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DETERMINATION OF CENTRIFUGAL-COMPRESSOR PERFORMANCE ON BASIS OF STATIC-PRESSURE MEASUREMENTS IN VANELESS DIFFUSER

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SUMMARY

The use of measured static pressures in a vaneless-diffuser passage for determining centrifugal-compressor performance was investigated. The general effects of diffuser-wall-surface friction were studied to locate the regions in the vaneless-diffuser passage where the most valid evaluation can be made by the static-pressure method. The compressor ratings based on measured static pressures and on measured total pressures are compared.

For the most accurate determination of compressor performance from measured static pressures, a wall-surface friction correction is required. The friction correction, however, is small at the diffuser entrance and in the region of low kinetic energy at the diffuser exit. At the diffuser entrance, the efficiencies determined from measured total pressures were considerably higher than those based on measured static (calculated total) pressures for the range of volume flow and tip speed except at low volume flows at 1200 feet per second. At a diffuser radius of about $1\frac{1}{2}$ impeller radii, the compressor efficiencies determined from measured static pressures showed good agreement with those based on measured total pressures at a tip speed of 800 feet per second; but at 1200 feet per second the efficiencies using measured static pressures were higher for the entire flow range. At the exit of the 34-inch diffusers, the two methods gave nearly the same results for the entire speed and flow range.

INTRODUCTION

The conventional methods of evaluating centrifugal-compressor performance are based on measurements taken upstream of the impeller entrance and downstream of the collector outlet. Although this technique determines the combined performance of the impeller, the diffuser, the collector, and the associated ducting, it provides little information on the individual performance of these components. Because the characteristics of a compressor are essentially determined by the individual performance of the impeller and the diffuser, it is of great value to evaluate separately the performance of these components.

If a reliable determination of the available energy of the air can be made at the impeller exit, the effect of other components is eliminated and an evaluation of the impeller alone is approached. However, not all the existing total pressure (which is a measure of the apparent available energy) can be converted to static pressure because of the nonuniform velocity and energy distribution, the equalization of which is accomplished by mixing with a consequent loss of energy. Therefore, the use of pitot tubes for total-pressure surveys at the impeller exit is undesirable, inasmuch as in this method the impeller is assumed to have the unavailable kinetic energy that is lost in the equalization of the total-pressure gradient. Furthermore, because the pitot-tube indication in a turbulent stream is fundamentally high, this technique always overrates impeller performance.

A more conservative value of total pressure and a closer approximation to the useful energy of the stream can be obtained through the use of measured static pressures. Dynamic pressures can be calculated on the basis of conservation of angular momentum and continuity of flow from the measured static pressures and the total temperature measured in the outlet pipe; the corresponding total pressures can then be determined. The use of measured static pressures therefore has advantages over the total-pressure-survey method in that it does not credit the impeller for flow energy associated with nonuniformity of velocity, the static-pressure wall taps are easily installed, and running time is short in comparison with that required for a pitot-tube survey.

One of the principal disadvantages of the method of determining total pressure from static pressure is that a uniform, full-channel flow is assumed, whereas the actual effective flow area is unknown and is affected by separation, vortices, and recirculation. When this method is applied at a station in a vaneless diffuser, the exact path of the air through the diffuser is also unknown but can be approximated by the use of the proper friction coefficient. Another error may result from the assumption that the equalization of the energy gradient takes place with a 100-percent loss. Inasmuch as static-pressure wall-tap measurements are fairly reliable, the greatest error in calculated total pressure results from the effect of these unknown factors on the calculated dynamic pressure.

The use of measured static pressures in a vaneless diffuser in determining centrifugal-compressor performance was investigated at the NACA Lewis laboratory. The performance ratings presented, which have been designated compressor performance, represent an evaluation 1090 of impeller and diffuser determined from measurements between the impeller entrance and various points along the vaneless-diffuser flow path. The effect of diffuser-wall-surface friction is investigated to determine the regions in the diffuser passage where the choice of friction coefficient is a critical factor in the determination of total pressure from measured static pressure. Compressor ratings determined from static-pressure calculations are compared with the corresponding ratings based on measured total pressure. Experimental data are used from three types of centrifugal impeller, each of which was investigated under similar conditions with similar vaneless diffusers. Data for the three impellers were taken from performance investigations reported in references 1 to 3.

APPARATUS

Installation. - Data from three centrifugal impellers, each having a different geometric shape, were used. The design details of these impellers (figs. 1 to 3) are as follows:

Designation	Impeller	Inducer section	Entrance tip diameter (in.)	Entrance blade-root diameter (in.)	Exit diameter (in.)	Axial depth (in.)	Height of passage at exit (in.)	Number of blades	Source of data (reference)
A	Experimental, large axial-plane curvature and large axial depth	Conventional bucket, die bending	6.80	1.32	12.00	4.88	1.00	14	1
B	Experimental, with constant-blade-loading inducer	Parabolic machined	8.00	2.85	12.00	3.84	0.66	18	2
C	Conventional production	Conventional bucket, die bending	7.27	2.59	12.00	2.80	0.76	22	3

The three impellers were investigated in a variable-component compressor setup in accordance with the recommendations of reference 4, except that only one radial outlet pipe was used. Previous results had shown that no appreciable difference existed in performance when one outlet pipe was used. Each of the three impellers was investigated with a 34-inch vaneless diffuser of similar design. Over-all performance was investigated in accordance with reference 4.

Instrumentation. - Inasmuch as the three impellers were investigated as part of other research programs rather than for the specific purpose of evaluating performance on the basis of static pressures, the instrumentation in the diffusers was different for the three units (fig. 4). In all the diffusers, however, static-pressure taps of 0.020-inch bore were drilled normal to the wall surface in the passage from the diffuser entrance to the diffuser exit. These taps were installed on both the front and rear wall opposite one another at each measuring station. For impellers A and B, six static-pressure wall taps were equally spaced at a 9.00-inch radius on the front diffuser plate.

For impellers A and B, total-pressure measuring stations were located in the diffuser passage at radii of 9.00 and 16.75 inches. At the 9.00-inch radius, a cylindrical tube 0.10 inch in diameter with a 0.020-inch hole drilled in its side was used for total-pressure readings at the midpoint of the diffuser passage and at 0.063 inch from each wall. At the 16.75-inch radius, three total-pressure rakes were equally spaced around the diffuser periphery. These rakes were shielded and insensitive to yaw over a wide range of angles. For impeller A, each rake contained three pressure tubes having a 0.016-inch opening; the tubes were located at the midpoint of the diffuser passage and approximately 0.10 inch from each wall. For impeller B, each rake contained four tubes, similar to those used for impeller A, with the two outer openings located approximately 0.10 inch from each wall and the two inner openings equally spaced across the rest of the diffuser passage.

For impeller C, total-pressure tubes of 0.093-inch diameter with a 0.031-inch-diameter hole drilled in the side of the tube were located at radii of 6.50, 8.00, and 16.50 inches from the impeller axis. Total-pressure readings were obtained by rotating these tubes for a maximum reading at the midpoint of each of four equal lengths across the diffuser passage.

