Conceptual Design of a Two Spool Compressor for the NASA Large Civil Tilt Rotor Engine

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ABSTRACT

This paper focuses on the conceptual design of a two spool compressor for the NASA Large Civil Tilt Rotor engine, which has a design-point pressure ratio goal of 30:1 and an inlet weight flow of 30.0 lbm/sec. The compressor notional design requirements of pressure ratio and low-pressure compressor (LPC) and high pressure ratio compressor (HPC) work split were based on a previous engine system study to meet the mission requirements of the NASA Subsonic Rotary Wing Projects Large Civil Tilt Rotor vehicle concept. Three mean line compressor design and flow analysis codes were utilized for the conceptual design of a two-spool compressor configuration. This study assesses the technical challenges of design for various compressor configuration options to meet the given engine cycle results. In the process of sizing, the technical challenges of the compressor became apparent as the aerodynamics were taken into consideration. Mechanical constraints were considered in the study such as maximum rotor tip speeds and conceptual sizing of rotor disks and shafts. The rotor clearance-to-span ratio in the last stage of the LPC is 1.5% and in the last stage of the HPC is 2.8%. Four different configurations to meet the HPC requirements were studied, ranging from a single stage centrifugal, two axi-centrifugals, and all axial stages. Challenges of the HPC design include the high temperature (1,560 °R) at the exit which could limit the maximum allowable peripheral tip speed for centrifugals, and is dependent on material selection. The mean line design also resulted in the definition of the flow path geometry of the axial and centrifugal compressor stages, rotor and stator vane angles, velocity components, and flow conditions at the leading and trailing edges of each blade row at the hub, mean and tip. A mean line compressor analysis code was used to estimate the compressor performance maps at off-design speeds and to determine the required variable geometry reset schedules of the inlet guide vane and variable stators that would result in the transonic stages being aerodynamically matched with high efficiency and acceptable stall margins based on user specified maximum levels of rotor diffusion factor and relative velocity ratio.

INTRODUCTION

Engine System Study and Compression System Requirements

A study of the notional Large Civil Tilt Rotor (LCTR-2) vehicle mission, references 1 and 2, identified the vehicle thrust requirements at the takeoff and cruise conditions, for which a thermodynamic cycle study of a notional three spool turboshaft engine for LCTR-2 was performed in reference 3 with the Numerical Propulsion System Simulator (NPSS) thermodynamic cycle code of reference 4. The results of the LCTR-2 engine system study are illustrated by the schematic diagram in Figure 1. The focus of this study reported herein is to perform a conceptual sizing of the two-spool compressor to meet the pressure ratio and flow requirements of the turboshaft engine gas generator for the LCTR-2 vehicle. Benefits of the two-spool gas generator over a single-spool configuration from an engine system perspective are noted in reference 3, while additional benefits are outlined in this paper. The engine system model study determined the specifications for the compressor design point to be at a flow rate of 30.0 lbm/sec and overall pressure ratio of 30:1. The pressure ratio was produced by two spools: a low pressure compressor (LPC) and a high pressure compressor (HPC), where the work split was based on minimizing the number of turbine stages required to drive each spool. The work split between the LPC and the HPC was determined a priori by the engine system model with the NPSS code with only slight variation from that during this conceptual design study of the HPC and LPC with the mean line compressor codes. Note that the turbine also has to be assessed for feasibility to meet the cycle requirements, but has not yet been addressed in detail beyond the WATE code assessments.

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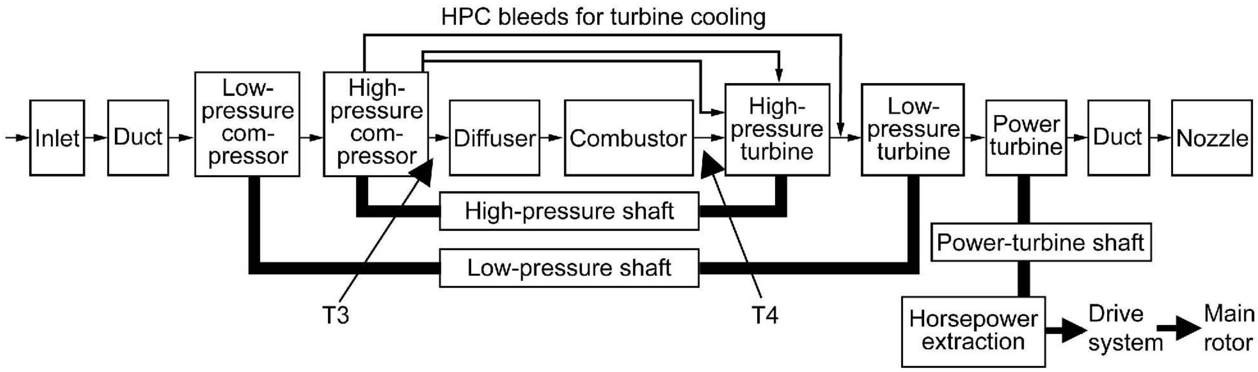

Figure 1. Thermodynamic cycle study of the three-spool engine (two-spool gas generator) with NPSS.

The engine flow path was created with the NASA software WATE (Weight Analysis of Turbine Engines) (ref. 5), which uses the thermodynamic data from NPSS to provide information such as a flow path schematic, an estimate of the weight of the engine and individual components, and stress analysis of rotating components such as blades and disks. The maximum performance conditions for each engine component were determined from the aerodynamic design point and off-design cases. This data, along with material properties and the design rules for stress and stage-loading limits, were used to determine an acceptable engine layout and weight, and provided an estimate for the shaft sizes required to transmit the torque through the three spools of the engine.

Compressor Conceptual Design Objectives

There are four goals for the compressor conceptual design effort. The first goal is to give an early indication of where the technical challenges may exist for a 30:1 pressure ratio two spool compressor at the low flow rate of 30 lbm/sec. Relative to that assumed for the engine system studies, the conceptual design effort is anticipated to provide an improved estimate of the overall compressor aerodynamic performance as well as insight into potential barriers that will need to be considered during the preliminary and detailed design phases. The second goal is to create an improved estimate of the compressor performance characteristic maps at off-design operating conditions to update the initial maps utilized in the NPSS system model. This task will include determining how many stages will require variable geometry and defining the variable geometry schedule that will provide an acceptable compressor operating line free of choke and surge at all anticipated off-design speeds. The other important consideration while designing the variable geometry schedule is to assure the compressor achieves as high a level of efficiency as possible along the defined operating line. These will be done by aerodynamically matching each stage along the operating line by adjusting the setting angles of the variable inlet guide vane and stators. Although the engine operating line between the takeoff condition and the cruise condition only moves from 100% down to 93% of design speed, it is important to have acceptable aerodynamic matching of the stages during startup and idle as well. The resulting compressor characteristic maps will be provided to the NPSS system model that is utilized to generate the engine performance characteristics throughout the vehicle flight envelope. The third goal of the compressor conceptual design effort is to provide an improved estimate of the compressor size and weight by providing the compressor flow path coordinates and stage geometry (i.e., aspect ratios, chords, solidities) resulting from the conceptual design effort as input into the weight estimation code (WATE). The fourth goal is to document the procedure and methodologies for performing the LCTR-2 compressor conceptual studies and evaluating various design configuration options, as well as the map creation process for LPC – HPC configurations featuring variable geometry.

Design Codes and Conceptual Design Process

The conceptual design process started with obtaining the overall performance requirements of pressure ratio and mass flow for the compressor as modeled by the NPSS code. The NPSS code defined the overall pressure ratio and mass flow requirements at the sea level takeoff condition, as well as the LPC – HPC work split. These were used as the design point for the compressors. The information was passed to the WATE code to estimate shaft diameters for the low pressure spool, high pressure spool and the power turbine spool, as well as compressor disk space requirements. The WATE code was used iteratively to determine the appropriate shaft and disk sizes for the resulting compression system.

The conceptual design and flow analysis codes to enable sizing the LPC and the HPC of the LCTR-2 engine consist of three mean line codes with specialized capabilities (COMDES, TCDES, QUIK, refs. 6, 7, and 8), as well as a through flow code (T-AXI, ref. 7). The COMDES, TCDES, T-AXI and QUIK codes are utilized iteratively to produce the flow path geometry, as well as rotor blade and stator vane angles. This is a key part of the conceptual design process as the shafts and compressor disks impose a physical size limit on the compressor flow path hub dimensions. The compressor conceptual design was executed utilizing the three mean line compressor flow analysis and design codes and focuses on the compressor flow path and key

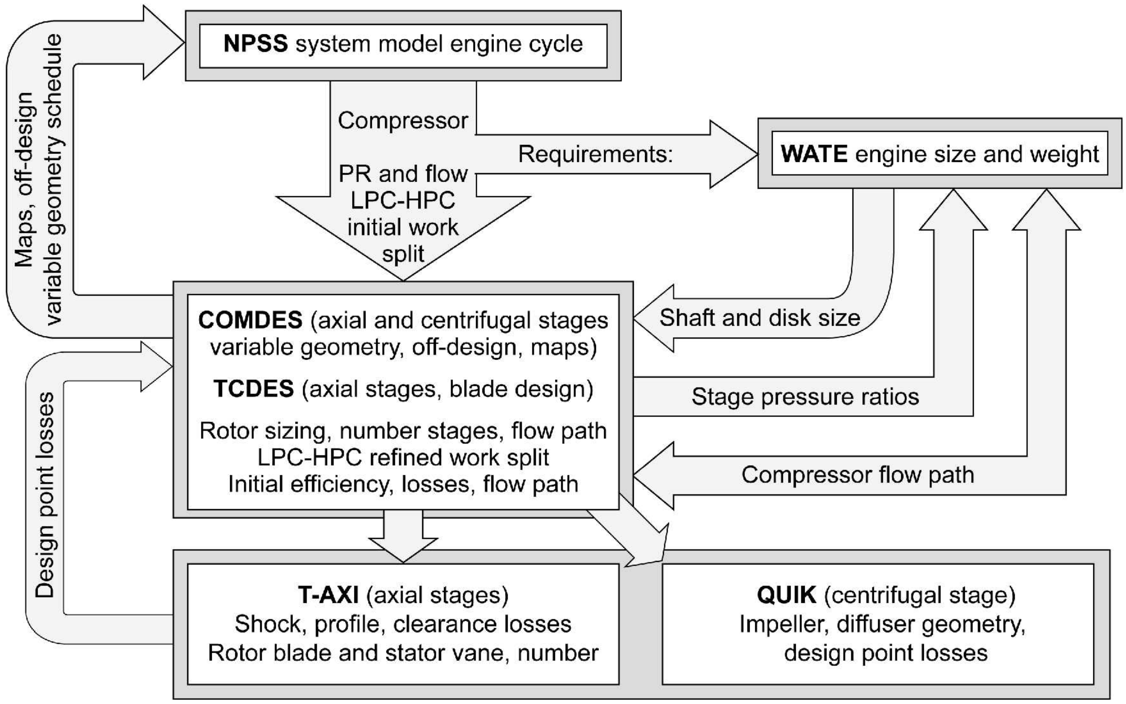

Figure 2. Conceptual design process for a two-spool axi-centrifugal compressor.

aerodynamic parameters of the rotor, stator and stage at the design point condition. The following schematic diagram of Figure 2 outlines the design procedure that was followed during the conceptual design process.

The COMDES compressor mean line flow code was utilized first to size the flow path by means of sizing the inlet and exit areas of the axial and the centrifugal rotors, with an initial estimate (based on user experience of similar components) for the rotor and stator losses and the centrifugal compressor impeller and diffusion system. The design point work input in terms of pressure ratio for each axial rotor was determined by assuming a maximum diffusion factor on the order of 0.50, with the blade solidity ranging from 1.28 to 1.46 that was specified at each blade tip. The purpose of utilizing a maximum diffusion factor limit was to reduce the likelihood of flow separation at the design condition, that is, to enable adequate surge margin. An iterative process was used that required variation of key design parameters to stay within the limits of maximum diffusion factor per stage. In addition, the level of relative velocity ratio for the rotors was limited to no higher than 1.91 at the design condition.

