JUN 11 1946

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1068

EFFECTS OF AXIAL-PLANE CURVATURE AND PASSAGE-AREA VARIATION ON FLOW CAPACITY OF RADIAL-DISCHARGE IMPELLER WITH CONVENTIONAL INLET BUCKETS

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Washington
May 1946

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SUMMARY

An experimental radial-discharge impeller was designed to have, by comparison with conventional impellers, a long radius of axial-plane curvature, a large axial depth, and a small inlet-blade root diameter. The impeller had conventional inlet-bucket bending and inlet and discharge diameters similar to an existing commercial impeller. This experimental impeller was tested as part of an investigation to improve the flow capacity of radial impellers. The performance effect of impeller passage area was investigated by tests of a series of three impeller front-shroud profiles. The impeller passages of profile 1 had constant area along the mean flow path taken in an axial plane. Profiles 2 and 3 had uniformly converging areas along the mean flow path.

The best performance for the experimental impeller was obtained with impeller blade profile 1, but the advantages over profiles 2 and 3 was apparently diminishing with increasing impeller tip speed. A comparison of specific flow capacity with that of a conventional radial-discharge impeller of approximately equal inlet and discharge diameters and with that of a mixed-flow impeller at an impeller tip speed of 1200 feet per second showed the experimental impeller to have a specific capacity 47 percent greater than the conventional radial-discharge impeller and 2 percent greater than the mixed-flow impeller. Most of the increase in volume flow capacity of this impeller, as compared with the conventional radial-discharge impeller, may be attributed to the gradual change in direction of the impeller passage resulting from the rear shroud profile of large radius of curvature and the large axial depth. A small increase in volume flow capacity was due to a slightly larger impeller-inlet annular area.

INTRODUCTION

The volume flow capacity of the radial-discharge centrifugal impeller must be improved to meet the increasing air-flow requirements of aircraft power plants. The volume flow capacity of a radial-discharge centrifugal impeller would be expected to depend, for fixed inlet and discharge diameters, on the entrance angle, the blade curvature that determines the rate of change of angular velocity of the air, the curvature of the passage in the axial plane, and the area and area variation along the passages. Parts of an investigation to determine the effects of the entrance angle of the blades and the blade curvature that determines the rate of change of angular velocity of the air have been reported in references 1 and 2.

The present investigation was made at the NACA Cleveland laboratory to find the effect of greatly decreasing the rate of turning the air in the axial plane as compared with conventional impellers, with consequent increase in axial depth, and to investigate the effect of changing the passage-area variation on this deep impeller. An experimental impeller was designed and constructed to have inlet and discharge diameters similar to an existing commercial production impeller but a greatly increased axial depth to permit more gentle curvature in the axial plane. The inlet buckets of the impeller blades conformed to conventional practice of die bending. The performance effect of varying the passage area was investigated by comparative tests of the experimental impeller with three profiles of the blades.

IMPELLER

The basic experimental impeller was designed and constructed to have a large axial depth to permit a gentle curvature in the axial plane with the inlet and discharge diameters approximately equal to those of an existing commercial production radial-discharge impeller. Both impellers had similar conventional inlet-blade bending with the entrance blade angle approximately the same in each case. Dimensions and details of the two impellers are as follows:

Impeller	Inlet-blade root diameter (in.)	Inlet-tip diameter (in.)	Outlet-tip diameter (in.)	Axial depth (in.)	Blade height at discharge tip (in.)	Number of impeller blades
Basic experimental	1.32	6.80	12.00	4.88	1.00	14
Conventional radial	1.97	6.80	12.23	2.44	.93	16

The small difference in outlet-tip diameters, which resulted from making the tip diameter of the basic experimental impeller 12.00 inches for convenience of test installation, was assumed to have negligible effects on the performance comparison of the two impellers. The reduced inlet-blade root diameter of the basic experimental impeller was used in conjunction with the increased axial depth to obtain the more gentle curvature in the axial plane as well as an increased inlet annular area. The small impeller hub resulting from the reduced inlet-blade root diameter necessitated a reduction in the number of impeller blades from 16 to 14.

A stub shaft with collar fastened to the rear of the impeller web by locking cap screws was used because the size of the hub prohibited use of a through shaft. The impeller was machined from an aluminum forging. The inlet buckets had a circular curvature and were formed by cold bending after machining. The inlet blade angle varied from 45° at the tip to 0° at the blade root; the angle was controlled by an axial taper along the leading edge of the blade. Figure 1 shows a photograph of the basic experimental impeller, designated impeller blade profile 1.

In order to determine the effect of convergence of impeller passage area on the performance of the impeller, three impeller blade profiles were designed and tested. For each blade profile, the rate of change in passage area was approximately constant along a mean flow path taken in an axial plane. The ratios of impeller-discharge passage area to impeller-inlet passage area (the passage areas in each case were normal to the mean flow line) were as follows:

Impeller blade profile	Ratio of discharge passage area to inlet passage area
1 (basic experimental)	1.00
2	.77
3	.65

The area ratio of impeller blade profile 1 was approximately the same as that of the commercial production impeller, which was used for comparative purposes in this report. Dimensions of the experimental impeller, including the three blade profiles, are shown in figure 2.

APPARATUS AND TEST PROCEDURE

Test setup. - The experimental impeller was tested in combination with a vaneless diffuser in a variable-component supercharger

test rig. The vaneless diffuser was 34 inches in diameter and its design was similar to diffusers that in previous tests had shown good pressure conversion over a wide range of operating conditions. The same diffuser was used with each of the three impeller blade profiles; but in each case the diffuser throat section was changed to form a smooth contour between the impeller front shroud and the front diffuser wall. The variable-component supercharger test rig was as described in reference 3 except that a flat-plate front collector cover was used for simplification of instrument installation. The experimental impeller was driven by a 1000-horsepower aircraft engine in conjunction with a speed-increaser gear.