METHOD OF ANALYSIS

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The adiabatic efficiency was determined at radial positions in the vaneless diffuser from direct total-pressure measurements and total pressures based on measured static pressures. Calculations were made in accordance with the standard procedures given in reference 5 using the total temperatures measured in the outlet pipe and the total pressures obtained from either direct measurements or calculations. The use of outlet-pipe temperature assumes that there is no change in total temperature from the impeller exit to the outlet-pipe measuring station in the thermally insulated setups. Actually some loss in total temperature does occur through the collector casing between the diffuser exit and the outlet-pipe measuring station (reference 6). The adiabatic efficiencies based on direct total-pressure measurements and on total pressures derived from measured static pressures can be compared because both efficiencies involve the same total temperature. The volume flow was corrected to standard conditions by the method given in reference 7. The analysis of performance was made for actual impeller tip speeds of 800 and 1200 feet per second.

Total pressure from measured static pressure. - Determining the efficiency at any point in the diffuser from the measured static pressure requires the calculation of the total pressure of the air stream. Under the assumption of uniform and full flow in a diffuser passage, the radial component of velocity can be expressed as a function of the local static temperature by means of the continuity equation. In the ideal case, the tangential component of velocity is diffused according to the law of conservation of angular momentum. The local static temperature and the total pressure can then be determined by the energy equation. Under actual conditions, however, the tangential component of velocity is reduced as a result of diffuser-wall-surface friction, mixing, and recirculation. The effect of friction on the tangential velocity can be accounted for by modifying the basic equation for angular momentum, which can then be used to calculate the total pressure. The methods of calculating these quantities are presented in appendix A.

In order to indicate the effect of diffuser-wall-surface friction on total pressure and adiabatic efficiency, calculations were made with a friction factor of 0.005; the friction factor is defined in reference 8. An analysis of flow in a vaneless diffuser and the calculation of a theoretical friction factor for smooth surfaces (reference 9) indicated that the value of 0.005 is of reasonable average magnitude for the diffusers used in this investigation, although higher apparent friction coefficients may exist near the diffuser entrance.

The magnitude of change in total energy that results from applying the friction correction at any point in the diffuser passage depends upon the coefficient of friction used, the length of the flow path between the impeller exit and the point in the diffuser (a function of volume flow and impeller tip speed), and the ratio of the dynamic energy to the total energy. The percentage changes in total pressure and dynamic pressure resulting from the friction correction are designated the total-pressure correction factor and the dynamic-pressure correction factor, respectively. These two correction factors are defined in appendix B.

Measured total pressure. - An accurate determination of the available energy from total-pressure readings requires the mass-flow integration of the total-pressure gradient across the diffuser passage. Because the error introduced by using the comparatively simple arithmetic average of measured total pressures was small, an arithmetic average across the diffuser passage was used in the calculation of all performance curves presented.

RESULTS AND DISCUSSION

Effects of Diffuser-Wall-Surface Friction on Calculated Total Pressure and Adiabatic Efficiency

The total and dynamic pressures calculated by assuming a friction coefficient of 0.005 were appreciably different from the respective pressures obtained with friction ignored. The significance of the friction correction at the various stations through the diffuser is shown by a study of the manner in which the total- and dynamic-pressure corrrection factors vary.

Total-pressure correction factor is dependent upon the ratio of the dynamic pressure to the total pressure and upon the dynamic-pressure correction factor. The ratio of the dynamic pressure to the total pressure decreases with increased diffuser radius, as shown in figure 5 for impeller A at a typical operating point for two impeller tip speeds. The dynamic-pressure correction factor, which is a function of flow-path length, increases, however, with an increase in diffuser radius. The opposing direction of variation of these two controlling terms results in a total-pressure correction factor that varies with diffuser radius in the manner shown in figure 6. The largest total-pressure correction factor occurred over an approximate range of diffuser radii from 8.00 to 14.00 inches with a negligible correction at the diffuser entrance and a reduced correction at the diffuser exit. The total-pressure correction factor increased with decreasing volume flow as a result of the increase in flow-path length. At a given volume flow, the total-pressure correction factor increased with an increase in impeller tip speed because of the increase in flow-path length and the increase in the ratio of dynamic to total pressure. The application of the friction correction is therefore seen to be most critical at high speeds, at low flows, and in the range of diffuser radii from 8.00 to 14.00 inches. In a diffuser of a size sufficient to decrease the kinetic energy appreciably, the friction correction is small in the region of low kinetic energy.

Compressor adiabatic efficiencies for impeller A determined from measured static pressures in the diffuser for frictionless flow and for flow with diffuser-wall-surface friction considered are compared in figure 7. The comparison was made at impeller tip speeds of 800 and 1200 feet per second and at several radii in the diffuser passage. The maximum variation between the efficiency curves was 0.05 for a tip speed of 1200 feet per second at a diffuser radius of 9.00 inches, which corresponds to $1\frac{1}{2}$ impeller radii.

Compressor Adiabatic Efficiency Determined from Calculated and Measured Total Pressure

<u>Performance at diffuser entrance.</u> - Compressor adiabatic efficiencies near the impeller exit determined from measured total pressures and from calculated total pressures (with friction considered) for impeller C at tip speeds of 800 and 1200 feet per second are shown in figure 8. Impeller C was the only impeller in which total-pressure measurements were taken at the diffuser entrance. The measured-total-pressure efficiencies were considerably higher than the calculated-total-pressure efficiencies over the volume-flow and tip-speed ranges with an exception at low volume flow at a tip speed of 1200 feet per second, where the calculated total-pressure efficiencies were 0.02 higher. The maximum difference between the two curves was 0.10 and occurred at the high volume flows. In general, this tendency for the measured total-pressure efficiencies to be greater than the calculated total-pressure efficiencies is the result of two effects:

(1) Total-pressure tubes inherently give high readings in a pulsating stream.

(2) The method of calculating total pressures neglects the unusable kinetic energy of the air stream, giving a conservative but more valid evaluation of performance.

At the diffuser entrance, the air stream has the greatest turbulence, pulsation, and nonuniformity of velocity gradient and the ratio of kinetic energy to total energy is a maximum; a maximum difference between the two methods of determining efficiency therefore exists at this station. The reversal of the two curves in the low-volume-flow range (fig. 8, 1200 ft/sec) is the result of some variation from the flow conditions assumed for calculation - probably recirculation, because maximum recirculation exists at the low volume flows (reference 10). The presence of recirculation results in an overvaluation of the tangential velocity of the air, which in turn makes the calculated total pressures and the corresponding efficiency too high.

Performance at diffuser radius of about $1\frac{1}{2}$ impeller radii. - Compressor adiabatic efficiencies at about $1\frac{1}{2}$ impeller radii for impellers A, B, and C determined from calculated total pressures (with friction considered) and from measured total pressures at tip speeds of 800 and 1200 feet per second are shown in figure 9. At 800 feet per second for impeller A, the two performance curves were almost the same over the volume-flow range. For impellers B and C, good agreement existed between the two performance curves with the calculated-total-pressure efficiencies slightly lower than the measured-total-pressure efficiencies at the high volume flows.

At a tip speed of 1200 feet per second, the calculated-total-pressure efficiencies for the three impellers A, B, and C were higher than the measured-total-pressure efficiencies over the volume-flow range from surge to flow cut-off; a maximum difference of 0.04 to 0.05 existed at the low flows and the difference diminished with increasing volume flow. This discrepancy may be a result of a recirculation at low volume flows. In addition (see fig. 6), the total-pressure correction factor is maximum in this part of the diffuser and increases with increasing tip speed and decreasing volume flow. The choice of friction factor and the application of the friction correction is most critical in the operating range where the maximum difference between the two curves occurs.

Performance at diffuser exit. - Compressor adiabatic efficiencies at the diffuser exit determined from measured and calculated total pressures (with friction considered) for impellers A, B, and C at 800 and 1200 feet per second are compared in figure 10.

