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COMPRESSOR CONFIGURATION AND DESIGN OPTIMIZATION FOR THE HIGH RELIABILITY GAS TURBINE

Final Report

By
Fred H. Boenig
Diana L. Day

April 1980

Work Performed Under Contract No. AC03-79ET15425

United Technologies Corporation
Power Systems Division
South Windsor, Connecticut

and

Pratt & Whitney Aircraft
Government Products Division
West Palm Beach, Florida

U. S. DEPARTMENT OF ENERGY

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DOE Technical Monitor
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ABSTRACT

The purpose of this program has been to develop a preliminary design of a low aspect ratio/high through-flow compressor configuration to be compatible with the Electric Power Research Institute/Department of Energy/Reliable Advanced Liquid Fueled Engine (EPRI-DOE/RALFE) program and to evaluate the design for use in the Reliable Engine.

Our objective was to define the benefits of low aspect ratio and high through-flow (HTF) in a large industrial gas turbine in which high reliability and cost-of-electricity (COE) are major design considerations. These benefits have been identified, in aircraft gas turbines, as reduced number of stages, with reduced number of parts, and increased aerodynamic loading capability. The impact of this design concept on Cost-of-Electricity (COE) and reliability is the determining factor in evaluating these benefits. To make full use of the High Through Flow (HTF) concept it was necessary to design a diffuser to decelerate the flow from the exit of the new Reliable Engine compressor to the combustor that has been configured in the EPRI program. This required some innovative analyses of higher inlet Mach number diffusers.

The compressor and diffuser preliminary designs have been completed to define size and performance characteristics. The compressor has 9 stages and a predicted adiabatic efficiency of 88.35%. The diffuser selected is a conventional straight-wall configuration with an equivalent conical angle of 8-degrees. An alternate diffuser configuration has also been recommended because of its excellent performance potential in high Mach number applications. The HTF compressor configuration appears to offer equivalent COE and reliability as compared to the Baseline Reliable Engine configuration, but at more conservative aerodynamic loading levels.

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SUMMARY

Based on the results of the EPRI Reliable Engine High Through-Flow compressor conceptual trade activities, three candidate high through-flow compressor configurations were selected for preliminary design optimization. The configurations incorporated low aspect ratio, low inlet tip speed, and low inlet specific flow in a compressor with a higher exit Mach number than the baseline Reliable Engine design. Flowpaths of the three configurations are shown in Figure S-1. Case 1 has a nearly constant outside diameter; case 2 has a constant mean diameter; and case 3 has features of both cases 1 and 2.

Following the preliminary optimization study, one candidate was selected as the HTF configuration for the balance of the program. It represented the only configuration that could satisfy the physical constraints of the Reliable Engine design, and showed the most potential to minimize Cost-of-Electricity. Design optimization of the HTF compressor included both one-dimensional and two-dimensional analyses. The final configuration was a 9-stage, 848 lb/sec compressor with a pressure ratio of 14:1 and a predicted adiabatic efficiency of 88.35 percent.

A diffuser also was configured to decelerate the flow from the compressor exit to the combustor with minimum total pressure loss. The diffuser selected was a conventional straight-wall design. An alternate diffuser configuration also was considered as having potentially better performance.

Mechanical design studies were completed to verify the feasibility of the design, complete preliminary sizing, and incorporate the new compressor and diffuser into the Reliable Engine cross-section. The cross-section with the HTF compressor is shown in Figure S-2.

Reliability of the selected HTF compressor was calculated to be slightly better than the base case Reliable Engine compressor design, with a Mean Time Between Failure (MTBF) of 54,348 hours (HTF) vs 54,054 hours Reliable Engine; a 0.54 percent increase. The Cost-of-Electricity (COE) for a power plant incorporating

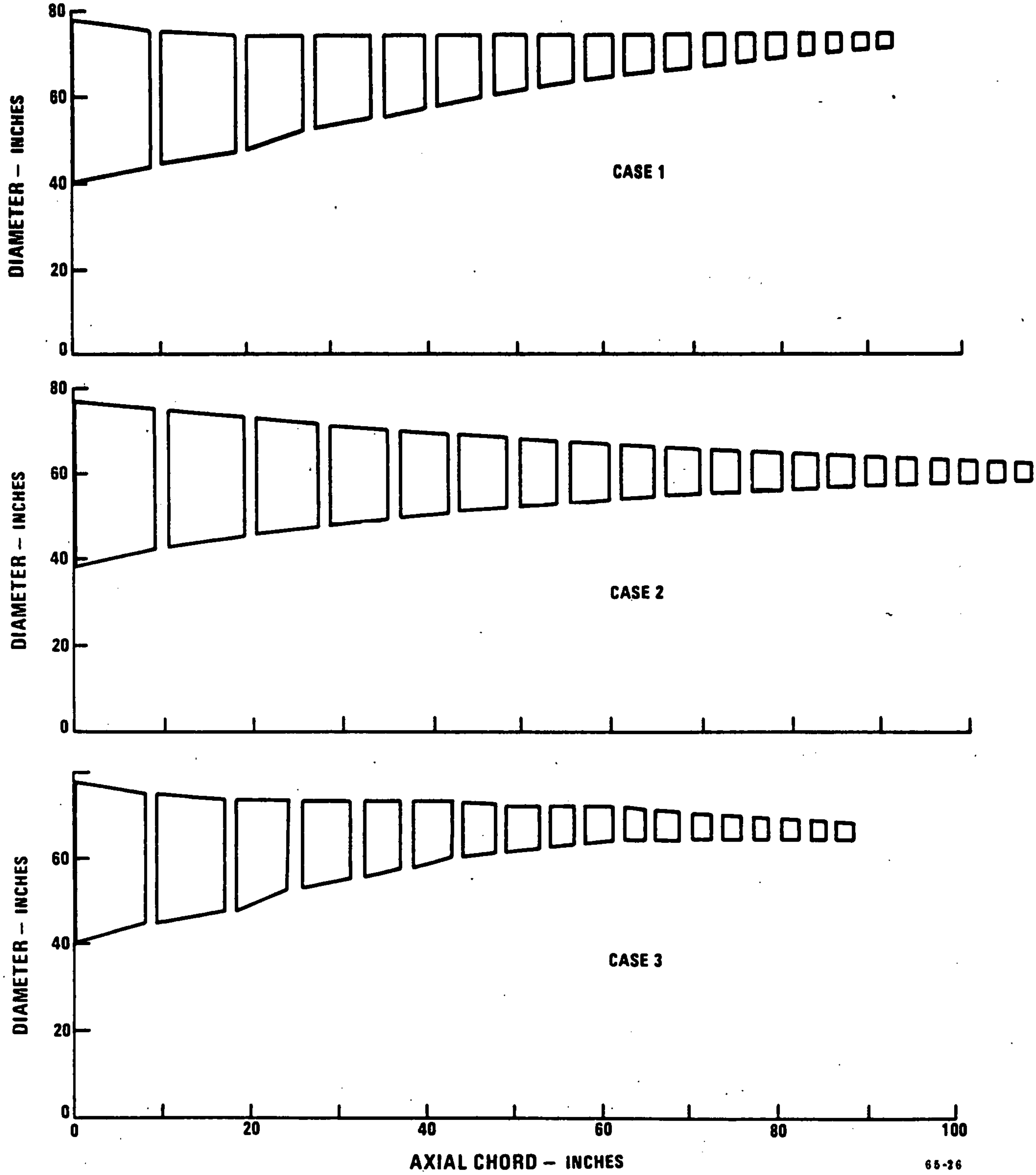
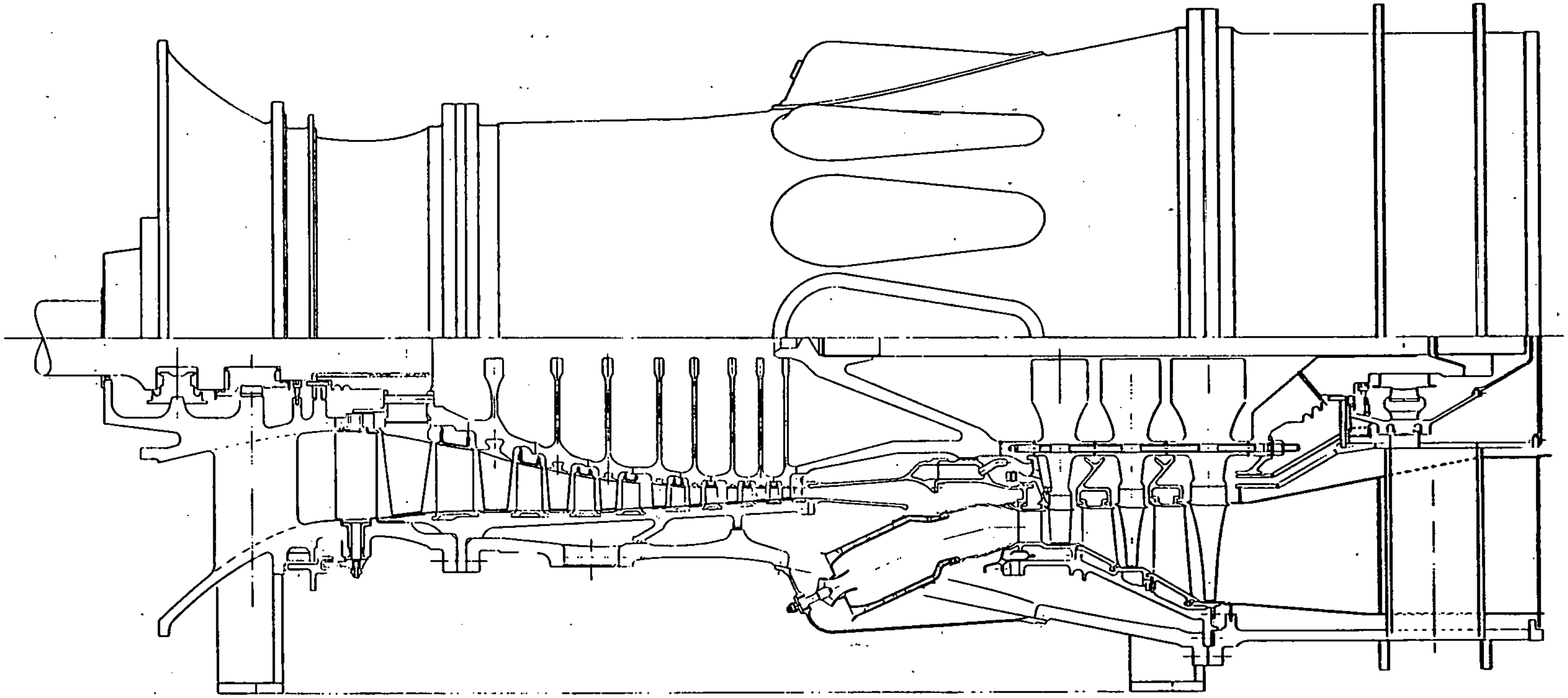


Figure S-1. Flowpaths for Three High-Throughflow Compressor Candidates



65-37

Figure S-2. Reliable Engine Cross-Section the HTF compressor was 90.369 mills per kilowatt-hour; an increase from the Reliable Engine configuration of 0.26 mills per kW hr (0.29 percent). The HTF compressor exhibited considerably lower aerodynamic loading levels. Considering the preliminary nature of these designs and the small performance and reliability differences, the HTF configuration appears to be equivalent to the Reliable Engine configuration.

In an industrial, baseload application, the efficiency of the gas turbine will have great impact on the power plant COE. Both the HTF and the baseline compressor designs could benefit from further detailed analysis directed toward increased efficiency. For example, the effect that advanced subsonic airfoil design would have on the efficiency of the compressor should be determined. Secondly, a design study of the alternate "bled diffuser" configuration could provide an advanced technology diffuser, with potentially improved performance.

Section 1

INTRODUCTION

To meet increasing demand for more reliable electric power generation facilities, the Electric Power Research Institute (EPRI) contracted with United Technologies Corporation (UTC) to design a new more reliable combined-cycle power plant. The compressor of the turbine engine to be used in the power plant has undergone substantial parametric study under EPRI contracts. The primary objectives of the studies were to maximize reliability and minimize Cost-of-Electricity (COE).

Background Analyses

Parametric studies were performed for EPRI on the major compressor variables of inlet, tip speed, inlet specific flow, average aspect ratio, and exit Mach number using an agressive compressor design as the baseline configuration. The parametric ranges are shown in Figure 1-1.

Results of the first parametric analyses indicated optimum performance could be attained with low inlet specific flow, low average aspect ratio and low inlet tip speed. One configuration from the EPRI study was selected as the baseline for further conceptual design work and was ultimately selected for the Reliable Engine. The flow path for this constant outside diameter (COD) compressor is shown in Figure 1-2.

A second parametric study investigated the effects of High Through-Flow velocity on the compressor performance and configuration. High Through-Flow (HTF) has been shown to have benefits in gas turbine compressors which include potentially fewer stages and airfoils or lower blade loadings for a given number of stages and airfoils, with no detriment to reliability. The benefits in an industrial gas turbine application in which reliability and COE are major objectives was previously undetermined. The results of the two parametric matrices are illustrated in Figures 1-3 through 1-7. These figures show trends in length, efficiency and number of airfoils vs. each of the parametric variables.