The shaft rotational speeds for the axial rotors in the low pressure compressor were determined from historical compressor experience where the maximum allowable tip speed of 1,500 ft/sec is set due to anticipated structural limits of compressor materials at standard inlet conditions. In addition, the axial rotor maximum blade tip speed limit was also driven by aerodynamic concerns (e.g., shock losses) which were set to limit rotor-inlet tip relative Mach number to less than 1.49. The tip speeds of the subsequent axial rotors would thus be reduced where a tapered tip flow path is used. However, the tip speed of the centrifugal impeller in the high pressure compressor last stage was sized with consideration for the limits of the maximum allowable tip speed for a range of operating temperatures for selected impeller materials. In one case the maximum allowable tip speed of the centrifugal at exit temperature conditions was used to set the HPC shaft speed. The absolute flow angle entering each rotor was arbitrarily set at 0° at the design point operating condition. The work distribution between the axial stages, as defined by work coefficient, was set to be nearly uniform, while the work of the centrifugal stage is typically higher by design intent.

The axial rotors were sized to have a nominal inlet absolute Mach number of 0.50, while in the latter stages of the high pressure compressor this limit was reduced in an effort to keep the blade spans preferably no smaller than 0.50 in. for manufacturing concerns and to minimize blade tip leakage and its associated parasitic losses. The stator loss coefficients utilized in the sizing study were determined from correlations to diffusion factor loading levels (ref. 6).

During the initial sizing process, an initial compressor design was obtained iteratively based on user assessment of an aesthetically acceptable flow path using the COMDES code. The reason for using the COMDES code initially was its ability to size an arbitrary flow path through both axial and centrifugal compressors. The blade solidity and the work distribution per stage was specified, the velocity triangles and other key aerodynamic and blade parameters were determine iteratively by concurrent assessment of the flow path coordinates with this mean line code. The analysis code was run with the work specified as an input item in terms of rotor exit blade angles per stage and the resulting pressure ratio and temperature rise determined as output items of the analysis. In this fashion, each stage of the multistage compressor was designed in sequence.

The resulting flow path and stage parameters of temperature rise per stage, blade solidity, inlet Mach numbers and rotor and stator aspect ratios and spacing coefficients, which are a percentage multiplier of the resulting axial blade chord were next input into the TCDES code to determine blade numbers and to provide a refined estimate of the flow path axial and radial coordinates. Note that initial guesses for the losses, aerodynamic blockages and deviation were input into both the COMDES and the TCDES codes, as in this phase of the conceptual design these codes were primarily utilized to provide a reasonably accurate estimate of the flow path and blade geometry.

The next step was to estimate the design point axial compressor efficiency and stator losses as well as overall compressor pressure ratio and efficiency with the use of the T-AXI through flow analysis code. This code has loss models for shock, rotor tip clearance and diffusion factor profile losses for axial rotors and stators. The TXSET code, a part of the TCDES/T-AXI package of codes, was used to generate the computational grid utilized by T-AXI. Using the values estimated by the T-AXI code, the initial losses for the axial rotor and stator utilized in COMDES and TCDES were updated and the codes were once again executed until a reasonable convergence between the output results was achieved in all three flow codes.

The resulting finalized flow path dimensions which were iteratively obtained from running the COMDES and TCDES codes were drawn with the use of a computer aided design code to assist in updating the final flow path sketch. The flow path sketch was also iterative, since there was an initial sketch made based on earlier design iterations with the COMDES code.

In the HPC configurations which utilized a centrifugal compressor, the initial sizing was done with the COMDES code using initial values of losses for the rotor and radial diffuser. The losses were then estimated by running the QUIK code (ref. 8) in the analysis mode on the resulting centrifugal stage geometry and determining impeller and diffuser losses based on extensive loss models within the code. Since the centrifugal stage has the capability to do more work than an axial, it has a higher limit on diffusion factor, typically in the range of from 0.75 to 0.85. Using the value for efficiency and loss estimates obtained from QUIK, the COMDES rotor and diffuser loss estimates were updated. The work through the centrifugal compressor in terms of pressure ratio was sized with a goal of obtaining a more nearly optimal specific speed which is between 0.60 to 0.90 as will be shown later in the report. One limitation of pressure rise capability for centrifugal compressors is the structural limitation on peripheral tip speed at the high operating temperatures, and is a strong function of the selected impeller material.

Utilizing the aforementioned codes and methodology, the aerodynamic parameters for each individual rotor and stator blade row were calculated at the leading edge and trailing edge (hub, mean, and tip) to determine the number of stages that would be required to produce the required overall pressure ratio. The compressor off-design analysis and map generation, including variable geometry resets, were accomplished with the COMDES mean line code for the axial compressor stages, which was previously utilized to model the characteristic maps from representative tested axial compressors in reference 6.

Low Pressure Compressor (LPC) Conceptual Design

The focus was to design an all axial low pressure compressor (LPC) to meet the flow and pressure ratio requirements from the engine cycle study. The flow rate of 30 lbm/sec, the maximum rotor tip speed criteria of 1,500 ft/sec, the hub-to-tip radius ratio of 0.48 and the inlet absolute Mach number of 0.5 were used to size the LPC first-stage axial compressor. The radius ratio limit was determined iteratively from COMDES and WATE due to the physical limitation of the rotor drum, or disk size and the sizes of the power turbine shaft, the low pressure shaft and high pressure compressor shaft. These shaft dimensions were taken from the engine system study and modeling with the WATE code as part of this report. The design criteria resulted in a shaft speed of 24,800 rotations per minute. Using the design process previously described resulted in a six stage axial compressor producing a pressure ratio of 10.83:1 at 84.2% adiabatic efficiency and an aesthetically pleasing flow path, as shown in Figure 3.

The T-AXI code was utilized next to provide through flow code level of analysis of the flow field through the axial stages of the LPC and to provide an estimate of the losses through the rotors and stators. The value of rotor tip clearance that was utilized was 0.012 in. and is based on the traditional practice of assuming a running clearance on the order of 0.001 in. of radial clearance per inch of rotor diameter. The clearance was specified as the average ratio of clearance gap to inch of rotor tip radius for the 6 stages. The variation of rotor tip clearance gap to blade span ratio is from 0.3% at the LPC first stage, to 1.5% at the sixth stage. These levels of clearance are at typical levels for traditional compressors. The computational grid for the T-AXI code was generated by the TXSET code (companion code to TCDES) and is shown in Figure 4, and includes the tip clearance gap.

There is a small difference between the flow path generated by the TCDES/T-AXI codes compared to the final flow path, due to the requirements in T-AXI that the flow be axial at the first and last stages. Nevertheless, the flow

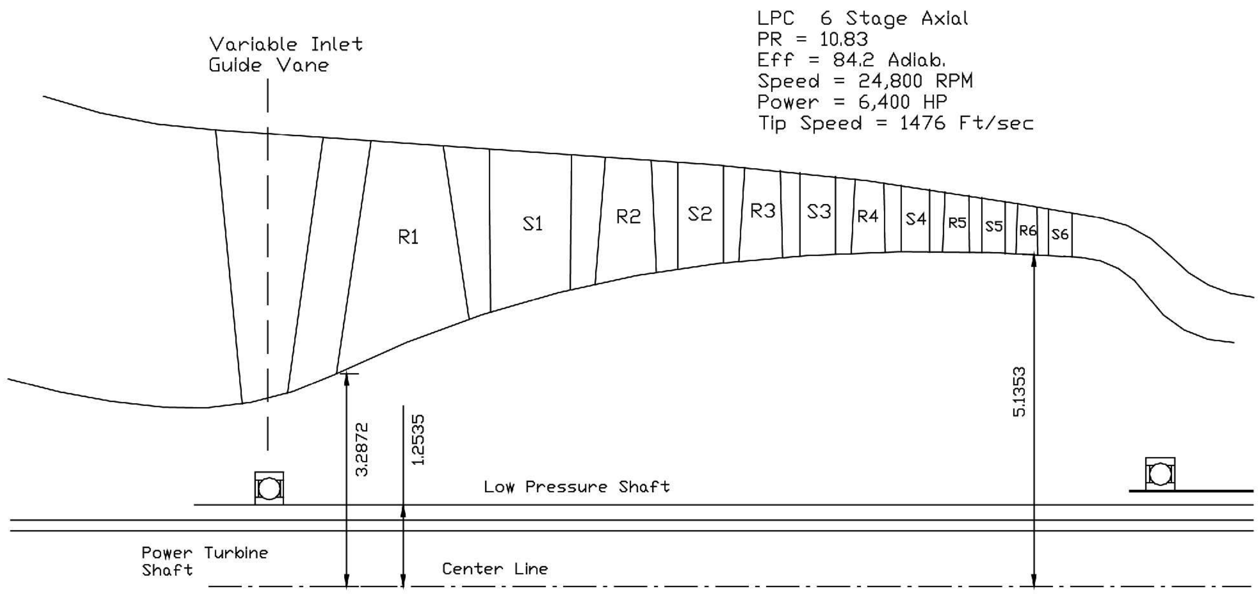

Figure 3. Six axial stage low-pressure compressor flow path illustrating rotors (R) and stators (S).

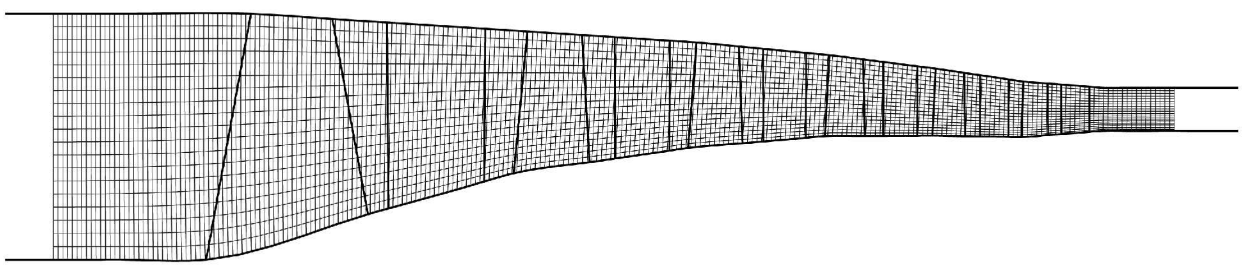

Figure 4. Six axial stage low pressure compressor grid for T-AXI analysis.

analysis results are for the most part quite consistent between the three flow codes (COMDES, TCDES, T-AXI). The complete design point mean line analysis results of the LPC with the T-AXI analysis code are listed in reference 10, and also summarized in Table 1.

The six axial stage LPC will produce an overall pressure ratio of 10.83 at an adiabatic efficiency of 84.2% at the design corrected flow of 30.0 lbm/sec and 100% corrected shaft speed of 24,800 rpm. The first 4 stages are transonic with inlet tip Mach numbers of 1.49, 1.33, 1.16, 1.05 at the first, second, third and fourth stages, respectively. Stage 1 was loaded slightly less in terms of diffusion factor, since the tip relative Mach number is the highest there. The result of this conceptual study is that this level of performance is achievable in a six stage LPC.

High Pressure Compressor Spool Conceptual Design

The goal of the HPC was to produce a pressure ratio of 2.77:1 in order to achieve the required overall pressure ratio of 30:1 for the two spools per the engine cycle optimization. The actual flow rate into the HPC is 30.0 lbm/sec, Because of the elevated temperatures and pressures from the LPC spool the corrected flow rate at the inlet to the HPC is 4.06 lbm/sec. One of the advantages of having a two spool compressor over the single spool case study in reference 10 is that the hub-to-tip ratio of the HPC stages can be reduced while still maintaining the ability to have a high work coefficient, or high pressure ratio per stage. A further advantage is the larger blade span and reduced clearance-to-span ratio of the HPC stages. As a consequence, the rotor tip leakage losses are reduced, and an increase in efficiency can be realized. The two spool concept also permitted the investigation of whether the specific speed of the centrifugal compressor stage can be varied to maximize the efficiency potential.

To produce the 2.77:1 pressure ratio required by the HPC, four possible configuration options were studied. Due to the elevated exit temperature, material selection for the rotor is critical as it limits the maximum physical speed, and therefore optimum specific speed, of the centrifugal impeller. A notional tip speed limit curves for an impeller.

Table 1. LPC Compression System Stage Performance.

