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OVERALL PERFORMANCE IN ARGON OF 4.25-INCH SWEPTBACK-BLADED CENTRIFUGAL COMPRESSOR

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16. Abstract

Overall compressor performance is presented as a function of equivalent weight flow. At the design equivalent weight flow of 0.581 lb/sec (0.263 kg/sec) and the design equivalent tip speed of 949 ft/sec (289 m/sec), the compressor produced an overall total-pressure ratio of 1.92 and an adiabatic efficiency of 0.795. The peak efficiency at design speed was 0.819 and occurred at an equivalent weight flow of 0.508 lb/sec (0.230 kg/sec).

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SUMMARY

A 4.25-inch (10.8-cm) diameter centrifugal compressor applicable for a nominal 6-kilowatt, single-shaft, Brayton-cycle spacepower system was tested using argon gas as the working fluid. Overall performance is presented as a function of equivalent weight flow, and a comparison with predicted performance is made.

At the design equivalent weight flow of 0.581 pound per second (0.263 kg/sec) and the design equivalent tip speed of 949 feet per second (289 m/sec), the compressor produced an overall total-pressure ratio of 1.92 and an adiabatic efficiency of 0.795. The peak efficiency at design speed was 0.819 and occurred at an equivalent weight flow of 0.508 pound per second (0.230 kg/sec). As the speed was reduced from the design value to 50 percent of design, the peak efficiency increased from 0.819 to about 0.83.

INTRODUCTION

The NASA Lewis Research Center is presently engaged in a research program on small centrifugal and axial-flow compressors for application to closed Brayton-cycle space electric power generation systems. The program is directed primarily toward establishing the performance level that can be achieved by these machines, and toward determining the effect of Reynolds number on their performance. In the case of the centrifugal compressors, the effect of tip clearance and diffuser vane setting angle on flow range and performance is also being investigated.

As part of this program, a study was conducted on a 6-inch (15.2-cm) diameter radial-bladed centrifugal compressor. This compressor was designed for a two-shaft, 10-kilowatt, Brayton-cycle space power system. The overall performance of this compressor was reported in reference 1. A peak efficiency of 78 percent, at design speed, was attained. The effect of Reynolds number on the performance of this compressor was reported in reference 2. A progressive degradation in performance resulted from decreasing Reynolds number. This degradation in performance is compared to the predicted performance based on the commonly used 0.2 power relation of loss with Reynolds number. The effect of changing the diffuser vane setting angle on the flow range and performance of the 6-inch (15.2-cm) centrifugal compressor was reported in reference 3. For the range of diffuser vane setting angles tested, the flow rate at which the peak efficiency occurred varied from 100 to 60 percent of design flow without any significant change in efficiency level.

More recently studies have been conducted on a 4.25-inch (10.8-cm) diameter sweptback-bladed centrifugal compressor designed for a single-shaft, Brayton-cycle system having a nominal electrical power output of 6 kilowatts. The compressor design and fabrication was accomplished under contract by the AiResearch Manufacturing Company of Phoenix, Arizona (ref. 4). The overall performance of the compressor, using argon gas, was obtained at Lewis and is reported herein. The report presents the overall performance for six speeds from 50 to 100 percent of design speed, for weight flows from maximum flow to compressor surge. For all tests, the compressor inlet conditions were set at nominal design values of total pressure and temperature, 20 psia (13.8 N/cm^2 abs) and 540° R (300 K).

COMPRESSOR DESIGN

The compressor was designed for a nominal 6-kilowatt, single-shaft, Brayton-cycle space electrical power system. The working fluid specified for the system was a mixture of helium-xenon (HeXe) gas having a molecular weight of 83.8. Argon gas was used for this test program instead of the design gas mixture of helium-xenon because it had the same specific-heat ratio and was more readily available. The compressor design is presented in reference 5. A summary of the compressor design point values for the reference design power level and the specified working fluid are given in table I.

TABLE I. - COMPRESSOR DESIGN PARAMETERS

[Working fluid, HeXe mixture with molecular weight of 83.8.]

Inlet total pressure, P_1, psia; N/cm^2 abs	13.5; 9.31
Inlet total temperature, T_1, $^\circ$R; K	540; 300
Weight flow rate, W, lb/sec; kg/sec	0.756; 1.667
Compressor total-pressure ratio, P_6/P_1	1.9
Compressor total-temperature ratio, T_6/T_1	1.37
Compressor efficiency, η_{1-6}	0.80
Impeller total-pressure ratio, P_3/P_1	2.03
Impeller efficiency, η_{1-3}	0.895
Rotative speed, RPM, rpm	36 000
Impeller tip speed, U_{t3}, ft/sec; m/sec	667; 203
Specific speed, N_s	0.11

The compressor design values for argon gas and equivalent values based on standard inlet pressure and temperature are presented in table II for the test fluid.

The compressor impeller has 15 blades that are curved backward at the exit 30° from the radial. The impeller inlet blade tip diameter is 2.60 inches (6.60 cm) and the hub diameter is 1.44 inches (3.66 cm). The impeller exit diameter is 4.25 inches (10.8 cm) and the exit blade height is 0.205 inch (0.521 cm). Design velocity diagrams for the impeller inlet and outlet are shown in figure 1 for argon as the working fluid.

From the impeller exit to the diffuser vane inlet, there is a vaneless diffuser section about 0.085 inch (0.216 cm) long in the radial direction. There are 17 vanes in the vaned diffuser section. These diffuser vanes are a constant height of 0.213 inch (0.541 cm) and are an integral part of the scroll assembly. The diffuser design velocity diagrams are shown in figure 2.