RELIABLE ENGINE COMPRESSOR MATRIX

		AVG AR = 1.20			AVR AR = 1.50			AVG AR = 1.80		
FLOWPATH	SPEC. FLOW / TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	10		14				15		11
COD	37.5					6				
COD	35.0	22		18				19		23
CMD	40.0					2				
CMD	37.5		8		4	1	5		9	
CMD	35.0					3				
CID	40.0	24		20				21		25
CID	37.5					7				
CID	35.0	12		16				17		13

AR1 (CMD rows) AR2 (CID rows)

FREE TURBINE: B (1250) A (1350)

DESIGN NUMBER

HTF RELIABLE ENGINE COMPRESSOR MATRIX

		AVG AR = 1.20			AVR AR = 1.50			AVG AR = 1.80		
EXIT MACH No.	SPEC. FLOW / TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
.55	40.0	H10		H14				H15		H11
.55	37.5					H6				
.55	35.0	H22		H18				H19		H23
.45	40.0					H2				
.45	37.5		H8		H4	H1	H5		H9	
.45	35.0					H3				
.35	40.0	H24		H20				H21		H25
.35	37.5					H7				
.35	35.0	H12		H16				H17		H13

DESIGN NUMBER

65-27

Figure 1-1. Reliable Engine and HTF Compressor Matrices

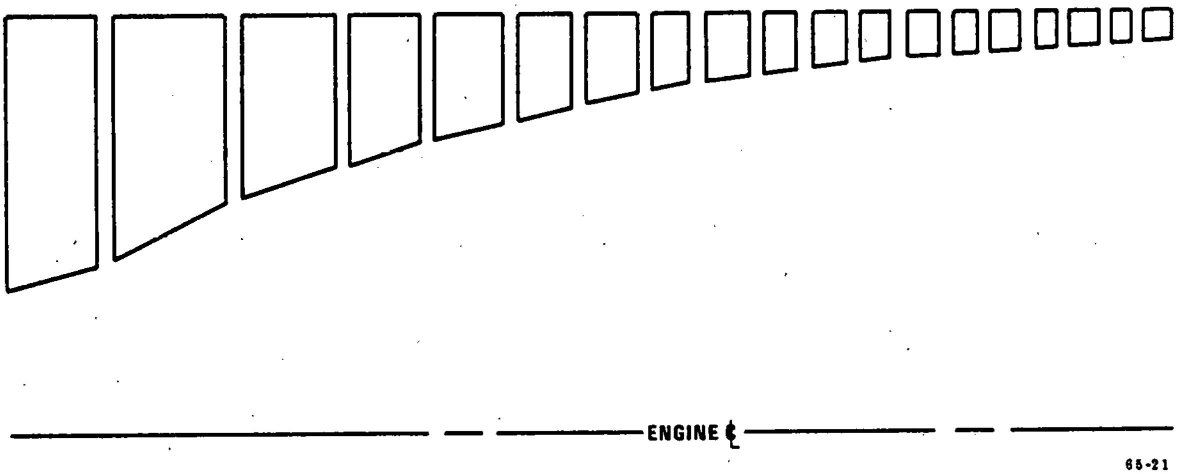


Figure 1-2. Reliable Engine Compressor Flowpath

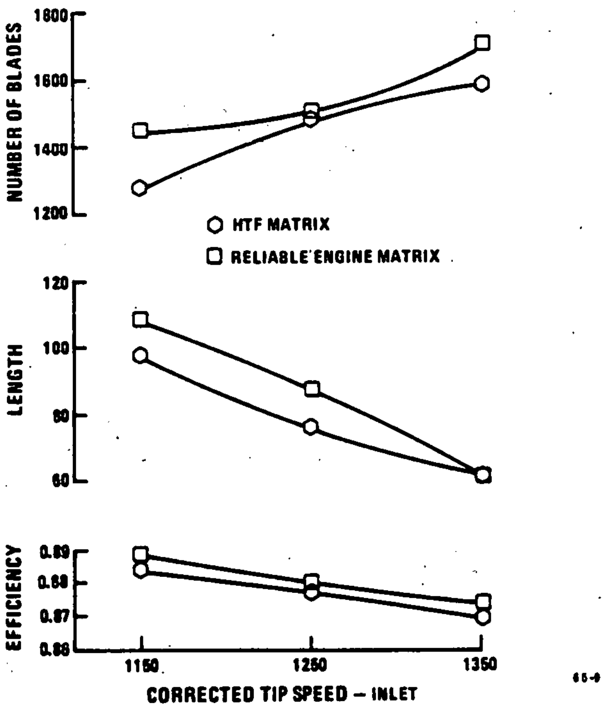


Figure 1-3. Effects of Tip Speed

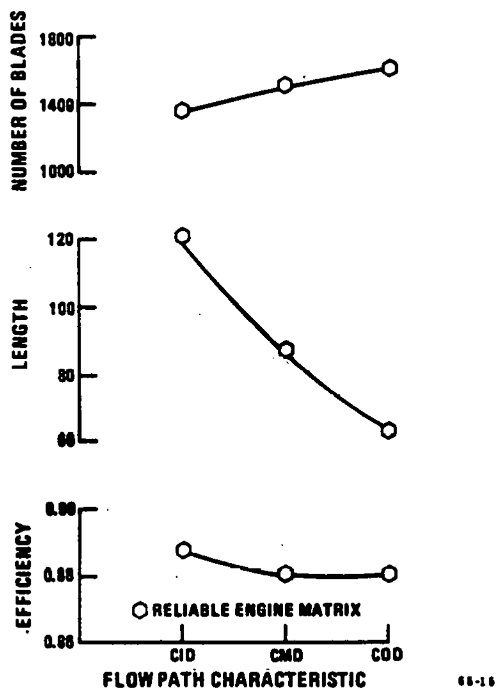


Figure 1-4. Effects of Flowpath

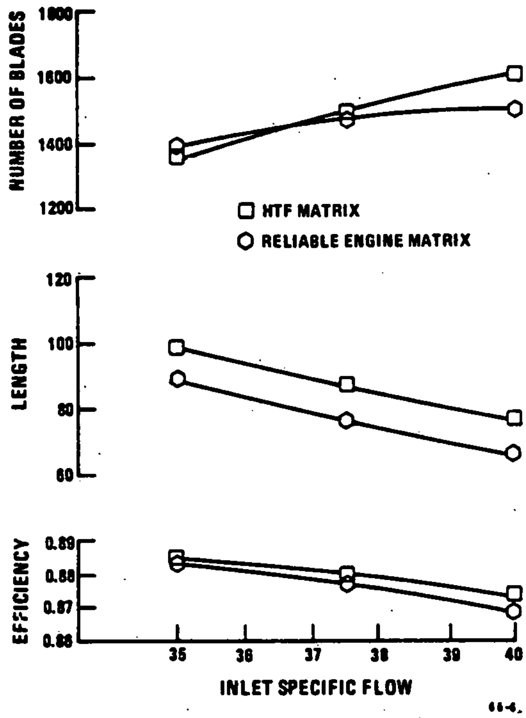


Figure 1-5. Effects of Specific Flow

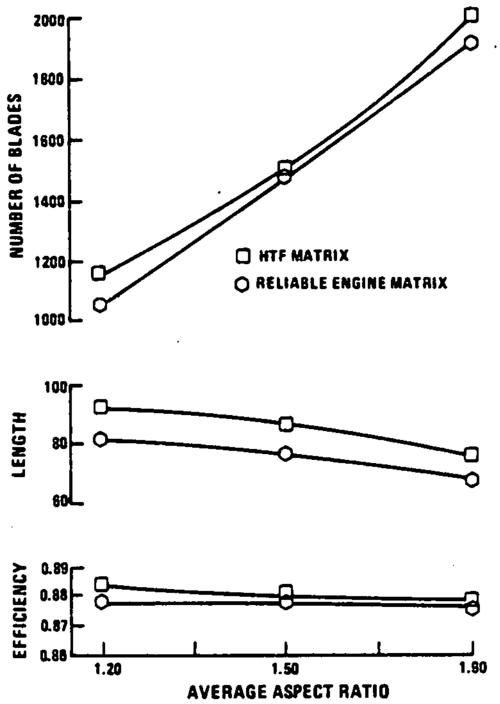


Figure 1-6. Effects of Average Aspect Ratio

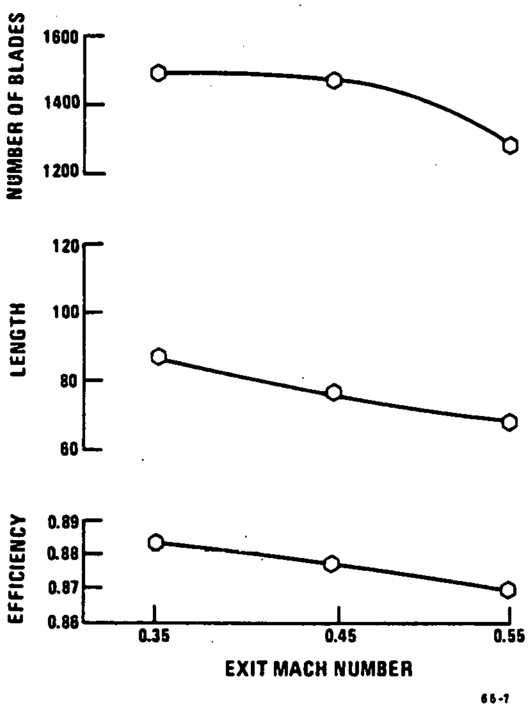


Figure 1-7. Effects of Exit Mach. No.

DOE Compressor Candidate Selection

This DOE compressor optimization program was planned to capitalize on the EPRI compressor parametric studies by isolating the most promising candidates from the matrix and permitting selection of one final HTF configuration for further refinement and integration in the Reliable Engine defined under EPRI program RP1187-1.

The physical (engine dictated) constraints the Reliable Engine placed upon the compressor preliminary design were the determining factors in the selection of the final compressor configuration. The first of these constraints was the required minimum hub diameter of 40 inches. This would enable the front bearing to fit under the inlet guide vane-first stage rotor, thus shortening the overall rotor length permitting use of a 2-bearing system.

The second constraint was a maximum critical speed parameter of 130. This parameter is defined as compressor length squared divided by exit stage root diameter. When the value of this parameter exceeds 130, practical two bearing rotor systems become very difficult to achieve due to lack of rotor stiffness. With the very large flow size of this compressor, this parameter eliminates all of the Constant-Inner-Diameter (CID) and most of the Constant-Mean-Diameter (CMD) configurations because of their extreme length and small exit stage diameter.

The third major constraint was that of exit stage hub-to-tip diameter ratio (H/T). Because of the high pressure ratio of this compressor, the exit H/T diameter ratios become very high, especially for a constant-outer-diameter (COD) configuration. UTC's test experience limit for exit H/T is 0.936, so this was selected as the limiting value. This constraint eliminated the COD compressors and some of the CMD configurations from further consideration as shown in Figure 1-8.

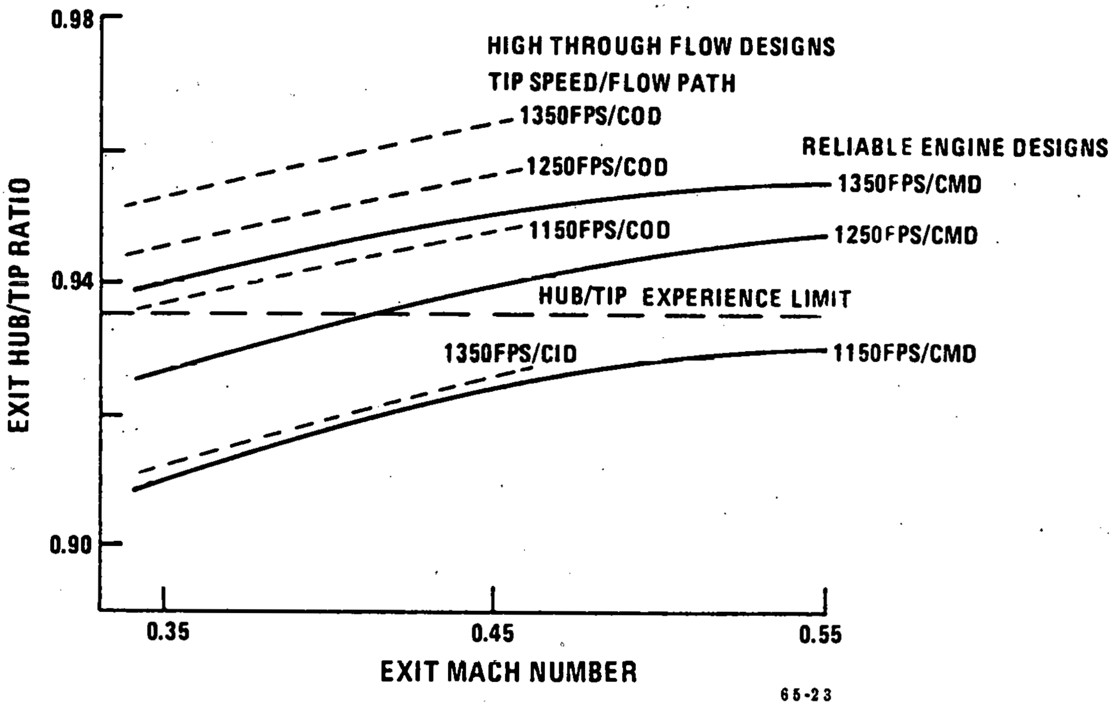


Figure 1-8. Exit Hub-Tip Trends

Utilizing the results of the Baseline Reliable Engine and the HTF matrices, and attempting to satisfy the three mechanical constraints mentioned above, three HTF configurations were selected for preliminary analysis. Because of the definite trends toward improved compressor performance with low aspect ratio, low specific flow and low inlet tip speed, these parameters were minimized for each candidate configuration. The remaining primary design parameters were exit Mach number and flowpath shape.