	1	2	3	4	5	6
Rotor inlet						
Flow rate, corrected, lbm/sec	30.00	18.04	11.81	8.38	6.35	5.01
Mach, absolute	0.52	0.51	0.42	0.38	0.35	0.34
Mach, relative at tip	1.49	1.32	1.17	1.05	0.97	0.90
Tip speed, ft/sec	1493.	1438.	1404.	1364.	1322.	1282.
Flow angle, relative, deg	64.3	63.7	66.1	67.0	67.0	66.3
Blade angle, deg	58.1	57.6	60.0	60.8	60.9	58.5
Rotor blade number	22	34	42	48	58	72
Rotor exit						
Blade angle, deg	48.8	48.8	52.1	53.3	52.3	50.6
Flow angle, absolute, deg	46.4	46.1	45.9	43.5	42.4	40.4
Deviation angle, deg	3.8	3.8	3.5	3.3	3.3	3.4
Diffusion factor	0.47	0.49	0.48	0.45	0.45	0.43
Relative velocity ratio	1.91	1.81	1.72	1.63	1.58	1.54
Tip speed, ft/sec	1476.	1427.	1393.	1352.	1310.	1271.
Stator						
Vane number	40	68	83	100	123	128
Diffusion factor	0.42	0.51	0.50	0.46	0.42	0.42
Stage						
Pressure ratio	1.833	1.658	1.505	1.393	1.326	1.282
Temperature ratio	1.215	1.179	1.141	1.113	1.095	1.083
Exit temperature, °R	630.0	743.0	848.1	943.8	1033.6	1119.4
Temperature rise, °R	111.3	113.0	105.0	95.7	89.8	85.8
Efficiency, adiabatic	88.4	86.9	87.6	87.5	87.3	86.5
Work coefficient	0.318	0.346	0.339	0.330	0.332	0.339
Horsepower	1175.	1195.	1115.	1022.	964.6	928.0

made from forged Inconel 718, as well as more advanced high Nickel content high strength material were used as a guide to limit the maximum impeller tip speed as a function of exit temperature. The actual maximum allowable tip speed at the elevated impeller exit temperatures would be determined after final material selection with detailed aero-structural-thermal analyses of the final impeller blades and disk. However, a final design of any portion of this two-spool compressor is out of the scope of this current study.

The first three options utilized a centrifugal stage as the last stage with varying number of axial stages supercharging it. The effect of supercharging the centrifugal impeller on specific speed and efficiency is noted. In addition, the overall efficiency of the HPC as a function of the number of axial stages supercharging it was noted. The fourth HPC option that was studied eliminated the centrifugal stage for consideration of an all-axial four stage compressor. The detailed results for the four HPC options are listed in reference 10.

HPC Configuration Flow Path Sizing Option #1: Single Centrifugal Stage

The first option assumes there are no axial stages supercharging the centrifugal, that is, only a single centrifugal stage will produce the 2.77:1 pressure ratio at a corrected mass flow rate of 4.06 lbm/sec. Two different single stage centrifugal cases were sized under this option, both running at the same tip speed of 1,982 ft/sec, but at different shaft speeds of 38, 620, and 45,000 rpm. The key design features of the two centrifugals are listed in Table 2.

Table 2. Two different single stage centrifugals to meet the HPC 2.77:1 pressure ratio.

	Centrifugal 1	Centrifugal 2
Shaft rpm	38,620.	45,000.
Impeller tip radius (inch)	5.88	5.06
Impeller tip speed (ft/sec)	1,982.	1,987.
Impeller exit height (inch)	0.380	0.410
Inlet hub-to-tip radius ratio	0.742	0.742
Specific speed	0.55	0.65
Stage efficiency	83.4	85.1
Power (Horsepower)	4,805.	4,716.

Note that the physical tip speed of near 1,980 ft/sec is considered excessively high for traditional rotor materials such as Inconel 718 at nearly 1,560 °R exit temperature, but may be achievable by other advanced high strength Nickel based materials developed for high strength at elevated temperatures.

The lower shaft rpm case was studied initially, but its resulting specific speed was considered low, resulting in lower efficiency potential. The higher shaft speed case was studied in order to increase the specific speed to a more favorable range between 0.60 to 0.90, in an effort to increase its efficiency potential. The centrifugal specific speed versus efficiency potential will be illustrated later in Figure 14. Both impellers were designed to have the same exit tip speed and a back sweep angle of 25°, but with different diameters. Table 2 shows the comparison between these two centrifugal stages.

The exit air temperature for both centrifugal cases is on the order of 1,551 °R. Either one of these two single stage centrifugal stages could produce a pressure ratio of 2.77, but their efficiencies are different because their specific speeds are different. While the centrifugal impeller running at 38,620 rpm has an adiabatic efficiency potential of 83.4%, the impeller running at 45,000 rpm would have an efficiency potential of 85.1% adiabatic due to its more favorable specific speed that is between 0.60 to 0.90, as will be discussed later. Note that these efficiencies do not include additional losses that may be incurred through the transition duct, or goose neck. In addition to specific speed, there are other factors to consider that influence the level of efficiency centrifugals can achieve, such as the hub-to-tip ratio as well as the rotor tip clearance to span ratio, both of which are indicators of the potential for parasitic leakages at the rotor shroud. However, in the case of these two centrifugals, the hub-to-tip ratios are the same at 0.742.

The overall axial length of the single centrifugal stage configuration option, including the transition duct, is 6.46 in. The severity of the transition duct could have been reduced by increasing the axial length, but the duct loss optimization was not within the scope of this study. Figure 5(a) and (b) illustrate the two different single centrifugal stages which were sized to meet the HPC flow and pressure ratio requirements. The goose-neck transition duct from the LPC illustrated in Figure 6 has a large change in radius from inlet to exit, and therefore may be prone to experience excessive pressure losses, thereby possibly reducing the HPC efficiency.

When evaluating the overall effect of the 45,000 rpm high pressure spool with the WATE engine sizing code, it was found that the high pressure turbine disk could not be designed sufficiently, since it did not have adequate radial space between the flow path and the shaft. Therefore, although the high speed impeller would prefer to run at a higher rpm for improved specific speed, and consequently higher efficiency potential, it places a severe technical challenge on the structural design of the high pressure turbine disk. In this case the HPC compressor efficiency would likely need to be traded for turbine disk structural design considerations.

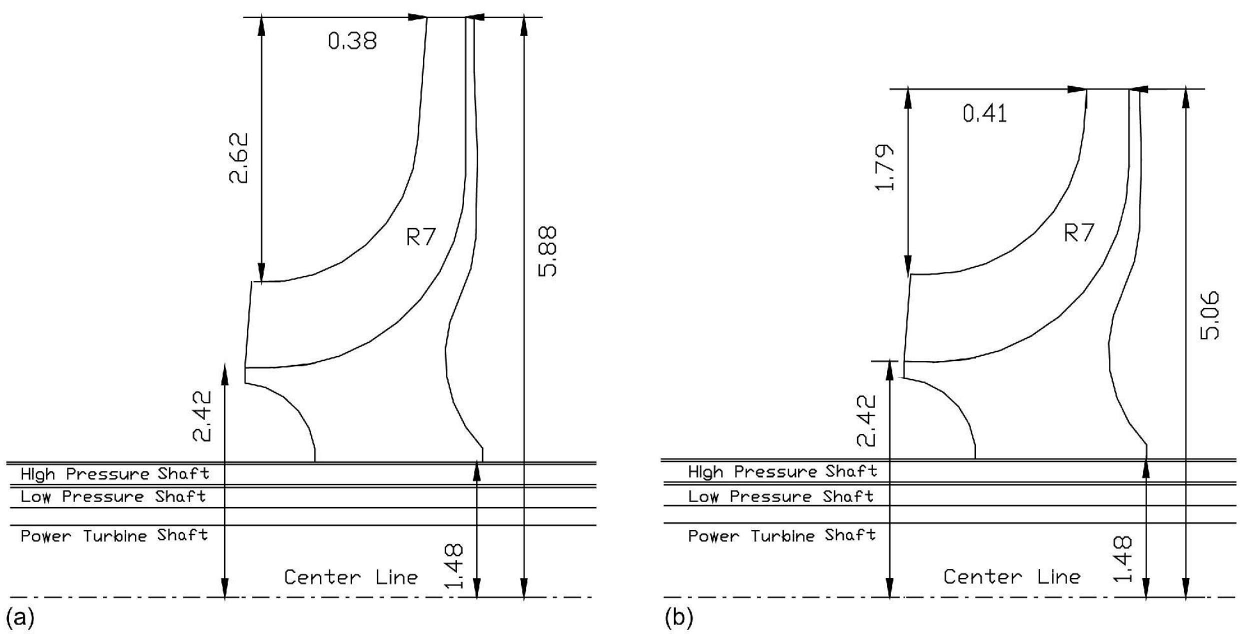

Figure 5. HPC Option #1: One Centrifugal stage. (a) Low specific speed impeller, 38,620 rpm, 5.88 in., (b) High specific speed impeller, 45,000 rpm, 5.06 in. Both Impellers have a tip speed of near 1980 ft/sec.

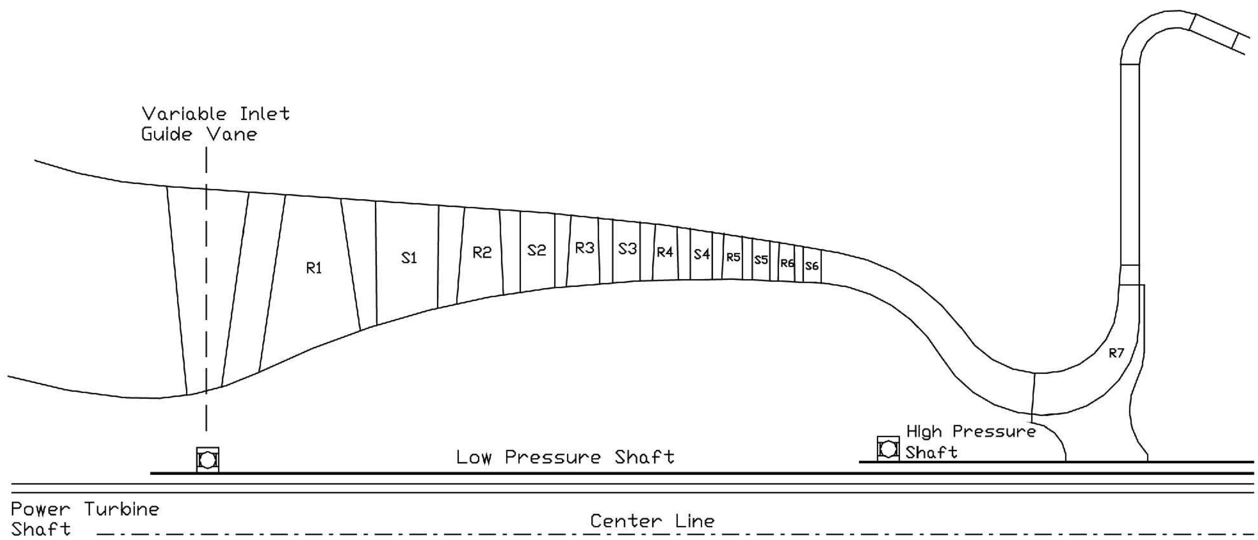

Figure 6. Two spool compressor. Six stage axial LPC and a single centrifugal stage HPC.

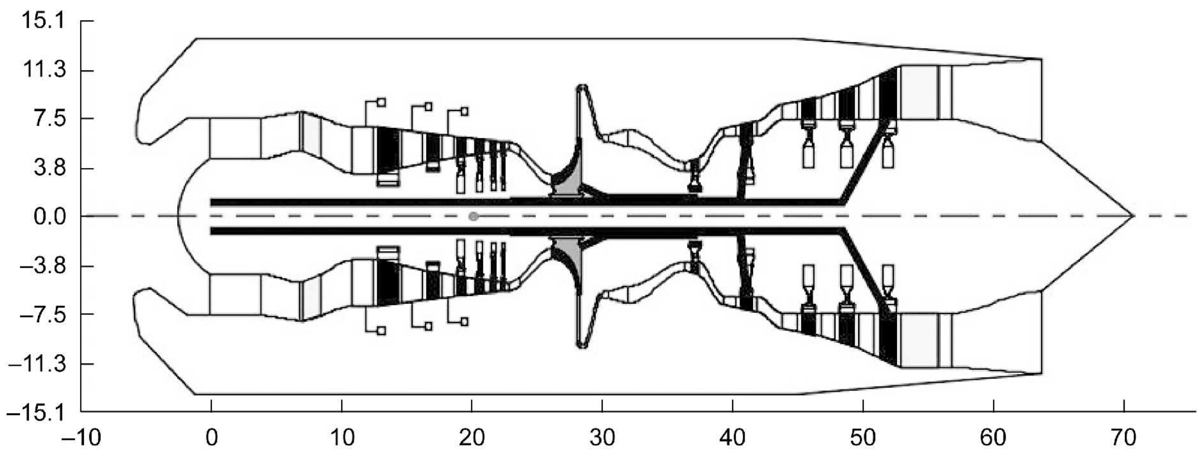

Figure 7. Three-spool engine generated by WATE code with compressor flow path using HPC Option #1.

