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OVERALL PERFORMANCE IN ARGON OF A 16.4-CENTIMETER (6.44-IN.) SWEPTBACK-BLADED CENTRIFUGAL COMPRESSOR

by Carl Weigel, Calvin L. Ball, and Edward R. Tysl

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16. Abstract

Overall compressor performance is presented as a function equivalent weight flow. At the design equivalent weight flow of 0.69 kg/sec (1.52 lb/sec) and the design equivalent tip speed of 325 m/sec (1066 ft/sec), the compressor produced an overall total pressure ratio of 2.28 and an adiabatic efficiency of 0.80. The peak efficiency at design speed was 0.808 at an equivalent weight flow of 0.65 kg/sec (1.43 lbm/sec).

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SUMMARY

A 16.4-centimeter (6.44-in.) diameter centrifugal compressor, designed for a 10-kilowatt two-shaft closed-loop Brayton cycle space-power system, was tested using argon as the working fluid. Overall performance is presented as a function of equivalent weight flow and compared with the predicted performance.

At the design equivalent weight flow of 0.69 kilogram per second (1.52 lbm/sec) and the design equivalent tip speed of 325 meters per second (1066 ft/sec), the compressor produced an overall total pressure ratio of 2.28 and an adiabatic efficiency of 0.80. At design speed, the compressor produced a peak efficiency of 0.808 and a total pressure ratio of 2.33 at an equivalent weight flor of 0.65 kilogram per second (1.43 lbm/sec). As the speed was decreased to 80 percent of design, the peak efficiency increased to 0.82, while further reduction in speed resulted in a decrease in efficiency.

It can be concluded that improved performance at design flow may be achieved by decreasing the diffuser vane setting angle. This conclusion is based on the facts that the peak efficiency at design speed was attained at a flow rate less than design and that the diffuser vanes appear to be operating at an incidence angle that is lower than design.

At design speed the 16.4-centimeter (6.44-in.) compressor attained a peak efficiency that was about 3 percent higher than that obtained from a previously reported 15.2-centimeter (6-in.) compressor which was designed for the same overall conditions. In comparing the differences in performance, it should be noted that different impeller designs were used, sweptback blades for the one, and radial blades for the other, and that the vaned diffusers were also different designs.

In comparing the efficiency of the 16.4-centimeter (6.44-in.) compressor with that of a scaled 10.8-centimeter (4.25-in.) compressor, no significant scale effect on efficiency was apparent.

INTRODUCTION

The NASA Lewis Research Center is currently engaged in a program to develop small centrifugal and axial flow compressors for application in closed Brayton cycle space electric power generation systems. The program is directed primarily toward establishing the performance level that can be achieved by these small machines.

As part of meeting the objectives of this program, the performance of a 15.2-centimeter (6-in.) radial bladed centrifugal compressor has been investigated (refs. 1 to 3). This compressor was designed for a two-shaft 10-kilowatt Brayton cycle space-power system. A peak adiabatic efficiency of 0.78 was attained at design speed using argon gas as the working fluid.

A 10.8-centimeter (4.25-in.) sweptback bladed centrifugal compressor was designed for a single-shaft Brayton cycle system having a nominal electrical power output of 6 kilowatts (ref. 4). This 10.8-centimeter (4.25-in.) compressor attained a peak adiabatic efficiency of 0.819 at design speed (ref. 5).

In addition to the performance investigation of the 15.2-centimeter (6-in.) radial bladed centrifugal compressor for the two-shaft 10-kilowatt Brayton cycle system, a 16.4-centimeter (6.44-in.) sweptback bladed centrifugal compressor was designed for the same operating conditions. The AiResearch Mfg. Co. of Pheonix, Arizona, designed and fabricated the compressor under NASA contract.

The 16.4-centimeter (6.44-in.) compressor and the 10.8-centimeter (4.25-in.) compressor are identical scale models of a larger compressor. One of the objectives of the 16.4-centimeter (6.44-in.) sweptback-bladed compressor test program, was to compare its performance with that of the 15.2-centimeter (6-in.) radial bladed machine and that of 10.8-centimeter (4.25-in.) scale model.

This report presents the overall performance of the 16.4-centimeter (6.44-in.) sweptback bladed centrifugal compressor obtained at Lewis. The overall performance, using argon gas, is presented for six speeds from 50 to 100 percent of design speed for weight flows from near maximum to surge. The compressor inlet conditions, for all tests, were set at a total pressure of 4.14 newtons per square centimeter (6 psia) and a total temperature of 298 K (536° R). The peak efficiency of this compressor, is compared with that obtained from a 15.2-centimeter (6-in.) and a 10.8-centimeter (4.25-in.) compressor.

SYMBOLS

c_p specific heat at constant pressure of argon, 524 J/(kg)(K); 0.125 Btu/(lbm)($^{\circ}$R)

D diameter, m; ft

f_{cw}	compressor work factor, $gJc_p\left[(T_6 - T_1) + \Delta T_L\right]/U_{t3}^2$
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, 9.807 m/sec^2; 32.17 ft/sec^2
ΔH_{is}	isentropic specific work, (N)(m)/kg; (ft)(lbf)/lbm
J	mechanical equivalent of heat, 1.00 (m)(N)/J; 778.16 (ft)(lb)/Btu
N_s	specific speed, RPM $\sqrt{Q}/60\ (g\ \Delta H_{is})^{3/4}$
P	total (stagnation) pressure N/cm^2; psia
p	static pressure, N/cm^2; psia
Q	volume flow, m^3/sec; ft^3/sec
R	gas constant (argon), 208.13 (m)(N)/(kg)(K); 38.683 (ft)(lbf)/(lbm)(^{o}R)
Re	Reynolds number, $\rho_1 U_{t3} D_{t3}/\mu_1$
RPM	impeller rotational speed, rpm
T	total (stagnation) temperature, K; ^{o}R
ΔT_L	reduction in gas temperature at station 6 resulting from heat loss to lube oil system, K; ^{o}R
ΔT_{vd}	gas temperature rise associated with vector diagrams, K; ^{o}R
ΔT_w	gas temperature rise associated with windage, K; ^{o}R
U	impeller wheel speed, m/sec; ft/sec
V	absolute gas velocity, m/sec; ft/sec
w	weight (mass) flow rate, kg/sec; lbm/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_1/10.1 N/cm^2; P_1/14.7 psia
η_{1-6}	adiabatic temperature rise efficiency, $T_1\left[(P_6/P_1)^{(\gamma-1)/\gamma} - 1\right]/\left[(T_6 - T_1) + \Delta T_L\right]$
θ	ratio of compressor inlet total temperature to NASA standard sea-level temperature, T_1/288.2 K; T_1/518.7^o R
μ	dynamic viscosity, (N)(sec)/m^2; lbm/(sec)(ft)
ρ	density, kg/m^3; lbm/ft^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

