NASA/CR—2008-215175

System Study for Axial Vane Engine Technology

Patrick R. Badgley, Michael R. Smith, and Cedric O. Gould
Advanced Technologies, Inc., Starkville, Mississippi

July 2008

NASA STI Program . . . in Profile

Since its founding, NASA has been dedicated to the advancement of aeronautics and space science. The NASA Scientific and Technical Information (STI) program plays a key part in helping NASA maintain this important role.

The NASA STI Program operates under the auspices of the Agency Chief Information Officer. It collects, organizes, provides for archiving, and disseminates NASA's STI. The NASA STI program provides access to the NASA Aeronautics and Space Database and its public interface, the NASA Technical Reports Server, thus providing one of the largest collections of aeronautical and space science STI in the world. Results are published in both non-NASA channels and by NASA in the NASA STI Report Series, which includes the following report types:

- TECHNICAL PUBLICATION. Reports of completed research or a major significant phase of research that present the results of NASA programs and include extensive data or theoretical analysis. Includes compilations of significant scientific and technical data and information deemed to be of continuing reference value. NASA counterpart of peer-reviewed formal professional papers but has less stringent limitations on manuscript length and extent of graphic presentations.

- TECHNICAL MEMORANDUM. Scientific and technical findings that are preliminary or of specialized interest, e.g., quick release reports, working papers, and bibliographies that contain minimal annotation. Does not contain extensive analysis.

- CONTRACTOR REPORT. Scientific and technical findings by NASA-sponsored contractors and grantees.

- CONFERENCE PUBLICATION. Collected papers from scientific and technical conferences, symposia, seminars, or other meetings sponsored or cosponsored by NASA.

- SPECIAL PUBLICATION. Scientific, technical, or historical information from NASA programs, projects, and missions, often concerned with subjects having substantial public interest.

- TECHNICAL TRANSLATION. English-language translations of foreign scientific and technical material pertinent to NASA's mission.

Specialized services also include creating custom thesauri, building customized databases, organizing and publishing research results.

For more information about the NASA STI program, see the following:

- Access the NASA STI program home page at *http://www.sti.nasa.gov*

- E-mail your question via the Internet to *help@sti.nasa.gov*

- Fax your question to the NASA STI Help Desk at 301–621–0134

- Telephone the NASA STI Help Desk at 301–621–0390

- Write to:
  NASA Center for AeroSpace Information (CASI)
  7115 Standard Drive
  Hanover, MD 21076–1320

NASA/CR—2008-215175

System Study for Axial Vane Engine Technology

Patrick R. Badgley, Michael R. Smith, and Cedric O. Gould
Advanced Technologies, Inc., Starkville, Mississippi

Prepared under Cooperative Agreement NNC–04VN19P

National Aeronautics and
Space Administration

Glenn Research Center
Cleveland, Ohio 44135

July 2008

This report contains preliminary findings, subject to revision as analysis proceeds.

Trade names and trademarks are used in this report for identification only. Their usage does not constitute an official endorsement, either expressed or implied, by the National Aeronautics and Space Administration.

This work was sponsored by the Fundamental Aeronautics Program at the NASA Glenn Research Center.

Level of Review: This material has been technically reviewed by NASA technical management.

Available from

NASA Center for Aerospace Information
7115 Standard Drive
Hanover, MD 21076–1320

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161

Available electronically at http://gltrs.grc.nasa.gov

Contents

List of figures

Appendix A figures

Appendix B figures

List of tables

System Study for Axial Vane Engine Technology

Patrick R. Badgley, Michael R. Smith, and Cedric O. Gould
Advanced Technologies, Inc.
Starkville, Mississippi 39760

1.0 Introduction

The purpose of this engine feasibility study was to determine the benefits that can be achieved by incorporating positive displacement axial vane compression and expansion stages into high bypass turbofan engines. These positive-displacement stages would replace some or all of the conventional compressor and turbine stages in the turbine engine, but not the fan. The study considered combustion occurring internal to an axial vane component (i.e., Diesel engine replacing the standard turbine engine combustor, burner, and turbine); and an axial vane compressor and axial vane turbine replacing conventional turbine engine inline continuous flow combustors.

Advanced Technologies Inc. (ATI) examined two engine sizes; a 10,000 lb Sea-Level-Static (SLS) thrust class engine and a 25,000 lb SLS thrust class engine. These Baseline Engines (appendix A) are referred to as the PAX 50 and PAX 150 engines respectively. The 10,000 lb thrust class engine was examined with both a conventional turbine engine combustor and within-vane-component-combustion. The 25,000 lb thrust class engine was examined with a conventional turbine engine combustor. NASA provided ATI with the turbofan engine data generated by the NPSS cycle analysis code for each baseline turbofan engine. All initial analyses were conducted for standard sea level conditions as specified in the contract. The final analyses were performed at engine inlet pressures equal to fan downstream pressures.

ATI developed thermodynamic analytical and mechanical models of axial vane compressors, expanders and internal axial vane combustion systems appropriate for the engines analyzed in this effort. The thermodynamic model accounts for inlet losses, exhaust losses and other flow pressure losses, heat losses, friction losses, and internal leakage losses. ATI has provided NASA with the thermodynamic analytic models (analytic descriptions, computational flow charts and input/output parameters) for the compressor, expander and internal axial vane combustion system in a form suitable for translating the analytic model into computer code that will interface with the NPSS code. The analytic models provide for output of pressures, temperatures, and velocity profiles at different locations in each device. Using the thermodynamic analytic models provided by ATI, NASA can code the thermodynamic computational component models that interface with the NPSS cycle analysis code. The mechanical model of each configuration consists of a three-dimensional drawing of the converged component, a listing of weight values, and a bill of material listing the component's materials and specifications.

ATI performed analysis of axial vane turbine hybrid engine cycles in each thrust class using the NPSS data and the AV-Design Program developed by ATI. A series of different engine configurations at different operating conditions were analyzed and modeled to yield an optimized engine to match the sea-level thrust ratings of each baseline turbofan engine. Solid models and engine weights were generated for each of the optimized engines. The results of this feasibility study revealed slight specific weight advantages for the axial vane components in turbofan engines, but significant specific fuel consumption advantages for the Diesel engine and hybrid engine using axial vane technology.

2.0 Description of Axial Vane Mechanism

A simplified cut-away model of an axial vane machine is shown in figure 1. The axial vane mechanism is unique in that a multiplicity of straight vanes, contained in a cylindrical rotor, travel parallel to the axis of rotation and compress or expand fluids in the same manner as reciprocating piston mechanisms. Since the vanes move parallel to the axis of rotation, the mechanism is always dynamically balanced and there are minimum energy losses due to reciprocating motion and accelerations typical of piston mechanisms.

The vanes in the axial vane mechanism, shown in figure 1, follow the cam surfaces in the end housings, alternately compressing and/or expanding fluids depending upon the application of the mechanism. In this particular mechanism, the exterior portion of the vane at the outer edge of the rotor travels on the exterior housing such that both the axial motion of the vane and the rotational motion of the rotor create friction. The resultant rubbing velocity limits the maximum rotational speed of the mechanism due to limitations on the rubbing velocity of the vane/housing and/or the vane/rotor materials. In addition, leakage of compressed fluids can occur at the tip of the vanes and along both the top (exterior) and bottom (interior) edges of the vanes.

The “Winged Rotor” concept developed by ATI eliminates or greatly minimizes leakage along the top and bottom of the vanes and de-couples the vane rubbing velocity from the effects of rotational speed. An illustration of the ATI Winged Rotor Concept (Patent Pending) is shown in figure 2. In the Winged Rotor device a cylindrical sleeve is attached to the outer periphery of the rotor and grooves in the sleeve and a similar “Wing” on the main shaft minimize leakage

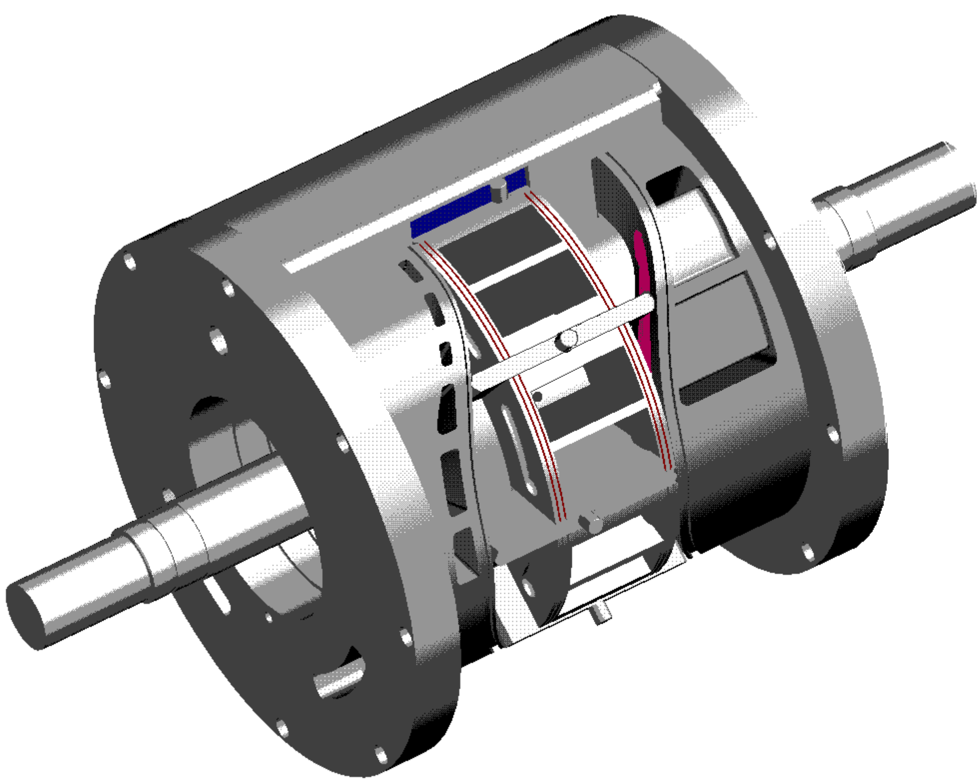


Figure 1.—Cut-away of typical axial vane rotary device.

over the top and under the bottom of the vane. A major advantage of the Winged Rotor Concept is that it decouples the vanes from the rubbing velocity caused by rotation of the rotor and permits much higher rotational speeds that yield much higher specific power density. The maximum allowable rubbing velocity between the vanes and rotor or the vanes and outer sleeve determines the limiting rotational speed of the ATI Winged Rotor device. The decoupling of the vanes from the rotational rubbing velocity permits the maximum rotational speed to be increased by about five-fold in engine applications. More detailed explanations of the construction and operation of the ATI Axial Vane Engine and AV-components are provided in appendix B.

The higher rotational speeds made possible with the ATI Winged Rotor Mechanism permit the design of weight sensitive devices with significantly increased specific power density. This feature is especially important in weight-critical aviation applications such as compressors, expanders, or Diesel engines. The design programs discussed in the instruction manual (appendix B) and in the design programs are based on utilization of the ATI Winged Rotor Concept to minimize friction losses, leakage, heat losses, and installed weight and to maximize power density, reliability, and life of the engine/device. These analysis and design programs were used in this feasibility study to analyze the performance of the various axial vane devices.

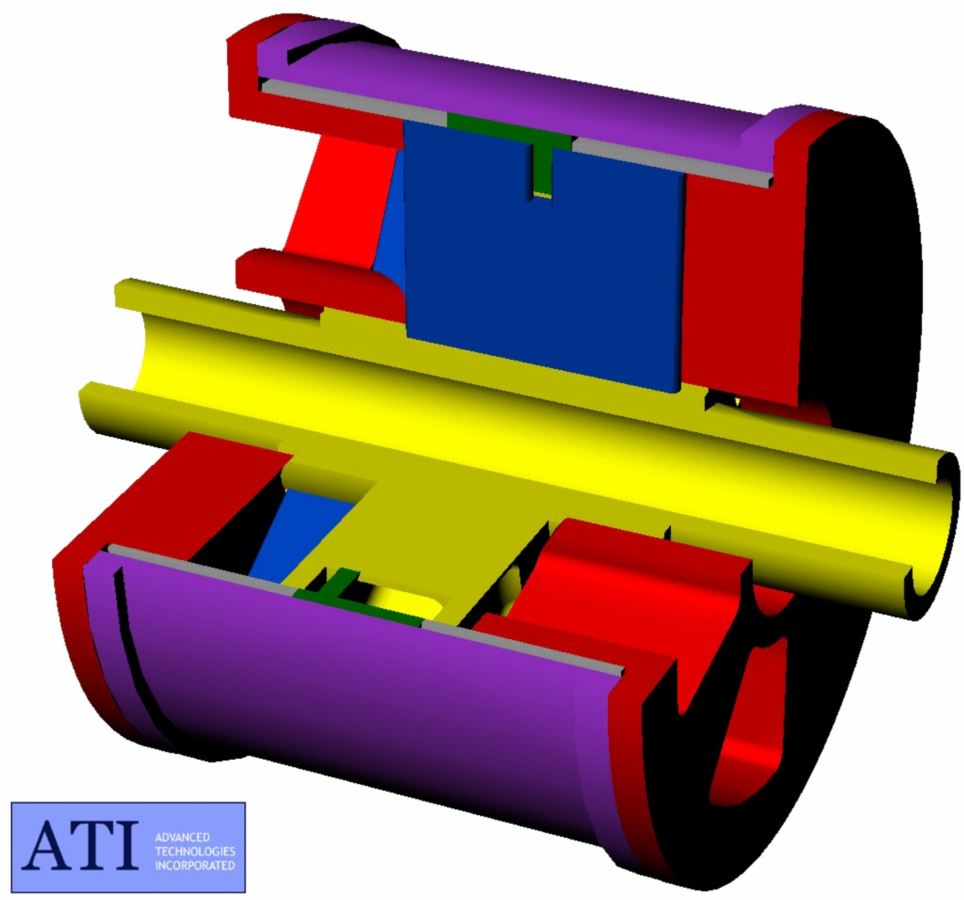


Figure 2.—Cut-away of ATI winged rotor axial vane device.

3.0 Design of Axial Vane Diesel Cycle Engine

3.1 Description of the Axial Vane Rotary Diesel Cycle Engine

A schematic of a twelve-vane rotary Diesel engine showing the cylindrical outer diameter surface of the rotor unwrapped onto a plane surface is shown in figure 3. The vanes are shown in their slots in the moving rotor. The fixed pin track is shown in the outer housing. The combustion clearance volumes are shown schematically by the two parallel lines on the face of the rotor and are better seen as the pockets on the face of the rotor in figure 1.

The main and auxiliary intake ports are used to effect the Miller cycle. When the auxiliary intake port is kept closed, the intake is shut off early in the cycle before the chamber achieves maximum volume. This lowers the effective compression ratio while leaving the expansion ratio unchanged. There are twelve chambers on each side of the diesel engine for a total of 24 chambers. The standard four-stroke cycle is completed in each chamber every engine revolution which gives the engine twenty-four widely overlapping power strokes per revolution. Each chamber is 30° wide and the power stroke occurs over 60°. This is equivalent to a four-stroke reciprocating engine with 48 cylinders. The engine torque and exhaust flow are almost steady which results in the low noise levels of the engine.

The ATI Axial Vane Rotary Diesel Engine employs a true compression ignition diesel cycle running at an extremely lean fuel/air ratio and compression ratios of 15:1 or higher. The engine utilizes continuous fuel injection and extended duration combustion. Both of these concepts are unique to this engine design and account for the extraordinary thermal efficiency. This cycle also approximates constant volume combustion, which has been a goal in internal combustion engines since Sadi Carnot first identified the constant volume cycle as having the highest possible theoretical efficiency.

3.2 Diesel Cycle Modeling

The first task in this project was to develop a cycle simulation for the Axial Vane Diesel Engine. It was decided that a "Spreadsheet" type simulation would be written to include the effects of leakage, combustion, and heat transfer using physical geometry, cam shape and ambient conditions as inputs. To provide additional background on the operation of the axial

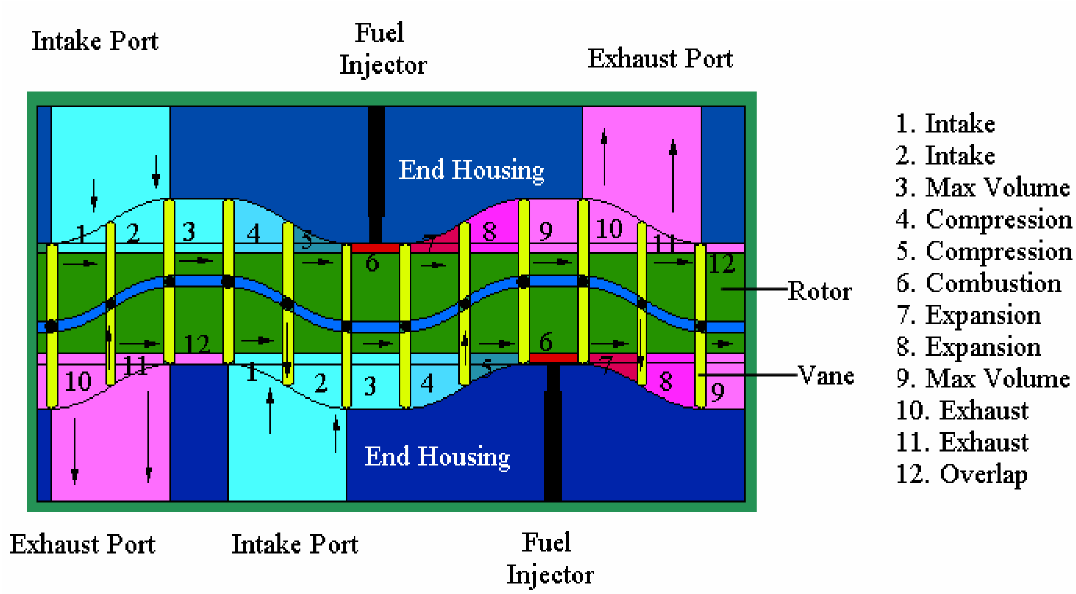


Figure 3.—Schematic of axial vane rotary diesel engine.

vane devices, the simulation tools, and to make it easy to select inputs, an “Instruction Manual for Axial Vane Mechanism Design Program” was prepared and is attached as appendix B to this report. This simulation tool was used to design Diesel engines suitable of serving as core engines for the PAX 50 and PAX 150 turbofan engines. A sample run for a Diesel engine is included in the "Instruction Manual" (appendix B) along with details of how the computational model was constructed and the detailed printout for each 1° of rotation of the engine. Due to the large size of the spreadsheet program (24 letter-size sheets), a Summary Sheet was included in the program to document all program inputs and the results of the engine sizing analysis. A printout of the Summary Sheet is obtained by selecting sheet "2 PRINOUT” of the spreadsheet and printing out the displayed information.

3.2.1 Design of Axial Vane Diesel Engines at Sea Level Conditions

The axial vane design program was used to design axial vane Diesel engines with the same fan power as the current baseline PAX 50 and PAX 150 high bypass ratio turbofan engines at sea level conditions. To assure these designs were conservative; it was assumed the leakage flow was always sonic despite the fact that for a large portion of the cycle time the pressure ratio would have warranted a lower leakage flow rate. Also, the wall temperature for the heat transfer was fixed at 500 °R. This assumption results in heat transfer losses that are higher than they would have been if higher wall temperatures had been used and results in a conservative estimate of net power output. The major constraint on the engine designs was to maintain the existing turbofan nacelle size. This limited the rotor diameter of the PAX 50 Diesel Core Engines to 20 in. and the PAX 150 engines to 40 in.

Axial Vane PDQ, a “Microsoft Visual Basic” program developed by ATI, was used to identify possible engine configurations that would meet the above constraints. The AV-PDQ Program is based on thermodynamic analysis and does not include the effects of friction, leakage, or cooling losses. However, the program is useful for quickly conducting preliminary analyses of multiple configurations that may meet the design constraints. The AV-PDQ program can be used for sizing axial vane Diesel engines, compressors or expanders.

The AV-PDQ program was used in a preliminary analysis of rotary Diesel engine configurations that could be candidates for use in the PAX 50 and PAX 150 engines. This analysis was run to determine the effect of engine rotor diameter and BMEP on the power output of various size rotary diesel units. Examples of the output from the AV-PDQ Program are shown in figures 4 and 5 for the PAX 50 and PAX 150 engines. The results of this preliminary analysis are presented in figure 6. The specifications for the baseline PAX 50 and PAX 150 turbofan engines are listed in the table. In this analysis engine RPM is limited by the maximum rubbing velocity of the vanes on the wings of the rotor and assuming the use of conventional materials for the vane and the wings. The Vane Velocity shown in the output from the AV-PDQ Program is based on the assumption the vanes are running on the cam surface and the resultant velocity is the vectorial sum of the rotor peripheral velocity and the vane transverse velocity. The use of low-friction coatings and/or thermal barrier coatings could increase the operating speed substantially.

The range of BMEP analyzed varied from 100 psi to 300 psi. Based on prior experience, it was estimated that BMEP of 140 to 170 could be obtained in a normally aspirated AV-Diesel Engine and that turbocharging or super charging will be required to obtain pressures in the 200 to 300 psi range. At the outset of this study, it was anticipated that multiple diesel engine units would likely be required to produce the required power while keeping the engine diameter within the prescribed limits. The maximum rotor diameter was limited to about 80 percent of the diameter of the Low Pressure Compressor (LPC) in the turbofan engine to provide space for induction and exhaust manifolds along the sides of the engines.

Using a conservative 140 BMEP, there appear to be two potential engine configurations that could be candidate configurations for the PAX 50 and PAX 150 Diesel engines. For the PAX-50 engine, a 20 in. and a 25 in. rotor diameter could provide the required 8200 BHP by using three 20-in.-diameter rotor engine units or two 25 in. units. Since the 25 in. diameter exceeds 80 percent of the LPC diameter, the 20-in. diameter unit was selected for further analysis using the AV-Design Program. Similarly, the 40-in. diameter engine unit was selected for further analysis for the PAX 150 core engine.

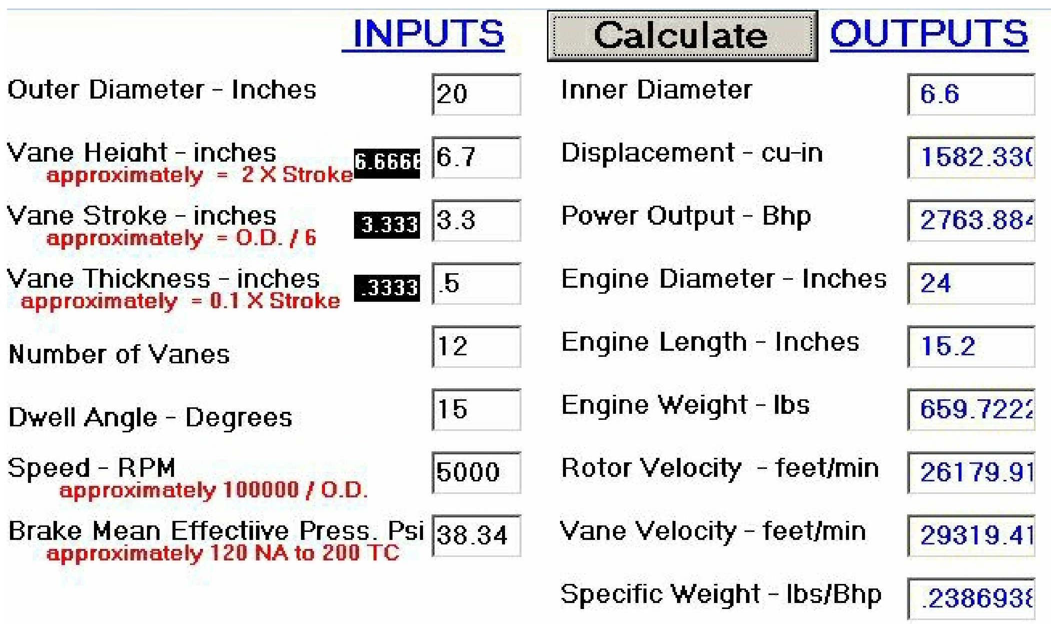


Figure 4.—AV-PDG analysis for PAX 50 diesel engine analysis. (Note: BMEP = 138.34 psi.)

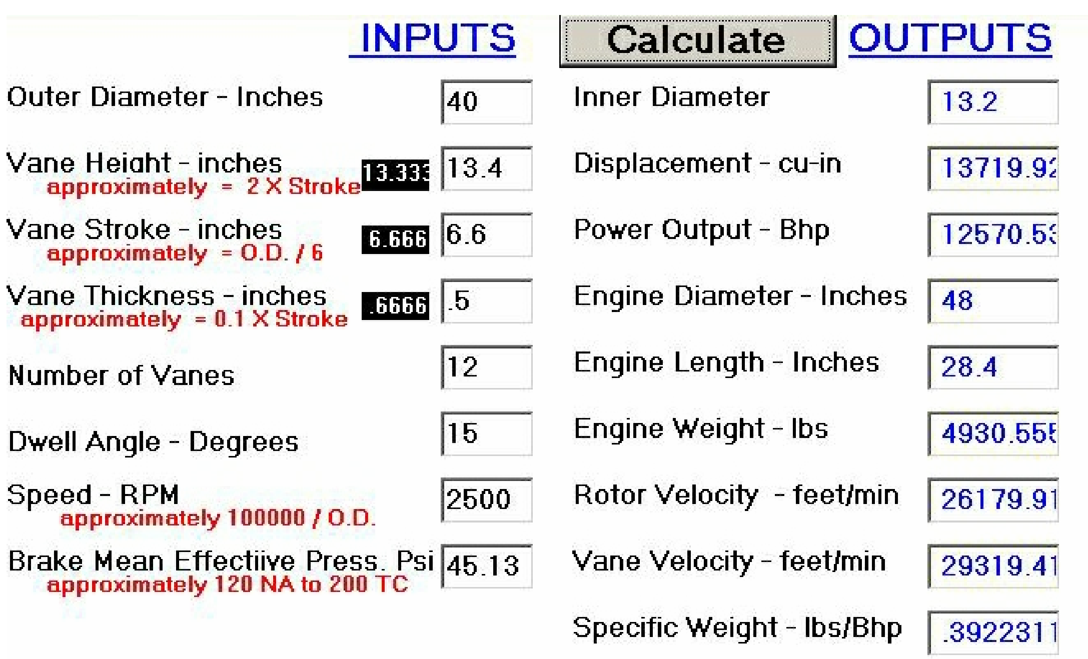


Figure 5.—AV-PDQ analysis for PAX 150 diesel engine. (Note: BMEP = 145.13 psi.)

