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Compressor Study to Meet Large Civil Tilt Rotor Engine Requirements

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Summary

A vehicle concept study has been made to meet the requirements of the Large Civil Tilt Rotorcraft vehicle mission. A vehicle concept was determined, and a notional turboshaft engine system study was conducted. The engine study defined requirements for the major engine components, including the compressor. The compressor design-point goal was to deliver a pressure ratio of 31:1 at an inlet weight flow of 28.4 lbm/sec. To perform a conceptual design of two potential compressor configurations to meet the design requirement, a mean-line compressor flow analysis and design code were used. The first configuration is an eight-stage axial compressor. Some challenges of the all-axial compressor are the small blade spans of the rear-block stages being 0.28 in., resulting in the last-stage blade tip clearance-to-span ratio of 2.4 percent. The second configuration is a seven-stage axial compressor, with a centrifugal stage having a 0.28-in. impeller-exit blade span. The compressors' conceptual designs helped estimate the flow path dimensions, rotor leading and trailing edge blade angles, flow conditions, and velocity triangles for each stage.

Introduction

A notional study of the Large Civil Tilt Rotorcraft (LCTR) vehicle mission has been done in Reference 1. To meet the LCTR vehicle thrust requirements at the takeoff and cruise conditions, a thermodynamic cycle study of a notional turboshaft engine was performed in Reference 2 with the Numerical Propulsion System Simulator (NPSS) thermodynamic cycle code. The results of the engine system model are illustrated by the schematic diagram in Figure 1. Utilizing the NPSS system model, the compressor flow and pressure ratio requirements to produce a pressure ratio of 31:1 at a corrected airflow rate of 28.4 lbm/sec were determined.

The focus of this study is to perform a conceptual sizing study of the compressor to meet the pressure ratio and flow requirements of the turbine engine for the LCTR vehicle. The conceptual design was done with a mean-line flow methodology and focuses on the compressor flow path and key aerodynamic parameters of the rotor and stage at the design point condition. Off-design performance is also estimated for the 100 percent speed line with the mean-line methodology. The purpose of the conceptual design is to have an initial estimate of the overall compressor that could meet the requirements, and to identify early on any potential technical barriers that will need to be considered during the preliminary and detailed design phases.

Compressor Conceptual Design

The purpose of the compressor conceptual design was to identify potential technical challenges towards meeting the goals. There are several options that can be considered for the shape of the flow path during the conceptual design process of multistage compressors. The constant-tip compressor flow path provides the most work capability in the fewest number of axial stages, but can result in excessively small blade spans in the rear stages. A tapered-tip axial-compressor flow path can generally produce less work per stage because of the reduction of rotor-tip diameter for each subsequent stage, and can result in blades with larger spans in the last stages than a constant-tip diameter flow path. A combination of a multistage axial compressor with a centrifugal compressor in the last stage can provide a more axially compact configuration in comparison to an all-axial compressor, but could have a larger outer diameter due to the impeller and radial diffuser. These conceptual design options for the 31:1 compressor were explored in this study with the use of a mean-line compressor design and analysis code. The mean-line compressor flow analysis and design code from Reference 3 was used in this study. The axial rotor-blade tip speed was limited to no higher than 1500 ft/sec because of anticipated structural limitations, based on historical compressor experience (Refs. 4 and 5), as well as concerns about having excessively high rotor-inlet tip relative Mach number. The diffusion factor as defined by Equation (1) was limited at the design point to be on the order of 0.53.

$$DF = 1 - \frac{W_2}{W_1} + \frac{(R_1\, C_{U1} - R_2 C_{U2})}{(R_1 + R_2) W_1\, \sigma} \qquad (1)$$

The diffusion factor is determined from flow and geometric quantities at the rotor leading edge (subscript 1) and trailing edge (subscript 2). The *R* refers to root-mean-square radius, while *C* refers to absolute velocity, and *W* is the relative velocity. The symbol σ is the blade solidity.

As this mean-line code does not generate blade shapes, but only estimates leading and trailing blade angles, the values for the number of blades and blade solidity at the rotor tip were obtained from the energy-efficient engine high-pressure compressor of Reference 6. Each rotor was sized to have an

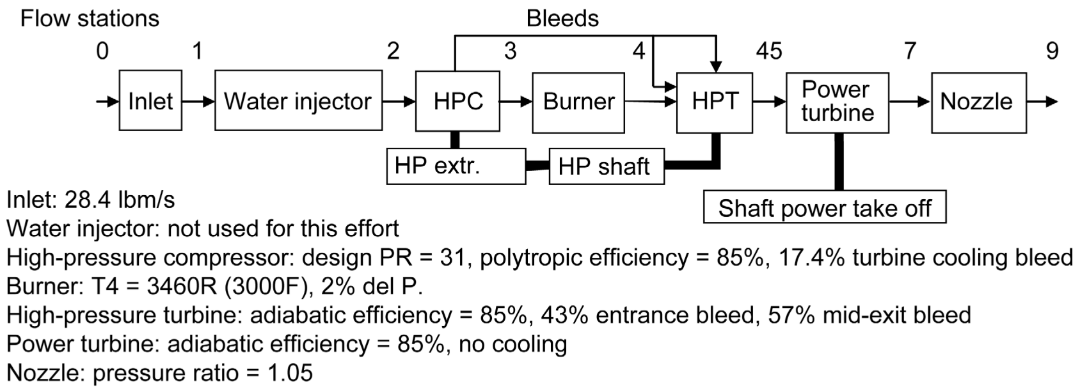

Figure 1.—The notional turboshaft engine system model from Reference 2 as determined by the Numerical Propulsion System Simulator thermodynamic cycle code, showing the requirements for the compressor to deliver a 31:1 pressure ratio at a flow rate of 28.4 lbm/s.

inlet absolute Mach number on the order of 0.50; however, in the latter stages, this limit was reduced in an effort to keep the blade spans as large as possible. The adiabatic efficiency for all rotors was input into the mean-line flow code as follows: tip: 84, mid: 92, hub: 93. The loss coefficient for the stators of 0.05 as defined by Equation (2) was also an input parameter.

$$\omega = \frac{P_{t4} - P_{t2}}{P_{t2} - P_{S2}} \tag{2}$$

The ω is the pressure-loss parameter through the stator P_{t4}–P_{t2} normalized with rotor-exit total pressure P_{t2} and rotor-exit static pressure P_{S2}. The mean-line compressor design analysis method of Reference 3 was used to perform the conceptual designs and analyses. The first conceptual design iteration focused on an all-axial stage design to estimate the overall geometric parameters of the compressor and the aerodynamic parameters for each stage. The aerodynamic parameters for each individual blade row were calculated at the leading edge and trailing edge to determine number of stages that would be required to produce the required pressure ratio. The aerodynamic performance of each rotor blade row was determined, including geometric parameters such as tip and hub radii and blade angles. The absolute flow angle entering each rotor was 0° at the design point operating condition. The work per blade row and the inlet and exit radii were varied, and the required exit blade angles were calculated. The diffusion factor and relative velocity ratios to achieve the prescribed work per blade row were limited to a maximum value to reduce the likelihood of flow separation. An iterative process was used that required variation of key parameters to stay within the limits of maximum diffusion factor per stage. The work split between stages was determined on the basis of the maximum work that could be achieved in a given blade row within the maximum diffusion factor limit. Also monitored during the design process was work coefficient per rotor as defined by Equation (3), where ΔH is the enthalpy rise and U_{Tip} is the rotor-inlet tip speed in feet per second.

$$\varphi = \frac{\Delta H}{U_{\mathrm{Tip}}^{2}} \tag{3}$$

For the off-design analysis a loss model correlated to rotor incidence was used (Ref. 3), as defined by Equation (4).

$$\Delta\eta = 0.0006\ i^3 - 0.0185\ i^2 + 0.1699\ i + 0.5187 \tag{4}$$

The incidence i at the rotor leading edge is the difference between the relative flow angle and the blade angle at the mean radius. The efficiency reduction at off-design values of rotor incidence is defined by Δη. As the rotor incidence changes at off-design flow rates, the rotor efficiency is determined by applying the loss model to the input value of rotor efficiency.

All-Axial Compressor

The blade tip maximum speed limit was 1500 ft/sec and the rotor-tip diameters were tapered or reduced in each successive stage. The flow rate of 28.4 lbm/sec, the maximum tip speed criteria and the hub-to-tip ratio of 0.36 was used to size the first-stage axial compressor, resulting in a design speed of 27 289 rotations per minute. To meet the overall pressure ratio requirement of 31:1, the flow path and each blade row were designed concurrently in an iterative process using the mean-line code. This process determined the pressure ratio that can be achieved in each stage. The number of stages that would be required to meet the overall pressure ratio was determined by an iterative process. The maximum tip speed of 1500 ft/sec was considered to be a structurally acceptable limit, as advanced blade and disk materials could support these speeds. The maximum tip speed limit was also imposed because of concerns about high rotor-inlet tip relative Mach number particularly in the first transonic stage. The taper of the tip flow path was kept at a rate such that the pressure ratio requirements could be met with eight stages without the addition of a ninth stage. Pressure ratios of this magnitude result in high gas temperatures at the compressor exit. The tapered-tip flow path axial-compressor configuration is favorable from a structural perspective, as the latter stages that experience the highest temperatures also operate at reduced tip speeds in comparison to the front-block compressor stages. The hub diameter was allowed to vary to keep the absolute Mach number near 0.50 at the rotor inlet, and within the limit of rotor diffusion factor at the design point. The flow path and the rotor designs were arrived at iteratively while keeping within limits of rotor diffusion factor and rotor-inlet Mach number.

The resulting flow path for the eight-stage axial compressor with tapered-tip diameter flow path is shown in Figure 2. The blade span of the last stage for this design is 0.281 in. at a tip diameter of 11.53 in. The rotor-tip clearance will be on the order of 0.011 in., resulting in the ratio of clearance to span in the first stage of 0.3 percent and the last stage having a clearance-to-span ratio of 4 percent.

Table I shows the results obtained with the mean-line compressor flow analysis and design code for the eight-stage axial compressor with tapered-tip flow path. The pressure ratio per stage tapers off in each subsequent stage, but the work split between the eight stages resulted in a relatively even distribution of power of near 1350 hp per stage, as shown in Table I.

Except for the tip speed and tip relative Mach number, all other rotor parameters in the table are calculated at the root-mean-square diameter. The rotor-inlet root-mean-square radius of each successive rotor increases from 4.72 in. in the first rotor inlet through the first five stages and then gradually becomes constant at 5.71 in. for the last three stages. The first five stages are transonic at the tip. The absolute Mach number at the inlet of each rotor was on the order of 0.50. The relative flow angle at each rotor inlet is on the order of 62° from the axial direction. The rotor exit blade angles are on the order of 40° to 46° for all rotors. The resulting peak overall adiabatic efficiency at 28.4 lbm/sec is 80.3 percent. The power required to drive this compressor is estimated to be 10 781 hp. The flow path coordinates of the rotor tip and hub radial and axial positions are listed in Appendix A. Note that the flow path is based on a mean-line conceptual flow analysis, and may change as the design and analysis progress to higher levels of fidelity. The design point performance for each stage is listed in Appendixes C and D. As a follow on to this study, it is planned to generate preliminary blade shapes with a turbomachinery design code that includes the losses due to tip leakages. Structural conceptual design, material selection, and analyses of the blades and disks is also planned, as the exit temperature of 1568 °R is high, and proper material selection will determine the viability of this compressor.