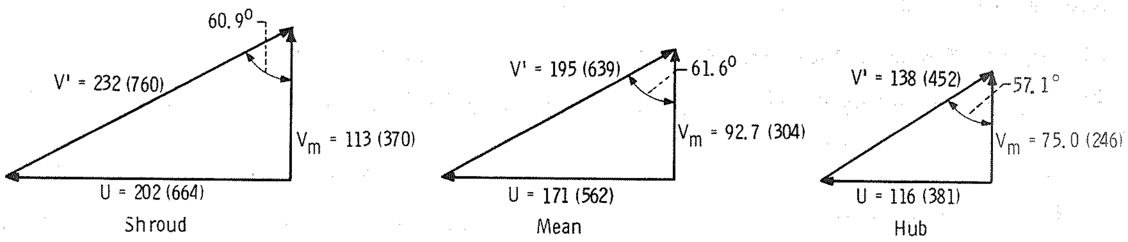
COMPRESSOR DESIGN

The compressor was scaled from an existing design for application in a 10-kilowatt Brayton cycle space power system using argon as the working fluid. A summary of the design point values for the reference design power level of 10 kilowatts is presented in the following table:

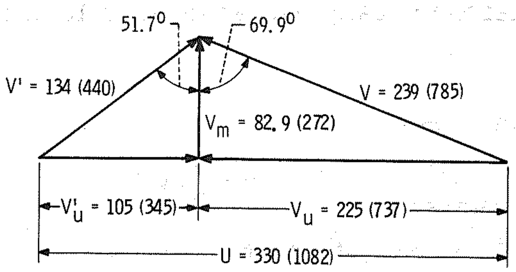
Inlet total pressure, P_1, N/cm^2; psia	4.14; 6.00
Inlet total temperature, T_1, K; ^{o}R	298; 536
Weight flow rate, w, kg/sec; lbm/sec	0.277; 0.611
Equivalent weight flow, $w\sqrt{\theta}/\delta$, kg/sec; lbm/sec	0.69: 1.52
Compressor total pressure ratio, P_6/P_1	2.30
Compressor total temperature ratio, T_6/T_1	1.48
Compressor efficiency, η_{1-6}	0.821
Impeller total pressure ratio, P_3/P_1	2.47
Impeller efficiency, η_{1-3}	0.897
Rotative speed, RPM, rpm	38 500
Equivalent speed, RPM/$\sqrt{\theta}$, rpm	37 900
Impeller tip speed, U_{t3}, m/sec; ft/sec	330; 1082
Equivalent impeller tip speed, $U_{t3}/\sqrt{\theta}$, m/sec; ft/sec	325; 1066
Specific speed, N_s	0.1057
Compressor work factor, f_{cw}	0.689
Reynolds number, Re	1.58×10^6

The compressor impeller has 15 blades that are curved backward (swept back) at the exit 30^o from the radial. The impeller inlet tip diameter is 10 centimeters (3.95 in.) and the inlet hub diameter is 5.56 centimeters (2.19 in.). The impeller exit diameter is 16.4 centimeters (6.44 in.) and the exit blade height is 0.79 centimeter (0.31 in.). The design velocity diagrams for the impeller inlet and outlet are shown in figure 1.

From the impeller exit to the diffuser vane inlet there is a vaneless diffuser section of about 0.33 centimeter (0.13 in.) long in the radial direction. There are 17 vanes in the vaned diffuser section. The diffuser vanes have a constant height of 0.820 centimeter (0.323 in.). The diffuser design velocity diagrams are shown in figure 2.

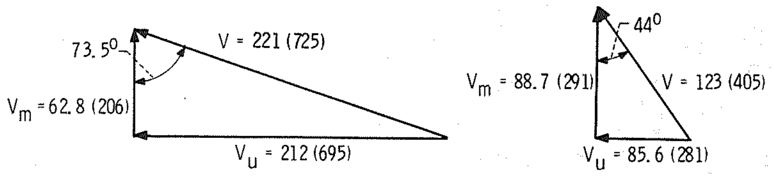


(a) Impeller inlet (station 2)



(b) Impeller exit mean (station 3)

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



(a) Diffuser inlet (station 4)

(b) Diffuser exit (station 5)

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

APPARATUS AND PROCEDURE

Test Apparatus

The impeller used in this investigation was made of titanium alloy and is shown in figure 3 with the vaned diffuser. The vaned diffuser was made of stainless steel. The compressor scroll assembly is shown in figure 4 and was also made of stainless steel to minimize the heat transfer by conduction from the scroll back along the shroud toward the impeller inlet. A phenolic heat barrier was used between the bearing housing and the diffuser section to help reduce the heat transfer from the gas to the bearing lubricating oil.

The impeller was cantilever mounted on a shaft supported by two angular contact bearings. Carbon face oil seals were mounted outboard of each bearing. A carbon shaft seal was also used between the compressor impeller and the carbon face oil seal. The static (cold) impeller clearance between the shroud and the blade tip at the impeller exit was set at 0.028 centimeter (0.011 in.). The compressor was enclosed in insulation as shown in figure 5 to minimize heat transfer out of the compressor between the inlet and outlet measuring stations.

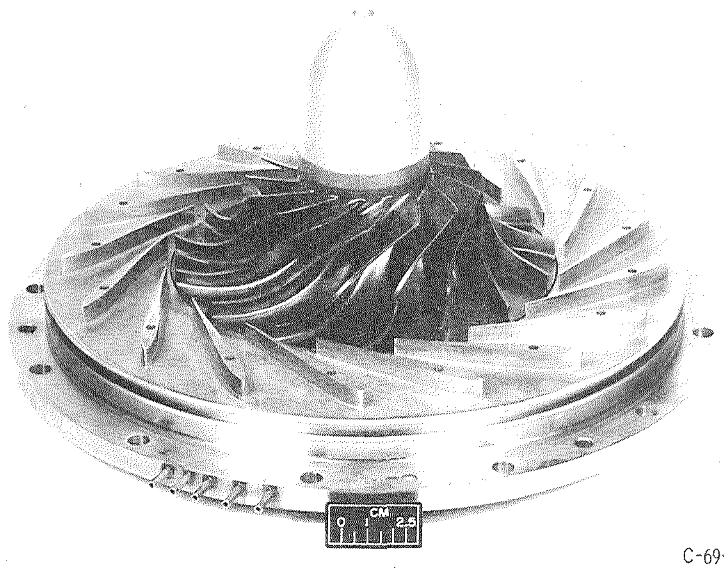


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Figure 3. - Centrifugal compressor impeller and vaned diffuser.