Based on the high tip speed for this single centrifugal stage HPC concept, it would need to be fabricated from an advanced technology high Nickel content material which has higher strength properties at elevated temperatures than Inconel 718. Given such a material, this single stage centrifugal compressor could produce the required HPC pressure ratio of 2.77 at near 1,980 ft/sec tip speed.

The complete two spool compression system utilizing HPC Option #1 with a single centrifugal stage is illustrated in Figures 6 and 7 illustrates the resulting engine based on this HPC option. Note that the transition ducts between the LPC and HPC, as well as the duct between the HPT and LPT are likely excessively severe, and would need further refinement to minimize duct losses before they can be considered feasible. The severe ducts may be caused by the large difference between the rotational speeds of the high pressure spool running at 38,620 rpm, and the low pressure spool running at 24,800 rpm. The larger the difference between the shaft speeds, the greater the severity in the transition ducts. Note that the ducts can be designed with less severity with a penalty on engine length and weight. The total engine pod weight for this current case is estimated to be 921.5 lbm and the total engine pod length is 69.7 in. The flow analysis output listing for the single centrifugal stage HPC is in reference 10.

The HPC design study also considered subsequent Options #2 and #3 in which a centrifugal stage is supercharged by axial compressor stages, in an effort to reduce the tip speed of the centrifugal impeller to a level that is achievable with traditional materials such as Inconel 718.

HPC Configuration Flow Path Sizing Option #2: Single Axial Stage Supercharging a Centrifugal Stage

A second option was studied by adding one axial stage in front of a centrifugal stage to produce the 2.77:1 pressure ratio. The design process outlined in Figure 2 was followed in an iterative fashion to result in agreement between the flow analyses obtained with the various codes utilized in the conceptual design process. Design iterations were performed in this process and each code was executed manually. However, a slightly different approach was utilized in this iteration by starting with setting the maximum allowable tip speed for a centrifugal impeller fabricated from Inconel 718 to a notional tip speed limit of 1,650 ft/sec, since that is maximum allowable for an impeller at the compressor exit temperature of near 1,560 °R. The flow path of the axial stage and the transition ducts from the LPC were sized such that they are aesthetically pleasing (neither would be excessively severe). In addition, the annular area at the axial rotor inlet was designed to be smaller than the annular area at the exit of the LPC in order to accelerate the flow and reduce the possibility of separation at the inner and outer walls. This is expected to reduce the anticipated duct losses, although quantifying the duct losses was not part of this study. Large wall curvatures in the transition duct reduce the possibility of local accelerations and decelerations due to curvature effects on local velocity, which can result in pressure losses and excessive aerodynamic blockages. The tip speed of the first axial rotor was kept at nearly the same value of 1,275 ft/sec as that of the previous stage 6 of the LPC. The resulting rotational speed of the HPC became 32,650 rpm to produce the required overall HPC pressure ratio. The shaft speed was also arrived at by the axial rotor disk size requirement for sufficient radial space between the flow path and the high pressure shaft, as determined by the WATE code.

The single axial stage produces a pressure ratio of 1.36:1 while the centrifugal produces a pressure ratio of 2.04:1. This case resulted in the centrifugal stage having a value of normalized specific speed of 0.511, which is below what is considered to be the optimum range and lower than the previous case, consequently the maximum efficiency potential of this centrifugal is 83.5% adiabatic per the QUIK code estimate, as well as the specific speed versus efficiency curve shown in Figure 12 which is taken from reference 9. This HPC option would produce the required 2.77:1 pressure ratio at an overall efficiency of 83.8%, since the axial stage is estimated to have an efficiency of 87.4% and the centrifugal is at 83.5% efficiency. This impeller has an exit radius of 5.80 in., and exit blade height of 0.35 in. and an inlet hub-to-tip ratio is 0.80. The overall axial length of this HPC configuration including the transition ducts is 8.09 in., as illustrated in Figure 8.

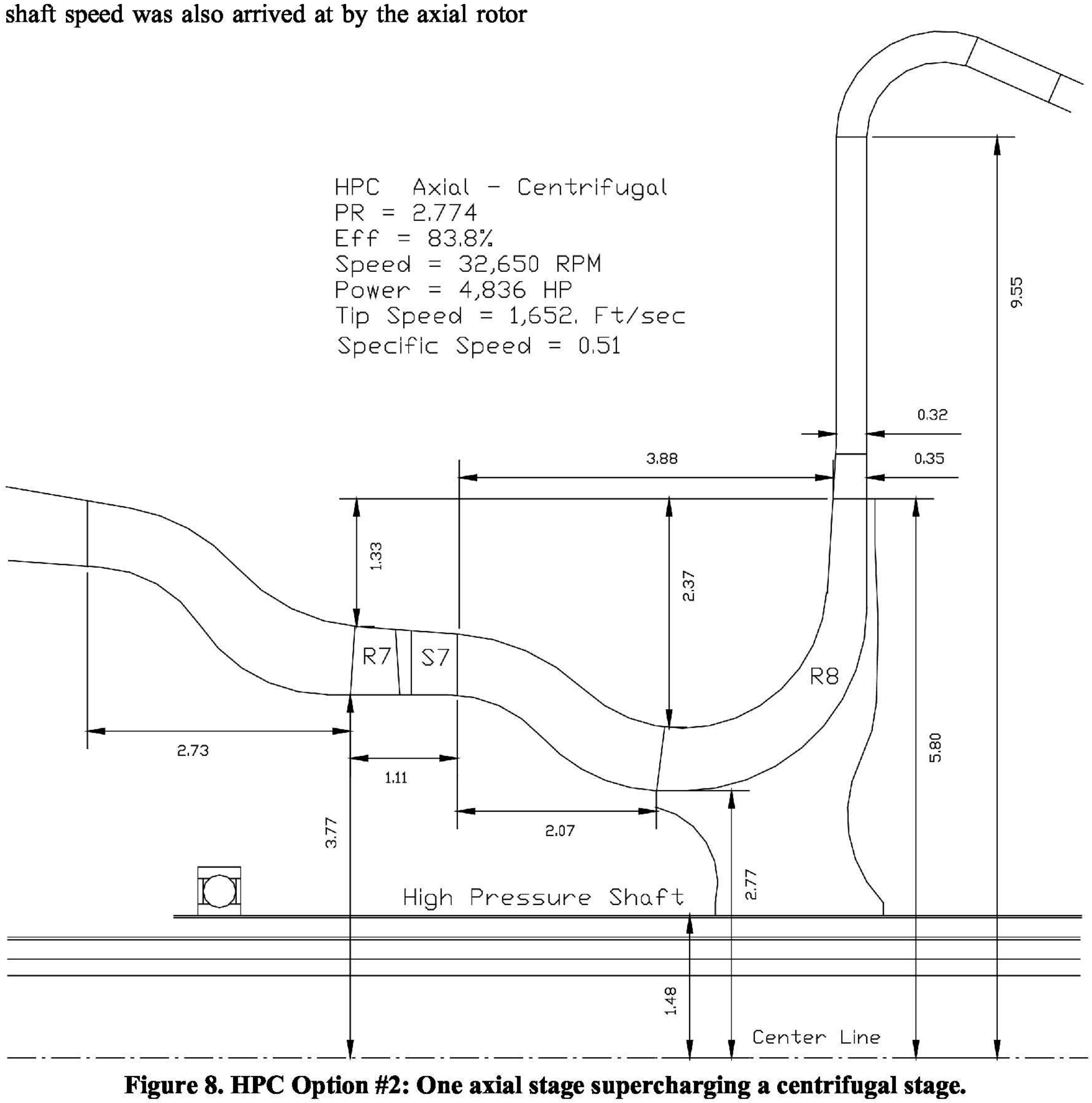

Figure 8. HPC Option #2: One axial stage supercharging a centrifugal stage.

In order to design this centrifugal stage to operate at a more favorable specific speed between 0.60 to 0.90, the shaft rotational speed would need to be increased, with the proportional reduction in impeller exit tip speed. However, this would need to consider the radial space required for structural design of the disk that supports the axial rotor. In addition, the high pressure turbine disk design requirements would need to be addressed, specifically whether there is adequate radial distance available between the turbine flow path and the high pressure spool shaft.

In addition to the specific speed, there are other factors that influence the efficiency, such as the hub-to-tip ratio at the inlet, which for this impeller is 0.80. This high inlet hub-to-tip ratio has a negative effect on the attainable level of efficiency. Note that this ratio for this impeller is slightly higher than in the previous single stage case, and is near the value considered to be the allowable limit for traditional aft-stage centrifugals. The impeller exit blade angle (back sweep) is nearly radial at 11.5°, resulting in the impeller exit absolute flow angle of 65.3° at the design point. The inlet radius of the vaned diffuser utilized for the conceptual design was at a 1.08 ratio with respect to the impeller exit radius, to avoid aeromechanical issues and is based on previous centrifugal compressor experience. This radius ratio is anticipated to result in adequate space allocated for mixing out the impeller exit blade-to-blade wakes prior to entering the radial diffuser vanes. The diffuser leading edge metal angle has nearly 0° of incidence with the flow exiting the impeller, which is at 65.3° from the radial direction. For the purposes of this study, a simple wedge diffuser configuration was sized, with a diffuser exit to inlet radius ratio of 1.525, resulting in an area ratio of 2.009 and a pressure recovery on the order of 0.60. In the process of striving to keep the centrifugal stage more compact, the selection of the diffuser radius ratio, and consequently its area ratio may be inversely proportional to its pressure recovery.

Table 3. HPC Compression System One Axial and Centrifugal Stage.

	Axial	Centrifugal
Rotor inlet		
Flow rate, corrected, lbm/sec	4.068	3.145
Mach, absolute	0.42	0.43
Mach, relative at tip	0.92	0.74
Tip speed, ft/sec	1275.	974.
Flow angle, relative, deg	60.3	51.1
Blade angle, deg	54.1	44.3
Rotor blade number	80	15/30
Rotor exit		
Blade angle, deg	34.3	11.5
Flow angle, absolute, deg	41.7	65.3
Deviation angle, deg	4.41	20.4
Diffusion factor	0.49	0.83
Relative velocity ratio	1.66	1.77
Tip speed, ft/sec	1266.	1652.
Stator		
Vane number	106	23
Diffusion factor	0.46	n/a
Stage		
Pressure ratio	1.358	2.042
Temperature ratio	1.102	1.264
Exit temperature, °R	1233.5	1558.6
Temperature rise, °R	114.1	325.1
Efficiency, adiabatic	87.4	83.5
Work coefficient	0.458	0.776
Horsepower	1245.	3592.

The design point overall mean line analysis code output listing for the two stages of this HPC compressor is summarized in Table 3 and is listed in reference 10.