Instrumentation. - Temperature and pressure measurements were made according to standards in references 3 and 4 wherever applicable. All air temperatures were measured with calibrated iron-constantan thermocouples and a potentiometer. Total-pressure measurements in the inlet and discharge ducts were made using total-pressure tubes of 0.093-inch outside diameter and 0.067-inch bore. Static wall taps of 0.020-inch bore were used in the inlet and discharge ducts. Other pressure measurements taken included static pressures on the front and rear walls of the vaneless diffuser at a radial distance of 0.38 inch from the impeller discharge tip.

Air-flow and pressure regulation was provided by throttle valves of the butterfly type in both the inlet and discharge ducts. A large orifice tank with a thin-plate orifice at the entrance to measure the quantity of air flow (reference 5) was attached to the inlet duct.

The desired constant speed was maintained with a speed strip and a stroboscopic light operated on 60-cycle current. An electric counter and a stop watch were frequently used to check the speed.

Tests. - Tests were conducted according to the procedure in references 3 and 4; all tests were made using inlet air at room temperature. For each constant tip speed, the volume flow was varied in a number of steps from wide-open throttle to pulsation, except at the flow cut-off point for the higher tip speeds where the insufficient driving power necessitated a small reduction in mass flow before the desired speed could be obtained. In the flow cut-off range, the maximum volume flow obtained was not affected by a small reduction in mass flow. For all throttle settings except wide-open throttle, a constant outlet total pressure of 10 inches of mercury above atmospheric pressure was maintained. Tests were made at impeller tip speeds of 800, 1000, and 1200 feet per second for each of the three impeller blade profiles.

COMPUTATIONS

Performance with vaneless diffuser and variable-component supercharger. - Computations of over-all adiabatic efficiency for the unit composed of the impeller, the vaneless diffuser, and the variable-component supercharger collector were made in accordance with reference 4. The flow parameters - corrected volume flow $Q_{1t}/\sqrt{\theta}$ and specific capacity $Q_{1t}/\sqrt{\theta}\,D_2^2$ - and the Mach number parameter $U/\sqrt{\theta}$ were computed according to the method of reference 6, where Q_{1t} is the volume flow at the total pressure at the impeller inlet, U is the actual impeller tip speed, D_2 is the impeller-discharge tip diameter, and θ is the ratio of the absolute inlet-air temperature to NACA standard sea-level air temperature. Two additional flow parameters calculated were corrected volume flow per unit impeller-inlet tip diameter squared $Q_{1t}/\sqrt{\theta}\,D_1^2$ and corrected volume flow per unit impeller-inlet annular area $Q_{1t}/\sqrt{\theta}\,A_1$.

Impeller performance. - Adiabatic efficiency η_{ad} was calculated at a point in the diffuser passage 0.38 inch from the impeller discharge for each test using calculated total pressures. The total pressure at this point in the diffuser (6.38-in. radius) was determined from the computed dynamic pressure and the measured static pressure. The calculations were made on the assumptions that there was no change in total temperature of the air from the impeller tip through the insulated system to the measuring station in the discharge duct, that the friction loss between the impeller discharge and the measuring point was negligible, and that the velocity was constant across the diffuser passage. The velocity and the density of the air were found from the measured static pressure, the continuity of flow, and the foregoing stated assumptions.

RESULTS AND DISCUSSION

Comparative Volume-Flow Characteristics of Basic Experimental Impeller

Impeller volume-flow characteristics were analyzed on a volume-flow-capacity basis for the given impeller size; these characteristics are shown by specific capacities and volume flows per unit impeller-inlet tip diameter squared. The flow-capacity effects of impeller-inlet annular area and impeller axial-plane curvature are shown by comparative volume flows on a basis of specific capacity and flow per unit impeller-inlet annular area.

Comparison with conventional radial-discharge impeller of approximately equal inlet and discharge diameters. - A comparison of the performance of the basic experimental impeller (profile 1) with that of the conventional radial-discharge impeller on a specific-capacity basis is shown in figure 3. Both impellers were tested with the same type of vaneless diffuser under similar test conditions. The performance data for the conventional radial-discharge impeller were taken from unpublished NACA tests. The basic experimental impeller at a tip speed of 800 feet per second had a 59-percent advantage in specific capacity over the conventional radial-discharge impeller and a 5-point higher peak adiabatic efficiency. At a tip speed of 1200 feet per second, the basic experimental impeller had a 47-percent advantage in specific capacity over the conventional radial-discharge impeller and a 1-point lower peak adiabatic efficiency. The basic experimental impeller and the conventional radial-discharge impeller were compared on a basis of corrected volume flow per unit impeller-inlet tip diameter squared (fig. 4). For the given inlet tip diameter, the same for both impellers, the basic experimental impeller had a 54-percent advantage in flow capacity at a tip speed of 800 feet per second and a 41-percent advantage at a tip speed of 1200 feet per second. The specific-capacity comparison and the comparison on a basis of corrected volume flow per unit inlet tip diameter squared showed the superiority in volume flow capacity of the basic experimental impeller.

In order to differentiate between the effect of the reduced inlet-blade root diameter, and the rear shroud of large radius of curvature and the large axial depth, the basic experimental impeller and the conventional radial discharge impeller were compared on a basis of corrected volume flow per unit of impeller-inlet annular area (fig. 5). As shown in figure 5, the basic experimental impeller had a greatly increased volume flow per unit of impeller-inlet annular area over the speed range, thus indicating that a large percentage of the increase in volume flow capacity experienced with the basic experimental impeller may be attributed to the gradual change in direction of the impeller passage resulting from the rear shroud profile of large radius of curvature and the large axial depth and that only a small part of the increase in volume flow capacity was due to a slightly larger impeller-inlet annular area.