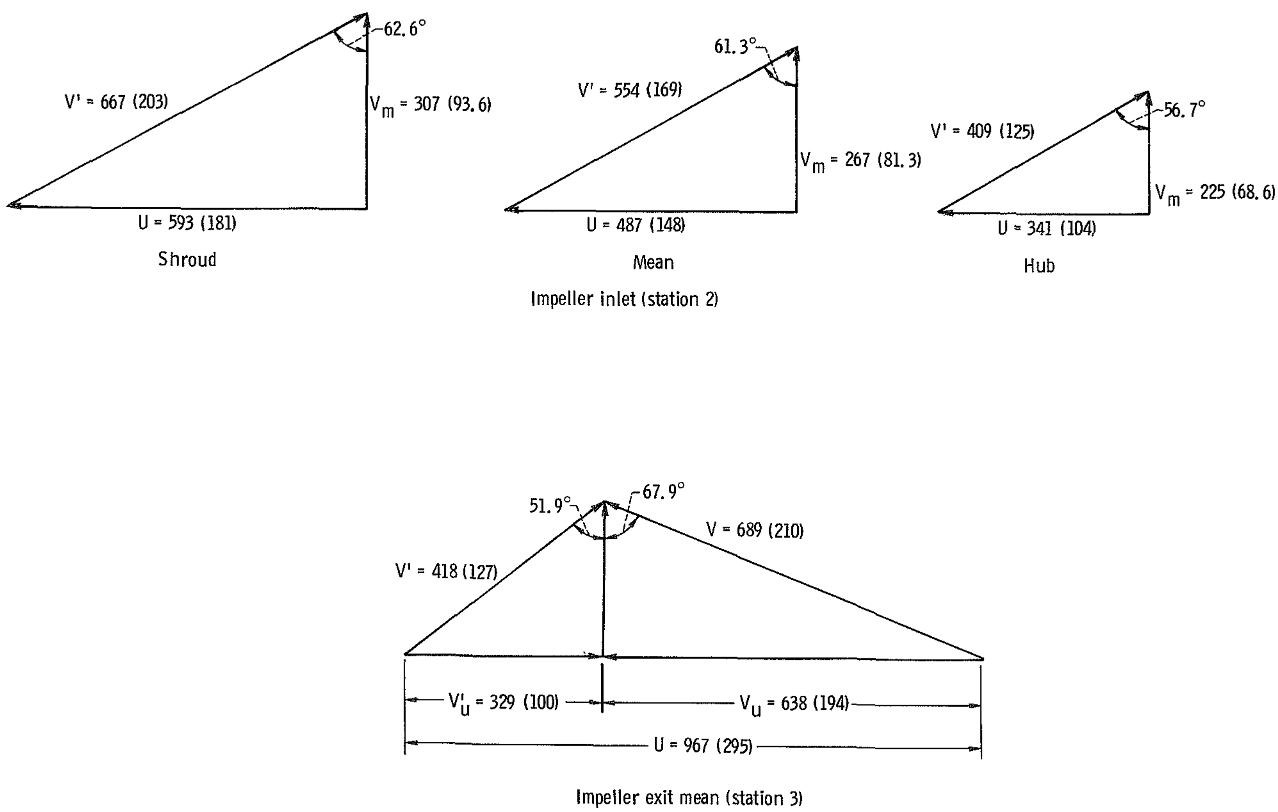
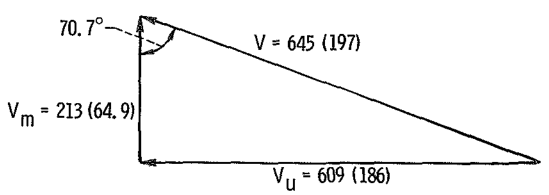


Figure 1. - Impeller design velocity diagrams for argon gas. (All dimensions in ft/sec (m/sec) unless otherwise indicated.)

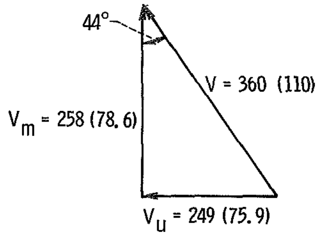
TABLE II. - COMPRESSOR DESIGN PARAMETERS

[Working fluid, argon]

Inlet total pressure, P_1, psia; N/cm^2 abs	20.25; 13.96
Inlet total temperature, T_1, ^oR; K	540; 300
Weight flow rate, W, lb/sec; kg/sec	0.785; 0.356
Equivalent weight flow, $W\sqrt{\theta}/\delta$	0.581; 0.263
Compressor total-pressure ratio, P_6/P_1	$^a1.9$
Compressor total-temperature ratio, T_6/T_1	$^a1.37$
Compressor efficiency, η_{1-6}	$^a0.80$
Impeller total-pressure ratio, P_3/P_1	$^a2.03$
Impeller efficiency, η_{1-3}	$^a0.895$
Rotative speed, RPM, rpm	52 200
Equivalent speed RPM/$\sqrt{\theta}$, rpm	51 176
Impeller tip speed U_{t3}, ft/sec; m/sec	968; 295
Equivalent impeller tip speed, $U_{t3}/\sqrt{\theta}$, ft/sec; m/sec	949; 289
Compressor work factor, f_{cw}	0.658
Specific speed, N_s	0.11
Reynolds number, Re	3.1×10^6



Diffuser inlet (station 4)



Diffuser exit (station 5)

Figure 2. - Diffuser design velocity diagrams for argon gas. (All dimensions in ft/sec (m/sec) unless otherwise indicated.)

