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RESEARCH MEMORANDUM

DESIGN AND TEST OF MIXED-FLOW IMPELLERS

V - DESIGN PROCEDURE AND PERFORMANCE RESULTS FOR TWO VANED DIFFUSERS TESTED WITH IMPELLER MODEL MFI-1B

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SUMMARY

The design procedure and experimental test results for a range of impeller speeds are presented for two vaned diffusers of 24 and 40 vanes. Both sets of vanes were designed to give approximately the same prescribed velocity distribution on the vane surfaces and were tested with the same impeller. For both diffusers the maximum efficiency was 0.78 for a pressure ratio of 3.25 at an impeller mean outlet speed of 1300 feet per second. At the design speed of 1400 feet per second, the efficiency was 0.77 for a pressure ratio of 3.73.

Pivoting the vanes of the 40-vane diffuser 5° and bending the vanes at inlet for the 24-vane diffuser 5° to match a reduced impeller weight flow increased the weight-flow range from choke to surge at design speed by 100 percent (from 0.6 to 1.2 lb/sec) in the 40-vane diffuser and 65 percent (from 1.2 to 2.0 lb/sec) in the 24-vane diffuser. The weight-flow range with the vaned diffuser is much shorter than that obtained with the vaneless diffuser at each operating speed.

INTRODUCTION

Mixed-flow impellers MFI-1 and MFI-2 have previously been investigated (refs. 1 to 4) with vaneless diffusers, because the wide range of weight flow obtained with this type diffuser allows a less restricted examination of impeller performance than with the use of vanes. In addition, the flow conditions leaving the impeller are such that a correct evaluation of impeller performance cannot be made from measurements near the impeller outlet (refs. 5 and 6, e.g.), and a more accurate evaluation can be made in a vaneless diffuser at a diameter approximately 1.5 times that of the impeller. Diffusion in the vaneless diffuser results primarily from the reduction in tangential velocity with increase in radius (due to conservation of angular momentum); therefore, the

diffuser-outlet diameter is more than twice the impeller diameter, so that the weight flow per unit frontal area is relatively low with this type diffuser. As a result, compressors for aircraft engines generally employ vaned diffusers (refs. 7 and 8, e.g.) with a vaneless transition section between the impeller outlet and the entrance to the vanes.

The vaneless transition section is especially desirable where flow leaving the impeller is supersonic, because it allows diffusion of the flow to subsonic without shock before entering the vanes if the axial-radial component is subsonic. Diffusion to a Mach number of approximately 0.9 before entering the diffuser vanes was considered low enough to avoid choking in this case. Diffusion to a Mach number of approximately 0.3 in the vanes while turning to the axial-radial direction was prescribed. Using these prescribed inlet and outlet conditions, two diffusers, one with 40 vanes and one with 24 vanes, were designed by the method of reference 9. The progressive area increase required for diffusion in the 40-vane diffuser was provided by an average divergence angle of 10° between the pressure and suction surfaces of the vanes. In the 24-vane diffuser, the increasing area was provided by a divergence angle up to 9° between the inner and outer walls with little to no divergence between the pressure and suction surfaces. The prescribed velocity distributions on the vane suction and pressure surfaces were approximately the same for both diffusers. The difference in vane height accounts for the difference in number of vanes.

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The objective of the investigation was to determine experimentally the over-all efficiency that could be obtained with vanes designed by the procedure followed herein and to compare the results obtained for two diffusers that have quite different geometrical shapes but have approximately the same velocity distribution on the vane surfaces.

SYMBOLS

The following symbols are used in this report:

f_s slip factor, ratio of absolute tangential velocity at exit to impeller speed at exit approximated by ratio of measured enthalpy rise to (impeller speed)^2/gJ

g acceleration due to gravity, 32.174 ft/sec^2

J mechanical equivalent of heat, 778.2 ft-lb/Btu

Q ratio of local velocity to velocity of sound at stagnation

U actual impeller speed based on 7.00-in. radius, ft/sec

w actual air weight flow, lb/sec

β angle between camber line of vane and a conic element through leading edge of vane (fig. 1)

δ ratio of inlet total pressure to NACA standard sea-level pressure of 29.92 in. Hg abs

η_{ad} adiabatic temperature-rise efficiency

θ ratio of inlet total temperature to NACA standard sea-level temperature of 518.688° R

APPARATUS AND INSTRUMENTATION

Apparatus

Two diffusers were tested, one having 40 vanes (figs. 1 and 2) and the other 24 vanes (figs. 3 and 4). Both were tested with impeller model MFI-1B of reference 4, except that for the 24-vane set the leading edges of the impeller as tested in reference 4 were swept backward (fig. 3) from the hub inlet to a point on the shroud 0.25 inch in the axial direction from the original leading edge, because the original leading edges had been damaged. This did not change the blade angle at inlet, and tests showed that it did not change the impeller performance.

For the 40-vane diffuser the wall profile is the same as that for the vaneless diffuser of impeller model MFI-1B of reference 4 from the impeller outlet to the sheet-metal extensions (fig. 1) which have a constant spacing of 0.360 inch throughout. The length of the vanes, measured in the axial-radial direction, was 3.5 inches as designed by the method of reference 9. The blade at exit was extended downstream 1/2 inch in the axial-radial direction with the expectation of reducing the length after the initial tests and retesting with the original design length to determine the effect upon performance. All performance results presented herein for the 40-vane diffuser are for the length extended to 4 inches.

For the 24-vane diffuser the outer diffuser wall is the same as for the 40-vane diffuser. The inner wall is the same to a radius of 8.3 inches (0.125 in. upstream of the vane inlet). At the radius of 8.3 inches it diverges from the outer wall at an angle of 3.5° to a radius of 8.83 inches, at which point the divergence becomes 9°. The design length of the vanes is 3.5 inches in the axial-radial direction. The end of the diffuser vanes is shown in figure 3. From the end of the

diffuser vanes as designed, the annulus continues to diverge for 0.5 inch, at which point the passage becomes a constant-area duct for the remaining 6.0 inches.

In the course of investigation, the 40-vane-diffuser inlet angle β was changed from 59.3° to 64.3° by turning the entire vane, filling in the gap left at the inner wall, and turning down the vane to fit the outer wall. The 24-vane inlet angle was changed from 61° to 66° by bending the vanes along the first 3/4 inch of travel along the vane at inlet. The remainder of the experimental setup is as described in reference 2. 3628

The vanes were cast from a 0.90-0.10 tin-zinc alloy (melting point approximately 390° F) and were attached to the inner diffuser wall. For added rigidity a brass strip was imbedded for the height of the vanes in the 24-vane castings for a distance of 2.00 inches from the leading edge and in the 40-vane castings for the entire length of the vanes.

Instrumentation

The instrumentation is the same as that in reference 2 except at the outlet measuring stations.

40-vane diffuser. - The outlet instrumentation station is shown in figure 1 for the 40-vane diffuser. Four spike-type calibrated thermocouple rakes with three probes each were spaced 90° apart. The probes were spaced on the rake so as to cover equal areas and were directed radially inward. Twelve unshielded 0.040-inch-diameter total-pressure probes directed radially inward were spaced around the periphery of the outlet measuring station so as to give the same coverage as the thermocouples. There were four static taps 90° apart on each wall at this station.

24-vane diffuser. - The description of the outlet instrumentation for this diffuser as given in reference 10 is repeated here for convenience. The outlet measuring station is located at the position shown in figure 3. There were eight static taps (four on the inner wall and four on the outer wall), 20 thermocouple probes, and 36 total-pressure probes. These were distributed throughout the 24 passages to give the coverage shown in figure 3 at four positions.

Vaneless-diffuser surveys. - Total-pressure surveys in the vaneless diffuser (prior to installing vanes) were made with the claw-type probe shown in figure 5. This probe was used to determine total pressure and angle of flow. Static pressures were measured on both walls in the planes of the surveys. Surveys were made at three positions at average

distances of approximately $\frac{7}{8}$, $1\frac{3}{4}$, and $2\frac{5}{8}$ inches from the impeller outlet (positions 1, 2, and 3, fig. 1). In addition, total-temperature surveys were made at position 1.

Outlet pipe. - Total pressure and temperature were measured 7 diameters downstream in the outlet pipe.

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PROCEDURE

Experimental Procedure

Total-pressure ratio and efficiency. - In general, the total-pressure ratio is taken herein as the ratio of the mass-averaged total pressures at the outlet measuring stations shown in figures 1 and 3 to the arithmetically averaged total pressures in the inlet tank. Specifically, the outlet-pipe total-pressure ratio is taken as the ratio of the arithmetic average of the total pressures measured in the outlet pipe to the arithmetically averaged total pressures in the inlet tank. A mass-average of the temperatures taken at the outlet stations shown in figures 1 and 3 was used for all adiabatic-efficiency calculations for both diffusers except as noted in the following discussion of temperature.

Temperature. - At the maximum efficiency points, there was a difference of less than 1° F between the temperatures measured in the outlet pipe and those at the outlet measuring station for the speeds of 1300, 1400, and 1600 feet per second at both angle settings for the 40-vane diffuser and for the speeds of 1300 and 1400 feet per second at both angle settings for the 24-vane diffuser. There were differences up to 2° for other points at these speeds. At speeds of 1100 feet per second and below, there were differences ranging from 4° to 10°. The temperatures measured at the outlet measuring station were always higher and were therefore used in computing efficiencies. A comparison of performance with the vaned and vaneless diffusers at these speeds showed that operation with vanes was restricted to weight flows for which rotating stall or surge caused similar discrepancies in measured temperatures for the vaneless diffuser. Inaccuracies at the outlet measuring stations caused by fluctuating flows that accompany stall or surge may account for discrepancies. Use of the higher temperature does not necessarily give efficiency values that are too low, inasmuch as the pressure measurements at the point may be proportionately too high for the same reason (fluctuating flows). The accuracy of results at these speeds (1100 ft/sec and below) is difficult to estimate.

For the 24-vane diffuser at speeds of 1500 feet per second and above, the outlet-pipe temperatures were used in computing over-all efficiency for the 24-vane diffuser. The reason for this is as follows: For the 24-vane diffuser at maximum efficiency, the temperatures in the outlet

pipe were 2° higher than those at the outlet measuring station at 1500 feet per second, 7° higher at 1600 feet per second, and 9° higher at 1700 feet per second.

These differences appeared to be the result of secondary flows, which caused the air leaving the impeller at different levels of energy input from hub to shroud to be distributed poorly for these speeds. For example, there were temperature gradients from the suction to the pressure surface at the outlet measuring station of the order of 15° at the speed of 1700 feet per second. Coverage at least equal to that given by the 36 total-pressure probes is necessary. Use of the temperatures in the outlet pipe brought the efficiencies as reported in reference 10 in line with those of the 40-vane diffuser for comparable pressure ratios and weight flows. Because of this factor, in addition to the agreement obtained between the outlet measuring station and outlet-pipe temperatures for the 40-vane diffuser, efficiencies based on the outlet-pipe temperatures are considered sufficiently accurate. 3628

Mach number. - Total pressure at the point and static pressure based on a linear variation of static pressure from one wall to the opposite wall were used to determine the Mach numbers for the surveys and for thermocouple Mach number corrections at the outlet measuring station.

Operating range. - The 40-vane diffuser was operated over a range of outlet meanline speeds (7-in. radius) from 700 to 1600 feet per second over a range of weight flows at each speed from incipient surge to choke. Above 1100 feet per second, refrigerated inlet air was used in order to avoid melting the diffuser vanes. The operation was repeated for two angles of blade setting at inlet, one at β of 59.3° and the other at 64.3°.

