) dy} \tag{2}$$

where h is the distance normal to the surface at station 1 which determines the amount of the boundary layer removed.

Similarly

$$c_Q = \frac{1}{\delta^*}\int_0^h \left(\frac{u}{U}\right) dy = \frac{\int_0^h \left(\frac{u}{U}\right) dy}{\int_0^\infty \left(1 - \frac{u}{U}\right) dy} \tag{3}$$

The integrals of equations (2) and (3) were computed from the data at station 1 and are plotted in figure 12 as a curve of slot and boundary-layer total-pressure-loss coefficient against flow coefficient. The corresponding curve for the total-pressure-loss coefficient for the type III slot ($\varphi = 30^\circ$, $w = 1.50$ in., and $\beta = 12^\circ$) is also shown.

Figure 12 shows that for a flow coefficient of 1.0 the deficiency in the boundary layer is about $0.67q_1$, or about 55 percent of the measured total-pressure-loss coefficient. The remaining 45 percent, about $0.55q_1$, represents the further loss attending the flow through the slot. Presumably the very low total pressure near the bottom of the boundary layer results in violent flow separation from the inner wall of the slot; nevertheless, the $0.55q_1$ loss seems remarkably high, since it even exceeds the average dynamic pressure at the throat of the slot which is only about $0.36q_1$. It is of interest to note that the best of the narrower slots ($\varphi = 30^\circ$, $w = 0.75$ in., and $\beta = 12^\circ$), although not as efficient as the 1.50-inch slot, at least gave values of ΔH_s that are more readily explained in terms of the commonly recognized diffuser losses. For this slot the inlet velocity at $c_Q = 1.0$ is $\frac{\delta^*}{w} = \frac{0.91}{0.75} = 1.21$ times the free-stream velocity. The inlet dynamic pressure is then $(1.21)^2 q_1 = 1.46q_1$. Since the diffuser expansion ratio is 2:1, one-fourth of this dynamic pressure (or $0.37q_1$) is lost at the diffuser outlet. An additional diffuser loss of about 0.15 times the dynamic pressure at the inlet (or $0.22q_1$) may be assumed. The calculated value of ΔH_s for this case is thus about $0.59q_1$, which is reasonably close to the measured value of $0.68q_1$. The total-pressure loss for the narrower slot thus lends itself to an approximate evaluation, whereas the loss for the wider one does not. A detailed study of the flow into the slot might show the origin of the total-pressure loss in the case of the wider slot and indicate methods of reducing its magnitude.

In figure 12 are also shown, for comparison, corresponding curves determined from the data of reference 1 (dashed lines). The diffuser slot used in those tests was inclined 40° to the wall, had a well-rounded front edge but a sharp rear edge, and had a slot width of $\frac{w}{\delta^*} = 1.55$, which compares with $\frac{w}{\delta^*} = 1.65$ for the present tests. The loss through the slot (the difference between the two dashed curves) is appreciably less than that found in the present tests, probably because of the relatively higher total pressure near the bottom of the boundary layer.

Remarks on optimum flow coefficient and optimum slot width.- The results of reference 2 indicate that the optimum flow coefficient will be about unity ($c_Q = 1.0$) for boundary layers which have a shape parameter near 1.8. Reducing the value much below 1.0 considerably decreases the effectiveness of the boundary-layer control, whereas increasing the value much above 1.0 results in relatively little further improvement while greatly increasing the necessary suction power and the amount of equipment. The velocity profiles of figure 7 may be considered as further evidence, for the curves show rapid reduction in both boundary-layer thickness and boundary-layer shape parameter as c_Q approaches 1.0, with little possibility of further improvement beyond this point.

For this flow coefficient of unity the curves of figure 11 show that the intermediate slot widths (1.13 to 1.25 in.) were appreciably more effective than the smaller slot width (0.75 in.) but not appreciably less effective than the largest slot widths (1.50 to 1.75 in.). For type III diffuser slots tested, inlet widths of the order of $1.58\delta^*$ appear to be adequate for $c_Q = 1.0$; or, in general, an inlet velocity of about 0.65 appears to be indicated. An approximately similar result was obtained in reference 1, where it was found that inlet-velocity ratios above 0.6 gave rapidly increasing pressure losses, whereas reducing the inlet-velocity ratio to as low as 0.2 effected a further reduction in total-pressure-loss coefficient of only 0.06. The larger inlet widths are definitely preferable when no diffuser or rounded edge can be provided; if a long inclined diffuser can be provided, higher inlet-velocity ratios appear acceptable and may even reduce the inlet losses.

CONCLUSIONS

Tests of three types of boundary-layer-control suction slots were made at a velocity of about 100 feet per second with a turbulent

boundary layer which had a displacement thickness of 0.85 inch and a shape parameter of 1.8. Results of these studies indicate the following conclusions:

1. The characteristics of the new boundary layer which is formed behind the slot is determined only by the quantity of air removed, provided that the slot inlet has rounded edges.

2. Nearly maximum effectiveness is obtained when the rate of air-flow removal is equal to the air which would pass at free-stream velocity through an area equal to the displacement thickness per unit span (flow coefficient $c_Q = 1.0$).

3. Total-pressure losses through the slot may be appreciably reduced by rounding the inlet edges, inclining the slot, and slightly diverging its walls. Adequate width, however, is the most important feature of a satisfactory slot.

4. The total-pressure coefficient for the best slot tested (slot angle $\varphi = 30^{\circ}$) was 48 percent less than that for a normal-opening sharp-edge slot of the same width for $c_Q = 1.0$.

5. The total-pressure loss in the boundary layer represented about 55 percent of the measured total-pressure coefficient for the best slot at $c_Q = 1.0$.

6. The optimum inlet-velocity ratio for a diffuser slot is about 0.60 to 0.65. The optimum may be lower for the less efficient types of slots and may be higher in certain cases if a long diffuser can be used.

Langley Memorial Aeronautical Laboratory
National Advisory Committee for Aeronautics
Langley Field, Va., March 10, 1947

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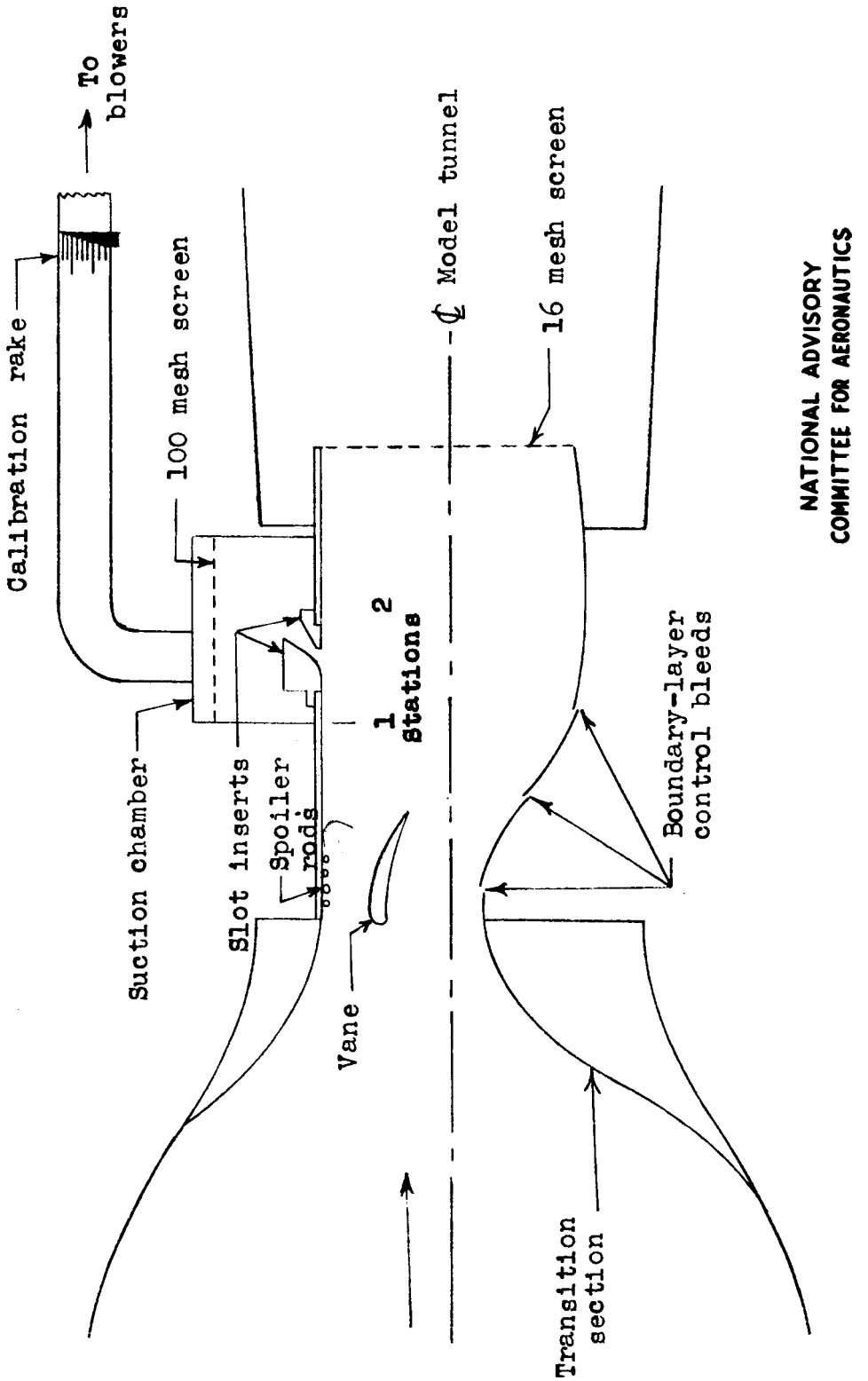
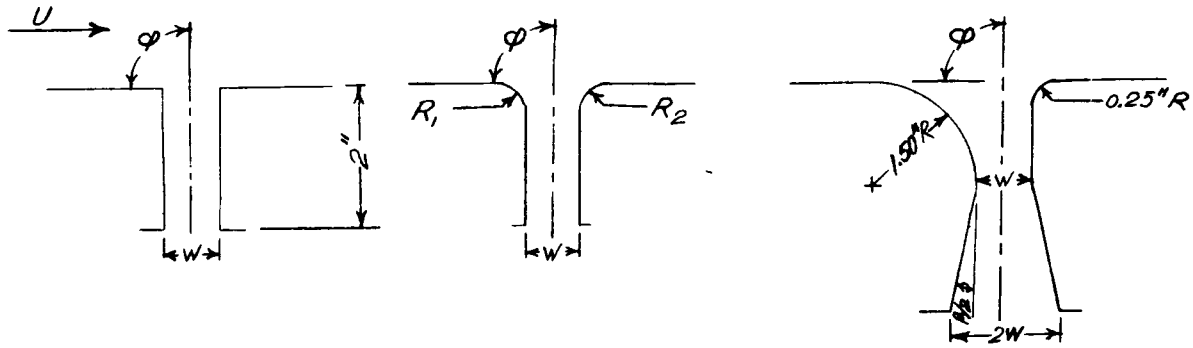