The candidate configurations were chosen to pinpoint the flowpath shape and exit Mach number for the final selection. Characteristics of these three configurations are shown in Table 1-1.

Table 1-1. HTF Compressor Candidate Characteristics

Design Paramters	CASE NO. 1	2	3
Average Aspect Ratio	1.2	1.2	1.2
Inlet Specific Flow lb/sec/ft^2	35.	35.	35.
Exit Mach No.	0.40	0.40	0.40
Inlet Tip Speed ~ ft/sec	1150	1200	1150
Flowpath	COD	CMD	CMD/COD
No. of Stages	9	10	9

The results of the HTF compressor parametric study also showed increased COE was due, to a large degree, to very high estimated diffuser losses. For this reason, the HTF preliminary design study included a diffuser preliminary design task to more accurately define actual diffuser losses. The diffuser study is discussed in detail in Section 3.

Section 2
COMPRESSOR DESIGN OPTIMIZATION

Configuration Selection

The three candidate compressor configurations selected using the parametric matrix results underwent further preliminary mean-line (one-dimensional) optimization in which the flowpath, average aspect ratio and average solidity of each configuration were optimized to maximize efficiency. The responsiveness of each candidate to the physical constraints (H/T, critical speed, 40-inch min. I.D.) was also examined.

Results of the preliminary optimization of the three candidate configurations are shown in Table 2-1. The Reliable Engine selected configuration is also shown, with adjustments made to accomodate the 40-inch minimum hub diameter requirement.

TABLE 2-1. OPTIMIZED DESIGN VALUES OF HTF COMPRESSOR CANDIDATES COMPARED WITH BASELINE RELIABLE ENGINE COMPRESSOR

CASE	EPRI RELIABLE ENGINE	HTF CASE NO. 1	HTF CASE NO. 2	HTF CASE NO. 3
Inlet Specific Flow ~lb/sec/ft^2	35.	35.	35.	35.
Exit Mach No.	.34	.42	.44	.40
Avg. Aspect Ratio	1.20	1.20	1.27	1.24
Inlet Tip Speed ~ ft/sec	1210	1210	1200	1210
No. of Stages	9	9	10	9
Adiabatic Efficiency	.888	.888	.885	.884
No. of Blades & Vanes	1100	1330	1250	1320
Length ~ Inches	93	92	107	88
L^2/D	126	121	195	119
Exit H/T	.936	.948	.931	.938
Clearance/Height	.011	.012	.011	.012
Avg. Solidity	1.269	1.269	1.274	1.277
Surge Margin	15%	20%	20%	20%
Avg. D-Factor	.464	.420	.418	.420
Delta COE ~ Mills/kWh	Base	+.68	+.51	+.38

The inlet geometries of Configurations No. 1 and No. 3 were altered to attain a minimum inlet diameter of 40 inches. As a result, inlet tip speed increased for both cases, to 1210 feet per second. Case No. 2 was not changed, because its ID was close to 40 inches and it was already likely to be much too long for a two-bearing configuration.

The Case No. 1 design was very similar to the base configuration proposed for the Reliable Engine but with a higher exit Mach number. The increased Mach number was attained by reducing the compressor exit annulus area. This, in turn, increased the exit H/T (diameter ratio) substantially over the acceptable limit of 0.936. This eliminated Case No. 1.

It should be pointed out that this limit was selected as a result of test experience which associated high losses with the high exit hub/tip ratios. However, the mean-line design system accounts for this effect in predicting performance and this restriction could be removed with the acquisition of supportive test experience.

Case No. 2 was selected as a candidate to determine the feasibility of a Constant-Mean-Diameter (CMD) design. Its length was found to be unacceptable for a simple 2-bearing design. Case No. 2 also required one more stage than Cases No. 1 and 3. This eliminated Case No. 2.

Candidate No. 3 was selected as a configuration that would meet all of the physical constraints. The flowpath average aspect ratio and exit Mach number were iterated to result in a configuration that had acceptable critical speed and H/T parameters. The design parameters and performance predictions for this configuration are shown in Table 2-2.

As the preliminary design of Case No. 3 proceeded a design review was made to verify the appropriateness of the loading levels; tip clearance and stage spacing design criteria being used for case No. 3 which was the only one of the three candidates meeting all physical and aerodynamic design requirements.

TABLE 2-2. HTF CONFIGURATION DESIGN VALUES - CASE NO. 3

Corrected Speed - RPM	3566.
Actual Speed - RPM	3600.
Pressure Ratio	14:1
Surge Margin - %	20.
Efficiency (adiabatic) %	88.35
Number of Stages	9
Variable Vane	IGV
Bleed	2 (S4, S6)
First Rotor Corrected Tip Speed - FPS	1210.
First Rotor Hub-Tip Ratio	0.515
First Rotor Inlet Specific Flow - $lbm/sec/ft^2$	35.0
Exit Hub-Tip Ratio	0.936
Exit Mach Number with Blockage	0.40
Rotor Average Aspect Ratio	1.23
Stator Average Aspect Ratio	1.29
Compressor Average Aspect Ratio	1.26
Rotor Average Solidity	1.29
Stator Average Solidity	1.24
Compressor Average Solidity	1.27
Rotor Average D-Factor	0.419
Stator Average D-Factor	0.429
Compressor Average D-Factor	0.424
Rotor Average $\Delta P/(P_t-P)$	0.404
Stator Average $\Delta P/(P_t-P)$	0.336
Compressor Average $\Delta P/P_t-P)$	0.370
Total Airfoils (Excluding IGV)	1304.

Loading Levels

A basic premise of the Reliable Engine and HTF programs was that the aerodynamic design was to be based on existing, proven designs. The intent of the program was to utilize the latest technology rather than to advance the art further through the demonstration of new levels of aerodynamic loading.

The original Reliable Engine Conceptual studies based on a high stage loading design. As shown in Figure 2-1, the baseline configuration loading levels exceeded previous design and test experience. The loading levels of the Reliable Engine were similar to those of the Department of Energy (DOE) and Energy Research and Development Agency (ERDA) axial-centrifugal compressor which has not yet been tested. The loading levels of the Reliable Engine parametric studies were, therefore, lowered a small amount to be comparable to present day compressors.

The loading levels were lowered even further in the HTF parametric studies as a result of the higher velocities (Mach numbers) through the compressor. These levels could have been raised by reducing the number of stages, but the step reduction from 9 to 8 stages would have raised the loadings above our experience level. Similarly, blade loadings could have been increased by reducing the number of blades. This was investigated, and the result would have been an efficiency decrease. Even though the main thrust of the program was directed toward high reliability, the dramatic impact of compressor efficiency on COE could not be ignored. As a result, the numbers of stages and blades were not altered and the lower loadings were retained as a benefit of using HTF concept.

Figure 2-1 shows that the HTF design does represent a substantial increase in loading and a reduction in number of stages from previous UTC industrial compressors. However, the resultant HTF loadings are at a level which is substantiated by a number of aircraft compressors, and therefore represents acceptable risk.

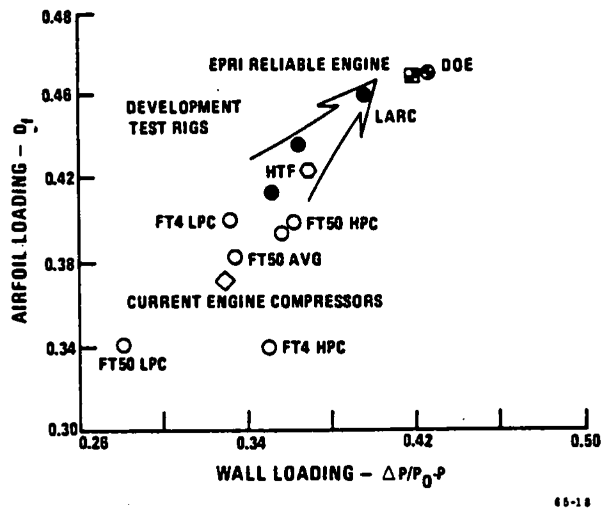


Figure 2-1. Compressor Loading Experience

Clearance

The blade tip clearance values for each of the parametric configurations ranged from 0.100 inches in the front end of the compressor to 0.040 inches in the rear. This resulted in an average clearance-to-blade height ratio of approximately 1 percent. An analysis was made of the effects of average clearance-to-height ratio on compressor efficiency using HTF configuration No. 3. The results presented in Figure 2-2 show that this is near optimum. The clearances selected were based on existing compressor experience. These values would be refined further in a detail design phase.

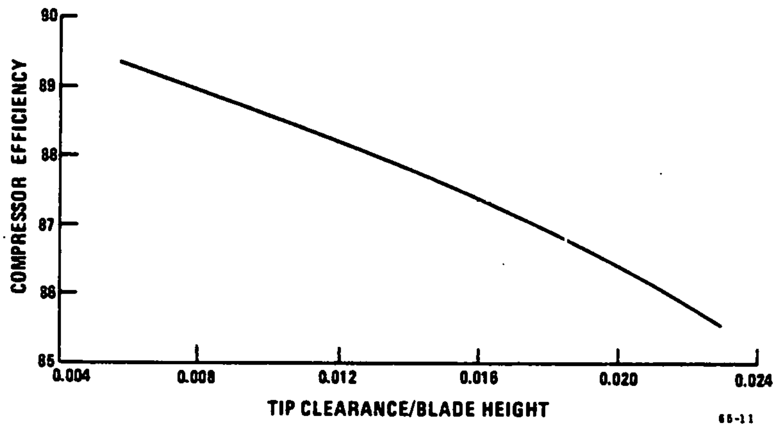


Figure 2-2. Effect of Blade Tip Clearance on Compressor Efficiency

Axial Spacing

Axial spacing between the compressor blade rows initially had been set at 1.20 inches at the mean line. Revised estimates distributed axial gaps from 1.20 inches at the root of the front stage to 0.8 inches at the root of the rear stages. The resulting compressor length remained virtually the same as the original length.

The preliminary Cost-of-Electricity calculated for the three HTF candidate configurations included relatively high diffuser losses. Later calculations of COE for the selected configuration will reflect the diffuser preliminary design work of this program and the subsequent improved performance.

Mean-Line Optimization

The selected compressor configuration underwent continued mean-line optimization of air angle and loading distributions. In conjunction with preliminary off-design analysis, this ensured that the compressor does not have any stall prone stages. Analysis of hot-day operation indicates that the HTF design integrated with Reliable Engine components will have adequate stall margin on a hot (100°F Amb.) day without the use of engine bleeds.

Evaluation of the effects of several preliminary design parameters on the selected design was completed as part of the optimization and is discussed below.

Surface Roughness

An estimate of the effect of blade surface finish on compressor performance is shown in Figure 2-3. Taking into account the extremely long time between overhaul required of the compressor blades, and the likelihood of surface finish degradation with time, the transient improvement in performance obtained with very smooth surfaces is not significant enough to warrant the extra polishing required. (The estimated roughness used for this design analysis is 300 micro-inches - which is an equivalent sand grain surface roughness.)

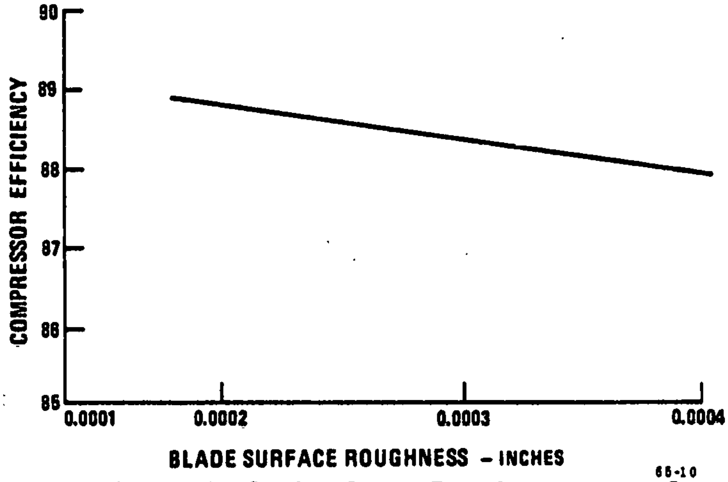


Figure 2-3. Effect of Airfoil Surface Roughness on Compressor Efficiency

Aspect Ratio

The effects of average blade aspect ratio on compressor efficiency, number of airfoils and compressor length are shown on Figure 2-4. The selected aspect ratio i.e., 1.26, represents the best compromise between compressor length for critical speed and number of airfoils for reliability.