Axial-Vane Diesel Engine Analysis using AV-PDQ Program

AV-PDQ, Version #5, 050531
Performed By: Mike Smith
16-Jun-05
Purpose of Analysis: To Determine the Effect of Engine Rotor Diameter & BMEP on Power Output
at Sea Level Conditions

		Estimated Brake Horsepower				
Rotor Diameter inches	**RPM**	**BMEP = 100**	**BMEP = 140**	**BMEP = 170**	**BMEP = 200**	**BMEP = 300**
3	33,300		67		81	
5	20,000		154		265	
10	10,000		770		1,055	
15	6,670		1,030			
20	5,000		3,024		4,260	
25	4,000		4,779			
30	3,330	4,779	6,786	7,124	9,558	14,337
35	2,860		9,265			
40	2,500		12,059		17,013	
45	2,220		15,269			
50	2,000	13,298	19,004	22,607	26,597	39,896
55	1,820		22,910			
60	1,670		27,226		38,347	

NASA GRC SPECS FOR PAX BASELINE ENGINES	
PAX 50 Equiv. 8,200 BHP	**PAX 150 Equiv.** 23,700 BHP
Fan & LPT = 18,000 RPM	Fan & LPT = 14,500 RPM
HPC & HPT = 7,000 RPM	HP Comp.& HPT = 5,380
Fan Max. Dia = 38.5 in.	Fan Max.Dia. = 68.3 in.
Max.Eng.Length = 109 in	Max.Eng.Length = 170.9 in.
POTENTIAL ENGINE CONFIGURATIONS @ BMEP=142	
3 - 20" OD x 15.4" Long	
2 - 25" OD x 18.8" Long	
	4 - 30" OD x 22" Long
1 - 35" OD x 25.4" Long,	3 - 35" OD x 25.4" Long
	2 - 40" OD x 28.8" Long
Max. LPC Dia. = 24 in.	**Max. LPC Dia. = 45 in.**

To be considered as a Potential PAX Engine Configuration
Max. Rotor OD must be = or < 0.80 Max.LPC.Dia. on Spec Dwg.

Light Blue indicates potential PAX 50 engine configuration.
Light Orange indicates potential PAX 150 engine configuration.

Figure 6.—Preliminary analysis of the effect of engine rotor diameter and BMEP on power output at sea level conditions.

3.2.2 PAX 50 Diesel Core Engine at Sea Level Conditions

The AV-Design Program was used to compute the performance of the Diesel engines to be used as the core engine for the PAX 50 Engine. The results of the final analysis are presented in figure 8 for a normally aspirated engine at sea level conditions. To meet the by-pass fan power requirement for 8132 hp, it was necessary to use three diesel units producing 2764 hp each and a total of 8293 hp. The engine rotor diameter of 20-in. resulted in a unit engine length of 15-in. A cross-section drawing of the PAX 50 engine with the three Diesel units is shown in figure 7.

Solid models of the components of the PAX 50 engine were constructed and assembled into a single rotor power module and into the complete three rotor core engine. An exploded view of a typical diesel engine module is shown in figure 9.

The single Diesel engine illustrated in figure 9 employs a slot-and-rail system of driving the vanes so the vanes do not actually rub on the cam surface. The green track incorporates an internal rail that has the same geometry as the cam surface while each vane has a slot that rides on an internal rail. The purpose of the slot-and-rail feature is to assure the vane tips do not rub on the cam surfaces that may be coated with thermal barrier materials or cause wear or galling of the cam surface. The wing components attach to the periphery of the rotor and extend over the cams to minimize gas leakage. The end cams attach to the outer housing to produce a leak-proof outer seal.

The assembly of the three Diesel engine units for the PAX 50 core engine is presented in figure 10. The three engines are mounted on a single splined shaft and the fan or gear box will mount to the extended shaft. The intake ports and exhaust ports are shown in the ends of the cams. Induction and exhaust ducts will be required to connect the engines axial ports. The space allocated between the engine units should provide space for the required ducts and accessories can be mounted on the rear unit.

A rendering of the three rotor core engine installed in the PAX 50 nacelle is shown as figure 11. A weight schedule and bill-of-materials for the engine is shown in table 1 including weights for each component and the assembly. For the weight estimate, it was assumed that the entire engine would be constructed of titanium alloys. It should be noted that no attempt has been made to weight reduce any of the components. This along with the fact that the final engine would likely incorporate ceramic and ceramic composite components means that the included weight estimate is conservative.

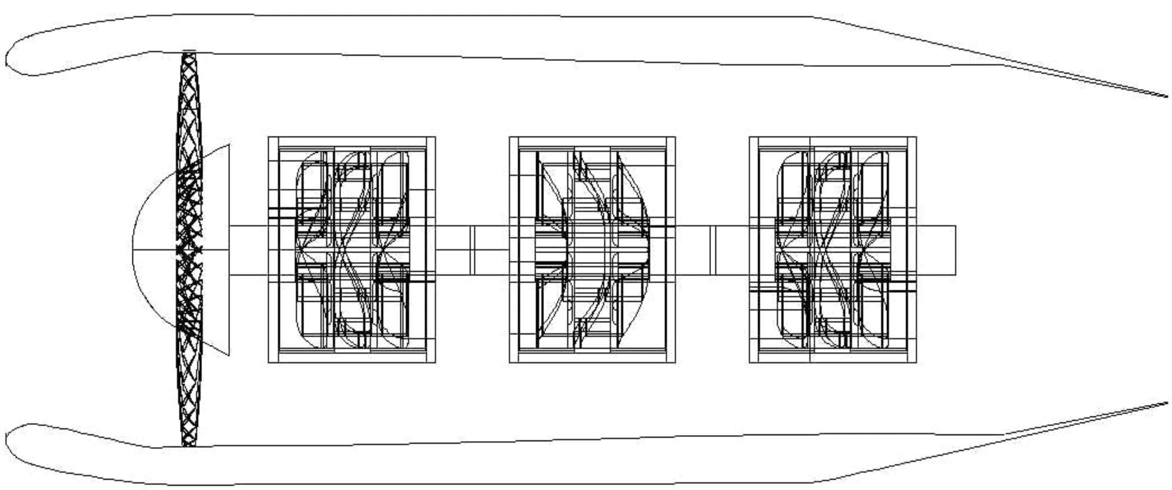


Figure 7.—Cross-section of PAX 50 engine with 3-rotor diesel core engine.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

8,200 Horsepower 3 Rotor Diesel Engine

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	3.30	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	6.70	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	20.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	6.60	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	13.30	
INPUT DESIGN VALUES		
SPEED (RPM)	5,000	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	524	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	14.70	Inlet Pressure
INITIAL TEMPERATURE (deg R)	530.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	15,711	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	65.94	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	4.88	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.1166	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	1,583	Total Displacement of 24 chambers.
COMPRESSION RATIO	14.21	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	343.16	
FUEL FLOW (lbs/min)	13.38	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	138.34	
INDICATED POWER (hp)	3,409	
FRICTION POWER (hp)	644	
HEAT LOSS POWER (hp)	529	
BRAKE HORSEPOWER OUTPUT (Bhp)	2,764	8,293.2 Bhp for 3 Rotary Diesel Units
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.2905	
EXHAUST TEMPERATURE (deg. F.)	1643	
Performed By:	Pat Badgley	Date Performed: June 28, 2005

Figure 8.—Performance analysis for PAX 50 rotary diesel core engine.

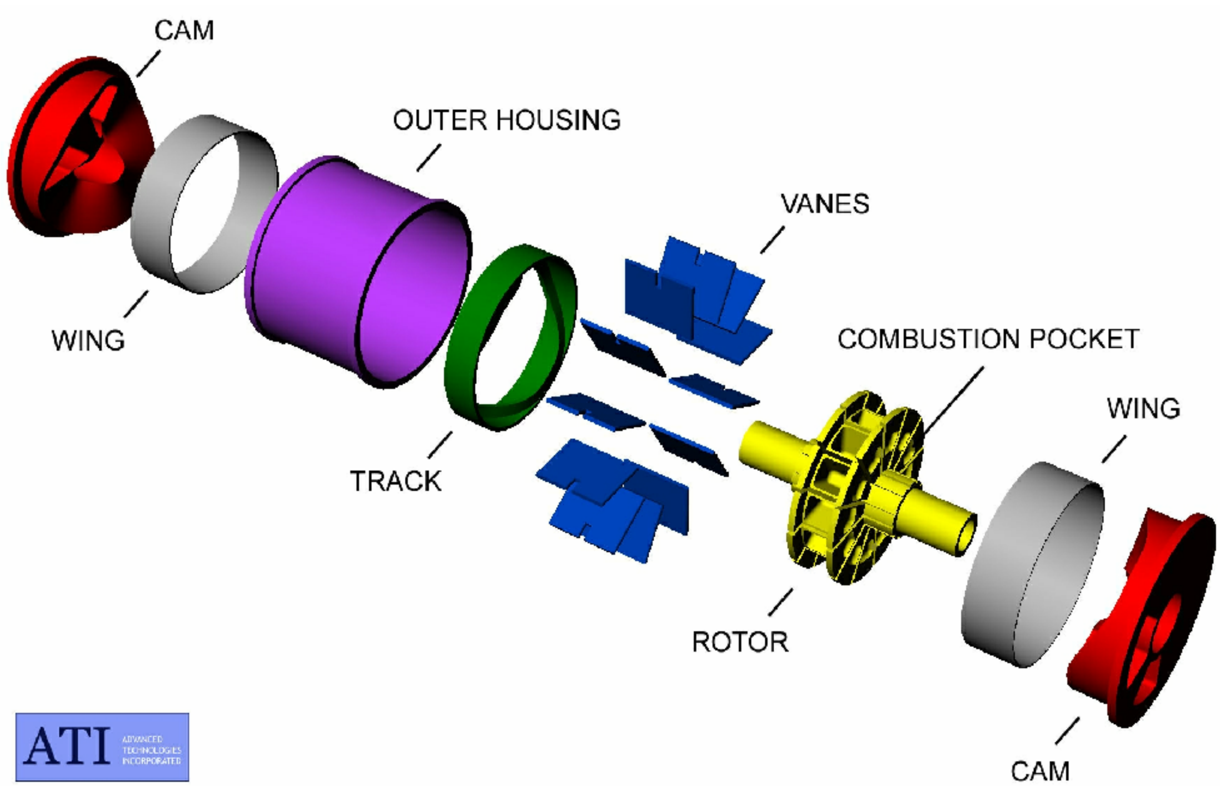


Figure 9.—Exploded view of axial vane rotary diesel engine.

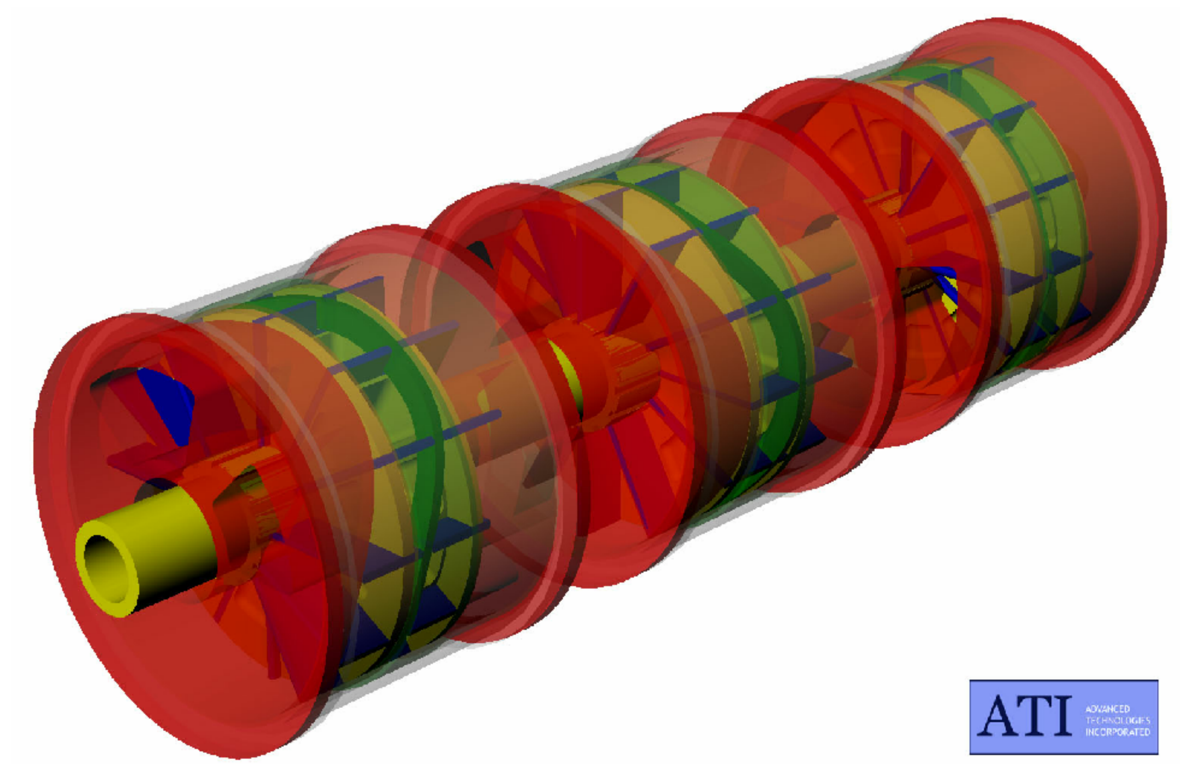


Figure 10.—Transparent model of PAX 50 engine with 3-rotor diesel core engine.

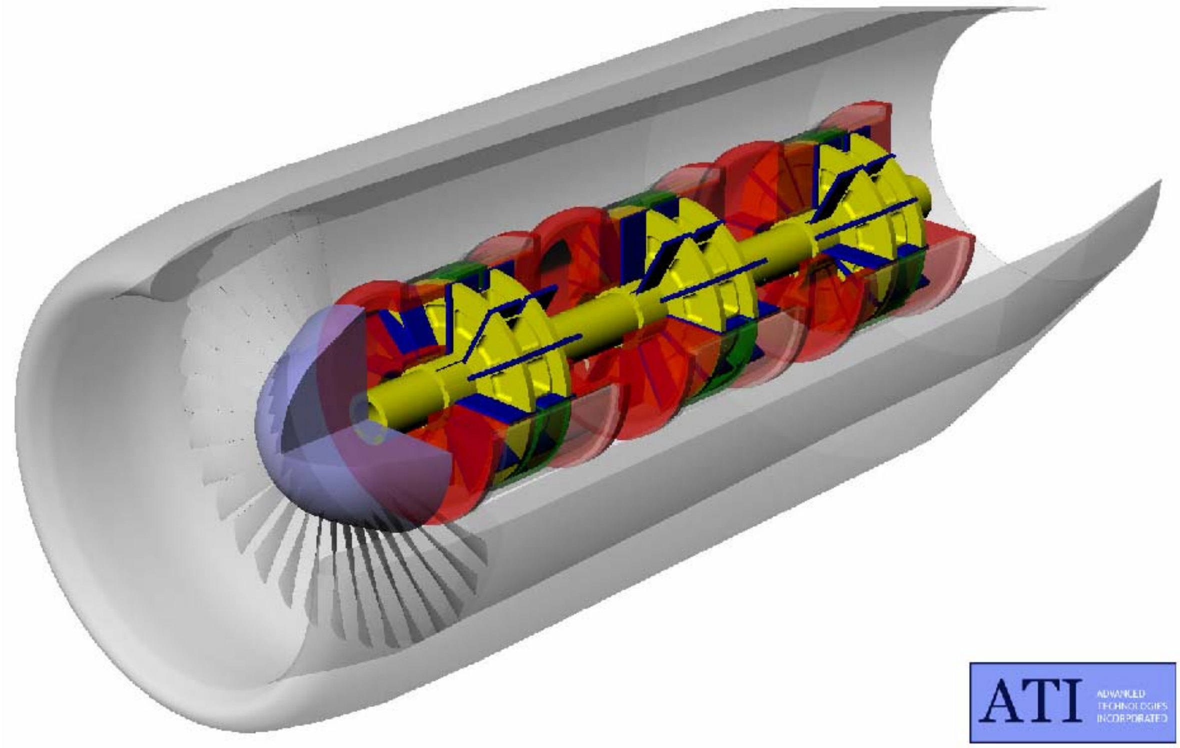


Figure 11.—Rendering of the PAX 50 engine with a 3-rotor diesel core engine.

TABLE 1.—WEIGHT ESTIMATE AND BILL OF MATERIALS
FOR PAX 50 DIESEL CORE ENGINE

Engine Weight for PAX 50 3-Rotor Diesel Core Engine

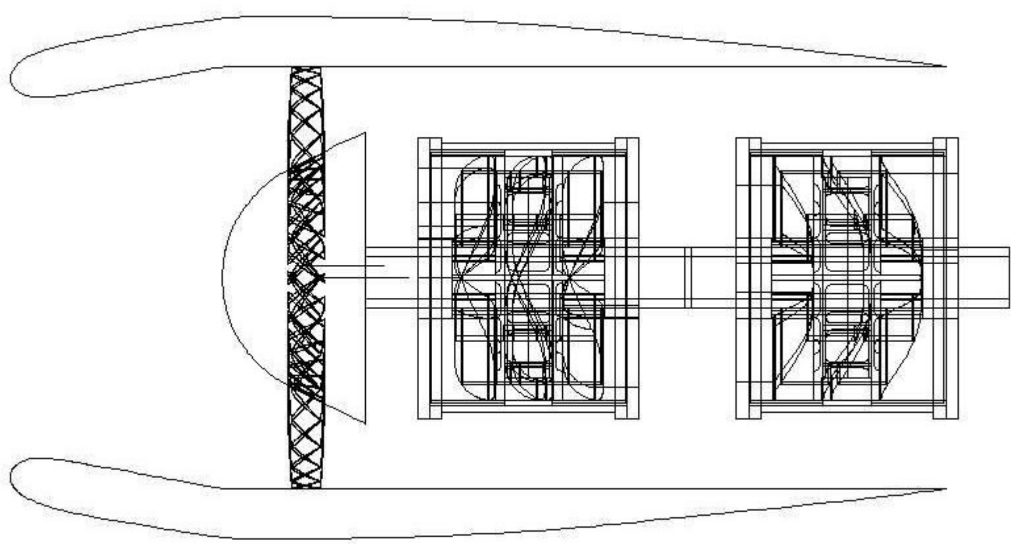
PAX 50 ENGINE with 3-20" x 15" Axial Vane Diesel Engines

Part	Number/ Module	Weight/ Part	Weight/ Module	
Cam	2	45.8	91.6	
Rotor	1	81.2	81.2	
Vane	12	5	60	
Wing	2	16	32	
Rail	1	20	20	
Rail Support	1	21	21	
Outer Housing	1	103	103	
Bare Weight One Diesel Unit			408.8	lbs
Bare Weight 3-Rotor Diesel Core Engine			**1,226.4**	**Lbs**
Estimated Accessory Weight (7% Bare Wt.)			**85.8**	**Lbs**
Estimate Total Weight PAX 50 3-Rotor Diesel			**1,312.2**	**Lbs**
Assuming all parts are made of titanium				

3.2.3 PAX 150 Diesel Core Engine at Sea Level Conditions

The AV-Design Program was also used to compute the performance of the Diesel engines to be used as the core engine for the PAX 150 Engine. The results of the final analysis are presented in figure 13 for a normally aspirated Diesel core engine at sea level conditions. To meet the by-pass fan power requirement for 23,664 hp, it was necessary to use two diesel units producing 12,560 hp each and a total of 25,120 hp. A cross-section drawing of the PAX 150 engine with the two Diesel core units is shown in figure 12. The 2-rotor Diesel core engine is slightly longer than the nacelle of the baseline PAX 150 turbofan engine. The 40-in. diameter of the rotor also results in an overall diameter of the engine (about 48 in.) that may restrict airflow from the turbofan when induction and exhaust ducts are added to the engine.

Turbocharging the engine could reduce the diameter of the engine, but would not significantly reduce the length of the engine or the space between engine units to connect the required ducts and accessories. To reduce the length of the core engine, it was suggested by the project COTR that the use of radial induction and exhaust porting be investigated to reduce the overall length of multi-unit core engines. This analysis was conducted and is discussed in section 3.4.



ATI ADVANCED TECHNOLOGIES INCORPORATED

Figure 12.—Cross-section of PAX 150 engine with diesel core engine.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

25,000 Horsepower 2 Rotor Diesel Engine

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	6.60	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	13.40	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	40.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	13.20	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	26.60	
INPUT DESIGN VALUES		
SPEED (RPM)	2,500	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	262	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	14.70	Inlet Pressure
INITIAL TEMPERATURE (deg R)	530.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	16,400	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	571.20	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	44.53	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.4666	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	13,709	Total Displacement of 24 chambers.
COMPRESSION RATIO	13.63	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	1,486.18	
FUEL FLOW (lbs/min)	57.96	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	145.13	
INDICATED POWER (hp)	15,170	
FRICTION POWER (hp)	2,610	
HEAT LOSS POWER (hp)	1,774	
BRAKE HORSEPOWER OUTPUT (Bhp)	12,560	25,120 BHP for 2-Rotor Core Engine
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.2769	
EXHAUST TEMPERATURE (deg. F.)	1597	
Performed By:	Mike Smith	Date Performed: June 20, 2005

Figure 13.—Performance analysis for PAX 150 rotary diesel core engine.

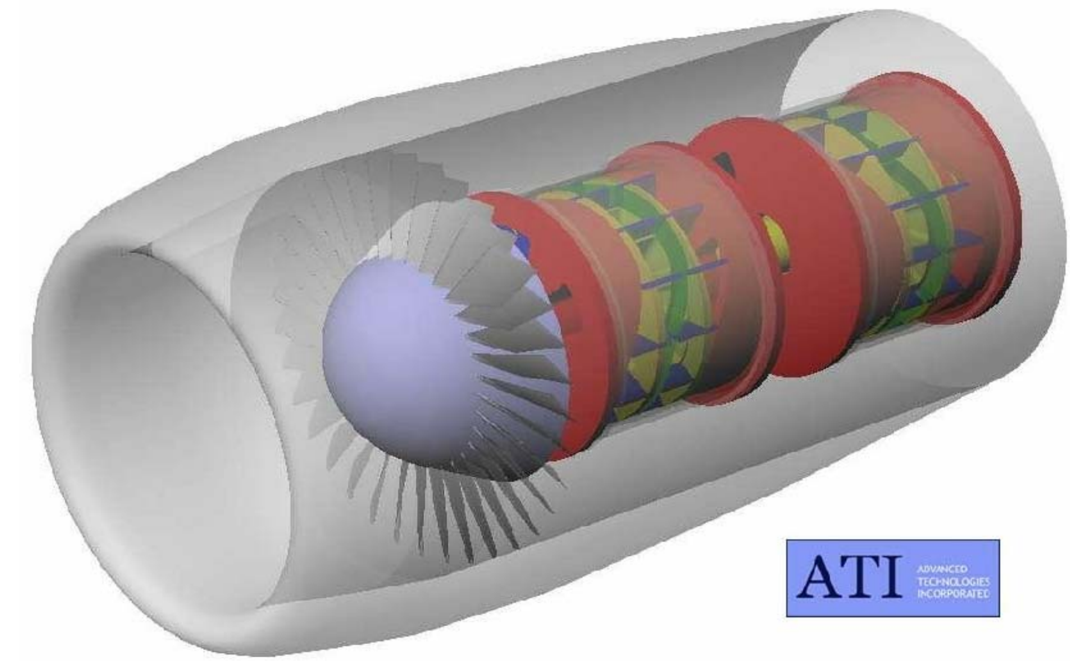


Figure 14.—Rendering of PAX 150 engine with 2-rotor diesel core engine.

TABLE 2.—WEIGHT ESTIMATE AND BILL OF MATERIALS AND FOR PAX 50 & PAX 150 DIESEL ENGINES

Weight Estimates for PAX 50 and PAX 150 Axial Vane Diesel Core Engines

Assumes All Parts are Made of Titanium

Weight Units are Pounds

		PAX 50 Axial Vane Diesel Engine		**PAX 150 Axial Vane Diesel Engine**	
Part	**Number/ Module**	**Weight/ Part**	**Weight/ Module**	**Weight/ Part**	**Weight/ Module**
Cam	2	46	92	210	421
Rotor	1	81	81	373	373
Vane	12	5	60	23	276
Wing	2	16	32	74	147
Rail	1	20	20	92	92
Rail Support	1	21	21	96	96
Outer	1	103	103	473	473
	Total per Unit		408.8	Total per Unit	1,878
Total Weight Per Unit		**3 Rotors**	1,226	**2 Rotors**	3,757
Accessories	7%		86	8%	301
TOTAL WEIGHT CORE ENGINE			1,312		4,057

3.3 Evaluation of Axial Versus Radial Intake and Exhaust Porting on Rotary Devices

The project COTR suggested investigation of the potential use of radial intake and exhaust porting on the axial vane components in an attempt to reduce the overall diameter and length of the engine units. Investigation of this concept revealed that radial porting could be used on axial vane Diesel engines, compressors and expanders. An example of radial porting on the PAX 150 Diesel core engine unit is shown in figure 15.

