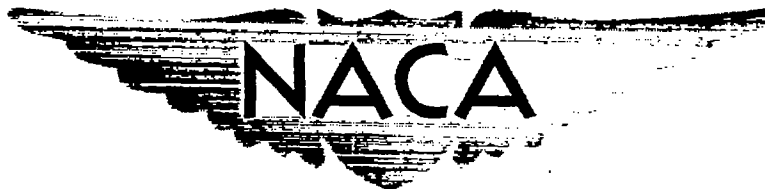


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RESEARCH MEMORANDUM

EVALUATION OF CENTRIFUGAL COMPRESSOR PERFORMANCE
WITH WATER INJECTION

By William L. Beede, Joseph T. Hamrick
and Joseph R. Withee, Jr.

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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RESEARCH MEMORANDUM

EVALUATION OF CENTRIFUGAL COMPRESSOR PERFORMANCE WITH WATER INJECTION

By William L. Beede, Joseph T. Hamrick, and Joseph R. Withee, Jr.

SUMMARY

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An investigation was conducted to determine the effects of water injection upon the performance of a double-entry centrifugal compressor. In order to determine the effects of varying water-air ratios, the compressor was operated at a constant equivalent impeller speed over a range of water-air ratios and weight flows. Operation over a range of weight flows at one water-air ratio and two inlet-air temperatures was carried out to obtain an indication of the effects of varying inlet-air temperature. An estimate of the magnitude of evaporation within the compressor impeller with water injection based upon actual performance data is also presented.

An increase in water-air ratio from 0 to 0.055 resulted in an increase in peak total-pressure ratio of 8.1 percent and a decrease in peak adiabatic efficiency of approximately 16 percent (actual numerical decrease, 0.762 to 0.638). The maximum equivalent air weight flow increased 3.6 percent and the maximum equivalent total weight flow (air and water) increased 9.4 percent. For this compressor, beyond a water-air ratio of 0.03 there was no increase in maximum air weight flow, a negligible rise in peak total-pressure ratio, and a decrease in peak adiabatic efficiency.

An increase in inlet-air temperature resulted in an increase in the magnitude of evaporation. This increased evaporation had the same effects upon the over-all compressor-performance characteristics as those resulting from an increase in the quantity of water injected.

Analysis of data indicated that the magnitude of evaporation in the compressor impeller was small.

INTRODUCTION

Injection of water at the compressor inlet is a standard method of turbojet-engine thrust augmentation. The effects of water injection on compressor performance are discussed herein.

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Theoretical investigations (reference 1) predict large gains in pressure ratio when water is injected if the compressor efficiency can be maintained near the value for dry compression.

Several conditions resulting from the injection of water may be expected to lower efficiencies. The unstable conditions which may exist in a mixture of air and water and the heat transfer from the air to the water droplets across a finite temperature difference contribute to lower efficiencies. The change of flow conditions within the impeller and the diffuser passages due to evaporative cooling of the air may result in greater flow decelerations. Flow decelerations generally are quite large in the conventional centrifugal impeller (reference 2) and additional decelerations tend to aggravate any boundary-layer growth that is present. Increased decelerations may also change the relation between flow area and flow volume within the impeller passages causing greater tangential velocity gradients across the impeller outlet with resultant higher mixing losses.

In order to determine the changes in centrifugal compressor performance which result from the injection of water and to determine qualitatively the distribution of vaporization within the compressor, an investigation was conducted at the NACA Lewis laboratory. A double-entry centrifugal compressor was operated at a constant equivalent impeller speed over a range of water-air ratios and weight flows. Operation over a range of weight flows at one water-air ratio and two inlet-air temperatures was carried out to obtain an indication of the effect of varying inlet temperature.