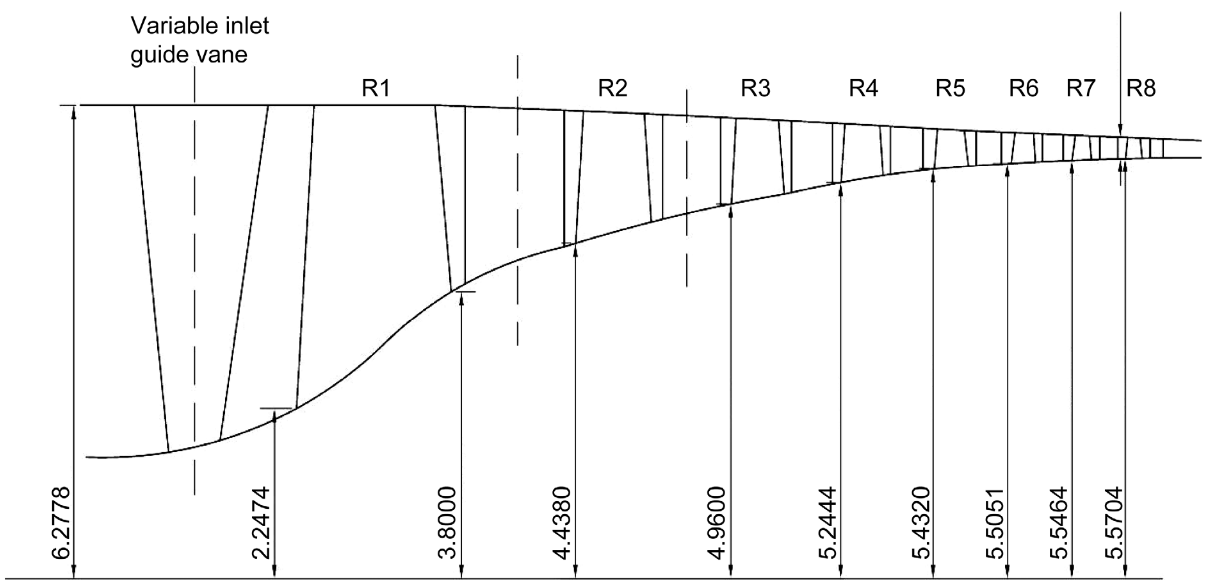

Figure 2.—Axial compressor conceptual design for an eight-stage compressor (units: inches).

TABLE I.—SUMMARY OF THE EIGHT-STAGE ALL-AXIAL COMPRESSOR WITH TAPERED-TIP FLOW PATH

	1	2	3	4	5	6	7	8
Rotor inlet								
Flow rate, corrected, lbm/sec	28.40	15.65	9.61	6.52	4.59	3.38	2.57	1.99
Mach absolute	0.53	0.54	0.47	0.45	0.47	0.44	0.40	0.36
Relative Mach at tip	1.50	1.34	1.19	1.10	1.05	0.98	0.92	0.86
Tip speed (*U*), ft/sec	1495	1475	1452	1434	1419	1407	1399	1392
Relative flow angle, deg	62.7	62.9	64.3	63.8	61.7	61.9	62.8	64.0
Blade angle, deg	57.6	56.7	58.4	57.5	55.6	55.4	56.2	57.6
Rotor exit								
Blade angle, deg	40.5	43.5	46.0	44.2	41.0	41.7	42.3	43.8
Absolute flow angle, deg	47.8	48.2	45.7	42.3	41.1	40.8	42.4	43.2
Flow deviation, deg	4.5	4.4	4.0	3.9	4.0	4.0	4.0	4.0
Diffusion factor	0.50	0.55	0.52	0.49	0.50	0.49	0.51	0.51
Relative velocity ratio	1.95	1.85	1.69	1.61	1.55	1.55	1.57	1.57
Exit temperature, °R	657.9	794.0	922.1	1054.0	1184.0	1313.0	1442.0	1568.0
Stage								
Pressure ratio	2.056	1.789	1.587	1.519	1.440	1.387	1.354	1.314
Temperature ratio	1.268	1.207	1.161	1.143	1.123	1.109	1.099	1.087
Efficiency, adiabatic	85.5	87.6	87.2	87.5	87.1	87.3	87.5	87.6
Work coefficient, φ	0.388	0.395	0.385	0.410	0.416	0.423	0.436	0.435
Horsepower	1391	1364	1290	1343	1336	1339	1367	1350

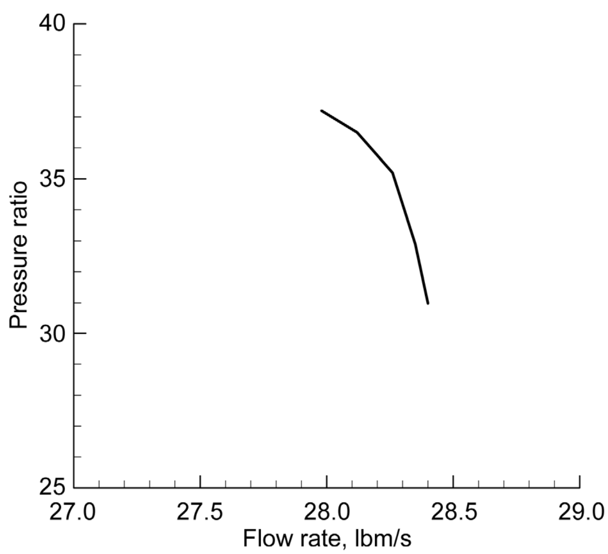

Figure 3.—Pressure ratio at 100 percent speed.

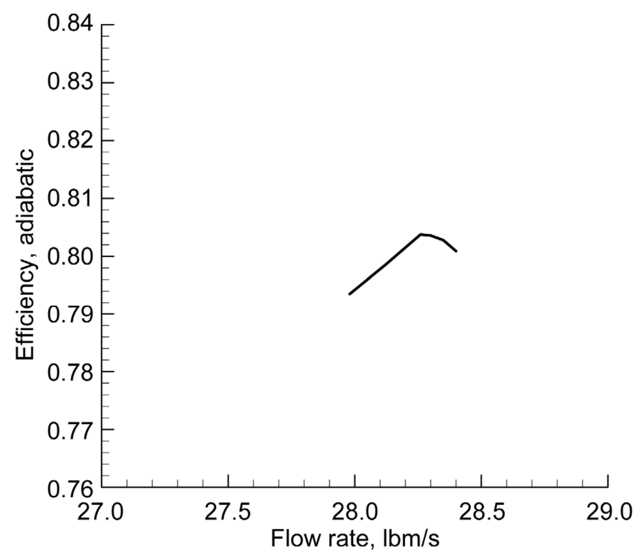

Figure 4.—Adiabatic efficiency at 100 percent speed.

All-Axial Compressor Off-Design Operation

The mean-line methodology was used to perform an off-design analysis of this all-axial compressor at the 100 percent speed line. The criteria that were used for predicting surge in the analysis are a maximum value of a diffusion factor of 0.60. The pressure ratio versus flow is shown in Figure 3, while the adiabatic efficiency versus flow rate is shown in Figure 4. Figure 3 illustrates the steep pressure rise characteristic as the flow is reduced from the design value of 28.4 lbm/sec. The off-design characteristics at a range of part speed operating conditions will be studied in more detail in the future, as it is important to understand the operability of this high-pressure compressor. In addition to the variable inlet guide vanes (IGV), there will likely be a need to have several stator vanes variable as well, to aerodynamically match the stages at part speed operating conditions with acceptable levels of rotor incidence.

Since the last stage blade span of 0.281 in. is low in comparison to traditional designs, a steeper tip-taper rate will also be studied in the future in order to increase the span of the last stage, possibly requiring additional axial stages.

Axial-Centrifugal Compressor

The second configuration that was studied focused on an axial-centrifugal compressor, since traditionally, rotary winged aircraft engines typically feature axial-centrifugal compressors. This design utilized a first-stage axial compressor that is close to the first stage of the previous all-axial case shown in Figure 2, but the tip flow-path taper through the subsequent axial stages is even steeper. The design shaft speed is 27 289 rotations per minute. This conceptual design iteration focused on adding one centrifugal stage to the back end of the compressor to take the place of several axial stages. In addition, the taper of the tip flow path was increased to make the rotor blade spans as large as possible. This further reduced the tip clearance-to-span ratio of the axial rotors, as well as their hub-to-tip ratio. This reduction was considered to be particularly important for the centrifugal impeller, as its efficiency can be negatively influenced by a high inlet hub-to-tip ratio. The increased rate of tip flow-path taper through the axial stages accommodates the transition to the centrifugal impeller with a hub-to-tip ratio as low as possible.

As the tapered-tip flow path already provided this transition to the centrifugal impeller, there was no need to use a transition duct after the last axial stage to further reduce the inlet hub-to-tip ratio. This was done in an effort to reduce pressure losses normally experienced in the "goose neck" of traditional axial-centrifugal compressors.

The centrifugal compressor can effectively produce enough pressure ratio to take the place of several rear-block axial stages. However, there are limitations on the impeller pressure ratio such as specific speed (Ref. 7), which is a normalized aerodynamic parameter that can be used to estimate the flow and pressure ratio where the centrifugal compressor will operate most efficiently. Specific speed N_S is defined by the following Equation (5).

$$N_S = \frac{N\left(Q^{1/2}\right)}{\Delta H^{3/4}} \tag{5}$$

Q is the volumetric flow rate in cubic feet per minute, while ΔH is the enthalpy rise through the rotor, and N is the shaft speed in rotations per minute. The range of centrifugal impeller specific speed that typically has the highest potential level of efficiency is 80 to 90. If the centrifugal is not designed to be in this range of specific speed, the maximum attainable efficiency will be limited. Another limitation of centrifugal compressors is the structural and material limitation of tip speed at high operating temperatures.

A mechanical design study for the LCTR engine has not yet been done, as it is currently only in the study phase. Rotordynamics analyses would be done on the engine system after the complete shaft assembly, including the turbines, bearings, and seals, has been sized. Likely, there will be mechanical and rotordynamics considerations that will influence the evolution of the final compressor flow path. The cross section for the compressor that resulted from this conceptual design is shown in Figure 5.

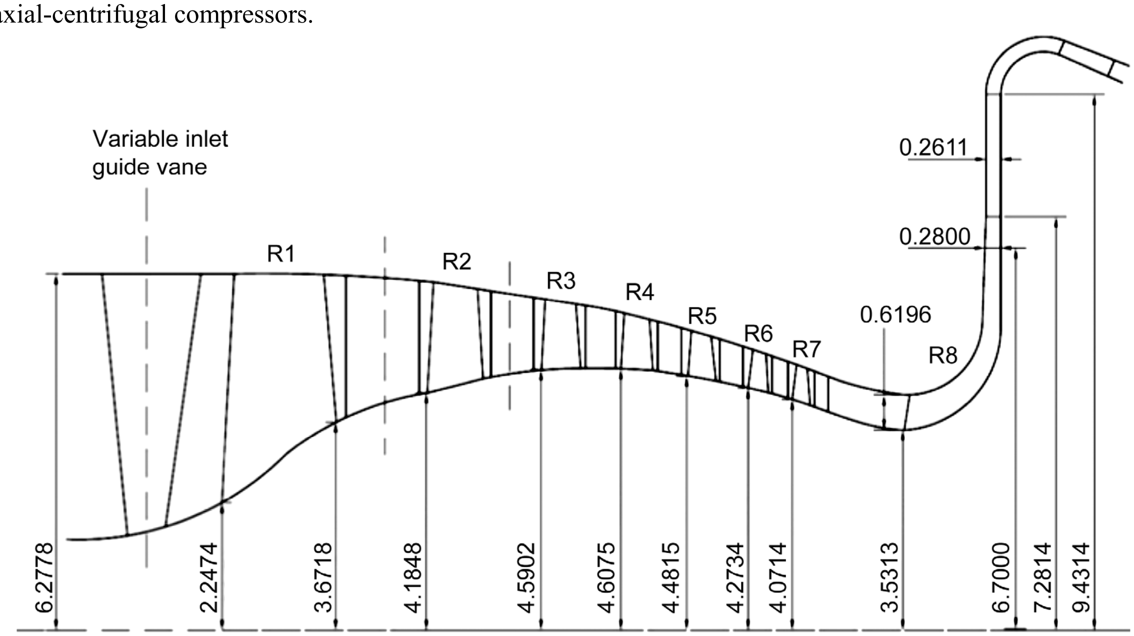

Figure 5.—Axial-centrifugal compressor conceptual design sizing featuring seven axial stages followed by a centrifugal.

The wall curvature of the flow path leading into the centrifugal stage has been sized to reduce curvature effects on local velocity, which has an effect on the pressure losses encountered in the duct. The flow path coordinates of the rotor tip and hub radial and axial positions are listed in Appendix A. Note that these coordinates are based on a mean-line conceptual flow analysis, and may change as the design and analysis progress to higher levels of fidelity. In addition, subsequent structural analyses may indicate that the limits on maximum tip speed may be different from 1500 ft/sec used in this conceptual design. The design point performance for each stage is listed in Appendixes C and D.