The range of operation for the 24-vane diffuser was from 1100 to 1700 feet per second at two inlet-angle settings of 61° and 66°. Otherwise, the operating conditions were the same as for the 40-vane diffuser.

Design Procedure

The vanes were designed to serve two functions: (1) to diffuse to a lower Mach number and (2) to turn the air to the axial-radial direction. The progressive increase in area required for diffusion in the 40-vane diffuser was provided by an average divergence angle of 10° between the pressure and suction surfaces. Because of the changing vane thickness from inlet to outlet, the divergence angle was less than 10° at inlet and was approximately 13° at outlet. The divergence for the 24-vane diffuser of 3.5° at inlet and 9° at outlet was between the inner and outer walls rather than from blade to blade.

The vanes were designed to match the average inlet flow condition at position 3 for 14 pounds per second and impeller-outlet meanline speed of 1400 feet per second. The vane camber and thickness distribution were determined from a prescribed velocity distribution on the vane surfaces by the method of reference 9. It was necessary to vary the velocity distribution in successive solutions for vane camber and thickness in order to make them compatible with the inlet conditions and the prescribed number of vanes. The number of vanes was prescribed in advance for the 40-vane diffuser only. The number of vanes for the 24-vane diffuser resulted from holding the velocity distribution approximately equal to that for the 40-vane diffuser and prescribing a divergence angle between the inner and outer walls. Since both diffusers have approximately the same velocity distribution on the vane surfaces, the blade loading per unit area is approximately the same; therefore, the 24-vane diffuser required fewer vanes because of the larger vane height. The larger angle β at inlet for the 24-vane diffuser than for the 40-vane diffuser resulted from the slightly higher vane height and thinner blades at inlet. The inlet thickness in the tangential direction is 0.097 for the 24-vane and 0.116 for the 40-vane diffuser. The setting up of the inlet geometry and the accompanying assumptions are presented in the following sections. Since the procedure for both sets of vanes is the same, only the procedure for the 40-vane diffuser is discussed except where noted.

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Diffuser-inlet conditions. - Before designing the diffuser vanes, the length of the vaneless transition section required to diffuse to a Mach number of approximately 0.9 and the flow angles across the passage at that point had to be determined. Only at the survey point $2\frac{5}{8}$ inches from the impeller outlet (position 3) was the absolute Mach number below 1. Mach number across the passage at position 3 is shown in figure 6 for weight flows of 13.0, 13.5, and 14.0 pounds per second at a speed of 1400 feet per second. Position 3 was chosen from consideration of Mach number, as the minimum radius at which to start the vaned section. It was evident from the surveys (fig. 6) that small changes in weight flow resulted in negligible changes in Mach number, as would be expected, inasmuch as the main flow component (tangential) remains approximately constant for a given impeller speed.

Inlet angle. - Surveys were made at three positions (fig. 1) to determine the angle of flow for weight flows of 13.5, 14.0, and 14.5 pounds per second at the speed of 1400 feet per second. The results are shown in figure 7. There is a variation in angle of approximately 12.5° at position 1 for all weight flows, whereas in the theoretical case for isentropic flow it is less than 3.0°. The 3.0° is based on the assumption of constant slip factor and constant axial-radial component of velocity from hub to shroud. Experimentally, there was a decrease in slip factor f_s (based on a temperature survey at position 1) of approximately 8.0

percent from hub to shroud, which resulted in less than 1° angle variation due to changes in tangential velocity. Therefore, the large angle variations shown in figure 7 are due primarily to variations in the axial-radial component of velocity. The variations are in the form of a fully developed boundary layer with maximum axial-radial velocity occurring at a distance from the outer wall that is approximately 80 percent of the distance from the inner to outer wall at position 1. For the 13-pound-per-second weight flow (fig. 7(a)), the boundary layer builds up rapidly along the inner wall from positions 1 to 3, causing a shift to boundary layers of approximately equal proportion attached to the opposing walls. However, for the 14-pound-per-second weight flow, the change in boundary-layer profile (fig. 7(c)) in going from positions 1 to 3 was small. 3628

The thick boundary layer attached to the outer wall is the result of a decelerating flow relative to the impeller along the shroud. An accelerating flow relative to the hub of the impeller resulted in a thinner boundary layer along the inner wall. The theoretical angle β into the diffuser, based on the isentropic design flow rate of 14.0 pounds per second with no boundary-layer allowance, is 55.4° at the inner wall and 58.1° at the outer wall. The minimum angle of 56.9° near the inner wall at position 1 in figure 7(c) lies between these two values. This agreement indicates that the only place for which boundary-layer allowance for the MFI-1B (ref. 4) resulted in the design velocity was at this point of minimum angle.

Number of vanes. - The number of vanes was chosen to given an average divergence angle of 10° between the pressure and suction surfaces. For a vane camber-line length of 4 inches and the vane height distribution provided by the vaneless diffuser walls shown in figure 1, approximately 40 vanes were required. An arbitrary thickness in the tangential direction at inlet of 0.116 inch (approximately 1/16-inch actual blade thickness) was chosen. This thickness caused an axial-radial blockage of 8.6 percent, the value used in determining the blade inlet angle and the Mach number just inside the vanes. The choice of a 10° divergence angle was based on results of tests with high-speed flow for diffusers of rectangular cross section with two parallel and two divergent walls as presented in reference 11. Total-pressure-loss coefficient, the ratio of mean total-pressure loss to mean dynamic head at inlet, was used in reference 11 as a basis of comparison of performance for diffusers of varying divergence angles. For the near-optimum divergence angle of 10.6° and outlet- to inlet-area ratio of 4, the pressure-loss coefficient was approximately 0.25 at an inlet Mach number of 0.65. For Mach numbers above 0.65, the loss coefficient rose sharply. A probable explanation is that, for Mach numbers above 0.65, smaller increases in area are required for diffusion. (Near Mach 1.0, a 2-percent change in area produces approximately a 15-percent change in Mach number.) Apparently, for all the diffuser configurations in reference 11, the area increase at

inlet produced a too-rapid drop in velocity for Mach numbers above 0.65, with consequent separation that choked the passage and produced a throat downstream of the geometrical throat. It was assumed herein that choking at design could be avoided by prescribing a velocity distribution with an approximately constant average gradient. With this assumption, the total-pressure-loss coefficient of reference 11 was extrapolated to give a value of 0.27 at the mean Mach number of 0.88 (from fig. 6).

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Inlet blade angle and Mach number. - In using the design method of reference 9, designing a single blade shape for the height provided between the inner and outer walls necessitated assigning a single value of blade angle from inner to outer wall at inlet. Choice of a single value was difficult in view of the large change in angle as measured at the survey positions. The design angle was taken herein as the arithmetic average for equal increments in height from the inner to outer wall at the flow rate of 14 pounds per second. The average angle is 61.5° (fig. 7(c), position 3).

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The Mach number just upstream of the inlet for design purposes was assumed equal to that at midpassage height (0.925), as shown in figure 6, in order to make some allowance for boundary-layer blockage. For the average angle of 61.5°, the through-flow component of Mach number is 0.442, as shown in figure 8(a). The blade blockage (equal to 8.6 percent) increases the through-flow Mach number to 0.483, with a resultant Mach number of 0.946 (fig. 8(b)). This value is equal to a Q value of 0.87, where Q is the ratio of the local air velocity to velocity of sound at stagnation conditions. The values of $Q = 0.87$ and inlet whirl Mach number of 0.813 were then used as the inlet conditions in applying the method of reference 9 to design the vanes. These resulted in the angle of 59.3° at inlet (fig. 8).

Velocity distribution. - In the design method of reference 9, the vane camber line, the vane thickness, and the number of vanes are determined from a prescribed velocity distribution on the vane surfaces and prescribed inlet and outlet conditions. In order to arrive at 40 vanes, it was first necessary to specify a preliminary velocity distribution that would give an acceptable approximate camber line and thickness distribution irrespective of the number of vanes. Then, to arrive at the desired value of 40 vanes, it was necessary to repeat the solution keeping the average velocity Q between blades constant and varying the differences in the pressure- and suction-surface values of Q in proportion to the ratio of the number of vanes obtained to the number of vanes desired. The final velocity distribution thus obtained is shown in figure 9. The velocity distribution for the 24-vane diffuser is also shown in this figure for comparison.

RESULTS AND DISCUSSION

Over-All Performance

The total-pressure ratio is plotted against equivalent weight flow for a range of speeds with adiabatic-efficiency contour superimposed in the upper parts of figures 11 and 12 for all vane configurations tested. Adiabatic efficiency and Mach number at the outlet measuring stations shown in figures 1 and 3 are plotted against equivalent weight flow in the lower parts of the figures. The weight-flow range from choke to surge with the vaned diffusers is much shorter than that obtained with the vaneless diffuser (fig. 10) at each operating speed. A maximum efficiency of 0.78 and pressure ratio of 3.25 were obtained at a speed of 1300 feet per second; at the design speed of 1400 feet per second, the maximum pressure ratio was 3.73 with a maximum efficiency of slightly over 0.77. Although the maximum efficiencies with both sets of vanes are not significantly different for the speeds of 1300 and 1400 feet per second, the Mach number at the outlet measuring station for the 40-vane diffuser for both angle settings is 0.20, whereas it is 0.30 for the 24-vane diffuser for both angle settings. Therefore, the 40-vane diffuser probably is the more efficient. A maximum pressure ratio of 4.90 at an efficiency of 0.75 was obtained at 1600 feet per second. The 1700-foot-per-second speed (for the 24-vane diffuser only) gave a pressure ratio of 5.4 at an efficiency of 0.71 (fig. 12(b)).

At speeds below 1300 feet per second, the relatively poor efficiency with vanes as compared with efficiencies in the vaneless diffuser probably resulted from having losses in the diffuser vanes that are out of proportion with over-all pressure ratio. For example, at a speed of 1100 feet per second for the 40-vane diffuser with β of 64.3° (fig. 11 (b)), the Mach number at the diffuser inlet is approximately 90 percent of that for 1400 feet per second. As a result, the losses incurred in diffusion are out of proportion with the over-all pressure ratio, which is only 75 percent of that at 1400 feet per second. In addition, operation at these speeds is in a range for which air flow in the impeller is unstable.

The over-all efficiencies based on ratio of total pressure in the outlet pipe to inlet total pressure are compared in figure 13 with the maximum efficiencies based on total pressure at the outlet measuring stations for both sets of vanes. The lower Mach numbers at the outlet measuring station for the 40-vane diffuser account for the smaller drop in efficiency in going from the outlet station to the outlet pipe.

For both sets of vanes, the performance characteristics at 1400 feet per second and above were noticeably improved by increasing the initial

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setting of the blade angle β by 5°. A detailed discussion of the results for two inlet-angle settings for each set of vanes is given in the following sections.

40-Vane Diffuser

Inlet $\beta = 59.3°$. - The performance of the 40-vane diffuser for the design inlet angle of 59.3° is shown in figure 11(a). At the design speed of 1400 feet per second, surging occurs just slightly above the weight flow corresponding to zero average angle of attack as obtained from the surveys (fig. 7). A large local angle of attack near the outer wall may have caused this surging above design weight flow, in which case warping of the vanes across the inlet to effect an angle variation that would conform with the measured angle of figure 7 should be investigated. At the time the vanes were designed, warping was considered, but it did not appear practical inasmuch as there was an 8° change in angle over a distance of only 5/32 inch adjacent to the outer wall. A smaller variation in angle of attack across the inlet can also be achieved by adjusting the vane angle for a lower weight flow. The surveys (fig. 7) show that the maximum variation in flow angle from inner to outer wall decreases with decreasing weight flow. In order to take advantage of this condition, the vanes were turned about their bases to effect an increase in the inlet angle β. The details of this change and the resulting performance are discussed in the next section.