Figure 1.- General arrangement of apparatus for tests of boundary-layer control suction slots.



Type I
Straight slot with sharp edges

Slot angle φ, (deg)	Slot width w, (in.)
90	0.38 .63 .75 1.50
60	0.38 .63 .75
45	0.38 .63 .75
30	0.38 .63 .75

Type II
Straight slot with rounded edges; w = 0.75 in.

Slot angle φ, (deg)	Edge radius R_1, (in.)	Edge radius R_2, (in.)
90	0.06 .13 .25 .38 1.50	0.06 .13 .25 .38 .38
60	0.06 .13 .25 .38	0.06 .13 .25 .38
45	0.06 .13 .25 .38	0.06 .13 .25 .38
30	0.06 .13 .25 .38	0.06 .13 .25 .38

Type III
Diffuser slot

Slot angle φ, (deg)	Slot width w, (in.)	Diffuser angle β, (deg)
90	0.75 1.25 1.75	12, 18, 24
60	0.75 1.25 1.75	12, 18, 24
45	0.75 1.25 1.75	12, 18, 24
30	.75 0.75 1.13 1.50	6 12, 18, 24

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Figure 2.- Sketches and dimensions of the three boundary-layer control slots.

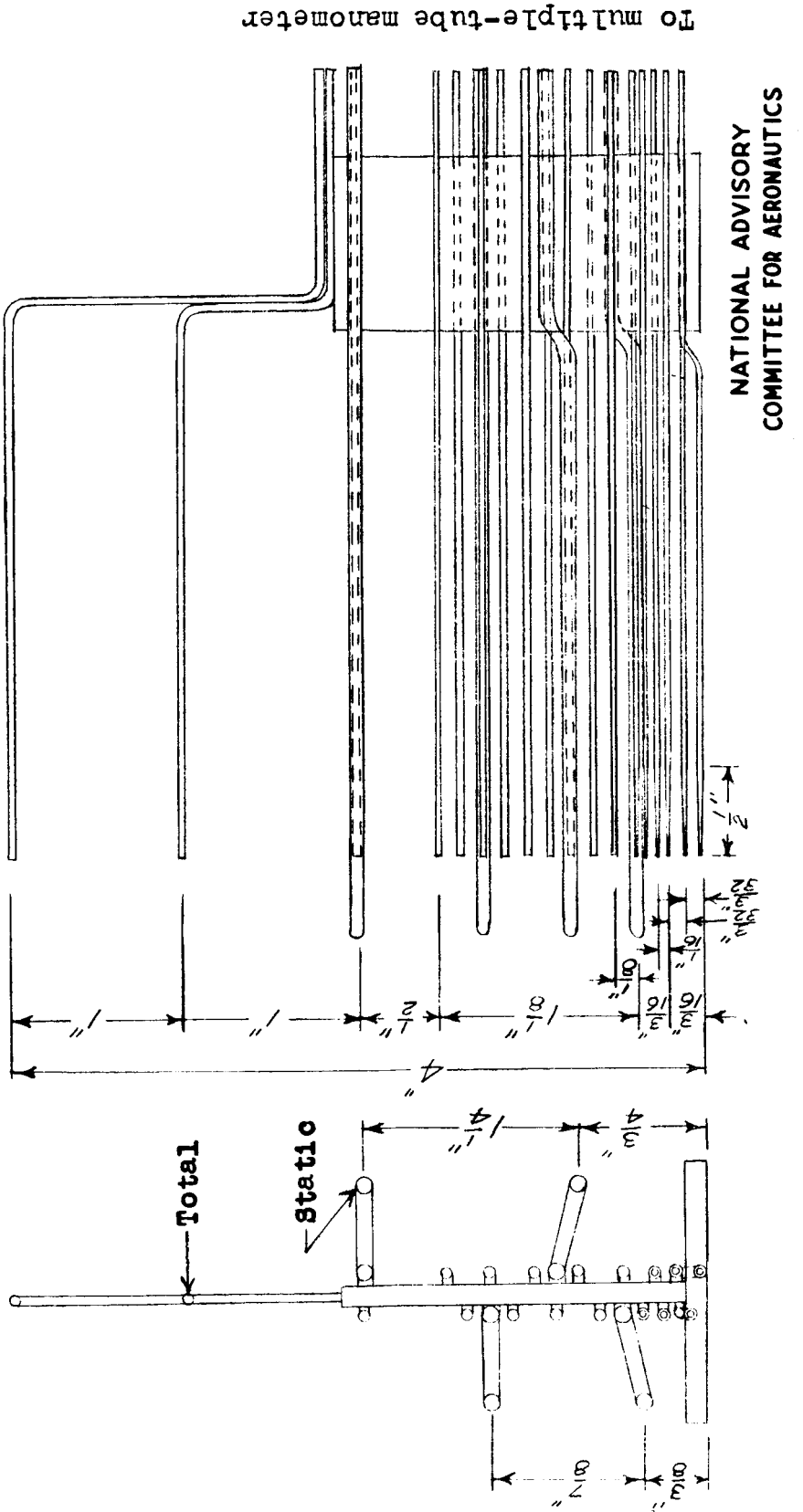


Figure 3.- Boundary-layer pressure measurement rake. Total-pressure tubes are 0.040-inch outside diameter and 0.026-inch inside diameter stainless steel. The lower six tubes are tapered to 0.030-inch outside diameter. Static-pressure tubes are 0.060-inch outside diameter with four 0.0135-inch-diameter orifices.