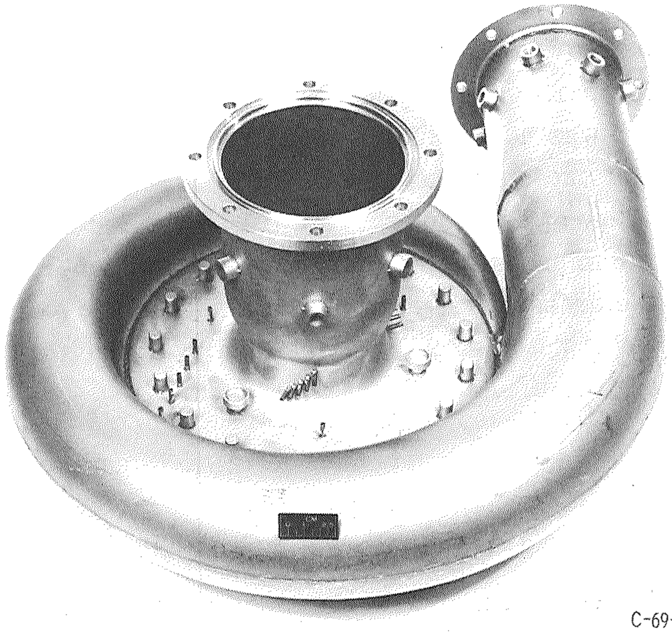

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Figure 4. - Compressor scroll.

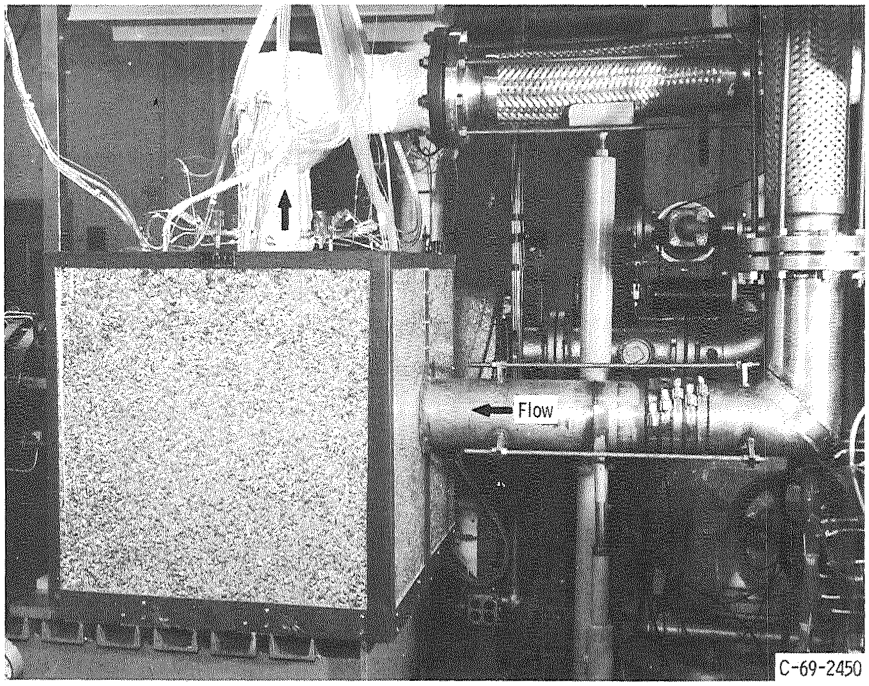

Figure 5. - Compressor fully enclosed in insulation.

Test Facility

An open-loop compressor test facility was used for this investigation. All tests were conducted using high purity argon gas (contaminants less than 50 ppm by volume). The inlet gas temperature was held constant by an automatically controlled 25-kilowatt electric heater. The inlet pressure was controlled by a remotely operated pressure control valve. The gas flow and exit pressure were controlled by a remotely operated valve downstream of the compressor. The argon gas was discharged into a laboratory exhaust system. The compressor was driven by a single-stage axial-flow air turbine. The drive turbine speed was automatically maintained by a throttle valve that regulated the air flow to the drive turbine.

Instrumentation

A cutaway view of the compressor with the locations of instrumentation and calculation stations is shown in figure 6. At the inlet (station 1) and exit (station 6), total pressure and total temperature were measured using combination total-pressure - total-temperature rakes. A typical rake is shown in figure 7. At stations 1 and 6, wall static pressure taps were equally spaced around the pipe as shown in figure 6. At station 2, static pressure taps were located in the shroud over the tip of the rotor leading edge. A wall static tap was located between stations 3 and 4 in the vaneless diffuser. Five wall static pressure taps were equally spaced from station 4 to 5 in the shroud along the midchannel of the vaned diffuser.

All static and total pressures were measured with strain-gage type of pressure transducers. All gas temperatures were measured using spike type of copper-constantan thermocouples. Flow was determined with a thin-plate orifice installed in the compressor gas supply line. Compressor speed was measured with a magnetic pick-up 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 near maximum to compressor surge for 50, 60, 70, 80, 90, and 100 percent of design equivalent speed.