SYMBOLS

The following symbols are used in this report:

$\frac{w}{a}$	water-air ratio by weight due to water injected at compressor inlet, (lb water/lb dry air)
W	actual air weight flow, (lb/sec)
θ	ratio of inlet total temperature to NACA standard sea-level temperature
δ	ratio of inlet total pressure to NACA standard sea-level pressure
$\frac{W\sqrt{\theta}}{\delta} (1.0 + \frac{w}{a})$	equivalent total weight flow, (lb/sec)

APPARATUS AND PROCEDURE

2199 Apparatus. - The double-entry centrifugal turbojet-engine compressor was driven by a 9000-horsepower variable-frequency induction motor through a speed increaser. The inlet air passed through a submerged flat-plate orifice into a stagnation tank housing the compressor. Water was injected into the inlet air stream through 14 orifice-plate-type nozzles, which were equally spaced around each of the front and rear compressor-inlet screens. The nozzles had a spray angle of approximately 65° and gave a relatively coarse spray.

During operation at a water-air ratio of zero, calibrated thermocouples were located at the compressor outlet. The remaining apparatus and instrumentation are the same as described in reference 3.

Procedure. - The effects of various water-air ratios on compressor performance were evaluated at ambient inlet-air temperature, which varied from 66° to 82° F, constant inlet-air pressure of 14 inches mercury absolute, and an equivalent impeller speed of 11,800 rpm (tip speed, 1545 ft/sec). The water-air ratio was varied from 0 to 0.055 and the weight flow was varied from conditions of choke to incipient surge. Data from these tests were used to make an estimate of the magnitude of evaporation at the inlet to the vaned-diffuser section of the compressor. Ambient temperature for these data varied from 66° to 73° F.

The effects of inlet-air temperature upon the magnitude of evaporation and the resultant effects upon over-all compressor performance were determined at an equivalent impeller speed of 10,000 rpm (tip speed, 1309 ft/sec) and a water-air ratio of 0.05. The inlet-air temperatures were approximately 60° and 100° F.

Computations. - At a water-air ratio of zero, the compressor efficiency is computed in accordance with the method of reference 4, using standard inlet and outlet total temperatures and total pressures. At water-air ratios of 0.01 to 0.055, the efficiency is the ratio of the isentropic work (reference 1) to the actual work as determined by use of a strain-gage torque meter.

An estimate of the magnitude of evaporation or vapor-air ratio at the inlet to the vaned-diffuser section of the compressor is determined by the method presented in the appendix.

RESULTS

Effect of Varying Water-Air Ratio

The variation of over-all compressor performance over a range of water-air ratios is presented in figure 1. An increase in water-air

ratio from 0 to 0.055 results in the following changes in over-all compressor performance:

- (1) An increase in peak total-pressure ratio of 8.1 percent (fig. 1(c))
- (2) An increase in maximum equivalent air weight flow of 3.6 percent and an increase in maximum equivalent total weight flow (air and water) of 9.4 percent (fig. 1(d))
- (3) A decrease in peak adiabatic efficiency of approximately 16 percent (actual numerical decrease 0.762 to 0.638) (fig. 1(e))

Effect of Varying Inlet-Air Temperature

The following results were obtained from a brief investigation of the effects of inlet-air temperature upon the magnitude of evaporation and the resultant effects upon the over-all compressor performance. At a water-air ratio of 0.05, a change in inlet-air temperature from approximately 60° to 100° F resulted in an increase in the magnitude of over-all evaporation from approximately 0.025 to 0.042 pounds of water per pound of air. These data consisted of 7 points at each temperature (60° and 100° F) and the spread of data at each temperature was within ± 0.002 . The over-all magnitude of evaporation is defined as the vapor-air ratio or specific humidity measured 34 inches downstream of the compressor outlet minus the inlet specific humidity. Because of the distance between the compressor outlet and the outlet measuring station, the over-all magnitude of evaporation is qualitative only; however, it does indicate that for a relatively small increase in inlet temperature a substantial increase in evaporation occurs through the compressor.