Inlet $\beta = 64.3°$. - The angle β was increased by 5.0° to an angle of 64.3°, which corresponds approximately to a flow angle of 66.5° upstream of the vanes. If the trend of the change in angle variation with decreasing weight flow shown in figure 7 continues with further weight-flow decrease, then the angle of 66.5° falls approximately halfway between the minimum and maximum flow angles (estimated 63° min. and 70° max.) for a weight flow of 12.0 pounds per second.

The resulting performance is shown in figure 11(b). At the speed of 1400 feet per second, there was a point increase in efficiency over that for the diffuser with β of 59.3°. The surge weight flow dropped 1.7 pounds per second to approximately 12.5 pounds per second with only 1.1-pound-per-second drop in maximum weight flow. This resulted in a 100-percent increase in weight-flow range (from 0.6 to 1.2 lb/sec). There was an increase of approximately 4 points in efficiency at the speed of 1600 feet per second, with approximately 200-percent increase in weight-flow range (from 0.3 to 0.9 lb/sec).

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24-Vane Diffuser

Inlet $\beta = 61.0^{\circ}$. - The performance of the 24-vane diffuser with inlet angle β equal to 61.0° is shown in figure 12(a). The performance is comparable to that for the 40-vane diffuser with β equal to 59.3° (fig. 11(a)). The primary difference between the two is in range of weight flow. At the speeds of 1300 and 1400 feet per second, the maximum weight flows are approximately the same; however, the surge point at each of these speeds for the 24-vane diffuser is approximately 0.8 pound per second below that for the 40-vane diffuser. As a result, at 1400 feet per second, the weight-flow range of the 24-vane diffuser is approximately twice that of the corresponding 40-vane diffuser. The reason for the difference is not apparent. 3628

Inlet $\beta = 66.0^{\circ}$. - The inlet angle for the 24-vane diffuser could not be increased by rotating the vanes as in the case of the 40-vane diffuser; therefore, the angle was increased to 66° by bending the blade near the inlet. The performance results are shown in figure 12(b). There was very little change in efficiency except for a 2-point increase at speeds of 1600 and 1700 feet per second. The surge weight flow at the design speed of 1400 feet per second dropped 1 pound per second, giving an increase of approximately 65 percent in total range (from 1.2 to 2.0 lb/sec); but there was a 200-percent increase in range where the pressure ratio is approximately constant. This can be considered a 200-percent increase in the useful range. The maximum weight flow did not change significantly at speeds of 1400 feet per second and above, but the surge line shifted to a position almost identical to that for the 40-vane diffuser with $\beta \approx 64.3^{\circ}$ (fig. 11(b)). This indicates that bending of the blades near the inlet to effect changes in the surge line is permissible or even more desirable than turning the entire vane. Also, the resulting change in surge line for both turning and bending shows that the diffuser vane inlet angle is the determining factor for surge in the cases where the surge line with vanes falls at a higher weight flow than the impeller surge line.

Total-Pressure-Loss Coefficient

In order to compare the loss during diffusion with that obtained in the diffuser with two parallel walls and two divergent walls (ref. 11), the total-pressure loss was computed at the maximum efficiency point for the 40-vane diffuser ($\beta = 64.3^{\circ}$) at the tip speed of 1400 feet per second. The computed loss coefficient based on test results was 0.29, compared with the extrapolated value of 0.27 as previously mentioned for the diffuser of reference 11. The computations that resulted in the value of 0.29 were based on total pressures obtained in the surveys at position 3, total pressures at the outlet measuring station, and an average of the opposing wall static pressures at each station. The outlet- to inlet-area

ratio based on flow area at the two stations was 3, whereas it was 4 in the diffuser of reference 11. There was some uncertainty as to whether or not the total pressures measured at position 3 should be corrected for mixing losses caused by poor flow distribution at the impeller outlet.

In the vaneless-diffuser tests with a 12-inch-diameter impeller in reference 12, the friction coefficients based on total pressure decreased from approximately 0.008 at the 7-inch radius to approximately 0.004 at the 9-inch radius. This change is attributed to a decrease in the mixing losses as the flow smooths out in moving to the 9-inch radius (radius ratio of 1.5). The length of the flow path from the outlet of the impeller used in the investigation reported herein to position 3 is approximately equal to a distance that would give an average radius ratio of 1.38. As a check on the total pressure measured at position 3 (4.2 with efficiency of 0.88), the pressure ratio at the radius ratio of 1.5 in the vaneless diffuser (ref. 4) was adjusted to obtain the pressure ratio at position 3. The adjustment was for skin-friction losses only using a total-pressure friction coefficient of 0.0042 and the method of reference 12. The measured and adjusted values of pressure ratio were in agreement. These results indicate that these diffusers are approximately as efficient as that of reference 11.

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SUMMARY OF RESULTS

The design procedure and experimental test results for a range of impeller speeds are presented for two vaned diffusers of 24 and 40 vanes. Both sets of vanes were designed to give approximately the same velocity distribution on the vane surfaces. The diffusers were tested with the same impeller and gave the following results:

1. For both diffusers at the design speed of 1400 feet per second, the maximum pressure ratio was 3.73 with an efficiency of 0.77; a maximum efficiency of 0.78 and a pressure ratio of 3.25 were obtained at 1300 feet per second.

2. Pivoting the vanes of the 40-vane diffuser 5° and bending the vanes at inlet for the 24-vane diffuser 5° to match a reduced impeller weight flow at the vane inlet increased the weight-flow range from choke to surge at design speed by 100 percent (from 0.6 to 1.2 lb/sec) in the 40-vane diffuser and 65 percent (from 1.2 to 2.0 lb/sec) in the 24-vane diffuser.

3. A maximum pressure ratio of 4.90 at an efficiency of 0.75 was obtained at 1600 feet per second for both diffusers. A pressure ratio of 5.4 at an efficiency of 0.71 was obtained at 1700 feet per second in the 24-vane diffuser.

4. The weight-flow range with the vaned diffusers is much shorter than that obtained with the vaneless diffuser at each operating speed.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, May 18, 1955

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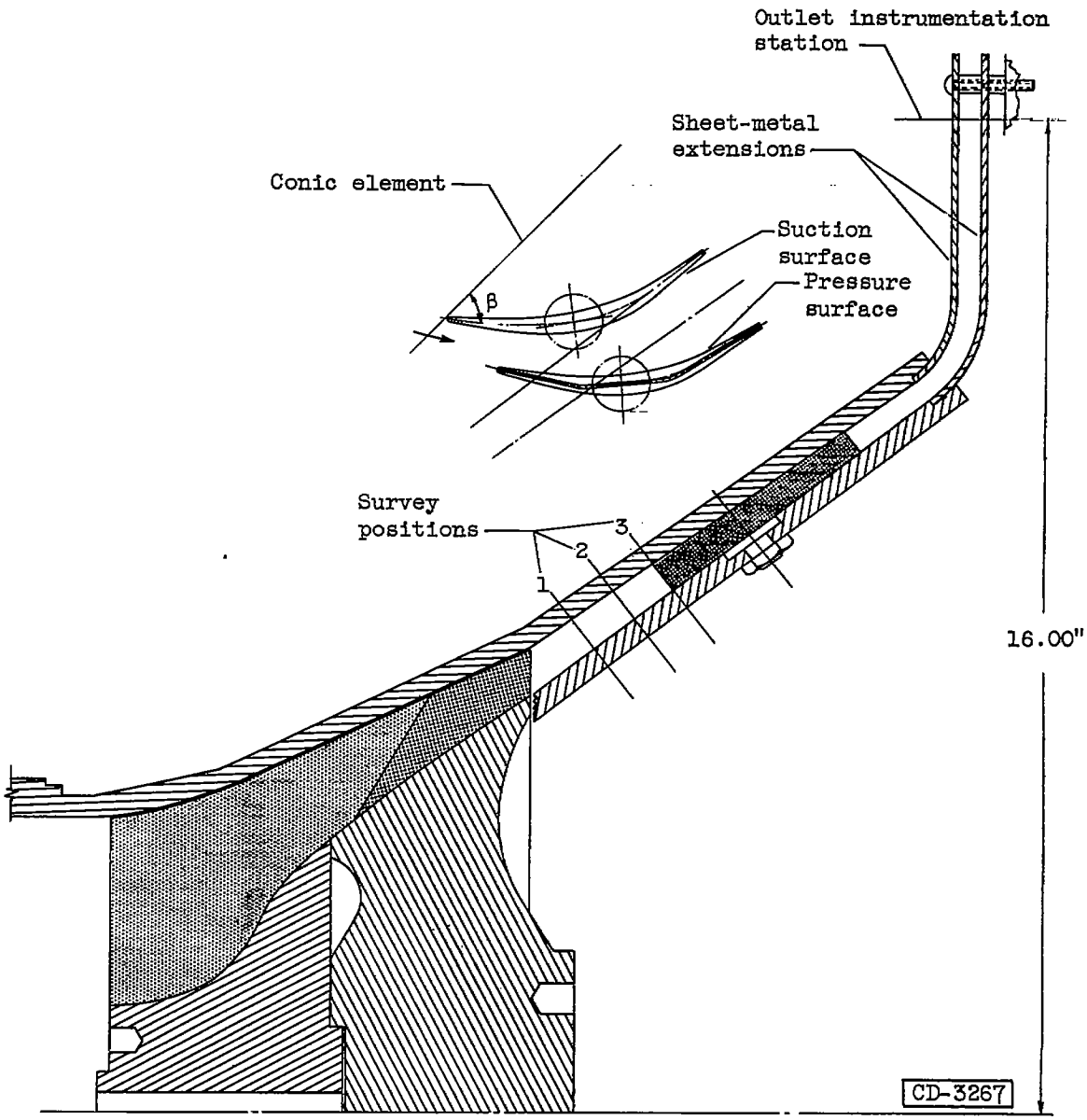


Figure 1. - Cross section of 40-vane-diffuser installation.