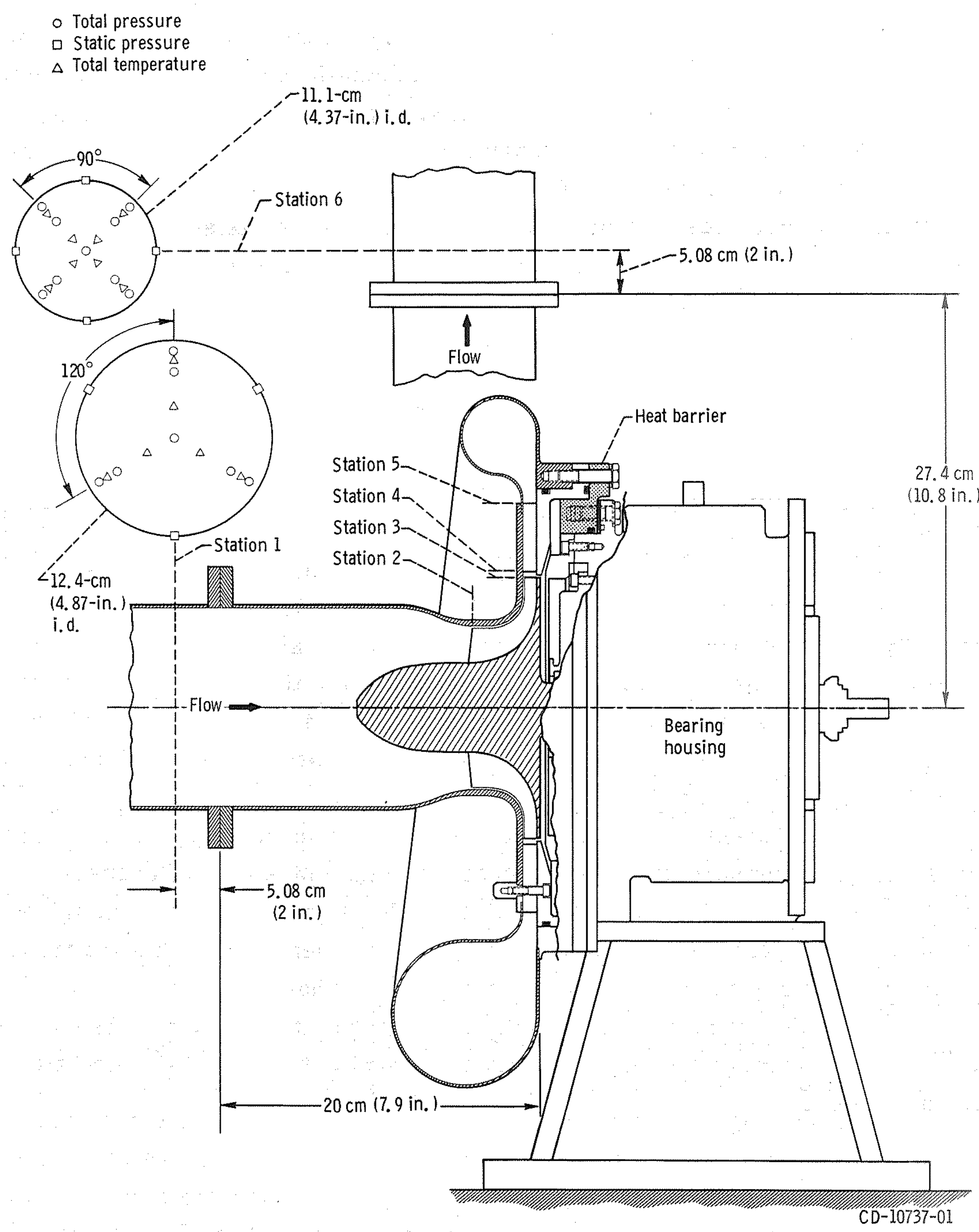

Figure 6. - Compressor cross section, showing instrument locations at stations 1 and 6.

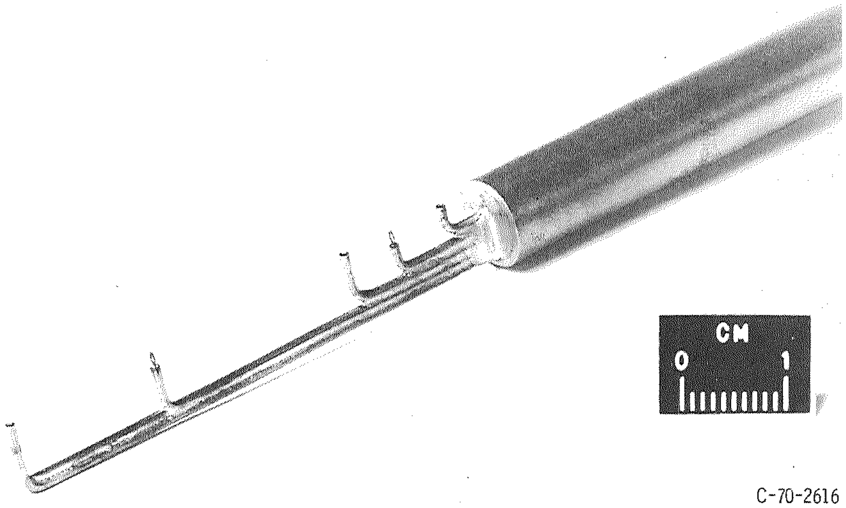

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Figure 7. - Combination total pressure and total temperature rake.

The compressor inlet conditions were held at nominal design values of total pressure and total temperature of 4.1 newtons per square meter (6 psia) and 298 K (536° R) for all tests.

CALCULATION PROCEDURE

The overall compressor performance was computed using the equations defined in the symbols list. The pressures and temperatures were measured at stations 1 and 6. However, analysis of the data indicated that the energy addition to the gas, based on the measured temperatures, was lower than anticipated. This was attributed to heat conduction affecting the exit-temperature measurements and heat transfer out of the system between the measuring stations. In calculating overall performance, the following procedure was used to adjust the measured temperature rise.

The six inlet thermocouple readings were in good agreement throughout the testing. However, the exit thermocouple readings (station 6, fig. 6) showed a radial temperature difference between the ring of thermocouples near the center of the pipe and those near the pipe wall. For each of the six speeds, the difference in temperature between the center ring of thermocouples and those near the exit pipe wall increased as the flow was decreased. The lower temperatures near the pipe wall were believed to be due to heat conduction down the rake (ref. 6). The outer ring of exit thermocouples was immersed in the gas stream only 1.02 centimeters (0.4 in.) from the wall. At design conditions the indicated temperature of these thermocouples was about 6 degrees lower than the indicated temperature of the inner ring of thermocouples. Calculations showed that this difference between the temperature readings could be attributed to heat conduction affecting the readings of the outer ring of thermocouples. The calculations indicated that the inner ring of thermocouples had a negligible conduction error because of the greater distance (3.94 cm or 1.55 in.) that they were immersed into the gas stream and the smaller cross-sectional area of the support for these couples (fig. 7). As a result, the exit gas temperature T_6 used in this report is an average of the center ring of thermocouple readings at station 6.