APPARATUS AND PROCEDURE

Test Apparatus

The impeller and scroll used in this investigation were made of stainless steel and are shown in figure 3. The impeller was cantilever-mounted on a shaft supported by two angular contact bearings. The bearings were lubricated and cooled by air-oil mist. To prevent oil leakage, a carbon face seal was located outboard of the bearing on the impeller end of the shaft. A labyrinth-type seal was used on the other end of the shaft. The static (cold) impeller clearance between the shroud and the blade tip at the impeller discharge was set at 0.008 inch (0.020 cm). The compressor was fully enclosed in insulation during testing, as shown in figure 4, to minimize the heat transfer out of the system between the inlet and outlet measuring stations.

Test Facility

A diagram of the open-loop-type compressor test facility used for this investigation is shown in figure 5. This facility was also utilized for the investigations reported in references 1 to 4. All tests were conducted using high-purity argon gas (contaminants less than 50 ppm by volume) that was supplied by high-pressure tube trailers. The inlet gas temperature was maintained constant by an automatically controlled 25-kilowatt electric heater. Inlet pressure was controlled by a remotely operated pressure control valve, and gas flow and exit pressure by a remotely operated globe valve installed downstream of the research compressor. The argon gas was discharged into a laboratory exhaust system. A single-stage axial-flow air turbine drove the compressor. The desired drive turbine speed was automatically maintained by a remotely controlled throttle valve that regulated the airflow to the drive turbine.

Instrumentation

A cutaway view of the compressor with the locations of instrument measurement and calculation stations is shown in figure 6. Overall compressor performance was computed using the pressures and temperatures at the inlet (station 1) and exit (station 6). A cross-sectional view of the instrumentation at stations 1 and 6 is shown in figure 6. At station 1 the instrumentation consisted of three wall-static-pressure taps equally spaced around the circumference and three combination total-pressure - total-temperature rakes.

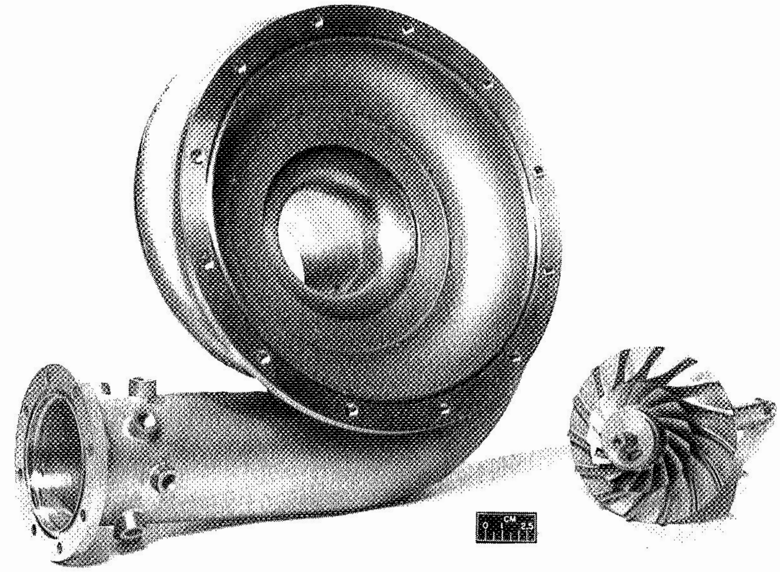


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Figure 3. - Compressor vaned diffuser - scroll assembly and impeller.

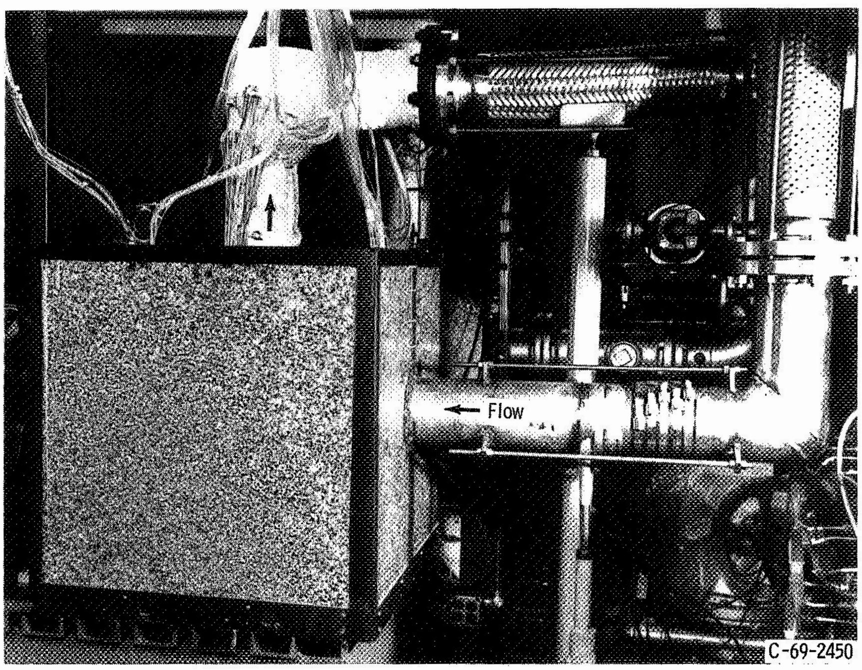

Figure 4. - Compressor fully enclosed in insulation.