Radial porting permits the axial vane units to be connected directly to each other which makes the Diesel core engine unit shorter and stiffer on the single drive shaft. In addition, the ducts for intake air and exhaust gases are shorter and have one less 90° bend at each port thereby increasing the aerodynamic efficiencies of the induction and exhaust systems. The radial ports also provide for easier assembly and disassembly of the ducts. The diameter of the engine housing is increased slightly in order to connect the engine units at the periphery of the cam end housings. Accessories can be mounted on the rear of the core engine.

Use of radial ports on the axial vane devices reduces the complexity of assembly and disassembly of the core engine units and should increase the reliability of the engine units due to the simpler system connections. A cross-section of the PAX 150 Diesel core engine with radial ports is presented in figure 16 and a solid model of the engine mounted in the original PAX 150 nacelle is shown in figure 17.

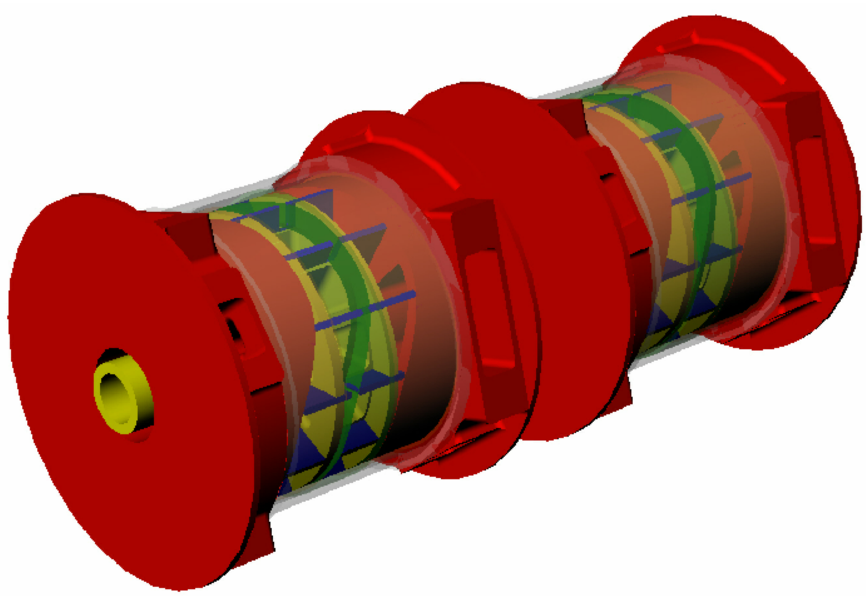


Figure 15.—Example of radial intake and exhaust ports on diesel engine.

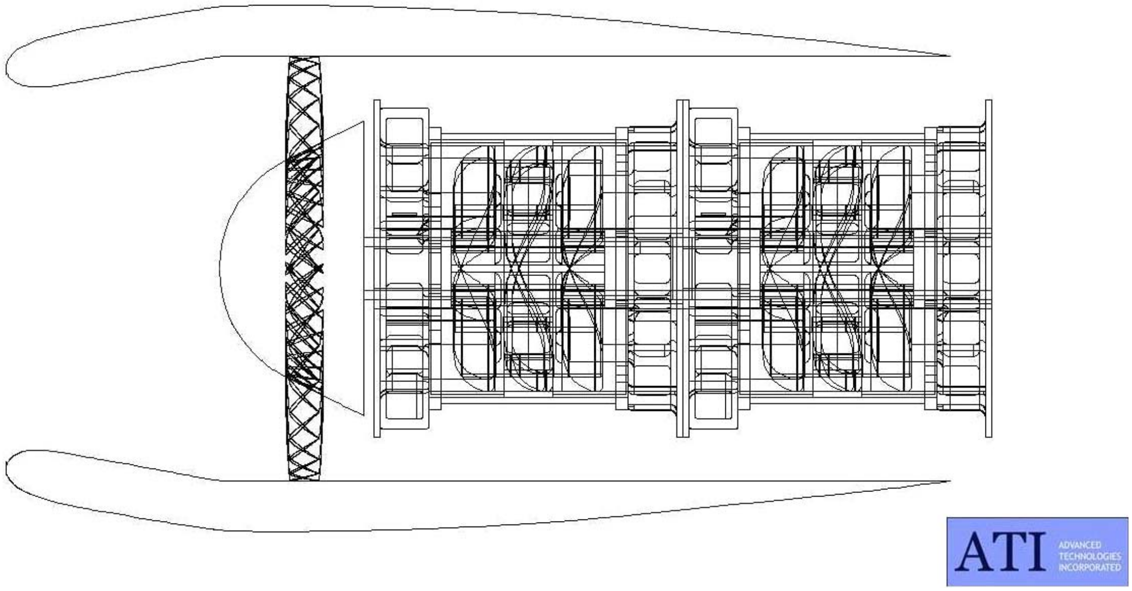


Figure 16.—Cross-section PAX 150 diesel core engine with radial porting.

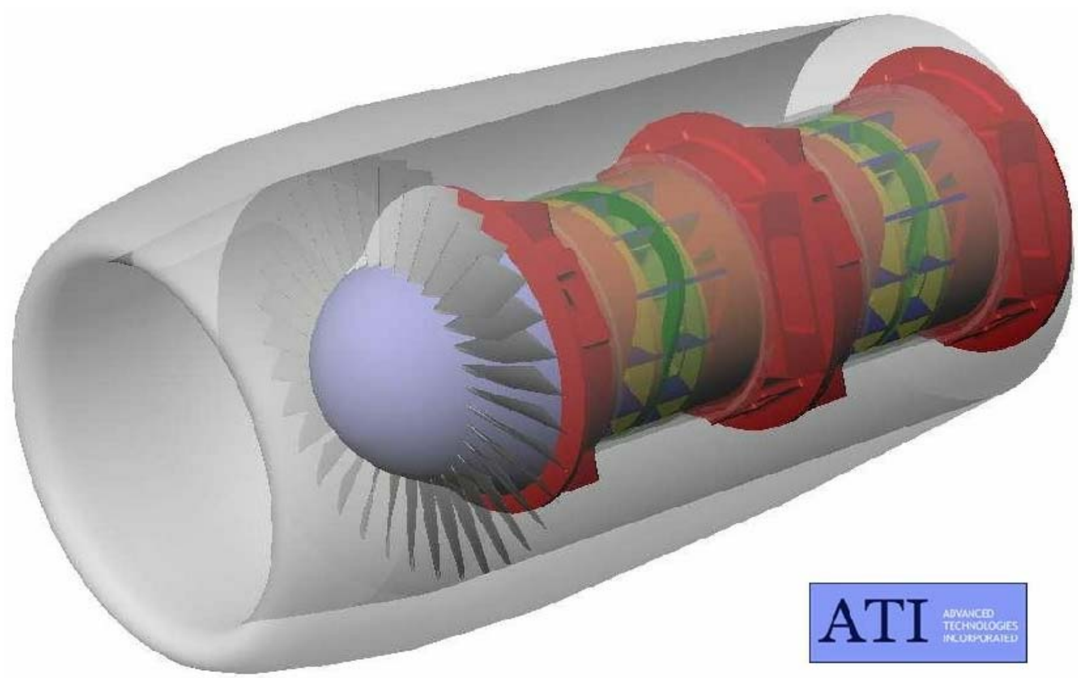


Figure 17.—Solid model of PAX 150 diesel core engine with radial porting.

3.4 Design of Diesel Core Engine Using Turbofan Pressure Boost

The statement of work for this feasibility study called for the analysis of the axial vane devices to be conducted for sea level conditions and this was accomplished in section 3.3. However; in analyzing the design of the PAX 50 and PAX 150 engines it was noted that the inlet for the turbofan core engines operated downstream of the by-pass fan and in an area of significantly increased pressure (23 psi versus 14.7 psi). Recognizing the increased air induction pressure would increase the specific power density and/or reduce the core engine size, it was decided to conduct performance analyses for the PAX 50 and PAX 150 engines using the higher pressure downstream of the fan section. These analyses are presented in figures 18 and 19 for core rotor diameters equal to the configurations evaluated at sea level. A comparison of the size, weight, and performance characteristics of the diesel core engines at sea-level and at the fan boosted pressure is presented in table 3.

TABLE 3.—COMPARISON OF ENGINE PERFORMANCE AT SEA-LEVEL VERSUS FAN BOOSTED PRESSURE

PERFORMANCE COMPARISON FOR SEA-LEVEL vs. FAN BOOSTED PRESSURES

PARAMETER	UNITS	PAX 50 ENGINE		PAX 150 ENGINE	
		DIESEL Sea Level	DIESEL BOOSTED	DIESEL Sea Level	DIESEL BOOSTED
ROTOR DIAMETER	Inches	20.0	20.0	40.0	40.0
NUMBER ROTOR UNITS	---	3	2	2	2
ENGINE SPEED	RPM	5000	5000	2500	2500
INLET PRESSURE	Psia	14.70	23.30	14.70	23.05
POWER	BHP	8,293	10,117	25,120	32,870
Increase		---	122%	---	131%
FUEL/AIR RATIO	---	0.03900	0.03900	0.03900	0.03900
AIR/FUEL RATIO	---	25.64	25.64	25.64	25.64
AIRFLOW RATE	Lb./Min	1,029	1,356	2,972	3,896
EXHAUST GAS TEMP	Deg. F	1643	2199	1597	2136
BSFC	Lb/BHP/Hr.	0.2903	0.3140	0.2769	0.2773
NUMBER ROTOR UNITS	---	3	3	2	2
BARE ENGINE WEIGHT	Lb.	1,226	1,226	3,757	3,757
ACCESSORIES WT.	Lb.	86	86	301	301
TOTAL CORE ENGINE WT.	Lb.	1,312	1,312	4,058	4,058
POWER DENSITY	BHP/Lb.	6.32	7.71	6.19	8.10
IMPROVEMENT BOOSTED vs. Sea Level Pressure			122%	---	131%

Note 1: DIESEL configuration is computed at Standard Sea Level Pressure (14.7 psia) and Temperature (530 Deg. Rankine).

Note 2: DIESEL BOOSTED is computed at Turbofan exit condition [inlet pressure (23.045/23.05 psig) and temperature (634-638 R)].

Note 3: Estimated Accessories Weight for DIESEL & DIESEL BOOSTED Configuration [PAX 50 = 0.7x Bare Wt.; PAX 150 = 0.08x Bare Wt.]

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

Fan Boosted 3-Rotor Axial-Vane Diesel Engine for 8,200 BHP PAX-50 Turbofan Engine

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	3.30	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	6.70	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	20.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	6.60	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	13.30	
INPUT DESIGN VALUES		
SPEED (RPM)	5,000	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	524	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between Vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	23.30	Inlet Pressure behind Turbofan
INITIAL TEMPERATURE (deg R)	637.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	14,378	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	65.94	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	4.88	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.1166	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	1,583	Total Displacement of 24 chambers.
COMPRESSION RATIO	14.21	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	452.55	
FUEL FLOW (lbs/min)	17.65	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	168.75	
INDICATED POWER (hp)	4,016	
FRICTION POWER (hp)	644	
HEAT LOSS POWER (hp)	912	
BRAKE HORSEPOWER OUTPUT (Bhp)	3,372	3 Units = 10,117
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.3140	
EXHAUST TEMPERATURE (deg. F.)	2199	
Performed By:	Mike Smith	Date Performed: 12-18-05

Figure 18.—Performance analysis of PAX 50 diesel engine with fan boost pressure.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

Fan Boosted PAX 150 2-Rotor Diesel Core Engine (40"@2500)

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	6.60	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	13.40	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	40.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	13.20	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	26.60	
INPUT DESIGN VALUES		
SPEED (RPM)	2,500	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	262	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	23.05	Inlet Pressure
INITIAL TEMPERATURE (deg R)	634.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	15,494	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	571.20	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	44.53	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.4666	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	13,709	Total Displacement of 24 chambers.
COMPRESSION RATIO	13.63	Ratio of Max. Real Vol. / Min. Real Vol.(Sht 1, Col. Q)
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	1,947.68	
FUEL FLOW (lbs/min)	75.96	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	189.90	
INDICATED POWER (hp)	18,444	
FRICTION POWER (hp)	2,009	
HEAT LOSS POWER (hp)	3,079	
BRAKE HORSEPOWER OUTPUT (Bhp)	16,435	32,870 BHP for 2-Rotor Core Engine
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.2773	
EXHAUST TEMPERATURE (deg. F.)	2136	
Performed By:	Mike Smith	Date Performed: Dec. 21, 2005

Figure 19.—Performance analysis for PAX 150 diesel core engine with fan boost.

3.5 Minimum Size PAX 50 and PAX 150 Engines with Fan Boosted Pressure

A second alternative with the availability of the higher fan boosted pressure is to reduce the diameter of the axial vane engines while producing the specified power for each installation. The performance analysis for the minimum size PAX 50 engine is presented in figure 20. The rotor diameter of the PAX 50 Diesel engine was reduced from 20.0 in. at sea-level pressure to 17.0 in. at the boosted pressure of 23.3 psig. The 15 percent reduction in diameter also permitted an 18 percent increase in engine speed and a 39 percent increase in specific power density to 8.80 BHP/lb while decreasing engine weight by 22 percent.

The performance analysis for the minimum size PAX 150 engine is presented in figure 21. The rotor diameter was reduced from 40.0 to 36.5 in. for a 9 percent reduction while increasing the specific power density 40 percent to 8.72 BHP/lb and reducing the engine weight by 34 percent.

The engine size reductions permitted by utilizing the fan boosted pressure is significant in that it makes the overall diameter of the Diesel engines with accessories more likely to fit within the limits of the existing nacelles and the engine weight reductions are highly significant. A summary of the engine analysis for the minimum size engines is presented in table 4.

TABLE 4.—COMPARISON OF MINIMUM-SIZE BOOSTED ENGINE WITH SEA-LEVEL ENGINES

COMPARISON OF CORE ENGINE TYPES FOR PAX 50 AND PAX 150 TURBOFAN ENGINES

PARAMETER	UNITS	PAX 50 ENGINE			PAX 150 ENGINE		
		DIESEL CORE	DIESEL BOOSTED	MIN.SIZE BOOSTED	DIESEL CORE	DIESEL BOOSTED	MIN.SIZE BOOSTED
ROTOR DIAMETER	Inches	20.0	20.0	17.0	40.0	40.0	30.5
NUMBER ROTOR UNITS	---	3	3	3	2	2	2
ENGINE SPEED	RPM	5000	5000	5900	2500	2500	3600
INLET PRESSURE	Psia	14.70	23.30	23.30	14.70	23.05	23.05
POWER	BHP	8,293	10,117	8,352	25,120	32,870	23,340
FUEL/AIR RATIO	---	0.03900	0.03900	0.03900	0.03900	0.03900	0.03900
AIR/FUEL RATIO	---	25.64	25.64	25.64	25.64	25.64	25.64
AIRFLOW RATE	Lb./Min	1,029	905	1,088	2,972	3,896	2,734
EXHAUST GAS TEMP	Deg. F	1643	2199	2125	1597	2136	1992
BSFC	Lb/BHP/Hr.	0.2903	0.3140	0.2983	0.2769	0.2773	0.2742
IMPROVEMENT BSFC vs. Sea-Level Engi		---	92.47%	97.33%	---	99.84%	100.97%
NUMBER ROTOR UNITS	---	3	3	3	2	2	2
BARE ENGINE WEIGHT	Lb.	1,226	1,226	857	3,757	3,757	2,480
ACCESSORIES WT.	Lb.	86	86	92	301	301	198
TOTAL CORE ENGINE WT.	Lb.	1,312	1,312	949	4,058	4,058	2,678
POWER DENSITY	BHP/Lb.	6.32	7.71	8.80	6.19	8.10	8.72
IMPROVEMENT BHP/Lb vs. Sea-Level		---	122.0%	139.2%	---	130.9%	140.8%

Note 1: DIESEL configuration is computed at Standard Sea Level Pressure (14.7 psia) and Temperature (530 Deg. Rankine).
Note 2: DIESEL BOOSTED is computed at Turbofan exit condition [inlet pressure (23.3/23.05 psig) and temperature (637/634 R)].
Note 3: HYBRID configuration is computed for Axial-Vane Compressor and AV-Expander operating at Turbofan exit condition.
Note 4: Estimated Accessories Weight for DIESEL & DIESEL BOOSTED, [PAX 50 = 0.7x Bare Wt.; PAX 150 = 0.08x Bare Wt.]

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

Minimum Size Fan Boosted 3-Rotor Axial-Vane Diesel Engine for PAX-50 Turbofan Engine

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	3.20	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	6.10	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	17.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	4.80	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	10.90	
INPUT DESIGN VALUES		
SPEED (RPM)	5,900	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	618	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between Vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	23.30	Inlet Pressure behind Turbofan
INITIAL TEMPERATURE (deg R)	637.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	14,484	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	45.98	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	3.36	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.0870	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	1,103	Total Displacement of 24 chambers.
COMPRESSION RATIO	14.39	Ratio of Max. Real Vol. / Min. Real Vol.(Col. Q, Sheet 1)
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	372.31	
FUEL FLOW (lbs/min)	14.52	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	169.35	
INDICATED POWER (hp)	3,356	
FRICTION POWER (hp)	571	
HEAT LOSS POWER (hp)	743	
BRAKE HORSEPOWER OUTPUT (Bhp)	2,784	3 Units = 8,352
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.3129	
EXHAUST TEMPERATURE (deg. F.)	2125	
Performed By:	Mike Smith	Date Performed: May 14, 2006

Figure 20.—Minimum size PAX 50 diesel engine with fan boost to meet specifications.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

Minimum Size Fan Boosted PAX 150 2-Rotor Diesel Core Engine (30.5"@3600)

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	5.50	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	11.00	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	30.50	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	8.50	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	19.50	
INPUT DESIGN VALUES		
SPEED (RPM)	3,600	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	377	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	23.05	Inlet Pressure
INITIAL TEMPERATURE (deg R)	634.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	15,875	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	278.48	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	21.72	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.2808	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	6,684	Total Displacement of 24 chambers.
COMPRESSION RATIO	13.62	Ratio of Max. Real Vol. / Min. Real Vol.(Sht 1, Col. Q)
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	1,367.37	
FUEL FLOW (lbs/min)	53.33	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	192.07	
INDICATED POWER (hp)	13,346	
FRICTION POWER (hp)	1,676	12.6% of Indicated Power
HEAT LOSS POWER (hp)	1,955	14.7% of Indicated Power
BRAKE HORSEPOWER OUTPUT (Bhp)	11,670	23,340 BHP for 2-Rotor Core Engine
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.2742	
EXHAUST TEMPERATURE (deg. F.)	1992	
Performed By:	Mike Smith	Date Performed: Dec. 21, 2005

Figure 21.—Minimum size PAX150 diesel engine with fan boost to meet specifications.

4.0 Design of Axial Vane Hybrid Turbine Engine

The basic axial vane mechanism has been explained in previous sections on the Axial Vane Diesel Engine. This section pertains to the use of the axial vane mechanism as a replacement for axial flow turbomachinery such as the compressor and turbine sections of high bypass ratio turbofan engines. The objective of the study of this application of axial vane technology was to determine size implications and efficiency implications of the compressor and expander for such a hybrid engine. A schematic of the hybrid turbine engine is shown in figure 22. The compressors and expanders of the hybrid engine are mounted on a common shaft that connects the fan, compressors, and expanders. The compressors and expanders would be attached to the shaft so individual units could be removed for servicing or replacement.

The above schematic represents a hybrid engine with the compressor intakes operating at sea level ambient conditions. By mounting the compressor intake behind the fan, the compressor inlet pressure could be increased from 14.7 psia to about 23 psia for the PAX 50 engine. The combustor section could be mounted inline with the compressors and expanders or in parallel outside of the engine nacelle. Similarly, the exhaust from the expanders could be used to drive a turbo charger that could further increase inlet pressure to the compressors and significantly increase the power output and specific power density of the hybrid engine. Figure 23 is a picture of a hybrid axial vane unit consisting of a compressor and an expander section showing the axial direction of airflow into and out of each component. It also shows that the compressor and expander can be different sizes to fit the application. Note there are four inlet ports and four outlet ports on each compressor and each expander.

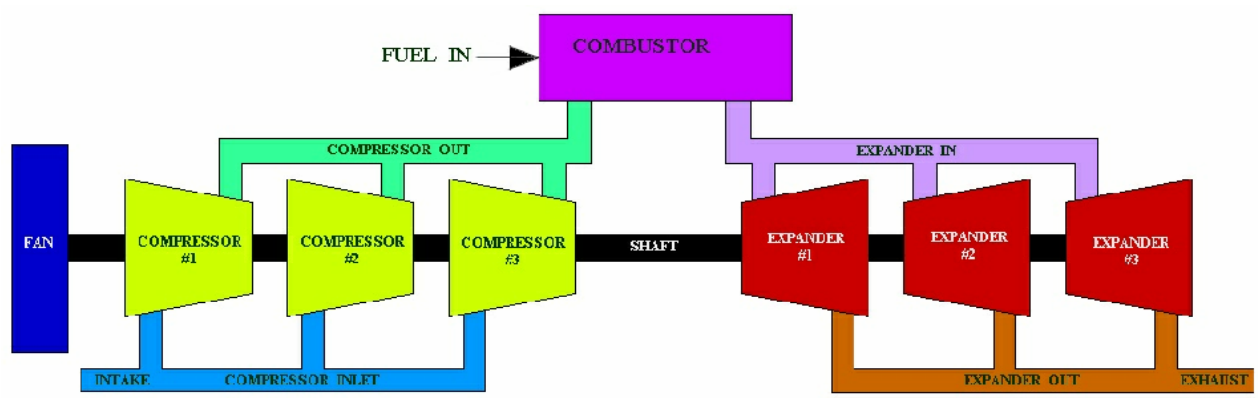


Figure 22.—Schematic of hybrid turbine engine with axial vane compressors and expanders.

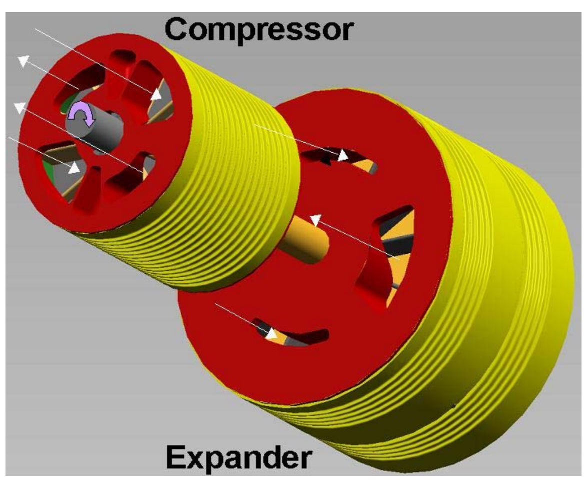


Figure 23.—Model of axial vane compressor and expander with axial intake and exhaust ports.

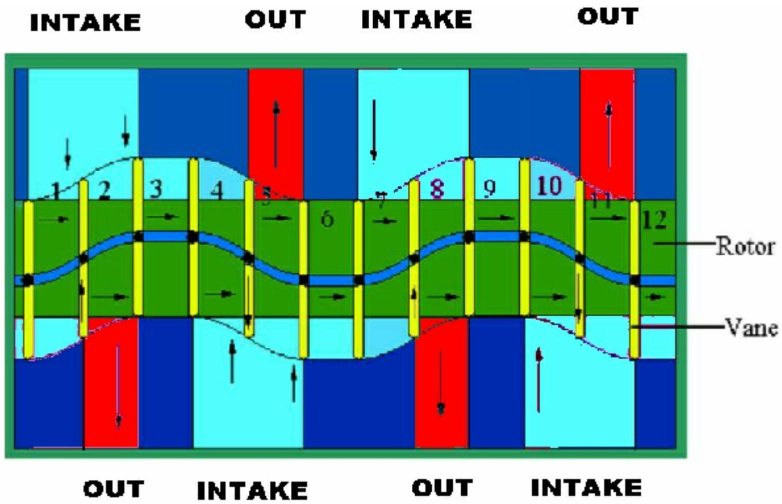


Figure 24.—Schematic of axial vane compressor cycle.

4.1 Compressor for Hybrid Turbine Engine

The axial vane compressor is basically the same as the axial vane Diesel engine except that it does not use fuel injection and has two intake and two outlet ports on each cam end-housing of the device. A schematic of the axial vane compressor operation is shown in figure 24. The compression cycle occurs twice for each chamber on each revolution and the two cycles occur on both sides of the rotor. This means that the displacement of the mechanism is twice as large as a compressor (or expander) as it is for a 4-stroke cycle rotary engine. It should also be noted that the pressure distribution on the two sides of the rotor are exactly 90° out of phase which means that there are no net axial or radial loads on the rotor and; therefore, no loads on the bearings other than the weight of the rotor, shaft, and vanes.

