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PERFORMANCE OF THE 19XB 10-STAGE AXIAL-FLOW COMPRESSOR WITH ALTERED BLADE ANGLES

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PERFORMANCE OF THE 19XB 10-STAGE AXIAL-FLOW COMPRESSOR WITH ALTERED BLADE ANGLES

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SUMMARY

Previous performance data of the 19XB axial-flow compressor indicated that the outlet guide vanes and possibly the inlet guide vanes were stalling. Calculations were made to determine if these adverse conditions could be eliminated and if the manufacturer's design specifications could be more nearly approached by altering the blade angles of the first few compression stages as well as the outlet guide vanes. With the blade angles altered, experimental data were taken at compressor speeds of 8500 to 17,000 rpm with inlet-air conditions of 14 inches of mercury absolute and 59° F.

The temperature-rise efficiency increased with speed from 0.70 at 8500 rpm to 0.74 at 13,600 rpm and dropped gradually to 0.70 at 17,000 rpm. At the design speed of 17,000 rpm, the pressure ratio at the peak efficiency point was 3.63. The maximum pressure ratio at design speed was 4.15 at an equivalent weight flow of 29.8 pounds per second. The altered compressor operated very near the design specifications of pressure ratio and equivalent weight flow.

At the high speeds, the peak adiabatic temperature-rise efficiency was increased 0.02 to 0.06 by altering the blade angles. The peak pressure ratio was increased 0.29 at design speed (17,000 rpm) and 0.05 and 0.18 at 11,900 and 13,600 rpm, respectively. The equivalent weight flow through the altered compressor was reduced 2 pounds per second at 15,300 and 17,000 rpm, as was expected from the design calculations. As extreme caution was taken not to surge the compressor violently, the point of minimum air flow may not have been reached in the present investigation and in a previous investigation. A true comparison of the pressure ratios obtained at the high speeds therefore cannot be made.

INTRODUCTION

The performance of the 10-stage axial-flow compressor from the 19XB jet-propulsion engine was investigated at the NACA Cleveland laboratory at the request of the Bureau of Aeronautics, Navy Department. The performance of the compressor received from the manufacturer is presented in reference 1. At an inlet pressure of 14 inches of mercury absolute with all interstage instruments installed and with no air leakage past the rotor rear air seal of the original compressor, the equivalent weight flow through the compressor was approximately 2 pounds per second higher than the 30 pounds per second for which the unit was designed and the adiabatic temperature-rise efficiency at the design speed was 0.68, somewhat lower than had been expected. The design pressure ratio of the unit was 4.17 but the maximum pressure ratio developed was 3.87 (reference 1). The results of interstage surveys indicated that the outlet guide vanes had been operating in a stalled condition. Oil-flow patterns on the compressor blading and calculations substantiated this indication and provided evidence to show that the inlet guide vanes were also probably stalled.

In an effort to approach more nearly the design specifications of the compressor with a few simple changes, the possibilities of altering the blade angles of the inlet guide vanes, the first several stages, and the second row of outlet guide vanes were analytically considered. This analysis showed that substantial improvements in performance could be expected by changing the angles of the inlet guide vanes, the rotor and the stator blades of the first stage, the rotor blades of the second stage, and the second row of outlet guide vanes.

The performance characteristics of the altered compressor were determined for a range of equivalent speeds from 8500 to 17,000 rpm with inlet-air conditions of 14 inches of mercury absolute and 59° F. The investigation was made with interstage instruments installed and with no leakage past the rotor rear air seal; the performance of the altered compressor can therefore be compared with that of the original compressor (reference 1) for speeds of 8500 to 17,000 rpm.

BLADE-ANGLE ALTERATIONS

Lift coefficients for the tip sections of the inlet guide vanes and the first rotor blades were calculated from the design information furnished by the manufacturer and were found to be 1.4 and 0.7, respectively. The high lift coefficient at the tip of the inlet guide vanes indicated that a portion of the blades might be stalled, in which case a similar condition might exist at the tip of the first-stage rotor blades. The patterns of oil and dust particles that accumulated on the blades during operation indicated that a stalled condition apparently extended over approximately the outer 10 percent of the inlet guide vanes and the outer 15 to 20 percent of the first-stage rotor blades. The scope of the analytical investigation to improve the performance of the 19XB compressor by eliminating the stalled condition was confined to the possibilities of altering the blade angles of the compressor by plastically deforming the blades within one-fourth inch of the mounts. The investigation was first directed toward improving the air flow at the inlet with minimum changes to the blading of the compressor. In this analysis, the radial distribution of axial velocities after each blade row was determined on the assumption that radial pressure equilibrium exists between the blades rows; that is, the pressure forces balance the centrifugal forces.