The effects of inlet-air temperature upon the over-all compressor performance are shown in figure 2. Representative changes in the performance variables are given in the following table:

Inlet-air temperature (°F)	Peak adiabatic efficiency	Maximum equivalent air weight flow (lb/sec)	Maximum equivalent total weight flow (lb/sec)	Peak total-pressure ratio
60	0.672	90.4	94.9	3.34
100	.656	93.2	97.8	3.50

Magnitude of Evaporation within Impeller

Calculations were made to obtain an estimate of the magnitude of evaporation within the impeller for data corresponding to zero angle of attack of the diffuser vanes (see appendix). Because installation of instruments at the impeller outlet was not feasible, this analysis was carried out from data obtained at the inlet to the vaned diffuser section. The magnitude of evaporation therefore includes the evaporation which occurs in the vaneless diffuser. As indicated in figure 3, the magnitude of evaporation is small.

DISCUSSION OF RESULTS

Effect of Varying Water-Air Ratio

The effects of water injection upon the peak total-pressure ratio, maximum equivalent total weight flow, and peak adiabatic efficiency are shown in figure 1. For this compressor, beyond a water-air ratio of 0.03 there was no increase in maximum air weight flow, a negligible rise in peak total-pressure ratio, and a decrease in peak adiabatic efficiency. In order to arrive at a detailed quantitative analysis of the observed effects of water injection, it is necessary to determine the state of the mixture at a series of stations along the flow path. Inasmuch as installation of instrumentation necessary to determine the state was not feasible, a qualitative analysis is given.

Peak total-pressure ratio. - Maximum gains in total-pressure ratio were obtained for water-air ratios up to 0.03, after which no appreciable increase was noted. Estimated values of static temperature and the quantity of water evaporated at the diffuser inlet indicate that the air was not saturated in the impeller nor at the diffuser inlet although droplets were present in the stream. Measurements obtained by use of the instruments described in reference 3 of actual quantities of water evaporated through the compressor showed that 60 to 80 percent of the water was evaporated at all water-air ratios for ambient inlet temperatures. The increase in the quantity of water evaporated at water-air ratios above 0.03 indicates that for evaporative cooling at the higher water-air ratios, the decrease in efficiency due to changes in flow conditions and thermodynamic losses offsets any gains in pressure ratio that might have been obtained.

Maximum equivalent total weight flow. - Although no gain in maximum equivalent air weight flow occurs beyond a water-air ratio of 0.02, the equivalent total weight flow increases over the entire range of water-air ratios. The increase in maximum equivalent total weight flow beyond a water-air ratio of 0.02 is due solely to the weight of water injected into the compressor. In the operation of the compressor both

with and without water injection, the flow was limited by choking at the inlet to the vaned diffuser section. With water injection the net effect of an increase in density and a decrease in through-flow velocity as effected by the cooling of the air stream is an increase in the maximum air weight flow. Inasmuch as choking occurs very near the impeller outlet only that evaporation which occurs within the impeller (and the short vaneless diffuser section) contributes to an increase in air weight flow. The water-air ratios at which the maximum equivalent air-weight-flow and total-pressure-ratio curves tend to level off would not be expected to coincide inasmuch as they do not represent the same compressor operating points.

Peak adiabatic efficiency. - The maximum rate of decrease in peak adiabatic efficiency occurred until a water-air ratio of 0.03 was reached, after which the curve tended to level off. There are several reasons for the large decrease in adiabatic efficiency which accompanies the injection of water; the most probable primary reason being the thermodynamic losses due to the evaporative cooling which occurs within the diffuser. The high rate of diffusion, which produces a rapid increase in static temperature, results in an unstable mixture of water droplets, superheated vapor, and air. The losses which result from this unstable condition in addition to those caused by heat transfer from the air to the droplets across a temperature difference may be quite large for high water-air ratios. An estimate of the amount of evaporation indicated that the evaporation within the impeller is very small and as a result the quantity of water evaporated in the diffuser becomes greater as the water-air ratio is increased. It would be difficult to eliminate this inefficiency due to unstable evaporation in the diffuser because water droplets were present at the compressor outlet even at the lowest water-air ratio ($w/a = 0.01$). The cooling in the impeller and the diffuser may also lower the aerodynamic efficiency of the compressor. Reference 2 shows that decelerations in flow are quite severe in this type compressor. Cooling of the air results in greater decelerations (more severe velocity gradients), which are highly conducive to boundary-layer build-up and separation. Experimental data show that the reduced velocities at the impeller outlet cause larger negative angles of attack at the inlet to the vaned diffuser section, which also contributes to lower over-all compressor efficiency.