The complete two spool compression system utilizing HPC Option #2 with a single axial followed by a centrifugal stage is illustrated in Figure 9. Although this configuration requires further detailed analysis, it can potentially meet the

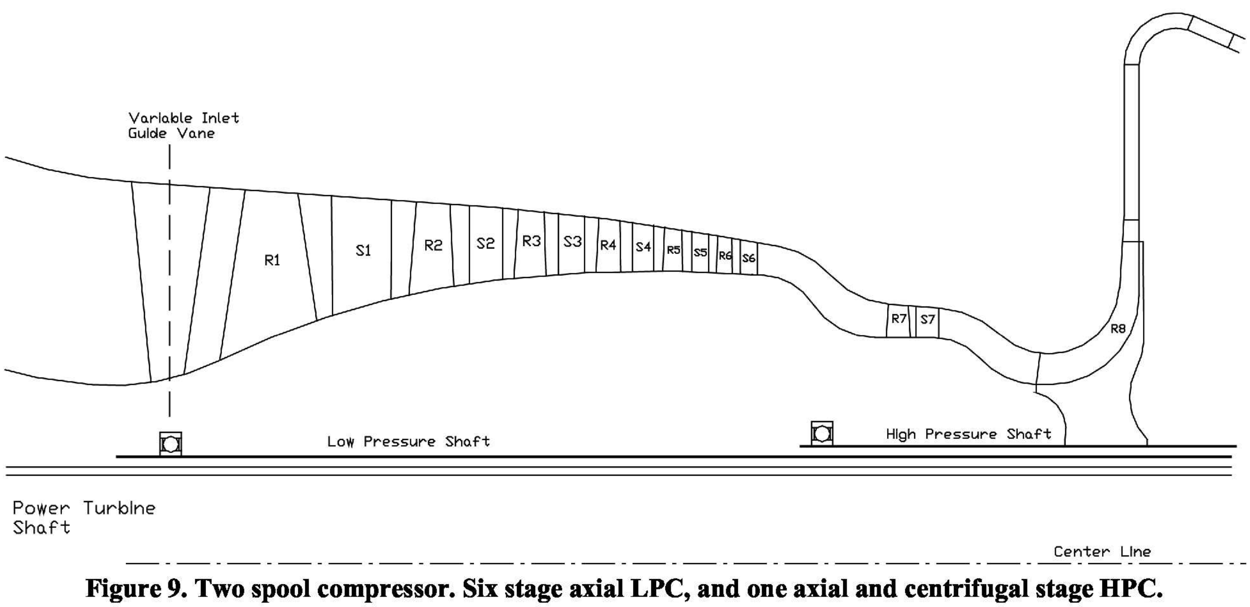

Figure 9. Two spool compressor. Six stage axial LPC, and one axial and centrifugal stage HPC.

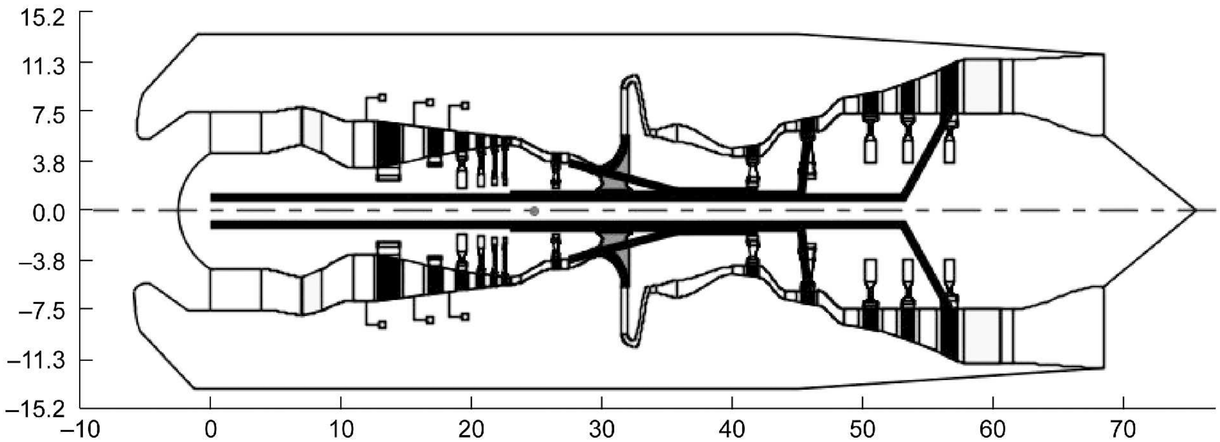

Figure 10. Three-spool engine generated by WATE code with compressor flow path using HPC Option #2.

LCTR-2 engine requirements. The total engine pod weight for this case is estimated to be 959.7 lbm and the total engine pod length is 74.3 in. The reduced radial diffusion system diameter at a diffuser exit to impeller radius ratio of 1.5 resulted in an HPC with a more compact radius than the power turbine. Figure 10 illustrates the resulting engine configuration for the single axial stage axi-centrifugal HPC option.

HPC Configuration Flow Path Sizing Option #3: Two Axial Stages Supercharging a Centrifugal Stage

A third option was studied by adding two axial stages in front of a centrifugal stage to produce the HPC overall 2.77:1 pressure ratio at 28,500 rpm. The second axial stage further reduces the work required by the rear stage centrifugal compressor consequently reducing its tip speed requirements to a modest 1,462 ft/sec. The flow path of the axial stage and the transition ducts were sized to reduce abrupt wall curvatures. The goose neck transition duct between the LPC and the HPC was sized such that the annular area at the HPC inlet is smaller than the annular area at the LPC exit, to accelerate the flow and reduce the possibility of separation at the inner and outer walls and as a consequence reduce the anticipated duct losses.

The tip speed of the first axial rotor was sized on the order of 1,275 ft/sec such that supporting disks would have adequate radial space between the flow path and the high pressure shaft. The design point diffusion factor of the two axial stages was limited to on the order of 0.50 at the design condition. Due to disk and shaft sizing constraints determined with the WATE code, the hub-to-tip ratio of the axial stages became 0.88. The two axial stages produce a pressure ratio of 1.35:1 and 1.29:1 respectively, while the centrifugal stage produces a relatively low pressure ratio of 1.60:1. This HPC option would produce the required 2.77:1 pressure ratio at an overall efficiency of 85.2%, an increase over the previous two cases, since the axial stages are expected to have an efficiency of 87.6 and 88.2%, while the centrifugal is at 83.5% efficiency. This case resulted in the centrifugal stage having a value of normalized specific speed of 0.538, which does not fall within the desired range between 0.60 to 0.90. This impeller has a diameter of 11.76 in., an exit blade height of 0.38 in., and an inlet hub-to-tip ratio is 0.81. The overall axial length of this HPC configuration including the transition ducts is 8.81 in., and would likely even be longer as the transition duct between the axial stages and the inlet to the centrifugal may need to be extended beyond that illustrated in Figure 11. Figure 12 illustrates the engine configuration that resulted when incorporating HPC Option #3. The total engine pod weight for this case is estimated to be 987.5 lbm and 75.5 in. and the total engine pod length is 75.5 in.

Summary of Centrifugal HPC Results

Figure 13 illustrates the historical variation of maximum attainable efficiency versus specific speed, with the above three cases featuring a centrifugal stage in the HPC. Also shown in Figure 13 is a potential increase in the level of maximum attainable efficiency that may result from centrifugal technology development efforts aimed at improving on the current state-of-the-art. Areas of potential efficiency improvement and pressure loss reductions include the impeller, the radial diffuser, the 90° bend, and the deswirler vane. Note that the specific speed for the above cases which were considered to be feasible from an overall engine design consideration remained nearly constant at between 0.51 and 0.55, and therefore their design point efficiencies were expected to be nearly constant. Option #1 with the single stage centrifugal rotating at 45,000 rpm shaft speed is shown on Figure 13 for reference, although its use in the current study was limited by the design constraint of inadequate radial space for the high pressure turbine disk. However, all the HPC options in this study are at a higher level of specific speed and therefore higher efficiency potential than the centrifugal stage from reference 11 which was a single spool compressor that produced the same overall pressure ratio as the two spool compressor in this study. Note that even though the specific speed range with the historically highest efficiency potential is between 0.60 and 0.90, three of the centrifugals in this study do not fall into that range due to other constraints such as inadequate

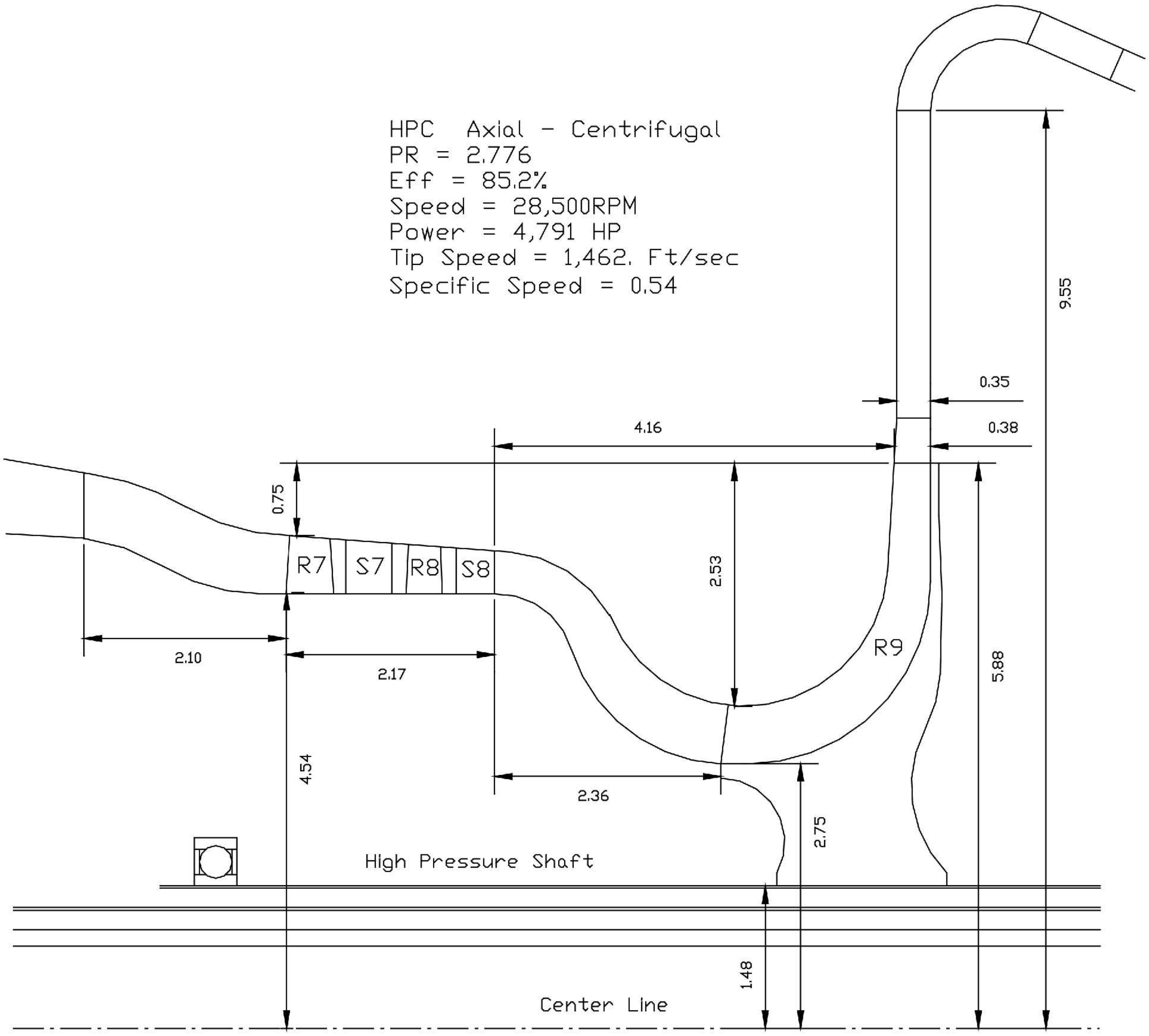

Figure 11. HPC Option #3: Two axial stages with one centrifugal stage high pressure compressor.

