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EFFECT OF SLOTTED CASING TREATMENT ON PERFORMANCE OF A MULTISTAGE COMPRESSOR

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16. Abstract

A J85-13 engine was equipped with a compressor case that allowed changes to the case wall over the rotor tips of six of its eight stages. The engine was tested with four inlet configurations: uniform inlet flow, 180° circumferential distortion, hub radial distortion, and tip radial distortion. Slotted inserts were installed in the first three stages, and the compressor was mapped under similar conditions. Overall compressor performance obtained with tip treatment (slotted inserts) was inferior to the performance for the compressor's normal operating range (base data). Pumping capacity with the slotted inserts was reduced. Overall compressor efficiency was reduced 1 to 2 percent with the slotted rings installed for 90 and 100 percent corrected engine speeds.

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EFFECT OF SLOTTED CASING TREATMENT ON PERFORMANCE OF A MULTISTAGE COMPRESSOR

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SUMMARY

A J85-13 engine was equipped with a compressor case that allowed changes to the casewall over the tips of six of its eight compressor stages. This engine was run at Lewis in an altitude test facility at four inlet conditions: uniform inlet flow, 180° circumferential distortion, and hub and tip radial distortion. Compressor stalls were induced by closing the exhaust nozzle.

Baseline compressor maps were generated for the four inlet conditions with solid (untreated) compressor case inserts. Slotted compressor case inserts, which provided the tip treatment over the rotor blades, were then installed in the first three compressor stages, and the compressor was mapped under the same four inlet conditions.

Overall compressor performance obtained with tip treatment (slotted inserts) was generally inferior to the base data. For 100 percent of rated corrected engine speed, the stall pressure ratio with tip treatment was lower than the baseline ratio; however, for 80 percent corrected engine speed, the stall pressure ratio was greater with tip treatment. Pumping capacity with tip treatment for all inlet configurations was reduced. Overall compressor efficiency was reduced 1 to 2 percent with the slotted rings installed for 90 and 100 percent corrected engine speeds. For 80 percent of rated corrected engine speed, the overall compressor efficiency increased (less than 1/2 percent) for tip treatment with the clean inlet and with tip-radial distortion.

These results indicate that tip treatment will not improve the performance of the J85-13 compressor in its normal operating range. This engine is hub critical, therefore, tip treatment had no beneficial effect on stall margin.

Most stalls occurred in the sixth or eighth stage for the baseline and the seventh stage for the slotted inlet configuration; however, it was very difficult to locate the stall sites precisely.

INTRODUCTION

A program was established at Lewis to evaluate the results of tip treatment on a multistage compressor. Grooved, slotted, and porous tip treatments have been shown to have the desirable effects on single-stage compressors (see refs. 1 to 3). These tip treatments have been effective in increasing the flow range, stall margin, and distortion tolerance of the compressor.

An analytic study (ref. 4) showed that, if tip treatment could modify the individual stage characteristic of the J85 as it had in the single-stage compressor work, a gain from 16.8 to 29.0 percent in the stall margin could be realized.

To evaluate this prediction of increased performance experimentally, a special compressor case was fabricated for the J85-GE-13 turbojet engine. The case was equipped with removable inserts over the tips of the rotor blades of stages 1, 2, 3, 6, 7, and 8. Three sets of inserts were provided: one set was left untreated for baseline testing; one set was machined with circumferential grooves; and one set was machined with blade angle slots. The results of an experimental program conducted with the circumferential grooved inserts and with the slotted inserts in the last three stages are reported in reference 5.

This report presents the results obtained with the slotted inserts installed in the first three compressor stages. Data obtained with slotted casing inserts are compared with those of the baseline case at three corrected engine speeds (80, 90, and 100 percent of rated), with the inlet undistorted, with 180° circumferential distortion, with hub radial distortion, and with tip radial distortion. Overall compressor maps are presented.

APPARATUS

The compressor case of this J85-13 engine was designed to permit insertion of casing wall treatment over the rotor tips of the first three and the last three of its eight stages. This was effected by providing segmented rings (fig. 1), T-shaped in cross section, which slipped into mating grooves in the compressor case. Figure 2(a) shows half of the compressor case with rings in place; figure 2(b) shows the case with the rings removed. The eighth-stage ring and groove are not apparent in these photographs; this ring is located between the compressor case and the main frame.

Figure 1 is a sketch and dimensions of the slotted rings. The angle of the slots are parallel to the chords of the rotor blades at their respective stages.