Over the speed and volume-flow ranges, the two efficiency curves are almost the same. A maximum difference of 0.02 occurred at a tip speed of 1200 feet per second in the high-volume-flow range for impellers A and C.

SUMMARY OF RESULTS

An investigation of the use of diffuser static pressures for rating compressors and a presentation of compressor performance based on measured diffuser static and total pressures produced the following results:

1. For the most accurate determination of compressor performance from measured diffuser static pressures, a wall-surface friction correction would be required. For a 34-inch-diameter vaneless diffuser and a 12-inch-diameter impeller, a maximum friction correction occurred in the approximate range of diffuser radii from 8.00 to 14.00 inches. The friction correction was very small at the diffuser entrance and in the region of low kinetic energy at the diffuser exit.

2. At the diffuser entrance, the efficiencies determined from measured total pressures were considerably higher than those based on calculated total pressures over the volume-flow and tip-speed ranges, except at the low volume flow at a tip speed of 1200 feet per second, where the calculated total pressures were slightly higher.

3. At the diffuser radius of about $1\frac{1}{2}$ impeller radii, the efficiency curves for the three compressors, based on calculated total pressures (with friction considered) and measured total pressures, showed good agreement at a tip speed of 800 feet per second. At 1200 feet per second, the efficiencies determined from calculated total pressures were higher than the corresponding measured total-pressure efficiencies over the flow range from surge to flow cut-off.

4. The performance based on calculated total pressures and on measured total pressures gave nearly the same results at the exit of the 34-inch diffusers over the entire speed and flow range.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, February 15, 1949.

APPENDIX A

CALCULATION OF TOTAL PRESSURE FROM STATIC PRESSURE

Symbols

The following symbols are used in the calculations:

a	speed of sound, ft/sec
b	width of diffuser passage, ft
c_p	specific heat of normal air at constant pressure, 189.05 ft-lb/(lb)($^{\circ}$F)
f	friction coefficient
g	acceleration due to gravity, ft/sec^2
H	increase in total enthalpy per unit mass, ft-lb/lb mass
M	Mach number in diffuser passage, V/a
p	pressure, lb/sq ft absolute
R	gas constant for normal air, 53.50 ft-lb/(lb)($^{\circ}$F)
r	radius from impeller center, ft
T	temperature, $^{\circ}$R
U	impeller tip speed, ft/sec
V	velocity of fluid, ft/sec
W	flow rate, lb/sec
α	angle formed by absolute velocity with tangential component, deg
γ	ratio of specific heats
ρ	mass density of air, slugs/cu ft

Subscripts:

1 conditions at impeller exit

r radial component

s static, or true-stream, value

t total, or stagnation, value

θ tangential component

Calculations

The total pressure at any point in the vaneless-diffuser passage may be determined from the computed dynamic pressure and the measured static pressure. The calculations are made on the assumptions that total temperature of the air remains constant from the impeller exit through the insulated system to the measuring station in the outlet pipe and that the velocity is constant across the diffuser passage. The velocity and the density of the air may be found from the measured static pressure, the continuity of flow, the conservation of angular momentum, and the foregoing assumptions.

The air velocity, the static temperature, and the density at a point in the diffuser passage are found from the relations

$$T_s = T_t - \frac{V^2}{2c_p g} \tag{1}$$

$$V^2 = V_r^{\ 2} + V_\theta^{\ 2} \tag{2}$$

where V_θ is found by the equation for the angular momentum

$$\frac{d(rV_\theta)}{rV_\theta} = -f\,\frac{\csc\alpha}{b}\,dr \tag{3}$$

Equation (3) is given by Polikovsky and Nevelson in reference 8. For the purpose of integration, the term $(\csc\alpha)/b$ was assumed to be constant. Results of unpublished NACA investigations show only a small variation of the term $(\csc\alpha)/b$ with diffuser radius for vaneless diffusers similar to the diffusers used in this investigation. Equation (3) becomes by integration

$$V_\theta = V_{\theta,1} \frac{r_1}{r} e^{-f \frac{\csc \alpha}{b} (r-r_1)} \tag{4}$$

The tangential velocity at the impeller exit is found from the total-enthalpy increase in accordance with Euler's law, with friction and windage losses neglected

$$V_{\theta,1} = \frac{gH}{U} \tag{5}$$

From continuity of flow

$$V_r = \frac{W}{2\pi rb\rho g} = \frac{WRT_s}{2\pi rbp_s} \tag{6}$$

Combining equations (1) to (6) results in an equation in which T_s is the only unknown variable

$$T_s = T_t - \frac{\left[\frac{gH}{U} \frac{r_1}{r} e^{-f \frac{\csc \alpha}{b} (r-r_1)}\right]^2 + \left(\frac{WRT_s}{2\pi rbp_s}\right)^2}{2c_p g} \tag{7}$$

If equation (7) and the relation

$$\frac{V_r}{V_\theta} = \tan \alpha \tag{8}$$

are used, the solution for T_s can be made either graphically or by a series of approximations.

For ideal flow, equation (7) becomes

$$T_s = T_t - \frac{\left(\frac{gH}{U} \frac{r_1}{r}\right)^2 + \left(\frac{WRT_s}{2\pi rbp_s}\right)^2}{2c_p g} \tag{9}$$

The value of V and therefore of M may then be determined from equation (1).

The total pressure p_t can be found from Bernoulli's equation for compressible flow

$$p_t = p_s \left(1 + \frac{\gamma-1}{2} M^2\right)^{\frac{\gamma}{\gamma-1}} \tag{10}$$

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APPENDIX B

TOTAL-PRESSURE AND DYNAMIC-PRESSURE CORRECTION FACTORS

The following symbols are used in appendix B:

$p_{t,id}$ calculated total pressure for ideal flow

$p_{t,f}$ calculated total pressure considering friction

q_{id} calculated dynamic pressure for ideal flow

q_f calculated dynamic pressure considering friction

The difference between the total pressure calculated for ideal flow and the total pressure calculated by accounting for diffuser-wall-surface friction, divided by the total pressure calculated for ideal flow, is called the total-pressure correction factor:

$$\text{total-pressure correction factor} = \frac{p_{t,id} - p_{t,f}}{p_{t,id}} \tag{1}$$

This ratio is a measure of the error introduced by ignoring friction in the calculations. The dynamic-pressure correction factor is similarly defined:

$$\text{dynamic-pressure correction factor} = \frac{q_{id} - q_f}{q_{id}} \tag{2}$$