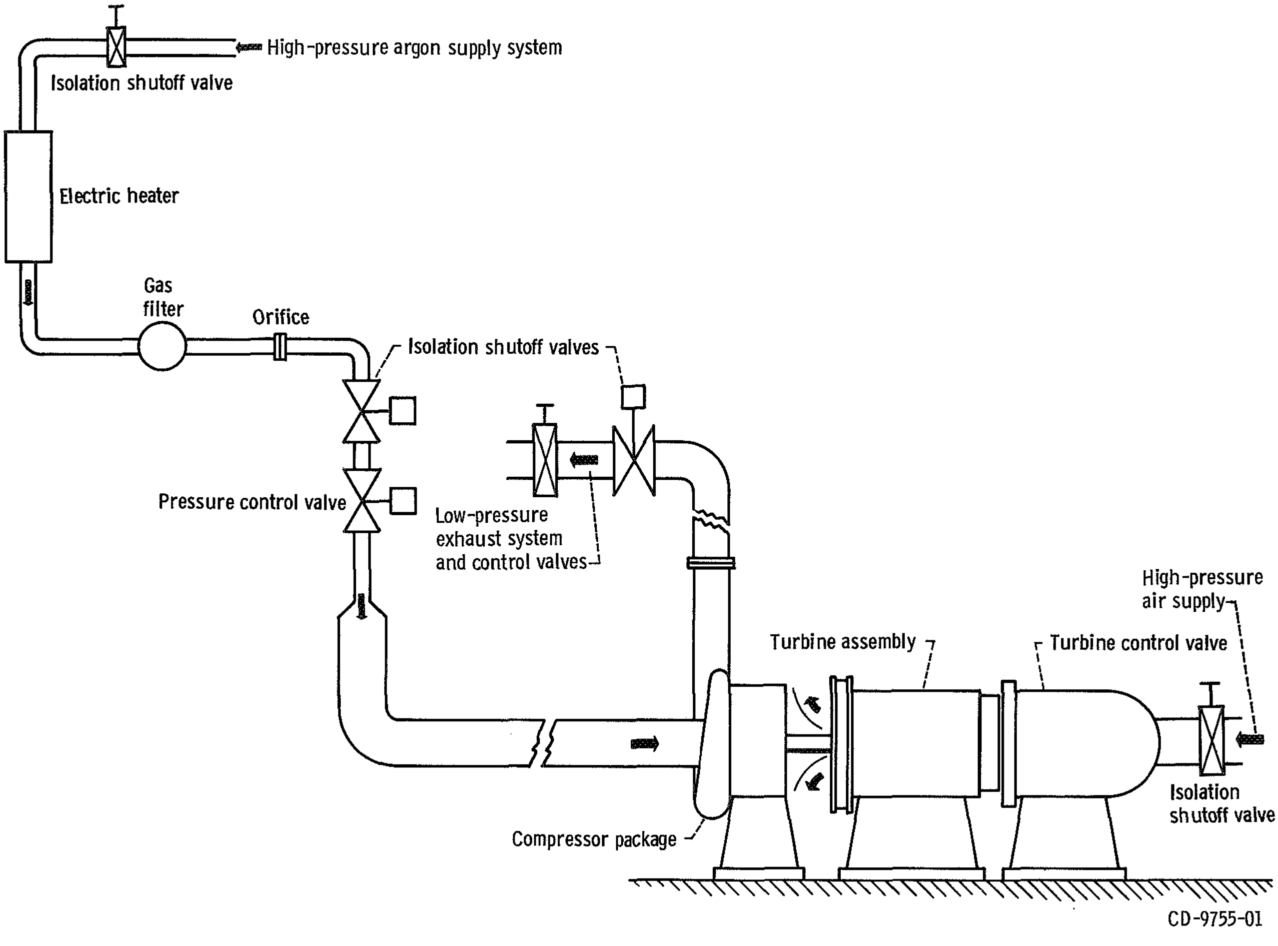

Figure 5. - Flow diagram of compressor test facility.