Gap-to-Chord Ratio

Figure 2-5 shows the effects of airfoil gap-to-chord ratio (the inverse of solidity) on efficiency and number of airfoils. The selected value of .789 was determined to result in a reasonable efficiency without an excessive number of airfoils.

Blade Thickness

The blade average thickness-to-chord ratio first selected for the design was 7 percent. Based on design experience, a value of 7 percent was judged to be adequate. For optimization purposes a study was made of the effect of varying the blade mean thickness-to-chord in the rear stages (4 to 9) of the compressor where the short chords lead to physically thin airfoils. The results, shown in Figure 2-6, indicate that compressor efficiency varies only slightly with blade thickness. The original selection of 7% thickness was confirmed.

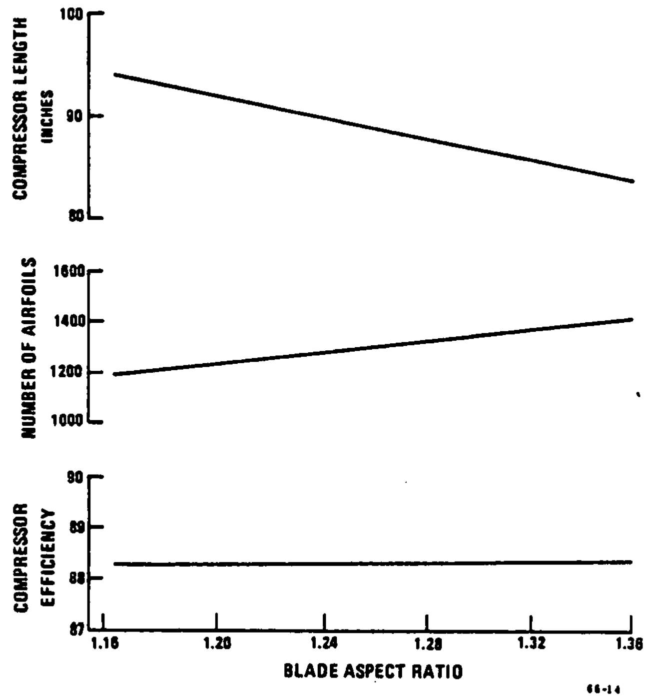


Figure 2-4. Effect of Aspect Ratio on Efficiency, Number of Airfoils, and Length

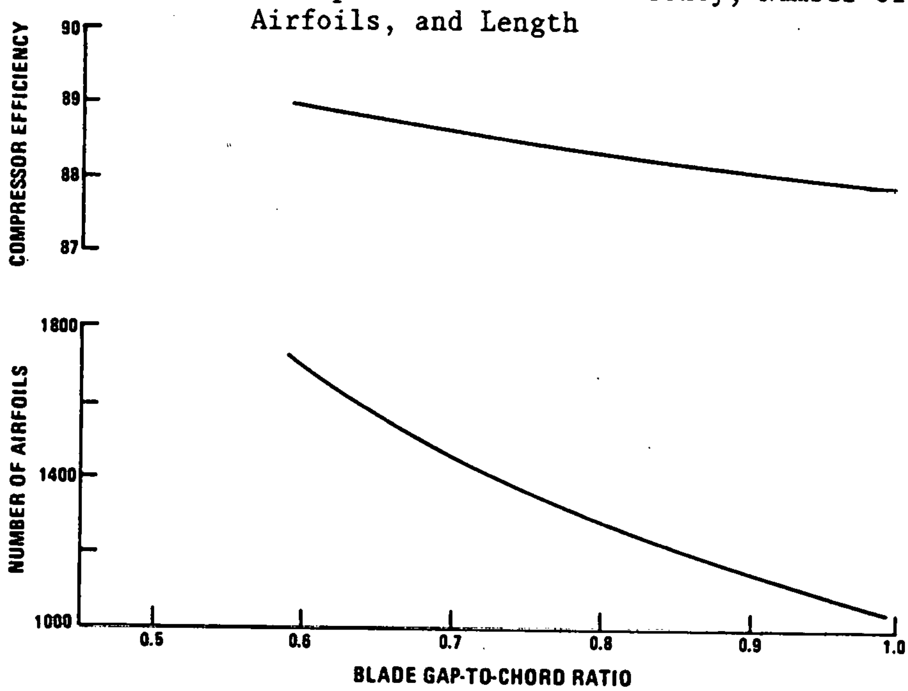


Figure 2-5. Effect of Gap to Chord Ratio on Efficiency and Number of Airfoils

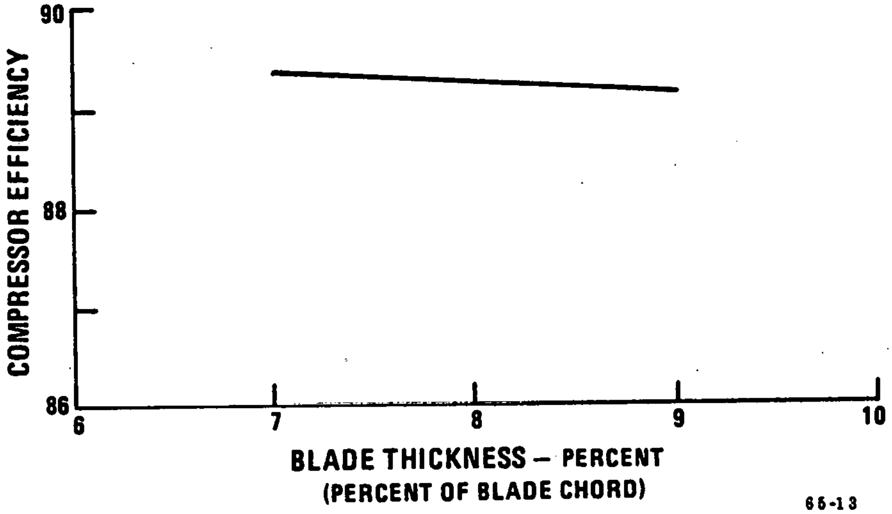


Figure 2-6. Effects of Blade Thickness on Efficiency

Streamline Analysis

Although one-dimensional analyses are useful for determining the gross effects of parameters such as flowpath, blade geometry, wheel speed, and work distribution; a more detailed analysis was needed for determining the radial distribution of flow conditions across the annulus. This was accomplished by using the Streamline Analysis Computer Program, which performs an axisymmetric, two-dimensional, inviscid analysis. Streamline locations were determined through the simultaneous solution of the radial equilibrium, angular momentum and continuity equations at specified calculation stations. Blade row performance was calculated through the specification of loss coefficient, pressure ratio and swirl angle at the appropriate stations.

The streamline analysis was a preliminary analysis characterized by specifying calculation stations at the blade entrances and exits only. Blade-row exit pressure and loss radial distributions were specified according to current design practice. Figure 2-7 illustrates typical radial profiles. The flow field information calculated by the streamline analysis was most valuable in determining flowpath changes. Flowpath changes were made to avoid local accelerations and decelerations along the wall and to obtain a smooth transition from entrance to exit. Figure 2-8 illustates the meridional Mach number distribution along the inner and outer walls. Figure 2-9 is a graphic representation of the streamline analysis output. Although beyond the scope of the study, streamline analyses provide a means by which a designer can determine and control loading levels; and as well as entrance and exit conditions such as Mach number and swirl angle. A detailed streamline analysis provides radius change and streamtube convergence information necessary for the complete definition of the blading.

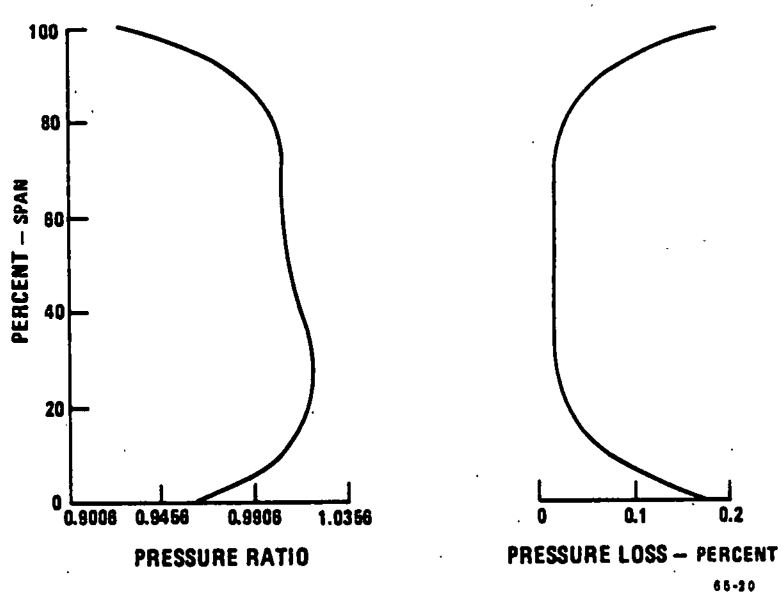


Figure 2-7. Representative Stage Exit Profiles from Streamline Analysis

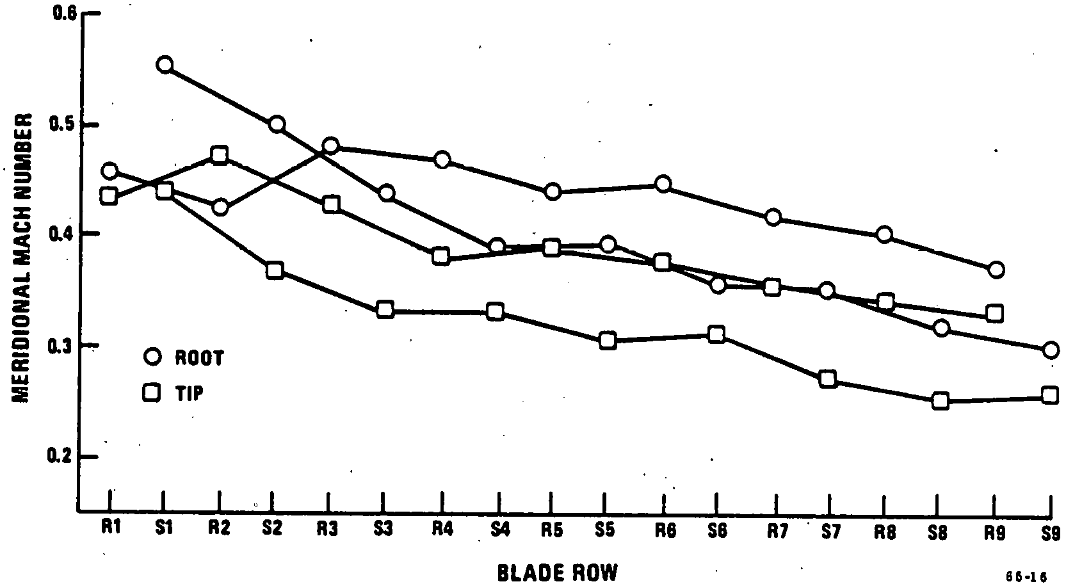


Figure 2-8. Meridianal Mach Number Distribution

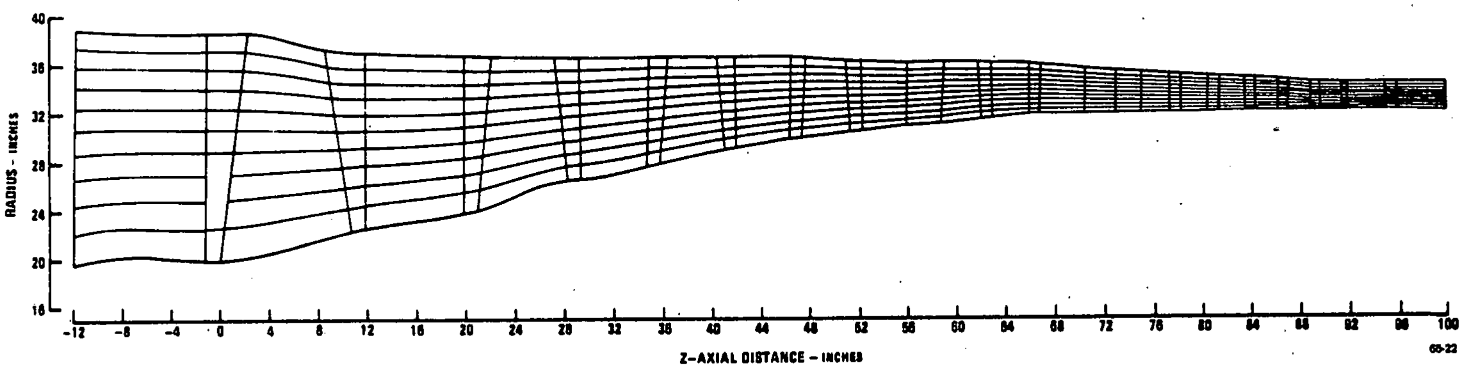


Figure 2-9. Graphic Representation of Streamline Analysis

Section 3
DIFFUSER/COMBUSTOR DESIGN

A comprehensive evaluation of the high through-flow compressor required definition of a diffuser/combustor configuration optimized for the High Through-Flow (HTF) design. The goal was not to exceed the original diffuser pressure loss of the High Reliability Gas Turbine (HRGT) with the higher diffuser inlet Mach numbers associated with the HTF design. A further goal was to accomplish the required diffusion without increasing the distance required between the rotor main bearings. This latter goal was imposed to prevent loss of rotor critical speed margin. Specific requirments were to define:

1) Diffuser/combustor configuration
2) Combustor cooling flow requirements
3) Combustor wall temperature distribution
4) Air flowpath
5) Combustor pattern factor
6) Combustor radial temperature profile
7) Diffuser pressure loss

Two combustor configurations were considered:

1) Rich Burn/Quick Quench (On Board)
2) External (SILO) lean burn

The five types of diffusers investigated were:

1) Straight Wall Equiangular
2) Bled Diffuser
3) Straight Wall Radial
4) Straight Wall on Angled Centerline
5) Curved

The diffuser types are shown schematically in Figure 3-1.