The engine's variable inlet guide vanes are linked to compressor third-, fourth-, and fifth-stage bleed doors such that when the guide vanes are fully closed, the bleed doors are fully open. These guide vanes were operated according to the manufacturer's schedule: bleed doors varied linearly from fully open at a corrected engine speed of 80 percent to fully closed at 94 percent. The inlet temperature bias on the bleed schedule was removed to allow comparison of results independent of small variations in inlet temperatures.

Since this was a stall program, a first-stage turbine nozzle with approximately 74 percent of the nominal flow area was used. This reduced area elevated the compressor operating line and allowed compressor stalls without excessive turbine-inlet temperatures. The exhaust nozzle was manually controlled to effect compressor stall. Six plates were added to the inner surface of the exhaust nozzle to increase the blockage. The range of the nozzle area was from 400 to 1130 square centimeters.

This engine was tested without distortion and with three classical screen patterns at the inlet: 180° circumferential, 360° tip radial, and 360° hub radial. The distortion screens were attached to a support structure 44.1 centimeters (about one compressor face diameter) upstream of the compressor face. Details of the support structure are given in reference 7.

Figure 3(a) shows the 180° circumferential distortion screen. All the screens used were made up of 0.081-centimeter-diameter wire, spaced 0.282 centimeter apart (9 mesh) yielding approximately 50 percent blockage. The radial distortion screen covered 40 percent of the inlet area. Figures 3(b) and (c) show the tip radial and hub radial distortion screens.

At 100 percent corrected engine speed, typical distortions ((average high pressure - average low pressure)/average face pressure) of 12 to 13 percent were observed. At 80 percent of rated corrected engine speeds, typical distortions were 6 to 7 percent, and at 90 percent corrected engine speed, 8 to 10 percent.

INSTRUMENTATION

Figure 4(a) shows the array of pressure probes located 3.7 centimeters upstream of the compressor face. The probes on each rake were area weighted radially. Figure 4(b) shows the array of 12 thermocouples located 7.9 centimeters upstream of the engine face. Closely coupled, high-response static-pressure transducers were provided at each stator row to locate the compressor stall sites.

Compressor discharge instrumentation was installed through the four bleed ports at the rear of the compressor. At each port three total-pressure, one static-pressure, and three total-temperature probes were used. A sketch of the pressure probe is presented in reference 5. At the compressor discharge the following weightings were used in averaging the data: 20 percent for the inner probe, 60 percent for the middle probe, and 20 percent for the outer probe.

Engine airflow was calculated from pressure and temperature measured in a plenum upstream of the inlet by using a previously determined inlet bellmouth calibration and station 1 static pressure as presented in appendix A. (The symbols are defined in appendix B.)

PROCEDURES

Tests were conducted with ambient inlet temperature and a nominal inlet pressure of 6.6 newtons per square centimeter. These inlet conditions yielded a Reynolds number index of 0.70 at the compressor face. The altitude chamber was maintained at approximately 3.1 newtons per square centimeter to ensure a choked exhaust nozzle.

For the two configurations, with and without casing treatment, the compressor was mapped with an undistorted inlet and with screens designed to produce 180° circumferential, 360° hub radial, and 360° tip radial distortions. Each mapping consisted of three constant corrected speed lines (80, 90, and 100 percent of rated). Several steady-state data points were taken along each speed line from wide-open exhaust nozzle to stall.

As data were taken along the constant corrected speed lines, turbine-discharge total temperature was observed on a control room gage and recorded. This gage was monitored carefully as the engine was stalled to obtain the turbine-discharge temperature at stall. Data obtained in this way were used to draw curves of compressor pressure ratio and corrected airflow as a function of turbine-discharge temperature. These curves were extrapolated to the turbine-discharge temperature at stall, thus determining the compressor pressure ratio and corrected airflow at the stall point.

As stall was approached, continuous data were recorded from the high-response interstage static-pressure transducers. These data were used to locate stall sites.

RESULTS AND DISCUSSION

Compressor Performance

Figure 5 presents the compressor efficiency and the compressor performance map for uniform inlet flow. The efficiency was computed to account for bleed flow at the lower speeds as shown in appendix A. (Symbols are defined in appendix B.) Baseline data were used to compare results obtained with untreated rings with various types of inlet distortion, with tip treatment with uniform inlet flow, and with tip treatment with various types of inlet distortion.