Comparison with a mixed-flow impeller. - The flow capacity of the basic experimental impeller was also compared with that of a mixed-flow impeller, which was tested with the same type of vaneless diffuser under similar test conditions. The performance data for the mixed-flow impeller were taken from unpublished NACA test data. The comparison was on a basis of specific capacity, corrected volume flow per unit impeller-inlet tip diameter squared, and corrected

volume flow per unit of impeller-inlet annular area at an impeller tip speed of 1200 feet per second. (See figs. 3, 4, and 5, respectively.) The basic experimental impeller had a 2-percent advantage in specific capacity, a 71-percent advantage on a basis of corrected volume flow per unit impeller-inlet tip diameter squared, and a 43-percent advantage on a basis of corrected volume flow per unit of impeller-inlet annular area. The mixed-flow impeller had higher adiabatic efficiencies.

Effect of Passage-Area Variation on Performance of the Basic Experimental Impeller

A comparison of over-all adiabatic efficiency and specific capacity (equivalent in this case to corrected volume flow) for the basic experimental impeller (profile 1) and profiles 2 and 3 is shown in figure 6. Profile 1 attained a higher efficiency and a greater volume flow at speeds of 800 and 1000 feet per second. At 1200 feet per second, profiles 1 and 2 had about the same volume flow capacities with profile 3 having a slightly lower volume flow capacity. At this speed, profiles 2 and 3 had a 2-point advantage in peak over-all efficiency over profile 1.

An impeller-performance comparison of the three profiles based on adiabatic efficiencies at the impeller discharge is shown in figure 7. Profile 1 maintained higher adiabatic efficiencies over the flow range for impeller tip speeds of 800 and 1000 feet per second. At the highest speed, 1200 feet per second, profiles 1 and 2 had about the same performance curves with profile 3 having decreased adiabatic efficiencies over the flow range. The comparison of impeller efficiencies for the three impeller blade profiles showed the basic experimental impeller (profile 1) to have a larger volume flow capacity and a higher adiabatic efficiency over the range of speeds tested, although the advantages were apparently diminishing with increasing impeller tip speed. The effect of the passage-area changes on flow capacity, however, was slight as compared with the flow-capacity increase that was apparently due to decreased rate of turning in the axial plane.

For all three impeller blade profiles, peak adiabatic efficiencies occurred in the lower part of the flow range for the speeds tested. A comparison of the over-all and impeller efficiencies shows that the diffuser losses decrease for the same volume flow with smaller discharge passage area of the impeller. This condition may be attributed to mixing losses in the diffuser passage at the impeller discharge, which appear to be inversely related to the ratio of the impeller relative dynamic pressure to the impeller static pressure,

An examination of diffuser lampblack flow patterns and static pressure gradients along the diffuser walls indicated that no flow separation occurred in the diffuser.

Over-all Performance Characteristics of the Basic Experimental Impeller

In order to give more complete information as to the performance characteristics of the basic experimental impeller, figure 8 shows the over-all total-pressure ratios for the corrected impeller tip speeds of 792, 980, and 1181 feet per second plotted against corrected volume flow and specific capacity with curves of over-all adiabatic efficiencies superimposed.

SUMMARY OF RESULTS

Tests of an experimental impeller with three passage-area variations and comparison of performance of the basic experimental impeller with that of a conventional radial-discharge impeller of approximately equal diameters and with a mixed-flow impeller established the following results:

1. The basic experimental impeller has a greatly increased volume flow capacity as compared with the conventional radial-discharge impeller.

2. Most of the increase in volume flow capacity of the basic experimental impeller, as compared with the conventional radial-discharge impeller, may be attributed to the gradual change in direction of the impeller passage resulting from the rear shroud profile of large radius of curvature and the large axial depth. A small increase in volume flow capacity was due to a slightly larger impeller-inlet annular area.

3. The specific capacity of the basic experimental impeller was equivalent to that of the mixed-flow impeller.

4. The best performance for this experimental impeller was obtained with the impeller passage having constant area along the mean flow path taken in an axial plane. The advantages as compared with the impeller passages having increased passage convergence, however, was apparently diminishing with increasing impeller tip speed.

CONCLUSION

The volume flow capacity of a conventional-type radial-discharge impeller may be greatly increased by utilizing a rear shroud profile of large radius of curvature and a large impeller axial depth to give a gradual change in direction through the impeller passage.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, January 7, 1946.

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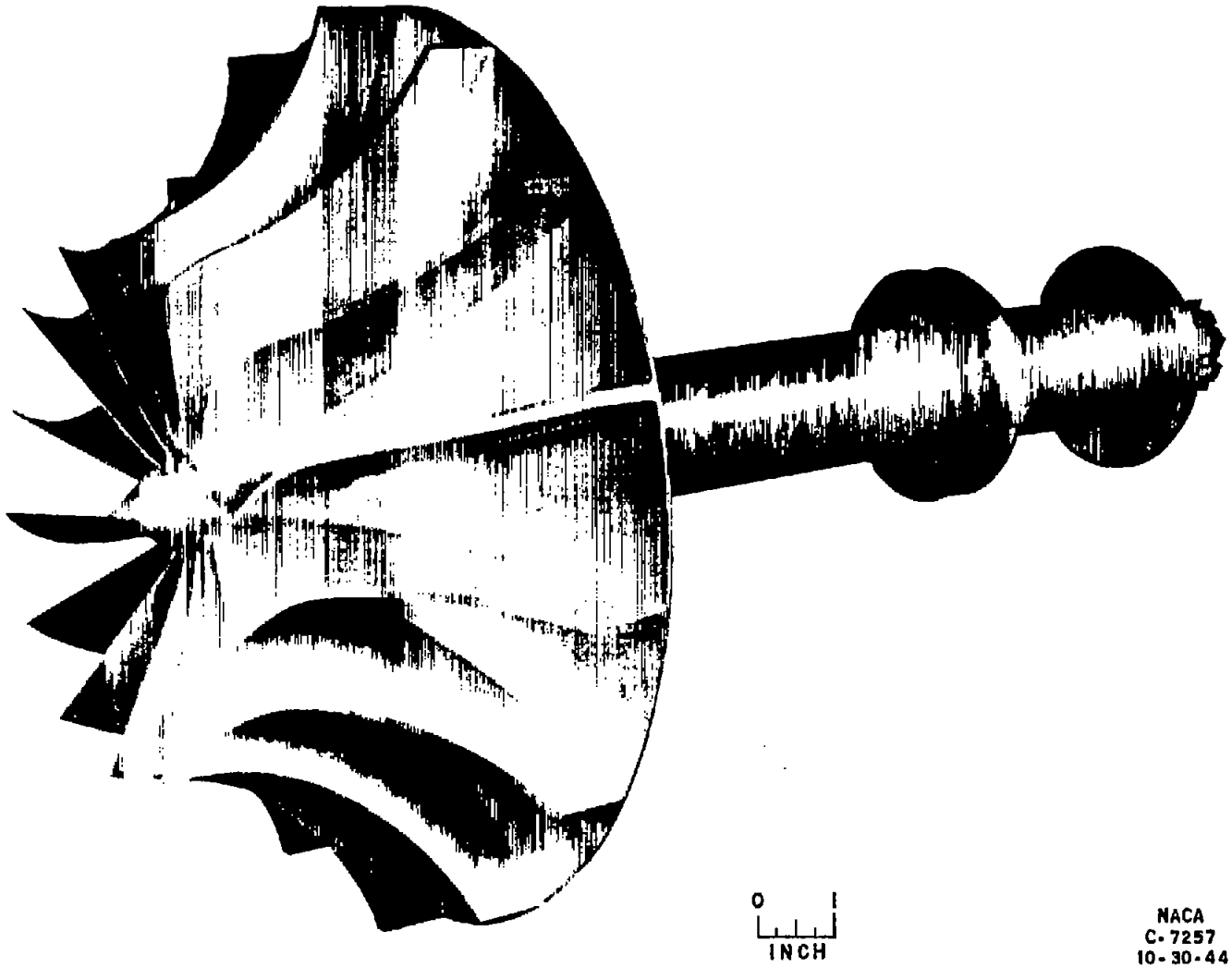