Effect of Varying Inlet-Air Temperature

Increasing the inlet-air temperature resulted in an increase in evaporation. The increased evaporation had the same effects upon the over-all performance characteristics as those resulting from an increase in the quantity of water injected.

Magnitude of Evaporation within Impeller

Calculations were made to obtain an estimate of the magnitude of evaporation within the compressor impeller. It was assumed that the effective flow area at the inlet to the vaned diffuser remained constant with and without water injection and that the slip factor also remained constant. Inasmuch as assumptions were made, the calculated value of magnitude of evaporation is not exact but is a reasonable estimate and serves to show that the amount of evaporation in the impeller is small.

SUMMARY OF RESULTS

The following results were obtained in an evaluation of the performance of a centrifugal compressor with water injection:

1. An increase in water-air ratio from 0 to 0.055 resulted in an increase in peak total-pressure ratio of 8.1 percent and a decrease in peak adiabatic efficiency of approximately 16 percent (actual numerical decrease, 0.762 to 0.638). The maximum equivalent air weight flow increased 3.6 percent and the maximum equivalent total weight flow (air and water) increased 9.4 percent.

2. The effects of water injection decreased as the water-air ratio was increased. At water-air ratios greater than 0.02, the maximum equivalent air weight flow was constant. At water-air ratios greater than 0.03, there was a negligible rise in peak total-pressure ratio and a decrease in peak adiabatic efficiency.

3. An increase in inlet-air temperature resulted in an increase in the magnitude of evaporation. This increased evaporation had the same effects upon the over-all compressor performance characteristics as those resulting from an increase in the quantity of water injected.

4. Analysis of data indicated that the magnitude of evaporation in the compressor impeller was small.

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APPENDIX - METHOD FOR DETERMINING MAGNITUDE
OF EVAPORATION AT INLET TO VANED DIFFUSER

The enthalpy change of the mixture with and without water injection is used as the basis for the computation of the magnitude of evaporation. As it was not feasible to install instruments at the impeller outlet, this analysis was carried out at the inlet to the vaned diffuser section, which results in a magnitude of evaporation that includes the evaporation occurring in the vaneless diffuser section. The enthalpy change is found by determination of the static temperatures with and without water injection at the inlet to the vaned diffuser section.

In order to compute the static temperature without water injection, the flow angle α of the air entering the vaned diffuser section of the compressor must be determined. Determining the static temperature with water injection required that the flow angle α remain constant and at the same value as that used in determining the static temperature without water injection. These conditions were satisfied by selecting data wherein the angle of attack of the air entering the vaned diffuser is equal to zero.

Static temperature without water injection. - In determining the static temperature, it is necessary to find the absolute velocity V of the air entering the vaned diffuser section. This computation is made by use of the general energy and continuity equations:

$$\frac{V^2}{2gJ} = c_p (T - t) = c_p \left(T - \frac{p}{\rho_s R} \right) \quad (A1)$$

and

$$W = \rho_s A V \quad (A2)$$

Then

$$\frac{V^2}{2gJ} = c_p \left(T - \frac{pVA}{WR} \right) \quad (A3)$$

where

V absolute velocity, ft/sec

g acceleration of gravity, ft/sec²

J	mechanical equivalent of heat, 778.26 ft-lb/Btu
c_p	specific heat at constant pressure, 0.24 Btu/(lb)(°F)
T	outlet total temperature, °R
t	static temperature at inlet to vaned diffuser, °R
p	static pressure at inlet to vaned diffuser, lb/sq ft
ρ_s	static density at inlet to vaned diffuser, lb/cu ft
R	universal gas constant for air, 53.35 ft-lb/(lb)(°F)
W	air weight flow, lb/sec
A	flow area at entrance to vaned diffuser, sq ft
v	radial velocity, ft/sec