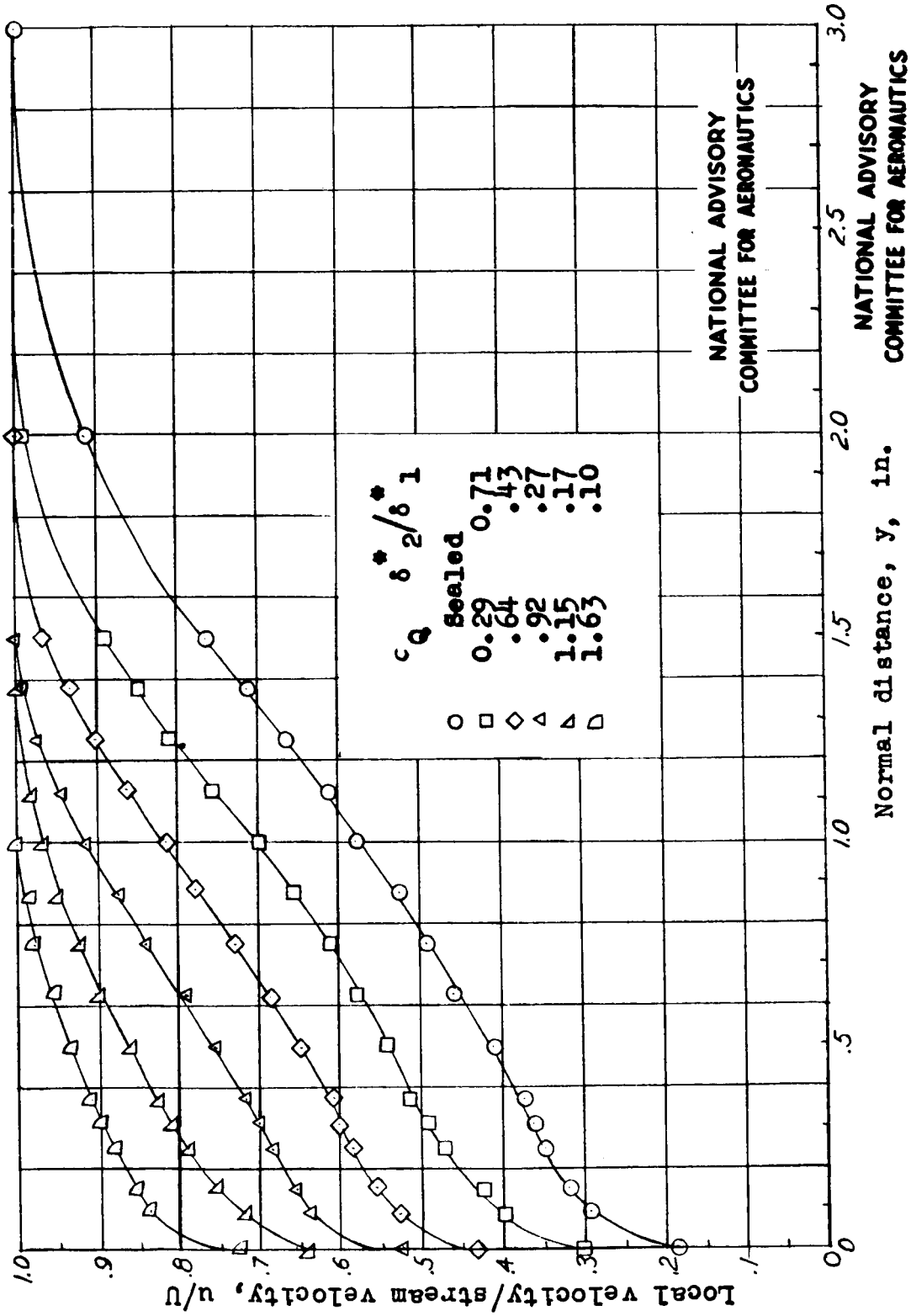
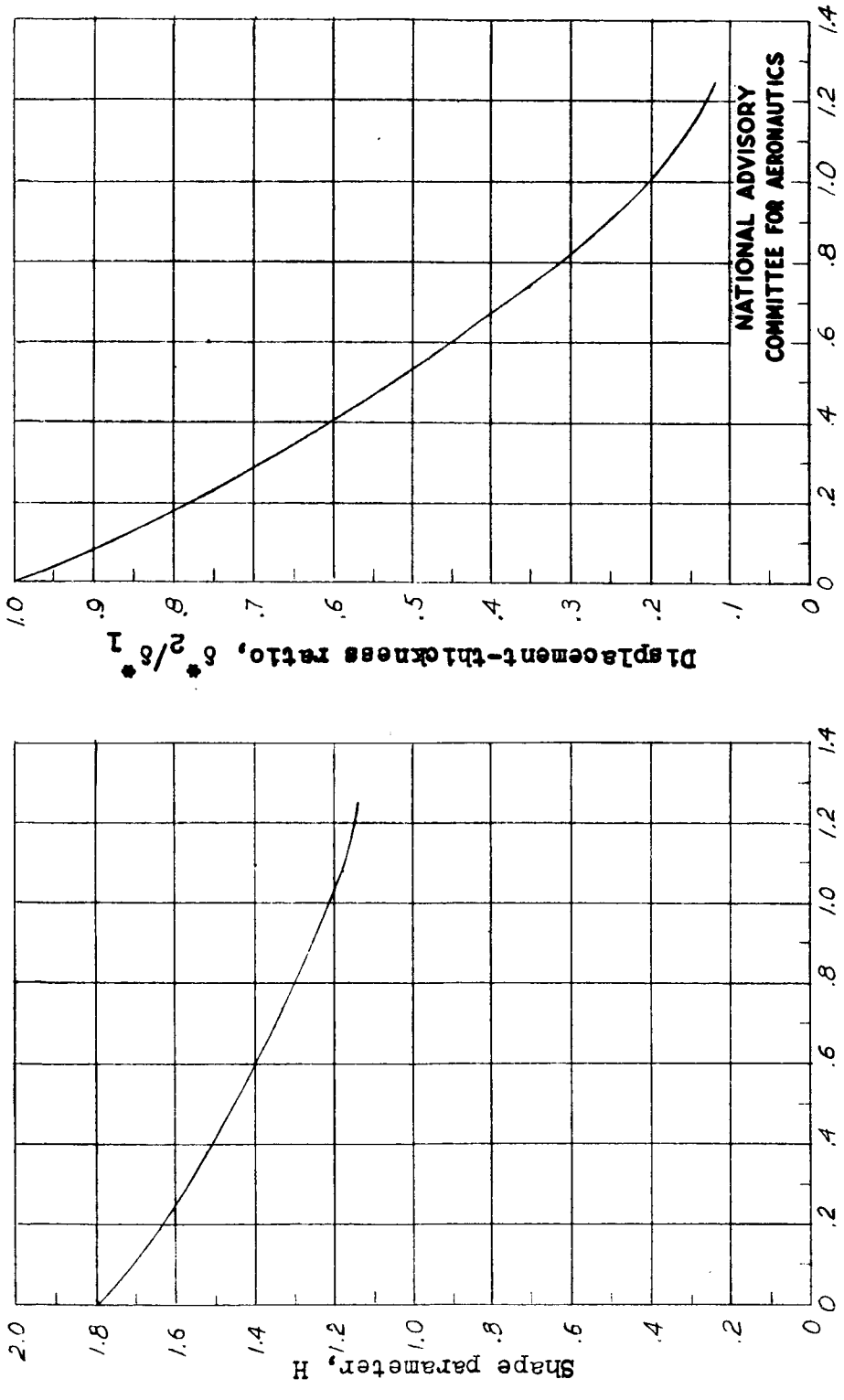


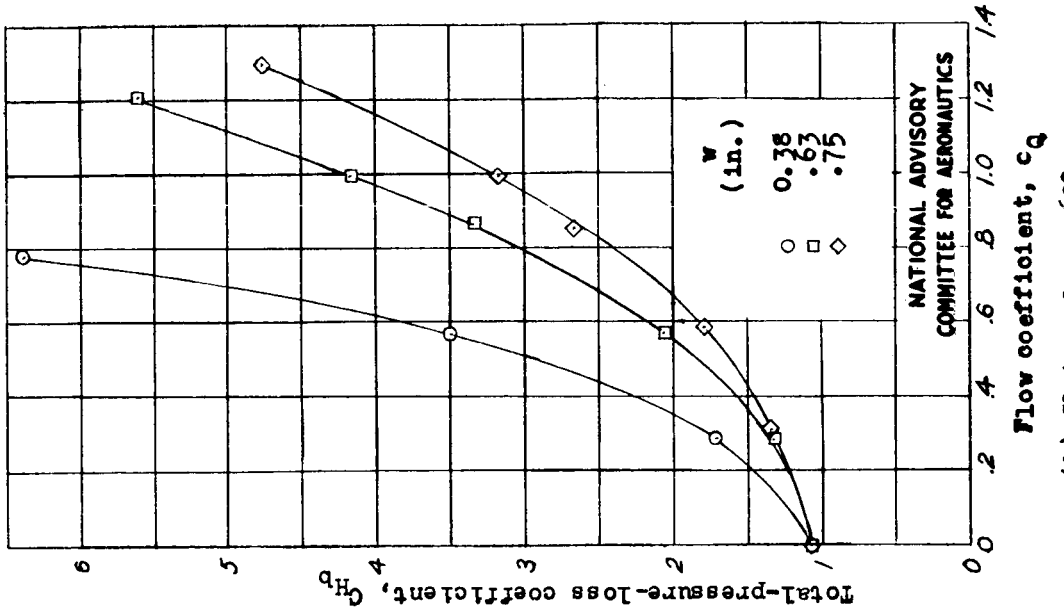
Figure 4.- Typical boundary-layer velocity profiles, with and without suction, at station 2 for type I slot.



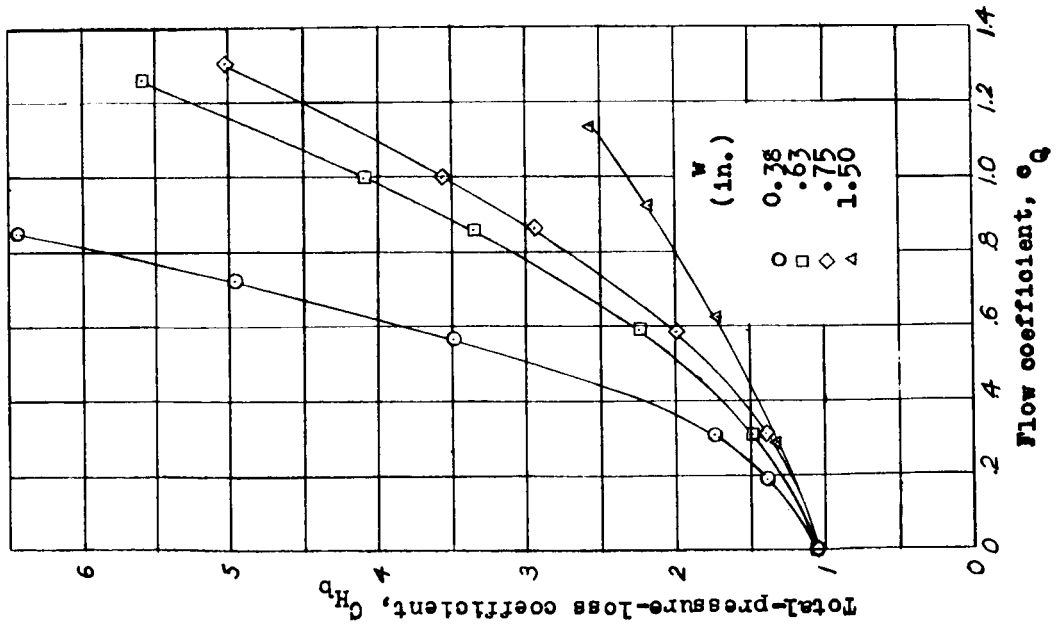
(a) Shape parameter, H.

(b) Displacement-thickness ratio, δ^*_2/δ^*_1.