4.1.1 Modeling Hybrid Compressors

To provide a design tool for analysis and design of the hybrid compressor, the Diesel engine cycle simulation spreadsheet was modified to model the axial vane compressor. Because the compressor operates at much lower pressure than a Diesel engine, it was decided to omit the leakage calculations to simplify the model.

4.1.2 Design of Compressors for Hybrid Turbine Engine

The NASA furnished data for the baseline PAX 50 and PAX 150 engines were used to design axial vane compressors to duplicate the burner inlet pressure and airflow rates. The maximum diameter of the compressors was also restricted to the existing nacelle sizes limits of the project specification. To meet the flow requirements, the PAX 50 required three compressor modules whereas two modules could meet the PAX 150 power requirement. The outlet ports were located at the point in the compression cycle where the internal pressure met the burner inlet pressure requirement. The design geometries and predicted performance of the respective PAX 50 and PAX 150 compressors are presented in figures 25 and 26. An exploded view of a typical axial vane compressor is shown in figure 27.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES

Developed by Advanced Technologies Inc.

March 2005

Pax 50 Compressor Design 3 Rotors

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	3.30	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	6.70	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	20.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	6.60	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	13.30	
INPUT DESIGN VALUES		
SPEED (RPM)	5,000	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	524	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	14.70	Inlet Pressure
INITIAL TEMPERATURE (deg R)	545.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
FUEL/AIR RATIO	0.0000	varies with Chamber Pressure & Temperature.
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	65.94	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	4.88	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.1166	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	1,583	Total Displacement of 24 chambers.
COMPRESSION RATIO	14.21	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	667.42	
FUEL FLOW (lbs/min)	0.00	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	-196.46	
INDICATED POWER (hp)	-3,565	
FRICTION POWER (hp)	360	
HEAT LOSS POWER (hp)	66	
BRAKE HORSEPOWER OUTPUT (Bhp)	-3,926	3 Compressors x 3926 = 11,778 Bhp
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.0000	
EXHAUST TEMPERATURE (deg. F.)	0	
Performed By:	Pat Badgley	Date Performed: July 28, 2005

Figure 25.—Summary of compressor design for PAX 50 hybrid turbine engine .

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

PAX 150 Compressor Design 2 Rotors

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Compressor
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	6.00	Estimated from Rotor Diameter / 6
VANE HEIGHT (inches)	12.00	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	36.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	12.00	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	24.00	
INPUT DESIGN VALUES		
SPEED (RPM)	2,778	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	291	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	23.05	Inlet Pressure
INITIAL TEMPERATURE (deg R)	634.55	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	416.09	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	0.02	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.3770	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	19,972	Total Displacement of 48 chambers.
COMPRESSION RATIO	760.22	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	3,150.41	
FUEL FLOW (lbs/min)	0.00	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	-274.73	
INDICATED POWER (hp)	-17,945	
FRICTION POWER (hp)	1,302	
HEAT LOSS POWER (hp)	360	
BRAKE HORSEPOWER OUTPUT (Bhp)	-19,246	2 Compressors = -38,492 Bhp
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.0000	
EXHAUST TEMPERATURE (deg. F.)	721	

Performed By: Pat Badgley Date Performed: July 31, 2005

Figure 26.—Summary of compressor design for PAX 150 hybrid turbine engine.

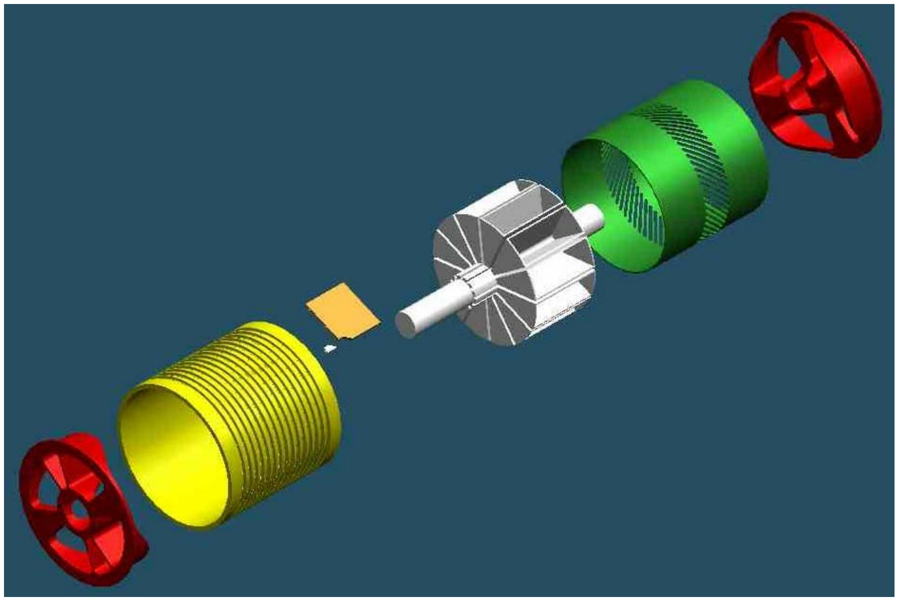


Figure 27.—Exploded view of typical axial vane compressor.

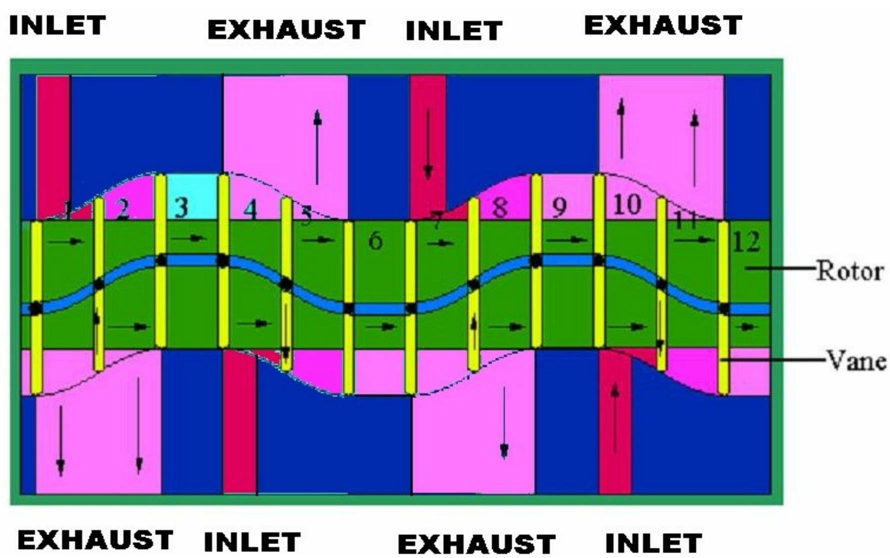


Figure 28.—Schematic of axial vane expander.

An exploded view of a typical compressor design (fig. 27) shows a two-piece vane design that allows the vanes to ride on the cam surface to reduce leakage. This design will be used if a material pair can be found for the vane and cam to allow long life using only self-lubrication. An alternative design using the pin and pin track or rail and slot can also be used. With the alternative designs, a thick thermal barrier coating can be applied to both the cam face and the face of the rotor to reduce heat losses and improve volumetric efficiency by allowing the surface temperatures to follow the gas temperature history continuously.

4.2 Expanders for Hybrid Turbine Engines

The axial vane expander is almost identical to the compressor with the exception that the inlet and exhaust ports are repositioned as shown in the expander schematic (fig. 28). Note that the expansion cycle occurs twice each revolution for each chamber and that there are no pressure induced bearing loads. Like the compressor, the expander with 12 vanes has 48 expansion cycles each revolution of the rotor and produces intake and exhaust pressure pulses at very high frequency.

4.2.1 Modeling the Hybrid Expanders for Hybrid Turbine Engines

Similar to the compressor, an expander model was developed from the basic Axial Vane Diesel Engine Design Spreadsheet. A very sensitive part of the modeling turned out to be the location of the inlet ports. The ports had to close at a position that would provide a volume large enough to hold the entire mass flow (per chamber) at the burner outlet density. The port closure position also directly controls the expansion ratio. This means that moving the port to an earlier position would reduce the mass flow and moving it to a later position would reduce the expander power output. An exploded view of a typical expander unit is shown in figure 29 and a cutaway view is shown in figure 30 illustrating the split-vanes and ports.

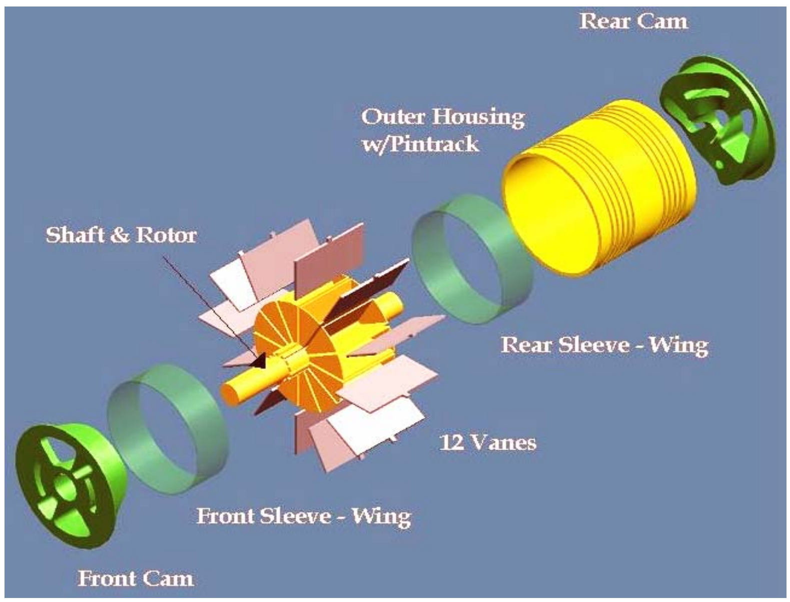


Figure 29.—Exploded view of axial vane expander.

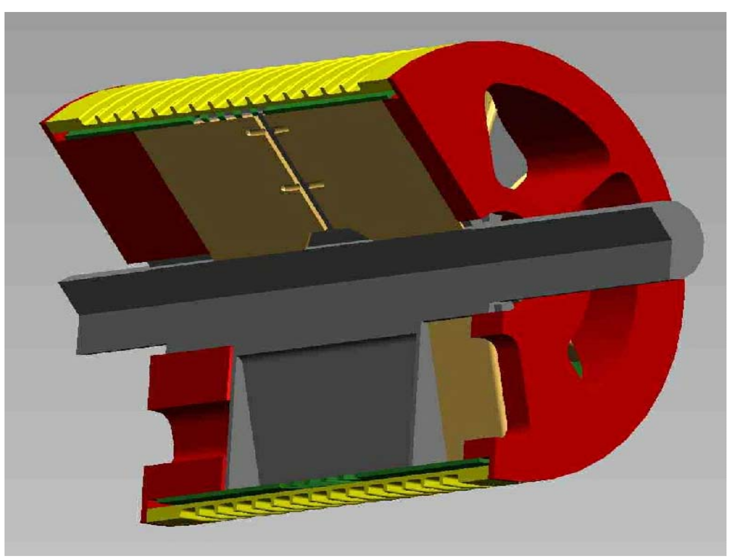


Figure 30.—Sectioned compressor or expander showing split-vanes and ports.

4.2.2 Design of Hybrid Expanders for Hybrid Turbine Engines

The rotor diameter of the hybrid expander was assumed to be 20 in., the same as the sea-level Diesel engine. The performance analysis for the axial vane expanders on the PAX 50 and PAX 150 Hybrid Turbine Engines are shown in figures 32 and 33. The weight schedule for the hybrid compressors, expanders, and the complete hybrid engine are presented in table 5.

Figure 31 is a solid model illustrating the compressor and expander construction by making the external components transparent. In this picture the compressor and expander are shown as being the same diameter.

TABLE 5.—WEIGHT SCHEDULE FOR PAX 50 AND PAX 150 COMPRESSORS AND EXPANDERS

WEIGHT SCHEDULE FOR HYBRID TURBINE ENGINES

HYBRID COMPRESSOR AND EXPANDER FOR PAX 50 & PAX 150 ENGINES

COMPRESSOR		PAX 50 Axial Vane Hybrid Engine		PAX 150 Axial Vane Hybrid Engine	
Part	Number/ Module	Weight/ Part	Weight/ Module	Weight/ Part	Weight/ Module
Cam	2	33.39	66.78	153.41	306.82
Rotor	1	59.19	59.19	271.99	271.99
Vane	12	3.65	43.74	16.75	200.98
Wing	2	11.66	23.33	53.59	107.19
Rail	1	14.58	14.58	66.99	66.99
Rail Support	1	15.31	15.31	70.34	70.34
Outer	1	75.09	75.09	345.01	345.01
		Total	298.02	Total	1,369.32
Total Weight Compressors		**3 Rotors**	**894**	**2 Rotors**	**2,739**
EXPANDER					
Part	Number/ Module	Weight/ Part	Weight/ Module	Weight/ Part	Weight/ Module
Cam	2	33.39	66.78	153.41	306.82
Rotor	1	59.19	59.19	271.99	271.99
Vane	12	3.65	43.74	16.75	200.98
Wing	2	11.66	23.33	53.59	107.19
Rail	1	14.58	14.58	66.99	66.99
Rail Support	1	15.31	15.31	70.34	70.34
Outer	1	75.09	75.09	345.01	345.01
		Total	298.02	Total	1369.32
Total Weight Expanders		**3 Rotors**	**894**	**2 Rotors**	**2,739**
TOTAL WEIGHT OF HYBRID CORE			**1,788**		**5,477**
ACESSORIES			**125**		**438**
TOTAL HYBRID TURBINE WEIGHT,lb.			**1,913**		**5,915**

Assuming all parts are made of titanium

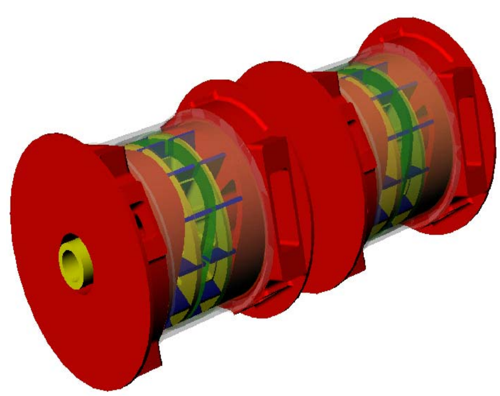


Figure 31.—Paired compressor and expander with space for burner between units.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

PAX 50 Expander Design 3-Rotors

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Expander
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	3.50	Estimated as Rotor Diameter / 6
VANE HEIGHT (inches)	7.00	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	20.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	6.00	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	13.00	
INPUT DESIGN VALUES		
SPEED (RPM)	5,000	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	524	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	345.00	Inlet Pressure
INITIAL TEMPERATURE (deg R)	2,500.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	71.15	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	0.00	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.1191	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	3,415	Total Displacement of 24 chambers.
COMPRESSION RATIO	644.28	Ratio of Max. Real Vol. / Min. Real Vol.(Col. Q, Sheet 1)
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	742.83	
FUEL FLOW (lbs/min)		
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	377.91	
INDICATED POWER (hp)	8,550	
FRICTION POWER (hp)	402	
HEAT LOSS POWER (hp)	452	
BRAKE HORSEPOWER OUTPUT (Bhp)	7,696	X 3 = 3 EXPANDERS = 23,087
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.0000	
EXHAUST TEMPERATURE (deg. F.)	1270	
Performed By:	Pat Badgley	Date Performed: 8/8/2005

Figure 32.—Performance analysis of PAX 50 expander for hybrid turbine engine.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

Pax 150 Expander Design 2 Rotors (36-in OD@2778)

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES	
PHYSICAL DIMENSIONS & DESIGN VARIABLES			
NUMBER OF VANES	12	12-Vane Expander	
NUMBER OF CAMS	2		
ANGLE BETWEEN VANES	30		
VANE THICKNESS (inches)	0.50	Estimated = 0.10 x Vane Stroke	
VANE STROKE (inches)	6.00	Estimated from Rotor Diameter / 6	
VANE HEIGHT (inches)	12.00	Estimated as 2 x Vane Stroke	
LIFT ANGLE (degrees)	75		
DWELL ANGLE (degrees)	15		
EVENT ANGLE (degrees)	90		
ROTOR OUTER DIAMETER (inches)	36.00	Estimated based on Chart: OD vs. BHP	
ROTOR INNER DIAMETER (inches)	12.00	Estimated = OD -2 Vane Height	
ROTOR MEAN DIAMETER (inches)	24.00		
INPUT DESIGN VALUES			
SPEED (RPM)	2,778	Limited by Vane Rubbing Velocity	
OMEGA (radians/sec)	291		
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam	
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam	
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings	
INITIAL PRESSURE (psia)	399.00	Inlet Pressure	
INITIAL TEMPERATURE (deg R)	2,960.00	Inlet Temperature	
Gas Constant, R =	53.30		
Ratio of Specific Heat, k =	1.3937		
COMPUTED VALUES			
DISLACEMENT/CHAMBER (cu. in.)	416.09	Adjust Displacement/Chamber to change Comp. Ratio	
POCKET VOLUME (cu.in.)	0.12	Pocket Volume is automatically adjusted when Disp. Per	
ROTOR CLEARANCE VOLUME (cu.in.)	0.3770	Chamber is changed.	
TOTAL DISPLACEMENT (cu. in.)	19,972	Total Displacement of 48 chambers.	
COMPRESSION RATIO	643.07	Ratio of Max. Real Vol. / Min. Real Vol.(Col.	
COMPUTED PERFORMANCE			
AIR FLOW (lbs/min	3,324.73	X 2 =	6649
AIR FLOW (lbs/min) X 2 =	6,649.46		
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	508.53		
INDICATED POWER (hp)	36,927		
FRICTION POWER (hp)	1,302		
HEAT LOSS POWER (hp)	2,156		
BRAKE HORSEPOWER OUTPUT (Bhp)	33,470	X 2 =	66939
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.0000		
EXHAUST TEMPERATURE (deg. F.)	1685		
Performed By:	Pat Badgley	Date Performed:	July 31, 2005

Figure 33.—Performance analysis of PAX 150 expander for hybrid turbine engine.

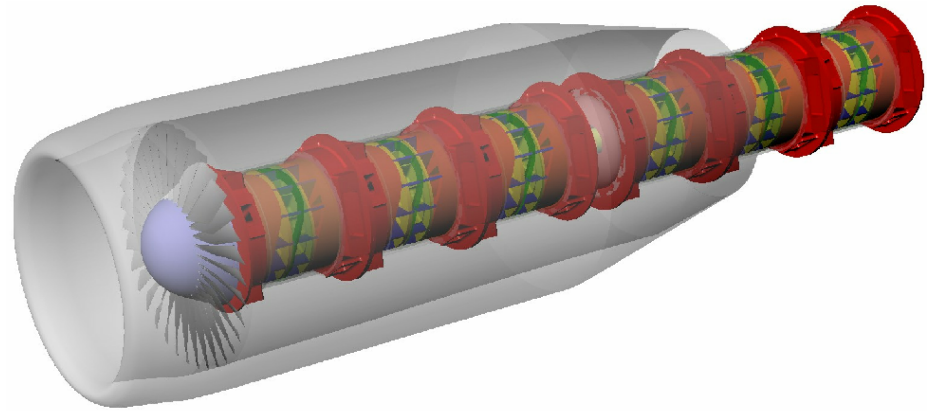


Figure 34.—Solid model of PAX 50 axial vane hybrid engine assembly.

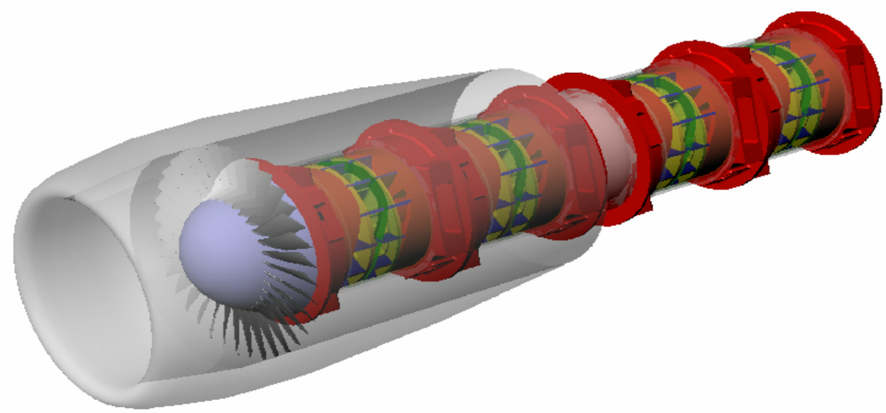


Figure 35.—Solid model of PAX 50 axial vane hybrid engine assembly.

Figures 34 and 35, respectively, are solid models of the assembled PAX 50 and PAX 150 axial vane hybrid engines including the combustors (burners), fans and nacelles. These axial vane components have radial inlet and outlet ports and mounting flanges to couple the modules together to make a rigid assembly.

4.2.3 Discussion of PAX 50 Expander and PAX 150 Expander

The solid models of the Axial Vane PAX 50 and PAX 150 hybrid engines are shown in figures 34 and 35, respectively. The PAX 50 hybrid engine has three compressors operating in parallel and three expanders operating in parallel. Located between the compressor outlets and the expander inlets will be a combustor, which can be an identical design and construction as the combustor (burner) on the current PAX 50 and PAX 150 engines. All of the rotating components share a common shaft, which runs the length of the engine and also drives the fan (with or without a gearbox). The shaft will have an external spline that fits a mating internal spline in each rotating component allowing the components and shaft to move axially relative to each other due to thermal expansion. The PAX 150 hybrid engine has two compressors operating in parallel and two expanders operating in parallel.

4.3 Thermodynamic Model of Hybrid Turbine Engine

As a check on the spreadsheet design programs, a thermodynamic analysis of the complete cycle for both the hybrid engines and also for the baseline turbomachinery type PAX 50 and PAX 150 engines was performed. This analysis is shown in table 6. This analysis relies on the

TABLE 6.—ENGINE CYCLE ANALYSIS FOR HYBRID AND TURBOMACHINE ENGINES

ENGINE CYCLE ANALYSIS with NASA INLET CONDITIONS

(Inlet Conditions behind Fan)

Pat Badgley August 28, 2005

Thermodynamic Analysis Program		HYBRID AXIAL VANE		TURBOMACHINE		NASA DATA	
Inlet Conditions		PAX 50	PAX 150	PAX 50	PAX 150	PAX 50	PAX 150
Pressure =	psia	23.3	23.045	23.3	23.045	23.3	23.045
Temperature =	Deg R.	638	634	638	634	638	634
Density =	lbs/ cu in	5.70504E-05	5.6782E-05	5.70504E-05	5.6782E-05		
Cp =		0.24	0.24	0.24	0.24		
K=		1.4	1.4	1.4	1.4		
R=	ft lb/Deg R	53.345	53.345	53.345	53.345		
Enthalphy =	Btu/lb	57.16	56.2	57.16	56.2		
Pr1 =		5.13	5.04	5.13	5.04		
Volumetric Efficiency =	percent	100	100	100	100		
Displacement =	cu in	6850	39945	7740	44266.64		
Fuel/Air Ratio =		0.02258	0.0249	0.02258	0.0249	0.02258	0.0249
Fuel Heating Value =	Btu/lb	18600	18600	18600	18600		
Speed =	rpm	5555	2778	5000	2500	18000	5380
First Estimate of Pcout	psia	345	399.4	345	399.4	345	399.4
Compressor Efficiency =	percent	98	98	86.5	86	86.5	
Combustor efficiency =	percent	100	100	100	100	100	
Turbine Efficiency =	percent	98	98	91.5	91.5		
Pr2 =		75.96	87.35	75.96	87.35		
Enthalphy 2 =	Btu/lb	232.9	246.1	232.9	246.1		
Enthalphy Rise =	Btu/lb	175.74	189.9	175.74	189.9		
Actual Rise =	Btu/lb	179.33	193.78	203.17	220.81		
Actual Enthalphy 2 =	Btu/lb	236.49	249.98	260.33	277.01		
Tcout =	Deg R.	1,357	1,409	1,448	1,512	1,448	1,518
Enthalphy Comp =	Btu/lb	172.66	186.00	194.40	210.72		
Work of Compression =	Btu/lb	123.28	132.80	138.80	150.45		
	Btu/min	267,616	836,790	306,453	945,434		
	hp	6,305	19,715	7,220	22,274	12,000	39,173
Enthalphy Cout =	Btu/lb	229.82	242.20	251.56	266.92		
Airflow =	cu in/min	38,051,750	110,967,210	38,700,000	110,666,600		
	lbs/min	2,170.87	6,300.94	2,207.85	6,283.87	2,200.00	6,284.00
Fuel Flow =	lbs/min	49.02	156.89	49.85	156.47		
	lbs/hr	2,941	9,414	2,991	9,388		
Heat Input =	Btu/min	911,738	2,918,219	927,271	2,910,314		
Percent Recuperation	percent	0.00	0.00	0.00	0.00		
Comb Enthalphy Rise =	Btu/lb	410.71	451.89	410.71	451.89		
Comb Out Enthalphy =	Btu/lb	640.53	694.09	662.27	718.81		
Comb Tout =	Deg R.	3,068.71	3,291.87	3,159.31	3,394.87	2,792.00	2,960.00
Pr3 =		2,130	2,900	2,440	3,330		
Pr4 =		143.85	167.33	164.79	192.14		
Enthalphy 4 =	Btu/lb	296.76	313.70	312.10	329.70		
Isentropic Enthalphy Ch	Btu/lb	343.77	380.39	350.17	389.11		
Actual Enthalphy Change	Btu/lb	336.89	372.78	320.41	356.03		
Enthalphy Expand Out =	Btu/lb	303.64	321.31	341.86	362.77		
Exp Out Temperature =	Deg R	1,611	1,677	1,753	1,737	1,446	1,528
Work of Expansion =	Btu/lb	249.79	276.72	240.98	284.09		
	Btu/min	554,511	1,787,034	544,072	1,829,648		
	hp	13,064	42,103	12,818	43,107	20,229	63,039
Mechicanical Efficiency =	percent	90.00	90.00	99.00	99.00		
Work Out =	hp	6,083	20,149	5,542	20,624	8,132	23,664
BSFC=	lbs/hp-hr	0.4835	0.4672	0.5397	0.4552	0.3580	0.3950
Total Available Enthalphy	Btu/lb	60.86	64.32	73.20	54.00		
	Btu/min	132,127.70	405,276.72	161,614.68	339,329.24		
	hp	3,113	9,548	3,808	7,995		

thermodynamic properties of air as tabulated in reference 1. The inlet pressure for the hybrid and turbomachine engines was assumed to be 23.3 psia that was the same as the computed pressure for the NASA analysis of the baseline PAX 50 and PAX 150 engines. The fuel/air ratios for the hybrid and turbomachine engines were assumed to be the same as used in the NASA analysis of the baseline engines. Table 6 shows that the predicted performance of the hybrid and turbomachinery engines are very close, but that the computed performance data supplied by NASA shows much better performance for the turbofan engine. The major difference between the model and the real turbomachinery type engines is that the current model does not account for bleeds and reintroduction of cooling air. Also this model does not account for the presence of combustion products and the additional mass flow that they introduce, nor does it account for the changed gas properties in the calculations.