Effect of Distortions

Tip-radial. - Figure 6 presents the effect of tip-radial distortion on compressor performance with untreated rings. Figure 6(a) shows a drop in overall compressor efficiency from 1 to 2 percentage points with inlet distortion. Figure 6(b) shows a 4 percent drop in stall pressure ratio for the 100 percent speed.

Hub-radial. - Figure 7 presents the effect of hub-radial distortion on the compressor performance with untreated rings. Figure 7(a) shows an increase in compressor efficiency with hub-radial distortion for the three corrected engine speeds. Figure 7(b) shows a sizable loss in the corrected airflow capacity of the compressor at the two higher speeds. Because compressor pressure ratio did not fall off as fast as corrected engine airflow, the map shows a slight net increase in overall compressor stall margin. This shift of the compressor stall lines indicates that this compressor is sensitive to hub-radial distortion.

Effect of Tip Treatment

Undistorted. - Figure 8 presents the effect of slotted tip treatment in the first three stages with uniform inlet flow. Figure 8(a) shows a 3 percent drop in compressor efficiency at the higher speed, the other speeds not being greatly affected. Figure 8(b) shows an increase with tip treatment in the stall pressure ratio at the lower two speeds and a drop in the stall pressure ratio at the 100 percent of rated speed. Pumping capacity is down for all speeds with tip-treatment.

Tip-radial distortion. - Figure 9 presents the combined effect of tip-treatment and tip-radial distortion. Figure 9(a) shows a drop in compressor efficiency with slotted rings and distortion for the two higher speeds. No significant trend exists at 80 percent rated speed. Figure 9(b) shows a 4 percent loss in stall pressure ratio at the 100 percent of rated speed due to the tip treatment with distortion.

Hub-radial distortion. - Figure 10 presents the combined effect of tip treatment and hub radial distortion. As stated previously, hub-radial distortion increased the baseline compressor efficiency. Figure 10(a) shows that tip treatment with hub-radial distortion decreases the compressor efficiency. However, at the two lower speeds the efficiency is still higher than that of the baseline. Figure 10(b) shows loss in pumping capacity of 2 to 3 percent. Tip treatment with hub-radial distortion lowered the stall pressure for the higher two speeds; however, the stall pressure ratio was increased for the 80 percent speed.

Circumferential distortion. - Figure 11 shows a 2 to 3 percent loss in pumping capacity for the compressor with tip treatment with an 180° circumferential distortion. No efficiency was calculated for the 180° circumferential distortion because of the lack of instrumentation to obtain sufficiently accurate data at the compressor exit. Data were not obtained for the 180° circumferential distortion with the untreated rings, nor are stall data available for this configuration.

The results of this report and of reference 5 show that tip-treatment will not improve the performance of the J85-13 compressor in its normal operating range. For this engine the hub stall margin is less than the tip stall margin; therefore, tip treatment was not beneficial to stall margin.

Stall Sites

A summary of stall sites is presented in table I. These stall sites were determined from recordings of the high-response interstage static-pressure transducers located in the tip wall of each stator row. The stage exhibiting the first drop in tip static pressure is listed as the location of the stall origin. The judgment of which pressure dropped off first was often subjective; arguments could be made for two or more stage pressures. This ambiguity is indicated by the multiple entries in the table.

For untreated rings, with or without distortion, the sixth and/or eighth stages of the compressor appeared to be the predominate stall sites. For tip treatment, with or without distortion, the seventh stage was the predominate stall site. Thus, no significant effect of tip treatment was observed on the location of the first stage to stall.

SUMMARY OF RESULTS

Compressor performance of a J85-13 engine with slotted casing treatment in the first three stages was compared with that for the untreated case. Data were obtained for rotor speeds from 80 to 100 percent of rated with no inlet distortion and with screen-induced 180° circumferential, 360° hub radial distortion, and 360° tip radial distortions. The following results were obtained:

1. Casing treatment was not effective in extending flow range or increasing stall margin.

2. Slotted tip-treated inserts reduced the stall pressure ratio at high rotor speed (100 percent corrected engine speed).

3. Slotted inserts reduced the compressor efficiency at 90 and 100 percent rotor speeds; however, the inserts increased the compressor efficiency at 80 percent rotor speeds.

4. Slotted inserts reduced the pumping capacity at all rotor speeds.

The results of this report and reference 5 show that tip treatment will not improve the performance of the J85-13 compressor in its normal operating range. This engine is hub critical; therefore, tip treatment did not have a beneficial effect on stall margin.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 10, 1975,
505-05.