Figure 5.- Variation with flow coefficient of boundary-layer characteristics at station 2 for type I slots.

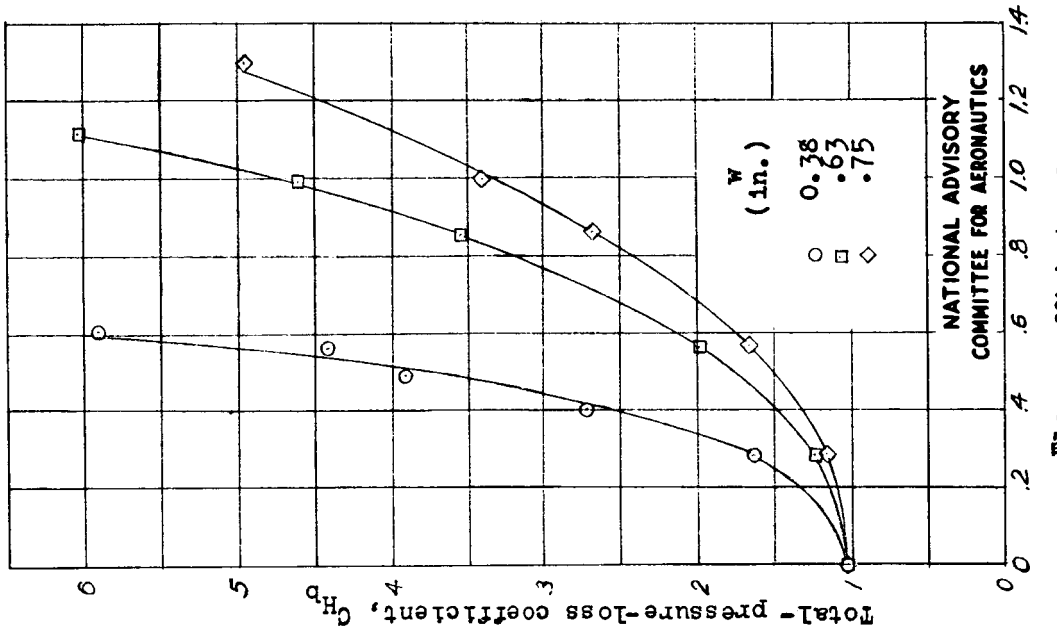


(b) Slot angle = 60°.

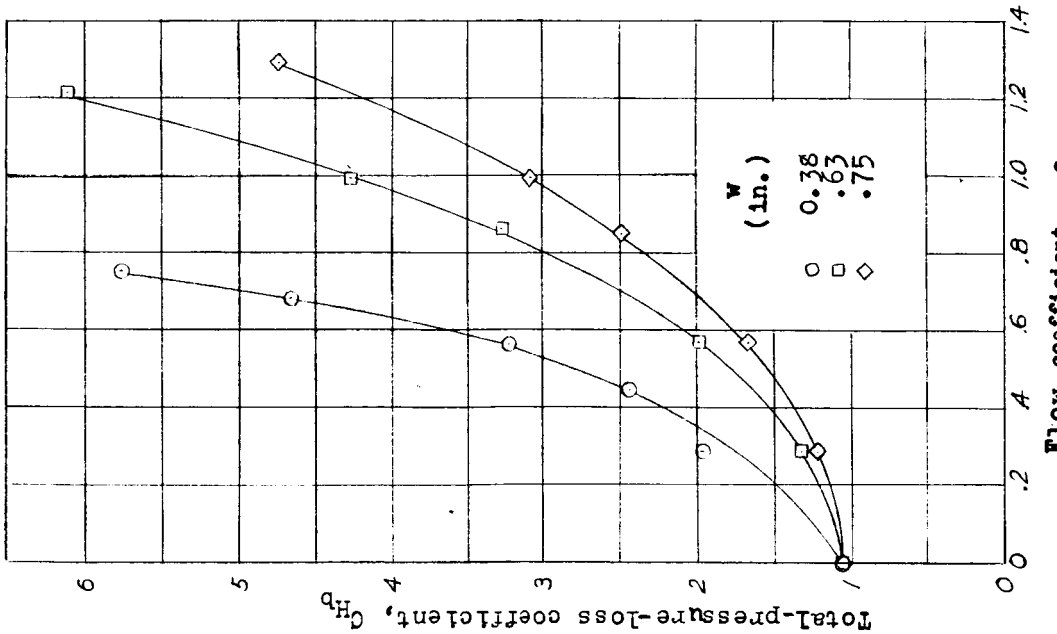


(a) Slot angle = 90°.

Figure 6.- Variation of total-pressure-loss coefficient with flow coefficient for type I slots.



(d) Slot angle = 30°.



(c) Slot angle = 45°.

Figure 6.- Concluded.

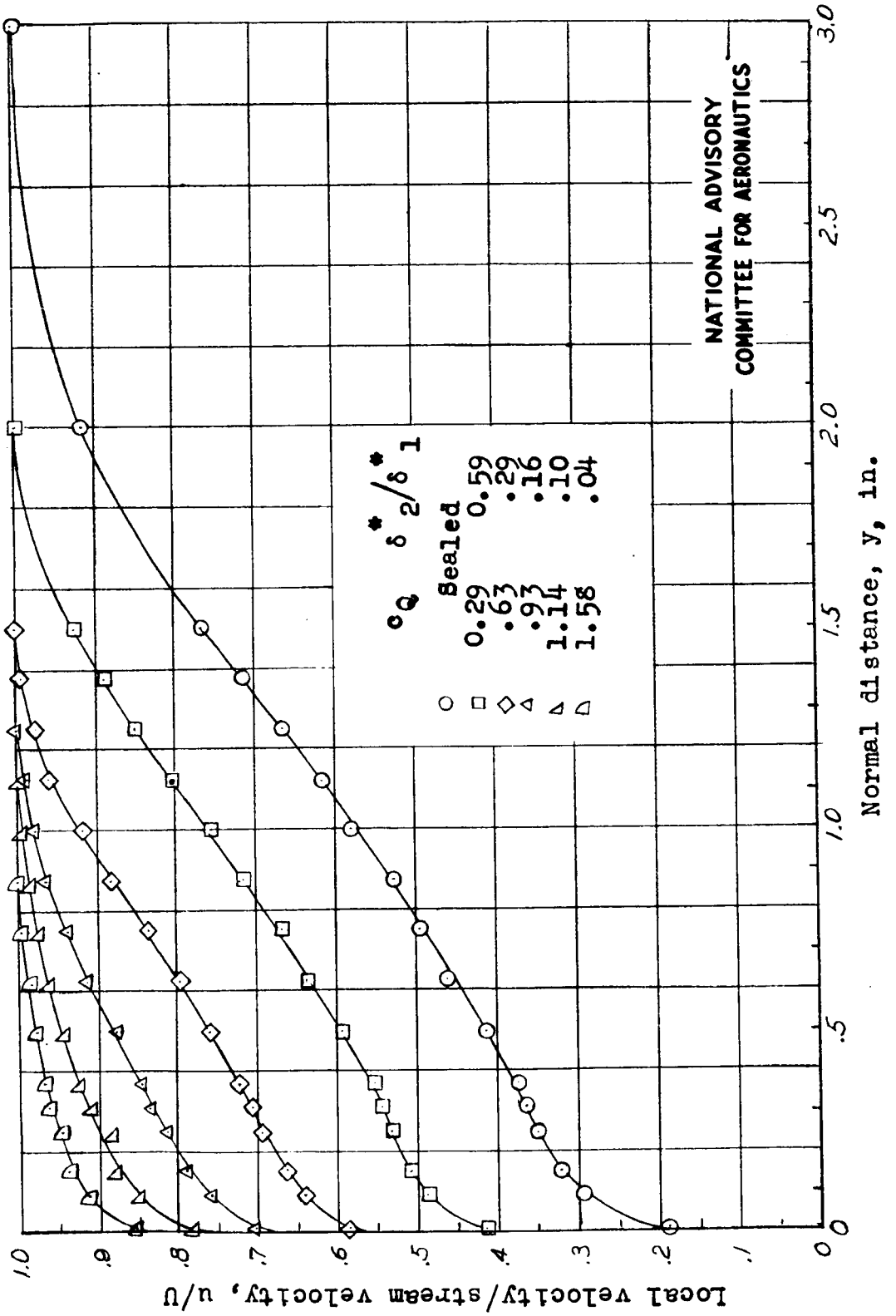
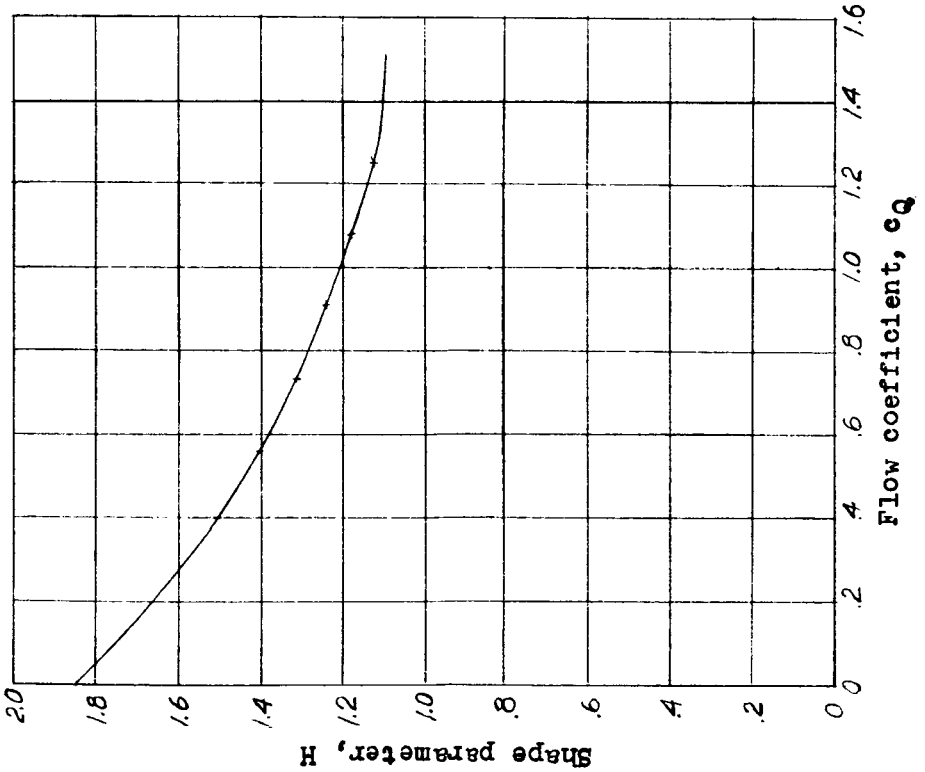
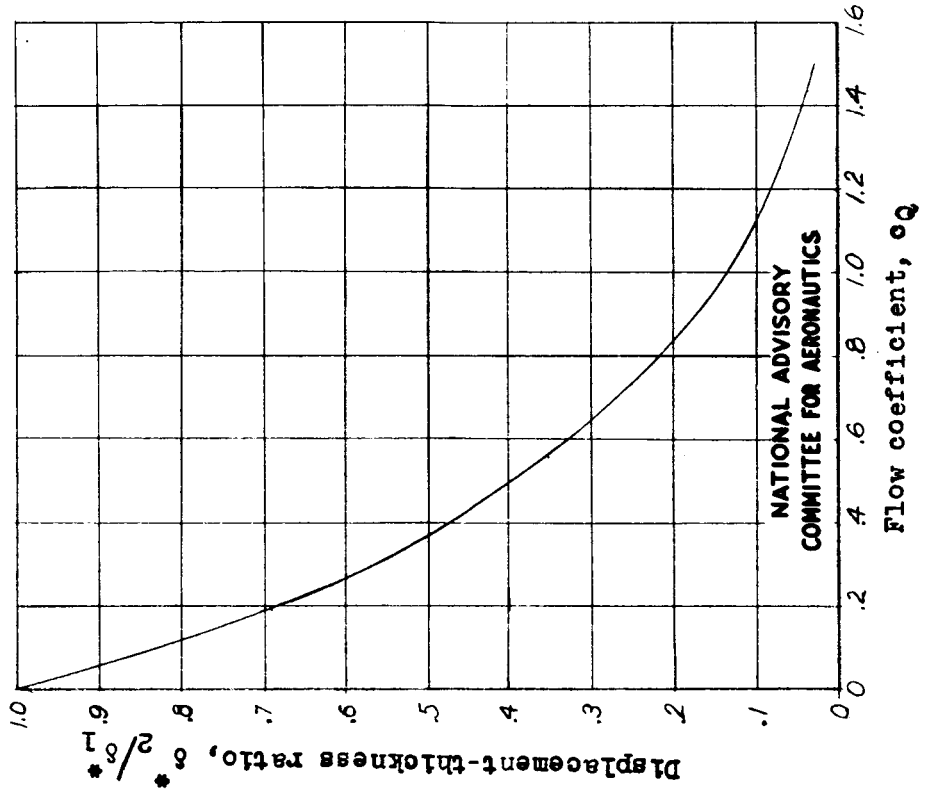