As discussed in reference 3, potential heat-leak paths exist and two of these are heat transfer from the impeller outlet to the impeller inlet through the stationary shroud and heat loss to the compressor oil supply. To minimize the heat transfer from the compressor discharge back to the compressor inlet, the stationary shroud and scroll assembly was made of stainless steel which has low conductivity. (An aluminum shroud and scroll assembly was used for the 15.2-cm (6-in.) radial blade compressor, reported in ref. 1.) To minimize the heat transfer from the compressor scroll to the bearing housing, a phenolic heat barrier was used (fig. 6).

Although considerable effort was used to minimize the heat loss from the compressor, measurements of the oil temperature rise indicated a heat loss through the wall behind the rotor to the oil. To determine this heat loss, as discussed in reference 3, the oil flow temperature rise due to bearing and seal friction was measured over the speed range, with the compressor evacuated to eliminate aerodynamic heating. The oil temperature rise due to friction was then substracted from the oil temperature rise measured during performance testing to determine the heat loss from the compressor gas to the oil. This oil-temperature-rise difference was added to the measured gas temperature (center ring thermocouple readings) at the compressor exit to obtain a corrected exit temperature. The performance data presented in this report were based on this corrected exit temperature. The magnitude of the heat loss effects on compressor efficiency are shown in the appendix.

RESULTS AND DISCUSSION

16.4-Centimeter (6.44-In.) Centrifugal Compressor Performance

The following results were obtained from a 16.4-centimeter (6.44-in.) sweptback-bladed centrifugal compressor using argon as the working fluid.

The compressor overall total pressure ratio as a function of equivalent weight flow for six speeds is shown in figure 8. At the impeller design equivalent tip speed of 325 meters per second (1066 ft/sec) and design equivalent weight flow of 0.69 kilogram per second (1.52 lbm/sec), the compressor produced an overall total pressure ratio of 2.28. The design total pressure ratio is 2.30. A peak total pressure ratio of 2.39 was

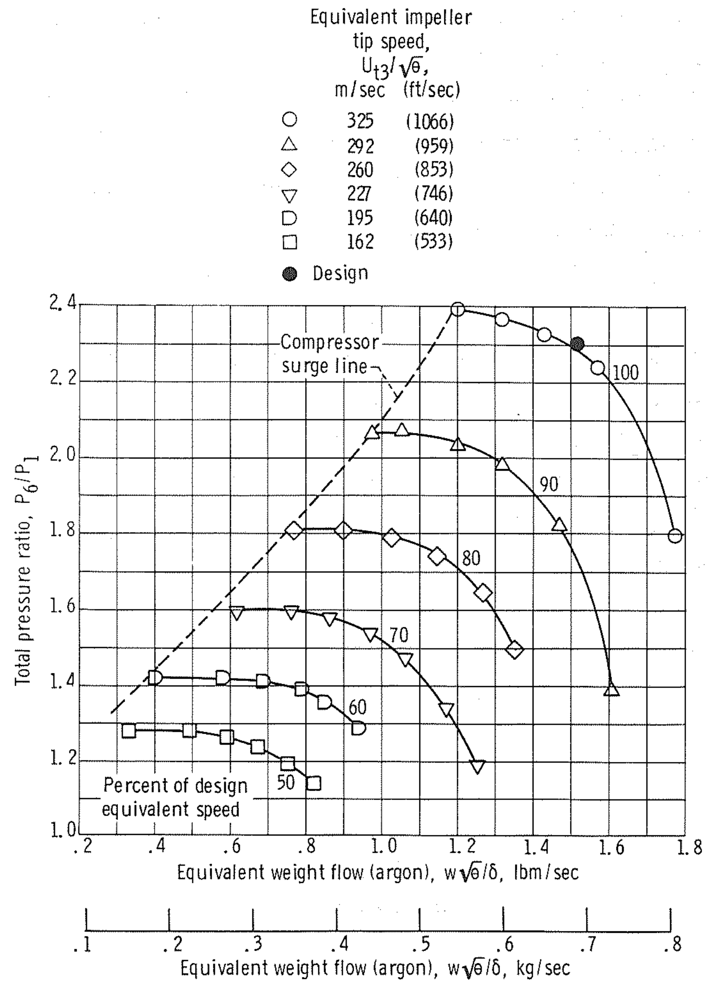

Figure 8. - Overall performance of 16.4 centimeter (6.44 in.) diameter centrifugal compressor.

attained at design speed near compressor surge.

The ratio of compressor total temperature rise to inlet total temperature as a function of equivalent weight flow is shown in figure 9. As mentioned in the calculation procedure, this temperature rise is corrected to account for the heat loss from the compressor gas to the lube oil. For each constant speed line, the total temperature rise ratio increased with decreasing flow. At the design weight flow and design speed, the total temperature rise ratio was 0.485, as compared with a design value of 0.481.

The adiabatic efficiency as a function of equivalent weight flow is shown in figure 10. At design conditions the compressor attained an efficiency of 0.80, as compared with the predicted design efficiency of 0.82. At design speed a peak efficiency of 0.808 was attained at an equivalent weight flow of 0.65 kilograms per second (1.43 lb/sec). As the speed was decreased to 80 percent of design, the peak efficiency increased to 0.82. As the speed was further reduced the peak efficiency decreased.

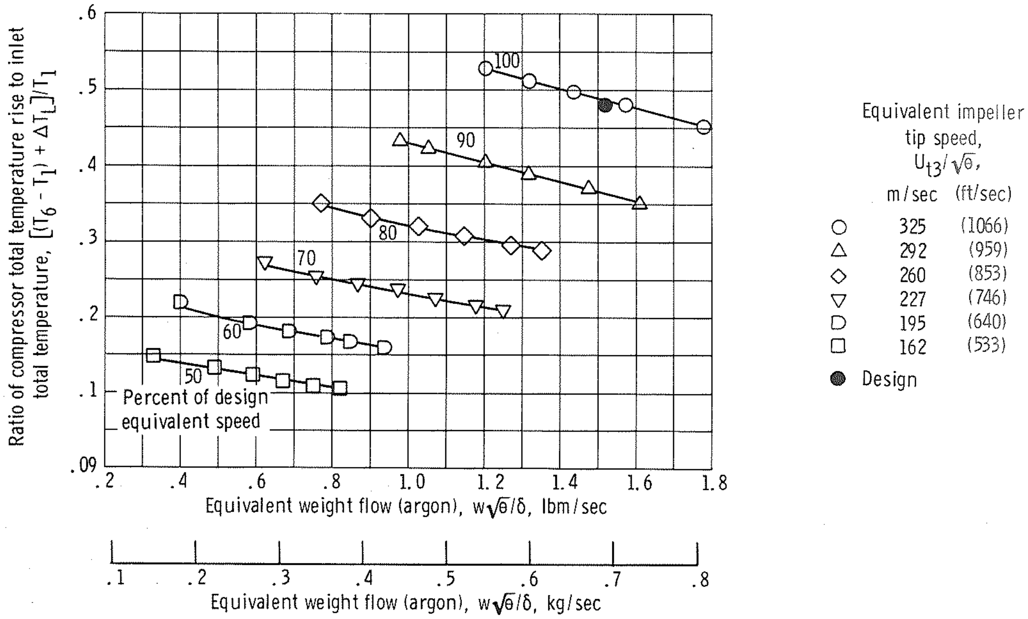

Figure 9. - Ratio of total temperature rise to inlet total temperature as a function of equivalent weight flow at six compressor speeds.