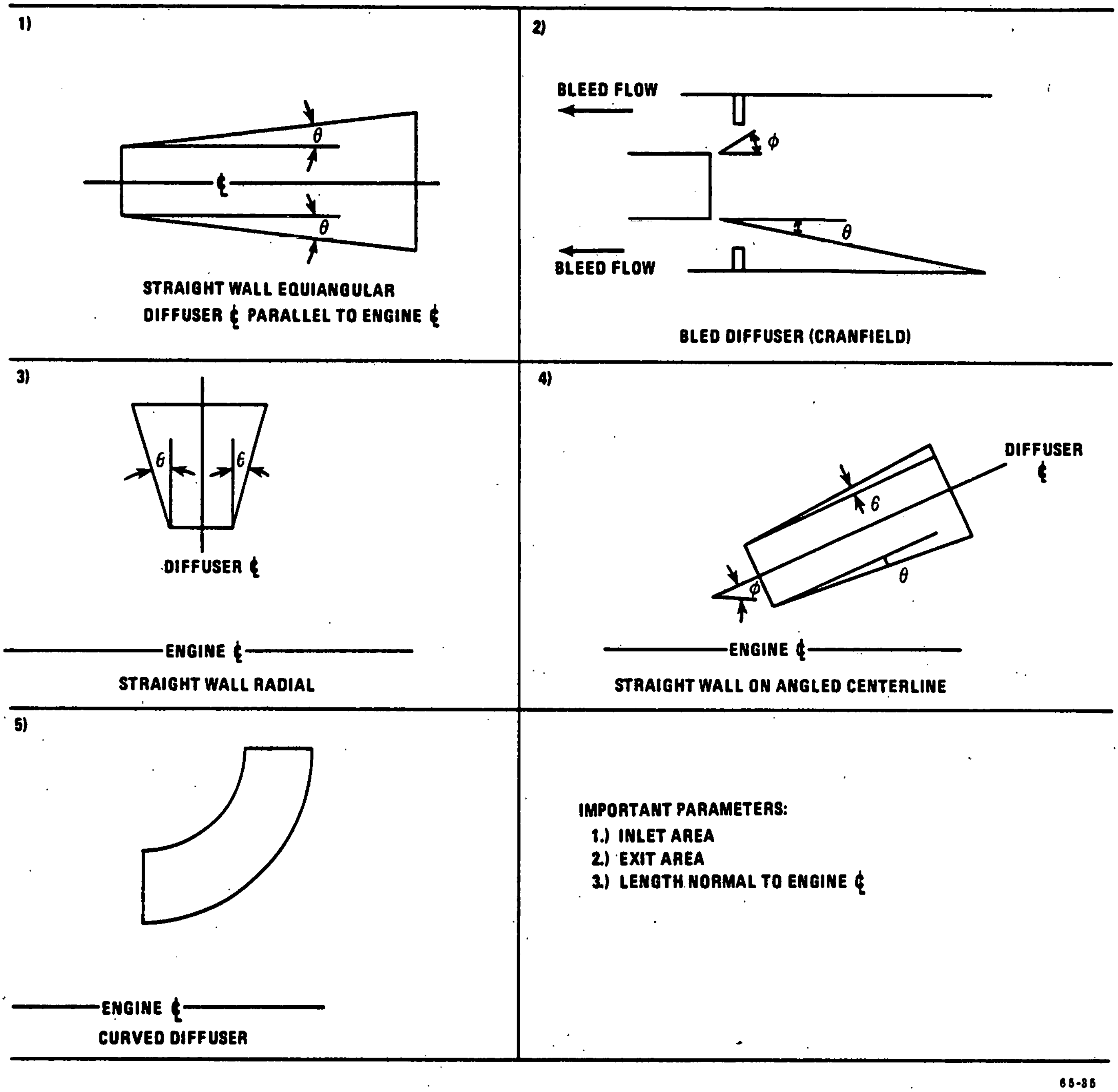


Figure 3-1. Five Types of Diffusers Studied

Diffuser/Combustor Selection

The straight wall equiangular and bled diffusers were more compatible with the rich burn/quick quench combustor (RBQQ), while the straight-wall angled and radial, and the curved diffusers were candidates for external silo-type combustors. The bled diffuser concept also could be applied to diffusers for silo combustors. The bled diffuser showed promising results from the preliminary analysis but it has had limited application to date. Further developmental work would be required before it could become the recommended configuration. The silo burner was considered an alternative design, with the on-board combustor configuration considered the primary candidate. This is the more widely used concept in industrial gas turbine engines. Most of the effort was therefore directed toward the on-board combustors with the more conventional straight wall equiangular diffuser.

The final configuration selected was an 8° equivalent conical angle straight wall diffuser shown in Figure 3-2. The final selection was based on a comparison of an 8° and a 9° diffuser of equal length which, as indicated in Figure 3-3, showed the risk of diffuser stall to be higher in the 9° design.

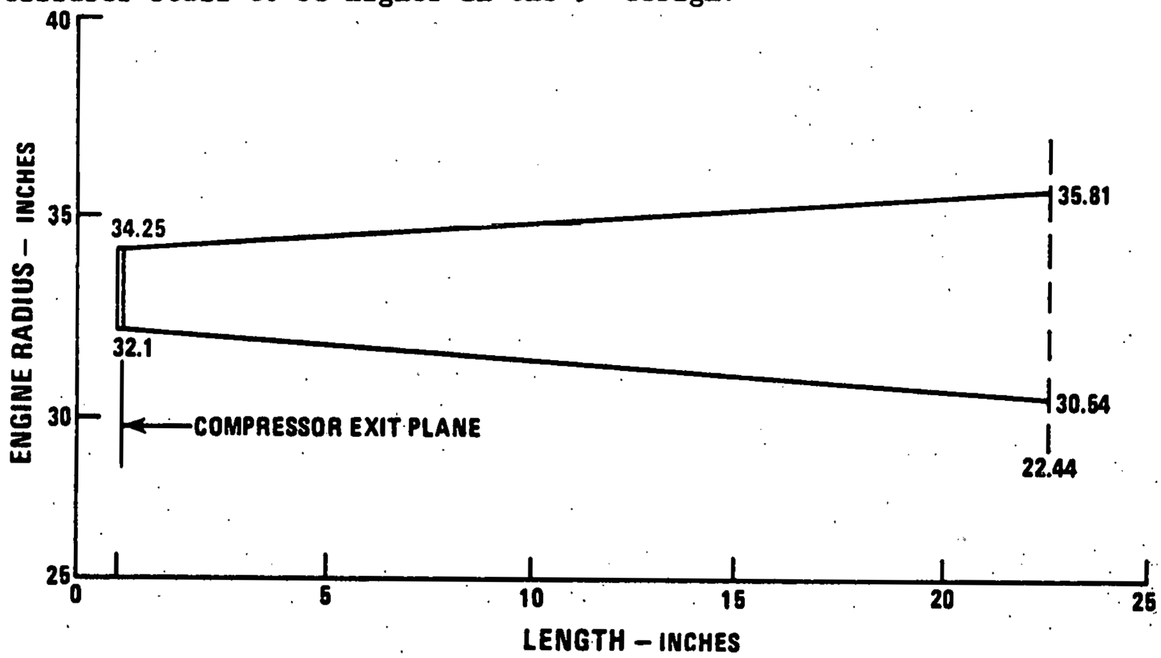


Figure 3-2. Equiangular 8-degree Diffuser

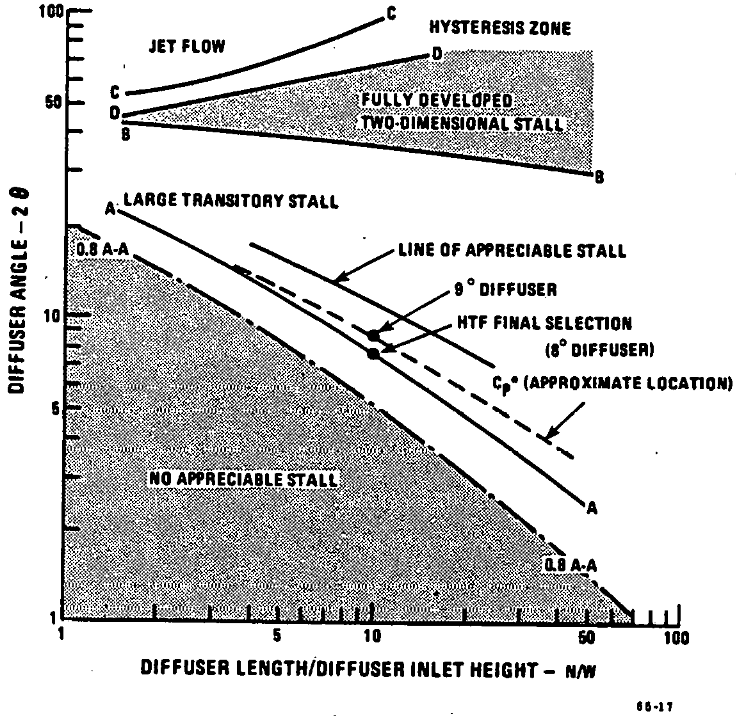


Figure 3-3. Diffuser Operating Regimes

Combustor Cooling Air Requirements; Air Flowpath

The goal pressure loss was met with the 8° diffuser and the RB/QQ combustor. The combustor cooling airflow is the same for the HTF design as for the Reliable Engine low aspect ratio compressor design. The silo combustor was eliminated from further consideration.

There is no major air flowpath change required in the new design to adapt it to the Reliable Engine. The only change is in the length of the diffuser. The combustor cooling air and the combustion air flowpaths are presented in Figure 3-4.

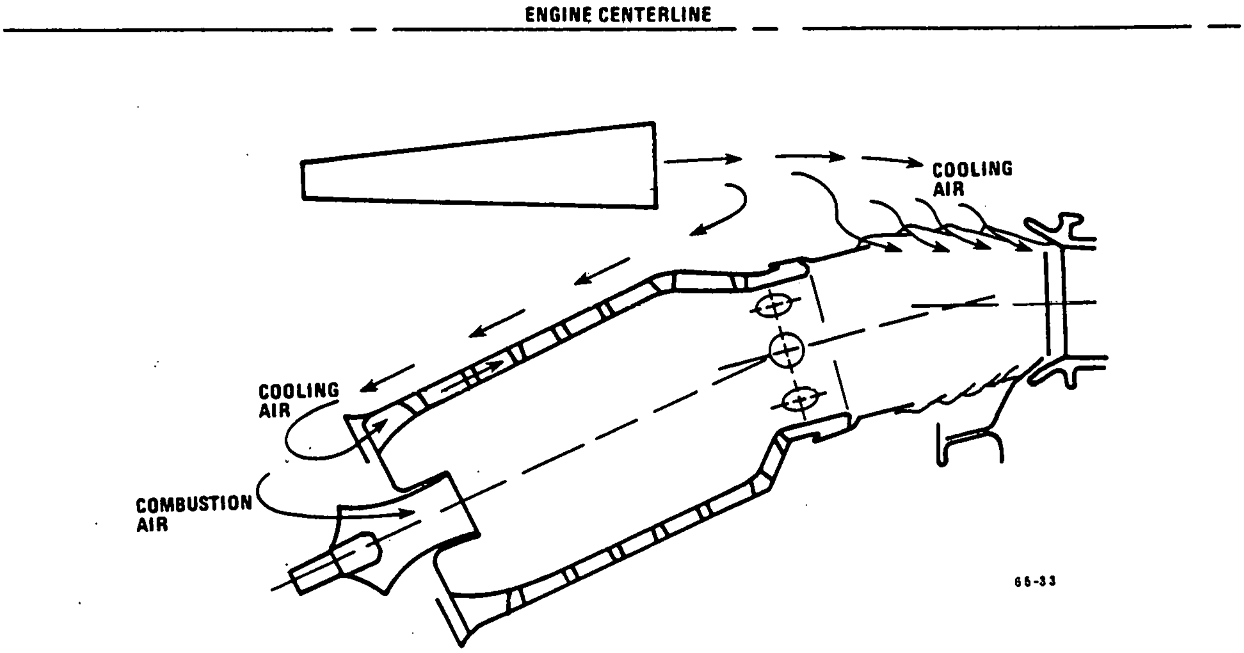


Figure 3-4. Schematic of Transition of Airflow from Diffuser to Combustor

The combustor wall temperature distribution will not be affected by the compressor diffuser selection and is unchanged from the EPRI Reliable Engine program. Figures 3-5 and 3-6 show the temperature distribution for the combustor I.D. and O.D., portions of the lean-burn transition liner. The louver design of the liner is overlayed on the figure to relate the temperature distribution to the air flowpath. The temperature distribution of the rich-burn section is presented in Figure 3-7. Finwall cooling is required to isolate the cooling flows from the combustion process.