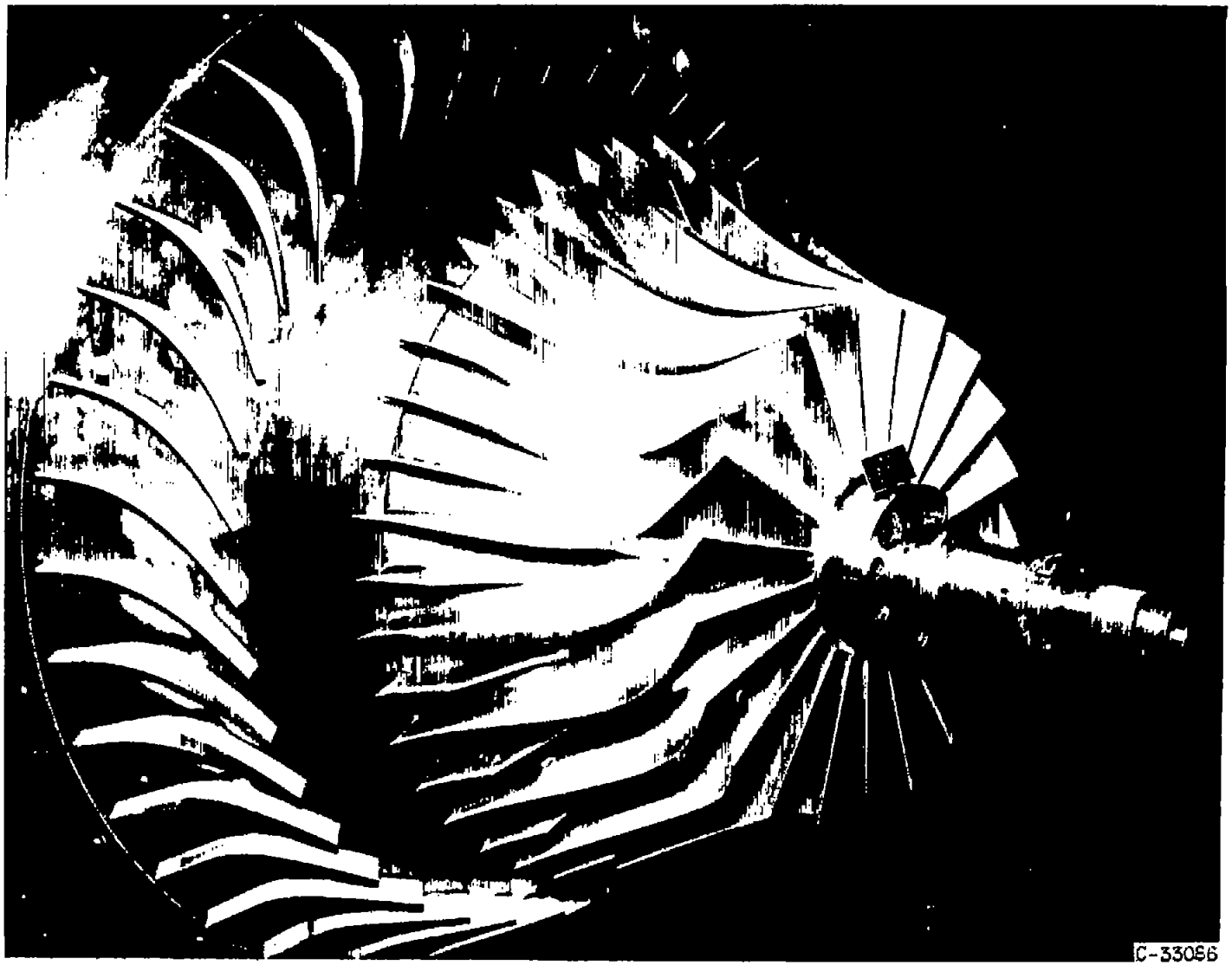


Figure 2. - Impeller model MFI-1B and 40-vane diffuser.

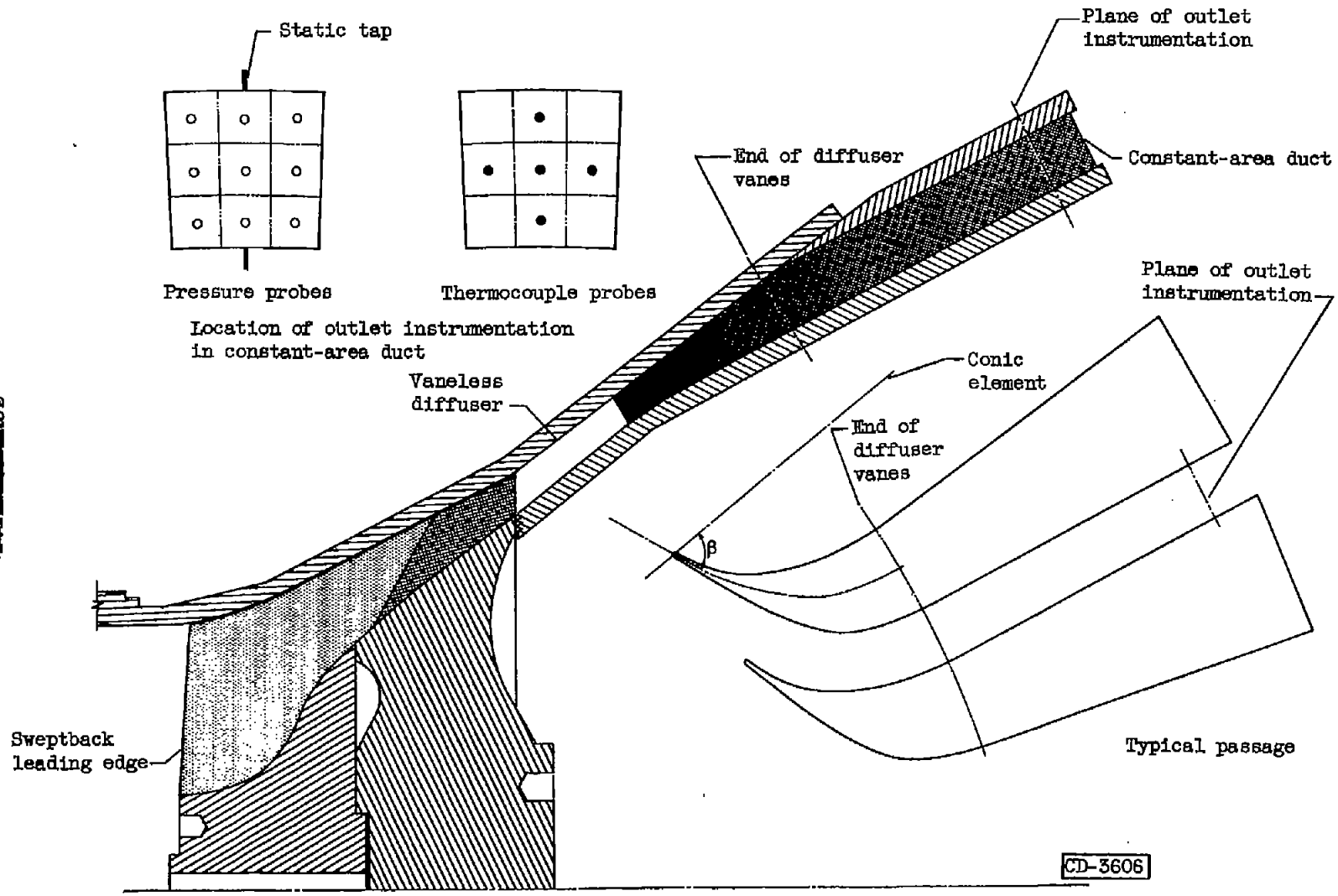


Figure 3. - Cross section of 24-vane-diffuser installation.

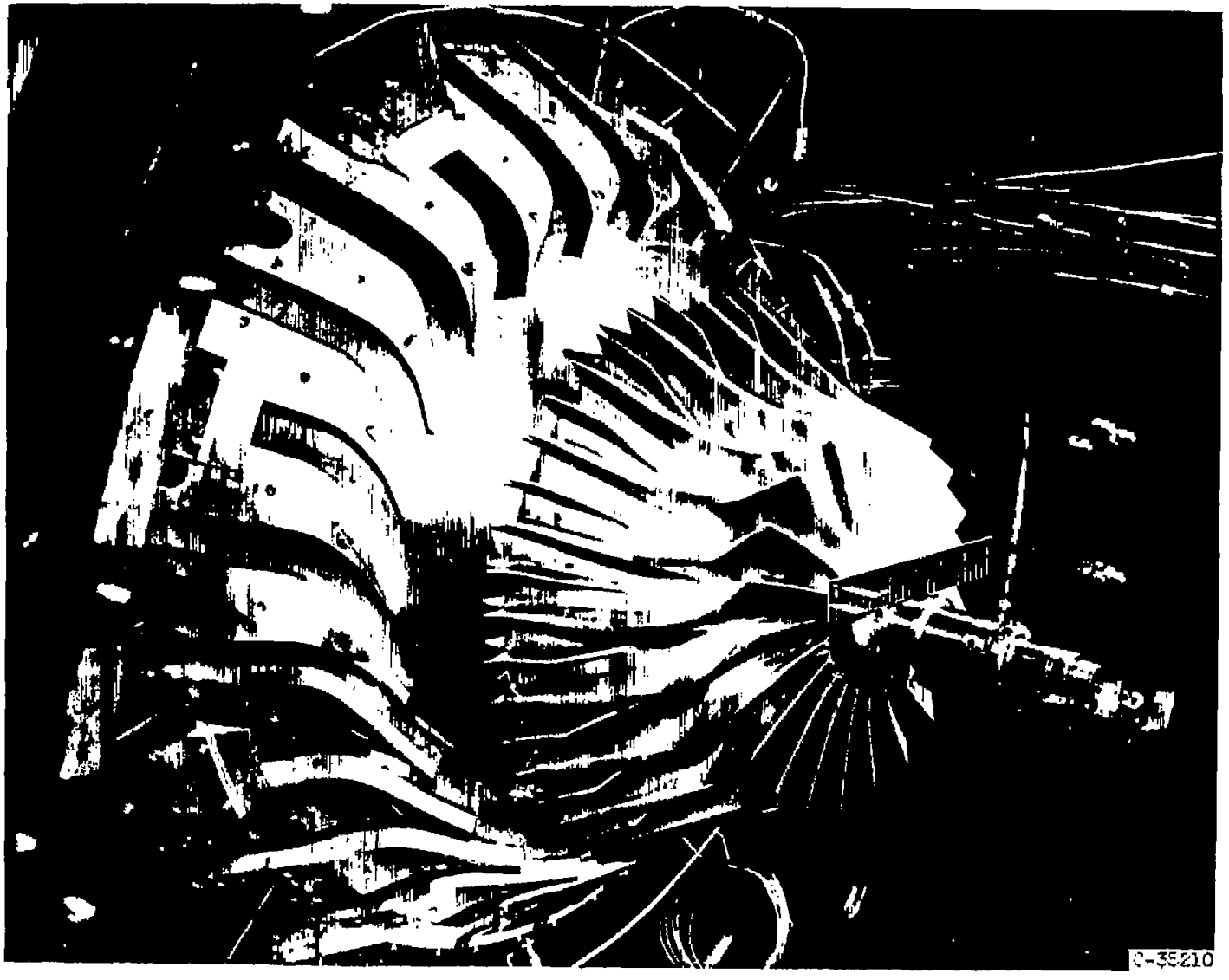


Figure 4. - Impeller model MFI-1B and 24-vane diffuser.