The stalling of the inlet guide vanes was relieved by reducing the lift coefficient at the tip section from 1.4 to 1.043. This change reduced the theoretical turning angle of the air from 20° to 14°, increased the angle of attack at the first-stage rotor blades, and increased the mean Mach number relative to the first-stage rotor blades from 0.8 to 0.9. Inasmuch as local sonic velocities and flow separation would be expected if the lift coefficient of the first-stage rotor blades was too high, the lift coefficient of these blades was reduced to 0.6.

The combined changes of the inlet guide vanes and first-stage rotor blades caused the absolute velocity of the air entering the first-stage stator blades to have a much lower tangential component than that of the original design. By reducing the lift coefficient of the first-stage stator blades to an extremely low value (-0.29 at the tip section), the relative tangential component of the air entering the second-stage rotor blades could be adjusted to such a value that an increase in the lift coefficient of the second-stage rotor blades to 0.73 at the tip section would not only compensate for the energy increment lost in the first-stage rotor blades but would also cause the flow at the second-stage stator blades to have the same configuration as the original design. These changes in the blade angles were designed to eliminate flow separation at the compressor inlet without changing the work done by the compressor at the design point. Because critical Mach numbers at the first-stage rotor blades will be obtained at lower values of mass flow than in the original design, optimum performance of the compressor would be expected at slightly lower mass flows than before (reference 2).

Because a high turning angle of the second row of outlet guide vanes was required to give an axial discharge, the lift coefficient for this row of blades was 1.1. The probability that stall was occurring was indicated by large total-pressure losses across the second row of outlet guide vanes (reference 1). These guide vanes were simply adjusted to lower the angles of attack in order to decrease the stalling tendency. The maximum lift coefficient of these blades in the new position was approximately 0.7.

The following table gives the original and altered blade angles measured from the tangent line along the lower surface of the airfoil to an axial plane:

Blade row	Blade angle, deg					
	Tip		Midspan		Hub	
	Original	Altered	Original	Altered	Original	Altered
Inlet guide vanes	20.0	14.0	16.0	13.5	12.0	13.0
First-stage rotor	38.4	52.0	32.5	37.0	26.6	22.0
First-stage stator	38.8	50.0	33.3	41.1	27.7	32.2
Second-stage rotor	39.3	41.5	34.1	36.4	28.9	31.2
Second row of outlet guide vanes	15.2	29.0	15.2	33.0	15.2	37.0

The decrease in losses resulting from the elimination of the stalled condition of the blades was expected to compensate for the decrease in the ideal pressure recovery.

EQUIPMENT AND PROCEDURES

The performance of the altered compressor was determined in the same setup as that described for the investigation with no air leakage in reference 1. Air was supplied to the compressor by the laboratory refrigerated-air system and was metered by a submerged adjustable orifice located in the inlet duct upstream of the compressor. The air was discharged into a collector fitted with two radial outlet pipes, which were, in turn, connected to the laboratory altitude-exhaust system.

Over-all measurements of pressure and temperature were taken as described in reference 1, using the methods recommended in reference 3. The inlet measurements were taken in a depression tank so large that the air velocity at the inlet measuring station was negligible; the total pressure could therefore be obtained with two wall static-pressure taps. The inlet-air temperature was measured with three unshielded thermocouples. Discharge measurements were taken halfway between the outlet guide vanes and the second bearing-support struts in a straight annular area. Six shielded axial-vent temperature probes were used to measure the temperature at the discharge measuring station. The discharge total pressures were calculated from the observed static pressures, total temperatures, and mass flow by the method given in reference 3.

Although interstage instruments were installed to give the same conditions of air flow as in reference 1, the interstage measurements were not recorded because the already large number of running hours to which the compressor had been subjected would have been greatly increased.

As in reference 1, data were taken with inlet-air conditions of 14 inches of mercury absolute and 59° F over a range of equivalent compressor speeds from 8500 to 17,000 rpm in increments of 1700 rpm.