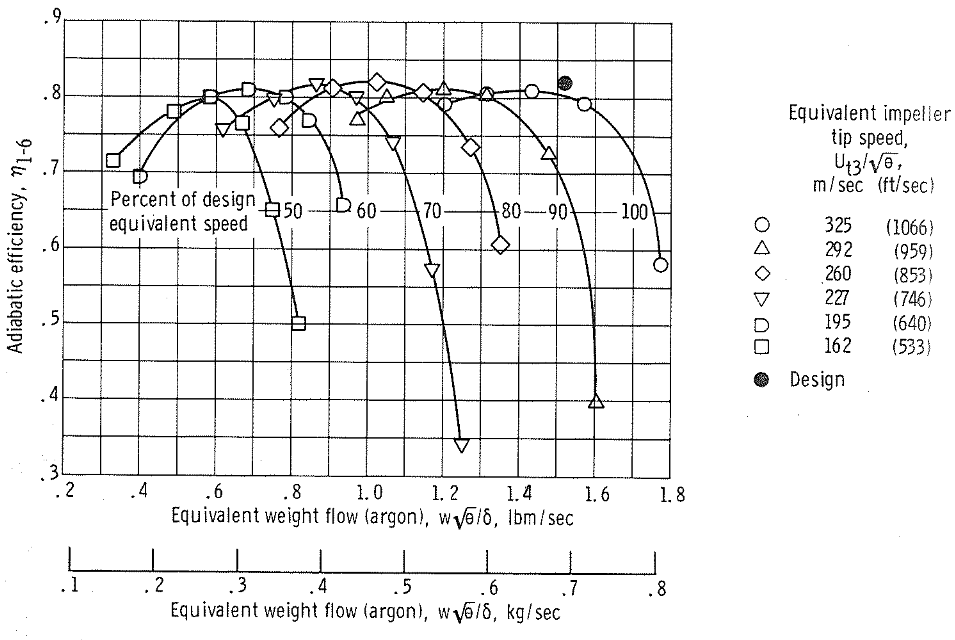

Figure 10. - Adiabatic efficiency as function of equivalent weight flow.

The compressor work factor as a function of equivalent weight flow for the six speeds is shown in figure 11. The work factor at design speed and flow was 0.685 as compared with the design value of 0.689.

The static pressure distribution through the compressor is shown in figure 12. These static pressures were measured at the five different flow rates at design equivalent speed (fig. 8).

The measured static pressure rise across the impeller and vaneless diffuser (station 2 to 4) was lower than the predicted design value. This indicates that the absolute velocity leaving the impeller is greater than design. The energy addition to the gas was slightly lower than design at design flow (fig. 11), so the tangential velocity leaving the impeller and that entering the diffuser may be assumed to be slightly lower than design. With the absolute flow velocity being greater than design and the tangential velocity slightly less than design, the flow angle as measured from the radial direction must be less 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. This was also noted in reference 2 for the performance of the 10.8-centimeter (4.25-in.) compressor.

At the maximum flow rate of 0.807 kilogram per second (1.78 lb/sec) the high through-flow velocity caused the diffuser performance to drop sharply. This may be

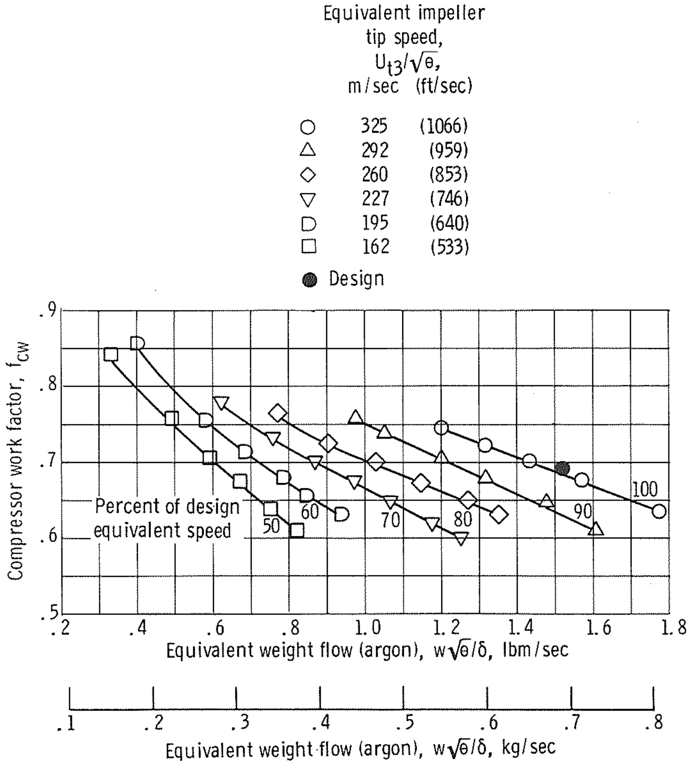

Figure 11. - Compressor work factor as function of equivalent weight flow.