With a known air-flow angle α and a substitution of $V \sin \alpha$ for the radial velocity v , the absolute velocity V is found directly from equation (A3). The inlet-air temperatures for the data selected for this investigation (where the angle of attack at the inlet to the vaned diffuser section is zero) varied from 66° to 73° F. Because of this variation in inlet-air temperature and because the compressor was operated at a constant equivalent impeller speed, corrections were applied for variations in inlet-air temperature and actual impeller speed in order to obtain a true absolute velocity V .

The static temperature is then obtained by using equation (A1)

$$\frac{V^2}{2gJ} = c_p (T - t) \quad (A1)$$

Static temperature with water injection. - In order to determine the static temperature with water injection, it is again necessary to select data wherein the angle of attack of the air entering the vaned diffuser is zero. The static temperature can then be determined by use of the continuity relation:

$$\frac{W_D}{W_W} = \frac{\rho_s D^2 V_D}{\rho_{s,W} A_W V_W} \quad (A4)$$

where s denotes static conditions and subscripts W and D represent values obtained with and without water injection, respectively. The tangential velocities are made equal by applying corrections for variations

in the actual impeller speed and by assuming that the slip factor is the same both with and without water injection. (Actual power measurements showed the slip factor to be constant within the accuracy of measurement.) As the air-flow angle α is constant (by selection of data) and the tangential velocities are equal, the absolute through-flow velocities of equation (A4) are equal and can be eliminated. If it is assumed that the effective flow area remains constant, equation (A4) can be written:

$$\frac{W_D}{W_W} = \frac{\rho_{s,D}}{\rho_{s,W}} = \frac{\frac{P_D}{Rt_D}}{\frac{P_W}{Rt_W}} \quad (A5)$$

Because of the small amounts of water used in this investigation, the change in R with water injection is negligible; therefore,

$$\frac{W_D}{W_W} = \frac{P_D t_W}{t_D P_W} \quad (A6)$$

where W is the actual air weight flow. In the solution of equation (A6), all values are corrected for variations in inlet-air temperature and actual impeller speed. To be rigorous the term W_W should include the weight of water which has evaporated in the impeller; however, as determination of this quantity is the purpose of the analysis, the weight of water is neglected. Results of the analysis indicate that this assumption is adequate.

Magnitude of evaporation. - The changes in the enthalpy of the mixture with and without water injection is determined using the computed inlet and outlet static temperatures. The quantity of water evaporated is then determined by assuming that the changes in enthalpy are due entirely to the evaporation of the water droplets and that the water vapor is at the same temperature as the surrounding air.