The compressor performance is presented as recommended in reference 3. Adiabatic temperature-rise efficiency η_T, over-all total-pressure ratio P_2/P_1, and pressure coefficient ψ are plotted against the equivalent weight flow $W\sqrt{\theta}/\delta$ for various values of equivalent speed $N/\sqrt{\theta}$ and equivalent tip speed $U_t/\sqrt{\theta}$, where θ is the ratio of inlet-air total temperature to NACA standard sea-level temperature and δ is the ratio of inlet-air total pressure to NACA standard sea-level pressure. Adiabatic temperature-rise efficiency contours are also shown on the curves of pressure ratio plotted against equivalent weight flow.

All measurements were made within the following limits:

Temperature, °F	±0.5
Pressure, inches of mercury	±0.02
Weight flow, percent	±1.0
Compressor speed, percent	±0.5
Compressor efficiency, percent	±1.0

Pressure and temperature measurements at the outlet may be in error owing to gradients in a radial and a circumferential direction. The interstage instruments probably had an adverse effect on the compressor performance, but this error was present in both the investigation of reference 1 and this investigation and should have little effect on the accuracy of the comparison.

RESULTS AND DISCUSSION

The performance of the altered 19XB compressor for speeds from 8500 to 17,000 rpm with inlet-air conditions of 14 inches of mercury absolute and 59° F is shown in figure 1. The adiabatic temperature-rise efficiency increased with speed from 0.70 at 8500 rpm to a peak of 0.74 at 13,600 rpm. With a further increase in speed, the efficiency gradually dropped to 0.70 at the design speed of 17,000 rpm. The peak pressure coefficient gradually decreased with an increase in speed, dropping from 0.27 at 8500 rpm to 0.23 at 17,000 rpm. The pressure ratio at the point of peak efficiency at design speed was 3.63. The maximum pressure ratio obtained was 4.15 at 17,000 rpm with an efficiency of 0.66 and an equivalent weight flow of 29.8 pounds per second. The altered compressor operated very near the original design conditions of pressure ratio and equivalent weight flow. Altering the compressor-blade angles improved the performance of the 19XB compressor for inlet-air conditions of 14 inches of mercury absolute and 59° F (fig. 2). At high speeds, the peak adiabatic temperature-rise efficiencies were increased from 0.02 to 0.06 over that obtained with the original compressor. The peak pressure ratio was increased 0.05 at 11,900 rpm, 0.18 at 13,600 rpm, and 0.29 at 17,000 rpm (the design speed). At 8500, 10,200, and 15,300 rpm, however, no changes were observed in the peak pressure ratios. The maximum pressure ratio and efficiency may not have been obtained with the original or the altered compressor because of the difficulty in determining the exact location of the surge point and a different degree of surging may have been obtained for each compressor.

The changes in the blade angles of the inlet guide vanes, the first-stage rotor blades, and the second-stage rotor blades resulted in the same calculated energy addition in the first two stages as for the original compressor and would therefore result in the same over-all pressure ratio for both compressors if the efficiencies are the same. Part of the observed increase in pressure ratio is attributable to the alleviation of flow separation at the inlet and outlet guide vanes. The comparatively large tangential component of air velocity behind the outlet guide vanes represents a loss in pressure ratio, which could be reclaimed by the proper conversion of the kinetic energy of rotation to static pressure.

As was expected from the design calculations, the equivalent weight flow through the altered compressor at the design speed was less than the corresponding flow through the original compressor. At the speeds of 15,300 and 17,000 rpm, this decrease amounted to 2 pounds per second. Little effect on the equivalent weight flow was observed at low speeds because the entire compressor is then operating at conditions far different from the design conditions.

SUMMARY OF RESULTS

The performance of the 19XB axial-flow compressor with altered blade angles is summarized as follows:

1. With inlet conditions of 14 inches of mercury absolute and 59° F, the peak temperature-rise efficiency increased with speed from 0.70 at 8500 rpm to a maximum of 0.74 at 13,600 rpm and dropped gradually to 0.70 at the design speed, 17,000 rpm. The pressure ratio at the point of maximum efficiency at design speed was 3.63. The maximum pressure ratio obtained at design speed was 4.15 with an efficiency of 0.66 and an equivalent weight flow of 29.8 pounds per second. The altered compressor operated very near the original design conditions of pressure ratio and equivalent weight flow.