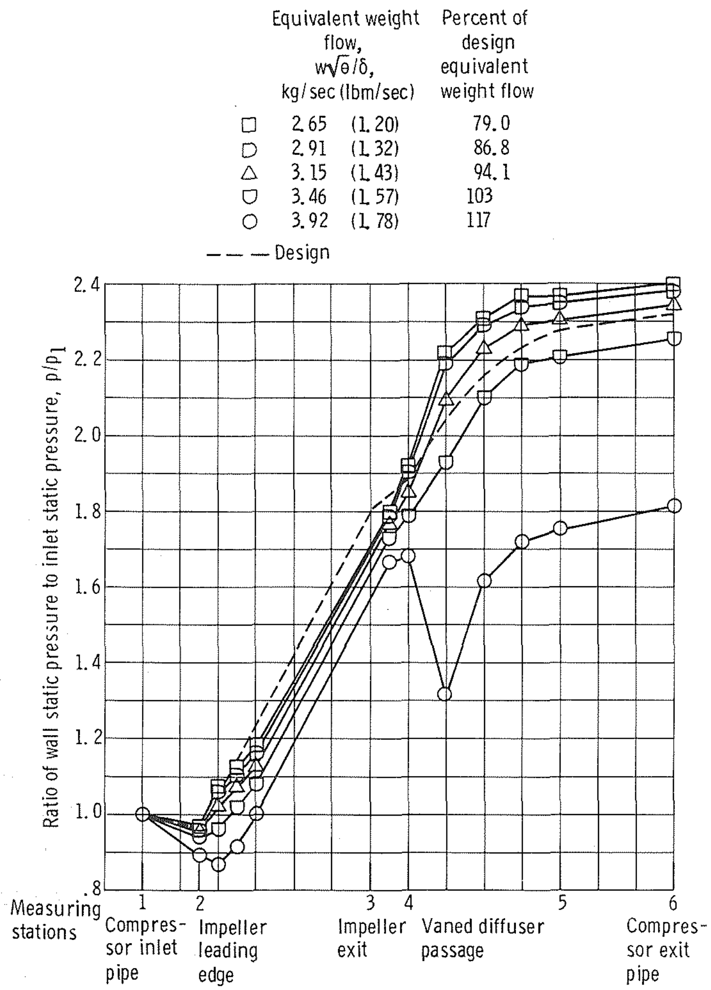

Figure 12. - Compressor static pressure distribution at design speed for five flow rates.

attributed to the diffuser vanes operating at much lower than design incidence angle.

The peak compressor efficiency for design speed was attained at a flow rate of 0.65 kilogram per second (1.43 lbm/sec) or 94 percent of design equivalent weight flow. Since the peak efficiency was attained at a flow rate less than design and it appears that the diffuser vanes are operating at lower than design incidence, improved performance at design flow may be achieved by decreasing the diffuser vane setting angle.

Comparison of the 16.4-Centimeter (6.44-In.) and the 15.2-Centimeter (6.0-In.) Compressors

The peak efficiency of the 16.4 centimeter (6.44-in.) compressor, at design speed, was about 3 percent higher than that attained for the 15.2-centimeter (6-in.) compressor, which was designed for the same overall parameters (ref. 1). When comparing the performances of these two compressors, it should be noted that the impeller designs are different (sweptback blades for one, radial blades for the other) and that the vaned diffuser designs are also different. The 15.2-centimeter (6-in.) compressor used an airfoil type of three-dimensional diffuser vane, and the 16.4-centimeter (6.44-in.) compressor used a two-dimensional (parallel walls) vane-island type of diffuser blading. The increase in performance of the 16.4 centimeter (6.44 in.) compressor over that of the 15.2-centimeter (6-in.) compressor is probably a result of the combination of the differences in impeller and diffuser designs.

Comparison of the 16.4-Centimeter (6.44-In.) and the 10.8-Centimeter (4.25-In.) Diameter Compressors

A comparison of the peak efficiency of the 16.4-centimeter (6.44-in.) compressor at 90 percent of design with that of the 10.8-centimeter (4.25-in.) compressor (ref. 5) at design speed, showed that the efficiency of the larger machine was slightly lower by about 1 percent. The 90-percent speed performance of the 16.4-centimeter (6.44-in.) machine was used for comparison with the design speed performance of the 10.8-centimeter (4.25-in.) machine so the pressure ratio of the two would be essentially the same. The lower efficiency of the larger machine may, in part, be the result of the correction for heat loss to the lubricating oil. The data presented for the 10.8-centimeter (4.25-in.) compressor did not account for any heat loss from the gas to the air-oil mist type of lubricating system. In most cases it is difficult to precisely account for the heat loss from these small compressors. The lower Reynolds number at which the performance of the larger compressor was taken may have also contributed to the slightly lower efficiency. The 16.4-centimeter (6.44-in.) compressor has a design Reynolds number of 1.58×10^6, compared with a design Reynolds number of 3.1×10^6 for the 10.8-centimeter (4.25-in.) compressor. In general it can be concluded that no significant scale effect on compressor efficiency is evident for the two compressors.

SUMMARY OF RESULTS

Overall performance of a 16.4-centimeter (6.44-in.) diameter centrifugal compressor with sweptback blades was obtained using argon gas. The compressor was designed for application in a 10-kilowatt two-shaft Brayton cycle space-power system and the following is a summary of the results:

1. At design equivalent weight flow of 0.69 kilogram per second (1.52 lbm/sec) and design equivalent tip speed of 325 meters per second (1066 ft/sec) the compressor produced an overall total pressure ratio of 2.28 and an adiabatic efficiency of 0.80.

2. The peak efficiency at design speed was 0.808 with a total pressure ratio of 2.33 at an equivalent weight flow of 0.65 kilogram per second (1.43 lbm/sec).

3. At 50 percent of design equivalent speed the compressor attained a peak efficiency of 0.798 with a total pressure ratio of 1.26 at an equivalent weight flow of 0.27 kilogram per second (0.59 lbm/sec).