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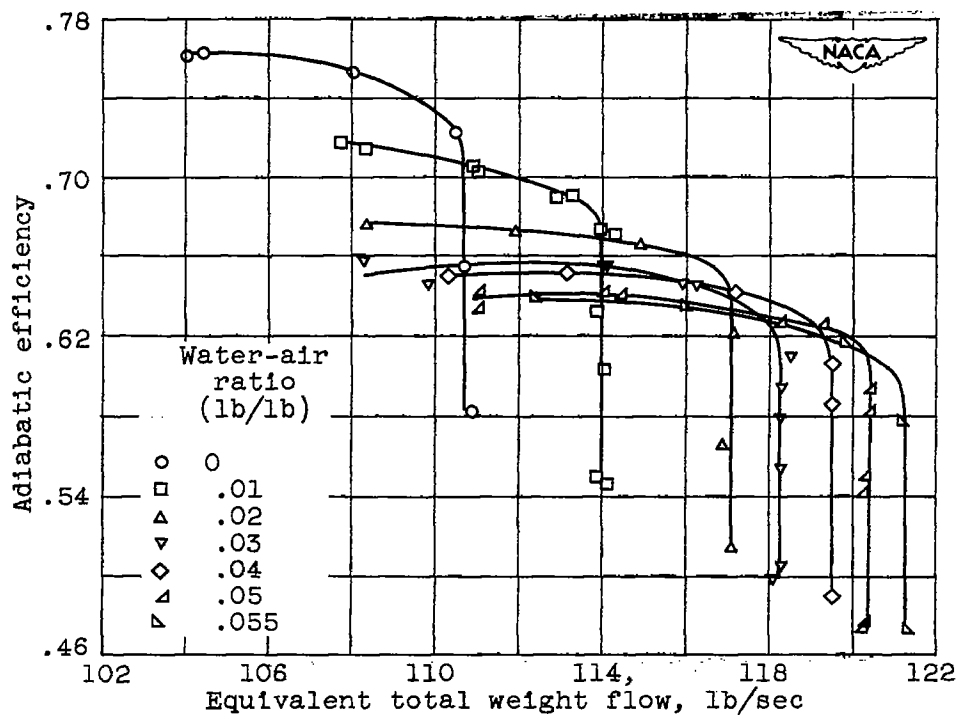
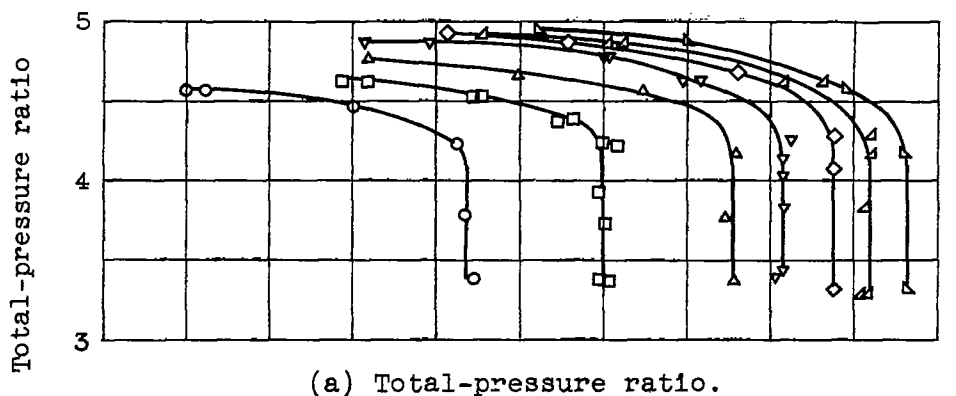
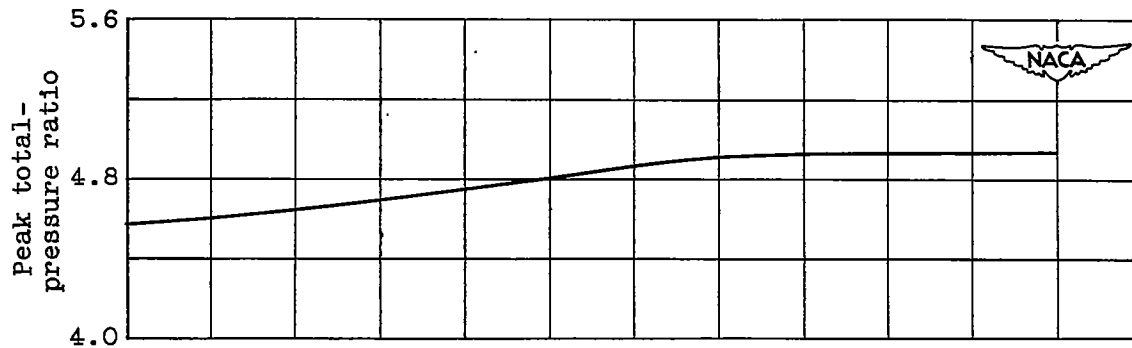
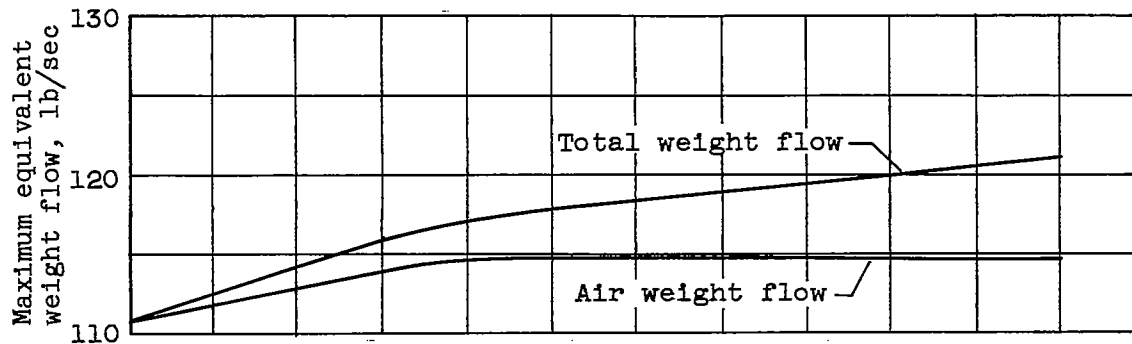