Figure 7.- Typical boundary-layer velocity profiles, with and without suction, at station 2 for type II slots.

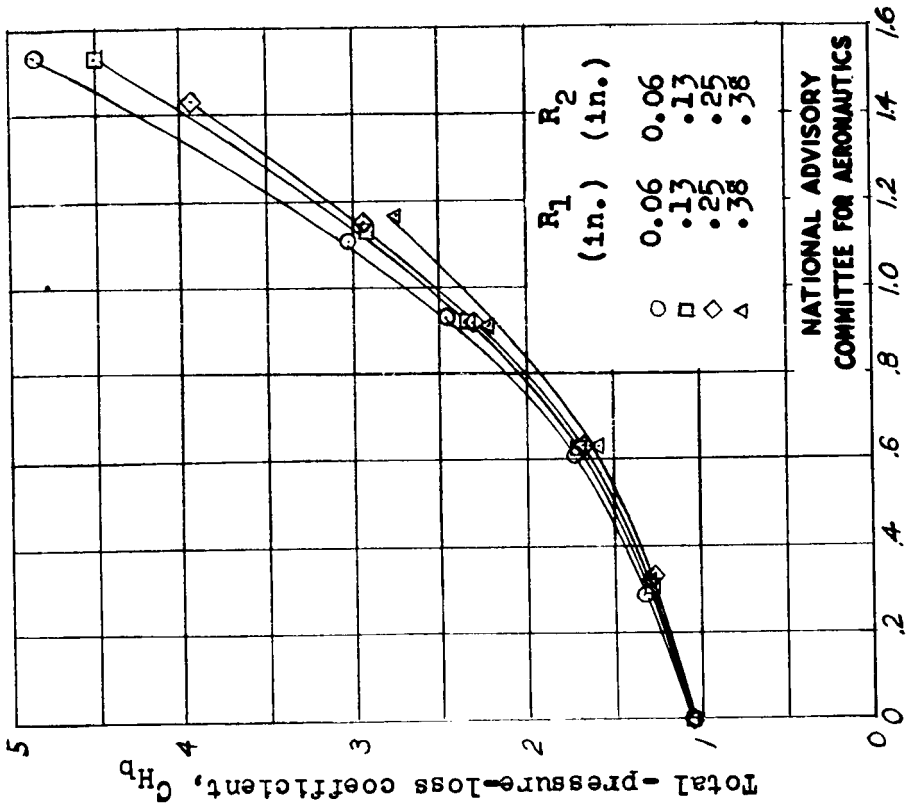


(a) Shape parameter, H.

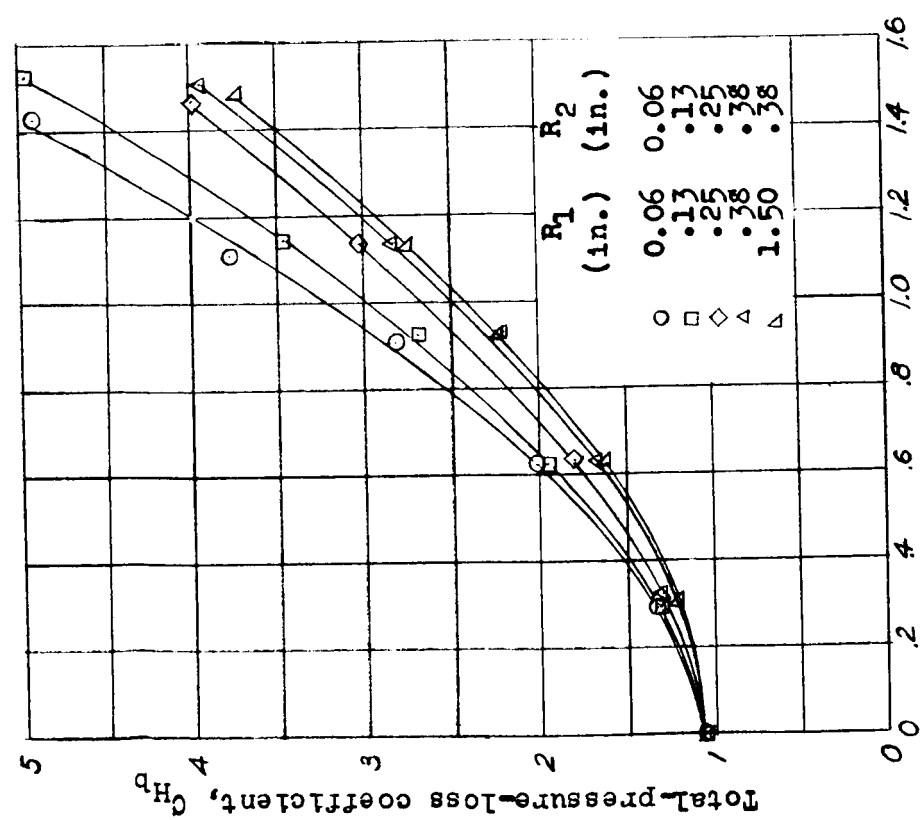


(b) Displacement-thickness ratio, δ^*_2/δ^*_1.

Figure 8.- Mean curves of boundary-layer characteristics at station 2 plotted against flow coefficient for type II slots.