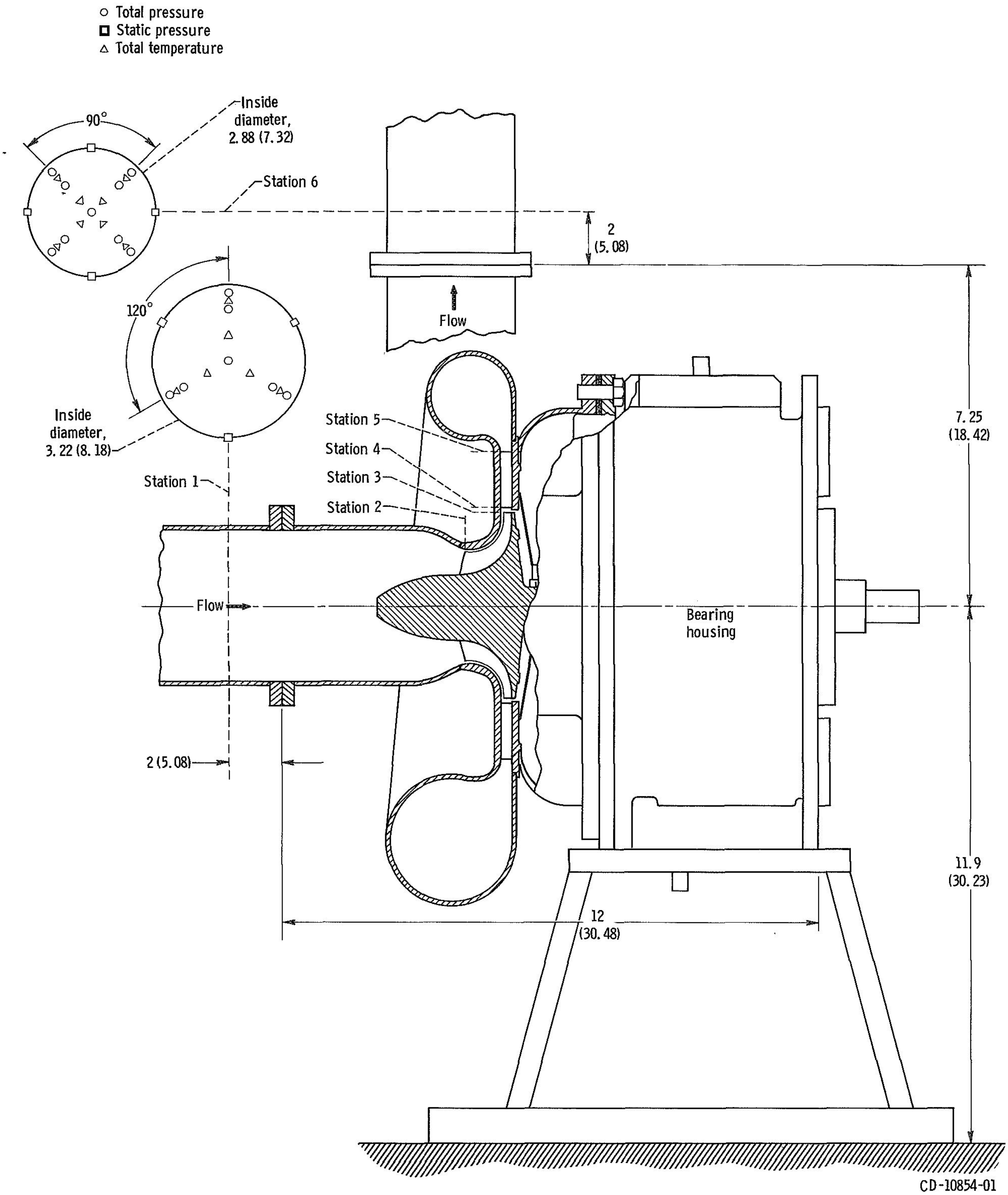

Figure 6. - Compressor cross section, showing instrument locations at stations 1 and 6. (All dimensions are in inches (cm).)

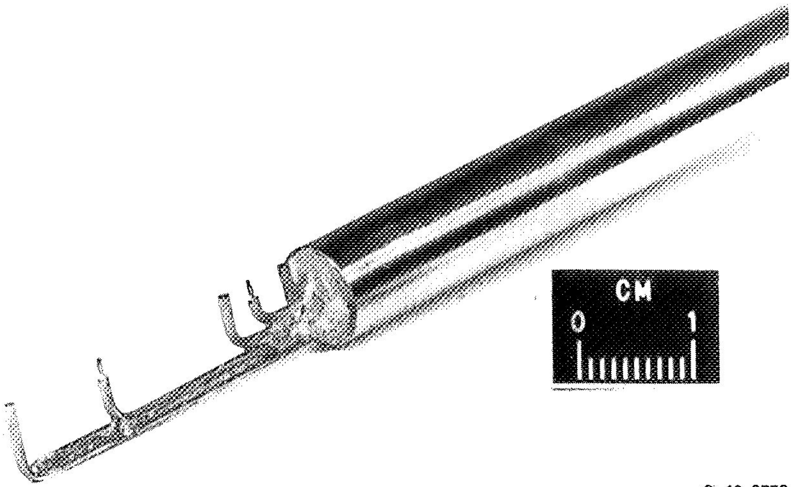

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Figure 7. - Combination total-pressure and total-temperature rake.

One of the combination rakes used for this investigation is shown in figure 7. The total-pressure heads on the rakes were radially located at the centers of three equal annular areas. At station 2, static-pressure taps were located in the shroud over the tip of the rotor leading edge. Five wall-static-pressure taps were equally spaced from station 4 to station 5 in the shroud along the midchannel of the vaned diffuser. At the compressor exit, station 6, the flow conditions were measured with four wall-static-pressure taps equally spaced around the pipe circumference and four combination total-pressure - total-temperature rakes. The exit rakes were the same design as the inlet station 1 combination rakes. The total-pressure heads were made of 0.040-inch (0.102-cm) outside-diameter stainless-steel tube with 0.0065 inch (0.017 cm) wall thickness. The wall-static-pressure taps were 0.030 inch (0.076 cm) in diameter.

All static and total pressures were measured with strain-gage-type pressure transducers. All temperatures were measured using spike-type copper-constantan thermocouples. Flow was determined from a thin-plate orifice installed according to ASME standards in the compressor gas supply line. Compressor speed was measured with the use of a magnetic pickup in conjunction with a gear mounted on the compressor shaft. All performance measurements were recorded by an automatic digital data acquisition system. The recorded data were processed through a high-speed digital computer to obtain the calculated performance parameters.

Test Procedure

Compressor test data were taken over a range of weight flows from maximum flow to compressor surge, for 50, 60, 70, 80, 90, and 100 percent of design equivalent speed. For all tests, the compressor inlet conditions were set and held at nominal design values of total pressure and temperature, 20 psia (13.8 N/cm^2 abs) and 540^0 R (300 K).