APPENDIX A

METHOD OF CALCULATIONS

Airflow. - Primary airflow at the engine was calculated from the following equation:

$$W_1 = C_D A \frac{P_{tot,p}}{R(T_p)} \left[\left(\frac{P_{s1}}{P_{tot,p}}\right)\right]^{1/\gamma} \sqrt{2gRT_p \frac{\gamma}{\gamma - 1}\left(1 - \frac{P_{s1}}{P_{tot,p}}\right)^{(\gamma - 1)/\gamma}}$$

Compressor efficiency. - The overall compressor efficiency was calculated from the following equation:

$$\eta = \frac{\left(\frac{P_{tot,3}}{P_{tot,2}}\right)^{(\gamma-1)/\gamma} - 1}{\frac{(W_3)T_3 + (B_1)T_{2.3} + (B_2)T_{2.4} + (B_3)T_{2.5}}{W_1 T_2} - 1}$$

APPENDIX B

SYMBOLS

A	flow area, m^2
B	bleed door flow rate, kg/sec
C_D	flow coefficient
g	acceleration due to gravity 9.8066 m/sec^2
P_s	static pressure, N/cm^2
P_{tot}	total pressure, N/cm^2
$P_{tot,p}$	plenum total pressure, N/cm^2
R	gas constant, J/kg·K
T	total temperature, ^oC
T_p	plenum temperature, ^oC
W	airflow rate, kg/sec
γ	ratio of specific heats
η	efficiency

Subscripts:

1	inlet airflow measuring station
2	compressor inlet
2.3	third stage stator
2.4	fourth stage stator
2.5	fifth stage stator
3	compressor discharge

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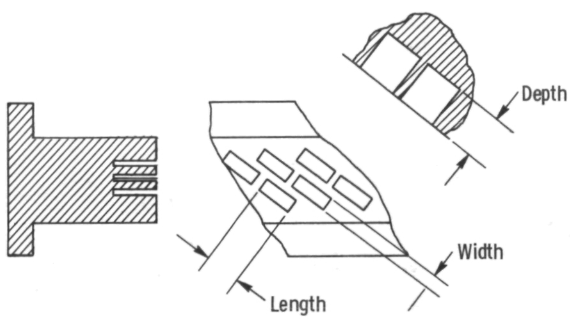
TABLE I. - SUMMARY OF STALL SITES

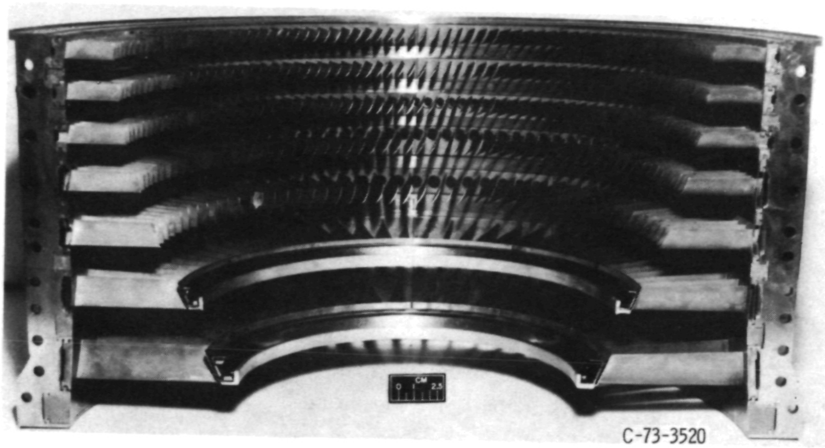
Inlet distortion	Corrected engine speed, percent of rated		
	80	90	100
	Stall site, stage		
Untreated rings			
None	6, 8	7	6, 8
180° circumferential	(a)	(a)	(a)
Tip radial	8	6, 8	7
Hub radial	6, 8	(b)	(b)
Slotted rings in stages 1, 2, and 3			
None	6	7	7
180° circumferential	(a)	(a)	(a)
Tip radial	7	6	8
Hub radial	6, 8	7	7

[a]Transient recordings were not available.
[b]Stall sites could not be determined.