The preliminary anaysis indicated that the transition section can be adequately cooled by the available air. The rich-burn portion of the Rich-Burn/Quick-Quench combustor cannot be cooled sufficiently by the compressor discharge air to meet the Reliable Engine life requirements. Other cooling approaches should be investigated. Three of the recommended approaches are:

1) Steam (gives a greater heat transfer)
2) Composite Walls (possibly ceramic on metal)
3) Change of Material (higher temperature alloy)

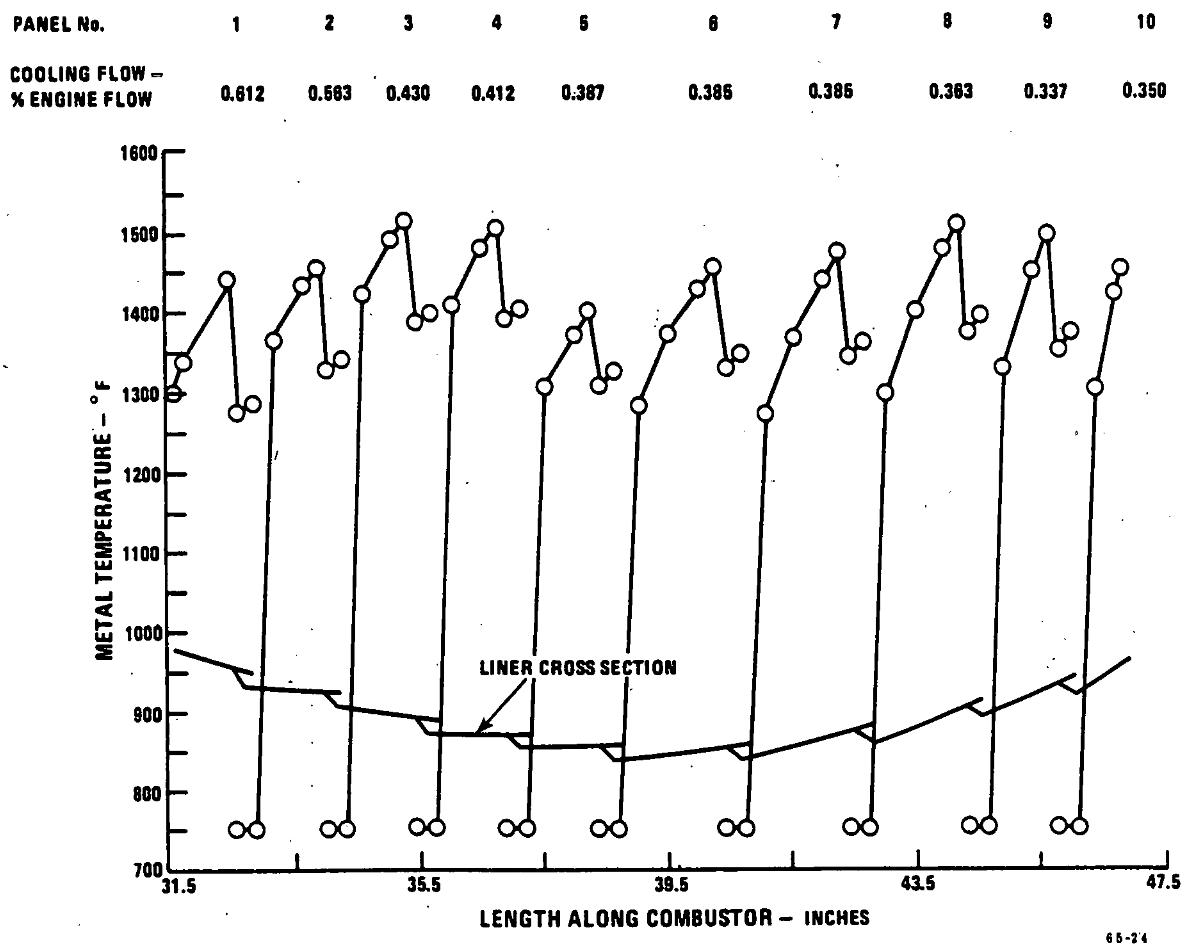


Figure 3-5. I.D. Metal Temperatures of Annular Combustor Section Inner Wall

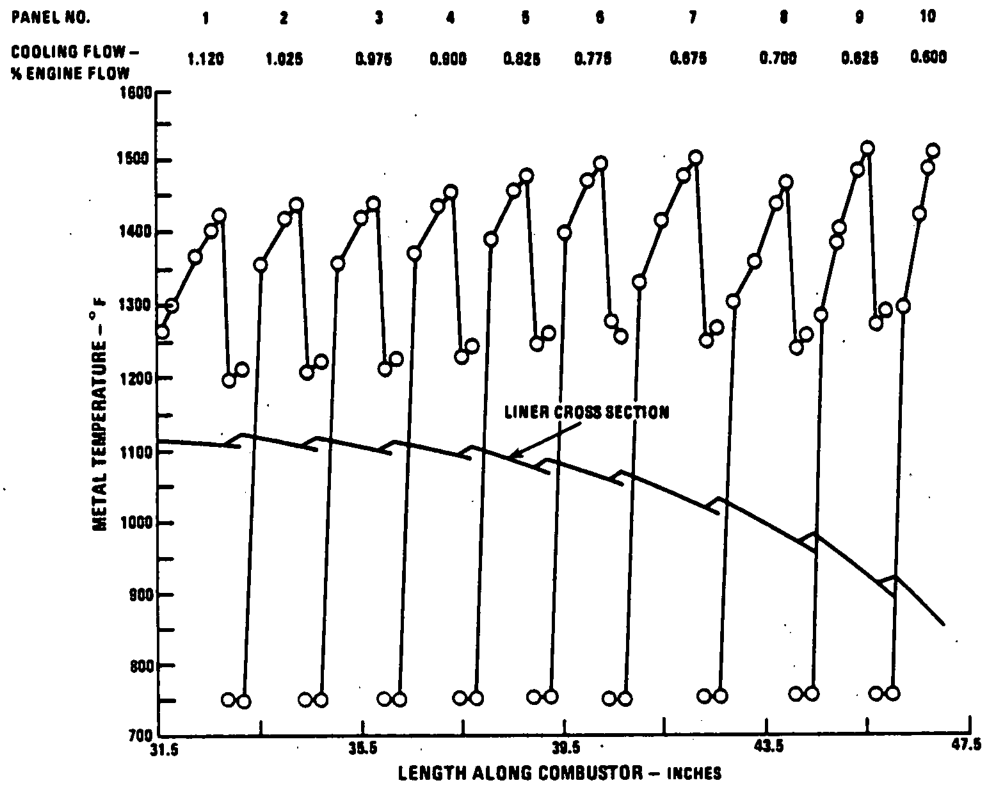


Figure 3-6. O.D. Metal Temperatures of Annular Combustor Section Outer Wall

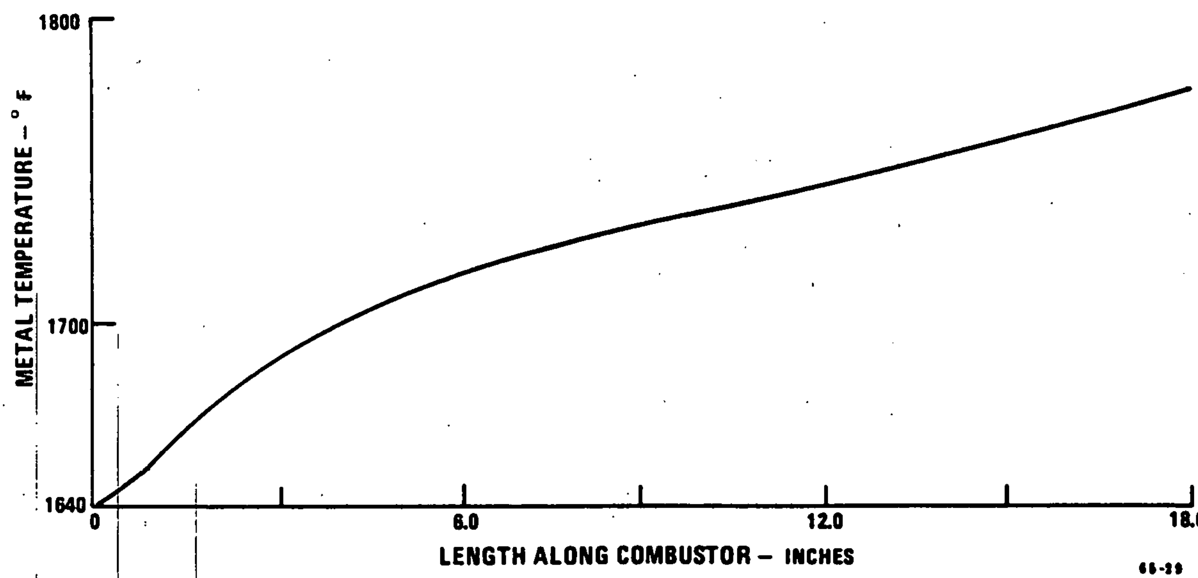


Figure 3-7. Metal Temperatures for Rich-Burn Section of RB/QQ Combustor

Pattern Factor and Radial Profile

Two other UTC engines have can-annular systems closely approximating the proposed system. The engines have pattern factors in the range of 0.3 to 0.35. At present, there is no analytical system for calculating pattern factor for th RB/QQ configuration. A conservative estimate is that the pattern factor will be in the range of 0.3 to 0.35.

UTC has a radial profile prediction system for the RB/QQ combustor design. The calculated radial profile is 1.055 at the peak and 0.721 at the wall. A 2% factor of safety is allowed for the peak value. The goal is to have the profile peak at 65% span.

Figure 3-8 shows the estimated combustor exit radial temperature profile. The profile is typical of industrial gas turbines and should present no special problem to the turbine.

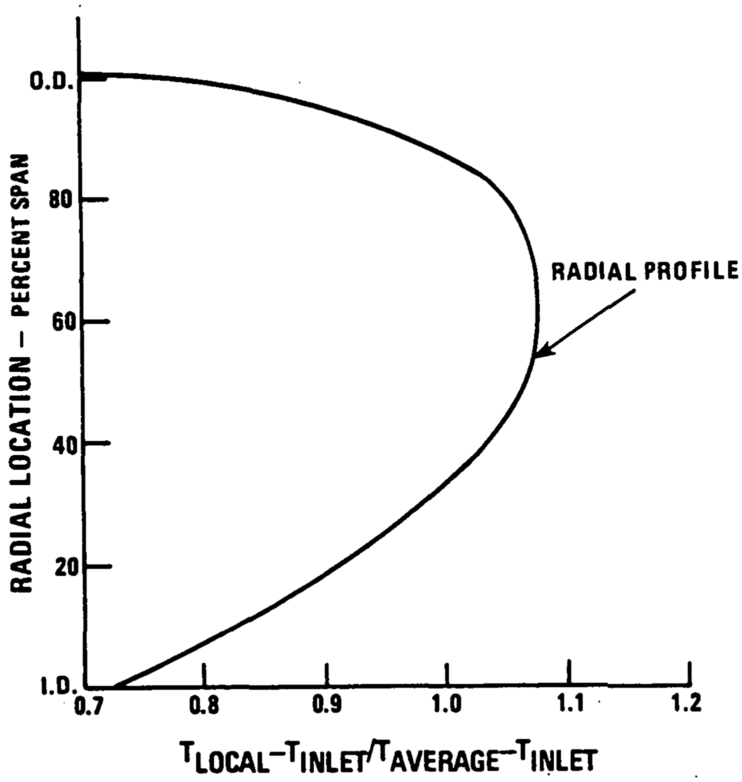


Figure 3-8. RB/QQ Combustor Exit Radial Profile

Pressure Loss Summary

The total pressure loss of the Reliable Engine diffuser/combustor system is 6.2%, broken down as follows:

Diffuser Loss	=	2.7%
Combustor Front End Loss	=	1.75%
Combustor Liner Loss	=	1.75%
		6.2%

The total pressure loss of the HTF diffuser/combustor system is essentially the same as for the reliable engine system and is developed as follows:

Diffuser Loss	=	2.9%
Combustor Front End Loss	=	1.75%
Combustor Liner Loss	=	1.75%
		6.4%

The 0.2 difference in pressure loss is within the accuracy of the prediction system. Therefore, we conclude that there is no change in the system pressure losses due to use of the HTF compressor concept.