The work split among the seven axial stages was tailored to be as even as possible and within the limits on rotor diffusion factor. The specific speed of the centrifugal compressor stage is 56 and is lower than the level considered to be near the optimum value for maximum efficiency potential. To be closer to what is considered to be the optimum specific speed, more of the work would have to be done by the axial stages, or more stages would be required, and the centrifugal compressor pressure ratio would need to be reduced (lower impeller exit diameter, and/or more back swept blade angle). Another way to increase the specific speed of the centrifugal stage is to have a two-spool compressor, as suggested in Reference 2. The second spool would rotate at a faster speed, allowing more flexibility to optimize the specific speed of the centrifugal stage. The two-spool compressor will be studied in more detail in the future.

Since the tip flow path taper is steeper than the previous all-axial compressor case, the blade spans for the rear block axial stages are larger, culminating with a 0.61 in. blade span of the seventh (last axial) stage rotor. This significant increase in blade span from the previous all-axial case was caused by two factors (1) the reduced rotor-tip diameters resulted in reduced pressure ratio per stage from the previous case and (2) as the rotor-tip diameter was reduced, the annular area in the flow path required a larger span to accommodate the increased volume flow. Even with a 0.62-in. impeller inlet span that was the result of reduced tip flow path through the axial stages, this design produced a high impeller inlet hub-to-tip ratio of 0.85. The centrifugal impeller exit height is 0.280 in., and the exit blade angle has a 20° back sweep from the radial direction. The tip clearance of this impeller will likely vary from inlet to exit. Table II summarizes the key stage-by-stage mean-line performance parameters for the seven-stage axial compressor followed by the centrifugal stage.

A reasonable axial running clearance of this impeller at the exit tip is on the order of 0.005 in., or near 2 percent of the exit span. It will be necessary to maintain tight axial clearances to minimize tip leakages and prevent a reduction in impeller efficiency.

TABLE II.—THE AXIAL-CENTRIFUGAL COMPRESSOR DESIGN POINT STAGE-BY-STAGE AERODYNAMIC PERFORMANCE

	1	2	3	4	5	6	7	8 Centrifugal
Rotor inlet								
Flow rate, corrected, lbm/sec	28.4	15.81	9.86	6.82	5.05	3.92	3.18	2.65
Mach absolute	0.53	0.50	0.49	0.44	0.41	0.40	0.37	0.29
Relative Mach at tip	1.50	1.31	1.16	1.03	0.93	0.85	0.78	0.65
Tip speed (*U*), ft/sec	1495	1452	1379	1319	1248	1169	1106	1595 (exit)
Relative flow angle, deg	62.7	63.6	62.0	62.4	61.6	59.7	59.2	60.6
Blade angle	57.6	57.4	55.7	56.2	55.2	53.2	53.0	54.5
Rotor exit								
Blade angle, deg	40.1	41.5	42.0	40.5	38.0	36.0	34.0	20.0
Absolute flow angle, deg	50.1	47.5	44.2	42.5	43.1	40.2	41.1	63.8
Flow deviation, deg	4.8	4.5	4.2	4.2	4.4	4.2	4.4	18.2
Diffusion factor	0.43	0.55	0.53	0.52	0.53	0.51	0.52	0.74
Relative velocity ratio	2.00	1.88	1.77	1.69	1.70	1.65	1.67	1.54
Exit temperature, °R	654.9	785.7	906.0	1017.	1119.	1212.	1298.	1577.
Stage								
Pressure ratio	2.031	1.755	1.553	1.431	1.351	1.284	1.241	1.818
Temperature ratio	1.263	1.200	1.153	1.123	1.101	1.083	1.071	1.215
Efficiency, adiabatic	85.8	87.5	87.4	87.2	87.3	87.1	86.8	82.8
Work coefficient, φ	0.379	0.403	0.408	0.418	0.438	0.450	0.470	0.718
Horsepower	1361	1296	1211	1126	1053	957	897	2935

The reduced tip radii of each subsequent axial rotor are also good from a structural perspective, since the operating temperatures of each subsequent axial rotor are higher from the previous stage, and a reduced blade tip speed in the latter rotors can result in a more structurally acceptable design. The material selection and structural design of the centrifugal impeller needs careful consideration as its tip speed is 1595 ft/sec at an exit temperature of 1565 °R. The feasibility of this rotor needs to be verified with structural and thermal analyses.

The resulting overall compressor efficiency at 28.4 lb/sec is 79.6 percent adiabatic. The power required to drive it is estimated to be 10 850 hp. The results from Tables I and II are shown in Appendix B as plots comparing the values obtained from the all-axial compressor study and the axial-centrifugal compressor.

As shown in Table II, the diffusion factor of the axial rotors at the design point ranges between 0.43 and 0.55. These values indicate that there is a surge margin available. The centrifugal impeller design point relative velocity ratio from inlet to exit is 1.54. The value of diffusion factor for the centrifugal impeller is higher at the design point (0.74) than the axial rotor diffusion factors, but this is not unexpected as centrifugal impellers are typically more highly loaded than axial blades.

Structural conceptual design, material selection, and analyses of the blades and disks are planned to determine whether this design is feasible, as the exit temperature of 1577 °R and impeller tip speed of 1595 ft/sec may be challenging with current material capabilities. The development of a material that can support these high tip speeds at high temperatures will determine the viability of this compressor.

Axial-Centrifugal Off-Design

The mean-line methodology of Reference 3 was used to perform an off-design analysis of this axial-centrifugal compressor at the 100 percent speed line. The criteria that were used for modeling the onset of surge were a maximum value of rotor diffusion factor of 0.60 for the axial rotors and relative velocity ratio of 1.95. Figure 6 and 7 show the compressor performance along the 100 percent speed line. As the flow rate is reduced to 27.7 lbm/sec, the maximum relative velocity ratio that is experienced in the axial rotors is 1.9 and 1.77 in the centrifugal impeller. Based on the relative velocity criteria, it appears that at the 100 percent speed, stall will be initiated in the axial rotors and not in the centrifugal impeller. However, the diffusion factor limit for the centrifugal compressors at surge needs further validation.

Further analyses of this compressor are planned to determine the variable geometry schedule that will be necessary to operate it with an acceptable surge margin at part speed.

The pressure ratio versus flow rate of the axial-centrifugal compressor is illustrated in Figure 6.

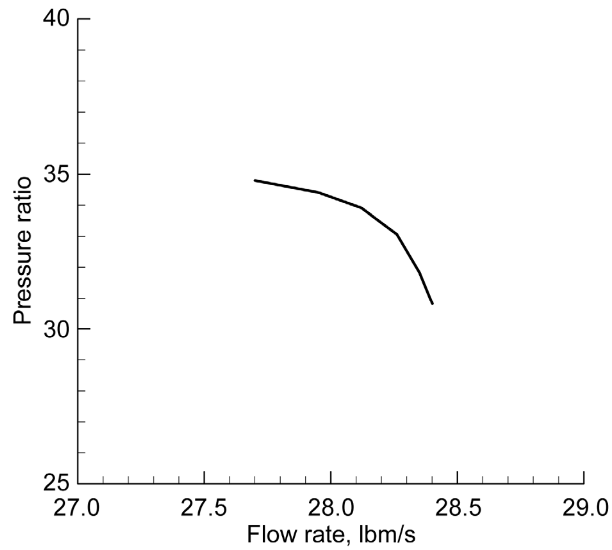

Figure 6.—Pressure ratio on the 100 percent speed line.

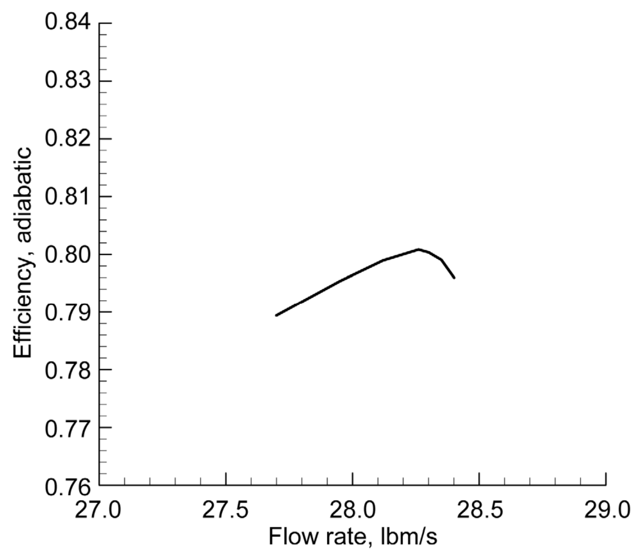

Figure 7.—Adiabatic efficiency on 100 percent speed line.

As illustrated in Figure 6, the pressure rise characteristic of this compressor is shallow as the flow is reduced from the design point value. The adiabatic efficiency versus flow rate is shown in Figure 7. The axial-centrifugal compressor pressure rise characteristic was compared to the all-axial compressor pressure rise characteristic in Figure 8. The axial-centrifugal compressor has less pressure rise as the flow is reduced, than does the all-axial compressor case. The axial-centrifugal compressor appears to have higher flow margin before surge is encountered, in comparison to the all-axial case. The reduced pressure rise to surge of the axial-centrifugal configuration and the additional flow margin are likely due to the centrifugal compressor. Figure 9 illustrates the efficiency characteristics of the all-axial versus the axial-centrifugal compressor. The peak efficiency of both compressors is at a flow rate of 28.3 lbm/sec.

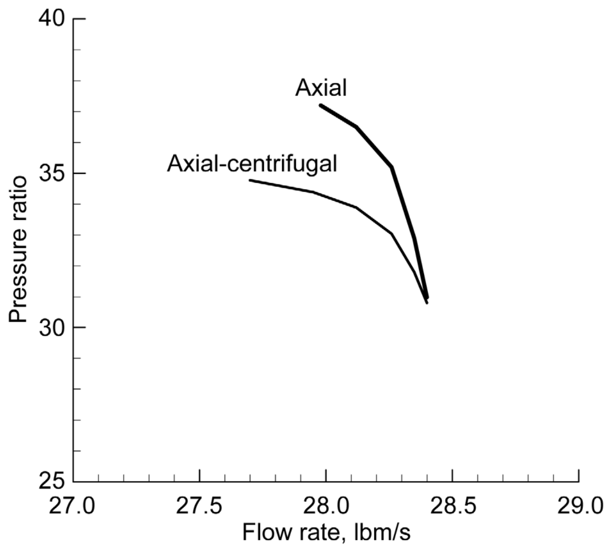

Figure 8.—Comparison of the all-axial to the axial-centrifugal configuration pressure rise characteristics.

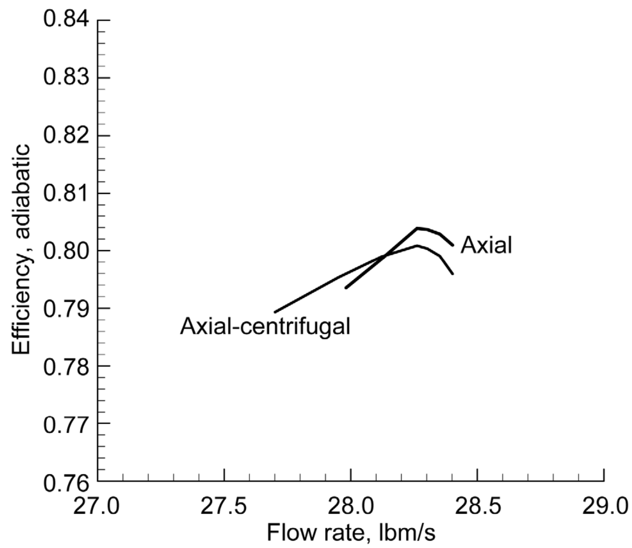

Figure 9.—Comparison of the all-axial to the axial-centrifugal configuration efficiency characteristics.