2. At high speeds, the peak adiabatic temperature-rise efficiencies were increased 0.02 to 0.06 by altering the blade angles. The peak pressure ratio was increased 0.29 at design speed and 0.05

and 0.18 at 11,900 and 13,600 rpm, respectively. The equivalent weight flow through the altered compressor was 2 pounds per second less than that through the original compressor at 15,300 and 17,000 rpm.

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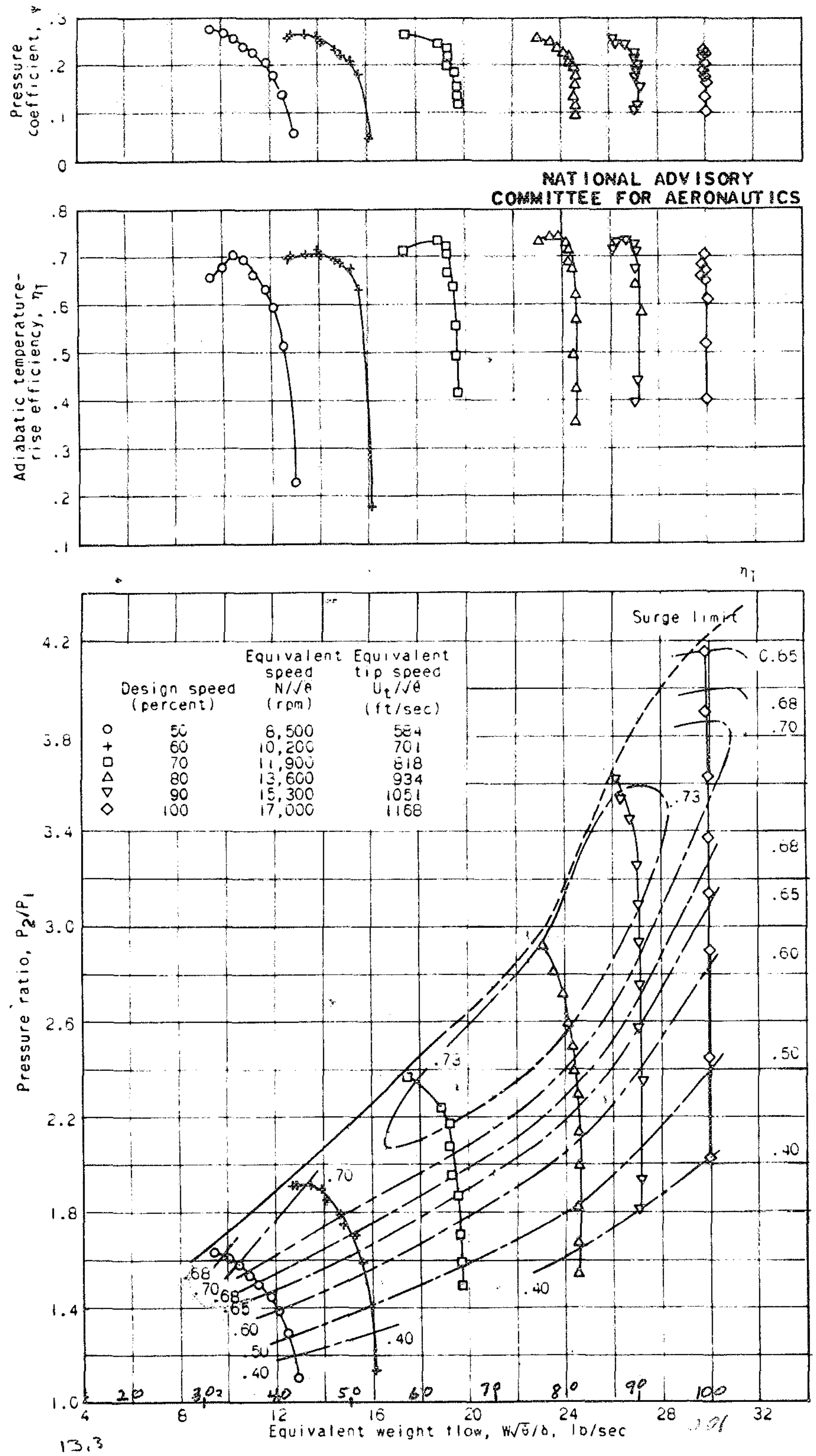

Figure 1. - Performance characteristics of altered 19XB compressor. Inlet-air pressure, 14 inches of mercury absolute; inlet-air temperature, 59° F.