Figure 1. - Basic experimental radial-discharge impeller having a large axial-plane curvature and constant passage area along a mean flow path taken in an axial plane.

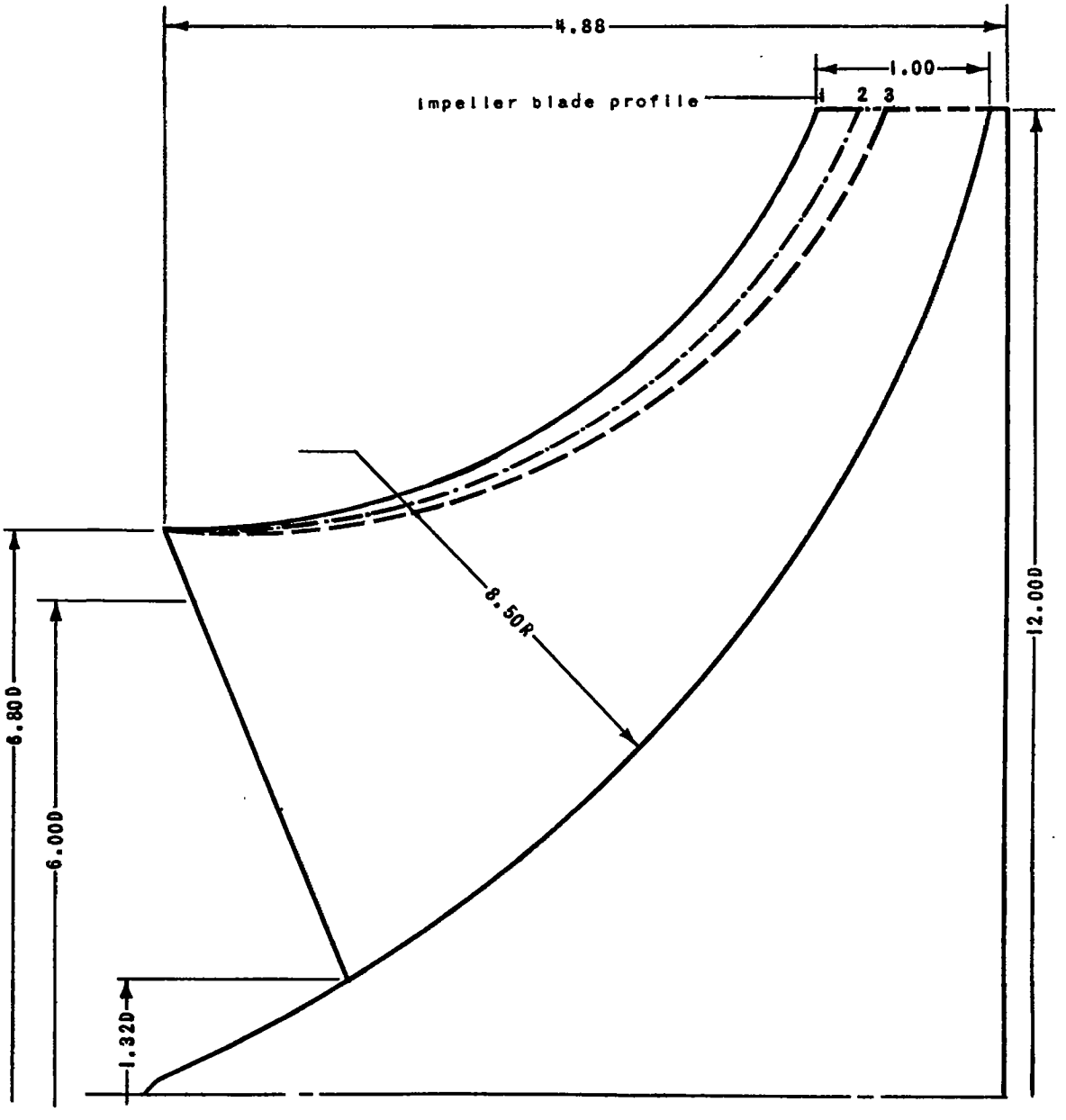

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Figure 2. - Design details of experimental radial-discharge impeller showing a comparison of passage variation for three impeller blade profiles.