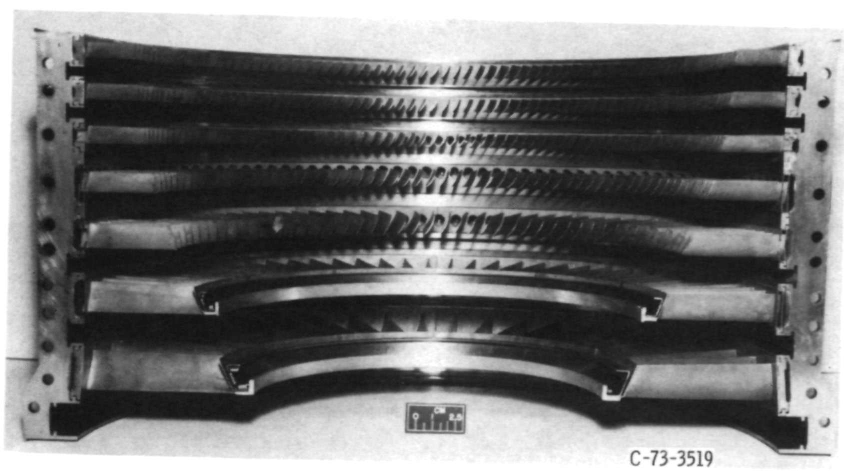


Stage	Number of pairs of slots	Width, cm	Length, cm	Depth, cm
1	240	0.208	0.932	0.610
2	364	.152	.536	.457
3	388	.137	.297	.406

Figure 1. - Sketch and dimensions of rings.

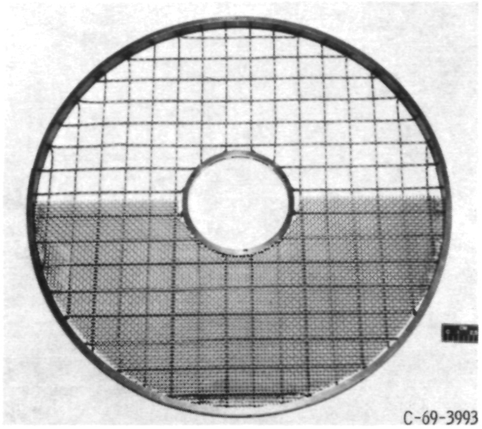


(a) With inserts for case wall treatment in place.

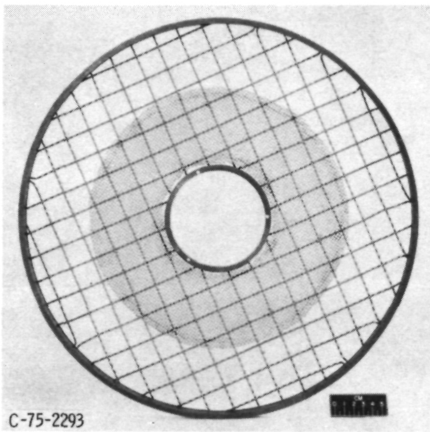


(b) Case wall treatment removed.

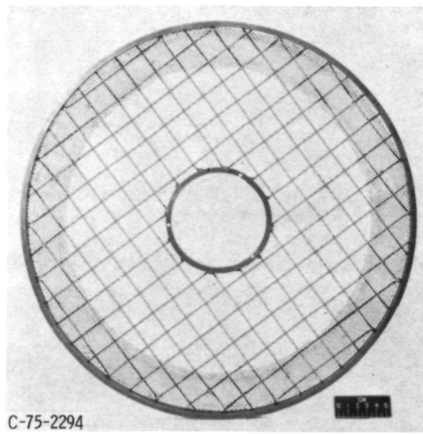
Figure 2. - Compressor case.



(a) 180° circumferential distortion.

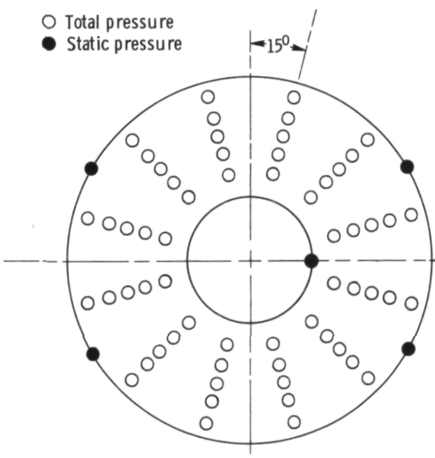


(b) Tip radial distortion.

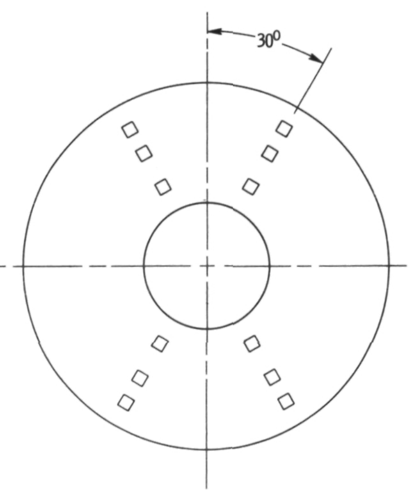


(c) Hub radial distortion.