Conclusions

A conceptual design study was made on two potential compressors to meet the Large Civil Tilt Rotorcraft (LCTR) engine pressure ratio and flow requirements. An all-axial compressor was sized with a tapered-tip flow path, resulting in the last-stage rotor span of 0.281 in., and a tip-to-span clearance ratio of 4 percent. The hub-to-tip ratio of the last rotor is high at 0.95. Further study of the all-axial configuration is planned to increase the blade span of the last stage by increasing the tip taper and likely requiring additional axial stages. A second compressor was studied, featuring an axial-centrifugal compressor configuration with seven axial stages followed by a centrifugal stage. The centrifugal compressor has a specific speed of 56, which is low even for centrifugals. The impeller inlet also has a high hub-to-tip ratio of 0.85 and an exit height of 0.28 in. The high hub-to-tip ratio and low exit height will require tight control of eccentricity and axial clearance for this compressor configuration to keep rotor-tip leakages at acceptable levels. It is planned to do a conceptual design study of a two-spool compressor in the future that can provide additional flexibility to increase the specific speed of the centrifugal stage. The result of this study shows that from an aerodynamic perspective, it may be feasible to meet the requirements of a 31:1 pressure ratio compressor for the LCTR engine with either the all-axial compressor with a tapered-tip flow path, or with an axial-centrifugal compressor featuring seven axial stages, followed by a centrifugal stage. The high rotor-tip speeds of the latter compressor stages operating at temperatures up to 1580 °R will require advanced alloy materials be selected that are typically used to make turbines. The selection of materials for these high-tip-speed rotors operating in a high-temperature environment is a technology that will enable the feasibility of these compressors. Further study is planned with higher fidelity turbomachinery design and flow analysis codes to estimate the compressor efficiencies with clearance effects. Preliminary structural and thermal analyses of the blades and vanes are needed to determine if the required tip speeds from this conceptual design study can be achieved with currently available advanced alloy rotor materials. The mechanical design of the compressor and turbine shafts will also influence the feasibility of the flow paths in this study.

Appendix A.—Compressor Coordinates

The coordinates listed below describe the hub and tip flow path contour for both the eight-stage all-axial compressor and the seven-stage axial-centrifugal compressor. LE is the blade or vane leading edge and TE is the blade or vane trailing edge.

Eight-stage all-axial compressor

	Tip		Hub	
	X	R	X	R
Rotor 1 LE	0.2218	6.2778	0.0000	2.2474
Rotor 1 TE	1.7847	6.2778	1.9957	3.8000
Stator 1 LE	2.1728	6.2634	2.1728	3.9049
Stator 1 TE	3.4517	6.2120	3.4517	4.3956
Rotor 2 LE	3.7025	6.2009	3.6033	4.4439
Rotor 2 TE	4.4893	6.1646	4.5798	4.7246
Stator 2 LE	4.7238	6.1534	4.7238	4.7615
Stator 2 TE	5.4735	6.1162	5.4735	4.9363
Rotor 3 LE	5.6755	6.1059	5.6149	4.9659
Rotor 3 TE	6.2246	6.0770	6.2972	5.0945
Stator 3 LE	6.3877	6.0682	6.3877	5.1098
Stator 3 TE	6.9196	6.0390	6.9196	5.2288
Rotor 4 LE	7.0800	6.0300	7.0314	5.2503
Rotor 4 TE	7.5312	6.0080	7.5753	5.3435
Stator 4 LE	7.6681	6.0001	7.6681	5.3576
Stator 4 TE	8.0874	5.9756	8.0874	5.4143
Rotor 5 LE	8.2775	5.9643	8.2373	5.4320
Rotor 5 TE	8.6236	5.9434	8.6770	5.4674
Stator 5 LE	8.7729	5.9343	8.7729	5.4744
Stator 5 TE	9.0998	5.9140	9.0998	5.5051
Rotor 6 LE	9.2801	5.9146	9.2282	5.5051
Rotor 6 TE	9.5195	5.9041	9.5425	5.5234
Stator 6 LE	9.6297	5.8993	9.6297	5.5280
Stator 6 TE	9.8973	5.8875	9.8973	5.5411
Rotor 7 LE	10.0574	5.8805	10.0159	5.5464
Rotor 7 TE	10.2404	5.8725	10.2656	5.5564
Stator 7 LE	10.3702	5.8668	10.3702	5.5601
Stator 7 TE	10.6039	5.8565	10.6039	5.5675
Rotor 8 LE	10.7352	5.8507	10.7071	5.5704
Rotor 8 TE	10.8959	5.8437	10.9208	5.5755
Stator 8 LE	11.0220	5.8381	11.0220	5.5776
Stator 8 TE	11.1900	5.8308	11.1900	5.5804

Seven-stage axial-1 centrifugal compressor

	Tip		Hub	
	X	R	X	R
Rotor 1 LE	0.2218	6.2778	0.0000	2.2474
Rotor 1 TE	1.7858	6.2647	2.0067	3.6718
Stator 1 LE	2.1728	6.2483	2.1728	3.7529
Stator 1 TE	3.4517	6.1508	3.4517	4.1597
Rotor 2 LE	3.7072	6.1313	3.6019	4.1848
Rotor 2 TE	4.4767	6.0112	4.5760	4.4313
Stator 2 LE	4.7146	5.9741	4.7146	4.4598
Stator 2 TE	5.4592	5.8578	5.4592	4.5707
Rotor 3 LE	5.6627	5.8261	5.5970	4.5902
Rotor 3 TE	6.1991	5.7423	6.2887	4.6127
Stator 3 LE	6.3668	5.7161	6.3788	4.6141
Stator 3 TE	6.8933	5.6171	6.8933	4.6103
Rotor 4 LE	7.0470	5.5695	6.9848	4.6075
Rotor 4 TE	7.4863	5.4694	7.5454	4.5781
Stator 4 LE	7.6303	5.4339	7.6303	4.5717
Stator 4 TE	8.0420	5.3223	8.0420	4.5016
Rotor 5 LE	8.2207	5.2718	8.1611	4.4815
Rotor 5 TE	8.5639	5.1714	8.6409	4.3913
Stator 5 LE	8.7134	5.1262	8.7134	4.3791
Stator 5 TE	9.1205	4.9985	9.1205	4.2940
Rotor 6 LE	9.2980	4.9408	9.2135	4.2734
Rotor 6 TE	9.5245	4.8652	9.5654	4.1882
Stator 6 LE	9.6303	4.8291	9.6303	4.1756
Stator 6 TE	9.9066	4.7329	9.9066	4.0952
Rotor 7 LE	10.0604	4.6780	9.9850	4.0714
Rotor 7 TE	10.2467	4.6101	10.2979	3.9663
Stator 7 LE	10.3712	4.5639	10.3712	3.9402
Stator 7 TE	10.6099	4.4735	10.6099	3.8536
Rotor 8 LE	12.0406	4.1509	11.9423	3.5313
Rotor 8 TE	13.3540	6.7000	13.6340	6.7000
Stator 8 LE	13.3722	7.2814	13.6340	7.2814
Stator 8 TE	13.3722	9.4314	13.6340	9.4314

Appendix B.—Plots of Compressor Parameters at the Design Point

Figures 10 to 23 present plots comparing the values obtained from the all-axial compressor study and the axial-centrifugal compressor (see Tables I and II in text).

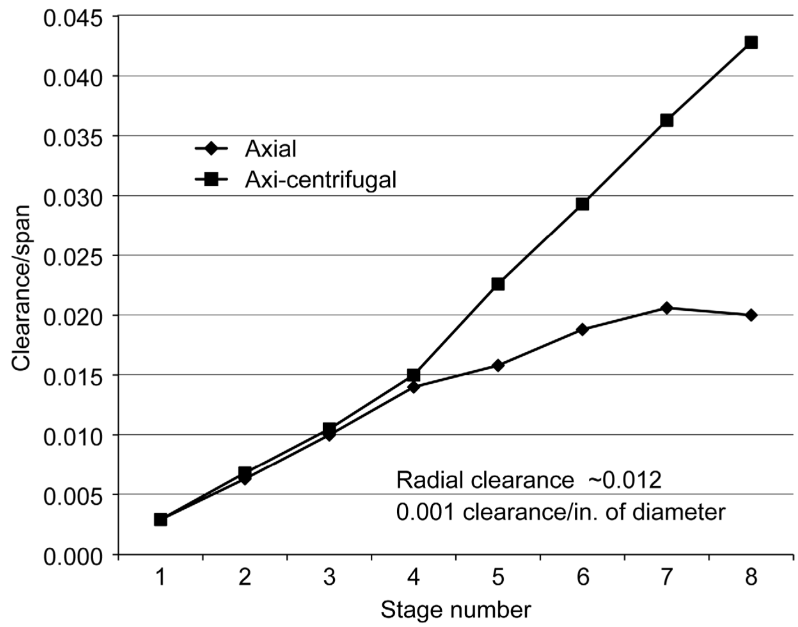

Figure 10.—Radial tip clearance and span comparison for the rotor leading edge.

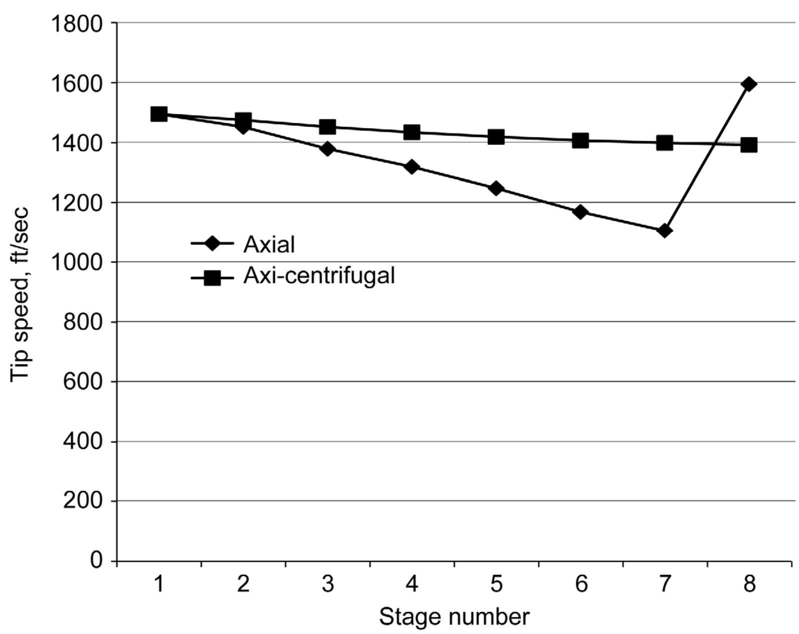

Figure 11.—Rotor peripheral tip speed versus stage.

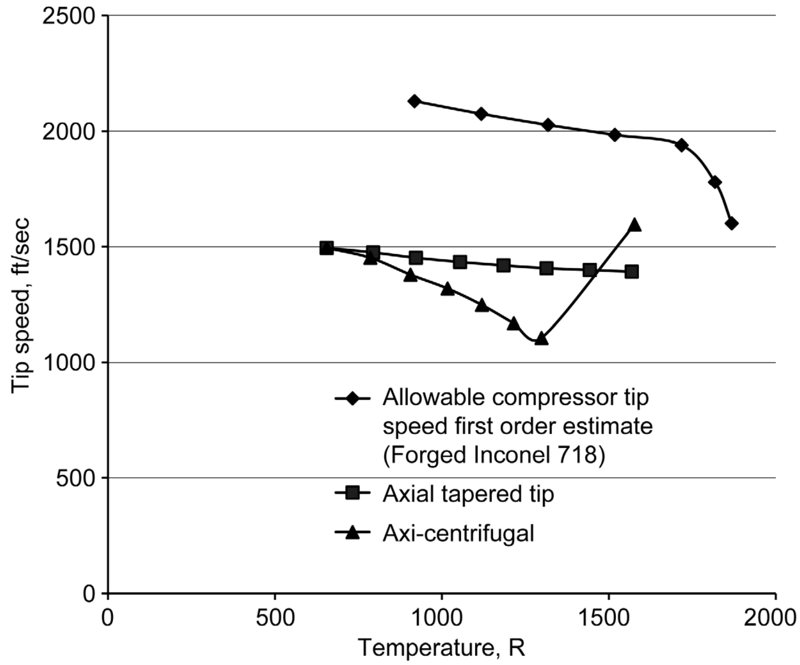

Figure 12.—Compressor stage exit temperature versus tip speed, compared to estimated maximum allowable for forged Inconel 718.