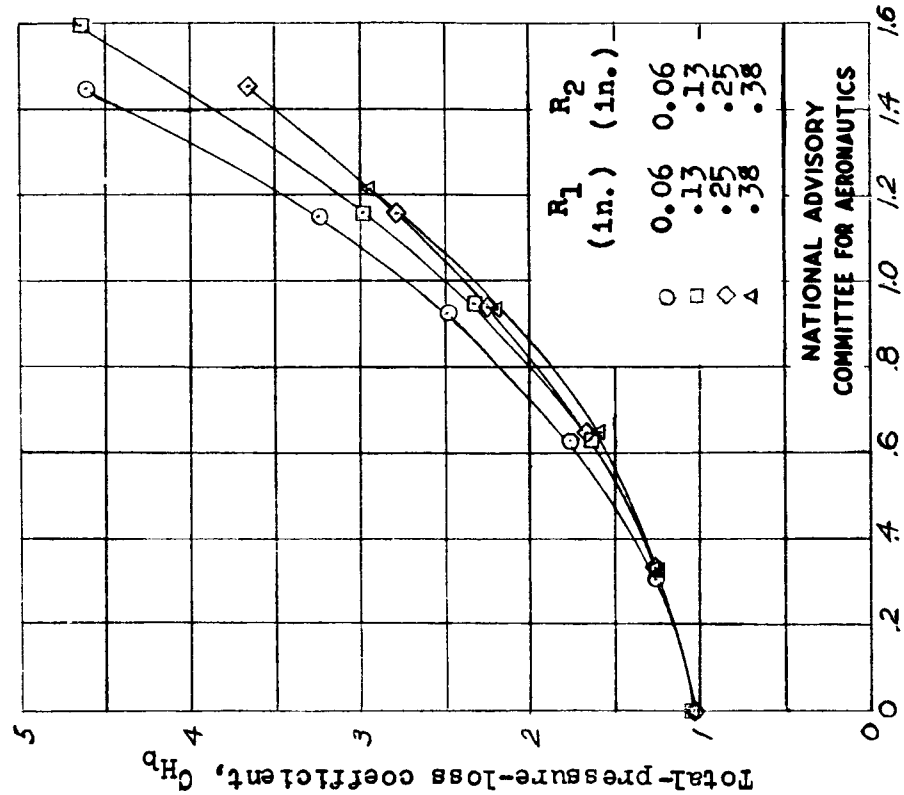


(b) Slot angle = 60°.

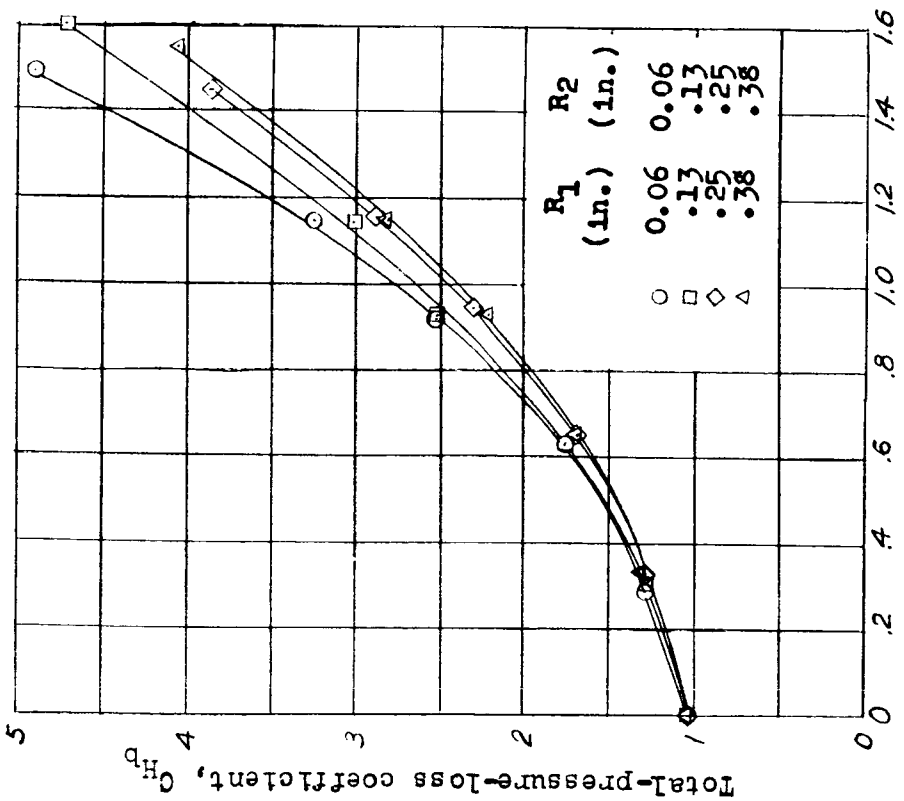


(a) Slot angle = 90°.

Figure 9.- Variation of total-pressure-loss coefficient with flow coefficient for type II slots. w = 0.75 inch.

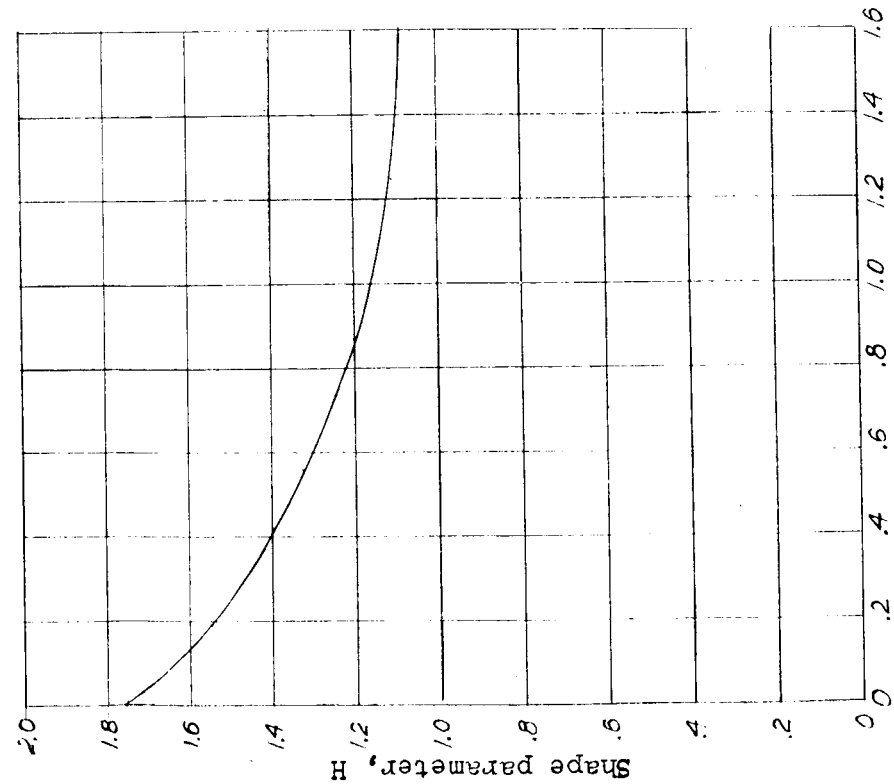


(d) Slot angle = 30°.

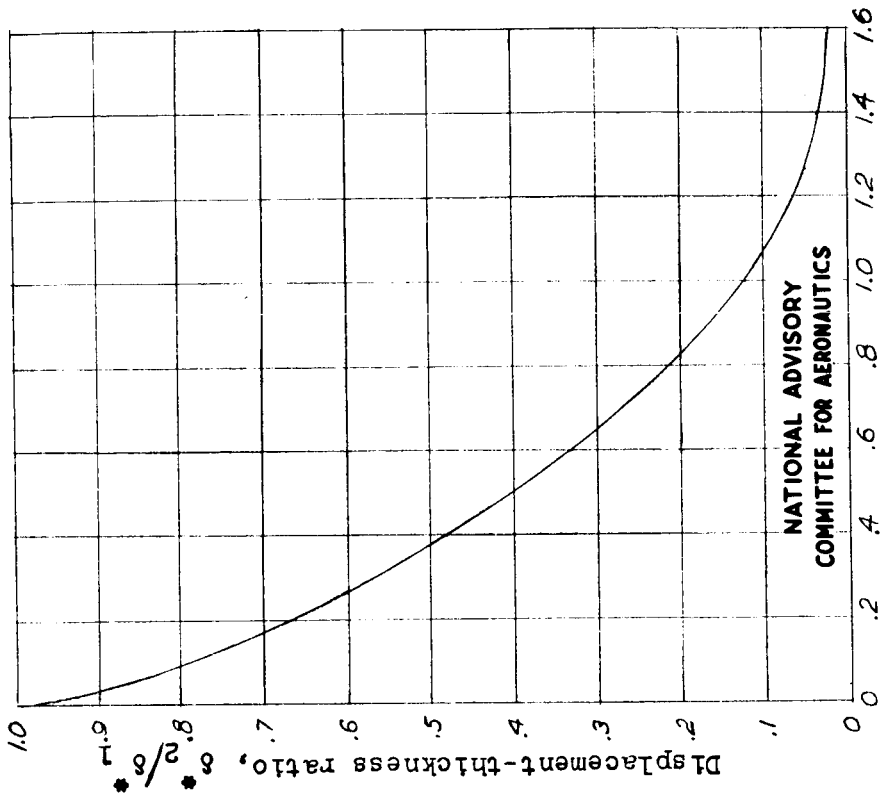


(c) Slot angle = 45°.

Figure 9.- Concluded.