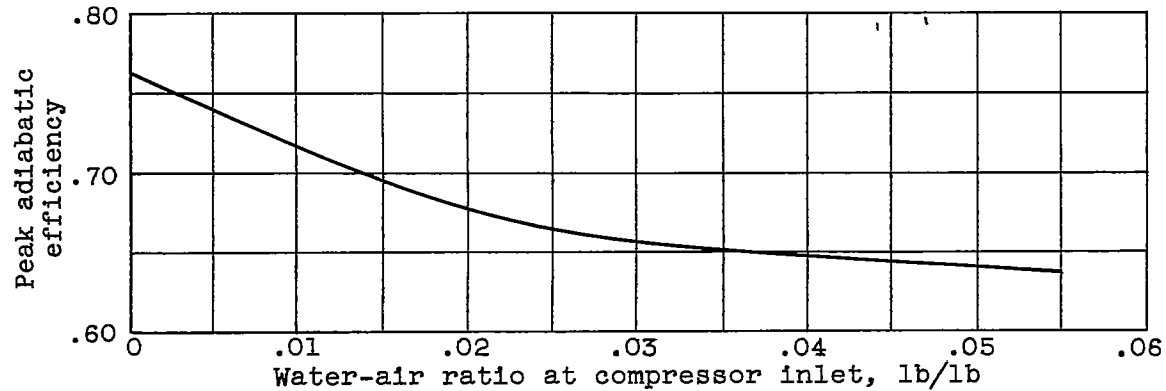
Figure 1. - Performance of centrifugal compressor with water injection at equivalent impeller speed of 11,800 rpm (tip speed, 1545 ft/sec), constant inlet pressure of 14 inches mercury absolute, and ambient inlet-air temperatures from 66° to 82° F.



(c) Peak total-pressure ratio.



(d) Maximum equivalent weight flow.



(e) Peak adiabatic efficiency.

Figure 1. - Concluded. Performance of centrifugal compressor with water injection at equivalent impeller speed of 11,800 rpm (tip speed, 1545 ft/sec), constant inlet pressure of 14 inches mercury absolute, and ambient inlet-air temperatures from 66° to 82° F.