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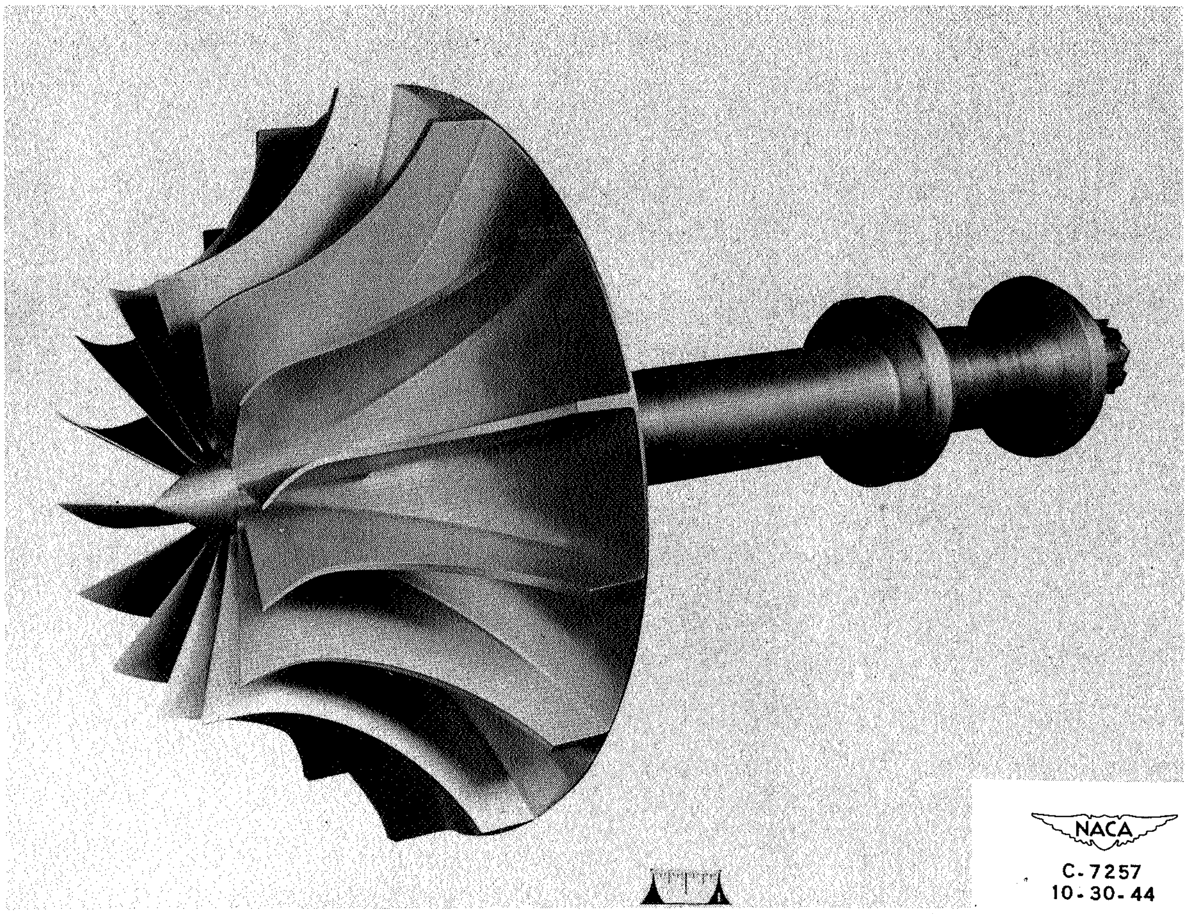

Figure 1. - Experimental impeller with conventional blade bending but large axial-plane curvature and large axial depth (impeller A).

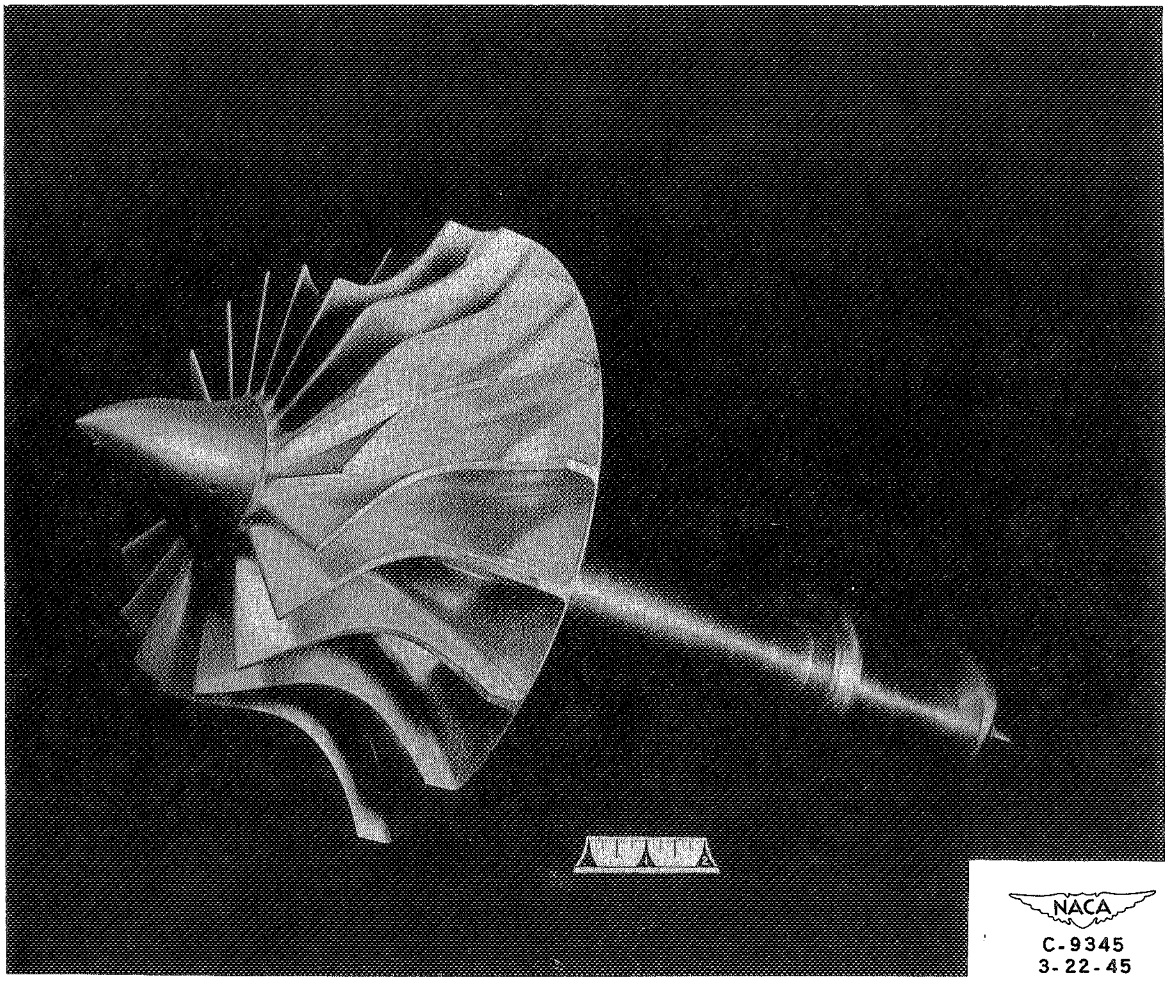

Figure 2. - Experimental impeller in combination with constant-blade-loading inducer (impeller B).