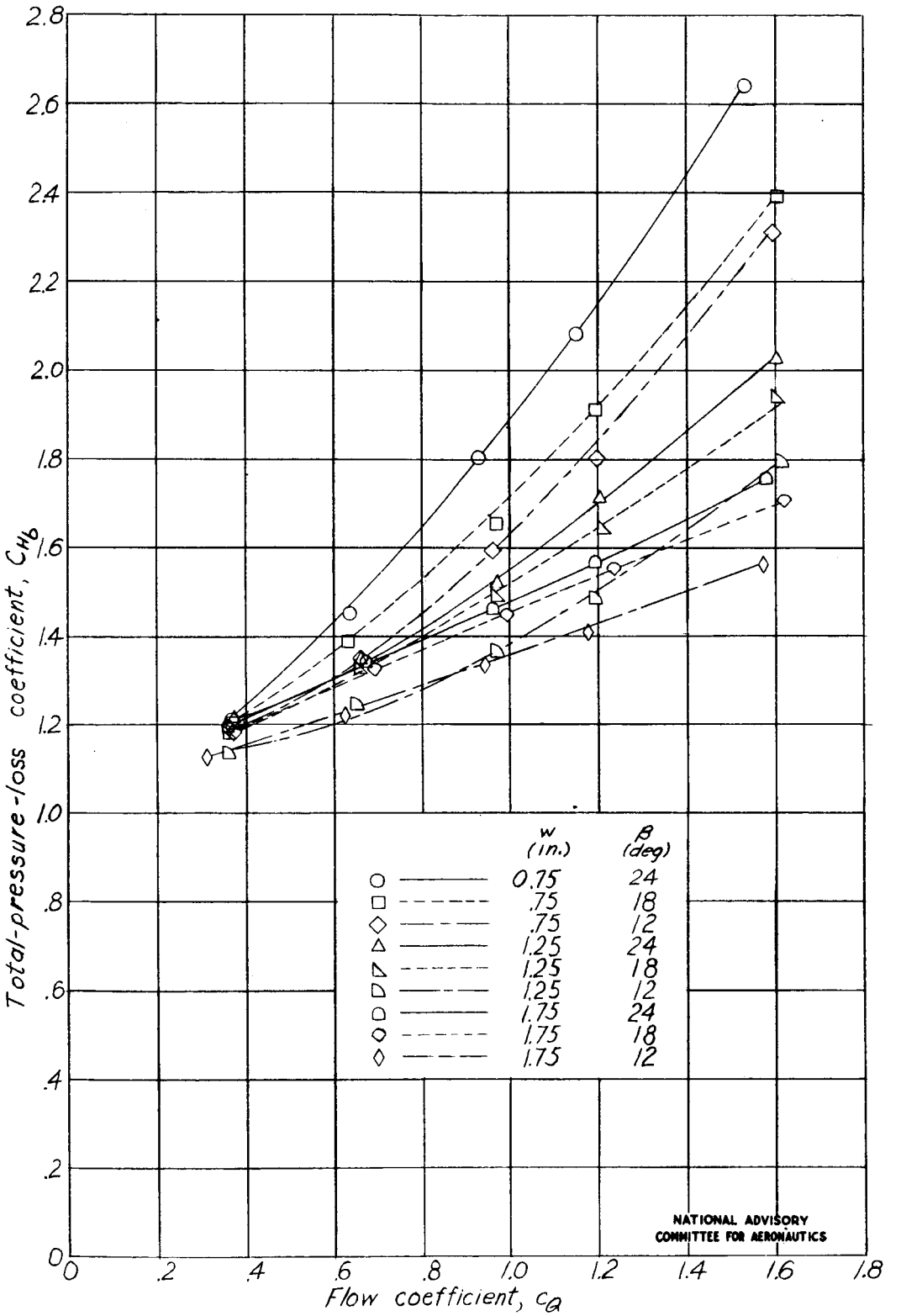


(a) Shape parameter, H.



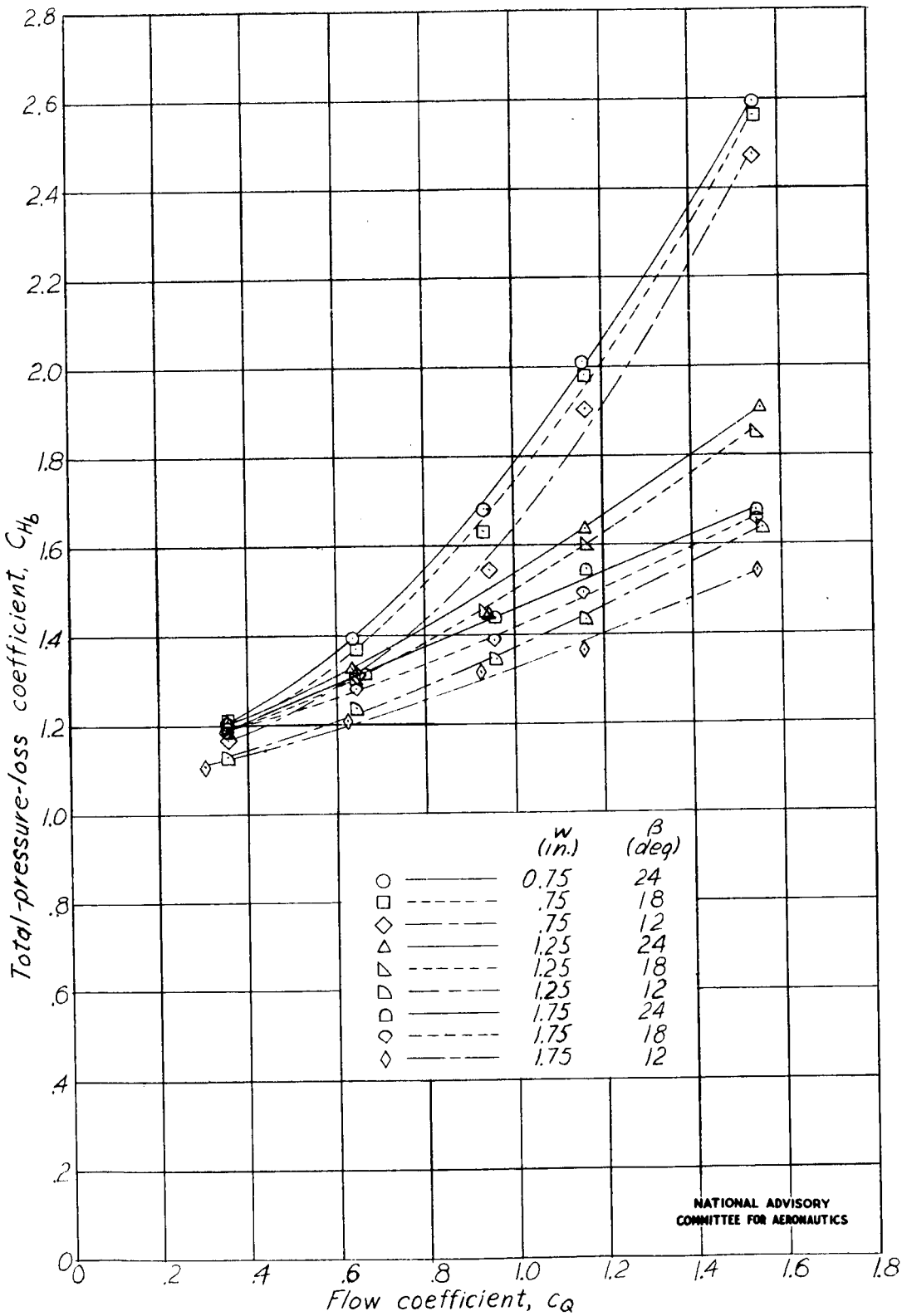
(b) Displacement-thickness ratio, δ^*_2/δ^*_1.

Figure 10.- Mean curves of boundary-layer characteristics at station 2 plotted against flow coefficient for type III slots.



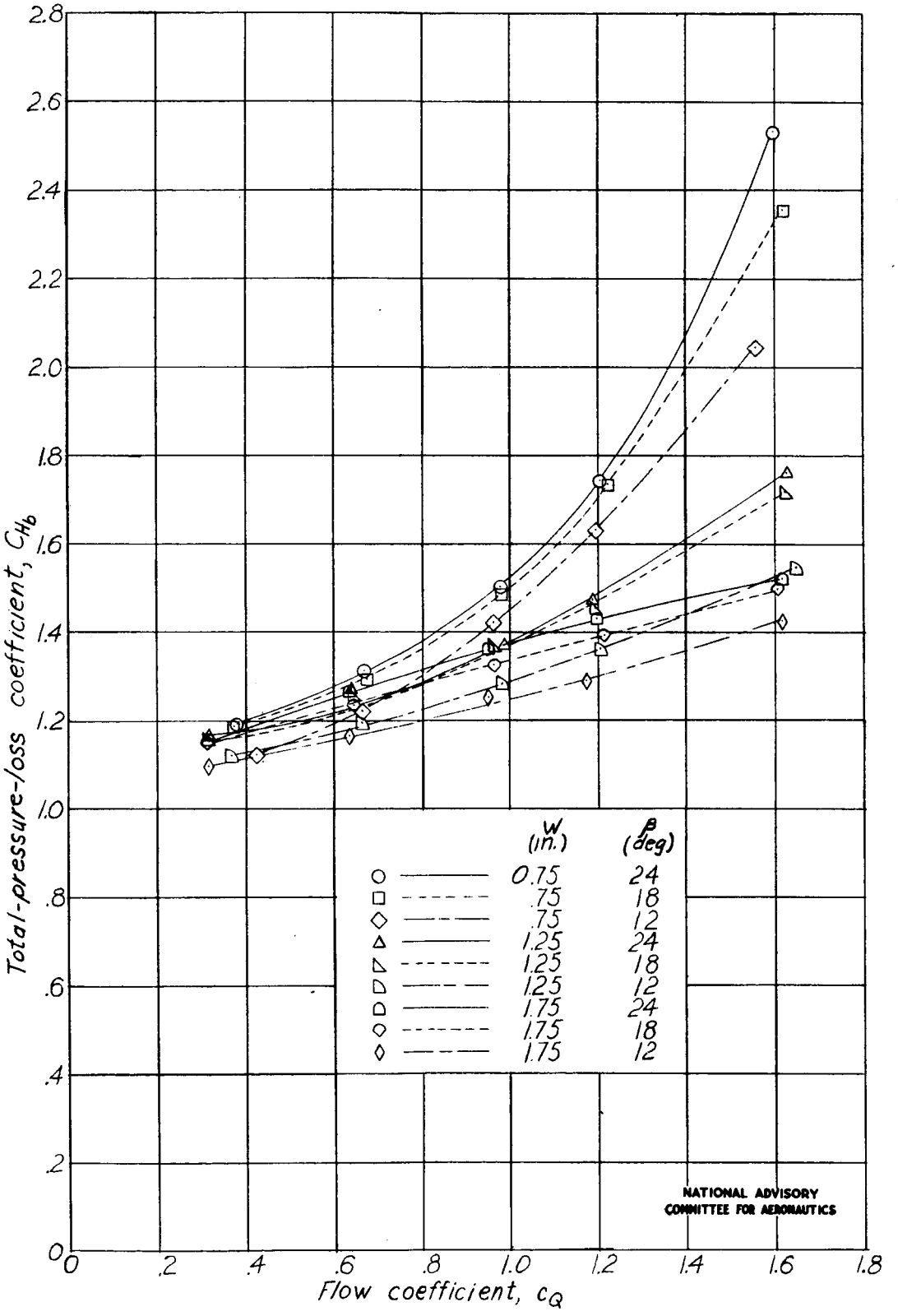
(a) Slot angle = 90°.