5.0 Summary and Conclusion

This project generated several analysis and design tools for axial vane rotary compressors, expanders, and Diesel engines. The project also generated specific technical data on the design of axial vane machinery for high specific output Diesel and positive displacement Brayton cycle engines for aircraft applications. Table 7 is a compilation of the Diesel, Hybrid and turbomachinery engines for the PAX 50 and PAX 150 applications. As the table clearly shows, the Axial Vane Diesel Engine shows the most promise for application in turbofan engines because the specific weight and specific volume match that of turbomachines while the specific fuel consumption is much lower. Another significant advantage for the Diesel engines is the fact that the specific fuel consumption of Diesel engines at reduced load and reduced speed changes very little as compared to turbomachinery engines. The engine weight reduction obtainable with the axial vane components is highly significant.

5.1 Comparison of PAX 50 Engines

Using the data shown in table 7, the Axial Vane Diesel Engine, the Axial Vane Hybrid Engine, and the Turbomachinery Engine have specific power density of 8.8, 4.29, and 4.98 hp/lb, respectively, when all are operating at the turbine inlet pressure behind the fan. The brake specific fuel consumption (lb/hp-hr) of the Diesel is 0.2983, the Hybrid is 0.3010 and the Turbomachine is 0.3665. It is also significant that the airflow in the Diesel engine is less than one half that of the hybrid and turbomachinery engines. This means that the net efficiency of the fan will increase because less air is bled off for the engine.

5.2 Comparison of PAX 150 Engines

Table 7 also shows the performance and weight estimates for the PAX 150 engines. The minimum size Axial Vane Diesel Engine has a specific power density of 8.72, the Axial Vane Hybrid Engine 4.69, and the Turbofan Engine 3.38 hp/lb, respectively. The brake specific fuel consumption (lb/hp-hr) of the Diesel is 0.2742, the Hybrid 0.3394 and the Turbofan 0.3945. Like the PAX 50, the airflow through the Diesel engine is roughly one-half that of the Turbofan engine which means that the actual thrust output of the engine’s fan will be greater and the overall efficiency further improved.

5.3 General Comments

It should be noted that these analyses of axial vane components all are based on the assumption the wall temperature is equal to ambient temperature which is highly conservative. Optimization of specific component designs should be based on thermal modeling and analysis.

TABLE 7.—COMPARISON OF PERFORMANCE AND WEIGHT OF ANALYZED ENGINE CONFIGURATIONS

COMPARISON OF CORE ENGINE TYPES FOR PAX 50 AND PAX 150 TURBOFAN ENGINES

PARAMETER	UNITS	PAX 50 ENGINE					PAX 150 ENGINE				
		DIESEL CORE	DIESEL BOOSTED	MIN.SIZE BOOSTED	HYBRID ROTARY	TURBO FAN	DIESEL CORE	DIESEL BOOSTED	MIN.SIZE BOOSTED	HYBRID ROTARY	TURBO FAN
ROTOR DIAMETER	Inches	20.0	20.0	17.0	20.0	24.4	40.0	40.0	30.5	36.0	44.5
NUMBER ROTOR UNITS	---	3	3	3	6	---	2	2	2	4	---
ENGINE SPEED	RPM	5000	5000	5900	5000	18000	2500	2500	3600	2778	14500
INLET PRESSURE	Psia	14.70	23.30	23.30	23.30	23.30	14.70	23.05	23.05	23.05	23.05
POWER	BHP	8,293	10,117	8,352	9,902	8,132	25,120	32,870	23,340	27,729	23,664
FUEL/AIR RATIO	---	0.03900	0.03900	0.03900	0.02258	0.02258	0.03900	0.03900	0.03900	0.02490	0.02490
AIR/FUEL RATIO	---	25.64	25.64	25.64	44.29	44.29	25.64	25.64	25.64	40.16	40.16
AIRFLOW RATE	Lb./Min	1,029	905	1,088	2,200	2,200	2,972	3,896	2,734	6,300	6,248
EXHAUST GAS TEMP	Deg. F	1643	2199	2125	2032	1103	1597	2136	1992	1685	1068
BSFC	Lb/BHP/Hr.	0.2903	0.3140	0.2983	0.3010	0.3665	0.2769	0.2773	0.2742	0.3394	0.3945
IMPROVEMENT BSFC vs. TURBO		126.24%	116.73%	122.87%	121.77%	---	142.48%	142.25%	143.86%	122.61%	---
NUMBER ROTOR UNITS	---	3	3	3	6	---	2	2	2	4	---
BARE ENGINE WEIGHT	Lb.	1,226	1,226	857	2,142	1,491	3,757	3,757	2,480	5,477	6,135
ACCESSORIES WT.	Lb.	86	86	92	168	141	301	301	198	438	864
TOTAL CORE ENGINE WT.	Lb.	1,312	1,312	949	2,310	1,632	4,058	4,058	2,678	5,915	6,999
POWER DENSITY	BHP/Lb.	6.32	7.71	8.80	4.29	4.98	6.19	8.10	8.72	4.69	3.38
IMPROVEMENT BHP/Lb vs. TURBO		126.9%	124.6%	176.6%	86.0%	---	183.1%	239.6%	257.8%	138.6%	---

Note 1: DIESEL configuration is computed at Standard Sea Level Pressure (14.7 psia) and Temperature (530 Deg. Rankine).

Note 2: DIESEL BOOSTED is computed at Turbofan exit condition [inlet pressure (23.3/23.05 psig) and temperature (637/634 R)].

Note 3: HYBRID configuration is computed for Axial-Vane Compressor and AV-Expander operating at Turbofan exit condition.

Note 4: Estimated Accessories Weight for DIESEL, DIESEL BOOSTED, & HYBRID Configuration [PAX 50 = 0.7x Bare Wt.; PAX 150 = 0.08x Bare Wt.]

Note 5: TURBOFAN is Baseline Turbofan Engine, Data by NASA GRC, Propulsion Branch.

5.4 Recommendations for Future Work

While this project has made significant progress in understanding the Axial Vane Machine for this application, and has shown that the axial vane engine has genuine promise, unanswered questions remain concerning the potential for axial vane components in a turbofan engine. Several of these issues are discussed in the following paragraphs:

5.5 Materials

Perhaps the biggest challenge to successful development of an axial vane Diesel engine is researching and optimizing materials for each of the axial vane parts. In no particular order, the key parameters that must be analyzed are strength, density, stiffness, thermal conductivity, thermal expansion, castability, machinability, and cost. Another family of properties will include the compatibility with thermal barrier coatings and/or tribological coatings. This effort could rely heavily on NASA's past and present work in these two areas. For the initial design of the Diesel and Hybrid engines, it was assumed for weight estimation purposes that titanium would be used exclusively. It is realized that it is likely that no titanium would be used in the final engine as other materials such as ceramic composites and some of the newer materials such as aluminides may prove better. Perhaps the largest improvement that will result from advanced materials will be a reduction in heat losses which will result in increased power output and reduced fuel consumption.

5.6 Tribology

The second key question regards friction and wear. If the engines were to be designed today, with minimal risks taken, they would use conventional liquid lubrication of the vane guide mechanism with the associated sealing and leakage problems. However, if the latest solid lubrication and air bearing technologies can be incorporated, the engines could be made oil-less. This would dramatically simplify the design, lower production costs, and reduce maintenance costs while improving life and reliability. The reduction in friction will also improve the power output and reduce fuel consumption. The previous and current work on tribology at NASA will be invaluable in guiding further development of the axial vane engine.

Reference

1. "Thermodynamic Properties of Air," by Joseph H. Keenan and Joseph Kaye, 1945

Appendix A
Baseline PAX 50 and PAX 150 Turbofan Specifications

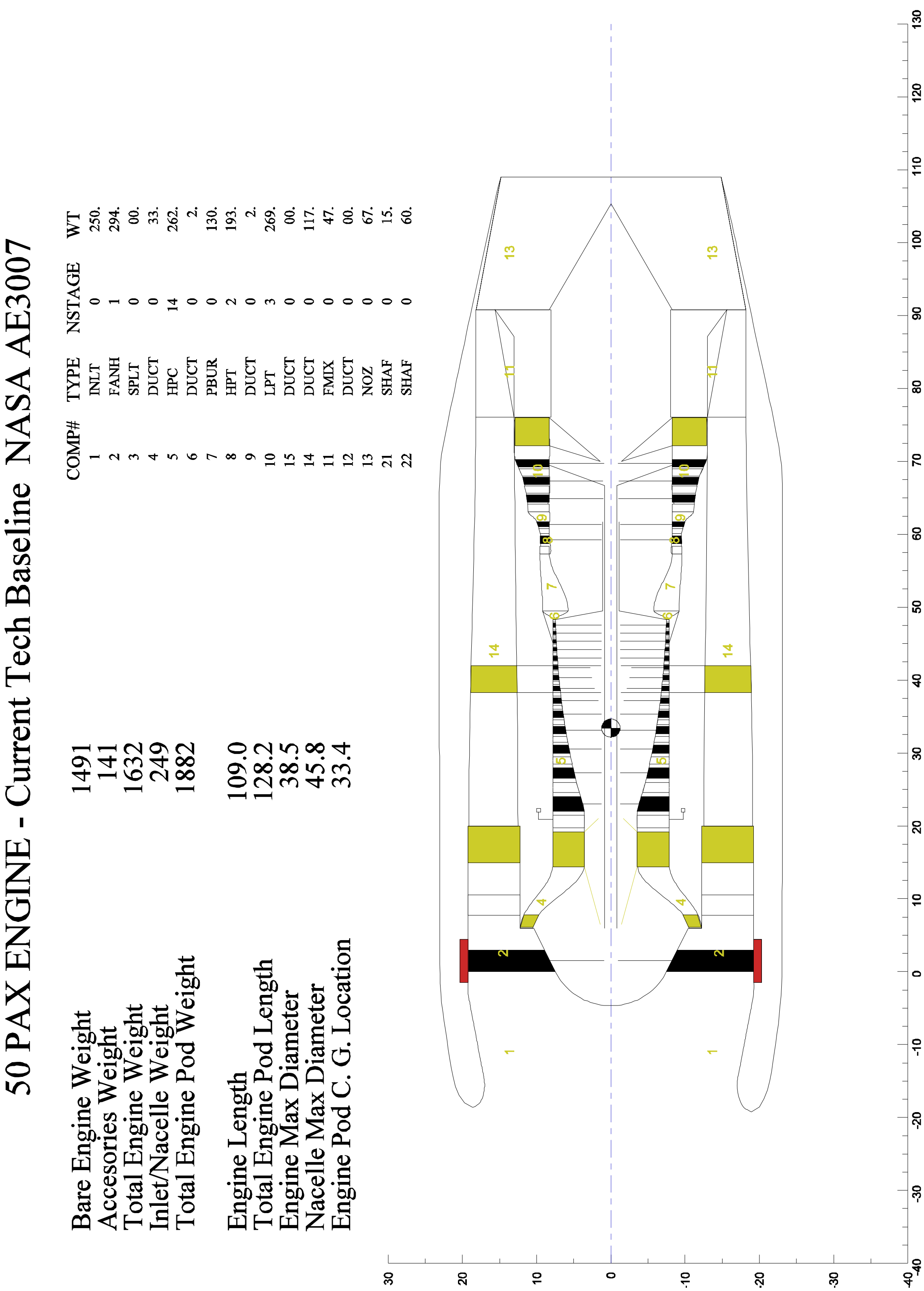


Figure 1–1.—Specifications for PAX 50 baseline 8,132 BHP turbofan engine.

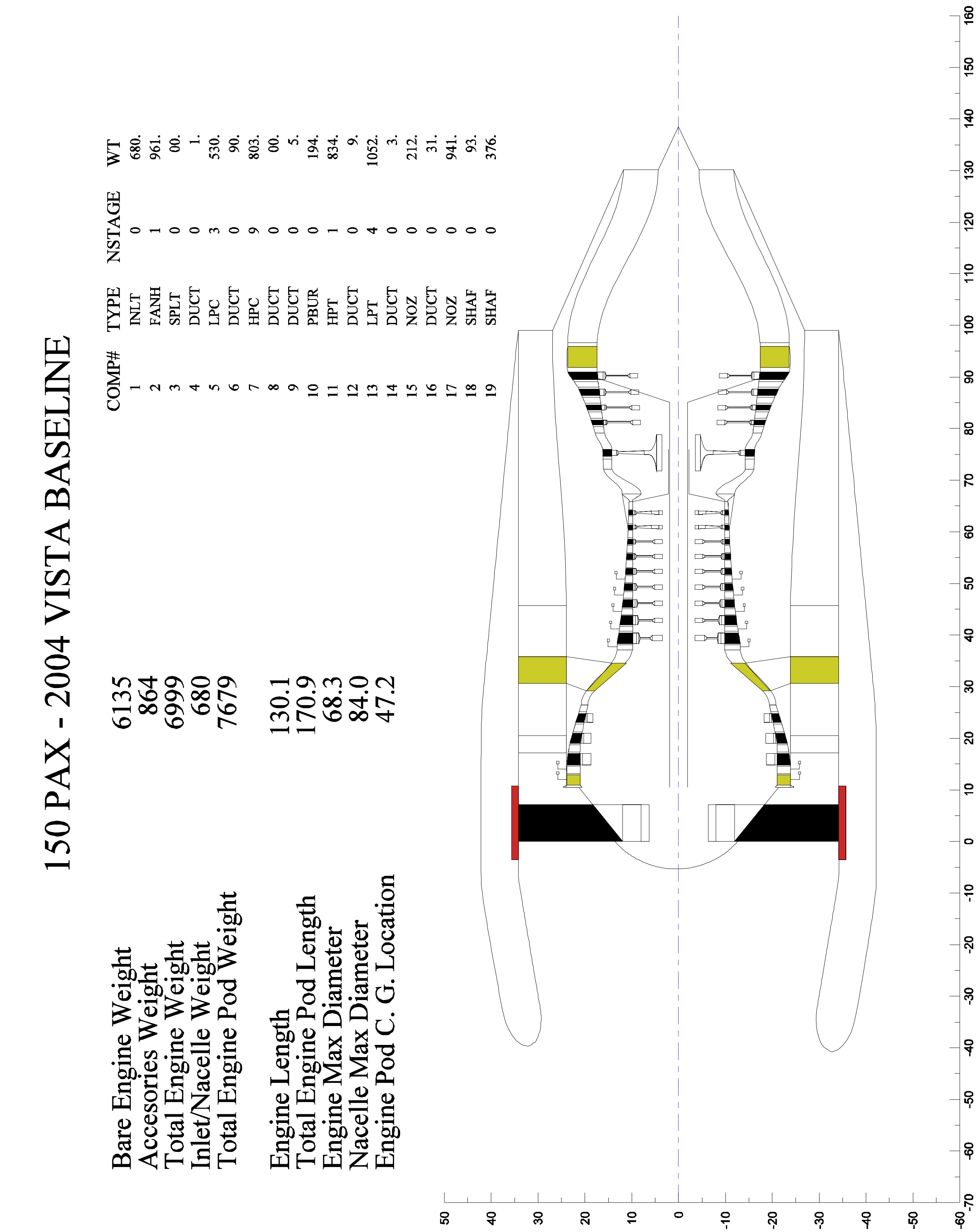


Figure 2–1.—Specifications for PAX 150 baseline 23,664 BHP turbofan engine.

Appendix B
Instruction Manual for Axial Vane Mechanism Design Program
Version 2.0

TABLE OF CONTENTS

DESIGN PROGRAM FOR AXIAL VANE MACHINES

Developed by
Advanced Technologies Inc.

INSTRUCTION MANUAL

SECTION 1: OVERVIEW

This manual is a guide to effectively utilizing the “Design Program for Axial Vane Machines” to perform design analyses on such machines including parametric optimizations of geometry and performance. The spreadsheet program can be used to evaluate and design compressors, expanders, and Diesel engines based on the ATI Winged Rotor Axial-Vane Mechanism. The Microsoft Excel (Microsoft Corporation) spreadsheet program provides (within the limitations of a spreadsheet type program) both numerical and graphical outputs of the following design and performance parameters:

- Cam Design including lift and velocity and comparisons of performance to Piston and Wankel type engines.
- Chamber Volume as a function of shaft angle for each chamber.
- Leakage Analysis for the flow over and under the sliding vanes and between the vanes and the cam surface.
- Temperature, Pressure, and Gas Weight histories in each chamber of the device through a complete 360° rotation.
- Heat Transfer through the Rotor, Outer Wing, Inner Wing and Cam along with overall heat transfer per degree of rotation of the rotor.
- Heat Release from the injection and burning of fuel.
- Friction Estimate and Power Loss due to vane sliding friction.
- Indicated Horsepower and Brake Horsepower Output.
- Exhaust Gas Temperature
- Fuel Consumption including Brake Specific Fuel Consumption

This manual includes a description of each working cell in the spreadsheet including the formula used in computing each cell value and the rationale behind each formula. In addition, the manual provides guidance in determining estimates of each input value and finally a step-by-step sequence to operating the program to obtain correct results.

SECTION 2: INTRODUCTION TO AXIAL VANE MECHANISMS

A simplified cutaway model of an axial vane machine is shown below. The Axial Vane Mechanism is unique in that a multiplicity of straight vanes, contained in a cylindrical rotor, travel parallel to the axis of rotation and compress or expand fluids in the same manner as reciprocating piston mechanisms. Since the vanes move parallel to the axis of rotation, the mechanism is always dynamically balanced and there are no large energy losses due to reciprocating motion and accelerations typical of piston mechanisms.

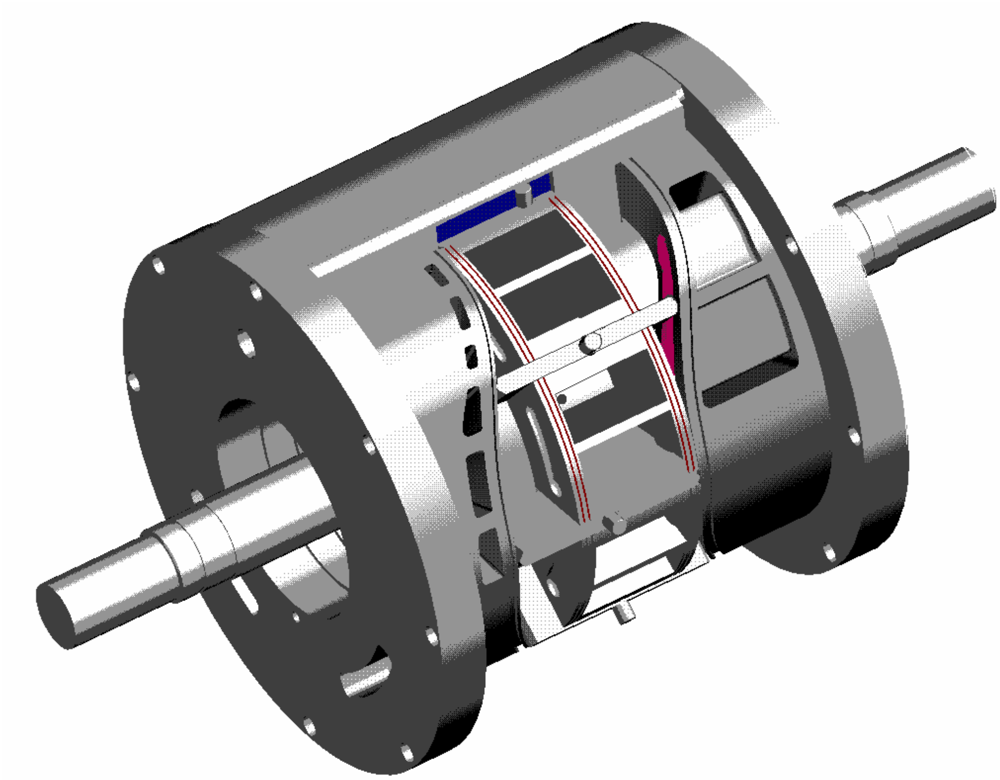


Figure 1–1.—Illustration of typical axial vane mechanism.

The vanes in the axial vane mechanism, shown in figure 1, follow the cam surfaces in the end housings, alternately compressing and/or expanding fluids depending upon the application of the mechanism. In this particular mechanism, the top edge of the vane at the outer edge of the rotor travels on the exterior housing such that both the axial motion of the vane and the rotational speed of the rotor create friction between the outer vane surface and the housing. The resultant rubbing velocity limits the maximum rotational speed of the mechanism due to limitations on the rubbing velocity of the vane/housing and/or the vane/rotor materials. In addition, leakage of compressed fluids can occur at the tip of the vanes and along both the top (exterior) and bottom (interior) edges of the vanes.

The vane shown in figure 1 illustrates a pin-and-track mechanism that drives the vane and absorbs dynamic vane loads such that the tip of the vane does not actually rub on the cam surface. Combustion pockets are shown in the face of the rotor in figure 1. The volume of the pockets is adjusted in the design process to obtain the desired compression ratio in diesel engines.

The “Winged Rotor” concept developed by ATI eliminates or greatly minimizes leakage along the top and bottom of the vanes and de-couples the vane from the frictional effects of rotational speed. An illustration of the ATI Winged Rotor Concept (Patent Pending) is shown in figures 2 and 3. In the Winged Rotor device, a cylindrical sleeve is attached to the outer periphery of the rotor and grooves in the sleeve ("Outer Wing") and a similar “Inner Wing” on the main shaft minimize leakage over the top and bottom of the vane. In some designs the groove in the outer wing may not be necessary due to centrifugal force of the vane being adequate to prevent major leakage over the top of the vane.

A major advantage of the Winged Rotor Concept is that it decouples the vanes from the rubbing velocity caused by rotation of the rotor and permits much higher rotational speeds (i.e., higher Specific Power Density). The maximum allowable rubbing velocity between the vanes and rotor or the vanes and outer housing of the original axial vane mechanism determined the limiting rotational speed. Decoupling of the vanes from the rotational rubbing velocity in the ATI Winged Rotor Mechanism permits the maximum rotational speed to be increased by about five-fold in engine applications.

The higher rotational speed made possible with the ATI Winged Rotor Mechanism permits the design of weight sensitive devices with significantly increased specific power density. This feature is especially important in aviation applications such as compressors, expanders, or diesel engines. The design programs discussed in this instruction manual and in the design programs are based on utilization of the ATI Winged Rotor Concept to minimize friction losses, leakage, heat losses, and installed weight; and to maximize specific power density, reliability, and life of the engine/device.

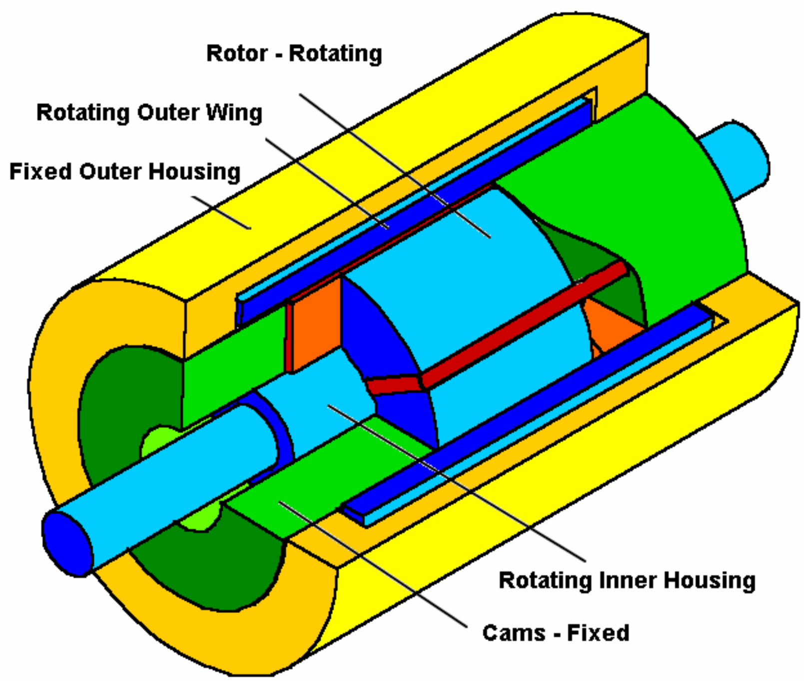


Figure 2–1.—Cutaway of ATI winged rotor axial vane mechanism.