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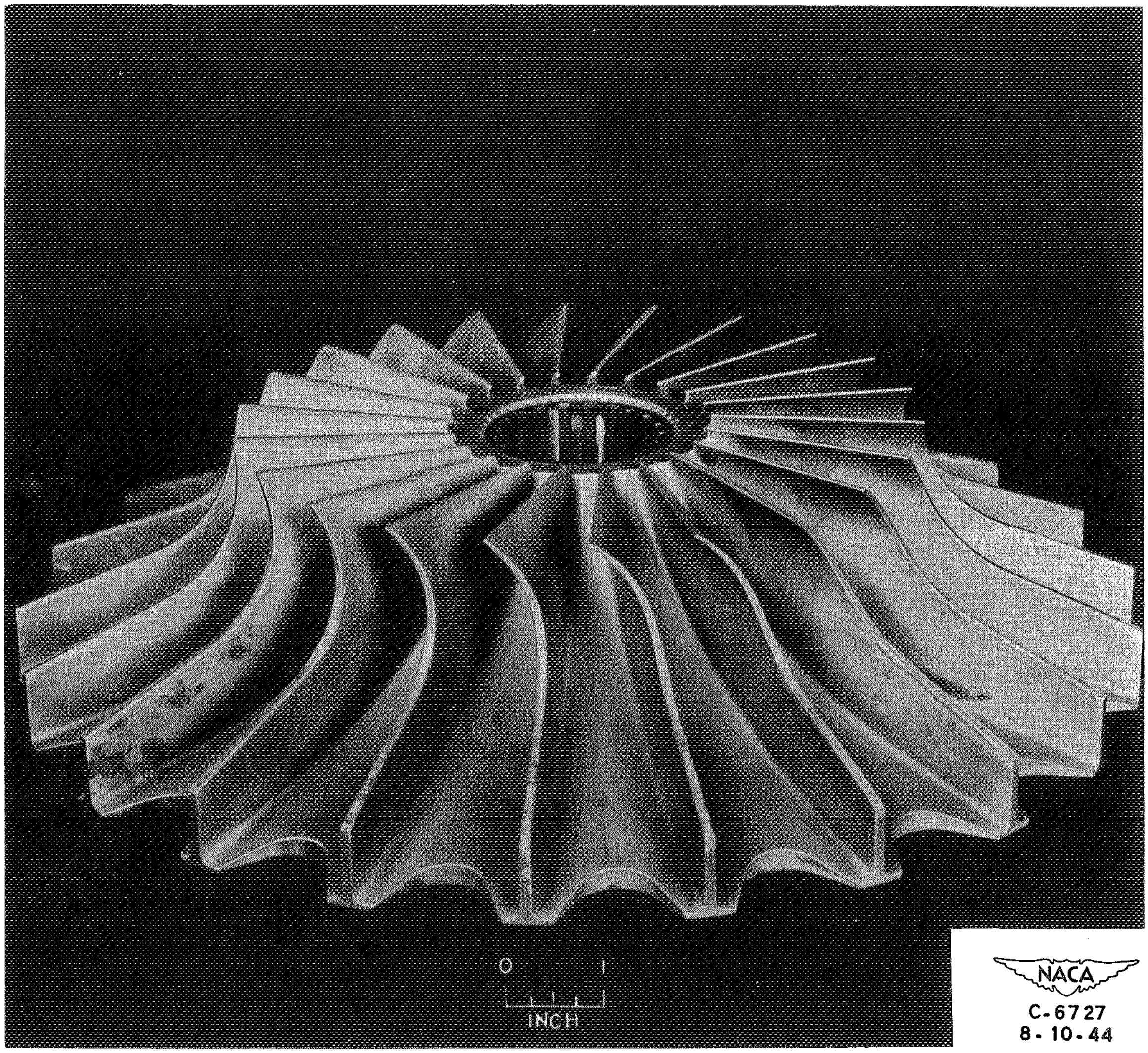
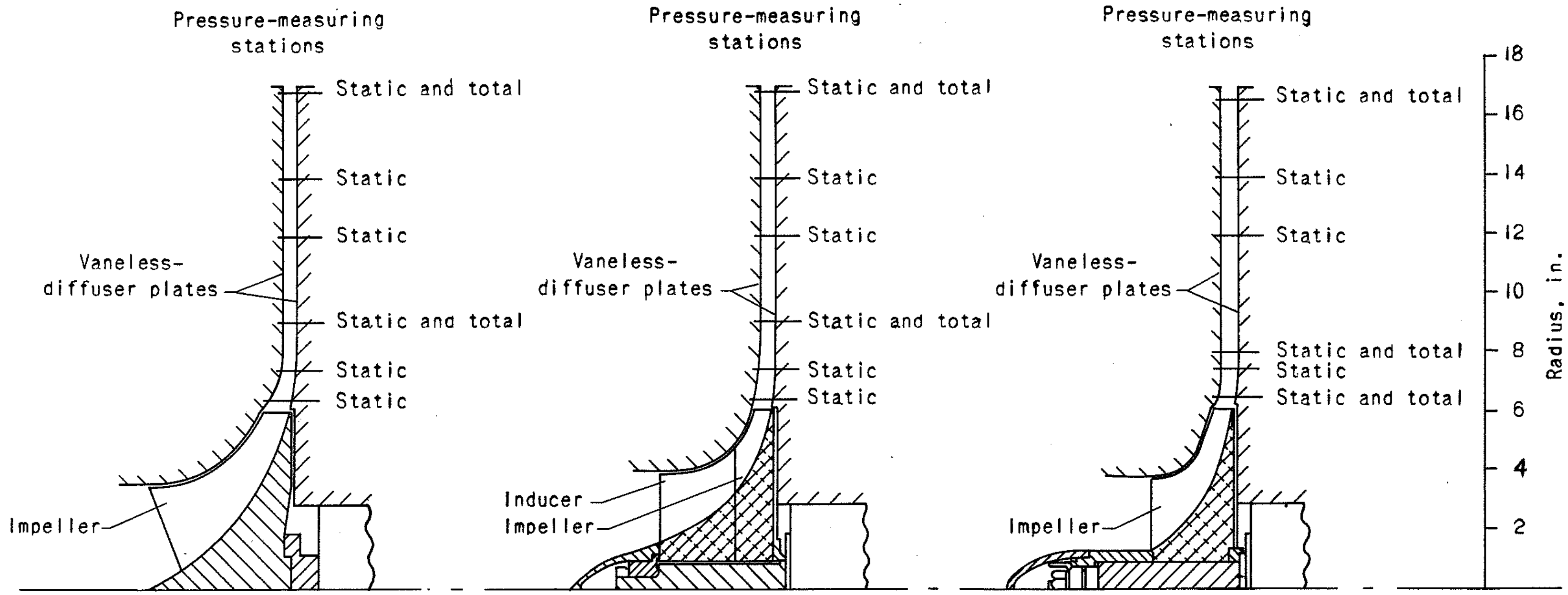


Figure 3. - Conventional production impeller (impeller C).



(a) Impeller A, experimental impeller with conventional blade bending but large axial-plane curvature and large axial depth.

(b) Impeller B, experimental impeller in combination with constant-blade-loading inducer.

(c) Impeller C, conventional production impeller.

Figure 4. - Installations of three impellers, A, B, and C showing locations of pressure-measuring stations in vaneless diffusers.