RESULTS AND DISCUSSION

The following results were obtained from a 4.25-inch (10.8-cm) sweptback-bladed centrifugal compressor using argon gas as the working fluid:

The compressor overall total-pressure ratio as a function of equivalent weight flow at six speeds is shown in figure 8. At the impeller design equivalent tip speed of

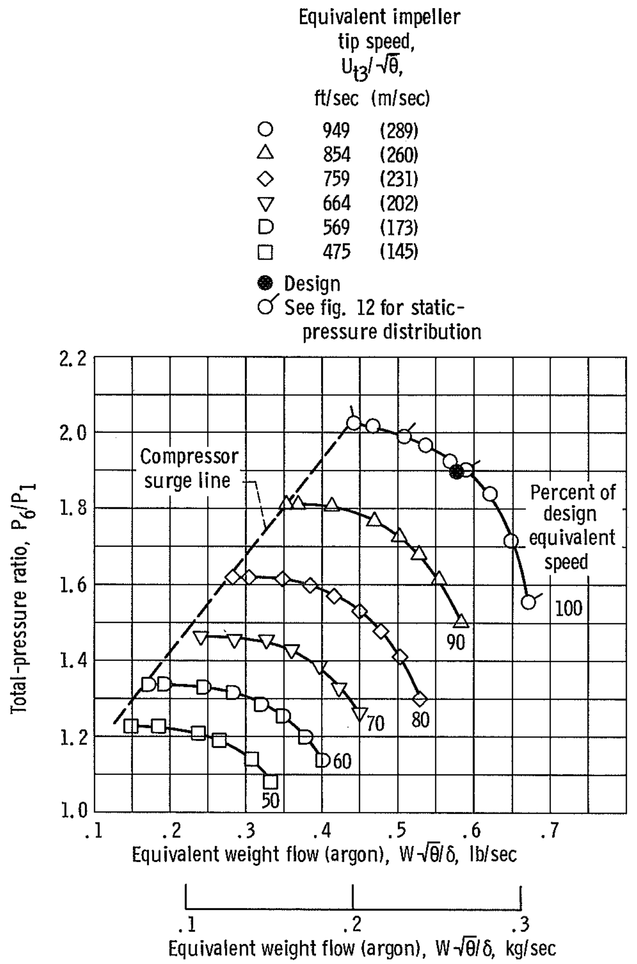

Figure 8. - Overall performance of 4.25-inch (10.8-cm) diameter centrifugal compressor.

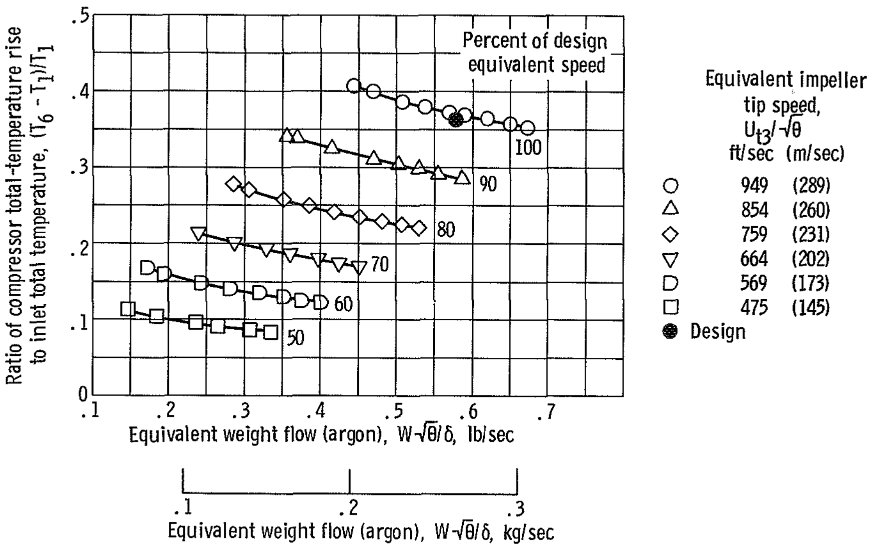

Figure 9. - Ratio of compressor total-temperature rise to inlet total temperature as function of equivalent weight flow at six speeds.

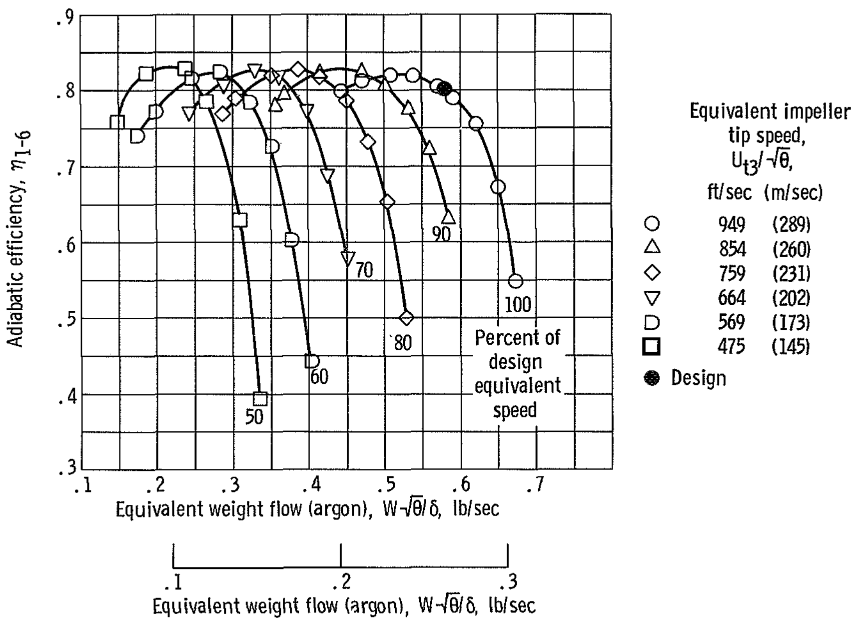

Figure 10. - Efficiency as function of equivalent weight flow.