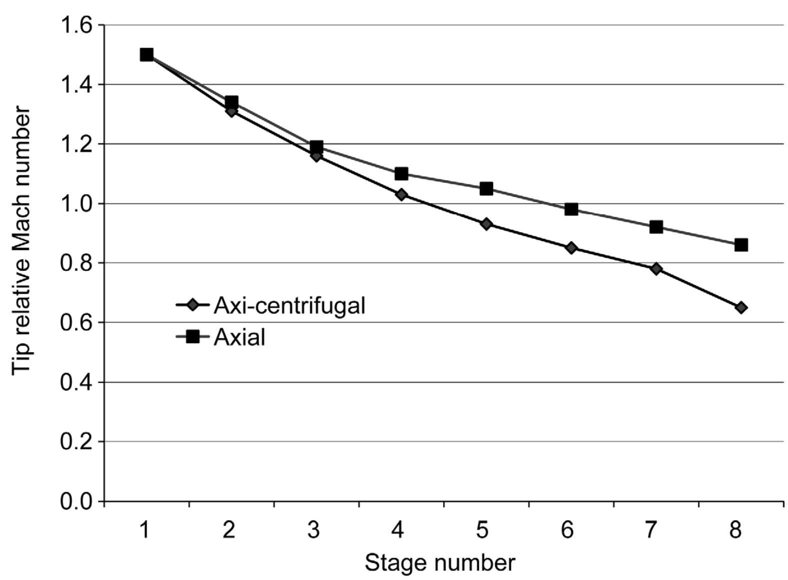

Figure 13.—Rotor inlet tip relative Mach number versus stage.

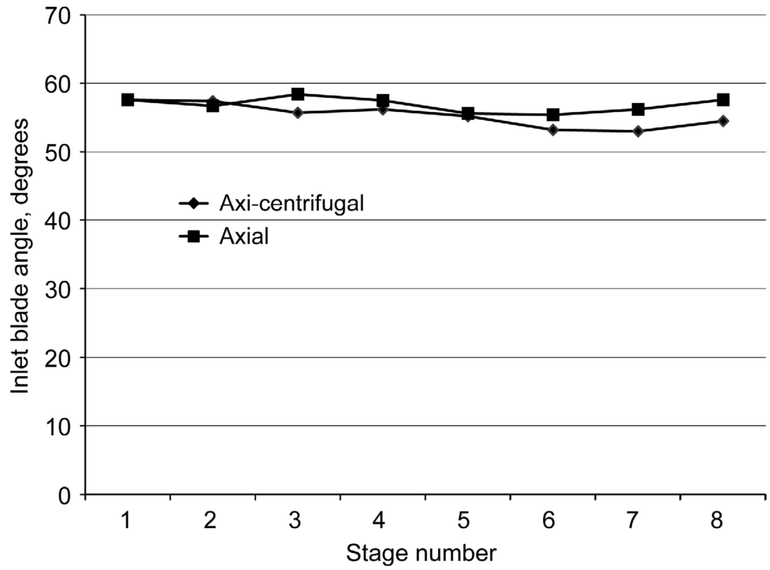

Figure 14.—Rotor inlet blade angle versus stage.

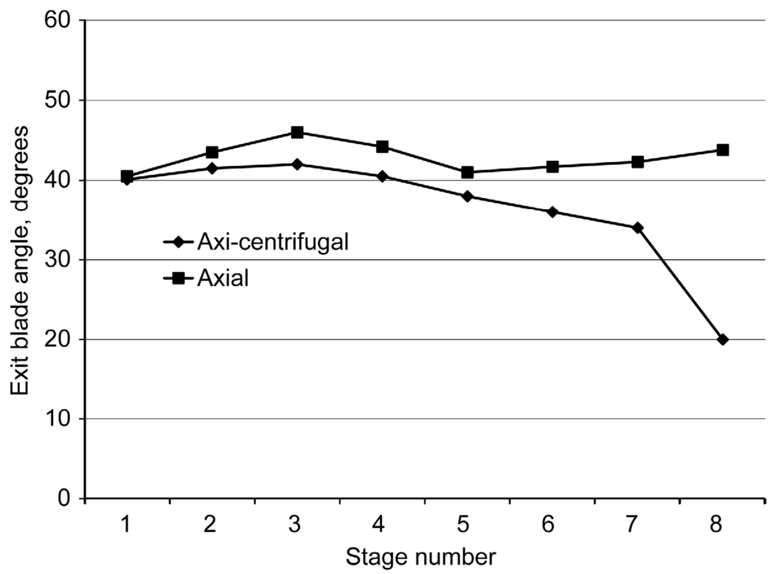

Figure 15.—Rotor exit blade angle versus stage.

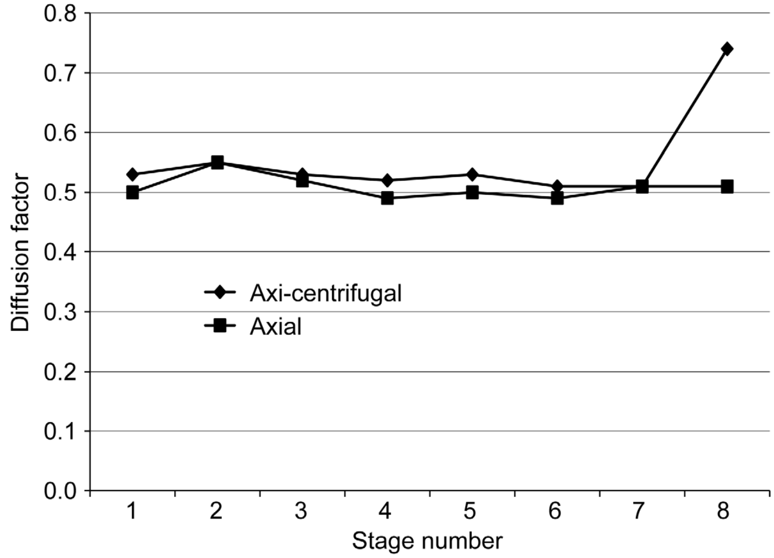

Figure 16.—Rotor diffusion factor versus stage.

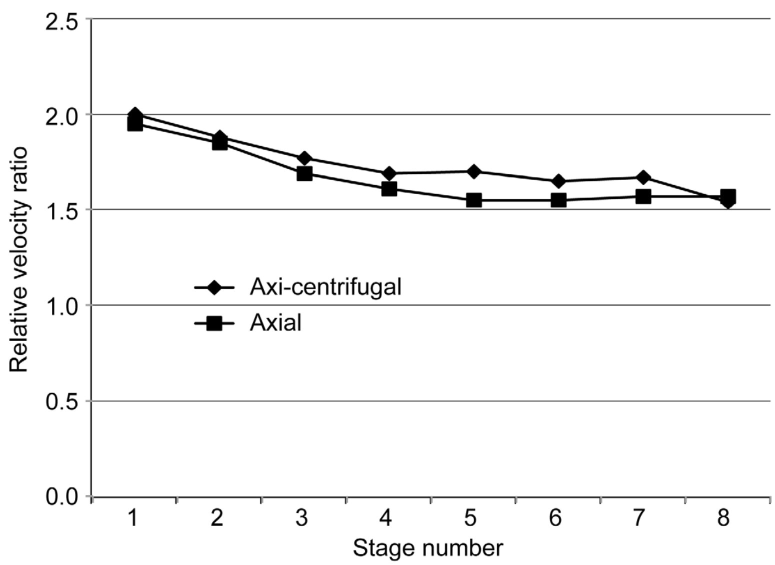

Figure 17.—Rotor relative velocity ration versus stage.

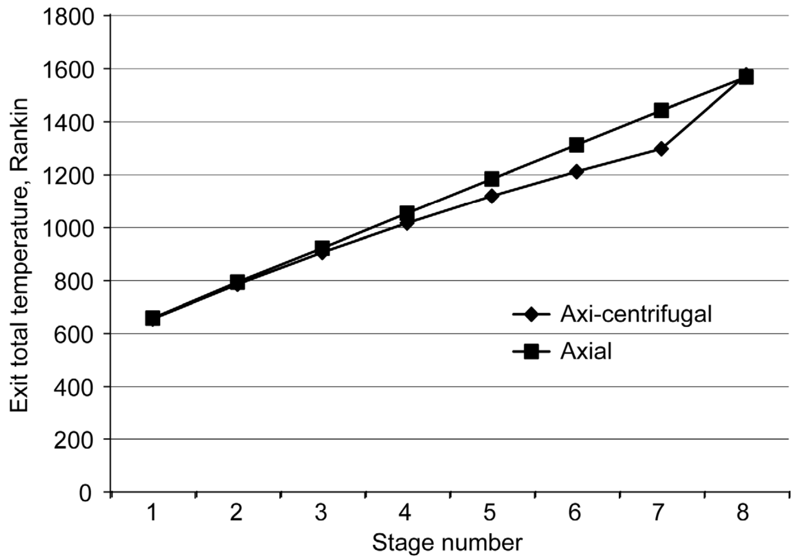

Figure 18.—Rotor exit total temperature versus stage.

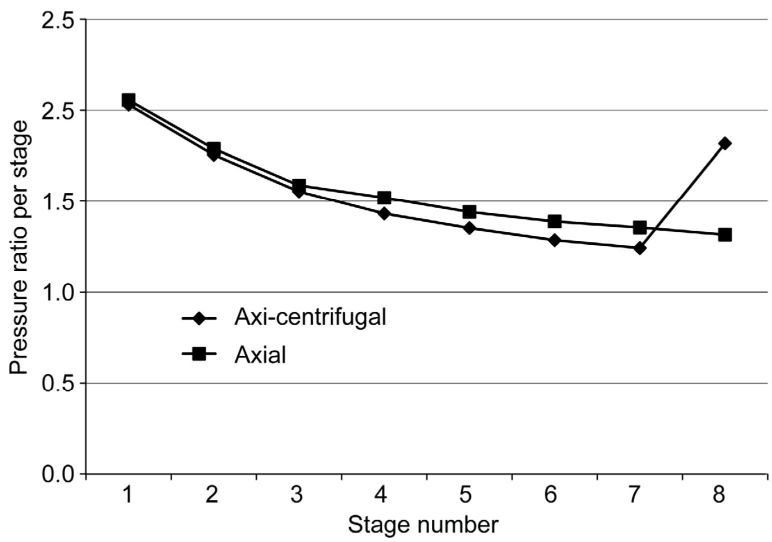

Figure 19.—Rotor total-to-total pressure ratio per stage.

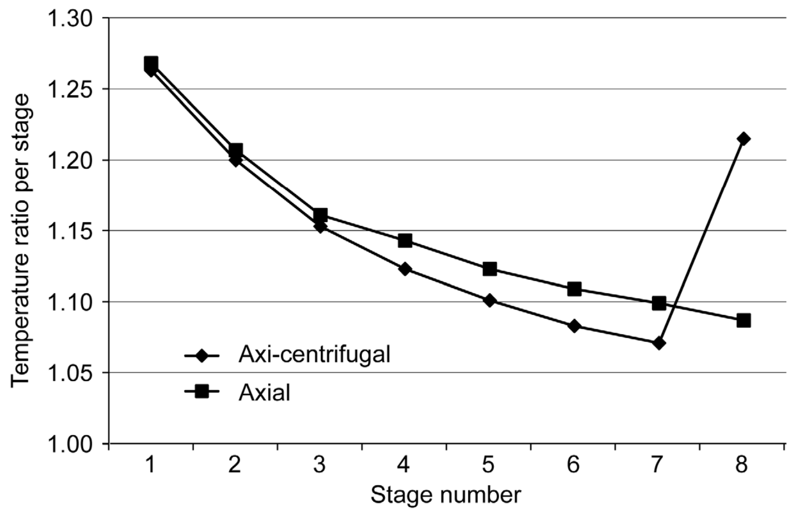

Figure 20.—Rotor total-to-total temperature ratio per stage.

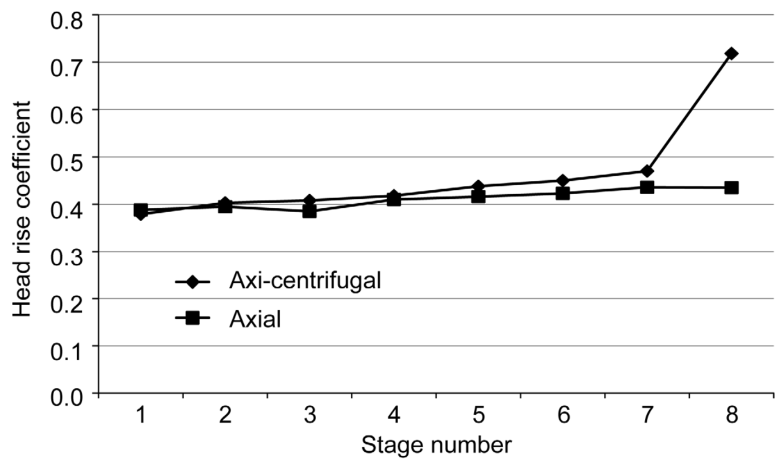

Figure 21.—Stage head rise coefficient.