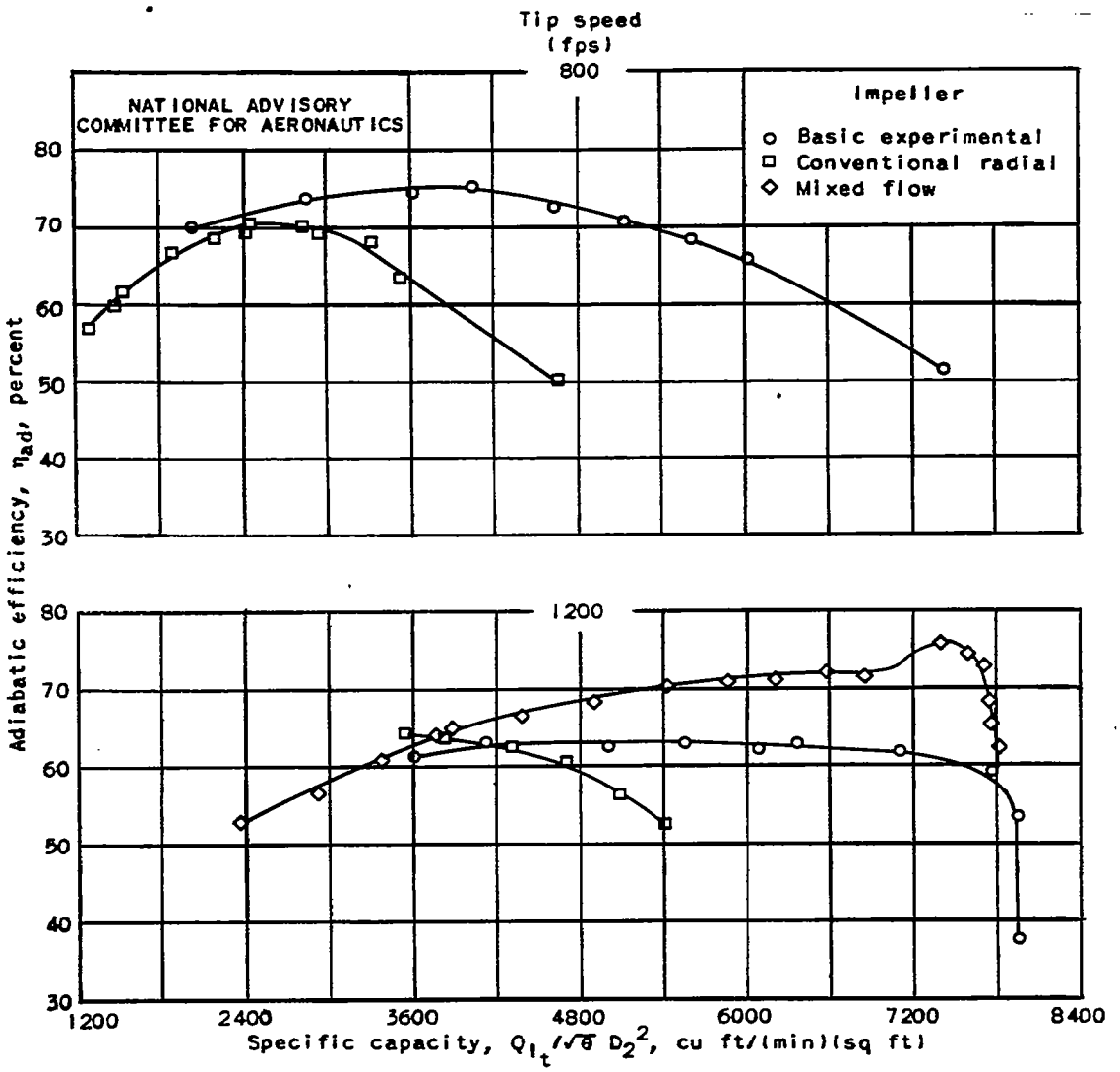

Figure 3. - Specific capacity of basic experimental radial-discharge, conventional radial, and mixed-flow impellers.