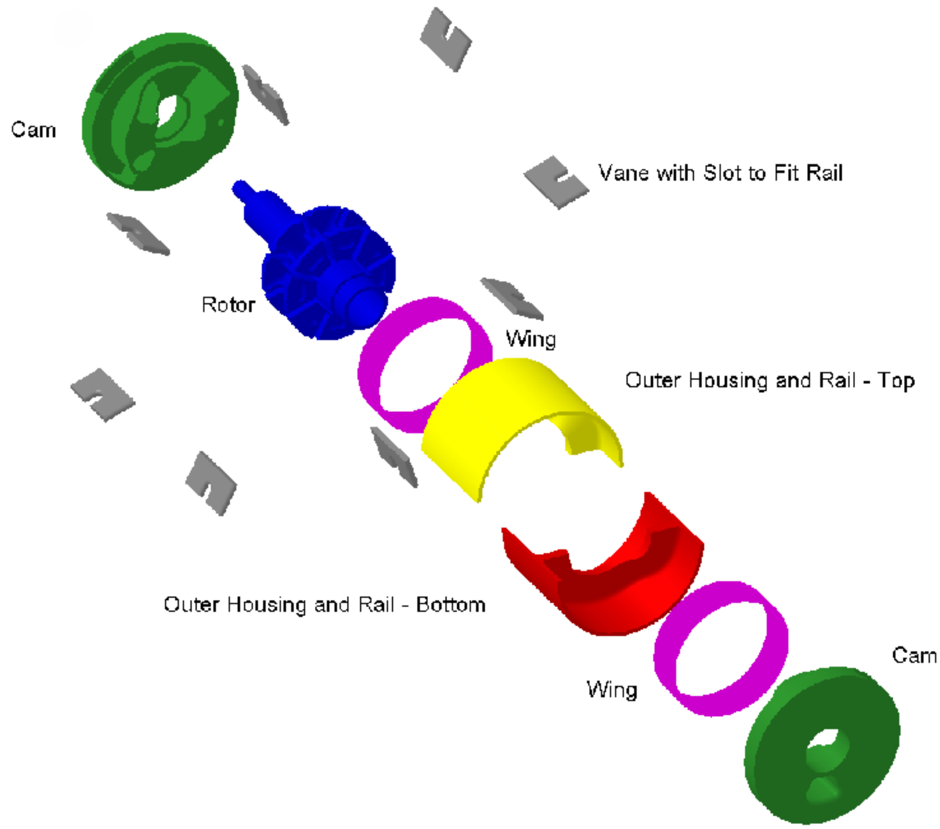


Figure 3–1.—Components of ATI “Winged Rotor” axial-vane mechanism.

The vanes and housings illustrated in figure 3, demonstrate a slot-and-rail system for driving the vane. This slot-and-rail system is designed to prevent the tip of the vane from rubbing on the cam. The vane clearance should be as small as possible to prevent rubbing on the cam and to minimize leakage across the vane tip. It should be noted that the effect of leakage in a chamber is minimized by the fact that any mass leakage moves into a trailing chamber and is swept forward, thereby minimizing mass loss as compared to a piston-type device.

SECTION 3: EXPLANATION OF AXIAL VANE DESIGN PROGRAMS

Two programs have been developed by Advanced Technologies Inc. to aid in the analysis and design of axial vane devices based on the “Winged Rotor” concept developed by ATI. The first program is a preliminary analysis and design tool, Axial-Vane PDQ, (AV-PDQ) that can be used to size an axial vane mechanism prior to conducting a detailed analysis of the mechanism using the Spreadsheet Design Program. The Axial-Vane PDQ program is based on fundamental thermodynamic modeling and does not include the effects of leakage, friction, and heat losses. The Axial-Vane PDQ program is written in Microsoft Visual Basic and permits the preliminary analysis and sizing of compressors, expanders, pumps, and diesel engines.

The second program, Axial Vane Design, includes a series of Microsoft Excel (Microsoft Corporation) spreadsheet programs have been developed by ATI to design and predict the performance of axial vane engines, compressors and expanders. These programs are stand-alone spreadsheets with all of the inputs, calculations, and outputs totally contained within the respective spreadsheet. The design program includes the effects of rotation of the rotor and vanes on the quantity and weight of air within each vane-chamber, the pressure and temperature within each chamber as it moves through the P-V cycle, leakage around the vanes, friction losses, the effects of fuel flow rates for diesel engines, and the resulting horsepower that is generated.

The current version of the Axial Vane Design Program is capable of designing Diesel engines and includes the effect of friction and heat losses. The program computes Indicated Horsepower (IHP) based on the Thermodynamic Power and then adjusts for friction and heat losses to compute Brake Horsepower (BHP). The spreadsheet program can also be used to predict the performance of Compressors, Expanders, and Pumps.

SECTION 4: EXECUTION OF PRELIMINARY DESIGN PROGRAM

The Axial-Vane Preliminary Design Program, AV-PDQ, is used for analysis of various design options and for approximate sizing of AV-devices prior to performing a detailed design using the Axial Vane Design Program. AV-PDQ is a Visual Basic program that requires Visual Basic_98 or a later version for execution.

The AV-PDQ program is started by implementing the RUN command in MS Windows. A window will appear giving options to analyze Axial Vane Devices such as Compressors, Expanders, Pumps, or Diesel Engines. The User selects the desired option and a new window appears requesting input for the device to be analyzed. As design variables are entered, the program provides prompts indicating reasonable input values that are based on prior design experience and operational limitations such as the maximum vane sliding velocity. The User may select the prompted values or input User defined values.

After all input variables are entered, the User selects the "Compute" option and the estimated power input or output is computed. The User may select the Print Option to obtain a copy of all input and computed values. When the User is satisfied with the preliminary sizing of the device, the results can be input into the Axial Vane Design Program for detailed analysis and design.

SECTION 5: EXECUTION OF THE AXIAL VANE DESIGN PROGRAM

The Axial-Vane Design Program has been constructed as a single large spreadsheet in the Excel program. The program includes an INPUT section that contains the design, operational variables, and the operating conditions. The program computes the properties in a typical chamber of the axial vane device as it rotates through 360°. The program computes the fluid mass, internal pressure, fluid temperature, combustion and heat loss, leakage, and friction losses in the chamber during the cycle. The output from the program is a listing of power input or output depending on the type of rotary device and related design information. For rotary Diesel engines, the program is capable of computing the Thermodynamic Horsepower, power losses due to friction and heat losses, the resultant Indicated Horsepower, Brake Horsepower, Brake Specific Fuel Consumption, and exhaust gas temperature.

5.1 User Inputs

All of the user inputs in the spreadsheet are have **RED** characters and are highlighted in YELLOW. A sample of the User Input section of the program is shown in Figure 5.1–1. Complete descriptions of all user inputs are listed in the following table:

Cell	Description	Units
J6	Vane Thickness – Width of Slot in Rotor	inches
J7	Vane Stroke – Maximum Distance Vane Moves Out of Rotor	inches
J8	Vane Height – Distance From Inner to Outer Diameter	inches
J9	Lift Angle – Angle on Cam From Minimum to Maximum Lift	degrees
J11	Event Angle – Angle on Cam From Start of Lift in One Direction to the Start of Lift in the Other Direction	degrees
J12	Speed – Rotation Speed of Shaft and Rotor	RPM
J15	Outer Diameter – Diameter of Rotor	inches
O4	Rotor Clearance – Axial Distance between Rotor and Cam	inches
M6	BMEP – Estimate of Brake Mean Effective Pressure	psi
O8	No. of Cam Lifts – Number of Cam Lifts per Revolution	
O9	No. of Vanes – Number of Vanes per Rotor	
O13	Minimum Volume per Chamber – Rotor Clearance Volume Per Chamber Plus Pocket Volume	cubic inches
O17	Pocket Length – Length of Pocket in Rotor Face	inches
W7	Tip Clearance – Distance From Tip of Vane to Cam	inches
W8	Diametral Clearance – Total Diametral Clearance Between Vane and Inner and Outer Wing	inches
BE6	Initial Pressure – Pressure in Chamber at Closure of the Intake Port	psia
BE7	Initial Temp – Temperature in Chamber at Closure of the Intake Port	°R.
CR5	R – Gas Constant	$\frac{ft-lb}{lb-°R.}$
CR6	k – Ratio of Specific Heats C_p/C_v	
CU5	Discharge Coefficient for Leakage Calculation	
EJ4	F/A Ratio – Fuel Flow Divided By Air Flow Estimate	
EO2	Fuel Lower Heating Value	Btu/lb
EO4	Effective Heating Value – Heating Value Reduced By Heat Transfer to Account for Heat Loss During Cycle	Btu/lb

TITLE OF ANALYSIS or DESIGN: **25,000 Horsepower Diesel Engine**
USER NAME: Mike Smith
DATE PERFORMED: May 26, 2005

			ROTOR CLEARANCE =	0.005	inches		
Estimated Displacement (cu.in.)	41,672	(Based on User Inputs)	ROTOR CLEAR VOL/CHAMB =	0.969	cu. In.	(Based on Rotor Clearance)	
VANE THICKNESS (inches)	1.00	25.40 mm	ESTIMATED BMEP	120.000	psi	(User Prelim. Est.)	
VANE STROKE (inches)	10.00		ESTIMATED BHP	21,092	BHP	(Based on Est. Disp. & Est. B	
VANE HEIGHT (inches)	20.00		No. OF CAM LIFTS =	2		180 Cam Cycle An	
LIFT ANGLE (degrees)	75.00	1.31 radians	NUMBER OF VANES =	12		(Limited to 12 in Spreadshee	
DWELL ANGLE (degrees)	15.00	0.26 radians	ANGLE BETWEEN VANES =	30		(Centerline of Vane to C.L. V	
EVENT ANGLE (degrees)	90.00						
SPEED (RPM) Auto Est. = 1754	1,670	(User Input RPM, Major BHP Variable)					
OMEGA (rev/sec)	174.88		MINIMUM VOL./CHAMBER =	128.000	cu. In.		
ROTOR PERIPHERAL VELOCITY (ft/min)	24,921		ROTOR CLEARANCE VOL. =	0.969	cu. In.		Piston Volume
OUTER DIAMETER OF ROTOR (in.)	57.00	(Major control of BHP)	POCKET VOLUME =	127.031	cu. In.		3,473.29
INNER DIAMETER OF ROTOR (in.)	17.00						868.32
MEAN DIAMETER OF ROTOR (in.)	37.00		POCKET LENGTH =	8.000	inches		
VOLUME FACTOR	6.46		CROSS SECT AREA =	15.879			
COMPRESSION RATIO	14.53		RATIO = 2 WIDTH =	5.635	DEPTH =	2.817704552	1,736.65
DISPLACEMENT/CHAMB (cu.in.)	1,736.65	**COMPUTED POWER**	RATIO = 3 WIDTH =	6.902	DEPTH =	2.300646133	128.33
TOTAL DISPLACEMENT (cu in)	41,679.51	**BHP = 25,029**	RATIO = 4 WIDTH =	7.970	DEPTH =	1.992417996	-45.00

R POSITION		VANE 1					VANE 2			REAL VOLUM	
RDegrees	Radians	LIFT	VELOCITY	VOLUME 1	VOLUME 2	NET VOL	LIFT	VANE VOL		AXIAL VANE	SINE WAVE
0	0.000	0.000	0.00	0.0	165.7	165.7	3.1	30.6	0	263.0	382.7
1	0.017	0.000	0.39	0.0	186.2	186.2	3.3	33.1	1	281.1	404.5
2	0.035	0.001	1.56	0.0	208.4	208.4	3.6	35.6	2	300.8	427.0
3	0.052	0.004	3.50	0.0	232.2	232.2	3.8	38.2	3	322.0	450.2
4	0.070	0.010	6.19	0.1	257.7	257.6	4.1	40.8	4	344.8	474.1
5	0.087	0.019	9.63	0.2	284.8	284.7	4.3	43.6	5	369.1	498.6
6	0.105	0.033	13.77	0.3	313.7	313.4	4.6	46.3	6	395.0	523.7
7	0.122	0.053	18.60	0.6	344.3	343.7	4.9	49.2	7	422.5	549.4
8	0.140	0.078	24.08	1.0	376.5	375.5	5.1	52.1	8	451.4	575.7

Figure 5.1–1.—Sample of INPUT SECTION of Axial-Vane Design Program

SECTION 6.0: PROGRAM OUTPUTS AND CELL BY CELL DESCRIPTIONS

A listing of each active cell in the spreadsheet is explained in the following section, with the exception of the input cells previously described in Section 5. Each cell includes a description and the formula used to calculate the value in the cell:

CELL	DESCRIPTION
J5	A simple estimation of the Total Displacement of the axial vane device per one revolution (OD is Outer Diameter, ID is Inner Diameter): Estimated Total Displacement Volume (cubic inches) = $\frac{\left(OD^2 - ID^2\right) \times \pi \times Stroke \times 2}{4} - VaneThickness \times Stroke \times Height \times NumberOfVanes \times 2$ $- ClearanceVolumePerChamber \times NumberOfVanes \times 2$
J10	Dwell Angle is an angular measurement in degrees of the length of the flat portion of the cam between rising and falling events: Dwell Angle (degrees) = $EventAngle - LiftAngle$
J13	Rotational speed of the device in radians per second: Ω = OMEGA = $\frac{Speed(rpm) \times 2 \times \pi}{60}$
J14	Peripheral Velocity is the velocity at the Outer Diameter: Peripheral Velocity (ft/min) = $\frac{OuterDiameter \times Speed \times \pi}{12}$
J16	The Inner Diameter (ID) is the diameter of the innermost radius of the cam Inner Diameter (inches) = $OuterDiameter - 2 \times VaneHeight$
J17	The Mean Diameter is simply the mean of the Outer and Inner Diameter: Mean Diameter = $\frac{OuterDiameter + InnerDiameter}{2}$
J18	The Volume Factor is the area of the cam face axially exposed for 1° of rotation. This number is used in calculating the volume of the chamber: Volume Factor (square inches) = $\frac{\pi \times \left(OD^2 - ID^2\right)}{4 \times 360}$

J19 Compression Ratio is the Maximum Volume in a chamber divided by the Minimum Volume in the chamber: Maximum and Minimum Volumes refer to the Real Volumes in Column Q.

Compression Ratio = $\frac{MaximumVolume}{MinimumVolume}$

J20 Displacement per Chamber is the volume displaced by each chamber calculated by subtracting the minimum volume from the maximum volume:

Displacement per Chamber = $MaximumVolume - MinimumVolume$

J21 Total Displacement of the device per revolution. This number should be very close to the value in cell J5, but is an accurate calculation of Total Displacement. It is used in all 4-stroke cycle engine calculations requiring the displacement of the device. When the device is a compressor or expander with more than one cam lift per revolution, the Displacement per Chamber must be multiplied by the number of cam lifts and number of vanes per revolution:

Total Displacement (cu. In.) =
$DisplacementPerChamber \times NumberOfCamLifts \times NumberofVanes \times 2$

A25:H564 The cells in this area of the spreadsheet are cam angles measured in both degrees and radians as noted and are used in calculating the vane travel (lift) for vanes one and two and the velocity of vane one.

I25:I564 This column is the lift of vane one. The formula used to calculate lift is a modified sine function as follows:

Lift (inches) = $VaneStroke \times \left(\frac{Angle}{LiftAngle} - \frac{Sin(2 \times \pi \times Angle / LiftAngle)}{2 \times \pi} \right)$

J25:J564 The velocity of Vane One is shown in this column:

Velocity (ft/min) = $\frac{VaneStroke \times \Omega \times LiftAngle}{12} \times \left(1 - Cos\left(\frac{2 \times \pi \times Angle}{LiftAngle} \right) \right)$

K25:K564 Column K along with column L is used to calculate the instantaneous volume of chamber one (chamber one is located between vane one and vane two). It is calculated by adding the volume created by the mean movement (lift) of vanes one and two times the volume factor of the previous volume.

Volume 1 (cubic inches) = $\frac{(Lift1 + Lift2) \times VolumeFactor}{2} + \Pr eviousVolume$

L25:L386 Volume 2 is shown in this column and is simply Volume 1 offset by 30° (this limits this spreadsheet to designing only a twelve vane device).

M25:M386 Column M is Volume 2 minus Volume 1 and is referred to as Net Volume measured in cubic inches. This is the net volume created by the vane motion and is used in column Q to create the Real Volume.

N25:N386 The Lift of Vane 2 is shown in Column N. It is calculated by offsetting the Lift on Vane 1 by 30°.

O25:O386 The volume displaced by the vanes is accounted for in this column. The vane displacement volume is shown in cubic inches:

Vane Volume (cu in) = $\frac{(Lift1 + Lift2) \times VaneThickness \times VaneHeight}{2}$

O5 This Cell is the Rotor Clearance Volume per Chamber in cubic inches, which consists of the minimum space between the face of the rotor and the cam:

Rotor Clearance Volume/Chamber = $\frac{\pi \times (OD^2 - ID^2) \times RotorClearance}{4 \times NumberOfVanes}$

O7 Estimated brake horsepower is computed in this cell based upon the assumed BMEP as a means of quickly sizing the device or engine. The actual brake horsepower is calculated in the spreadsheet program based upon factors including fuel air ratio, friction and heat losses:

BHP = $\frac{BMEP \times Speed \times Displacement}{396000}$

O16 Pocket Volume is the volume of the pocket in the face of the rotor that provides space for combustion when the rotor face is close to the face of the cam. The magnitude of the Pocket Volume is adjusted to control compression ratio:

Pocket Volume = $ClearanceVolume - RotorClearanceVolume$

O18 Cross Section Area is the average cross sectional area of the pocket in the face of the rotor calculated in square inches:

Cross Sect Area = $\frac{PocketVolume}{PocketLength}$

O19:O21 These three cells contain the average width of the pocket for three different ratios of width to depth of the pocket, two, three and four:

Width (inches) = $\frac{CrossSectArea}{Ratio}$

Q19:Q21 These cells contain the depth of the pocket for the same three ratios of Pocket width to depth:

Depth (inches) = $\sqrt{\dfrac{CrossSectArea}{Ratio}}$

Q25:Q386 The Real Volume is the actual instantaneous volume of Chamber One that is used for all of the performance calculations in units of cubic inches. The following two columns show the volumes for equivalent sine wave engines (Wankel) and piston type engines (with a stroke to length of connecting rod ratio of 2) using the same maximum and minimum volumes as the axial vane:

Real Volume (Axial Vane) = $NetVolume - VaneVolume + ClearanceVolume$

R19 Maximum Real Volume of the axial vane chamber used in calculating the volume of the sine and piston type engines.

R20 Minimum Real Volume of the axial vane chamber used in calculating the volume of the sine and piston type engines.

R16 Maximum Real Volume divided by two. Related to computed volume of Piston Engine.

R15 Maximum Real Volume multiplied by two. Related to computed volume of Piston Engine.

Z6 Leakage area between the tip of the vane and the face of the cam:

Tip Area = $TipClearance \times VaneHeight$

V13:AG373 Chamber Volume in cubic inches for all twelve chambers on one side of the rotor. These values are simply transferred from the Real Volume column (Q) and offset by the appropriate angle.

AH13:AS373 Vane Travel (lift) in inches for all twelve vanes.

AT13:BE373 Leakage Area around each of the twelve vanes including the Tip Area and the stroke times the diametral clearance:

Leakage Area (sq in) = $TipClearance + Stroke \times DiametralClearance$

BG8 Initial Density of the air in the chamber at the initial pressure and initial temperature conditions:

Initial Density (lbs/cu ft) = $\dfrac{Initial\,\mathrm{Pr}\,essure \times 144}{R \times InitialTemperature}$

BI8 Initial Density in lbs per cubic inch

BJ7 Initial weight of air in each chamber with the air at Initial Density:

Air Weight = $InitialDensity \times Max\,\mathrm{Re}\,alVolume$

BF13:BQ373 Weight of Air in Each Chamber calculated by taking the initial weight and subtracting the leakage thru the leading and trailing vanes.

BR13:CC373 Absolute Temperature of the Air in Each Chamber set to Initial Temperature while the intake port is open and calculated based on chamber pressure from that point on:

$$\text{Air Temperature (°R.)} = \Pr eviousTemp \times \left(\frac{\Pr eviousVol}{\Pr esentVol}\right)^{(k-1)}$$

CD13:CO373 Absolute Pressure of the Air in Each Chamber set to Initial Pressure while the intake port is open and calculated from that point on:

$$\text{Chamber Pressure (psia)} = \Pr evious\Pr ess \times \left(\frac{\Pr evVol}{\Pr esVol}\right)^{k} \times \left(\frac{\Pr esTemp}{\Pr evTemp}\right)$$

CP13:DA373 These columns calculate the leakage across each vane. To err on the conservative side, the flow calculation used is for choked or sonic flow realizing that some of the time the flow will actually be subsonic and produce less leakage:

Leakage Flow (lbs) =

$$Time/\deg \times 2.05 \times C_d \times Area \times Down\Pr ess\sqrt{\frac{\Pr essRatio^{\frac{k-1}{k}} \times \left(\Pr essRatio^{\frac{k-1}{k}} - 1\right)}{UpstreamTemp}}$$

CB13:DM373 The Pressure Ratios for the Leakage Flow equation are contained in these columns for each angle of rotation. The Pressure Ratio value is simply the larger of the upstream or downstream pressure divided by the smaller pressure value.

DN13:DY373 The cells in Column DN to DY contain the value of the downstream pressure for the leakage flow equation including the sign as to which direction the flow is going.

DZ13:EK373 The Upstream Temperature for the leakage flow calculation is contained in these columns.

EK6 The weight of fuel injected per chamber per revolution is calculated in this cell:

$$\text{Fuel Weight (lbs)} = \frac{Fuel/Air(ratio)}{AirWeight}$$

EK7 The weight of fuel injected per degree is the fuel weight per chamber divided by 30.

EK8 The amount of heat injected per degree is the Weight of Fuel Injected per Degree times the Effective Heating Value of the fuel times the ratio of injection duration (30°) to heat release duration (50°). This last ratio can be adjusted for faster or slower burning (heat release).

EL5 The Flow Rate of Air in lbs/min is presented:

Total Air Weight (lbs/min) = $AirWeight(lbs) \times NumberOfVanes \times Speed \times 2$

EM5 The Flow Rate of Air in lbs/hr is the rate in lbs/min times 60.

EL6 The Flow Rate of Fuel in lbs/min is presented:

Total Fuel Flow Rate (lbs/min) = $FuelWt(lbs) \times No.OfVanes \times Speed \times 2$

EM6 The Flow Rate of Fuel in lbs/hr is the rate in lbs/min times 60.

EP6 The Temperature Rise per degree due to fuel burning times the weight of air is:

Delta T (deg. R) = $\dfrac{FuelHeatPerDegree}{0.1725}$

EO8 The power equivalent for the total fuel flow is the fuel flow rate (lbs/min) times the heating value (Btu/lb)divided by (0.7068 times 60):

Fuel Power (HP) = $\dfrac{FuelFlow \times HeatValue}{0.7068 \times 60}$

EM13:EX373 The weight of fuel injected per degree is shown in these columns for the 30° period in which the chamber is rotating past the injector. The timing of the injection (injection location in the face of the cam) can be varied by moving these events.

EZ13:FK373 The Temperature Rise Per Degree Due to Fuel Burning is calculated by multiplying (Delta T) times the weight of air in one chamber.

FM13:FM373 This column contains the value for absolute pressure times the change in volume for each degree of rotation. The sum of the values in this column effectively integrate to give the value shown as follows:

FM375 P-V or the integral of pressure and volume are presented:

P-V (in-lbs) = $\int_0^{360} PdV = \sum FM13 : FM373$

FM376 The Indicated Horsepower per chamber is the integrated value of P-V times speed divided by 396,000.

FM377 The Total Indicated Horsepower is the power per chamber multiplied by the number of chambers (24).