Figure 3. - Screen pattern.



(a) Pressure instrumentation.



(b) Total temperature instrumentation.

Figure 4. - Compressor face instrumentation schematic as viewed from upstream.

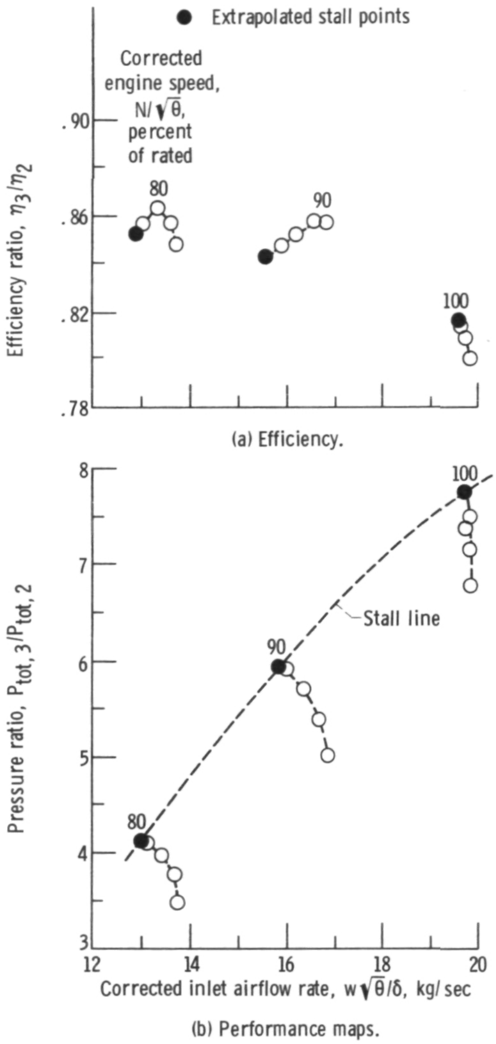


Figure 5. - Baseline compressor performance with untreated rings and uniform inlet flow.

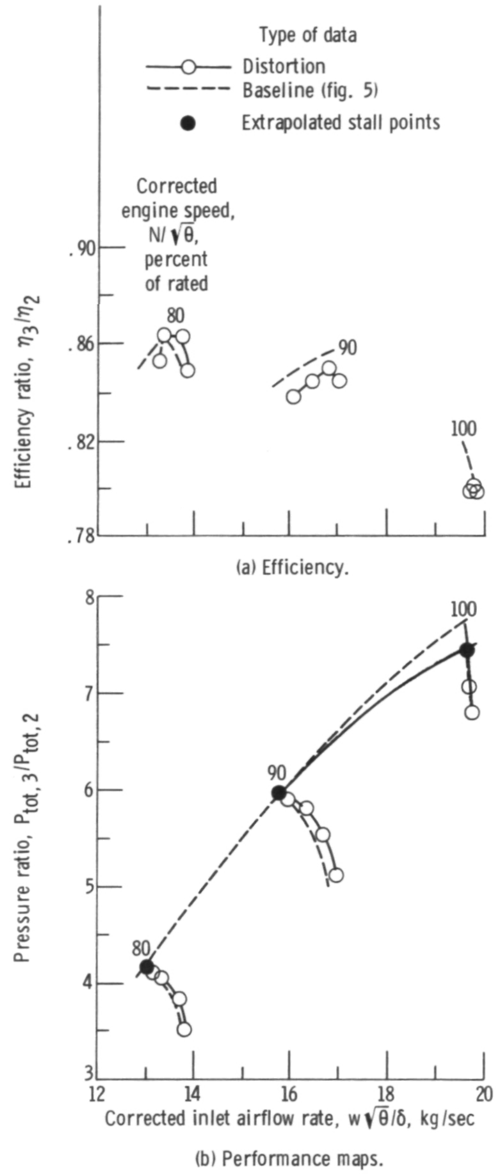


Figure 6. - Compressor performance with untreated rings and tip radial distortion.