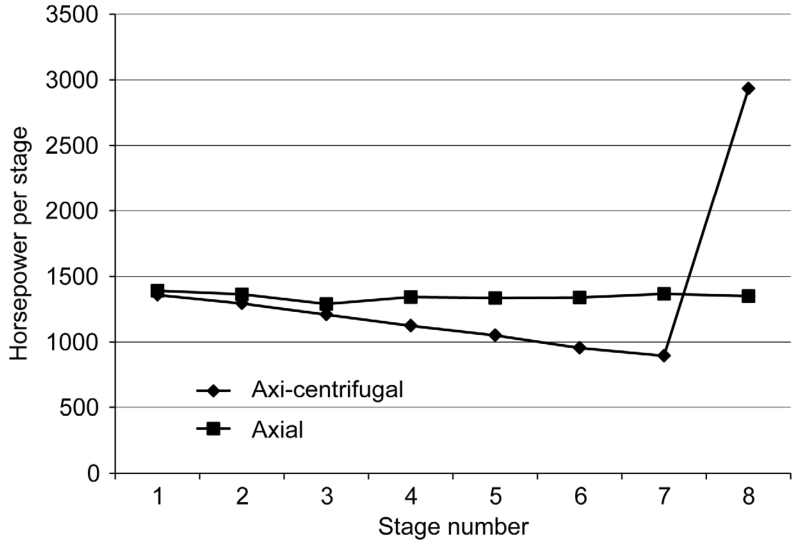

Figure 22.—Horsepower per stage.

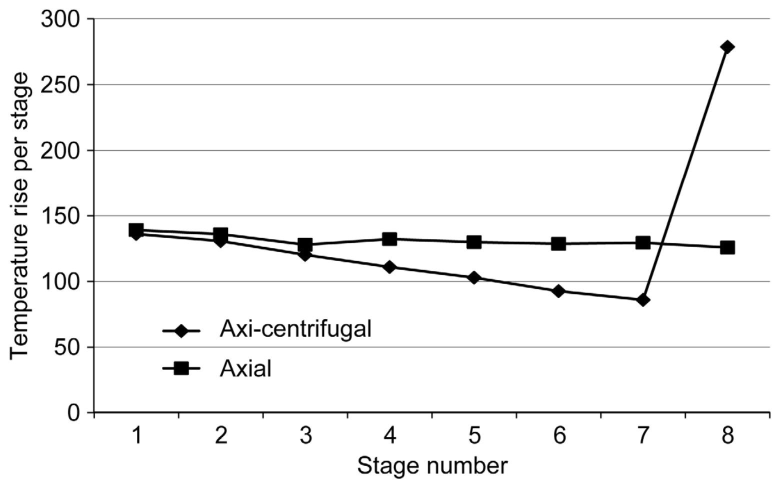

Figure 23.—Total temperature rise per stage.

Appendix C.—All-Axial Compressor Data

Below are the all-axial compressor mean-line flow analysis results at the design point operating condition.

```
   LCTR  8 Axial Compressor Stage 100% Speed

       COMPRESSOR INLET CONDITIONS, STAGE             1
RESET =  0.000 BLEED =  0.000  DP Inc 5.000   Kg/sec = 12.90425

     W act     RPM act        Pt           Tt          'POTS         POTH       AeroBl
        28.39   27289.000      14.613     518.670       1.100        0.800       0.980

     W cor     RPM cor        GAMMA        Cp          'R           NBLAD        THK
        28.39   27289.789       1.402       0.249      53.349      28.000       0.030

      ROTOR LEADING EDGE CONDITIONS, STAGE             1
           R1      Stator    Alfa      C1       CU1     Cm1     Abs MACH
  TIP      6.28      0.00    -0.02   636.07    -0.22   636.07     0.59
  MEAN     4.72      0.00    -0.02   578.24    -0.20   578.24     0.53
  HUB      2.25      0.00    -0.02   462.60    -0.16   462.60     0.42

         BetaFlo   BetaBlade Incid     U1      W1       Ps1      Ts1     Rel Mach
  TIP     66.96     62.00     4.96  1495.53  1625.38    11.58   485.21     1.50
  MEAN    62.76     57.66     5.10  1123.15  1263.44    12.05   490.74     1.16
  HUB     49.17     44.20     4.97   535.10   707.46    12.95   500.97     0.64

       ROTOR EXIT CONDITIONS, STAGE             1

     B2 axial    THK         AeroBl
     0.30       0.030       0.950

           R2      C2       Cu2      Cm2     Ao2      Mach2
  TIP      6.28   890.20   702.10   547.27  1222.90     0.73
  MEAN     5.19   874.11   647.99   586.67  1185.39     0.74
  HUB      3.80  1082.43   829.37   695.56  1143.21     0.95

         U2       W2       Wu2      Mach Rel2  Ws1/W2
  TIP   1495.53   963.86   793.43     0.79
  MEAN  1236.03   830.64   588.04     0.70     1.96
  HUB    904.94   699.65    75.57     0.61

           Pt2      PR       Ps2     Tt2       TR      Ts2       Eff2
  TIP     33.52     2.29    23.55   687.41     1.33   621.24     0.83
  MEAN    29.62     2.03    20.63   647.37     1.25   583.59     0.90
  HUB     28.63     1.96    16.06   639.26     1.23   541.65     0.91

           Alfa2  Beta FLO Beta BLADE Deviat   Slip F. Diff Fct Solidity
  TIP     52.06    55.40    51.50     3.90     0.93     0.57     1.32
  MEAN    47.84    45.07    40.50     4.57     0.93     0.50     1.75
  HUB     50.01     6.20     1.00     5.20     0.93     0.21     3.68

     STAGE EXIT CONDITIONS, STAGE             1

    DIFF LOSS       Effic      Pdisch       PR          TR         Ns          Ns nondim
         0.05       0.855      30.065      2.057       1.269     267.791        2.076

 Del Enthalpy   Del_H/U^2    GHP       Reynolds#
    867478.44       0.388    1391.703 1191539.375
```

COMPRESSOR INLET CONDITIONS, STAGE 2

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	30.065	658.012	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
15.63	24228.629	1.401	0.249	53.349	38.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 2

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	6.20	0.00	-0.02	720.54	-0.25	720.54	0.59
MEAN	5.39	0.00	-0.02	655.04	-0.23	655.04	0.54
HUB	4.44	0.00	-0.02	589.54	-0.20	589.54	0.48

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	63.97	57.82	6.15	1475.29	1642.07	23.76	615.16	1.34
MEAN	62.96	56.70	6.26	1283.25	1440.97	24.73	622.24	1.18
HUB	60.85	54.60	6.25	1056.87	1210.36	25.73	629.32	0.98

ROTOR EXIT CONDITIONS, STAGE 2

B2 axial	THK	AeroBl
0.20	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	6.16	885.95	663.11	587.53	1341.71	0.66
MEAN	5.49	881.34	657.35	587.06	1325.89	0.66
HUB	4.72	900.82	641.19	632.74	1303.07	0.69

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1466.71	995.48	803.60	0.74	
MEAN	1306.45	875.19	649.09	0.66	1.85
HUB	1123.55	795.64	482.36	0.61	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	56.83	1.89	42.40	813.88	1.24	748.46	0.84
MEAN	55.75	1.85	41.44	795.64	1.21	730.90	0.92
HUB	51.09	1.70	37.11	773.46	1.18	705.83	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	48.46	53.83	50.00	3.83	0.93	0.55	1.27
MEAN	48.23	47.87	43.50	4.37	0.93	0.55	1.46
HUB	45.38	37.32	32.50	4.82	0.93	0.50	1.77

STAGE EXIT CONDITIONS, STAGE 2

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.876	53.845	1.791	1.207	213.512	1.655

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
851000.19	0.396	1365.266	794049.563

COMPRESSOR INLET CONDITIONS, STAGE 3

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	53.845	794.328	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
9.59	22051.896	1.398	0.251	53.349	50.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 3

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	6.10	0.00	-0.02	699.40	-0.24	699.40	0.52
MEAN	5.56	0.00	-0.02	635.82	-0.22	635.82	0.47
HUB	4.96	0.00	-0.02	572.24	-0.20	572.24	0.42

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	64.29	58.20	6.09	1452.66	1612.48	44.87	754.12	1.19
MEAN	64.35	58.40	5.95	1323.90	1468.86	46.28	760.77	1.09
HUB	64.16	58.40	5.76	1181.18	1312.67	47.71	767.41	0.96

ROTOR EXIT CONDITIONS, STAGE 3

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	6.07	873.06	624.63	609.97	1449.37	0.60
MEAN	5.60	861.81	618.16	600.50	1440.31	0.60
HUB	5.09	848.71	568.87	629.84	1423.84	0.60

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1445.76	1022.89	821.12	0.71	
MEAN	1333.91	934.29	715.75	0.65	1.69
HUB	1211.78	900.01	642.91	0.63	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	88.57	1.64	69.33	938.26	1.18	875.00	0.84
MEAN	88.57	1.64	69.55	925.75	1.17	864.11	0.92
HUB	82.37	1.53	64.79	904.19	1.14	844.42	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	45.68	53.39	49.70	3.69	0.93	0.52	1.22
MEAN	45.83	50.00	46.00	4.00	0.93	0.52	1.34
HUB	42.09	45.59	41.50	4.09	0.93	0.46	1.51

STAGE EXIT CONDITIONS, STAGE 3

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.873	85.573	1.589	1.162	179.754	1.393

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
806058.38	0.386	1293.166	650593.000

COMPRESSOR INLET CONDITIONS, STAGE 4

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	85.573	922.735	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
6.50	20460.078	1.395	0.253	53.349	60.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 4

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	6.02	0.00	-0.02	724.66	-0.25	724.66	0.50
MEAN	5.65	0.00	-0.02	658.78	-0.23	658.78	0.45
HUB	5.24	0.00	-0.02	592.91	-0.20	592.91	0.40

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	63.20	57.00	6.20	1434.56	1607.43	72.33	879.86	1.10
MEAN	63.91	57.50	6.41	1344.94	1497.83	74.41	886.96	1.03
HUB	64.61	58.00	6.61	1248.91	1382.69	76.53	894.04	0.94

ROTOR EXIT CONDITIONS, STAGE 4

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	6.00	926.43	632.40	677.01	1543.43	0.60
MEAN	5.68	902.12	610.45	664.20	1536.64	0.59
HUB	5.34	916.89	621.65	673.98	1530.54	0.60

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1429.32	1045.68	796.93	0.68	
MEAN	1352.53	995.92	742.08	0.65	1.62
HUB	1271.10	935.97	649.46	0.61	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	132.14	1.54	103.67	1065.67	1.15	994.92	0.84
MEAN	132.17	1.54	104.75	1053.29	1.14	986.21	0.93
HUB	130.41	1.52	102.40	1047.67	1.14	978.38	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	43.05	49.65	45.80	3.85	0.93	0.51	1.19
MEAN	42.59	48.17	44.24	3.93	0.93	0.50	1.27
HUB	42.69	43.94	39.70	4.24	0.93	0.49	1.37

STAGE EXIT CONDITIONS, STAGE 4

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.876	130.176	1.521	1.144	148.354	1.150

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
840325.50	0.411	1348.141	567954.750

COMPRESSOR INLET CONDITIONS, STAGE 5

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	130.176	1055.544	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
4.57	19129.678	1.389	0.256	53.349	70.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 5

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.96	0.00	-0.02	798.75	-0.28	798.75	0.51
MEAN	5.70	0.00	-0.02	726.14	-0.25	726.14	0.47
HUB	5.43	0.00	-0.02	653.52	-0.23	653.52	0.42

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	60.63	54.10	6.53	1418.85	1628.47	108.87	1003.97	1.05
MEAN	61.85	55.60	6.25	1356.98	1539.27	112.20	1012.50	0.99
HUB	63.18	56.80	6.38	1292.16	1448.22	115.61	1021.01	0.92