Figure 11.- Variation of total-pressure-loss coefficients with flow coefficient for type III slots.



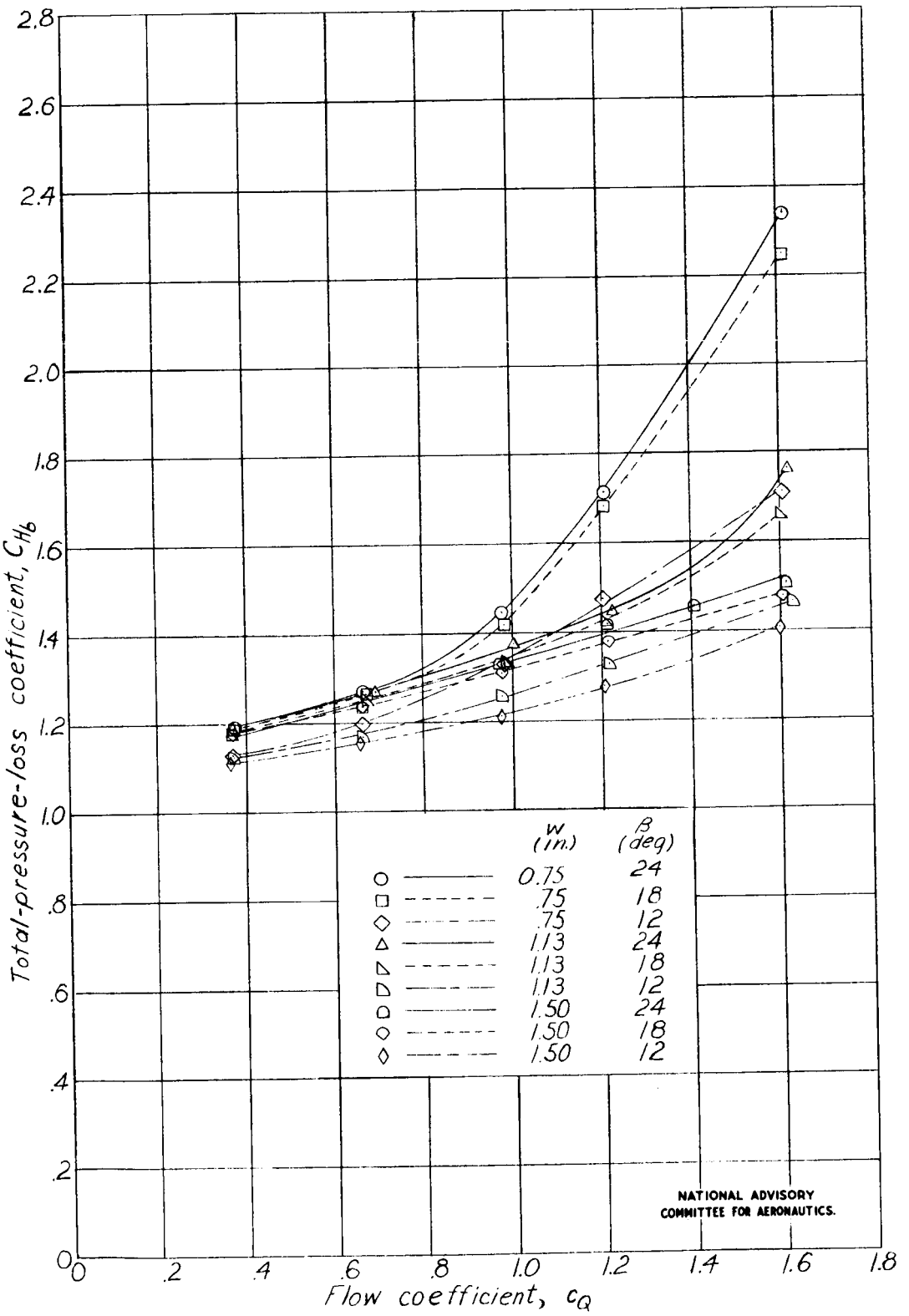
(b) Slot angle = 60°.

Figure 11.- Continued.



(c) Slot angle = 45°.

Figure 11.- Continued.



(d) Slot angle = 30°.

Figure 11.- Concluded.

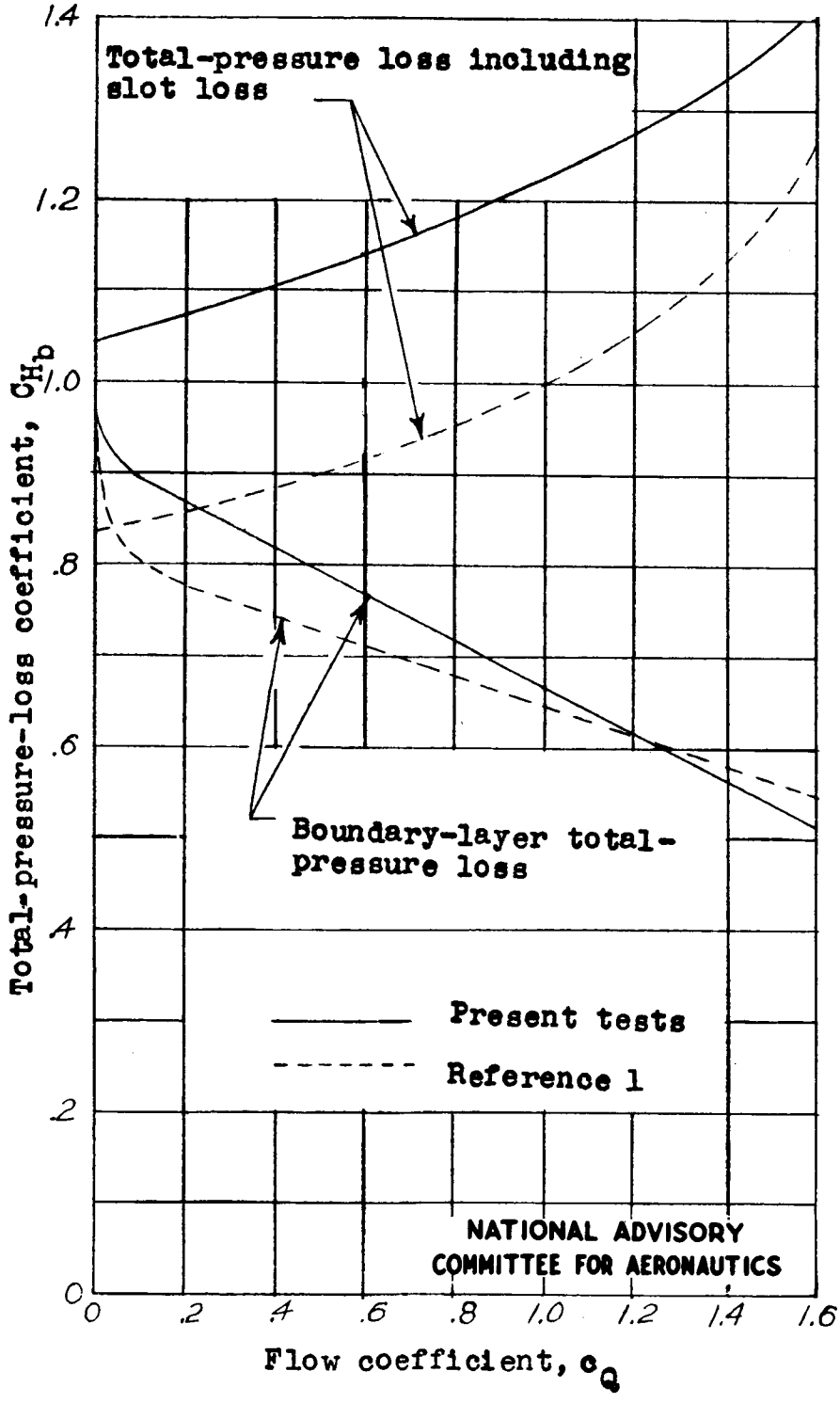


Figure 12.- Variation with flow coefficient of slot and boundary-layer total-pressure-loss coefficients.