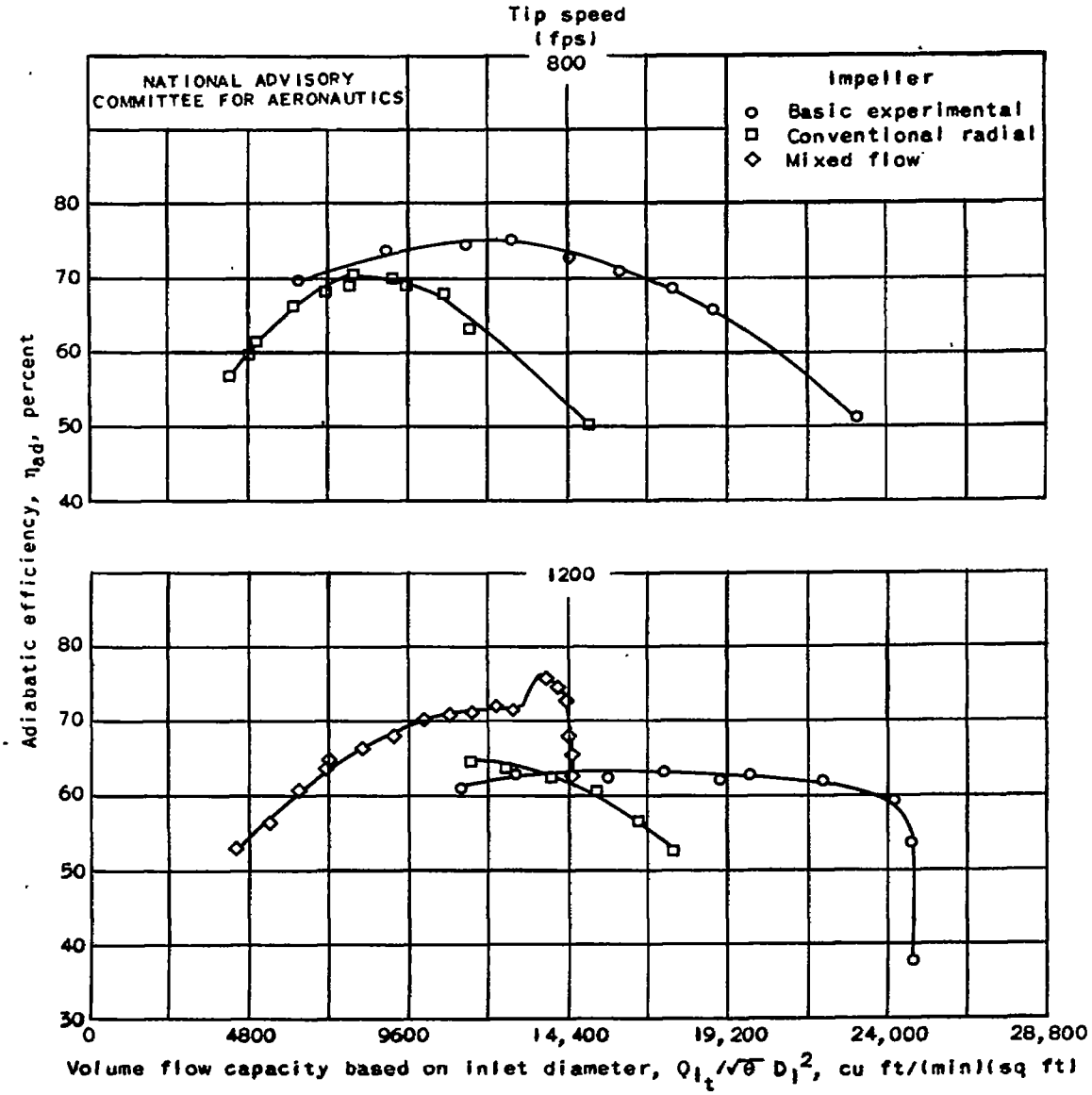

Figure 4. - Volume flow capacity based on inlet diameter of basic experimental radial-discharge, conventional radial, and mixed-flow impellers.

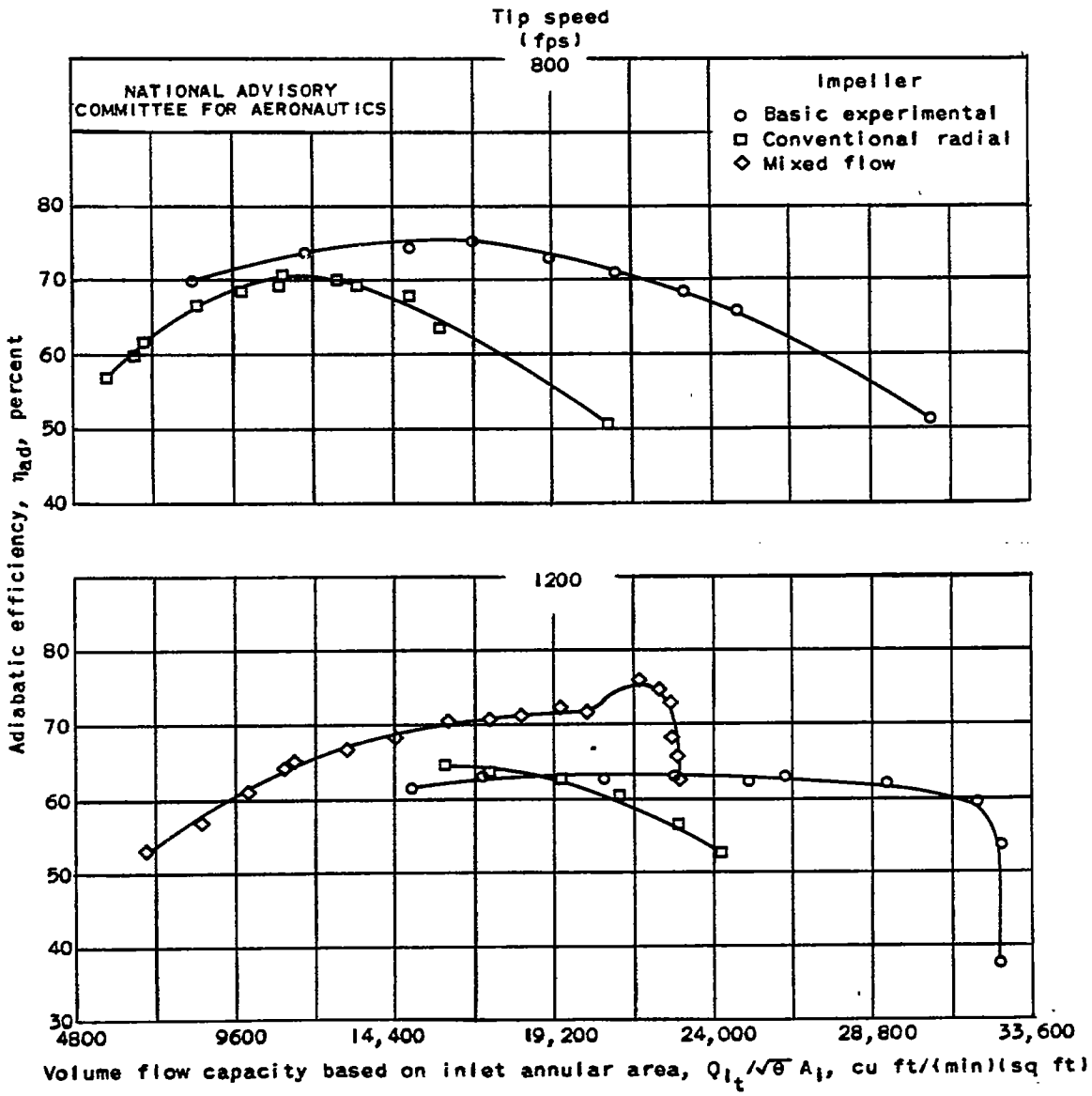

Figure 5. - Volume flow capacity based on inlet-annular area of basic experimental radial-discharge, conventional radial, and mixed-flow impellers.