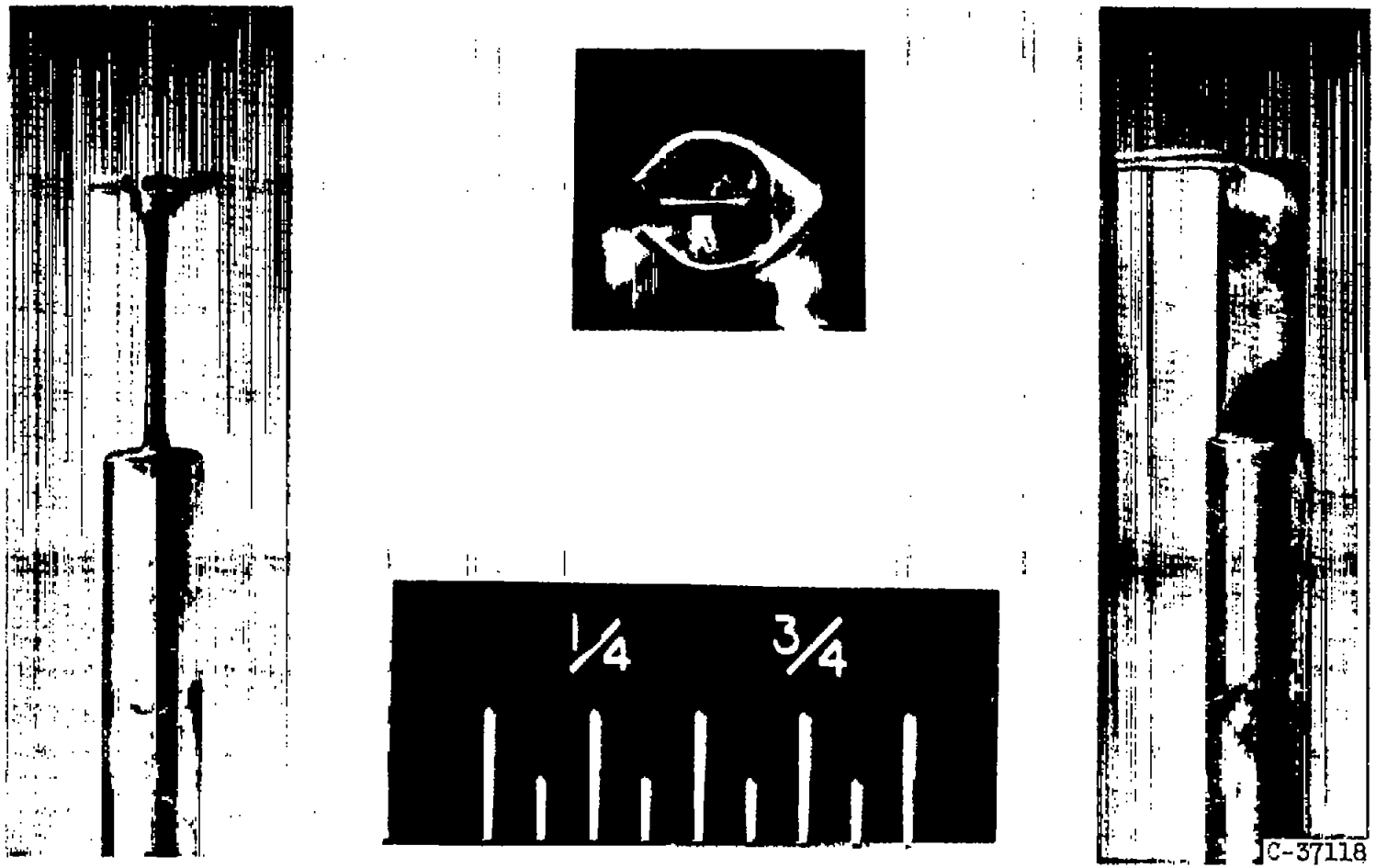


Figure 5. - Claw-type survey probe.

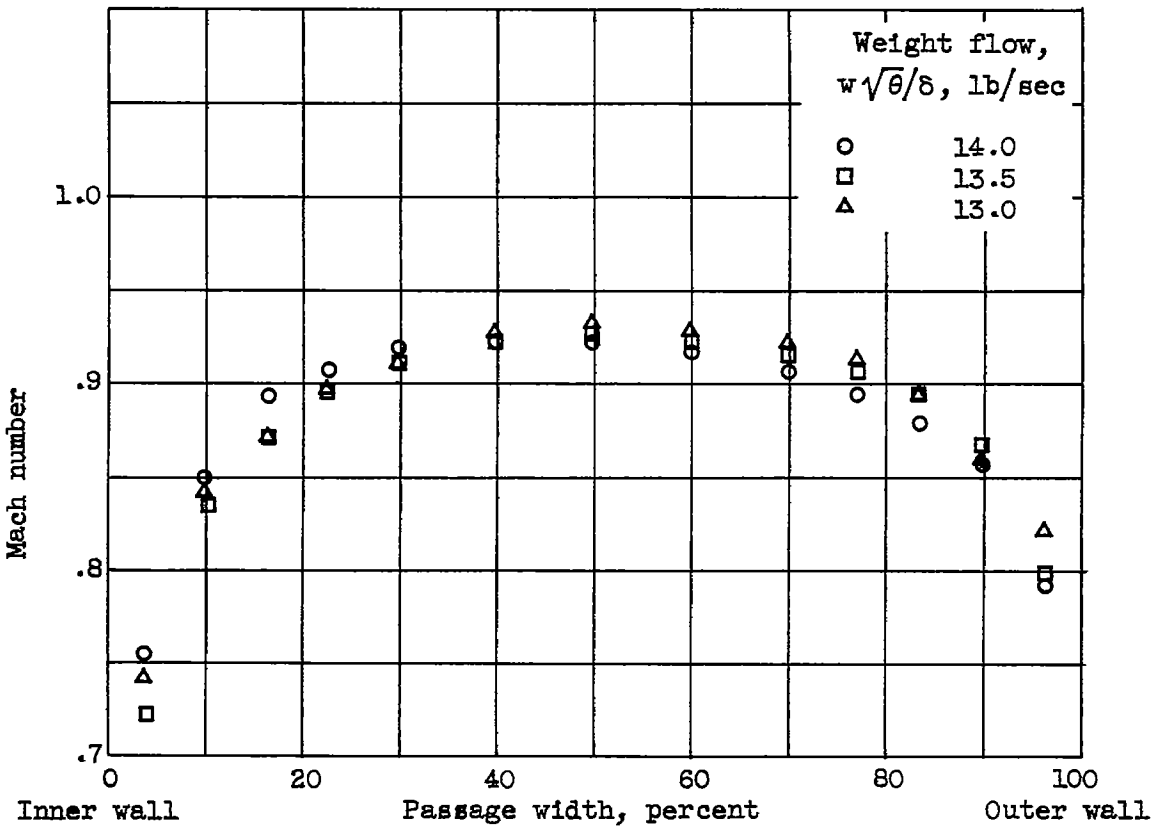
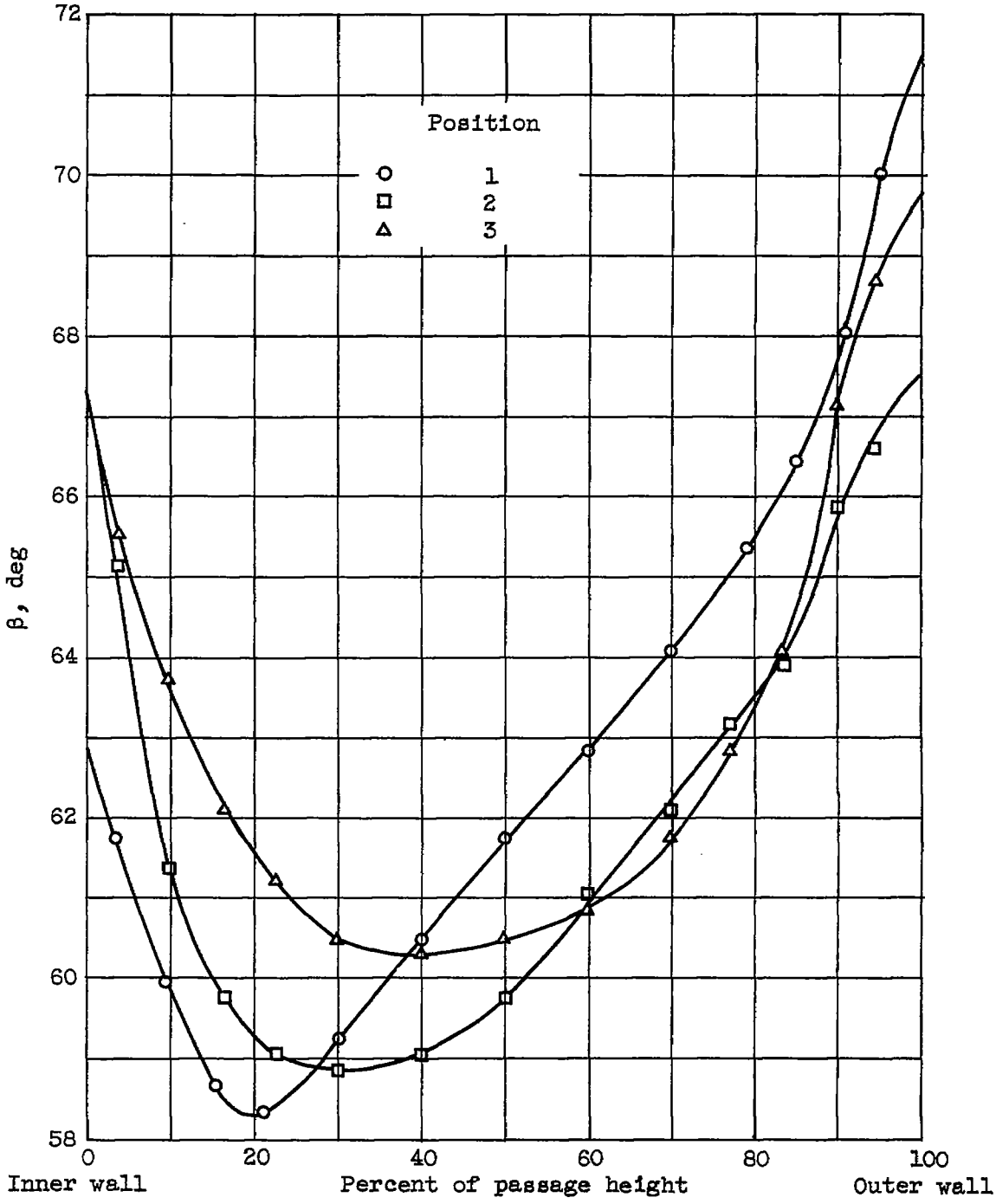
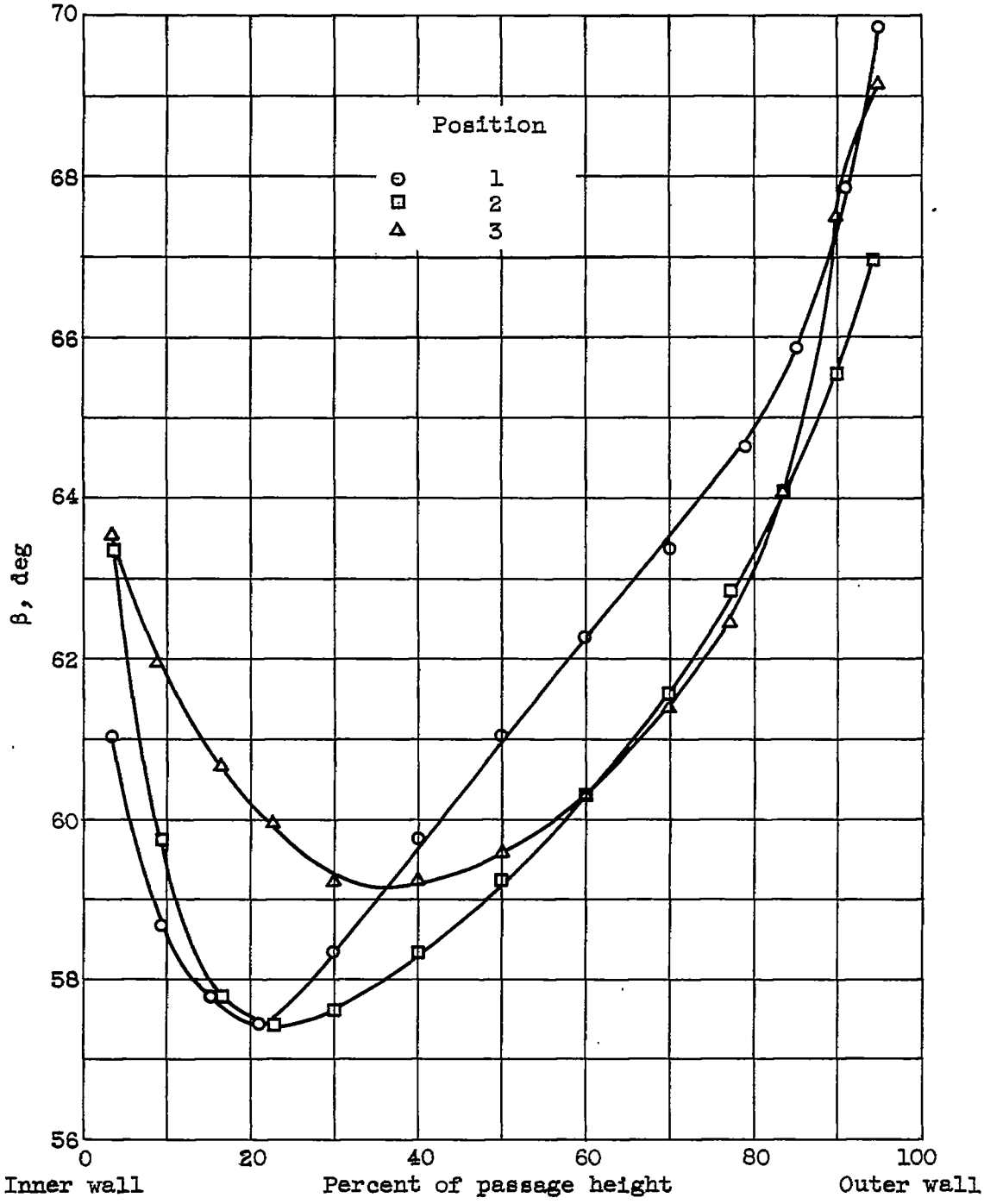


Figure 6. - Mach number distribution at position 3 for 1400 feet per second.