FH1,FH2,FH3 These are three constants (**F1,F2,F3)** used in fitting a curve to model the engine friction:

$$\text{F1} = 35.377 \times \left(\frac{TotalVolume}{2000}\right)^{(-0.0443)}$$

$$\text{F2} = 5000 \times \left(\frac{TotalVolume}{2000}\right)^{(-0.3316)}$$

$$\text{F3} = \frac{Speed}{F2}$$

FG5 Friction Mean Effective Pressure (FMEP) is calculated in this cell using the three constants shown above with dimensions of psi:

$$\text{FMEP} = \frac{F1}{F3} \times \left(0.24531 - 0.987 \times F3 + 2.6 \times F3^2 - 1.575 \times F3^3 + 0.715 \times F3^4\right)$$

FH6 Friction Horsepower (FHP) is calculated in this cell:

$$\text{FHP} = \frac{FMEP \times Speed \times TotalVolume}{396000}$$

FH8 Brake Mean Effective Pressure is the mean pressure that when multiplied by the speed and displacement and divided by a constant yields Brake Horsepower (net usable shaft power):

$$\text{BMEP (psi)} = \frac{BrakeHorsepower \times 396000}{Speed \times TotalVolume}$$

FO13:FR373 These four columns give the exposed area for each chamber in which heat transfer takes place:

$$\text{Rotor (sq in)} = \frac{\pi \times \left(OD^2 - ID\ \right)}{4 \times NumberOfVanes}$$

$$\text{Outer Wing (sq in)} = PrevValue + \frac{\pi \times OD \times (Lift2 - Lift1)}{360 \times 2}$$

$$\text{Inner Wing (sq in)} = PrevValue + \frac{\pi \times ID \times (Lift2 - Lift1)}{360 \times 2}$$

$$\text{Cam (sq in)} = PrevValue + \frac{RotorArea \times (Lift2 - Lift1)}{360 \times 2}$$

FT13:FW373 The heat transfer coefficient for each surface area for each degree of rotation is calculated using heat transfer data from a NASA report titled “THERMAL AND STRUCTURAL ANALYSIS OF A LOW HEAT REJECTION ROTARY ENGINE COMBUSTION CHAMBER” under Contract NAS3-24344 prepared by ADAPCO, May 8, 1997. Two Figures from this report, Figure 1: “Engine Cycle Simulation Chamber Pressure”, and Figure 3: “Engine Cycle Simulation Heat Transfer Coefficient”, are presented below to show that the heat transfer coefficient is basically proportional to chamber pressure. Figure 1 shows a peak chamber pressure of 700 psi and Figure 3 shows a peak Heat Transfer Coefficient of about 3.4 BTU/IN^2-HR-°F. These referenced data were used to establish the following empirical equation to estimate heat transfer coefficients at the chamber pressures within the axial vane engine during a specific design cycle.

$$\text{Heat Transfer Coefficient (BTU/IN}^2\text{-HR-°F)} = \frac{3.4}{700} \times Pressure$$

FIGURE 1: ENGINE CYCLE SIMULATION CHAMBER PRESSURE

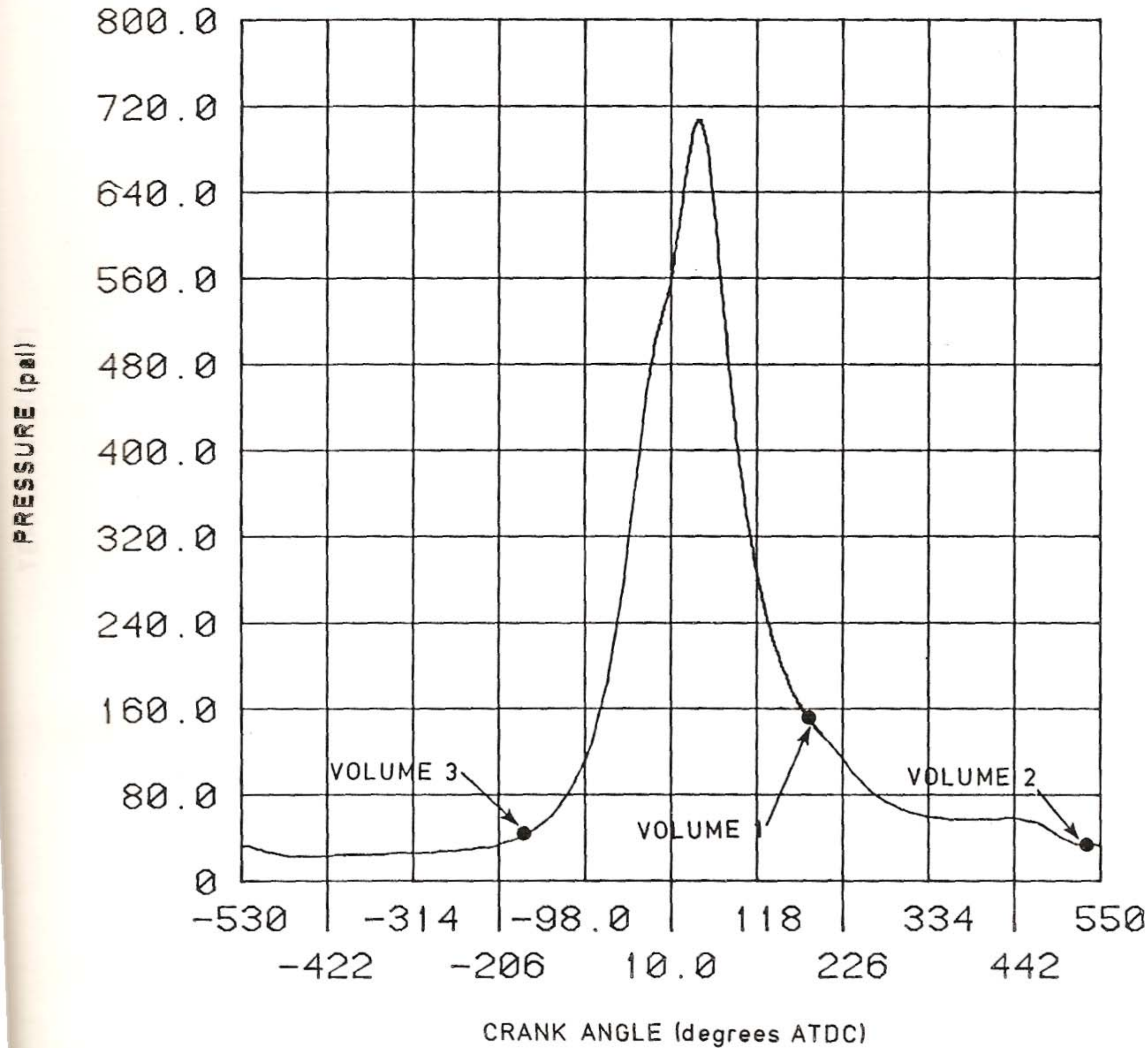
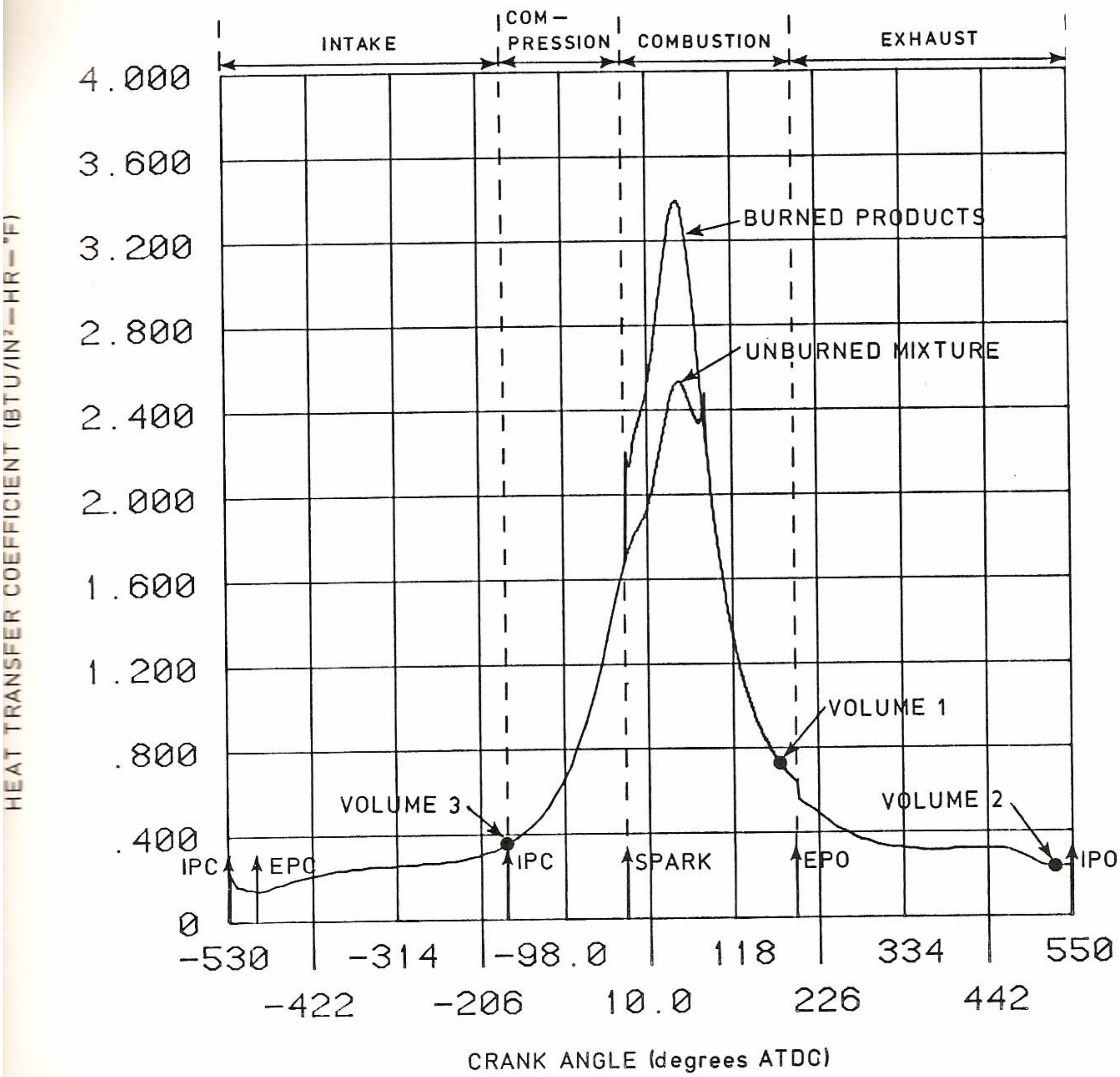


FIGURE 3: ENGINE CYCLE SIMULATION
HEAT TRANSFER COEFFICIENT



FY13:GB373 The Heat Transfer (BTU/degree) through each of the four surfaces is calculated in these cells using the heat transfer coefficient (HTC), the temperature of the gas, the temperature of the wall and the exposed areas:

$$\text{Heat Transfer (BTU/deg.)} = \frac{HTC \times Area \times (GasTemp - WallTemp)}{Speed \times 60 \times 360}$$

To make the heat transfer computations very conservative, a very cold wall temperature of 500 °R. was used for all walls. It is known that the walls will run hotter which will reduce the heat transfer significantly. To obtain realistic temperature profiles for each of the walls will require finite element modeling of the heat transfer processes across each wall.

GD13:GD373 The sum of the heat transfer through the four wall surfaces yields the total heat transfer in BTU/Degree). This is the instantaneous heat transfer per degree for the rotor, outer wing, inner wing and cam.

FY375:GB375 These cells contain the sums of the heat transfer per degree for the rotor, outer wing, inner wing and cam and have the units of BTU/revolution.

GB377 This cell contains the sum total of the heat transfer per revolution of the rotor, outer wing, inner wing and cam combined per chamber.

GB378 This cell is simply GB377 times speed and yields units of BTU/min per chamber.

GB379 This cell is GB378 times 60 to give BTU/hour per chamber.

GB380 This cell is GB378 divided by 60 to give BTU/sec per chamber.

GB381 The heat energy equivalent horsepower per chamber is GB380 divided by 0.706.

GB383:GB387 These cells are like the preceding five cells except that they represent all of the chambers in the device and simply multiply the cell (per chamber) by the number of chambers (No. of vanes X 2).

FV1:FV5 These five cells contain the equivalent heat flow (Btu/min) for the fuel input, the power output, friction, heat rejection and exhaust temperature. The only unknown quantity is the heat flow in the exhaust and it is calculated by subtracting the power output, friction power loss, and the heat rejection from the fuel input heat value.

FQ3 This cell is the specific heat for the exhaust mixture assuming an exhaust temperature of 1800 °F. and has a value of 0.275 Btu/lb/°F.

FQ4 This is the same heat flow number as calculated in FV5 and is calculated in the identical way.

FQ5 The engine exhaust temperature is calculated by the following equation:

$$\text{Exhaust Temp (°R)} = \frac{ExhaustEnergy}{C_p \times (WeightAir + WeightFuel)} + InitialTemperature$$

FQ6 $$\text{Exhaust Temp (°F)} = ExhaustTemp(°R) - 460$$

SECTION 7: DESIGN CONSIDERATIONS FOR DESIGN OF AXIAL VANE DEVICES

7.1 Discussion of Design Input Parameters and Design Aids:

Designing an axial vane device is an iterative process involving estimates of basic size and performance parameters followed by refinements to the design inputs after each design iteration. **As a guide to refining displacement and power estimates: displacement is roughly proportional the cube of linear dimensions and power is roughly proportional to the square of linear dimensions in all axial vane devices**.

7.1.1 Vane Thickness: The Vane Thickness is determined by vane mounting and support criteria, vane material properties, and two design parameters; vane stroke and chamber differential pressure . The peak stress in a vane is bending stress due to the pressure differential between the leading and trailing chambers surrounding the vane. Figure 7.1.1-1 is a plot of Vane Pressure Differential versus Vane Travel for a large Diesel type Axial Vane Engine (approx. 25,000 bhp) with a vane thickness of 3 inches and a vane stroke of 11 inches.

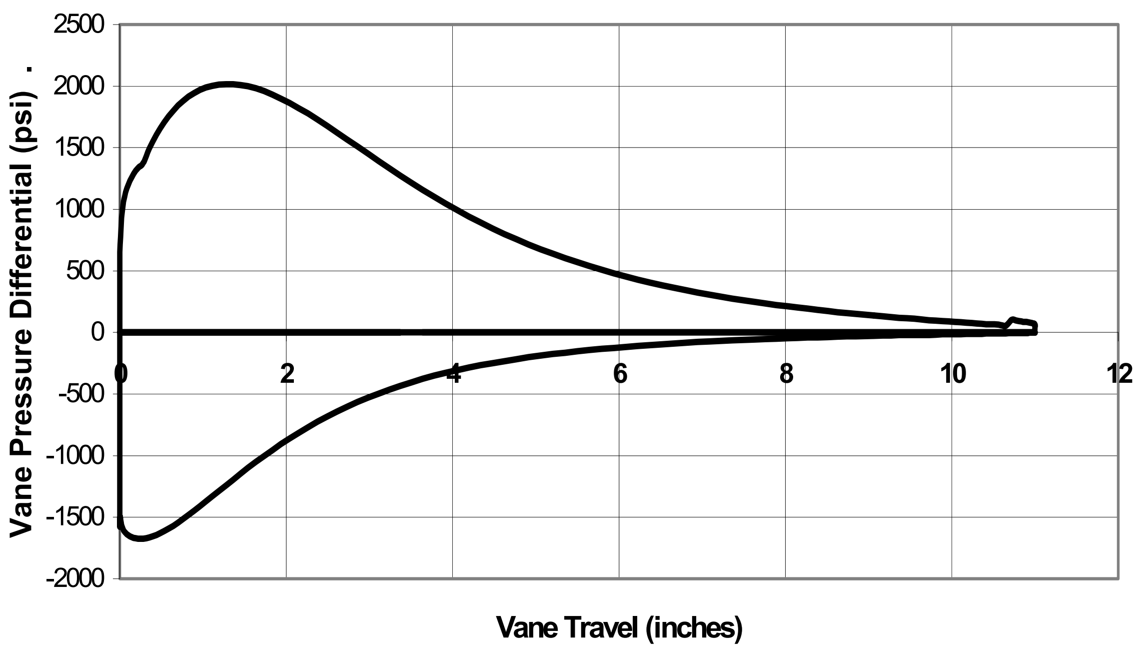


Figure 7.1.1-1.—Vane Pressure Differential versus Vane Travel

The Differential Pressure data in Figure 7.1.1-1 was generated using the ATI Axial Vane Analysis Program wherein the pressure in each chamber was computed for each 1° of rotation of the rotor. The pressure differential across the vane (“Vane Differential Pressure”) is computed as the difference in chamber pressure on the rear face of the subject vane less the chamber pressure on the advancing face of the advancing vane. Thus, during the intake stroke the chamber pressure on the rear vane face is less than the pressure on the advancing face and the Vane Pressure Differential (VDP) is considered to be negative (This sign convention is arbitrary; however, maximum bending stress occurs during the combustion/expansion stroke). Conversely, during the combustion/expansion stroke, the VDP is considered positive with the VDP producing positive vane bending stresses that tend to “bend” the vane toward the direction of rotation.

The term “Vane Stroke” refers to the maximum extension of the vane beyond the face of the rotor and is one of the most important parameters in the design of axial vane devices. Minimum vane extension occurs at the beginning of the intake cycle when the tip of the vane is nearly flush with the face of the rotor (i.e., Vane Travel = 0). The term “Vane Travel” refers to the instantaneous position of the tip of the vane with respect to the lateral face of the rotor. The maximum “Vane Travel” is equal to the “Vane Stroke” when the vane is fully cantilevered from the rotor.

In an axial vane Diesel engine, the 4-stroke Cycle is accomplished in one revolution of the rotor. Therefore, the vane reaches maximum stroke twice in each revolution of the rotor. For example, referring to Figure 7.1.1-1, the intake cycle begins with the vane tip nearly flush with the rotor (i.e., Vane Travel = 0). As the vane extends to maximum travel (i.e. Vane Stroke = 11 inches), the vane pressure differential remains approximately zero throughout the intake cycle. During the compression stroke, the Vane Pressure Differential increases from about zero to a negative value of about -1700 psi (i.e., chamber pressure on front face of the advancing vane is greater than pressure on the rear face). As the vane approaches the position of maximum compression and minimum vane extension (i.e., Vane Travel = 0), the VDP suddenly becomes positive with the chamber pressure on the vane rear face exceeding that of the front face. As the combustion/expansion stroke occurs, the Vane Travel moves from a position of minimum value to a position of maximum Vane Travel or the Vane Stroke length. During the exhaust stroke, the VDP is approximately zero as the vane moves from the position of maximum vane travel to the position of zero vane travel.

Figure 7.1.1-2 shows typical Vane Bending Stress (VBS) versus Vane Travel for the aforementioned large Axial Vane Diesel Engine. These data indicate the point in the vane travel history where the vane bending stress is maximized. As shown, the bending stress peaks at about 50% of the maximum Vane Travel (i.e, about 5.5 inches), also being about 50% of the design Vane Stroke. The VBS during the Intake Stroke and the Exhaust Stroke is approximately zero as the vane travels from positions of minimum travel to maximum travel and positions of maximum travel to minimum travel respectively.

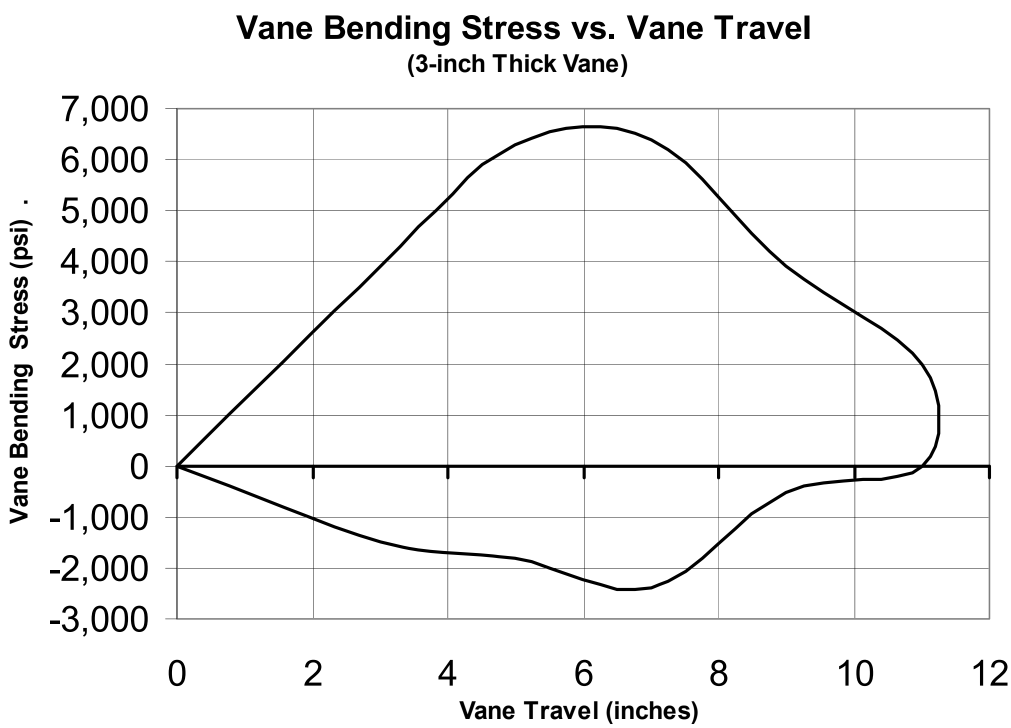


Figure 7.1.1-2.—Vane Bending Stress versus Vane Travel

Some generalizations and design guidelines are offered to expedite preliminary design of axial vane devices:

1. Maximum Vane Bending Stress occurs at approximately 50% of the design Vane Stroke (Ref. Figure 7.1.1.2).
2. At 50% Vane Stroke, the Vane Pressure Differential is approximately 30% of the maximum VDP computed by the Axial Vane Analysis Program (i.e., 560/2000 = 28% in Figure 7.1.1-1).
3. Vane Bending Stresses for non-cantilevered vanes should be evaluated using more sophisticated analysis tools such as finite element analysis.

Vane Thickness for a simple cantilevered vane can be estimated assuming uniform Vane Pressure Differential (VPD) across the vane and the vane at a position of 50% of design Vane Stroke. This leads to Figure 7.1.1-3 showing Vane Bending Stress versus Vane Thickness for different values of Vane Travel, per 1000 psi of VPD. The pressure at the midpoint in the vane stroke is about 30% of the peak pressure computed by the ATI Axial Vane Design Program (Fig. 7.1.1-1). It should be noted that the bending stresses discussed in this section are for vanes cantilevered from the rotor. Vane stresses in ATI Winged Rotor designs with grooves in the wings should be analyzed using finite element analysis codes based on the actual vane mounting configuration and the vane thicknesses should be adjusted accordingly.

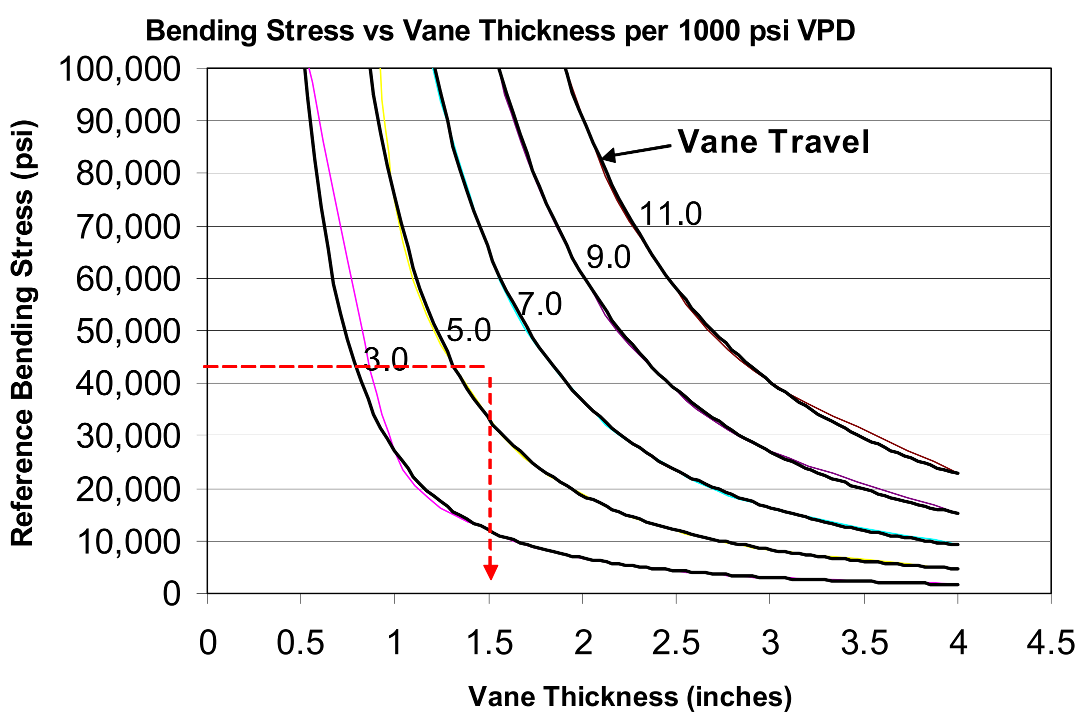


Figure 7.1.1-3.—Vane Bending Stress vs. Vane Thickness and Vane Stroke

Computing Vane Thickness

To estimate a preliminary vane thickness, use the total vane stroke and the allowable stress of the vane material in conjunction with Figure 7.1.1-3. To compute the vane thickness, first compute 30% of the peak VDP (0.30 x 2000 = 600 psi reference VDP). Secondly, the proposed material allowable must be adjusted by the ratio of the adjusted reference pressure (600 psi) to the chart baseline pressure of 1000 psi (i.e., 70,000 x 600/1000 = 42,000 psi Reference Bending Stress). Entering into the Reference Bending Stress chart at 42,000 psi, draw a horizontal line to intersect the Vane Travel plots at the half vane stroke point (5.5 inches). Then from that intersection, draw a vertical line to intersect the abscissa (Vane Thickness) at approximately 1.5 inches.

The assumption that chamber DP pressure is 30% of the peak chamber pressure is conservative as shown in Figure 7.1.1-2 where the chamber pressure at 50% of Vane Stroke is about 550 psi (27.5%). As the design is refined through several iterations, the vane thickness should be adjusted to reflect the computed maximum DP pressure for the design article.

7.1.2 Vane Stroke – Figure 7.1.2-1 is presented as a means of roughly predicting the required stroke of the vane as a function of desired power output for a naturally aspirated (non-turbocharged) engine with a BMEP of 120 psi. For turbocharged engines at higher BMEP's reduce the desired power by the ratio of the BMEP's.