The HTF diffuser can be incorporated in the Reliable Engine design without increasing the distance between the rotor main bearings. The rotor dynamic response will therefore not be affected.

Section 4
MECHANICAL CONFIGURATION STUDIES

There are no significant mechanical design differences between the high through-flow compressor and the compressor defined for the High Reliability Gas Turbine under Electric Power Research Institute (EPRI) Contract RP 1187-1.

Blade Attachment Sizing

Materials - Attachment size is influenced by the material properties of both blade and disk. The blade material, AMS 5616 (common designations: Greek Ascoloy, Uniloy 1415 NW) was chosen because of its reasonable cost and favorable experience in numerous existing gas turbine airfoil applications. The disk material, PWA-IM-3005 (common designation: Cameron Z448, was chosen for similar reasons. It has good strength and stability up to 700°F. It is hardenable in sections at least as thick as 12 inches based on industry experiences, which is sufficient for this application.

Stresses - Attachment geometry and stress margins of safety are determined from successful past experience. Bearing surface contact angles are 45°. Fillet radii are proportional to dovetail size to enhance low-cycle fatigue life.

Attachments for this preliminary design were sized by a computer synthesis program that iterates the dovetail design until the margin of safety of the stresses at selected locations is adequate. Locations and types of stresses analyzed are, Neck tensile stress, Tooth-bending stress, Tooth-bearing stress, Tooth-Shear stress, Combined stress.

The margins of safety used were developed from past aircraft and industrial engine experience. As noted below, blade and blade attachment size have little effect on disk size in the latter stages of the compressor. In these stages, consideration could be given in the final design to oversizing these attachments for increased reliability.

Disk Sizing

Two disks, the third stage and the eighth stage, were sized using a UTC disk synthesis computer program. Disk shapes for the remaining stages were interpolated from the definition these two disks. For this preliminary design, the following worst overspeed, worst temperature case assumptions were made:

Twenty-five percent overspeed at turbine startup with disk OD at 100% gas path temperature, disk ID at room temperature.

Using the assumed conditions, the disks were proportioned so that local stresses and average tangential stresses are within the capabilities of the PWA-IM-3005 material used in the disks. A cutaway view of the entire compressor is shown in Figure 4-1. During detail design, additional checks such as fatigue life, including fracture mechanics considerations, will be made.

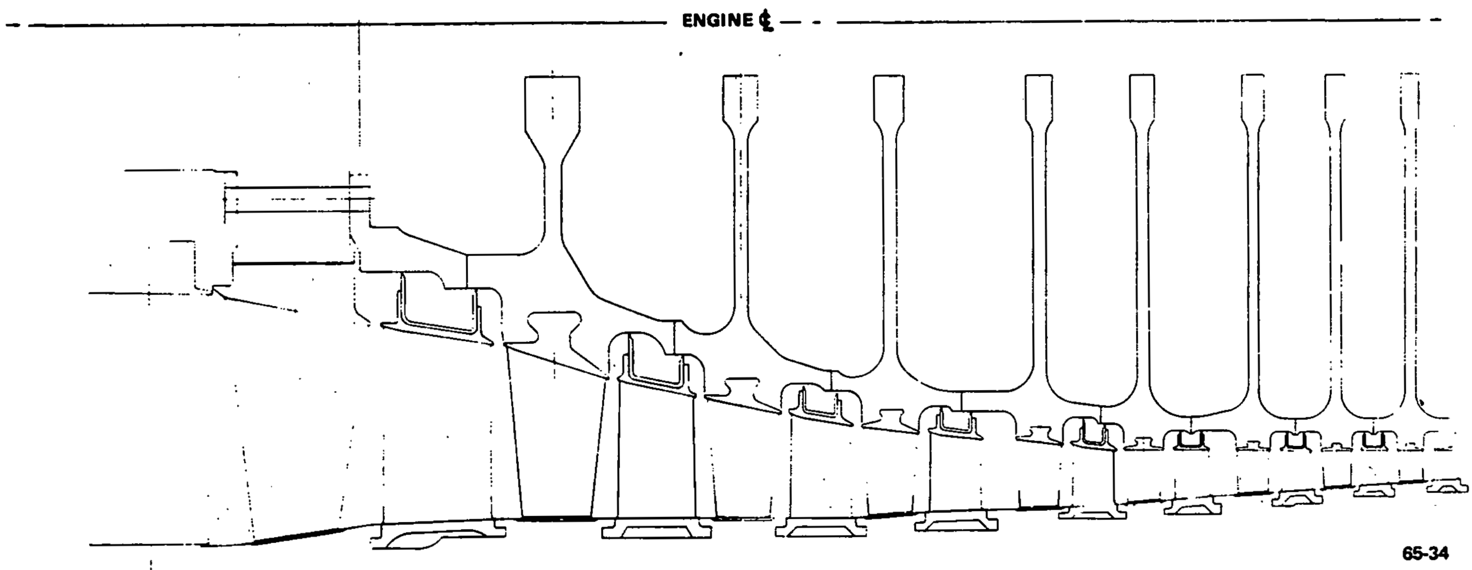


Figure 4-1. High Through-flow Compressore Cross-Section

Stresses in the aft disks were not strongly influenced by the airfoil pulls. The average tangential stress of the eighth stage disk, for example, consists of 82% disk body centrifugal pull stress and 18% blade centrifugal pull stress. The third stage by contrast has 73% body pull stress and 27% of its stress due to blade pull.

Flowpath Spacing

Final flowpath spacing considers airfoil deflections, compressor bleed port requirements etc. This preliminary design study used an estimated spacing based on prior successful compressor designs.

Compressor Integration

The EPRI High Reliability Gas Turbine Combined Cycle Program (RP1187) defined a single shaft direct drive gas turbine using a low aspect ratio compressor design as shown in Figure 4-2. The high through-flow compressor has been combined with the 8° diffuser and RB/QQ combustor and integrated with the balance of the Reliable Engine components. The resulting new design is shown in Figure 4-3. The overall length and diameter of the engine are unaffected by the HTF compressor introduction. The new diffuser exit is carried further aft before dumping in the combustor section plenum in order to obtain comparable dump pressure losses with the Reliable Engine baseline design. The increased diffuser length required some reduction in size of the turbine cooling air supply duct which underlies the combustor. Aside from this change no significant structural modifications were required. The baseline bearing-to-bearing distance is unchanged and the compressor structure is essentially unchanged so the rotor critical speeds should not be affected. The basic two bearing design is therefore still feasible.

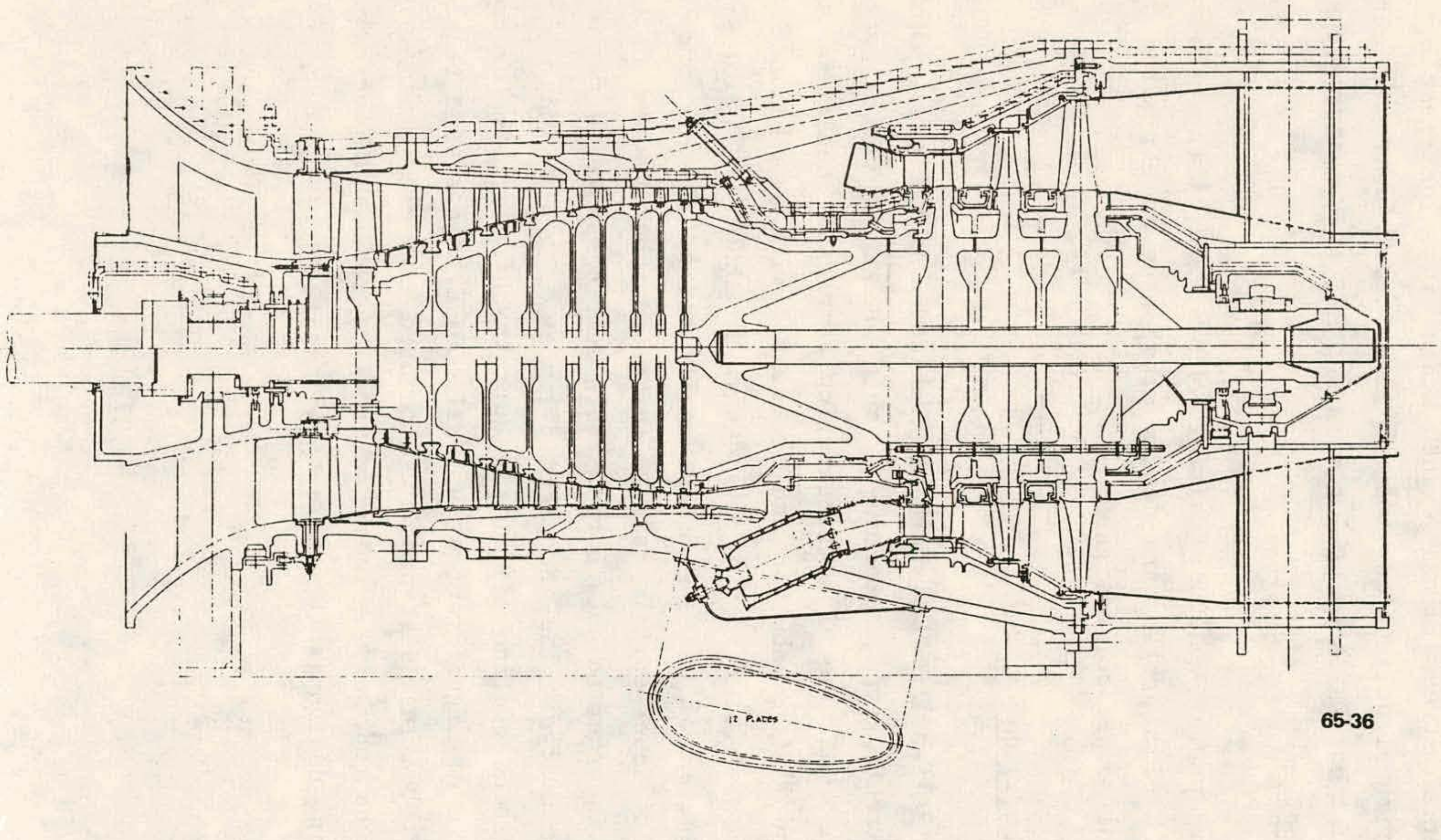


Figure 4-2. Baseline Reliable Engine from EPRI RP1187

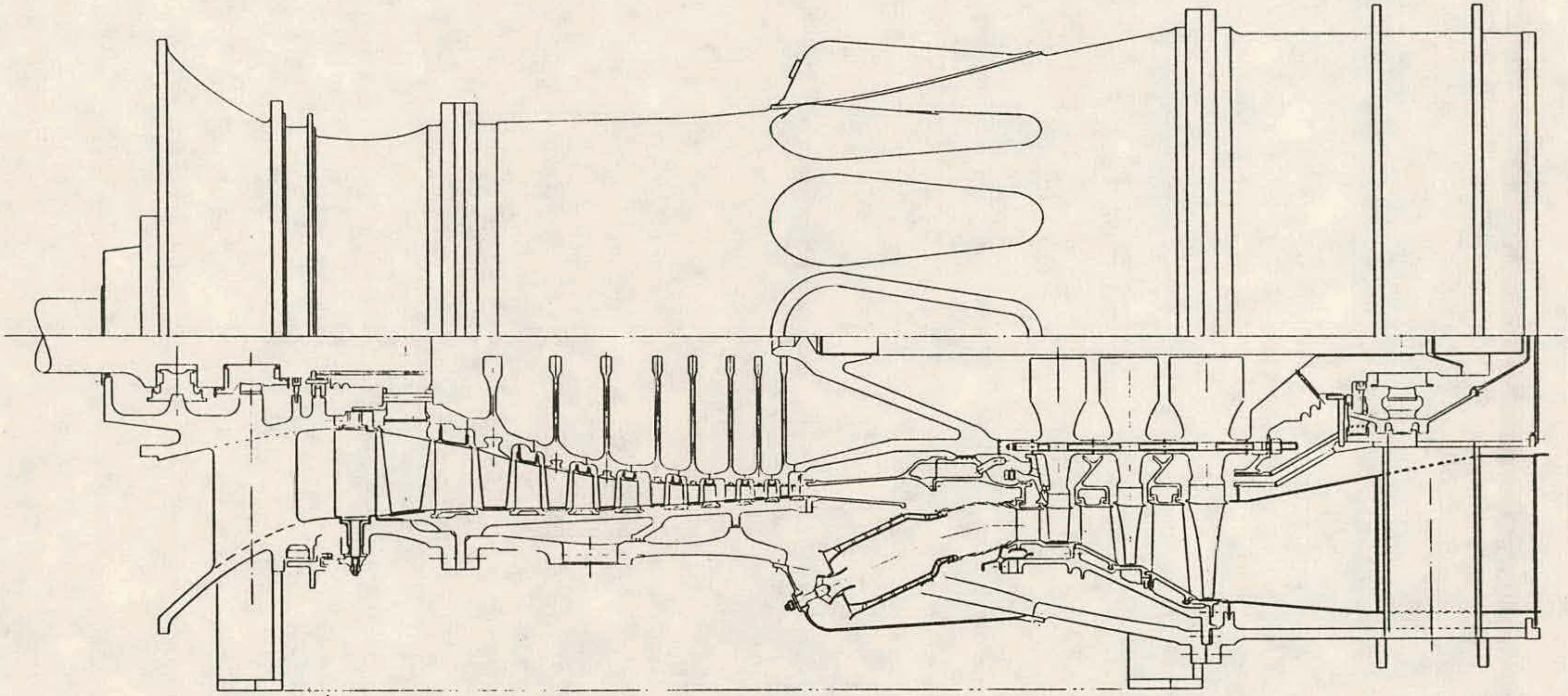


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Figure 4-3. Revised Reliable Engine with High Through-flow Compressor