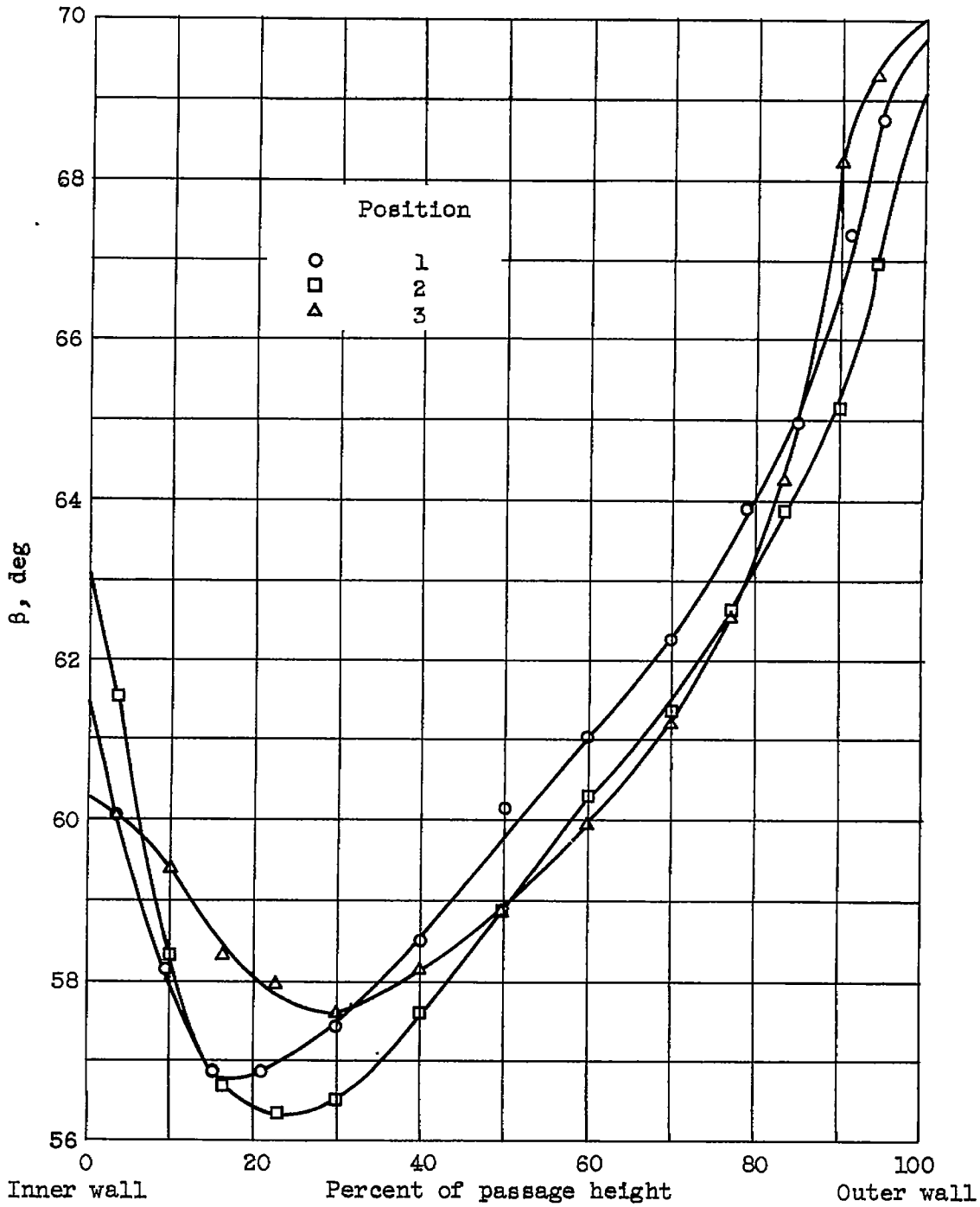
(a) Weight flow, 13.0 pounds per second.

Figure 7. - Outlet surveys at three positions for weight flows of 13.0, 13.5, and 14.0 pounds per second at 1400 feet per second.



(b) Weight flow, 13.5 pounds per second.

Figure 7. - Continued. Outlet surveys at three positions for weight flows of 13.0, 13.5, and 14.0 pounds per second at 1400 feet per second.



(c) Weight flow, 14.0 pounds per second.

Figure 7. - Concluded. Outlet surveys at three positions for weight flows of 13.0, 13.5, and 14.0 pounds per second at 1400 feet per second.

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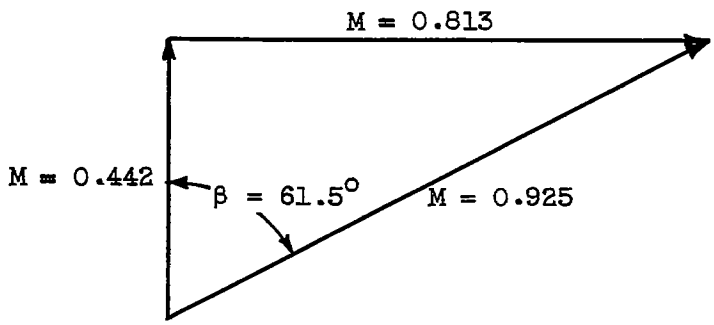
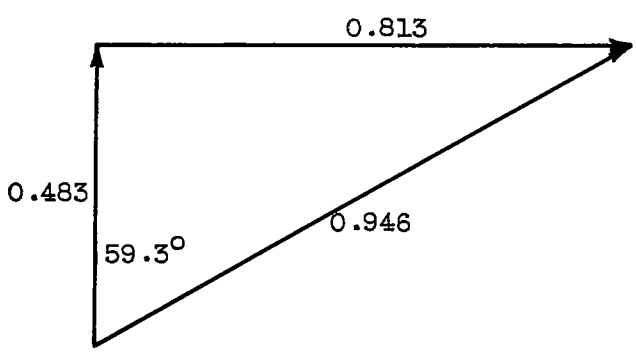


(a) Just upstream of vanes.

CK-4



(b) Just inside vanes.

Figure 8. - Mach number diagram at inlet to vanes of 40-vane diffuser.

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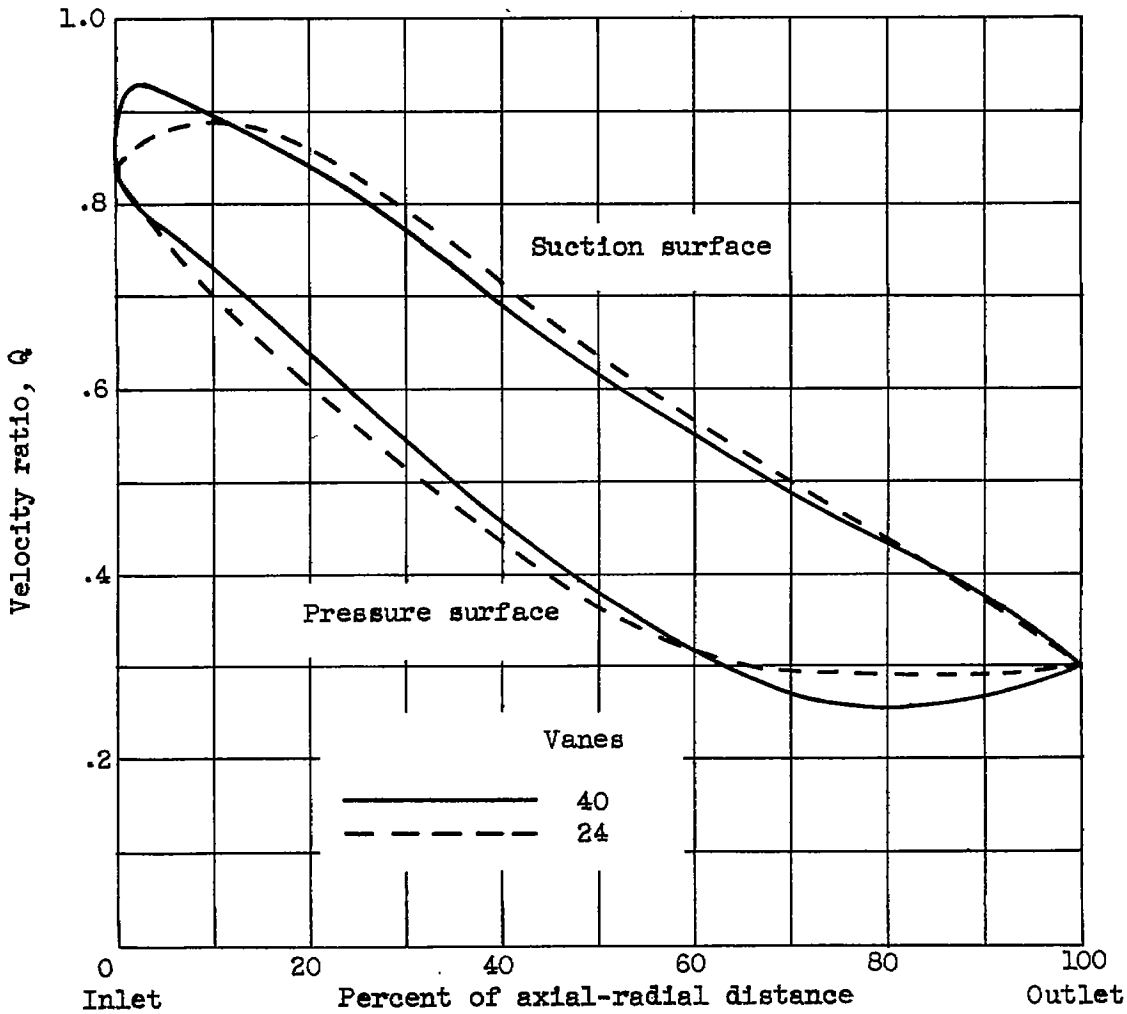

Figure 9. - Prescribed velocity distributions for design of diffuser vanes.