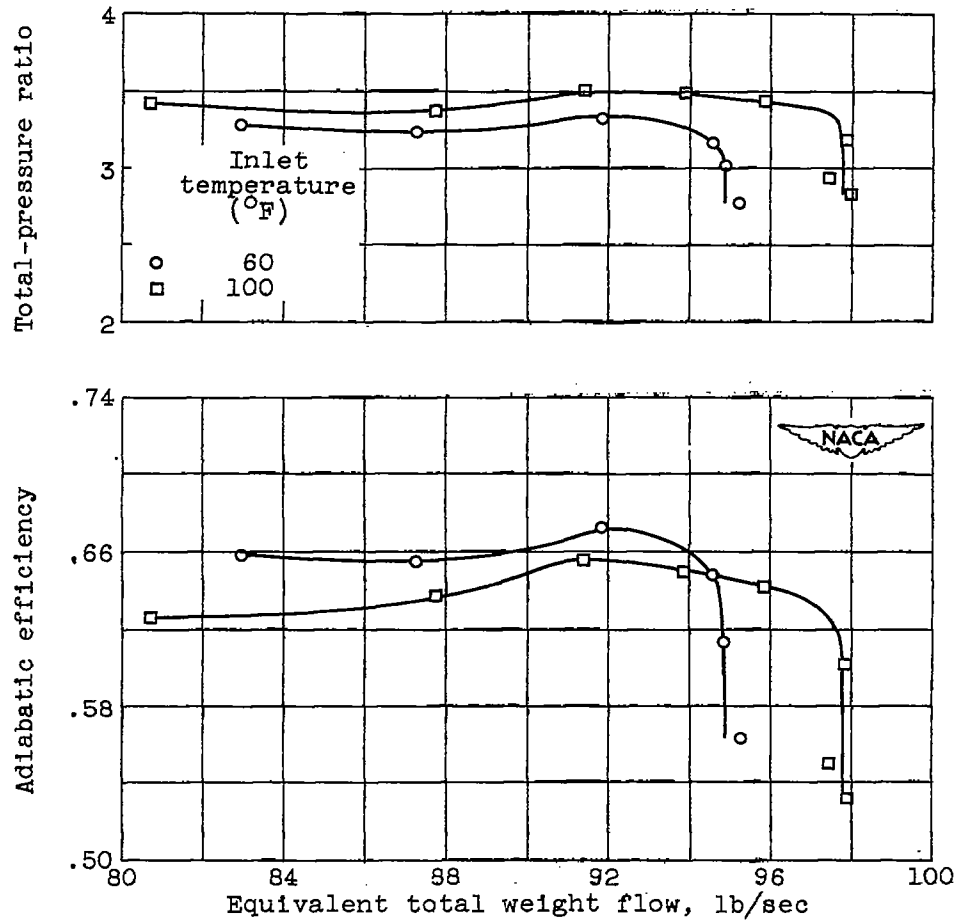


Figure 2. - Variation of centrifugal-compressor performance with inlet-air temperature at equivalent impeller speed of 10,000 rpm (tip speed, 1309 ft/sec), constant inlet pressure of 14 inches mercury absolute, and water-air ratio by weight of 0.05.

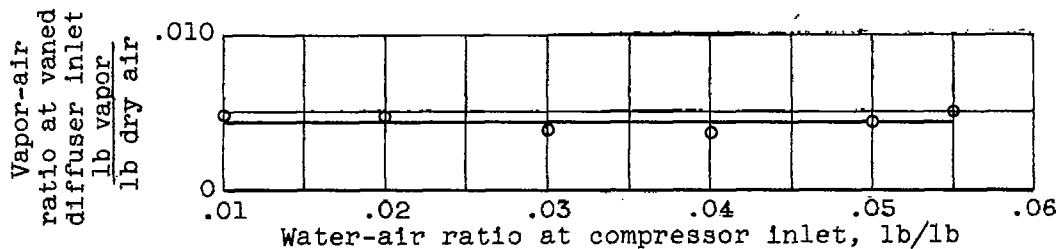


Figure 3. - Performance of centrifugal compressor with water injection at equivalent impeller speed of 11,800 rpm (tip speed, 1545 ft/sec), constant inlet pressure of 14 inches mercury absolute, ambient inlet-air temperatures from 66° to 73° F, and zero angle of attack at inlet to vaned diffuser.

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