ROTOR EXIT CONDITIONS, STAGE 5

B2 axial	THK	AeroBl
0.05	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.94	983.70	648.18	739.95	1634.40	0.60
MEAN	5.70	961.63	635.91	721.35	1630.91	0.59
HUB	5.46	931.93	562.69	742.88	1619.14	0.58

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1414.08	1064.96	765.90	0.65	
MEAN	1358.52	1021.04	722.61	0.63	1.56
HUB	1300.59	1047.07	737.90	0.65	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	191.71	1.47	150.33	1198.90	1.14	1119.92	0.84
MEAN	193.99	1.49	153.56	1190.66	1.13	1115.18	0.92
HUB	183.68	1.41	146.96	1170.00	1.11	1099.12	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	41.22	45.99	42.00	3.99	0.93	0.51	1.21
MEAN	41.40	45.05	41.00	4.05	0.93	0.50	1.27
HUB	37.14	44.81	41.00	3.81	0.93	0.42	1.33

STAGE EXIT CONDITIONS, STAGE 5

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.872	187.818	1.443	1.124	129.361	1.003

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
837881.94	0.419	1344.221	517435.875

COMPRESSOR INLET CONDITIONS, STAGE 6

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	187.818	1186.523	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
3.36	18042.957	1.383	0.259	53.349	80.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 6

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.91	0.00	-0.02	793.95	-0.27	793.95	0.48
MEAN	5.71	0.00	-0.02	721.77	-0.25	721.77	0.44
HUB	5.50	0.00	-0.02	649.60	-0.22	649.60	0.39

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	60.57	53.90	6.67	1407.11	1615.88	160.59	1136.17	0.98
MEAN	62.03	55.43	6.60	1359.21	1539.18	164.89	1144.50	0.93
HUB	63.62	57.10	6.52	1309.56	1462.03	169.25	1152.81	0.88

ROTOR EXIT CONDITIONS, STAGE 6

B2 axial	THK	AeroBl
0.05	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.90	981.22	654.68	730.87	1723.20	0.57
MEAN	5.71	951.16	626.87	715.37	1719.28	0.55
HUB	5.52	926.24	571.24	729.11	1711.00	0.54

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1404.58	1047.15	749.90	0.61	
MEAN	1360.01	1024.33	733.14	0.60	1.56
HUB	1313.92	1040.76	742.68	0.61	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	265.99	1.42	214.00	1328.46	1.12	1250.81	0.85
MEAN	267.36	1.42	217.66	1318.11	1.11	1245.14	0.93
HUB	257.59	1.37	211.50	1302.37	1.10	1233.18	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip	F. Diff Fct	Solidity
TIP	41.85	45.74	41.70	4.04	0.93	0.52	1.21
MEAN	41.23	45.70	41.70	4.00	0.93	0.50	1.25
HUB	38.08	45.53	41.70	3.83	0.93	0.44	1.30

STAGE EXIT CONDITIONS, STAGE 6

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.874	261.184	1.391	1.109	113.106	0.877

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
841334.13	0.426	1349.759	468731.500

COMPRESSOR INLET CONDITIONS, STAGE 7

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	261.184	1316.314	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
2.55	17130.342	1.376	0.263	53.349	82.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 7

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.87	0.00	-0.02	761.01	-0.26	761.01	0.44
MEAN	5.71	0.00	-0.02	691.83	-0.24	691.83	0.40
HUB	5.54	0.00	-0.02	622.65	-0.21	622.65	0.36

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	61.46	54.80	6.66	1398.96	1592.79	229.53	1270.68	0.92
MEAN	63.04	56.20	6.84	1359.76	1525.85	234.57	1278.23	0.88
HUB	64.74	57.90	6.84	1319.40	1459.13	239.67	1285.76	0.83

ROTOR EXIT CONDITIONS, STAGE 7

B2 axial	THK	AeroBl
0.05	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.87	963.51	659.92	702.04	1807.01	0.53
MEAN	5.71	938.35	639.43	686.75	1804.28	0.52
HUB	5.55	915.64	595.68	695.39	1798.28	0.51

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1397.06	1017.95	737.13	0.56	
MEAN	1359.94	995.37	720.51	0.55	1.58
HUB	1321.78	1005.38	726.10	0.56	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	358.21	1.37	296.07	1456.51	1.11	1382.65	0.85
MEAN	361.80	1.39	301.75	1448.54	1.10	1378.49	0.93
HUB	352.33	1.35	296.03	1436.04	1.09	1369.33	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip	F. Diff Fct	Solidity
TIP	43.23	46.40	42.30	4.10	0.93	0.53	1.22
MEAN	42.96	46.37	42.30	4.07	0.93	0.51	1.26
HUB	40.58	46.24	42.30	3.94	0.93	0.47	1.30

STAGE EXIT CONDITIONS, STAGE 7

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.876	354.474	1.357	1.099	98.583	0.764

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
860064.69	0.441	1379.809	427515.625

COMPRESSOR INLET CONDITIONS, STAGE 8

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.39	27289.000	354.474	1447.031	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
1.97	16338.298	1.368	0.267	53.349	84.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 8

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.84	0.00	-0.02	721.08	-0.25	721.08	0.39
MEAN	5.71	0.00	-0.02	655.53	-0.23	655.53	0.36
HUB	5.56	0.00	-0.02	589.98	-0.20	589.98	0.32

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	62.62	55.70	6.92	1391.89	1567.80	319.10	1406.66	0.86
MEAN	64.25	57.60	6.65	1358.91	1508.97	324.77	1413.34	0.83
HUB	66.00	59.00	7.00	1325.11	1450.70	330.49	1420.00	0.79

ROTOR EXIT CONDITIONS, STAGE 8

B2 axial	THK	AeroBl
0.05	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.84	933.48	648.48	671.46	1885.48	0.50
MEAN	5.71	912.48	632.67	657.54	1883.46	0.48
HUB	5.57	892.03	596.42	663.33	1878.86	0.47

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1390.20	1000.50	741.72	0.53	
MEAN	1358.65	979.50	725.98	0.52	1.59
HUB	1326.35	986.31	729.93	0.52	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	469.94	1.33	398.85	1581.94	1.09	1513.62	0.85
MEAN	475.61	1.34	406.42	1575.66	1.09	1510.38	0.93
HUB	466.31	1.32	400.91	1565.40	1.08	1503.02	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	44.00	47.85	43.80	4.05	0.93	0.53	1.21
MEAN	43.90	47.83	43.80	4.03	0.93	0.52	1.24
HUB	41.96	47.74	43.80	3.94	0.93	0.48	1.28

STAGE EXIT CONDITIONS, STAGE 8

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.877	467.192	1.318	1.088	88.784	0.688

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
851134.44	0.440	1365.482	393019.219

OVERALL EXIT CONDITIONS; ALL 8 STAGES

Del Enthalpy	GHP	EFFICIENCY	PR	TR
6755277.50	10837.5469	0.8032	31.7818	3.0353

Appendix D.—Axial-Centrifugal Compressor Data

Below are the axial-centrifugal compressor mean-line flow analysis results at the design point operating condition.

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    LCTR   7 Axial 1 Centrifugal Stage  100% Speed

       COMPRESSOR INLET CONDITIONS, STAGE            1
RESET =  0.000 BLEED =  0.000 DPInc 5.000 Kg/sec = 12.90909

     W act     RPM act        Pt          Tt         'POTS         POTH       AeroBl
        28.40   27289.000      14.613     518.670       1.100       0.800       0.980

     W cor     RPM cor        GAMMA        Cp         'R           NBLAD        THK
        28.40   27289.789       1.402       0.249      53.349      28.000       0.030

      ROTOR LEADING EDGE CONDITIONS, STAGE            1

          R1      Stator    Alfa     C1      CU1     Cm1     Abs MACH
  TIP      6.28     0.00    -0.02   636.97    -0.22   636.97     0.59
  MEAN     4.71     0.00    -0.02   579.07    -0.20   579.07     0.53
  HUB      2.25     0.00    -0.02   463.25    -0.16   463.25     0.42

        BetaFlo   BetaBlade Incid    U1      W1       Ps1      Ts1     Rel Mach
  TIP     66.93     62.00     4.93  1495.05  1625.29    11.57   485.12     1.50
  MEAN    62.72     57.56     5.16  1122.83  1263.54    12.04   490.66     1.16
  HUB     49.12     44.20     4.92   535.10   707.89    12.94   500.92     0.64

       ROTOR EXIT CONDITIONS, STAGE            1

     B2 axial    THK        AeroBl
     0.30       0.030       0.950

           R2      C2       Cu2      Cm2     Ao2      Mach2
  TIP      6.28    864.45   685.22   527.02  1222.52     0.71
  MEAN     5.14    870.86   667.80   558.97  1188.55     0.73
  HUB      3.67   1048.17   801.40   675.59  1140.89     0.92

        U2        W2        Wu2       Mach Rel2  Ws1/W2
  TIP   1495.05   966.22   809.83      0.79
  MEAN  1224.70   789.04   556.90      0.66     2.00
  HUB    874.41   679.52    73.00      0.60

          Pt2       PR       Ps2      Tt2       TR       Ts2        Eff2
  TIP     32.96     2.26     23.60    683.30     1.32    620.90     0.83
  MEAN    30.05     2.06     21.02    650.09     1.25    586.78     0.91
  HUB     27.51     1.88     15.93    631.26     1.22    539.70     0.92

          Alfa2  Beta FLO Beta BLADE Deviat   Slip F. Diff Fct Solidity
  TIP     52.44    56.95    53.20     3.75     0.93     0.57     1.32
  MEAN    50.07    44.89    40.10     4.79     0.93     0.53     1.75
  HUB     49.87     6.17     1.00     5.17     0.93     0.23     3.68

     STAGE EXIT CONDITIONS, STAGE            1

    DIFF LOSS       Effic      Pdisch       PR          TR         Ns          Ns nondim
         0.05       0.858      29.677      2.031       1.263     272.485       2.112

 Del Enthalpy   Del_H/U^2    GHP       Reynolds#
    848001.94       0.379    1360.967 1192642.500
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COMPRESSOR INLET CONDITIONS, STAGE 2

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	29.677	654.883	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
15.81	24286.434	1.401	0.249	53.349	38.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 2

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	6.10	0.00	-0.02	679.69	-0.23	679.69	0.56
MEAN	5.23	0.00	-0.02	617.90	-0.21	617.90	0.50
HUB	4.18	0.00	-0.02	556.11	-0.19	556.11	0.45

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	64.93	59.00	5.93	1452.66	1604.02	24.06	616.74	1.31
MEAN	63.62	57.42	6.20	1245.69	1390.71	24.93	623.05	1.14
HUB	60.84	54.50	6.34	996.62	1141.44	25.83	629.35	0.92

ROTOR EXIT CONDITIONS, STAGE 2

B2 axial	THK	AeroBl
0.20	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.98	865.82	633.85	589.80	1331.49	0.65
MEAN	5.26	873.23	644.05	589.68	1316.63	0.66
HUB	4.43	941.87	700.49	629.63	1297.19	0.73

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1424.09	986.07	790.23	0.74	
MEAN	1253.29	847.87	609.24	0.64	1.88
HUB	1055.20	722.67	354.71	0.56	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	53.91	1.82	40.57	799.56	1.22	737.07	0.84
MEAN	53.30	1.80	39.67	784.25	1.20	720.69	0.92
HUB	51.19	1.72	36.04	773.34	1.18	699.41	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	47.06	53.26	49.47	3.79	0.93	0.54	1.27
MEAN	47.52	45.93	41.46	4.47	0.93	0.55	1.48
HUB	48.05	29.40	24.00	5.40	0.93	0.54	1.85

STAGE EXIT CONDITIONS, STAGE 2

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.875	52.097	1.755	1.200	219.481	1.701

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
816707.63	0.403	1310.742	812599.375

COMPRESSOR INLET CONDITIONS, STAGE 3

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	52.097	785.718	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
9.86	22172.389	1.399	0.251	53.349	50.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 3

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.79	0.00	-0.02	729.10	-0.25	729.10	0.54
MEAN	5.22	0.00	-0.02	662.81	-0.23	662.81	0.49
HUB	4.59	0.00	-0.02	596.53	-0.21	596.53	0.44

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	62.14	56.00	6.14	1378.84	1559.96	42.62	742.01	1.16
MEAN	61.96	55.70	6.26	1244.19	1409.93	44.09	749.24	1.05
HUB	61.38	55.00	6.38	1093.07	1245.43	45.60	756.46	0.92