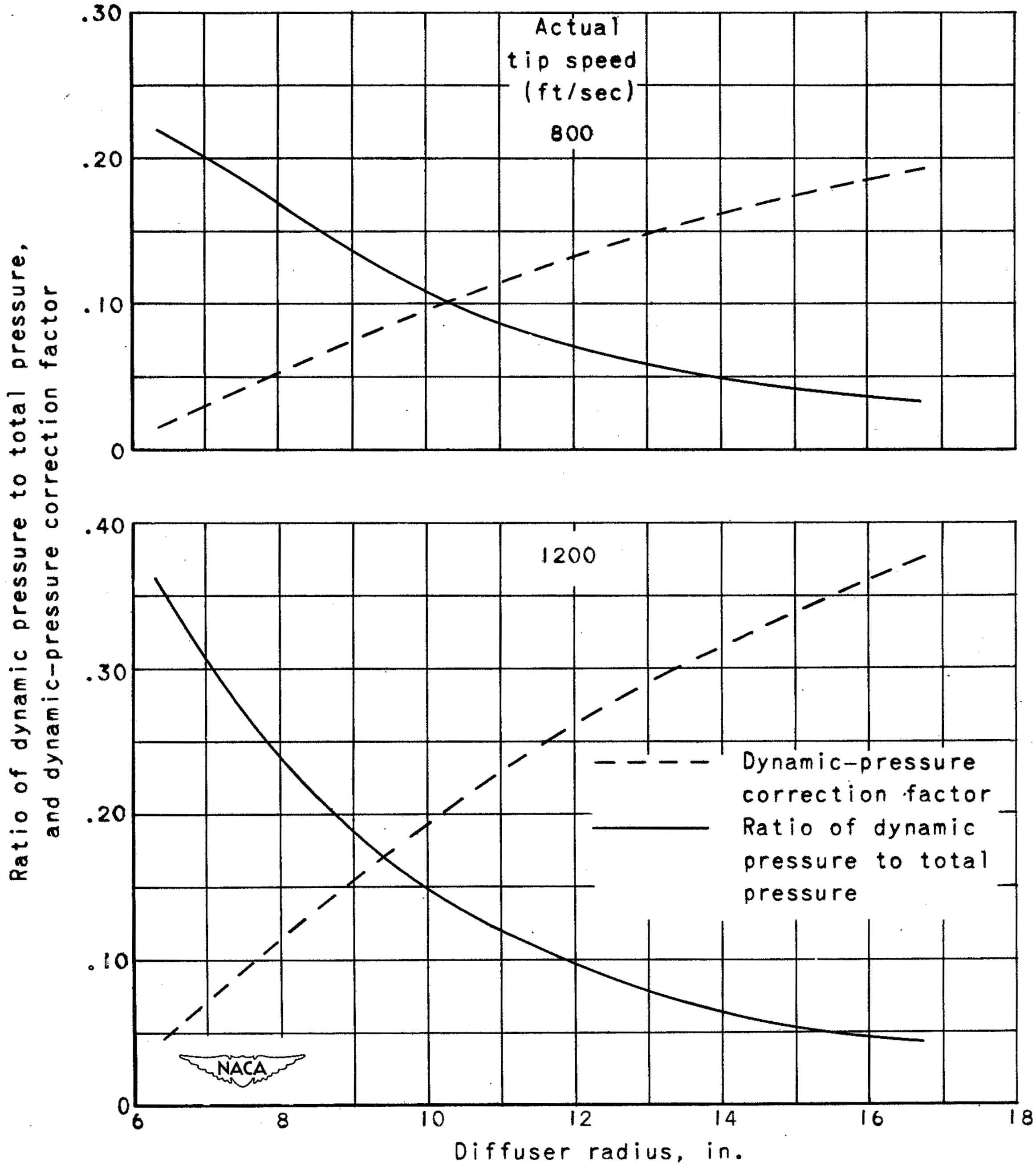


Figure 5. - Effect of diffuser-wall-surface friction on dynamic pressure and ratio of dynamic pressure to total pressure for impeller A at corrected volume flow of 4800 cubic feet per minute.

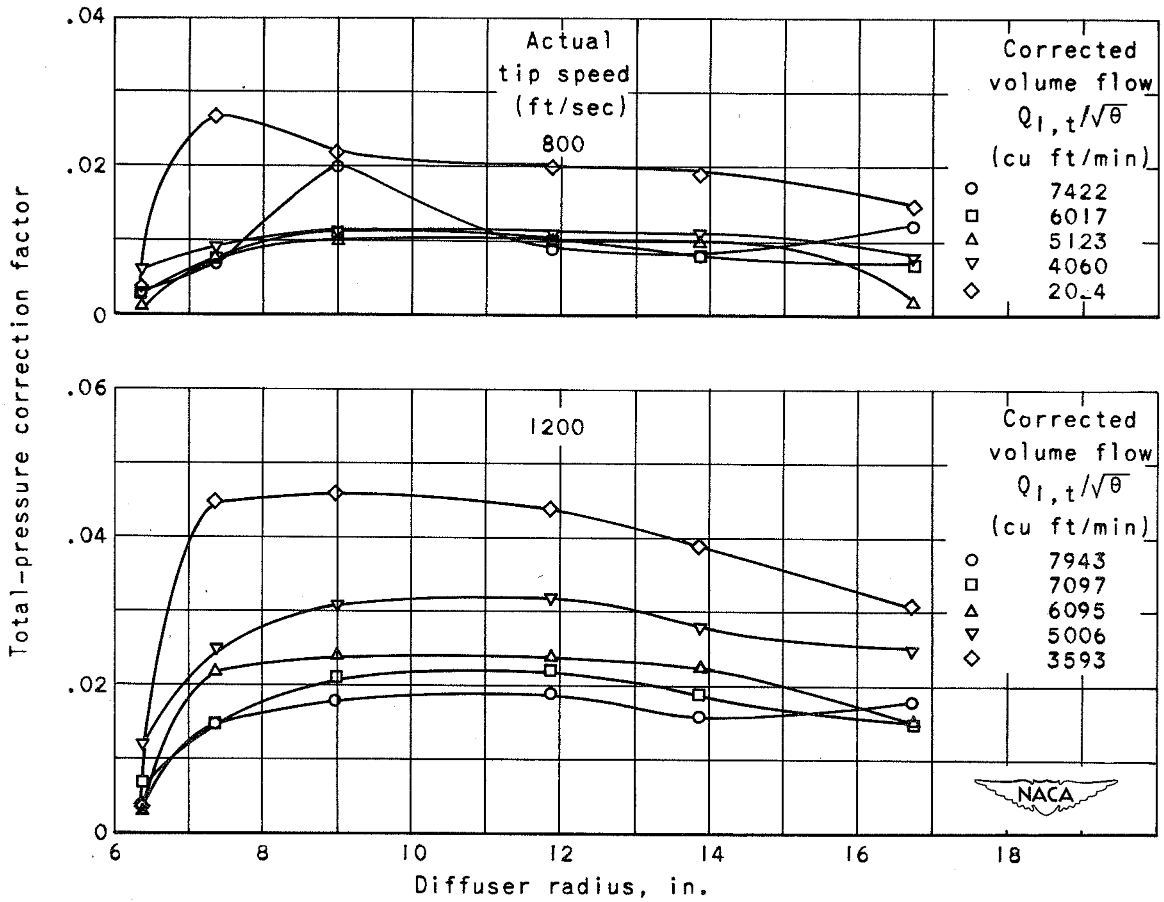


Figure 6. - Effect of diffuser-wall-surface friction on total-pressure correction factor through vaneless-diffuser passage for impeller A for range of corrected volume flows.