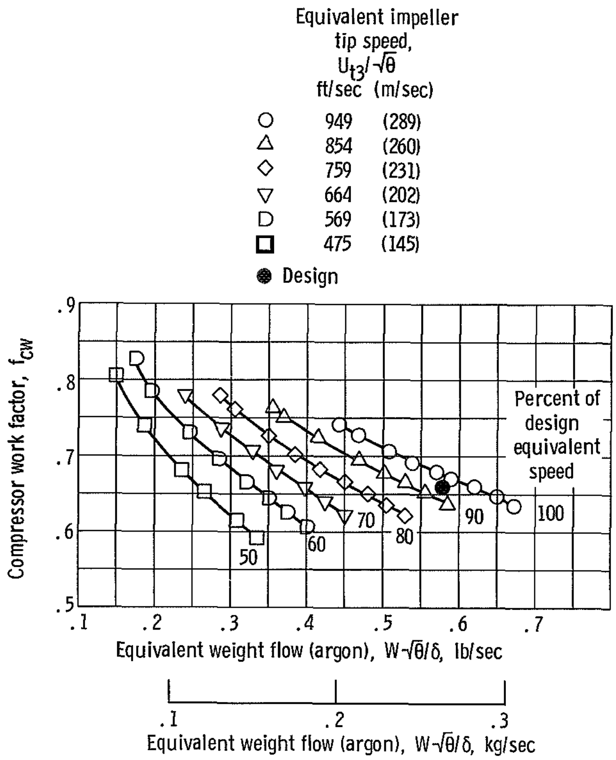

Figure 11. - Compressor work factor as function of equivalent weight flow.

949 feet per second (289 m/sec) and design equivalent weight flow of 0.581 pound per second (0.263 kg/sec), the compressor produced an overall total-pressure ratio of 1.92, which is slightly above the design value of 1.90.

The ratio of compressor total-temperature rise to inlet total temperature as a function of equivalent weight flow is shown in figure 9. For each constant-speed line the total-temperature-rise ratio increases slightly with decreasing flow, which is characteristic of the sweptback-bladed impeller design. At design weight flow and design speed the total-temperature-rise ratio was 0.37, as compared to the design value of 0.365.

The adiabatic efficiency as a function of equivalent weight flow is shown in figure 10. At design conditions the compressor attained an adiabatic efficiency of 0.795, which is in good agreement with the predicted design efficiency of 0.80. At design speed a peak efficiency of 0.819 was attained as the flow was reduced to about 0.508 pound per second (0.230 kg/sec) or 87.5 percent of design weight flow. At 50 percent of the design speed a peak efficiency of 0.83 was attained at a total-pressure ratio of 1.2 and an equivalent weight flow of 0.22 pound per second (0.099 kg/sec).

The compressor work factor, plotted as a function of equivalent weight flow for the six speeds, is shown in figure 11. The work factor for design speed and weight flow is

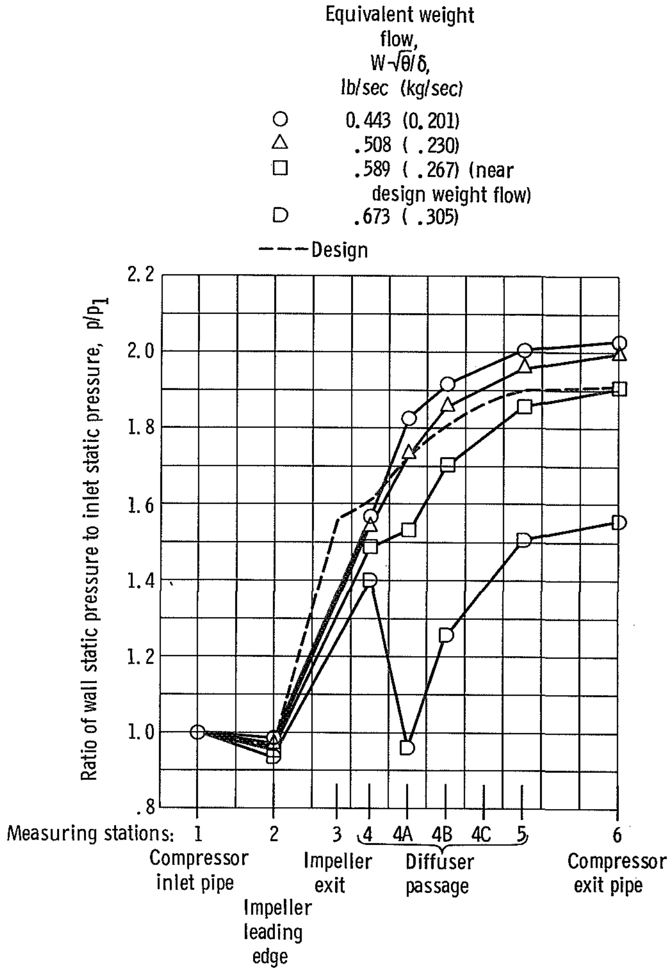

Figure 12. - Static-pressure distribution through compressor at design speed for four flow rates.

0.675 as compared to the design value of 0.658, thus indicating that the energy input to the gas was slightly higher than design.

The static-pressure distribution through the compressor is shown in figure 12. These static pressures were measured at four different equivalent weight flows at design equivalent speed. The corresponding operating points for each static-pressure-distribution curve are shown on figure 8 as the tailed points. The pressure transducer sensing the static pressure at diffuser station 4C failed during testing, so no pressures are shown at this location on figure 12.