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APPENDIX - HEAT LOSS EFFECTS ON COMPRESSOR EFFICIENCY

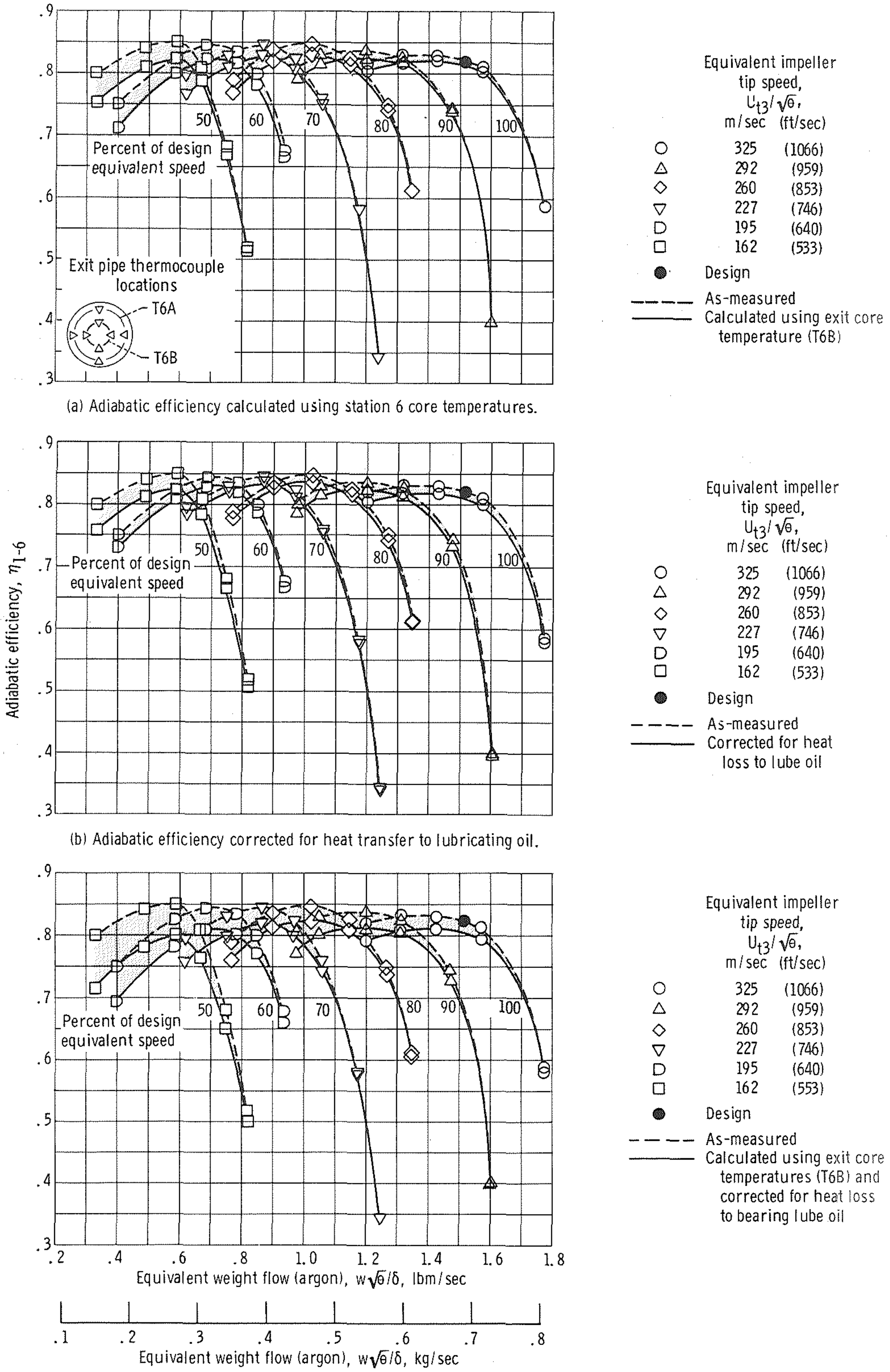
As discussed in the calculations procedure section of this report, the adiabatic efficiency η_{1-6} was computed to account for the heat loss from the compressor gas to the lubricating oil. And as was further explained, a conduction error caused the ring of exit thermocouples near the pipe wall to indicate a lower temperature than the ring of thermocouples near the center of the exit pipe. The effect that these problems had on the as-measured adiabatic efficiency is shown in this appendix.

The adiabatic efficiency as a function of equivalent weight flow for the 16.4-centimeter (6.44-in.) compressor is shown in figure 13. The inlet total pressure was 4.14 newtons per square centimeter (6 psia), and the inlet total temperature was 298 K (536° R).

Figure 13(a) shows the as-measured efficiency computed from an average of the temperatures of both the inner ring T_{6A} and outer ring T_{6B} of thermocouples at the compressor exit, compared with the efficiency that was computed using an average of the exit core thermocouples T_{6B} readings. This comparison shows the effect that the heat transfer along the support tubes of the rakes (fig. 7) had on the readings of the exit thermocouples near the pipe wall. The as-measured peak efficiency at design speed was 1 percent greater than the corrected efficiency, and this difference increased to 2.5 percent at 50 percent speed.

The decrease in efficiency that resulted from the correction made for the heat loss to the lubricating oil system is shown in figure 13(b). The peak as-measured efficiency here, too, was decreased 1 percent at design speed and 2.5 percent at 50 percent speed.

The combined effect of heat conduction on the thermocouple readings and heat loss to the oil as it affected the as-measured efficiency is shown in figure 13(c). This is the corrected efficiency used in the body of this report. At design speed, the peak corrected efficiency is 0.808 compared with the peak as-measured efficiency of 0.830. At 50 percent speed the peak corrected efficiency is 0.823, and the measured efficiency is 0.849, a decrease that reflects the more adverse conditions of lower pressures and velocities at the exit.



(a) Adiabatic efficiency calculated using station 6 core temperatures.

(b) Adiabatic efficiency corrected for heat transfer to lubricating oil.

(c) Adiabatic efficiency calculated using station 6 core temperature and corrected for heat loss to lubricating oil.

Figure 13. - Comparison of corrected values of adiabatic efficiency with as-measured values.

REFERENCES

1. Tysl, Edward R.; Ball, Calvin L.; Weigel, Carl; and Heidelberg, Laurence J.: Overall Performance in Argon of a 6-inch Radial-Bladed Centrifugal Compressor. NASA TM X-1622, 1968.

2. Ball, Calvin L.; Weigel, Carl, Jr.; and Tysl, Edward R.: Overall Performance of a 6-inch Radial Bladed Centrifugal Compressor with Various Diffuser Vane Setting Angles. NASA TM X-2107, 1970.

3. Heiderlberg, Laurence J.; Ball, Calvin L.; and Weigel, Carl: Effect of Reynolds Number on Overall Performance of a 6-inch Radial Bladed Centrifugal Compressor. NASA TN D-5761, 1970.

4. Anon.: Design and Fabrication of the Brayton Cycle High Performance Compressor Research Package. Rep. APS-5269-R, AiResearch Mfg. Co. (NASA-CR 72533), Nov. 29, 1967.

5. Weigel, Carl, Jr.; Tysl, Edward R.; and Ball, Calvin L.: Overall Performance in Argon of 4.25-inch Sweptback-Bladed Centrifugal Compressor. NASA TM X-2129, 1970.

6. Futral, S. M.; Kofskey, M.; and Rohlik, H.: Instrumentation Used to Define Performance of Small Size, Low Power Gas Turbines. NASA TM X-52522, 1969.

NASA-Langley, 1971 —— 1 E-6100

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