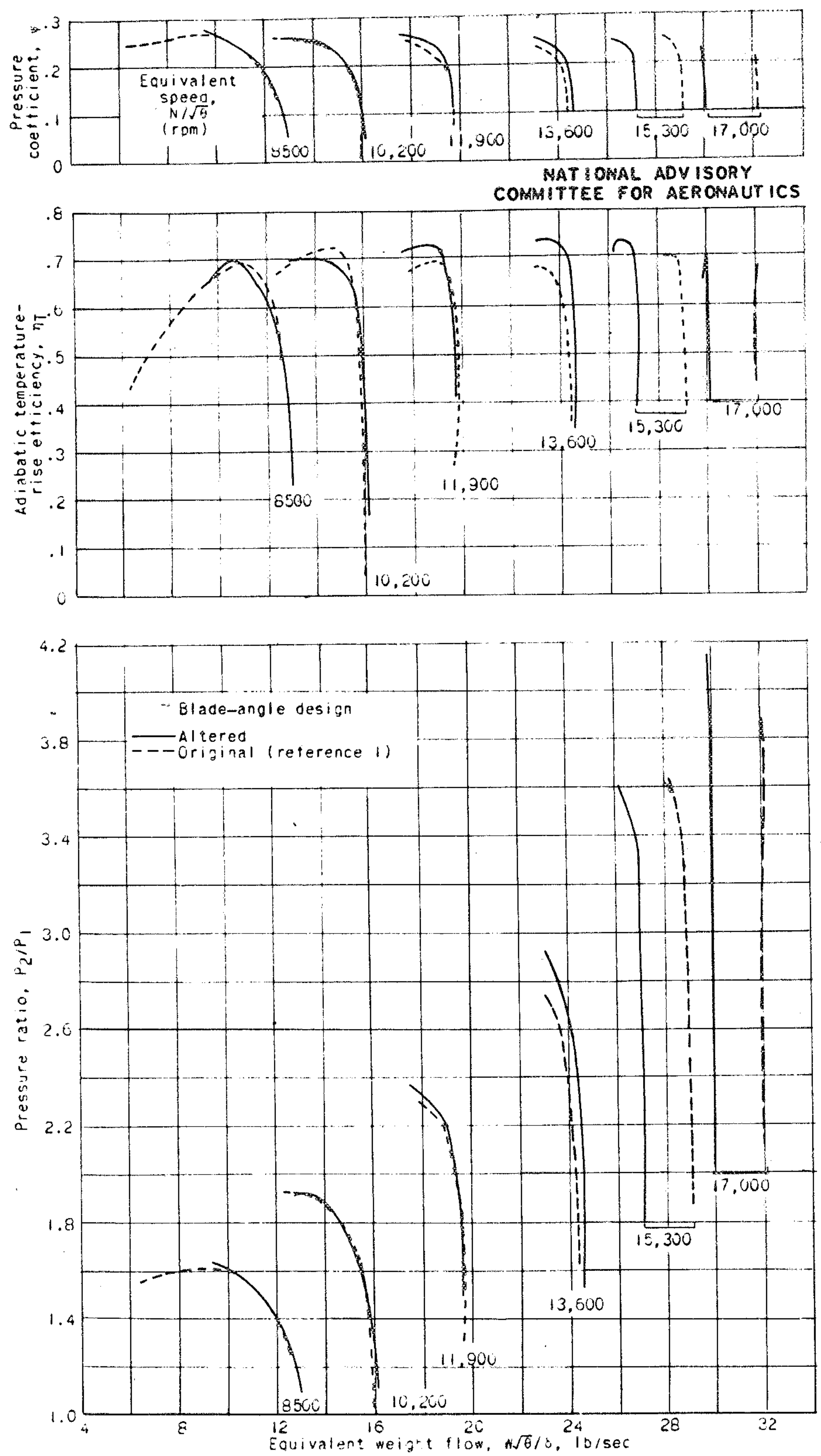

Figure 2. - Comparison of performance of original and altered 19XB compressor. Inlet-air pressure, 14 inches of mercury absolute; inlet-air temperature; 59° F.