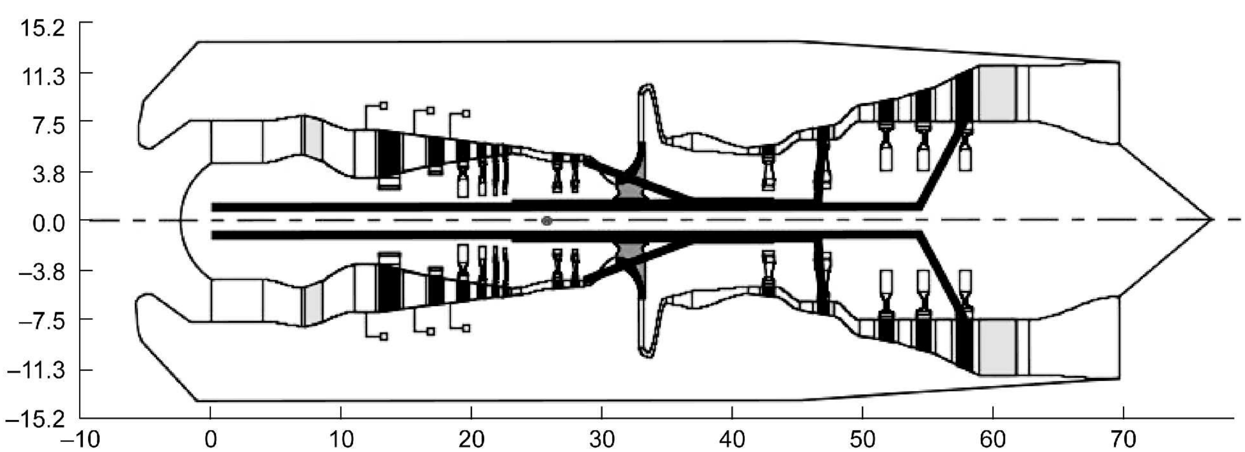

Figure 12. Three-spool engine generated by WATE code with compressor flow path using HPC Option #3.

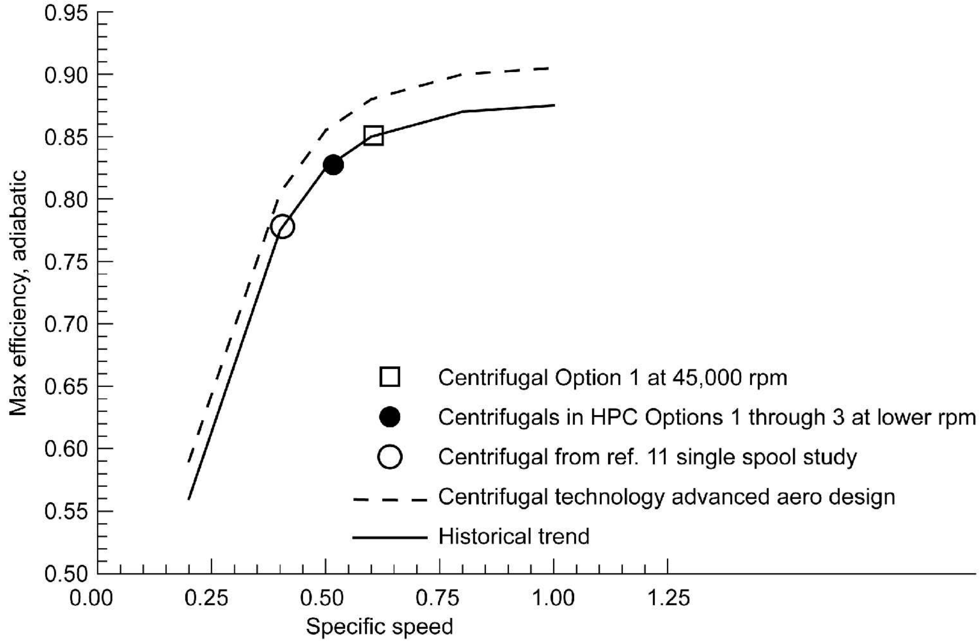

Figure 13. Centrifugal compressor specific speed versus maximum efficiency potential (ref. 9).

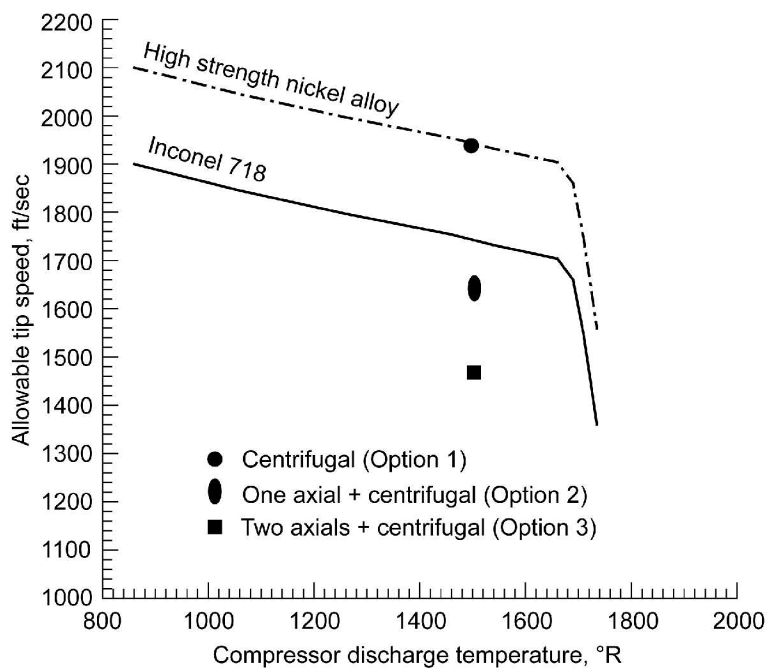

Figure 14. Exit temperature versus allowable tip speed limit for two different advanced Nickel alloy materials, and the three HPC options utilizing centrifugal impellers.

space for the high pressure turbine disk, as previously mentioned. Only the centrifugal running at 45,000 rpm falls in the favorable specific speed range. Also note that development of advanced compressor technology in the future may result in an increase of the absolute levels of maximum efficiency potential by 1-2 points as shown by the dashed line in Figure 13. Illustrated in Figure 14 are the tip speeds of the centrifugal impeller configurations compared to the impeller can potentially be met only with more advanced technology high strength Nickel alloy materials. The two-stage axial plus centrifugal case studied (Option #3) is not fully utilizing the capability of either of these impeller materials.

Note that any assumptions in this study if changed could alter some of the outcomes and potentially impact some of the results. For example, an advanced high strength impeller material with better properties at high temperature would enable centrifugal impeller tip speeds at 2,000 ft/sec and above, and could potentially impact the specific speed and consequently its efficiency potential. However, equivalent advances in turbine technology are also necessary to fully utilize improved capabilities on the compressor side. Specifically, advancements in turbine disk and blade materials could also enable running the high pressure spool at increased rotational speeds, resulting in a favorable increase to the centrifugal compressor specific speed to be within the favorable range for maximum efficiency potential. Any material change for the compressors and turbines may also impact the overall engine weight. In addition, if the work split between the LPC and the HPC were changed from what was assumed to be fixed in this study, it may impact design flexibility to improve the specific speed of the centrifugal compressor.

HPC Configuration Flow Path Sizing Option #4: All-Axial Four Stage HPC

This option considers a four-stage all axial HPC, which did not use a centrifugal compressor as a last stage, to produce the overall HPC pressure ratio of 2.77:1. A constant hub diameter option was utilized for sizing the multi-stage axial compressor to allow radial space for the disks, and that the hub-to-tip radius ratio of the latter stages would not become even larger. The design point diffusion factor of the axial rotors was limited to 0.50 at the design condition. As the various design criteria were applied, the tip speed of the first axial rotor became 1,275 ft/sec and the shaft speed became 32,650 rpm. As a result of the disk and shaft sizing constraints determined with the WATE code, the hub-to-tip ratio of the first axial stage became 0.84 to allow adequate space for the disks between the flow path and the shaft. The four axial stages produce a pressure ratio of 1.36, 1.31, 1.27, and 1.25, respectively. The last stage blade span became 0.44 in., and is at or below the limit of what is considered to be an acceptable design. The challenges with a low span blade include higher than traditional rotor clearance percentages, as well as the circumferential variation of clearance due to high hub-to-tip ratio and high relative surface finish in comparison to blade size. This four stage axial option would produce the required 2.77:1 pressure ratio at an overall efficiency of 87.0%, an increase over the previous three cases featuring centrifugal compressors. Figure 15 illustrates the four axial stage HPC option. The overall axial length of this HPC configuration including the transition ducts is 7.73 in.

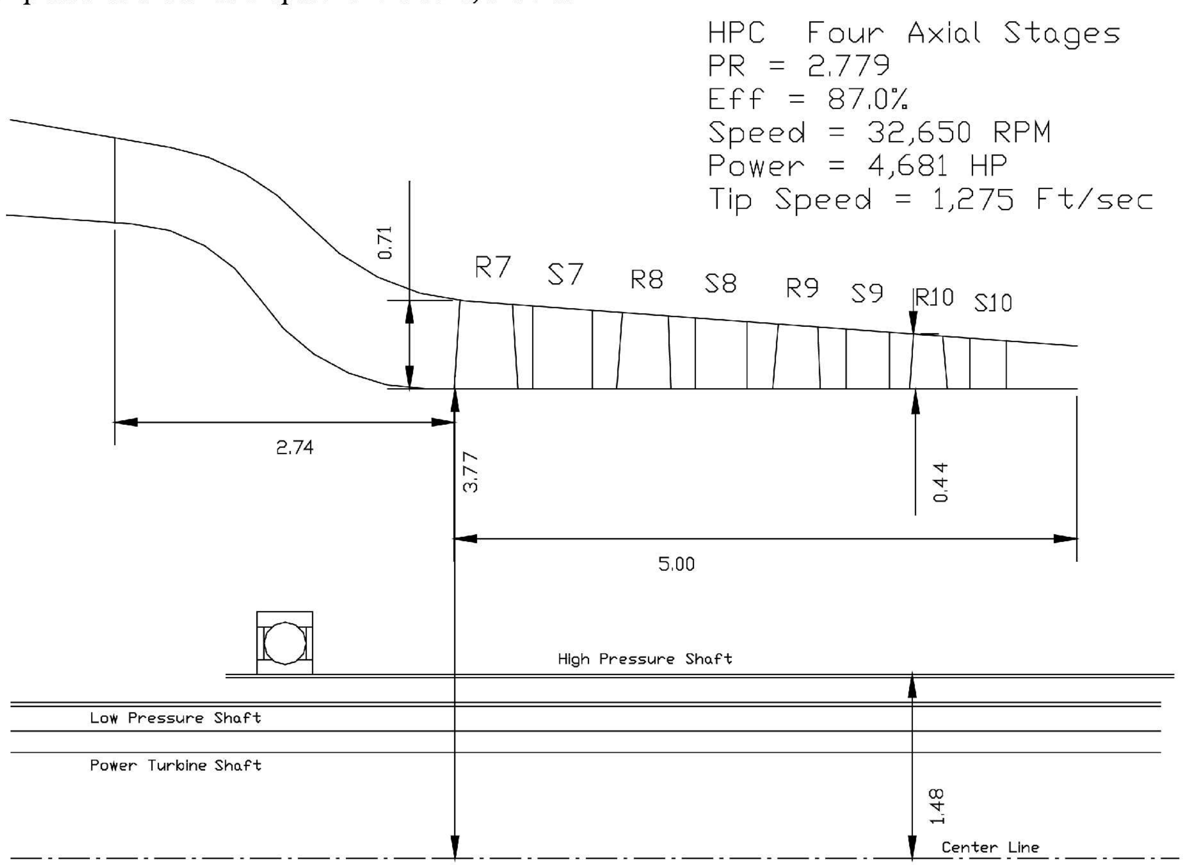

Figure 15. HPC Option #4: Four axial stages with last blade exit height of 0.44 in.

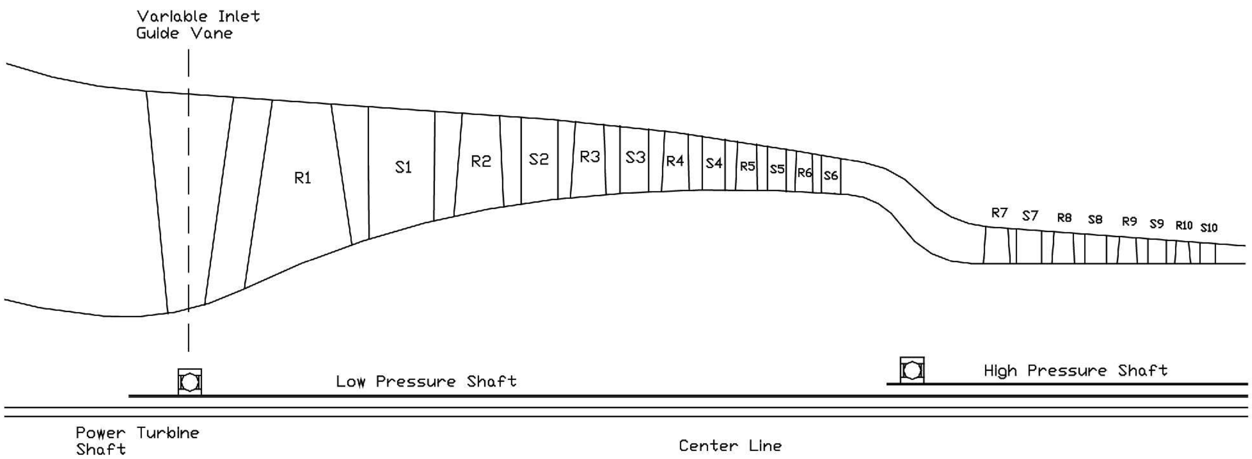

Figure 16. Two Spool compressor. Six axial stage LPC with all axial stage HPC.