The measured static-pressure rise across the impeller and vaneless diffuser (fig. 12, stations 2 to 4) was lower than the predicted design value. This low pressure rise in the impeller indicates that the absolute flow velocity leaving the impeller is greater than design. This may be attributed in part to more boundary-layer blockage than anticipated in the design. This blockage effect was also noted in the analysis of the data in reference 1. Since the energy addition to the gas was only slightly higher than design for design flow (fig. 11), the tangential velocity leaving the impeller and that entering the diffuser vanes is essentially the same as design. With the absolute flow velocity being higher than design and the tangential velocity being essentially the same as design, the flow angle as measured from the radial direction must be lower than design. Therefore, the incidence angle at which the diffuser vanes are operating must be lower than design and operating closer to flow choking than assumed in the design.

At the maximum flow rate of 0.673 pound per secone (0.305 kg/sec), the high throughflow velocity caused the diffuser performance to drop sharply. At this high flow rate the rapid drop in static pressure near the diffuser vane inlet shows that locally high velocities exist within the diffuser passage. This may be attributed to the diffuser vanes operating at much lower than design incidence angle. This trend in static pressure near the diffuser inlet also exists at near design weight flow (0.589 lb/sec, 0.268 kg/sec), further supporting the fact that the diffuser vanes are operating at lower than design incidence and closer to flow choking.

The peak compressor efficiency was attained at a weight flow rate of 0.508 pound per second (0.230 kg/sec), which corresponds to 87.5 percent of design flow. This fact, along with the indication that at design flow the diffuser vanes are operating at lower than design incidence, indicates that improved performance at design flow may be achieved by decreasing the diffuser vane setting angle and thus increasing the incidence angle.

CONCLUDING REMARKS

A 4.25-inch (10.8-cm) diameter centrifugal compressor designed for a nominal 6-kilowatt, single-shaft Brayton-cycle space power system was tested using argon gas. At design speed, design flow rate, and design inlet conditions, the overall total-pressure ratio was 1.92 and the overall adiabatic efficiency was 0.795. These values are in good agreement with the design values of 1.90 and 0.80. At design speed an overall peak adiabatic efficiency of 0.819 was attained at 87.5 percent of design equivalent weight flow. Because of the indication that the vaned diffuser was operating at a lower than design incidence angle at design flow, and because peak efficiency occurred at lower than design flow, an improvement in performance at design weight flow may be obtained by decreasing the diffuser vane setting angle as measured from the radial direction.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, August 10, 1970,
120-27.

APPENDIX - SYMBOLS

c_p	specific heat at constant pressure of argon, 0.125 Btu/(lb)(^{O}R); 524 J/(kg)(K)
D	diameter, ft; m
f_{cw}	compressor work factor, $gJc_p(T_6 - T_1)/U_{t3}^2$
f_s	slip factor, $V_{ut3}/U_{t3} = gJc_p \Delta T_{vd}/U_{t3}^2$
f_w	windage factor, $gJc_p \Delta T_w/U_{t3}^2$
g	gravitational acceleration, 32.17 ft/sec^2; 9.807 m/sec^2
ΔH_{is}	isentropic specific work, (ft)(lb)/lb; (N)(m)/kg
J	mechanical equivalent of heat, 778.16 (ft)(lb)/Btu; 1.00 (m)(N)/J
N_s	specific speed, RPM $\sqrt{Q}/60\ (g\ \Delta H_{is})^{3/4}$
P	total (stagnation) pressure, psia; N/cm^2 abs
p	static pressure, psia; N/cm^2 abs
Q	volume flow, ft^3/sec; m^3/sec
R	gas constant (argon), 38.683 (ft)(lb)/(lb)(^{O}R); 208.13 (m)(N)/(kg)(K)
Re	Reynolds number, $\rho_1 U_{t3} D_{t3}/\mu_1$
RPM	impeller rotational speed, rpm
T	total (stagnation) temperature, ^{O}R; K
ΔT_{vd}	gas temperature rise associated with vector diagrams, ^{O}R; K
ΔT_w	gas temperature rise associated with windage, ^{O}R; K
U	impeller wheel speed, ft/sec; m/sec
V	absolute gas velocity, ft/sec; m/sec
W	weight (mass) flow rate, lb/sec; kg/sec
γ	ratio of specific heat at constant pressure to specific heat at constant volume (for argon, 1.667)
δ	ratio of compressor inlet total pressure to NASA standard sea-level pressure, P_1/14.7 psia; P_1/10.1 N/cm^2 abs
η	adiabatic temperature rise efficiency, $T_1\left[(P_6/P_1)^{(\gamma-1)/\gamma} - 1\right]/(T_6 - T_1)$
θ	ratio of compressor inlet total temperature to NASA standard sea-level temperature, T_1/518.7^{O} R; T_1/288.2 K
μ	dynamic viscosity, lb/(sec)(ft); (N)(sec)/m^2

ρ density, lb/ft^3; kg/m^3

Subscripts:

is isentropic

m meridional component

t tip

u tangential component

1 station in inlet pipe upstream of compressor inlet flange (fig. 6)

2 station at impeller inlet (fig. 6)

3 station at impeller outlet (fig. 6)

4 station at diffuser vane inlet (fig. 6)

5 station at diffuser vane outlet (fig. 6)

6 station in exit pipe downstream of compressor scroll exit flange (fig. 6)

Superscript:

relative to impeller

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