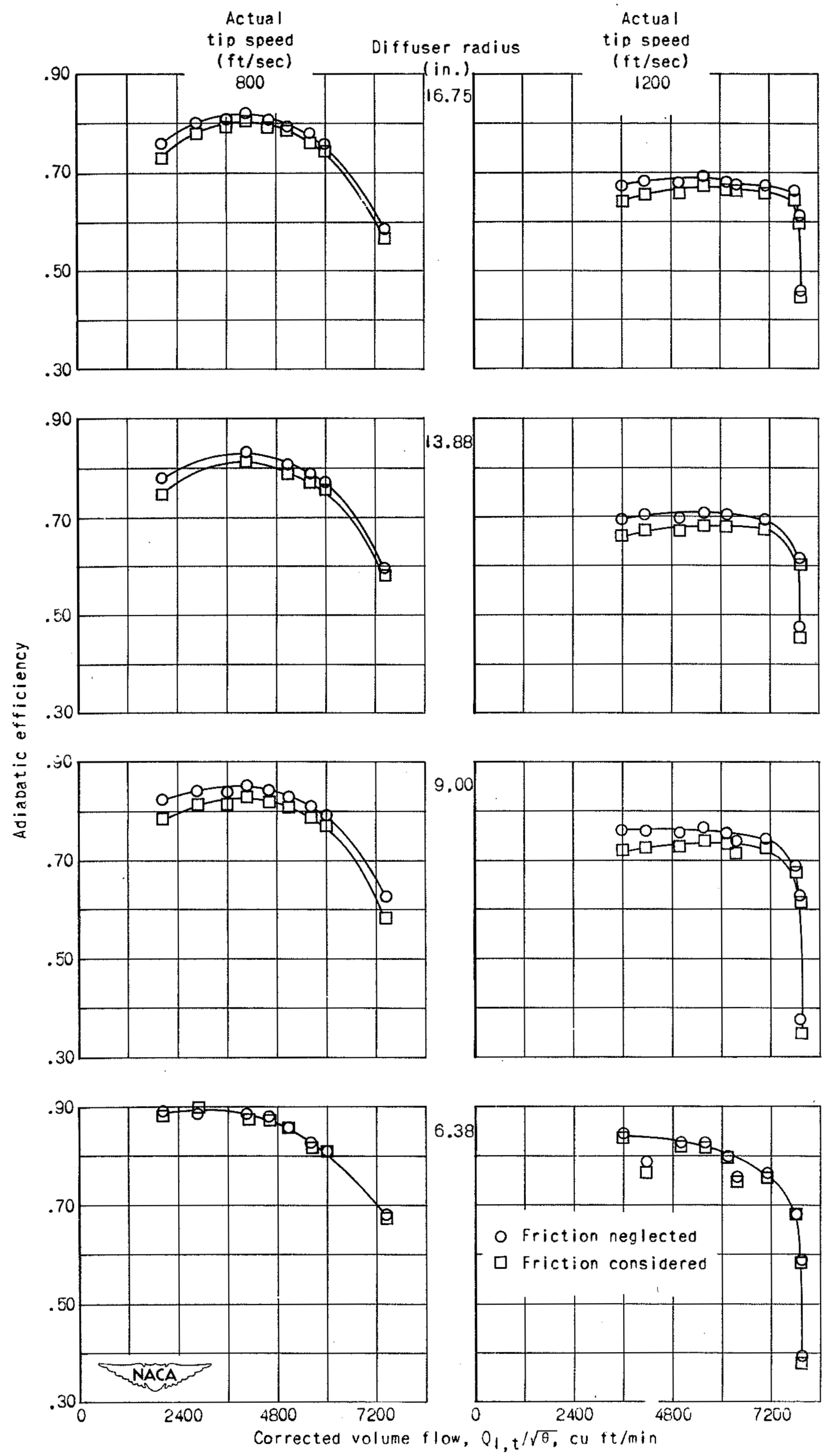


Figure 7. - Compressor adiabatic efficiencies for impeller A determined from measured static pressures for frictionless flow in diffuser and for flow with diffuser-wall-surface friction considered.

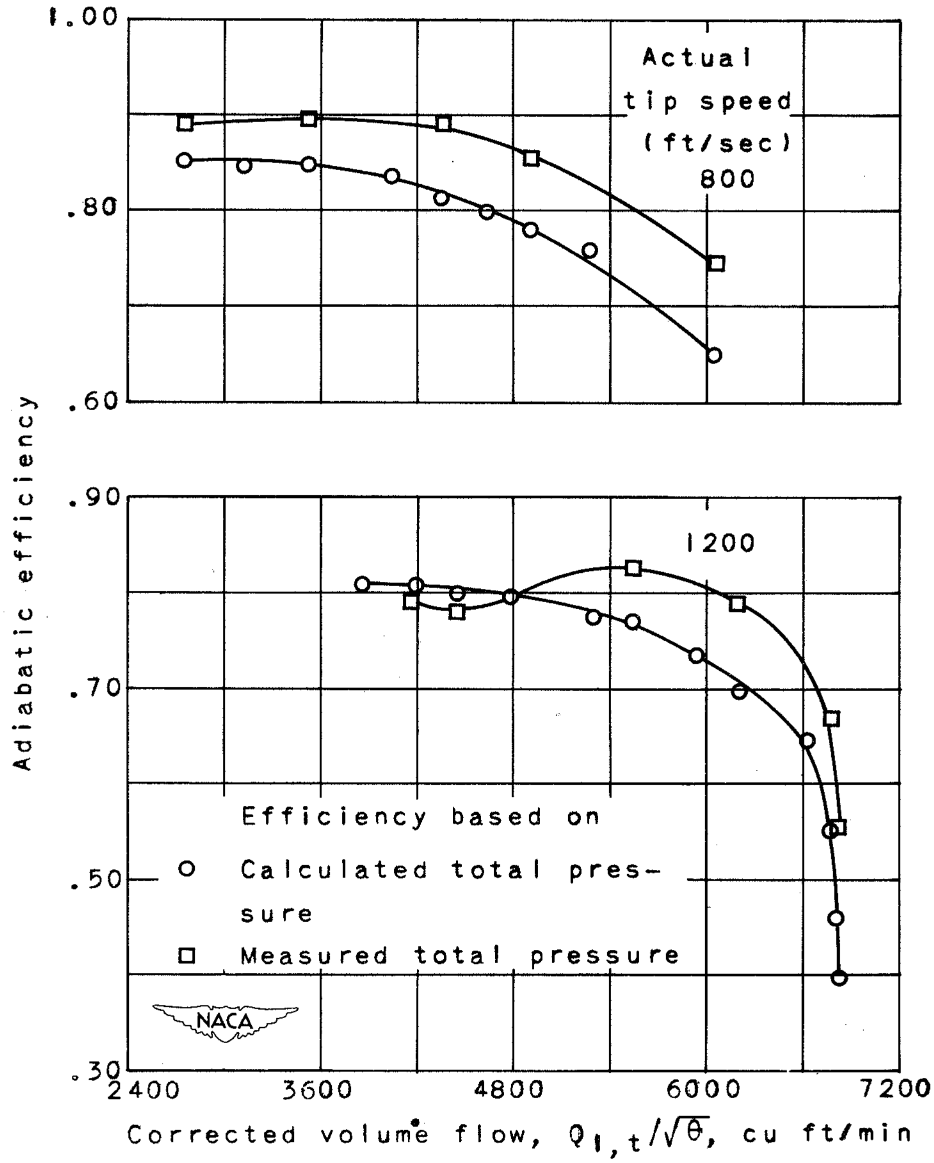


Figure 8. - Compressor adiabatic efficiencies for impeller C based on calculated and measured total pressures at impeller exit.