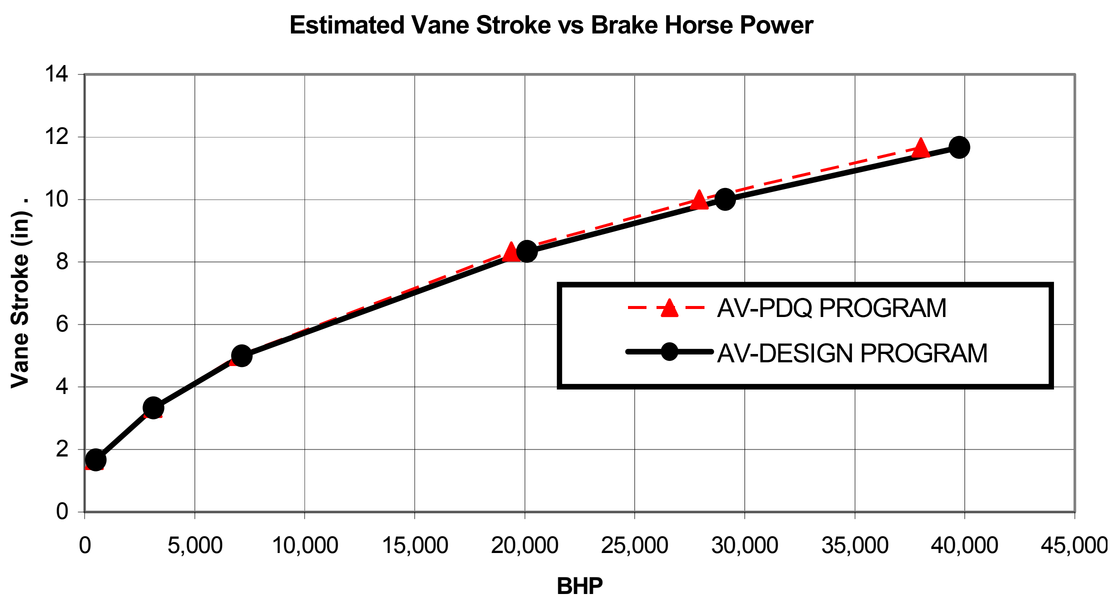


Figure 7.1.2-1.—Estimated Vane Stroke vs. Brake Horsepower

7.1.3 Vane Height – Vane Height is the radial dimension of the vane as it is installed in the rotor. Figure 7.1.3-1 is a graph that can be used to make a first estimate of vane height. Another way to make an initial guess is to make an estimate of the amount of device diameter that is needed for the shaft and bearings, subtract that value from the outer diameter of the rotor and divide the answer by two. For large displacement axial-vane devices the optimal vane height will be about twice the vane stroke. For devices with displacements below 100 cubic inches, the vane height to vane stroke ratio should be reduced to prevent vane to cam interference.

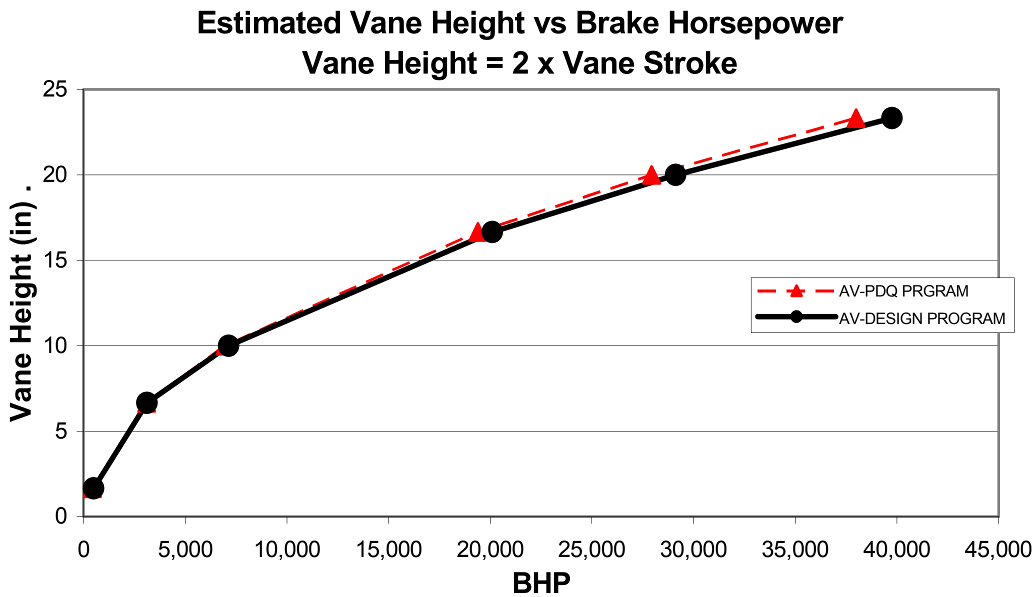


Figure 7.1.3-1.—Estimated Vane Height versus Brake Horsepower

7.1.4 Lift Angle – This measurement is the rotational angle from the start of vane travel in one direction to the end of vane stroke. For a twelve-vane mechanism this angle is between 60 and 75°. The smaller angle results in lower clearance volumes but higher velocities and accelerations of the vane.

7.1.5 Event Angle – This is the angle from the start of vane motion in one direction to the start of motion in the other direction. For devices with two cam lifts per revolution (such as a four stroke cycle engine) this angle is 90°.

7.1.6 Speed – The rotational speed of the axial vane rotor. The rotational speed and the outer diameter of the device determine the maximum velocities and accelerations for the vanes and the pin and pin track used to drive the vane. A rough rule of thumb for maximum speed of an axial vane device with the "winged rotor design" is 100,000 divided by outer diameter of the rotor in inches (i.e., 100,000/50 In. OD = 2,000 RPM).

7.1.7 Outer Diameter – The outer diameter of the rotor basically determines the size of the axial vane device. Figure 7.1.7-1 is a convenient means to make a first estimate of outer diameter for a desired power output of a Diesel engine. The results of the analysis show that the AV-PDQ program yields a reasonably accurate estimate of brake horsepower for preliminary estimates of engine power output versus rotor diameter.

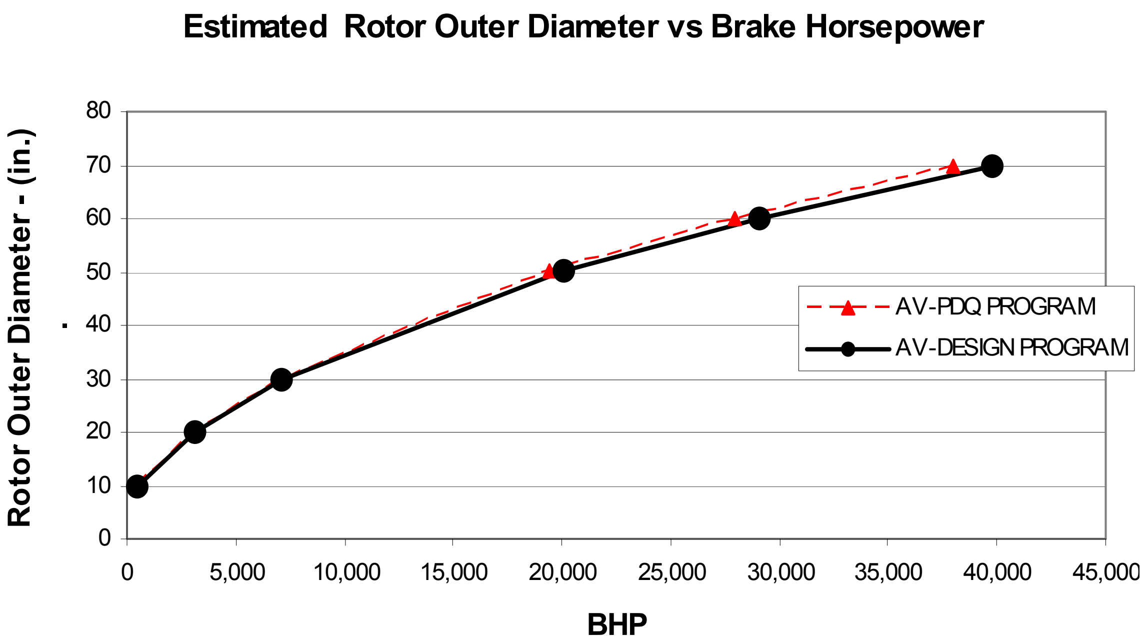


Figure 7.1.7-1.—Rotor Outer Diameter versus Power

7.1.8 Rotor Clearance – This is the axial clearance between the face of the rotor and the face of the cam when they are closest together at minimum vane stroke. This dimension should be as small as possible to minimize rotor clearance volume, which does not participate in the fuel burning process. A practical value for the clearance dimension is 0.005 inch when essentially zero axial motion is permitted by the bearings selected. If larger axial motions are allowed, then larger dimensions for rotor clearance are required.

7.1.9 BMEP – Brake Mean Effective Pressure (BMEP) measured in psi is a cycle averaged value that when multiplied by the displacement and the speed of an engine (times a constant) results in net, or brake power output. For naturally aspirated engines a typical BMEP is 120 psi. For turbocharged engines this value can be as high as 300 psi, but is more typically around 200 psi.

7.1.10 Number of Cam Lifts – This is the number of cam lifts per revolution. For four stroke cycle engines this value is 2. The number of cam lifts can be more than 2 for compressors and expanders.

7.1.11 Number of Vanes – Input the number of vanes per rotor. As a rule of thumb the minimum number of vanes is 8. Any number of vanes, odd or even, can be used making certain that the minimum distance between vanes (at the inner diameter) results in sufficient rotor strength and stiffness. The number is usually 12 vanes for Diesel engines less than 15 inches in diameter; however, large devices with diameters over 15 to 20 inches will be more efficient with an even larger number of vanes.

7.1.12 Minimum Volume – Minimum Volume is a measurement of the rotor clearance volume plus the combustion pocket volume. It is used to determine the compression ratio of the device and the required volume of the pocket (as the clearance volume is calculated and not input). The Minimum Volume is manipulated to change the Pocket Volume and produce the desired compression ratio. This ratio is typically around 15:1 for a Diesel cycle engine. Figure 7.1.12-1 provides a means to make an initial estimate of the required Minimum Volume.

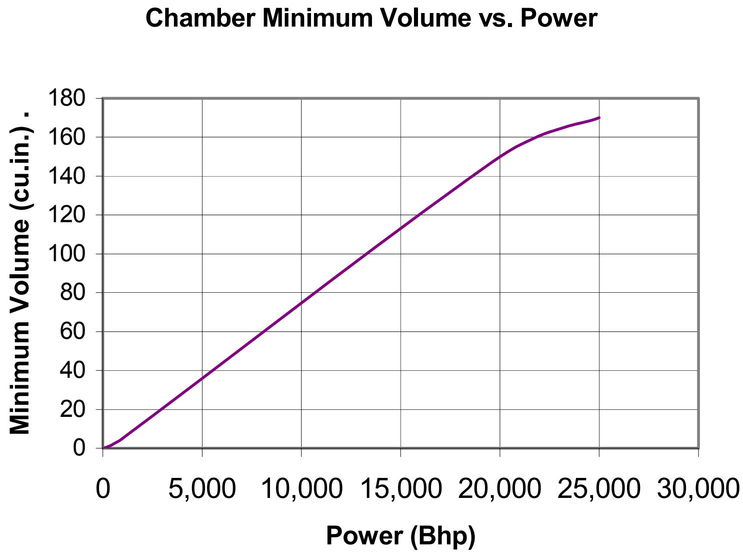


Figure 7.1.12-1 Estimated Chamber Minimum Volume versus Power

7.1.13 Pocket Length – This measurement is the arc length of the combustion pocket in the face of the rotor in each chamber. In the case of a Compressor or expander, the Pocket Length is zero. An easy means to set this value is to take the mean diameter between the inner and outer diameters and multiply this value by "π" to find the mean circumference. Subtract from this product the number of vanes times the thickness of the vanes. The result will be the maximum arc length available for construction of combustion pockets in the rotor. Divide the maximum arc length by the number of vanes to find the total maximum pocket lengths. Reduce this length by about an amount sufficient to support of the vanes on both ends of the pocket.

7.1.14 Tip Clearance – This dimension is the clearance between the tip of the vane and the face of the cam. This clearance is maintained by the mechanism used to position the vane. A typical value for this dimension is 0.005 inches with a larger dimension required for large devices to account for larger manufacturing tolerances and increased tolerances in the pin- and-pin track or slot-and-rail mechanism.

7.1.16 Diametral Clearance – Diametral Clearance is the Outer Diameter minus the Inner Diameter minus the Vane Height. This dimension needs to vary with engine size, materials, and detailed design of the engine. A value of 0.010 inch for a 10-inch diameter engine is typical. It should be noted that centrifugal force will cause the outer edge of the vane to have zero clearance at the outer wing and that most of the Diametral Clearance will occur at the inner edge of the vane where it travels in the slot of the inner wing.

7.1.17 Initial Pressure – The absolute pressure in the chamber at the point that the intake port is closed. A good approximation of this value for a naturally aspirated engine is to use atmospheric pressure. The value of atmospheric pressure will of course vary with altitude. In the case of a turbocharged engine, the turbocharger will increase the initial pressure depending on the pressure ratio of the turbocharger compressor.

7.1.18 Initial Temperature – This is an absolute temperature value determined in exactly the same way as initial pressure.

7.1.19 R – Gas Constant – This is also referred to as the Universal Gas Constant.

For air, R has a value of 53.3 $\frac{ft-lb}{lb-°R}$.

7.1.20 k – Ratio of Specific Heats – The Ratio of Specific Heats, symbolized as a lower case k, is the ratio of $\frac{Cp}{Cv}$. For air at normal sea level conditions k has a value of about 1.4.

7.1.21 C – Discharge Coefficient – A discharge coefficient is required in calculating leakage of air across the vane. A typical value to use for this calculation is 0.70.

7.1.22 F/A Ratio – F/A is an abbreviation for fuel air ratio, which is the mass flow rate of air divided by the flow rate of fuel with both measurements in pounds. A value of 0.04 to 0.05 should be used for naturally aspirated engines and a value of 0.025 to 0.030 for turbocharged axial vane engines. The larger this number, the richer the mixture, which results in higher combustion temperatures.

7.1.23 Fuel Heating Value – This is a measurement of the energy available in a pound of fuel measured in Btu/lb. A typical value for Diesel fuel is 18,600 Btu/lb.

7.1.24 Effective Lower Heating Value – This value is used as a means to circumvent a problem inherent in spreadsheets known as “circular references”. The only way to get around this problem is to use manual iteration. In this case the heat losses are calculated using the chamber pressure as one of the variables. Unfortunately, chamber pressure is dependant upon the amount of heat loss. To get around this problem the percentage of input fuel energy, which is lost because of heat transfer, is calculated and displayed in cell FB6. This value is automatically input into cell EO5 to reduce the total fuel heat input by the amount of heat lost. The Effective Heating Value should be adjusted for each change of parameters in the design as the optimal sizing of the engine is achieved. The computed ratio will vary with each change of parameters in the design and should be re-entered into cell EO5 until the output values converge. This process should typically not require more than three iterations.

SECTION 8: INSTRUCTIONS FOR DESIGN OF AXIAL-VANE DIESEL CYCLE ENGINE

A Diesel engine is the most complex device that can be analyzed or designed using the Axial Vane Design Program. Therefore, design of a Diesel engine has been used in the attached Axial Vane Design Program to illustrate use of the spreadsheets and interpretation of the results. The program was designed to operate using Microsoft Windows 2000 or a comparable operating system and Microsoft Excel. Initial User Input values for the program can be obtained from a preliminary analysis using the associated Axial-Vane Preliminary Design Program, AV-PDQ, or may be estimated from the charts provided in Section 7. Listed below is a guide for use of the Axial Vane Design Program.

A. Open the spreadsheet file titled "Axial Vane Design.xls"
B. The spreadsheet analysis can be started by pressing the first spreadsheet, 1 AV Design, in the Excel Workbook.
C. Each design parameter and spreadsheet cell location is listed below. Fill in the input values, which have red text and yellow background as follows:

I	Vane Thickness (inches)	J6
ii	Vane Stroke (inches)	J7
iii	Vane Height (inches)	J8
iv	Lift Angle (degrees)	J9
v	Event Angle (degrees)	J11
vi	Speed (RPM)	J12
vii	Outer Diameter (inches)	J15
viii	Number of Cam Lifts	O8
ix	Rotor O.D. Clearance (inches)	O4
x	Estimated Brake Mean Effective Pressure (psi)	O6
xi	Number of Vanes	O12
xii	Minimum Volume/Chamber (cubic inches)	O13
xiii	Pocket Length (inches)	O17
xiv	Tip Clearance (inches)	W7
xv	Diametral Clearance (inches)	W8
xvi	Initial Pressure (psia)	BE6
xvii	Initial Temperature (Degrees R.)	BE7

D. When values for all parameters have been input, the spreadsheet computations are automatically processed. The Effective Heat Value of the fuel (Cells EO5 & FB6) MUST BE MANUALLY ITERATED until the effective heat values in the two cells converge. The results of the analysis, including User Inputs and Outputs are listed in the PRINTOUT on Sheet 2 of the Spreadsheet
E. The results of the computations are also presented in graphical form at the bottom of the spreadsheet. Graphs of the various outputs for all chambers are shown in charts located at the bottom of the spreadsheet. Each chart is located immediately below the

section of the spreadsheet where the related values were computed. The spreadsheet program computes values for each 1° of rotation of each chamber in the AV-device. Each chart shows a plot of the related values for all chambers through a 360° revolution of the rotor.

F. The spreadsheet also contains detailed graphs of the parameters for a single chamber as it moves through one complete revolution. Each of the graphs can be called up via the tabs on the bottom of the spreadsheet as follows:

   Sheet 2: PRINTOUT - Summary of Performance Analysis (Inputs & Outputs)

   Sheet 3: Engine Types -Volume vs. Shaft Angle for Three Engine Types

   Sheet 4: Vane Travel

   Sheet 5: Chamber Volume

   Sheet 6: Leakage Area

   Sheet 7: Weight of Gas in Chamber

   Sheet 8: Temperature of Gas in Chamber

   Sheet 9: Pressure of Gas in Chamber

   Sheet 10: P-V Diagram

G. A summary of the input variables and the results of the computations are presented in the "Summary of Performance Analysis" table contained in Sheet 2 of the Spreadsheet Program (2 PRINTOUT).

H. The graph "Three Engine Types" (Worksheet 3) compares the cylinder/chamber volume relationships versus shaft angle for a Piston Engine, a Wankel Engine, and an ATI Axial Vane Device. The graph illustrates the volumetric superiority of the ATI Axial Vane Device.

I. Repetitive analyses can be conducted by adjusting the dimensions of the axial-vane device to produce the desired output. In the case of Diesel engines, the Minimum Volume per Chamber (Cell O13) should be changed as required to adjust the Compression Ratio by changing the Combustion Pocket Volume (Cell O16). The Brake Horsepower is most effectively changed by adjusting the Outer Diameter of the Rotor and re-scaling the sizing of vanes and compression ratio as explained above.

SUMMARY OF PERFORMANCE ANALYSIS

DESIGN PROGRAM FOR AXIAL VANE MACHINES
Developed by Advanced Technologies Inc.
March 2005

25,000 Horsepower Diesel Engine

{Title of Design: From Sheet 1 AV DESIGN)

PARAMETER	VALUE	COMMENTS & NOTES
PHYSICAL DIMENSIONS & DESIGN VARIABLES		
NUMBER OF VANES	12	12-Vane Diesel Engine
NUMBER OF CAMS	2	
ANGLE BETWEEN VANES	30	
VANE THICKNESS (inches)	1.00	Estimated = 0.10 x Vane Stroke
VANE STROKE (inches)	10.00	Estimated from Chart: Vane Stroke vs. BHP
VANE HEIGHT (inches)	20.00	Estimated as 2 x Vane Stroke
LIFT ANGLE (degrees)	75	
DWELL ANGLE (degrees)	15	
EVENT ANGLE (degrees)	90	
ROTOR OUTER DIAMETER (inches)	57.00	Estimated based on Chart: OD vs. BHP
ROTOR INNER DIAMETER (inches)	17.00	Estimated = OD -2 Vane Height
ROTOR MEAN DIAMETER (inches)	37.00	
INPUT DESIGN VALUES		
SPEED (RPM)	1,670	Limited by Vane Rubbing Velocity
OMEGA (radians/sec)	175	
ROTOR CLEARANCE (inches)	0.005	Minimum clearance between Rotor & Cam
TIP CLEARANCE (inches)	0.005	Clearance between Vane Tip and Cam
DIAMETRAL CLEARANCE (inches)	0.010	Total radial clearance between vane & Inner & Outer Wings
INITIAL PRESSURE (psia)	14.70	Inlet Pressure
INITIAL TEMPERATURE (deg R)	530.00	Inlet Temperature
Gas Constant, R =	53.30	
Ratio of Specific Heat, k =	1.3937	
HEAT VALUE (BTU/lb)	18,600	
EFFECTIVE HEAT VALUE (BTU/lb)	15,693	Computed in Cell FB6 based on cooling Heat Loss that
FUEL/AIR RATIO	0.0390	varies with Chamber Pressure & Temperature.
AIR/FUEL RATIO	25.64	
COMPUTED VALUES		
DISLACEMENT/CHAMBER (cu. in.)	1,736.65	Adjust Displacement/Chamber to change Comp. Ratio
POCKET VOLUME (cu.in.)	127.03	Pocket Volume is automatically adjusted when Disp. Per
ROTOR CLEARANCE VOLUME (cu.in.)	0.9687	Chamber is changed.
TOTAL DISPLACEMENT (cu. in.)	41,680	Total Displacement of 24 chambers.
COMPRESSION RATIO	14.53	Ratio of Max. Real Vol. / Min. Real Vol.(Col.
COMPUTED PERFORMANCE		
AIR FLOW (lbs/min	3,018.37	
FUEL FLOW (lbs/min)	117.72	
BRAKE MEAN EFFECTIVE PRESSURE (BMEP)(psi)	142.40	
INDICATED POWER (hp)	29,857	
FRICTION POWER (hp)	4,829	
HEAT LOSS POWER (hp)	3,830	
BRAKE HORSEPOWER OUTPUT (Bhp)	25,029	
BRAKE SPECIFIC FUEL CONSUMPTION (lbs/Bhp/hr)	0.2822	
EXHAUST TEMPERATURE (deg. F.)	1738	

Performed By: Mike Smith Date Performed: May 26, 2005

Figure 8–1.—Sample of PRINTOUT from Axial-Vane Design Program

REPORT DOCUMENTATION PAGE					*Form Approved* *OMB No. 0704-0188*
The public reporting burden for this collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing this burden, to Department of Defense, Washington Headquarters Services, Directorate for Information Operations and Reports (0704-0188), 1215 Jefferson Davis Highway, Suite 1204, Arlington, VA 22202-4302. Respondents should be aware that notwithstanding any other provision of law, no person shall be subject to any penalty for failing to comply with a collection of information if it does not display a currently valid OMB control number. PLEASE DO NOT RETURN YOUR FORM TO THE ABOVE ADDRESS.					
1. REPORT DATE *(DD-MM-YYYY)* 01-07-2008		**2. REPORT TYPE** Final Contractor Report			**3. DATES COVERED** *(From - To)*
4. TITLE AND SUBTITLE System Study for Axial Vane Engine Technology					**5a. CONTRACT NUMBER**
					5b. GRANT NUMBER
					5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S) Badgley, Patrick, R.; Smith, Michael, R.; Gould, Cedric, O.					**5d. PROJECT NUMBER** NNC-04VN19P
					5e. TASK NUMBER
					5f. WORK UNIT NUMBER WBS 984754.02.07.03.11.02
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) Advanced Technologies, Inc. P.O. Box 850 Starkville, Mississippi 39760					**8. PERFORMING ORGANIZATION REPORT NUMBER** E-16418
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) National Aeronautics and Space Administration Washington, DC 20546-0001					**10. SPONSORING/MONITORS ACRONYM(S)** NASA
					11. SPONSORING/MONITORING REPORT NUMBER NASA/CR-2008-215175
12. DISTRIBUTION/AVAILABILITY STATEMENT Unclassified-Unlimited Subject Category: 07 Available electronically at http://gltrs.grc.nasa.gov This publication is available from the NASA Center for AeroSpace Information, 301-621-0390					
13. SUPPLEMENTARY NOTES					
14. ABSTRACT The purpose of this engine feasibility study was to determine the benefits that can be achieved by incorporating positive displacement axial vane compression and expansion stages into high bypass turbofan engines. These positive-displacement stages would replace some or all of the conventional compressor and turbine stages in the turbine engine, but not the fan. The study considered combustion occurring internal to an axial vane component (i.e., Diesel engine replacing the standard turbine engine combustor, burner, and turbine); and external continuous flow combustion with an axial vane compressor and an axial vane turbine replacing conventional compressor and turbine systems.					
15. SUBJECT TERMS Internal combustion engines; Diesel engines; Turbomachinery; Thermodynamic cycles					
16. SECURITY CLASSIFICATION OF:			**17. LIMITATION OF ABSTRACT**	**18. NUMBER OF PAGES**	**19a. NAME OF RESPONSIBLE PERSON** STI Help Desk (email:help@sti.nasa.gov)
a. REPORT U	**b. ABSTRACT** U	**c. THIS PAGE** U	UU	78	**19b. TELEPHONE NUMBER** *(include area code)* 301-621-0390