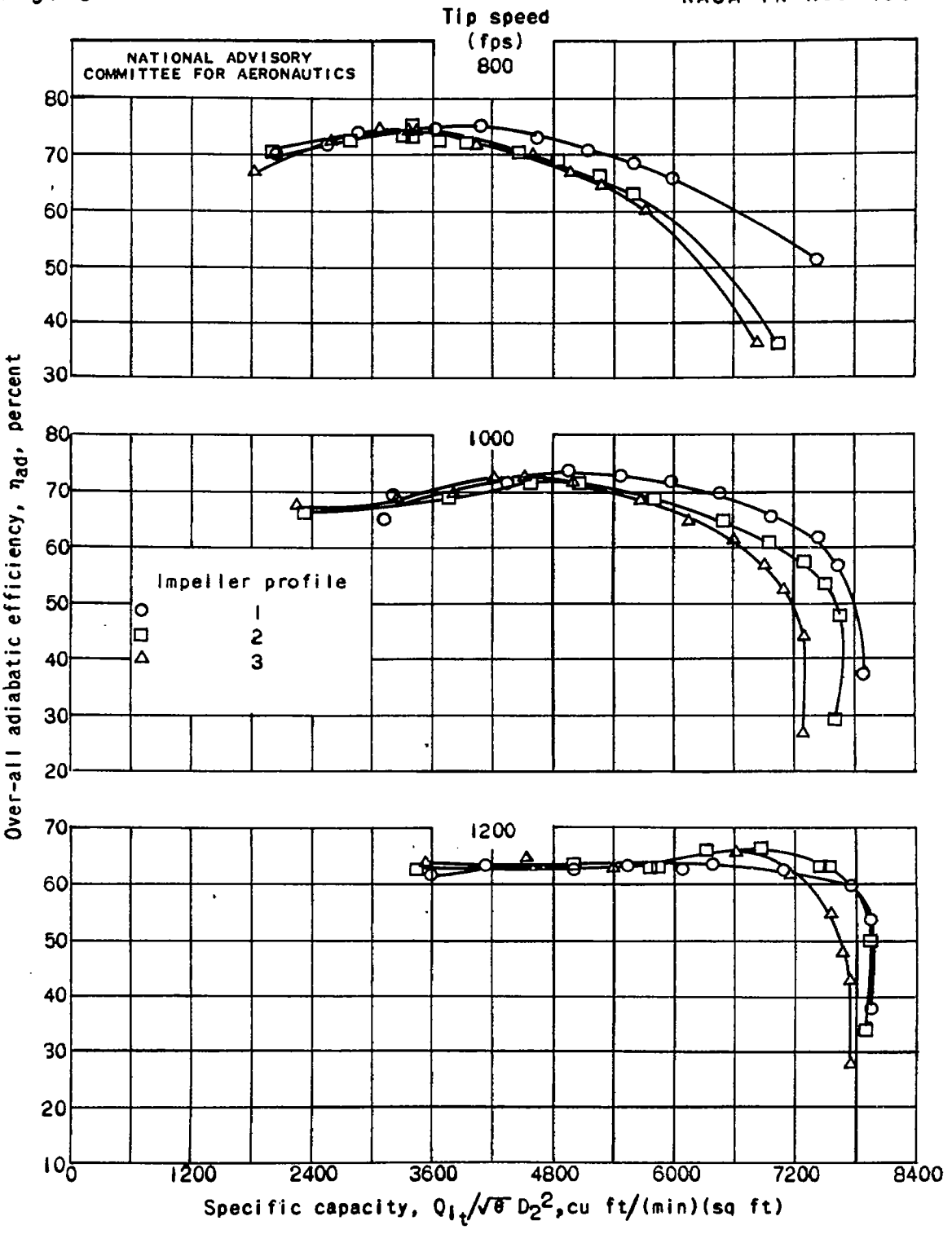

Figure 6. - Over-all adiabatic efficiencies of experimental radial-discharge impeller for three impeller passage-area variations.