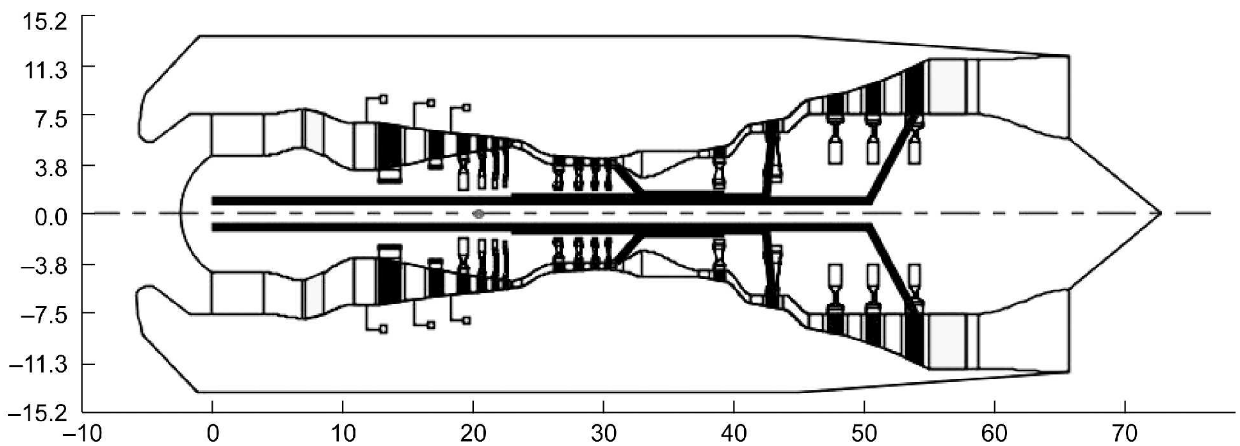

Figure 17. Three-spool engine generated by WATE code with compressor flow path using HPC Option #4.

The complete flow path for the two-spool compressor including the transition duct is shown in Figure 16 for the LPC – HPC compressor cross section. Figure 17 illustrates the engine configuration that resulted from this compressor option. The total engine pod weight for this case is estimated to be 892.2 lbm and 71.6 in.

HPC Design Refinement of Option #4: Four-Stage Axial Compressor

The flow path obtained from the initial sizing of the four stage axial flow path and blades with the COMDES code using assumed losses was input into the TCDES design code per the procedure outlined in Figure 3. Losses and the flow path were refined iteratively utilizing the TCDES and T-AXI codes respectively, which contain the loss models. The last stage rotor blade height is 0.44 in. The rotor tip radial clearance of 0.012 in. results in a variation of rotor tip clearance to blade span ratio in the four stage axial HPC. The first stage tip clearance ratio is 1.7%, while the fourth stage tip clearance ratio is 2.8%. These levels of clearance are higher than for traditional compressors, and can result in increased tip leakage losses.

The computational grid for the four-stage axial HPC, also generated with the TCDES code, is shown in Figure 18. The T-AXI through flow code utilized this grid to determine improved estimates of rotor efficiency levels and stator losses, based on the models within the code for shock, diffusion factor loading and tip clearance losses.

The output from the COMDES, TCDES and the T-AXI through flow analysis resulted in the overall HPC efficiency of 87.0% at the 2.77:1 pressure ratio. The hub diameter limitations have been determined from the conceptual mechanical design and analyses utilizing the WATE code. The HPC stage performances are listed in Table 4, with the detailed outputs listed in reference 10.

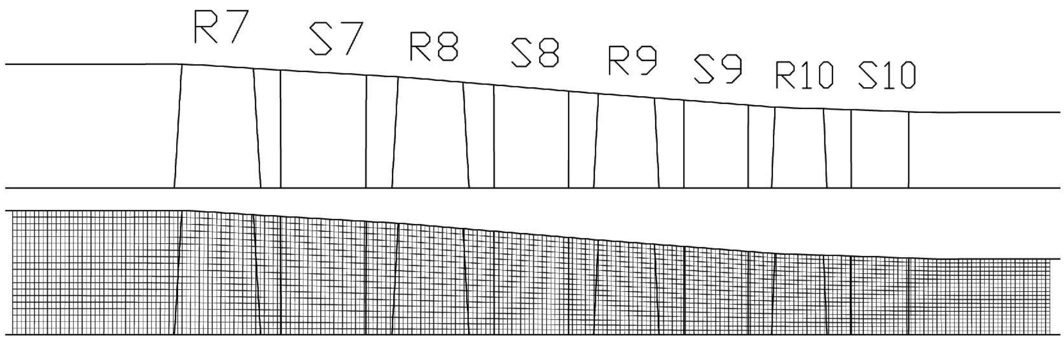

Figure 18. Computational grid and flow path for the four axial stage high-pressure compressor illustrating the rotors (R) and stators (S). The last stage rotor blade height is 0.44 in.

Table 4. HPC Compression System Four Axial Stage of Option #4.

	1	2	3	4
Rotor inlet				
Flow rate, corrected, lbm/sec	4.068	3.151	2.524	2.066
Mach, absolute	0.42	0.37	0.35	0.34
Mach, relative at tip	0.92	0.84	0.79	0.75
Tip speed, ft/sec	1275.	1247.	1221.	1198.
Flow angle, relative, deg	60.3	61.7	61.8	61.5
Blade angle, deg	54.1	55.6	55.8	55.6
Rotor blade number	59	61	70	78
Rotor exit				
Blade angle, deg	35.6	37.0	36.5	35.0
Flow angle, absolute, deg	42.4	42.8	41.9	41.6
Deviation angle, deg	4.4	4.4	4.3	4.4
Diffusion factor	0.50	0.50	0.48	0.48
Relative velocity ratio	1.69	1.66	1.62	1.60
Tip speed, ft/sec	1266.	1240.	1214.	1193.
Stator				
Vane number	82	81	94	112
Diffusion factor	0.50	0.50	0.48	0.50
Stage				
Pressure ratio	1.354	1.301	1.267	1.244
Temperature ratio	1.101	1.086	1.076	1.070
Exit temperature, °R	1232.2	1338.3	1440.4	1540.7
Temperature rise, °R	112.8	106.0	102.1	100.3
Efficiency, adiabatic	87.5	87.5	87.4	87.1
Work coefficient	0.453	0.449	0.456	0.470
Horsepower	1231.	1171.	1142.	1136.

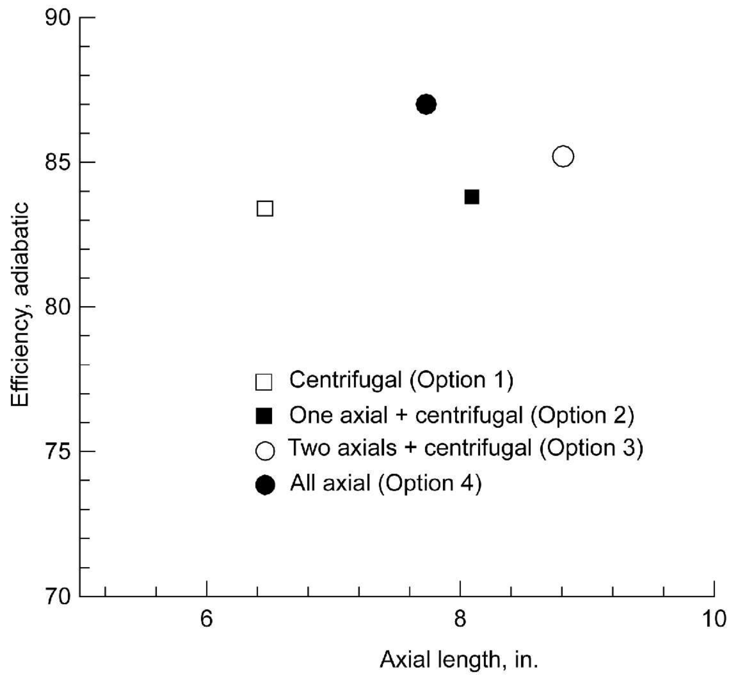

Figure 19. HPC efficiency versus axial length of the four design options cases studied.

Comparison of Four HPC Design Options

A comparison was made between the four options for the HPC flow path, illustrating the estimated efficiency versus the axial length in Figure 19. Note that the study utilized current estimates for compressor efficiency as well as the assumptions in the WATE code for sizing the turbines based on a certain level of technology. A further assessment of state-of-the-art turbine technologies, including cooling requirements and advanced materials could result in potential improvements to the LCTR-2 engine, and subsequently offer additional flexibility to determining the best work split between the LPC and the HPC from both the perspective of turbine and compressor aerodynamic and structural considerations. Also, potential future advancements in centrifugal compressor aerodynamics could result in additional realizable efficiency benefits, even at reduced levels of specific speed.

A comparison of total engine pod weight versus total engine pod length was made from the WATE code results for all four HPC design options and is illustrated in Figure 20. The resulting comparison indicates that the four stage all axial HPC option (Option #4), appears to have the lowest total engine weight and pod length, however, that option needs further assessment, as it has a 0.44 in. last stage exit blade span, which is at, or below the limit considered to be the minimum for axial blades.

LPC Off-Design Analysis:

The off-design compressor performance was estimated with COMDES. During the assessment of numerous combinations of variable geometry settings, it was determined that it would be adequate to vary only the variable geometry inlet guide vane (IGV) along with the stators in the first two stages, as resetting these three vanes resulted in acceptable stage matching throughout a wide range of flows at each speed line. Variable geometry stators in subsequent stages 3 to 6 would not have resulted in significantly widening the resulting overall flow range or operability of the LPC. The most suitable variable geometry schedule was determined for each speed line by manually varying the IGV and stator reset angles, that is, by specifying the inlet swirl angles at the rotor leading edges. The stage matching was achieved by varying the inlet guide vane and stators to result in acceptable levels of rotor incidence, relative velocity ratio and diffusion factor. A model consisting of a user defined function of rotor incidence in COMDES enables an estimate of the off-design performance by reducing the design point rotor efficiency. As discussed in reference 6, the off design efficiency lapse in the code was calibrated based on test data of highly loaded transonic compressor cases, with unique incidence being on the order of 6°. This enables sizing the variable geometry schedule to result in the overall performance at off-design speed and flow conditions. The performance

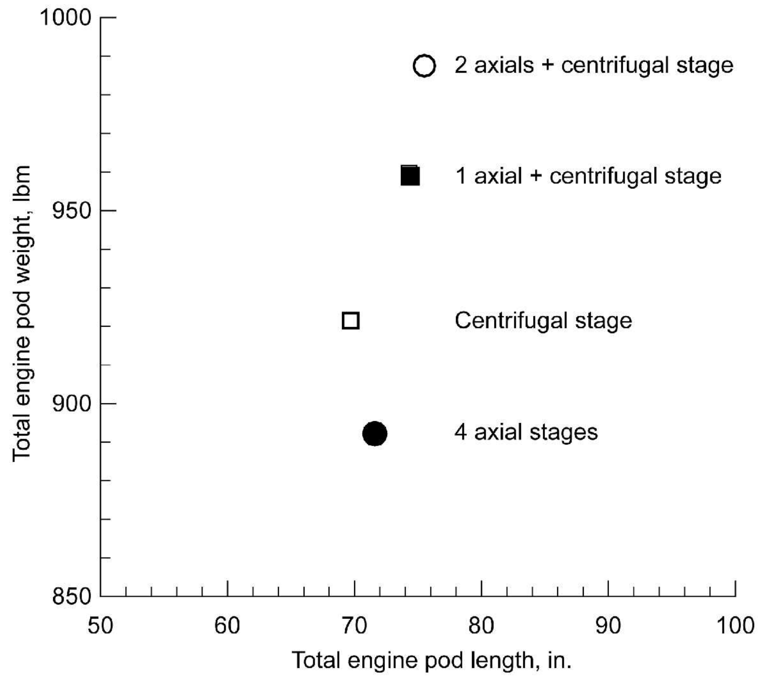

Figure 20. Total engine pod length versus total engine pod weight for the four HPC design options.

along each speed line for the LPC was estimated with COMDES by incrementally varying the incidence of rotor 1 to result in changing the air flow rate, until the rotor maximum diffusion limits were reached in any of the six axial stages. This was repeated for each speed line from 105 to 80%. Figure 21(a) and (b) illustrate the estimated compressor characteristic maps for pressure ratio and efficiency versus corrected flow and design corrected percent speed. Note that at the 105% speed line the variable geometry inlet guide vane was reset to 5° in the positive direction in order to manage the tip relative velocity of rotor 1, causing the speed line to appear slightly discontinuous in comparison with the lower speed lines. The criteria used to determine the onset of surge were the maximum values of rotor diffusion factor of near 0.60 and relative velocity ratio near 1.95. The estimated LPC map data are listed in reference 10.