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CK-4 back

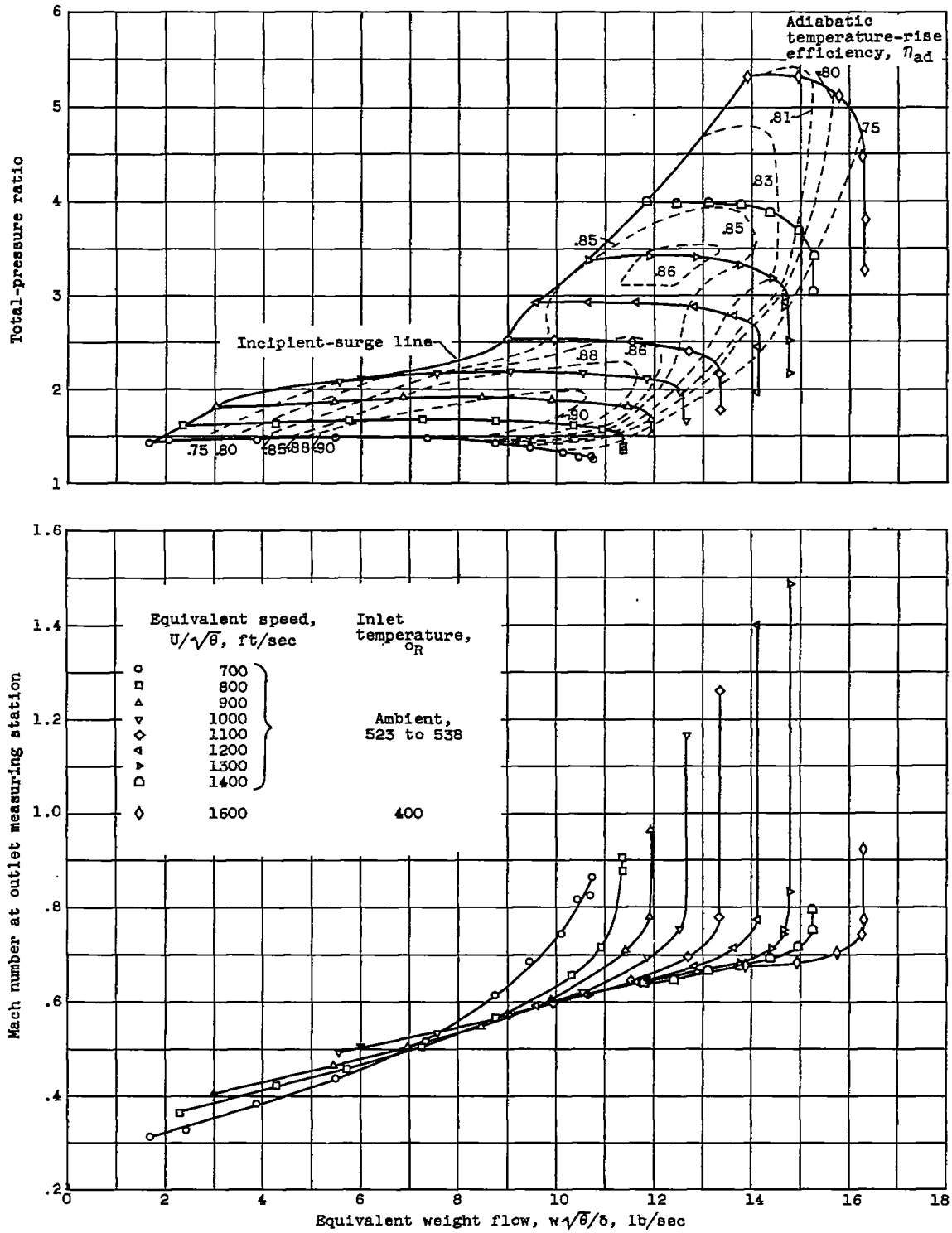

Figure 10. - Over-all performance characteristics of MFI-1B impeller at inlet-air pressure of 14 inches of mercury absolute.

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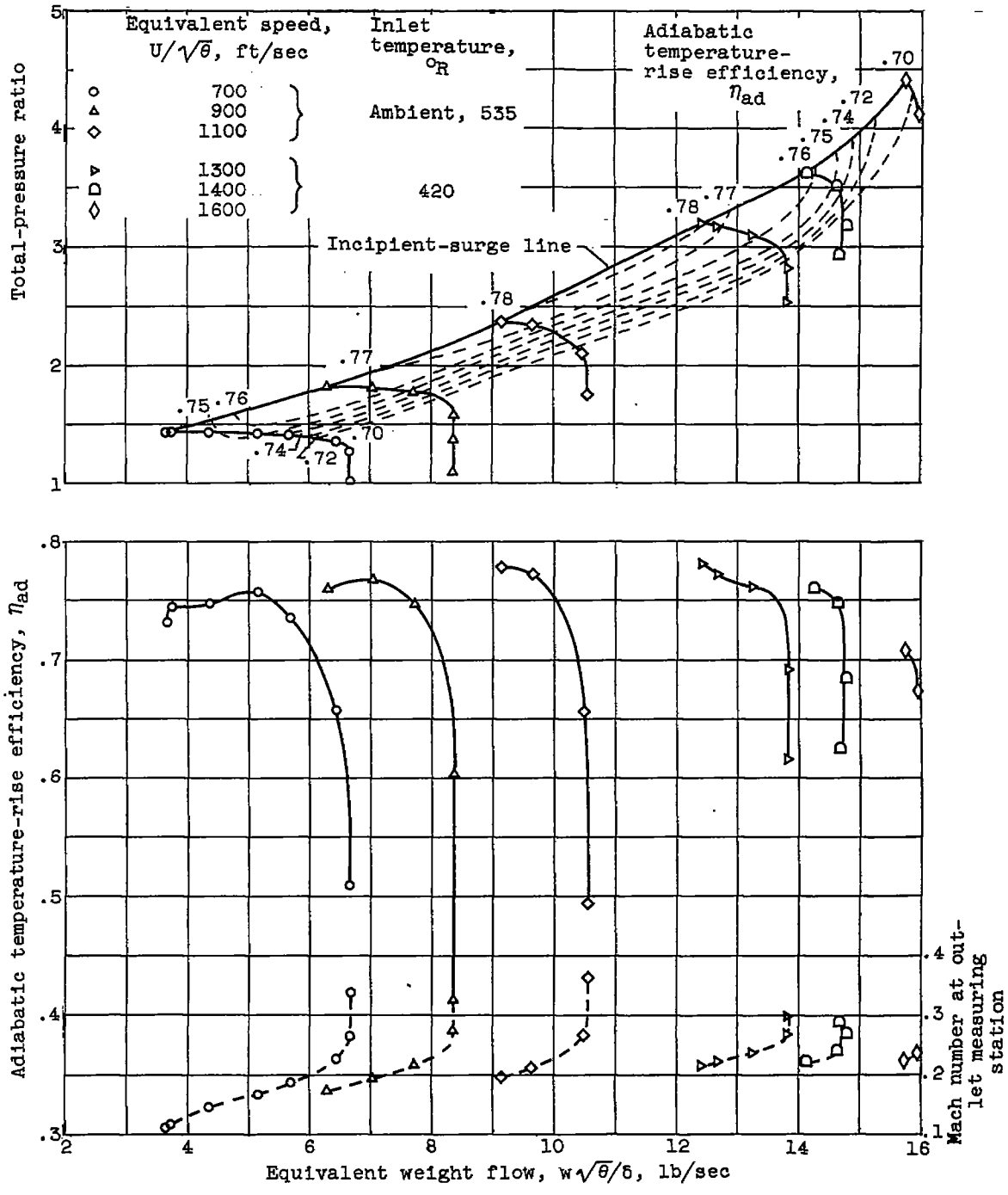
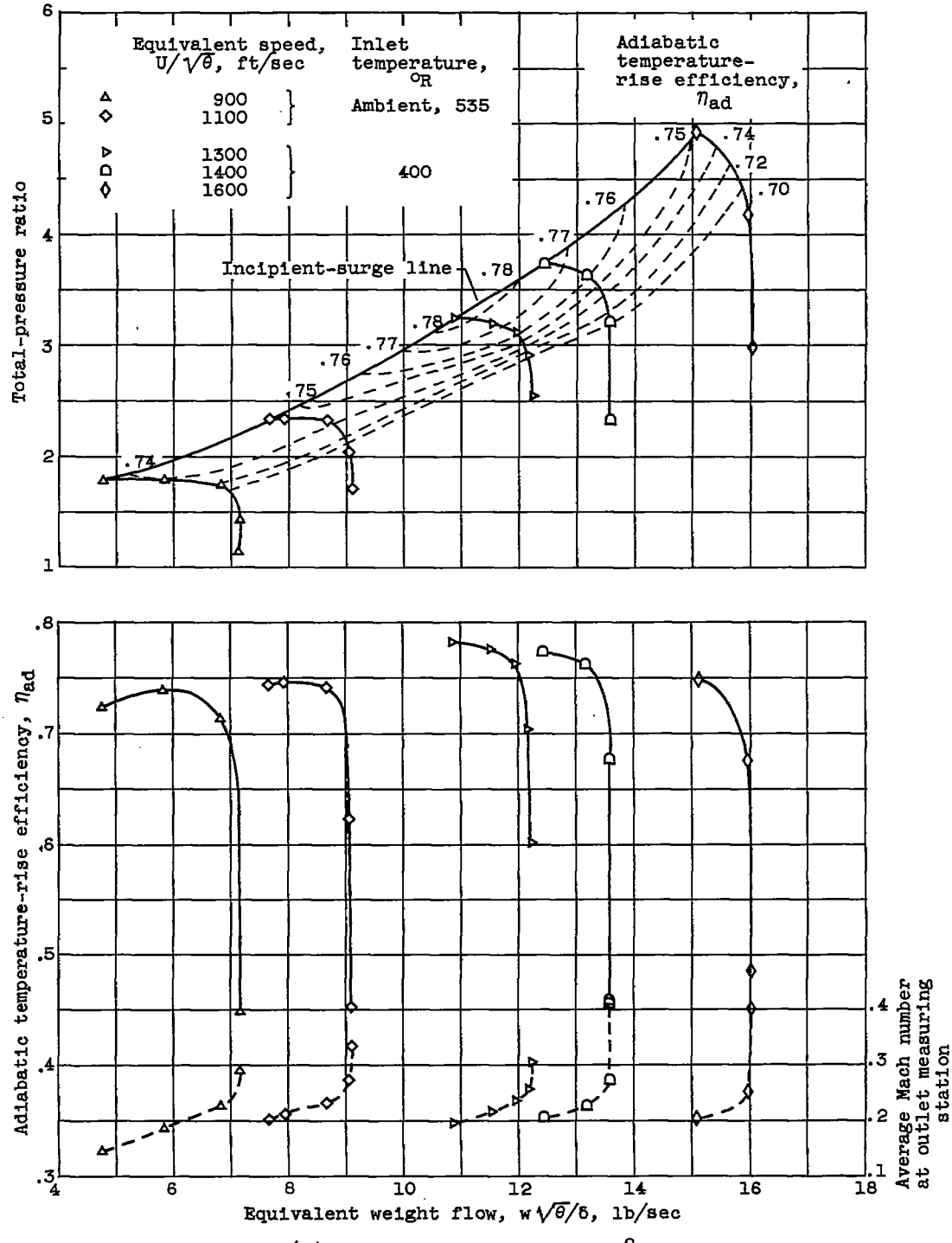