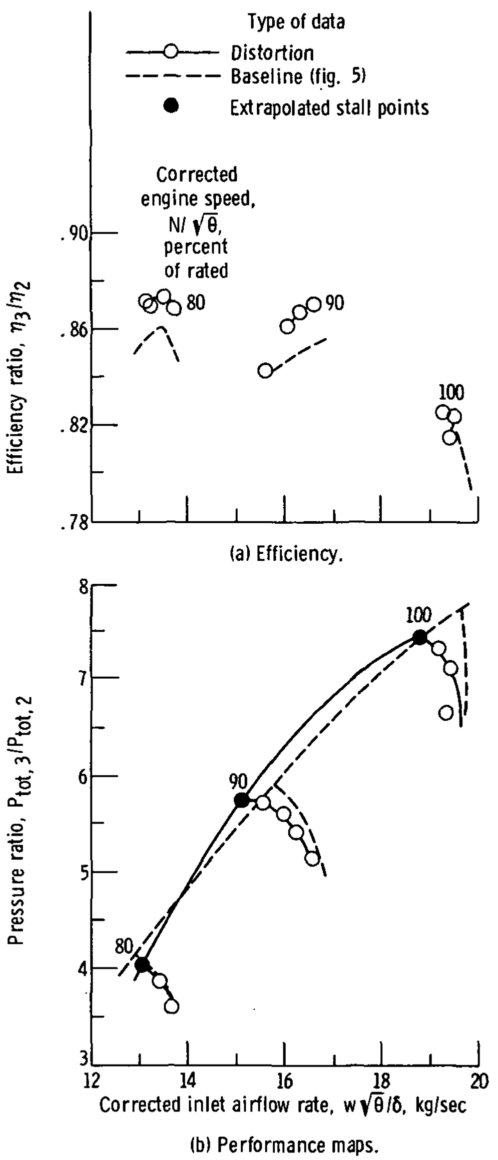


Figure 7. - Compressor performance with untreated rings and with hub radial distortion.

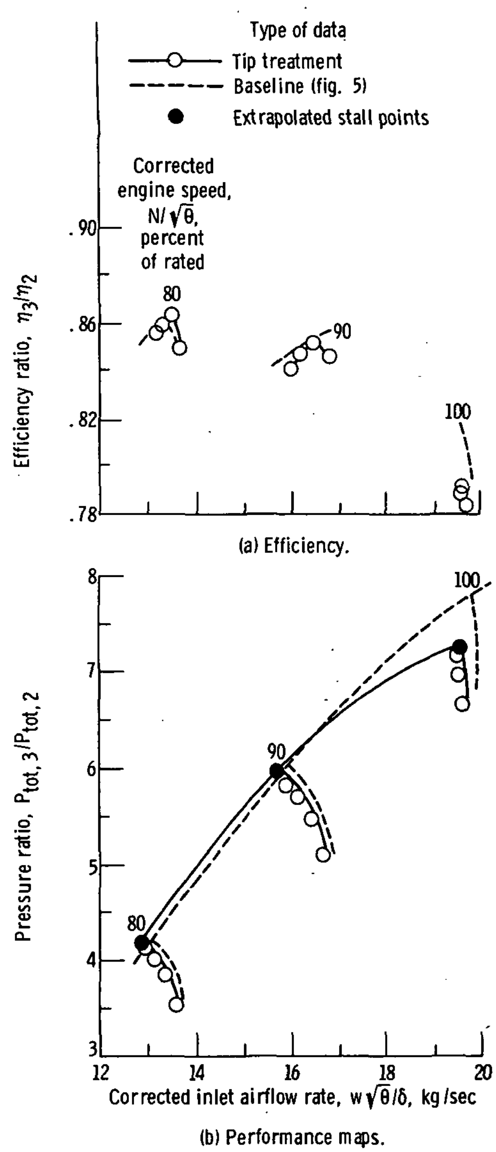


Figure 8. - Compressor performance with slotted rings in stages 1 to 3 and uniform inlet flow.