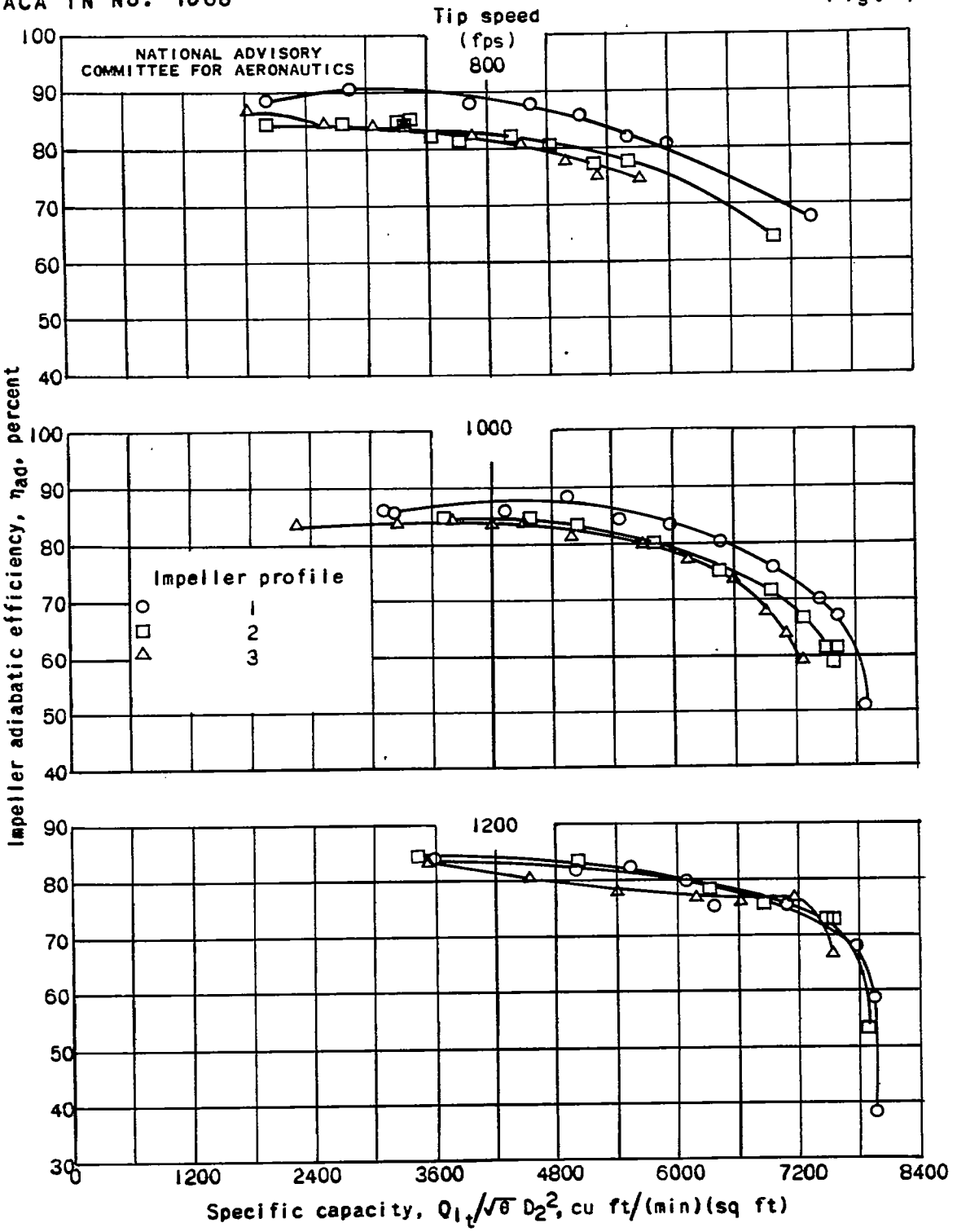

Figure 7. - Impeller adiabatic efficiencies of experimental radial-discharge impeller for three impeller passage-area variations.

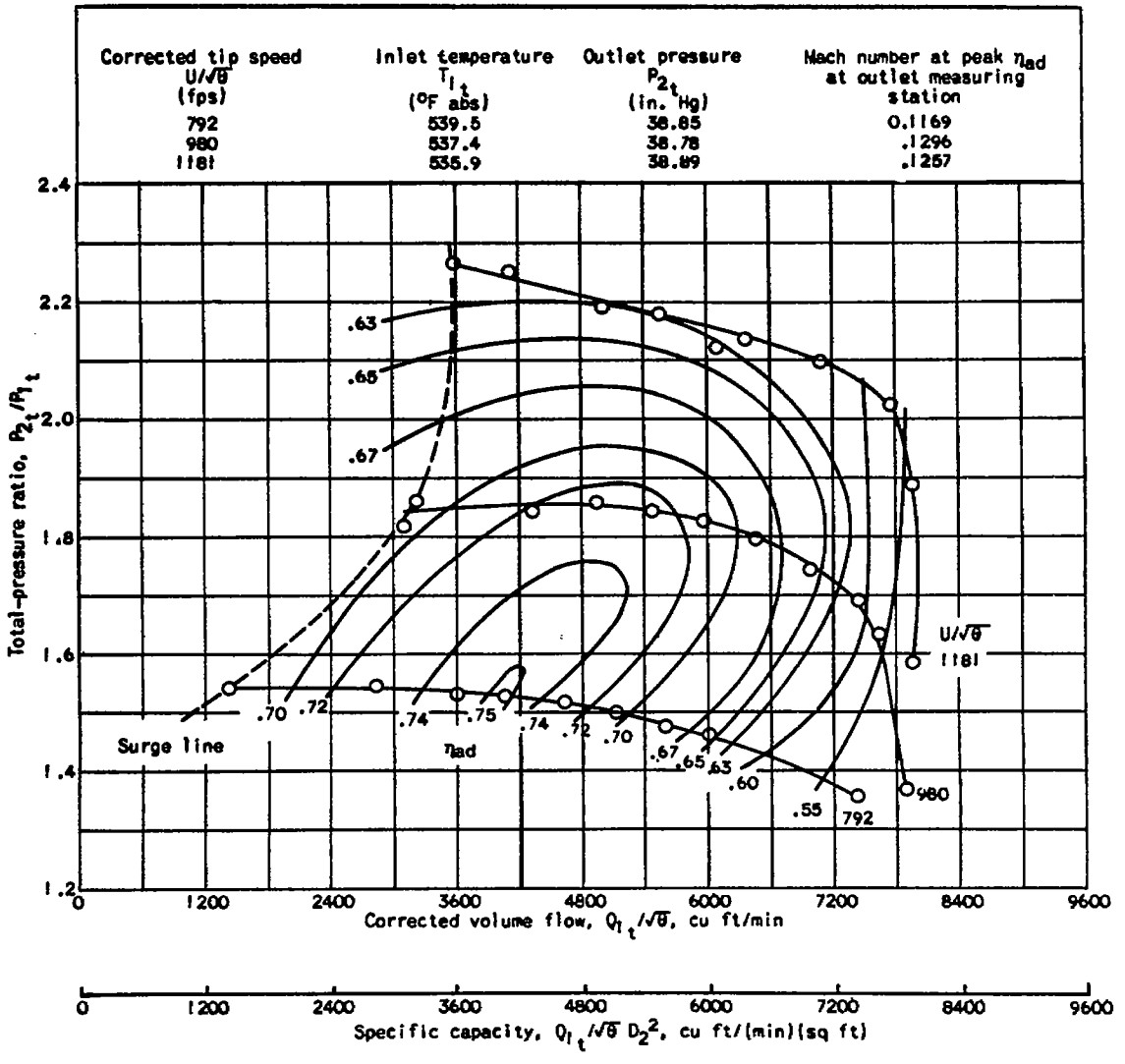

Figure 8. - Over-all performance characteristics of basic experimental radial-discharge impeller.

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