HPC Off-Design Analysis

The characteristic performance map for the HPC was also estimated at off-design speeds and flows using COMDES. Since there is no variable geometry in the HPC, the process was simpler and merely involved varying the flow along each speed line by means of incrementally changing the rotor incidence, and executing the multistage flow analysis. For the purpose of map generation the four-stage all axial compressor configuration was selected, even though the last stage blade height may be considered excessively small for reasons previously mentioned. The 105, 100, 95, 90, 85, and 80% of design corrected speed lines were executed for a range of corrected flows. The criteria used to determine the onset of surge were the same as for the LPC. The estimated HPC characteristic pressure ratio and efficiency maps are illustrated in Figure 22(a) and (b).

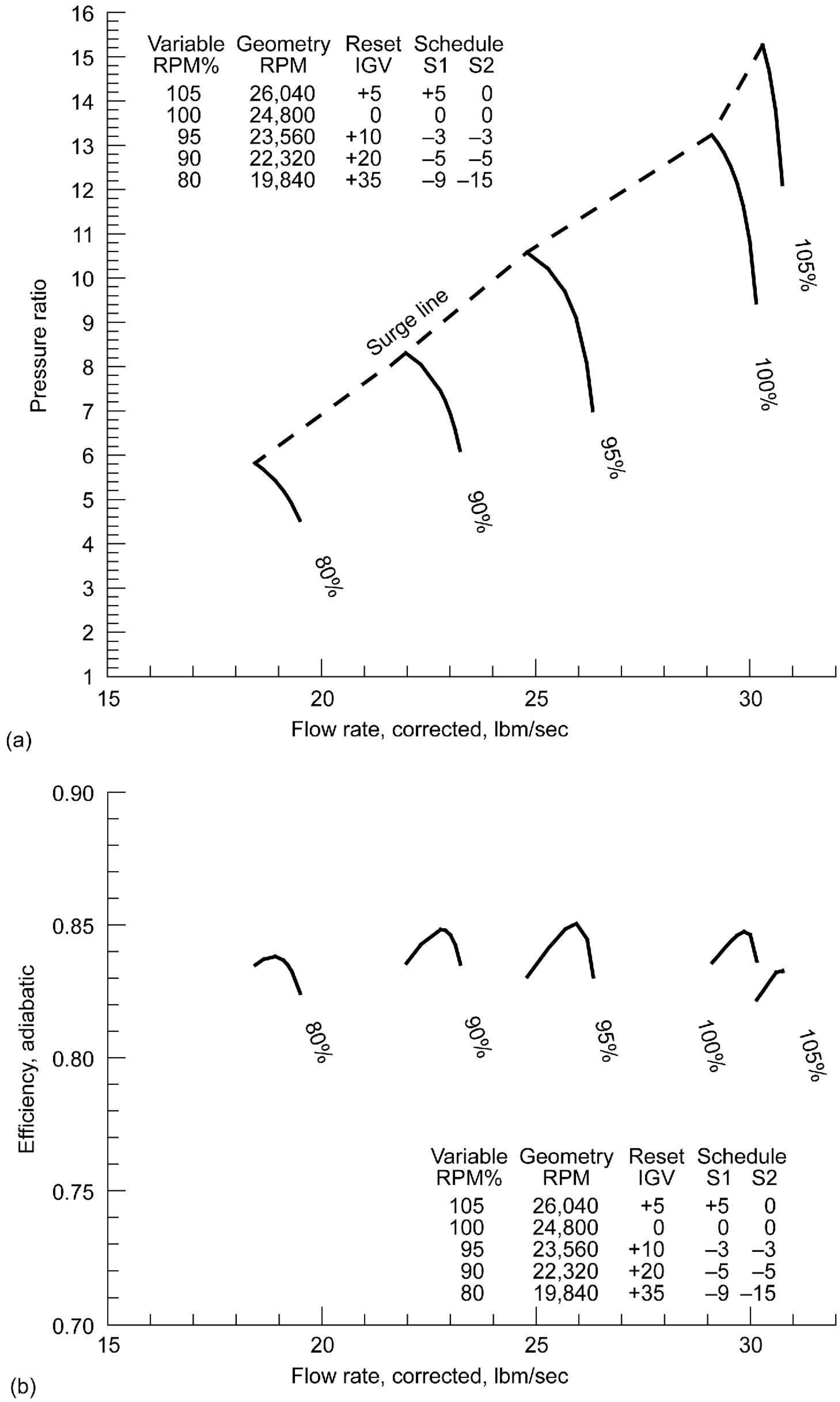

Figure 21. LPC performance maps with variable geometry schedule. (a) Pressure ratio. (b) Efficiency.

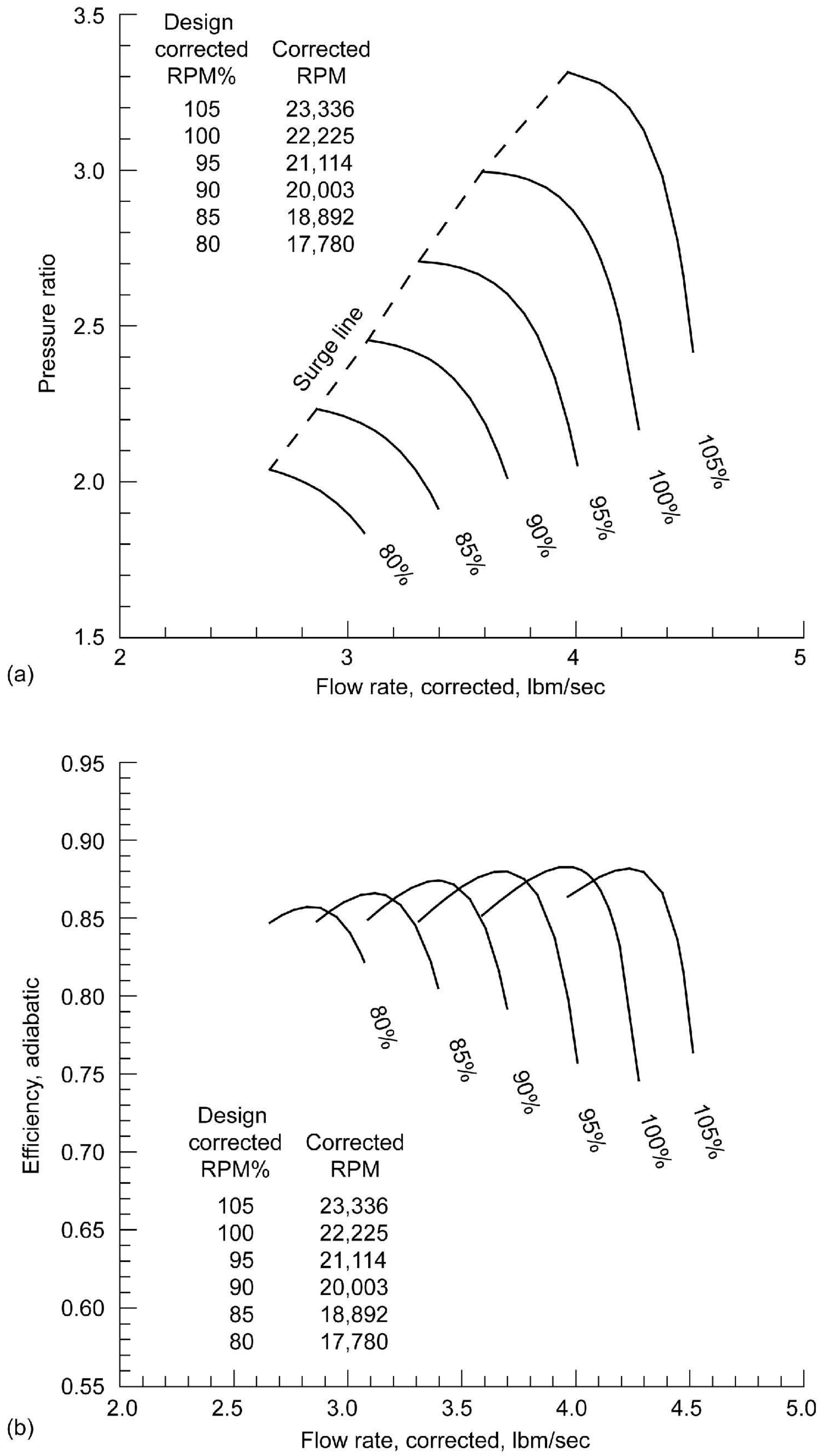

Figure 22. HPC performance maps. (a) Pressure ratio. (b) Efficiency.

CONCLUSIONS

A conceptual design study was made of a two spool compressor to meet the performance requirements of a notional LCTR-2 engine. The LPC requirements can be met with a six stage transonic axial compressor with an overall pressure ratio of 10.83:1 at the design point. Characteristic maps were produced for the LPC, with a variable inlet guide vane as well as variable stators in stages 1 and 2. Proper aerodynamic matching between the transonic stages is a challenge for maintaining stability and efficiency. Four options for the 2.77:1 PR HPC compressor were studied featuring an axial-centrifugal compressor configuration with zero, one and two axial stages ahead of the centrifugal compressor stage, as well as one option with four axial stages. The result of this study shows that from an aerodynamic perspective, it is feasible to meet the requirements of a 30:1 pressure ratio compressor for the LCTR-2 engine with a two spool axial-centrifugal compressor, comprising a six stage transonic LPC, followed by an HPC configuration which can be either a single stage centrifugal, an axi-centrifugal, or an all axial compressor featuring four stages.

Technical challenges for the LPC design include:

- High tip relative Mach number of the first stage, due in part to physical space limitation of the rotor drum.
- Aerodynamic matching of the transonic axial stages at off-design operation with variable geometry.

Technical challenges of the HPC designs include:

- Small blade span of last stage rotor of all-axial compressor at 0.44 in. may be excessively small.
- Large clearance-to-span ratio of 2.7% for the last stage axial rotor of the all axial stage option.
- Small exit blade span of centrifugal impeller ranging from 0.35 to 0.41 in. depending on configuration and specific speed.
- Increasing the specific speed of the centrifugal stage from a nominal range of 0.51 to 0.55 to a more optimum value is limited by the disk space and strength requirements of the high pressure turbine at increased speeds.
- High impeller inlet hub-to-tip radius ratio of 0.80 is set by shaft size requirements and can limit efficiency potential.
- Higher centrifugal impeller tip speeds up to 2,000 ft/sec could be required for a more optimum design and can only be achieved with the use of advanced nickel alloy materials with high strength characteristics at elevated temperatures in the range of 1,560 °R.

The two spool configuration of the gas generator is a thermodynamic and structural improvement over the single spool, due to improved efficiency levels caused by reduced rotor tip clearance ratios and improved centrifugal impeller specific speed. Modifying the LPC-HPC work split with consideration for turbine design requirements, may result in additional flexibility in optimizing the centrifugal stage specific speed, and consequently improve its efficiency potential.

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