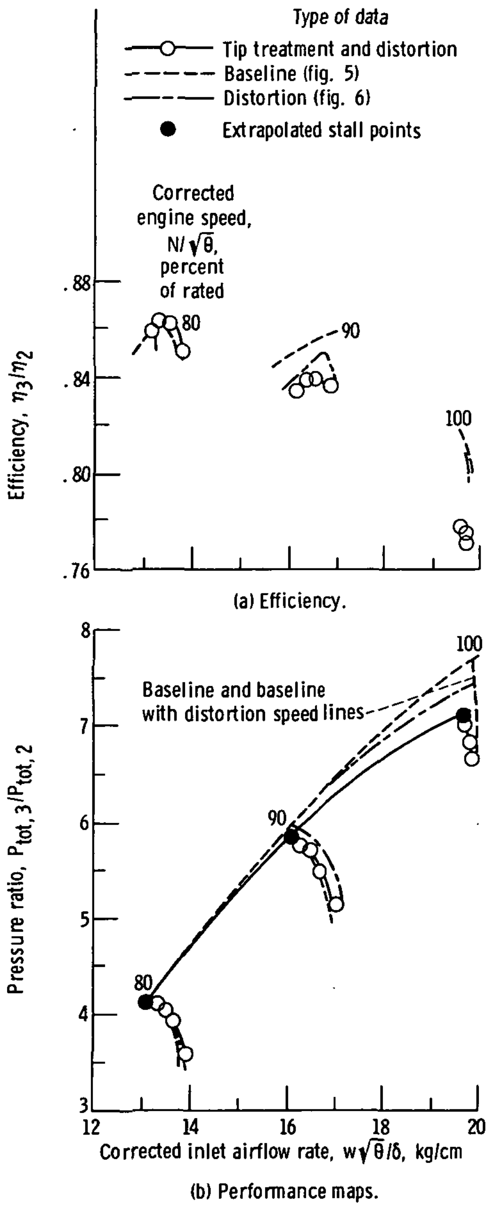


Figure 9. - Compressor performance with slotted rings in stages 1 to 3 and tip radial distortion.

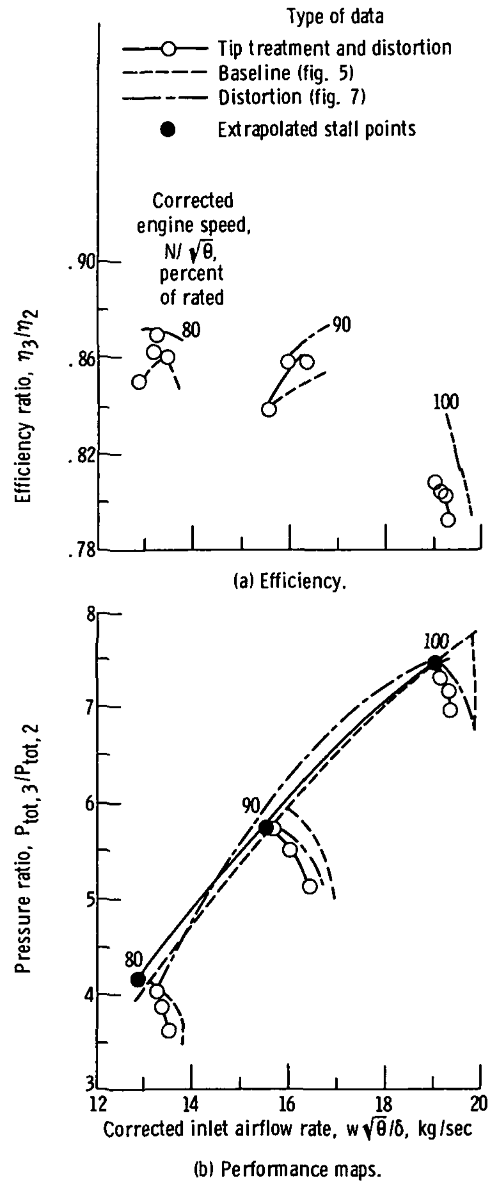


Figure 10. - Compressor performance with slotted rings in stages 1 to 3 and hub radial distortion.

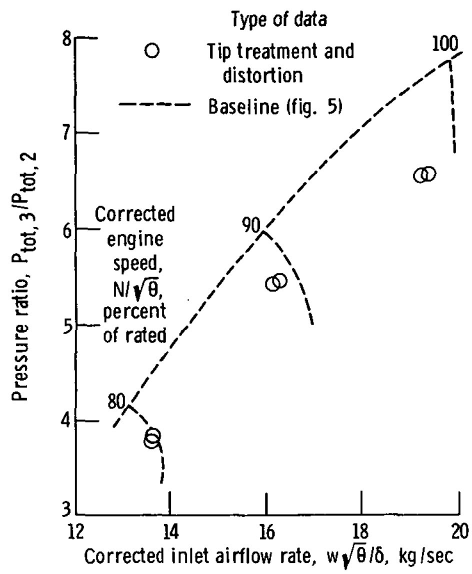


Figure 11. - Compressor performance with slotted rings in stages 1 to 3 and with 180^0 circumferential distortion.

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