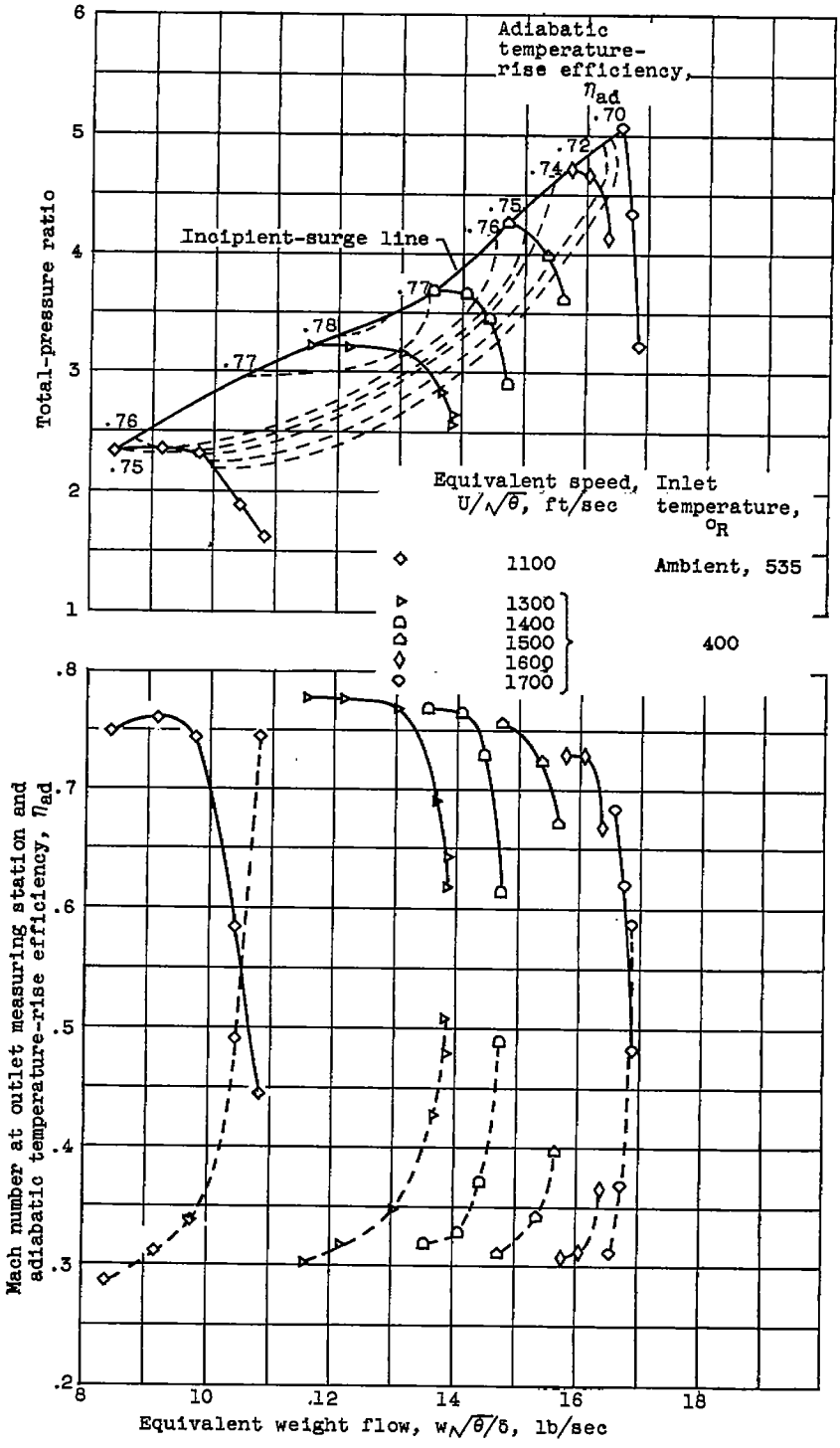
(a) Inlet vane angle, β, 59.3°.

Figure 11. - Over-all performance characteristics of MFI-1B impeller with 40-vane diffuser at inlet-air pressure of 14 inches of mercury absolute.



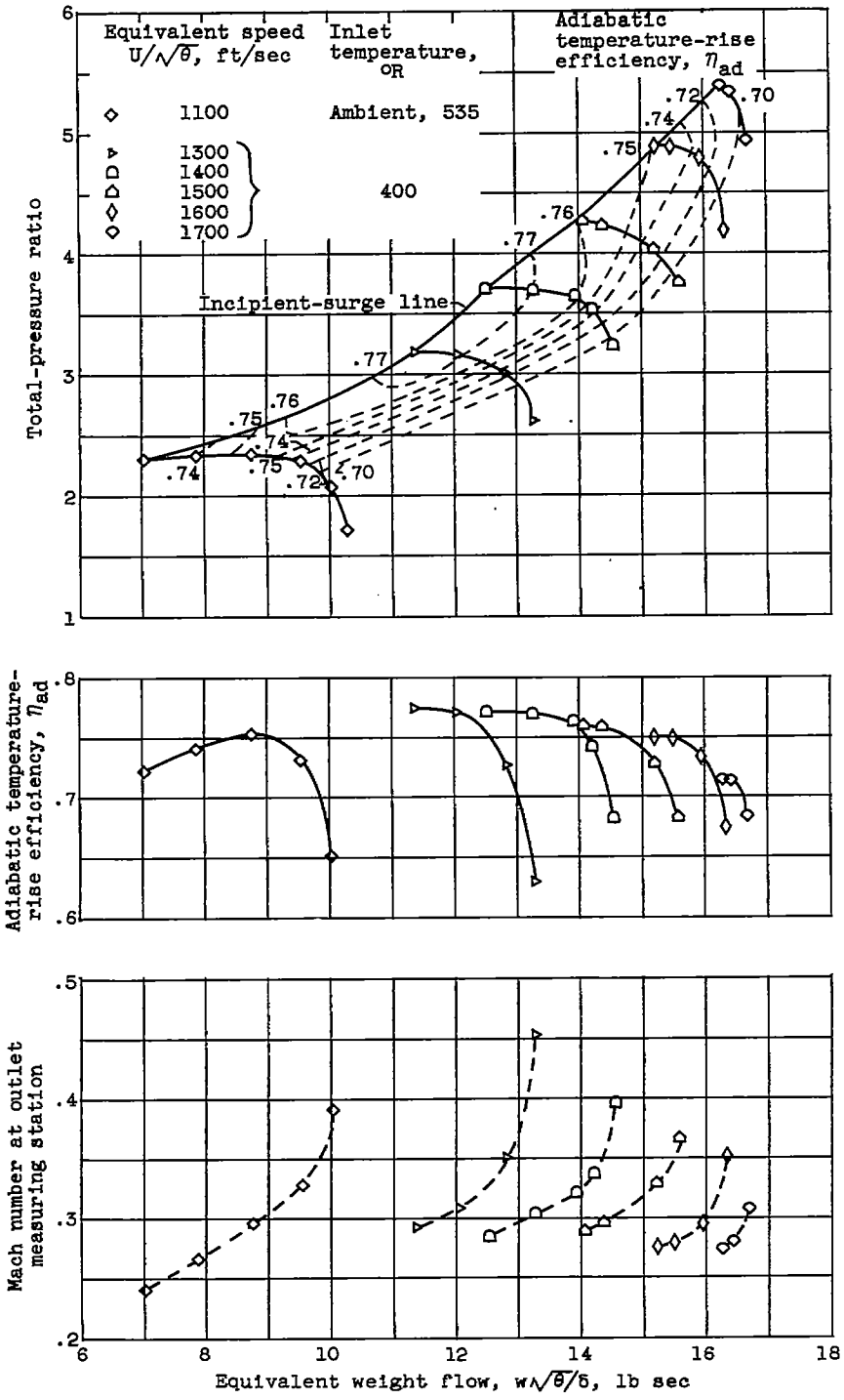
(b) Inlet vane angle, β, 64.3°

Figure 11. - Concluded. Over-all performance characteristics of MFI-1B impeller with 40-vane diffuser at inlet-air pressure of 14 inches of mercury absolute.



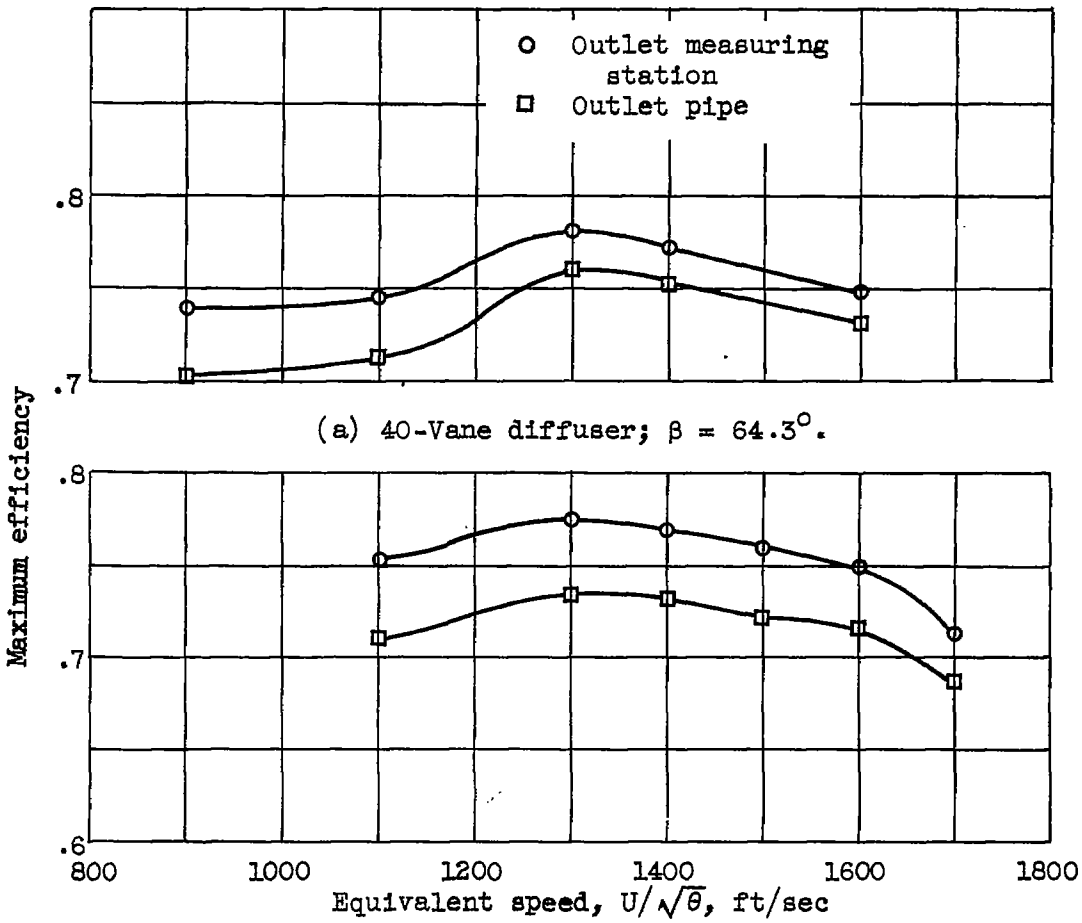
(a) Inlet vane angle, β, 61.0°.

Figure 12. - Over-all performance characteristics of MFI-1B impeller with 24-vane diffuser at inlet-air pressure of 14 inches of mercury absolute.



(b) Inlet vane angle, β, 66.0°.

Figure 12. - Concluded. Over-all performance characteristics of MFI-1B impeller with 24-vane diffuser at inlet-air pressure of 14 inches of mercury absolute.



(a) 40-Vane diffuser; $\beta = 64.3^{\circ}$.

(b) 24-Vane diffuser; $\beta = 66^{\circ}$.

Figure 13. - Comparison of efficiencies at outlet measuring station with those based on static pressure in outlet pipe.

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