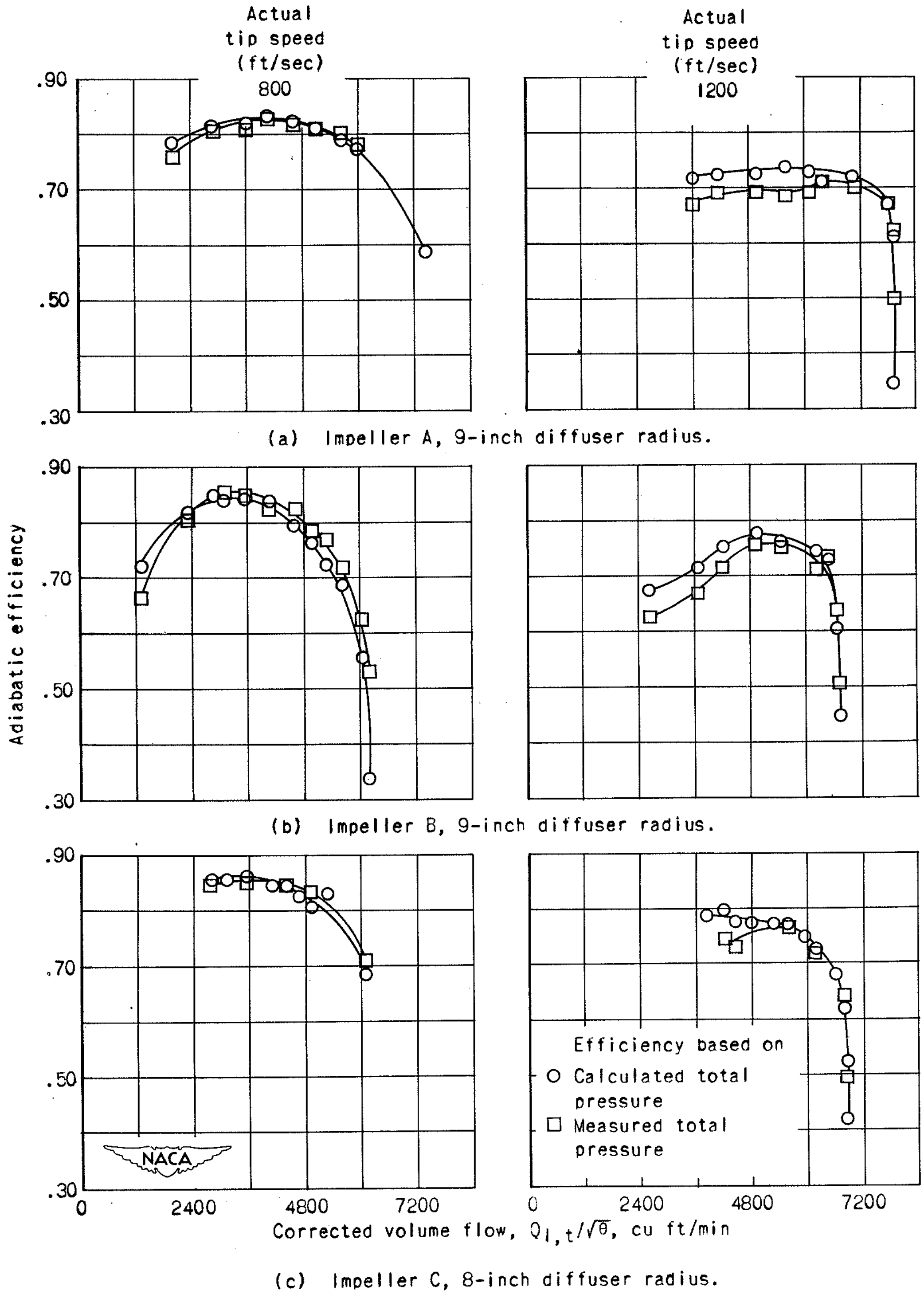
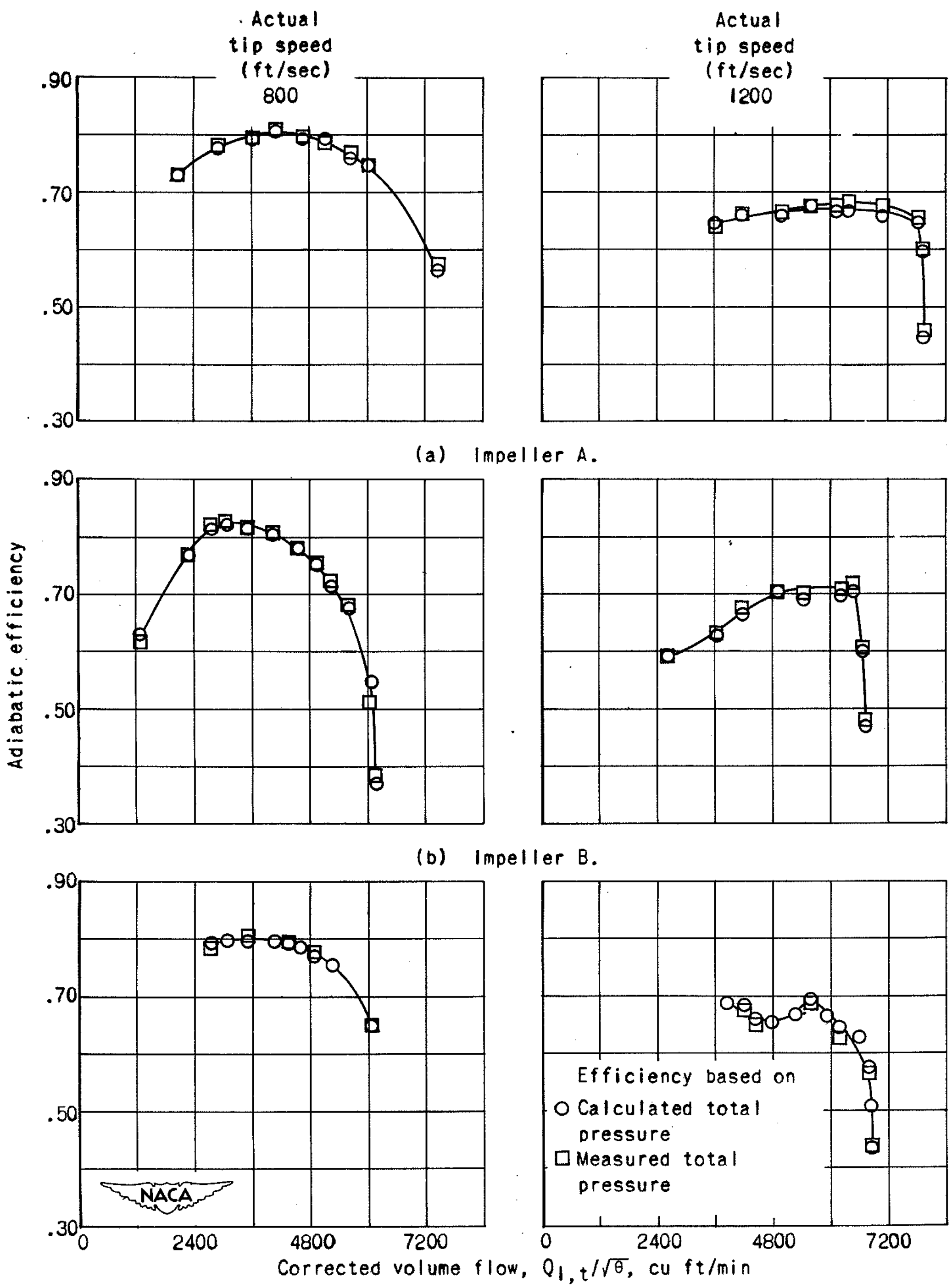


Figure 9. – Compressor adiabatic efficiencies with impellers A, B, and C based on calculated and measured total pressures at diffuser radius of about $1\frac{1}{2}$ impeller radii.



(c) Impeller C.

Figure 10. - Compressor adiabatic efficiencies with impellers A, B, and C based on calculated and measured total pressures at diffuser exit.

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