ROTOR EXIT CONDITIONS, STAGE 3

B2 axial	THK	AeroBl
0.15	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.71	868.09	609.75	617.89	1433.13	0.61
MEAN	5.19	855.65	596.31	613.63	1422.29	0.60
HUB	4.61	895.51	633.67	632.78	1411.88	0.63

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1359.79	971.77	750.04	0.68	
MEAN	1236.09	886.48	639.78	0.62	1.77
HUB	1098.55	785.19	464.88	0.56	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	83.00	1.59	64.79	917.93	1.17	855.37	0.84
MEAN	82.05	1.57	64.26	903.25	1.15	842.47	0.92
HUB	80.47	1.54	61.39	896.71	1.14	830.13	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	44.62	50.52	46.64	3.88	0.93	0.54	1.22
MEAN	44.18	46.19	42.01	4.18	0.93	0.53	1.36
HUB	45.04	36.30	31.47	4.83	0.93	0.53	1.54

STAGE EXIT CONDITIONS, STAGE 3

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.874	80.924	1.553	1.153	192.065	1.489

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
754495.94	0.408	1210.898	703666.375

COMPRESSOR INLET CONDITIONS, STAGE 4

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	80.924	905.963	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
6.82	20648.596	1.395	0.252	53.349	60.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 4

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.54	0.00	-0.02	697.68	-0.24	697.68	0.48
MEAN	5.09	0.00	-0.02	634.26	-0.22	634.25	0.44
HUB	4.61	0.00	-0.02	570.83	-0.20	570.83	0.39

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	62.12	56.00	6.12	1318.83	1492.21	69.06	866.19	1.03
MEAN	62.40	56.20	6.20	1213.05	1369.05	70.93	872.77	0.95
HUB	62.52	57.00	5.52	1097.12	1236.91	72.84	879.34	0.85

ROTOR EXIT CONDITIONS, STAGE 4

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.44	863.90	584.23	636.39	1519.69	0.57
MEAN	5.03	852.54	575.81	628.70	1512.51	0.56
HUB	4.58	879.96	603.22	640.68	1505.43	0.58

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1295.49	954.40	711.26	0.63	
MEAN	1197.27	884.01	621.46	0.58	1.69
HUB	1090.24	804.77	487.02	0.53	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	117.54	1.45	94.47	1025.79	1.13	964.21	0.84
MEAN	117.44	1.45	94.72	1015.10	1.12	955.13	0.92
HUB	115.96	1.43	92.08	1010.07	1.11	946.18	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	42.55	48.18	44.26	3.92	0.93	0.52	1.19
MEAN	42.49	44.67	40.51	4.16	0.93	0.52	1.29
HUB	43.28	37.24	32.62	4.62	0.93	0.52	1.43

STAGE EXIT CONDITIONS, STAGE 4

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.872	115.820	1.431	1.123	172.586	1.338

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
701661.88	0.418	1126.104	637040.500

COMPRESSOR INLET CONDITIONS, STAGE 5

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	115.820	1016.988	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
5.05	19488.920	1.391	0.255	53.349	70.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 5

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	5.24	0.00	-0.02	690.70	-0.24	690.70	0.45
MEAN	4.88	0.00	-0.02	627.91	-0.22	627.91	0.41
HUB	4.48	0.00	-0.02	565.12	-0.19	565.12	0.37

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	61.04	54.80	6.24	1247.86	1426.47	100.90	978.31	0.93
MEAN	61.60	55.20	6.40	1161.06	1320.17	103.26	984.70	0.86
HUB	62.10	55.50	6.60	1067.23	1207.79	105.66	991.09	0.78

ROTOR EXIT CONDITIONS, STAGE 5

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	5.14	849.48	575.31	625.00	1597.25	0.53
MEAN	4.78	844.07	576.55	616.48	1592.18	0.53
HUB	4.39	854.35	579.87	627.43	1585.22	0.54

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1224.05	900.82	648.73	0.56	
MEAN	1138.39	834.09	561.85	0.52	1.70
HUB	1045.75	781.48	465.88	0.49	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	158.30	1.37	130.71	1127.51	1.11	1068.43	0.84
MEAN	159.28	1.38	131.68	1120.00	1.10	1061.66	0.93
HUB	156.10	1.35	128.25	1112.16	1.09	1052.39	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	42.63	46.07	42.00	4.07	0.93	0.53	1.21
MEAN	43.08	42.35	38.00	4.35	0.93	0.53	1.30
HUB	42.74	36.59	32.00	4.59	0.93	0.52	1.42

STAGE EXIT CONDITIONS, STAGE 5

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.873	156.506	1.351	1.101	159.814	1.239

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
655986.81	0.438	1052.800	604502.250

COMPRESSOR INLET CONDITIONS, STAGE 6

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	156.506	1119.889	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
3.92	18571.986	1.386	0.257	53.349	80.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 6

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	4.91	0.00	-0.02	703.82	-0.24	703.82	0.44
MEAN	4.60	0.00	-0.02	639.84	-0.22	639.84	0.40
HUB	4.27	0.00	-0.02	575.85	-0.20	575.85	0.36

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	58.96	52.20	6.76	1169.27	1364.97	137.44	1080.07	0.85
MEAN	59.73	53.20	6.53	1096.05	1269.33	140.47	1086.66	0.79
HUB	60.50	54.20	6.30	1017.58	1169.39	143.54	1093.23	0.72

ROTOR EXIT CONDITIONS, STAGE 6

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	4.83	845.97	548.72	643.88	1661.44	0.51
MEAN	4.52	833.86	537.94	637.13	1656.75	0.50
HUB	4.18	864.62	579.20	641.95	1653.46	0.52

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1151.18	881.78	602.46	0.53	
MEAN	1076.35	834.15	538.40	0.50	1.65
HUB	995.91	765.34	416.70	0.46	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	202.28	1.29	169.75	1218.08	1.09	1159.98	0.85
MEAN	202.48	1.29	170.58	1209.89	1.08	1153.45	0.93
HUB	202.83	1.30	168.62	1209.54	1.08	1148.86	0.94

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip	F. Diff Fct	Solidity
TIP	40.44	43.10	39.00	4.10	0.93	0.52	1.21
MEAN	40.18	40.20	36.00	4.20	0.93	0.51	1.29
HUB	42.06	32.99	28.36	4.63	0.93	0.52	1.39

STAGE EXIT CONDITIONS, STAGE 6

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.871	200.887	1.284	1.083	154.647	1.199

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
596158.88	0.450	956.781	593802.938

COMPRESSOR INLET CONDITIONS, STAGE 7

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	200.887	1212.503	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
3.18	17848.609	1.382	0.260	53.349	82.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 7

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	4.65	0.00	-0.02	683.27	-0.24	683.27	0.41
MEAN	4.37	0.00	-0.02	621.16	-0.21	621.16	0.37
HUB	4.07	0.00	-0.02	559.04	-0.19	559.04	0.33

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	58.31	52.10	6.21	1106.55	1300.70	179.46	1175.31	0.78
MEAN	59.16	53.00	6.16	1040.27	1211.79	182.88	1181.46	0.72
HUB	60.04	53.80	6.24	969.47	1119.28	186.35	1187.61	0.66

ROTOR EXIT CONDITIONS, STAGE 7

B2 axial	THK	AeroBl
0.10	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	4.58	826.13	547.75	618.44	1721.39	0.48
MEAN	4.28	813.61	535.13	612.86	1717.06	0.47
HUB	3.97	836.11	563.38	617.81	1713.53	0.49

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1090.45	822.79	542.70	0.48	
MEAN	1020.07	781.52	484.95	0.46	1.67
HUB	944.47	725.89	381.09	0.42	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	251.41	1.25	215.16	1304.43	1.08	1249.53	0.84
MEAN	251.46	1.25	216.03	1296.52	1.07	1243.27	0.92
HUB	250.66	1.25	213.42	1294.39	1.07	1238.15	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	41.53	41.27	37.00	4.27	0.93	0.54	1.22
MEAN	41.13	38.35	34.00	4.35	0.93	0.52	1.30
HUB	42.36	31.67	27.00	4.67	0.93	0.53	1.39

STAGE EXIT CONDITIONS, STAGE 7

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.05	0.868	249.360	1.241	1.071	148.361	1.150

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
558713.00	0.470	896.684	584503.813

COMPRESSOR INLET CONDITIONS, STAGE 8

W act	RPM act	Pt	Tt	'POTS	POTH	AeroBl
28.40	27289.000	249.360	1298.448	1.100	0.900	0.980

W cor	RPM cor	GAMMA	Cp	'R	NBLAD	THK
2.65	17247.789	1.377	0.262	53.349	24.000	0.030

ROTOR LEADING EDGE CONDITIONS, STAGE 8

	R1	Stator	Alfa	C1	CU1	Cm1	Abs MACH
TIP	4.10	0.00	-0.02	562.70	-0.19	562.70	0.32
MEAN	3.81	0.00	-0.02	511.55	-0.18	511.55	0.29
HUB	3.50	0.00	-0.02	460.39	-0.16	460.39	0.26

	BetaFlo	BetaBlade	Incid	U1	W1	Ps1	Ts1	Rel Mach
TIP	60.05	54.00	6.05	976.38	1127.09	232.26	1273.45	0.65
MEAN	60.60	54.50	6.10	907.75	1042.12	235.03	1277.59	0.60
HUB	61.09	55.00	6.09	833.50	952.33	237.81	1281.71	0.54

ROTOR EXIT CONDITIONS, STAGE 8

B2 axial	THK	AeroBl
0.28	0.030	0.950

	R2	C2	Cu2	Cm2	Ao2	Mach2
TIP	6.70	1286.38	1137.24	601.22	1846.69	0.70
MEAN	6.70	1281.72	1149.70	566.57	1849.22	0.69
HUB	6.70	1281.17	1151.12	562.42	1849.51	0.69

	U2	W2	Wu2	Mach Rel2	Ws1/W2
TIP	1595.55	755.98	458.31	0.41	
MEAN	1595.55	720.96	445.85	0.39	1.54
HUB	1595.55	716.82	444.42	0.39	

	Pt2	PR	Ps2	Tt2	TR	Ts2	Eff2
TIP	455.63	1.83	330.98	1574.88	1.21	1442.97	0.84
MEAN	483.37	1.94	352.20	1577.91	1.22	1446.95	0.92
HUB	486.92	1.95	354.91	1578.25	1.22	1447.40	0.93

	Alfa2	Beta FLO	Beta BLADE	Deviat	Slip F.	Diff Fct	Solidity
TIP	62.14	37.32	20.00	17.32	0.85	0.75	1.50
MEAN	63.77	38.20	20.00	18.20	0.85	0.74	1.61
HUB	63.96	38.32	20.00	18.32	0.85	0.70	1.76

STAGE EXIT CONDITIONS, STAGE 8

DIFF LOSS	Effic	Pdisch	PR	TR	Ns	Ns nondim
0.17	0.828	453.331	1.818	1.215	55.924	0.434

Del Enthalpy	Del_H/U^2	GHP	Reynolds#
1828923.63	0.718	2935.257	554689.563

OVERALL EXIT CONDITIONS; ALL 8 STAGES

Del Enthalpy	GHP	EFFICIENCY	PR	TR
6760649.50	10850.2324	0.7961	30.8389	3.0405

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<tr><td colspan="6">14. ABSTRACT
A vehicle concept study has been made to meet the requirements of the Large Civil Tilt Rotorcraft vehicle mission. A vehicle concept was determined, and a notional turboshaft engine system study was conducted. The engine study defined requirements for the major engine components, including the compressor. The compressor design-point goal was to deliver a pressure ratio of 31:1 at an inlet weight flow of 28.4 lbm/sec. To perform a conceptual design of two potential compressor configurations to meet the design requirement, a mean-line compressor flow analysis and design code were used. The first configuration is an eight-stage axial compressor. Some challenges of the all-axial compressor are the small blade spans of the rear-block stages being 0.28 in., resulting in the last-stage blade tip clearance-to-span ratio of 2.4 percent. The second configuration is a seven-stage axial compressor, with a centrifugal stage having a 0.28-in. impeller-exit blade span. The compressors' conceptual designs helped estimate the flow path dimensions, rotor leading and trailing edge blade angles, flow conditions, and velocity triangles for each stage.</td></tr>
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