Section 5
COST OF ELECTRICITY AND RELIABILITY ASSESSMENT

Reliability

The selected high through-flow (HTF) design incorporates many of the configuration characteristics which enhance reliability. These are: lower aspect ratio blading, greater surge margin, reduced numbers of stages and airfoils, lower rotor L^2/D (shaft critical speed parameter), lower inlet specific flow (equivalent to inlet Mach number) and less tendency toward constant outer diameter (COD) shape of the flowpath. CID/CMD/ COD shapes refer to the flowpath as seen in an axial cross section, where CID, CMD, and COD are constant inside, mean and outside flowpath diameters, respectively. The selected HTF configuration has a slightly greater mean time between failure (MTBF) than does the base case compressor, as shown below:

Configuration	MTBF (hr)	Equivalent failure rate per 10^6 hrs
Selected HTF Compressor	54,348	18.4
Base Case HRGT Configuration	54,054	18.5

The key factor for higher compressor reliability is the use of low aspect ratio airfoils (long chords with proportionately thicker airfoil sections). However, the Reliable Engine as a base case and selected HTF configurations both have low aspect ratio and are also similar in many other physical characteristics. The one parameter which gives the extra reliability advantage to the HTF selected configuration is the design parameter called surge margin, which for the base case compressor is 15% and for the HTF is 20%. The greater surge margin provides assurance of stall free (reliable) starting plus greater ability to avoid compressor surge during operation, a cause of airfoil overstress. Table 5-1 presents the various compressor design parameters and characterizes their reliability impact.

TABLE 5-1
EFFECT OF DESIGN PARAMETERS ON RELIABILITY

PARAMETER	IMPACT ON RELIABILITY	PARAMETER/ RELIABILITY RELATIONSHIP	EFFECT OF USING LARGE VALUE OF PARAMETER
Aspect Ratio	Large	Inverse	Increasing aspect ratio yields thinner, more fragile, more numerous airfoils, cracks, FOD & erosion damage.
No. of Stages	Medium	Inverse	Increasing no. of stages increases probability of cracking, FOD & erosion damage.
No. of Airfoils	Medium	Inverse	Same. Increased no. of airfoils usually implies thinner airfoils.
Surge Margin	Medium-Small	Direct	Increased surge margin and reduced chance of surging means fewer overstressed airfoils and better starting reliability.
L^2/D (length ^{2}/rotor diam.)	Medium-Small	Inverse	Shaft critical speed parameter - affects stiffness & vibration problems.
Inlet Specific	Flow-Small	Inverse	Increased flow rate means more transonic airfoils with thin tip leading edge and reduced resistance to erosion.
Flow Path Shape	Small	Constant ID is better	Constant outside diameter leads to aft stage airfoils which are shorter and thinner, and more subject to erosion damage. Also COD geometry leads to larger thermal transients in the rear stage disks.

Reliability Assessment

Reliability of the baseline Reliable Engine compressor and HTF configurations was evaluated by applying factor analysis techniques to the parameters having reliability impact. These factors were: number of stages; number of airfoils; length^2/diameter; surge margin; aspect ratio; inlet specific flow and flowpath shape.

The factor analysis technique consisted of assigning weighted reliability indicators to the selected parameters, based on how these parameters affected the compressors sensitivity to foreign object damage (FOD), vibration, airfoil erosion, cracks, etc., the realibility indicators for each case were then summed, providing comparative reliability indicators as shown in Table 5-2.

This analysis assumed that only the flowpath hardware was subject to variation of the design parameters. In other words, cases and disks were assumed to be equally reliable for both configurations due to their heavy construction and conservative design assumptions.

A comparison of the reliability indicators in Table 5-2 reveals that while the base case Reliable Engine has higher indicator values due to its slightly lower aspect ratio and lower number of airfoils, the HTF configuration has higher indicator values in the surge margin, L^2/D and flowpath categories. The net result is a lower total reliability indicator value and higher estimated failure rate for the baseline Reliable Engine Compressor.

Table 5-2. Comparison of Reliability Indicators for HTF and Baseline Compressors

	Baseline Reliable Engine		Selected HTF Configuration	
	Value	Reliability Indicator	Value	Reliability Indicator
Aspect Ratio	1.2	17	1.24	15.9
Surge Margin	15%	3	20%	6
Number of Stages	9	6.4	9	6.4
Number of Airfoil	1089	6.3	1326	5.2
L^2/D	129	6.5	119	6.6
Inlet Specific Flow	35	5	35	5
Flowpath Shape	COD	1	CMD	1.8
Compressor Exit Mn	.34	0	.39	0
Reliability Indicator Total		45.2		46.9
Estimated Compressor Failure Rate (Per 10^6 hours)	13.3 (1)		13.2 (1)	

(1) User must add 5.2 to rate shown to get the total compressor failure rate (which includes cases, disks, seals)

Cost-of-Electricity

The Cost-of-Electricity (COE) calculated for each of the High Reliability Gas Turbine configurations is a levelized cost per kilowatt-hour (in cents) over the 30 year life of the power plant. The COE calculation procedure was designed to measure the impact of design and reliability changes on overall plant costs.

The variables used in the COE calculation are:

- *Compressor Efficiency
- Turbine Efficiency
- *Number of compressor stages
- *Number of compressor airfoils
- Number of turbine stages
- Number of turbine airfoils
- Power plant utility characteristic
- *Equivalent Unscheduled Outage Rate

* Factors which were affected by this study (compressor related variables).

A United Technologies Corporation (UTC) Computer Program (Ref. 1) calculates the plant capacity and heat rate of the design, and the subsequent COE. The program also calculates the delta COE from a reference design, which in this case, is Reliable Engine, defined in EPRI Program RP1187.

Power Plant Equivalent Unscheduled Outage Rate (EUOR) is determined by using the curve shown in Figure 5-1. The Utility Characteristic (M) is a parameter used to account for utility size and defines the effect of a single power plant unit outage on the equivalent outage rate. A utility characteristic of 1000 was used in this analysis. This implies that the power plant represents a relatively small portion of the total utility capacity.

The impact of each of the compressor-related variables on the overall plant COE is shown in Figures 5-2 and 5-3. The effect of compressor efficiency on COE is the overwhelming factor in this analysis.

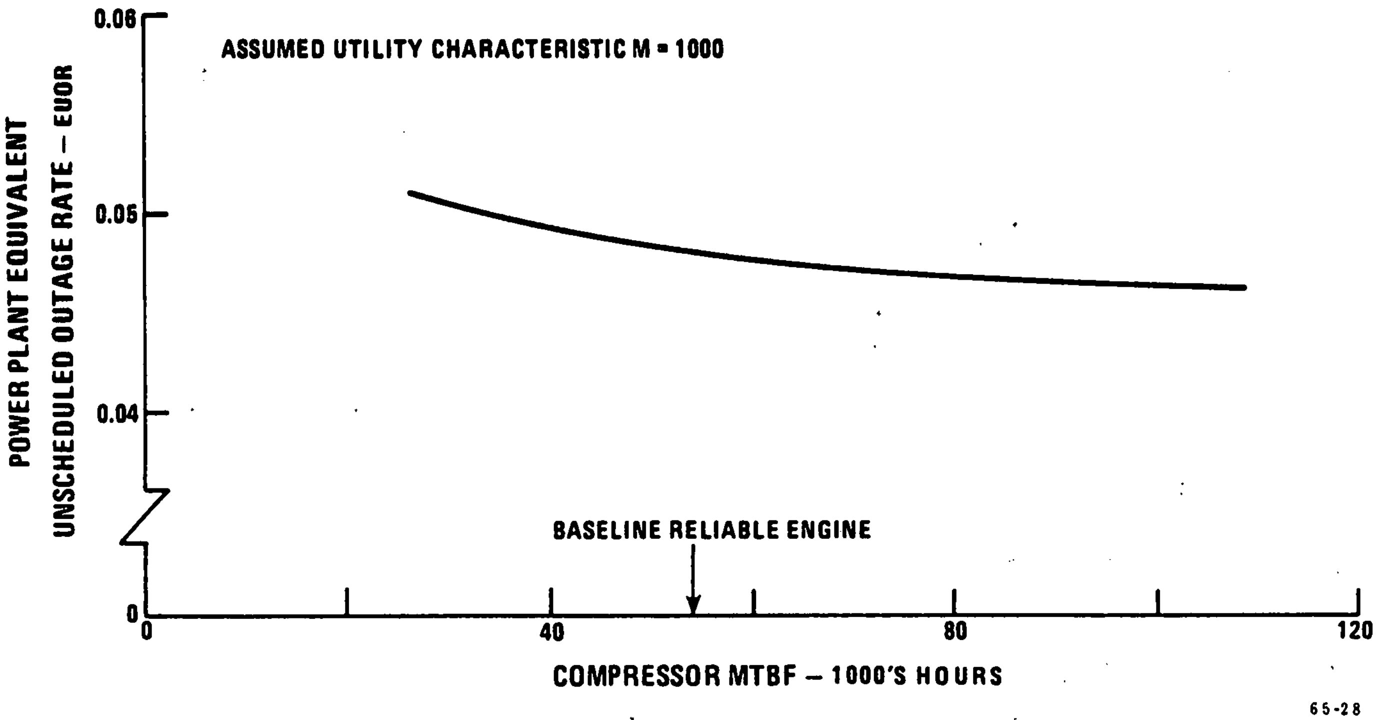

Figure 5-1. Overall Power Plant EUOR Variation with Compressor Failure Rate

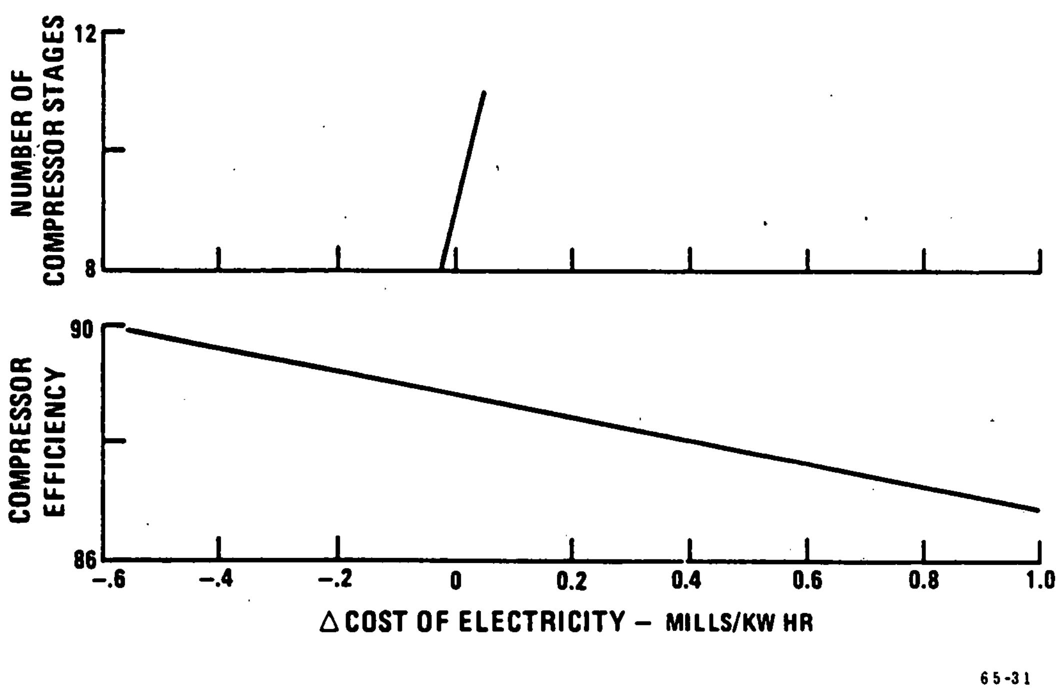

Figure 5-2. Compressor Efficiency and Number of Stages vs. Δ Cost of Electricity

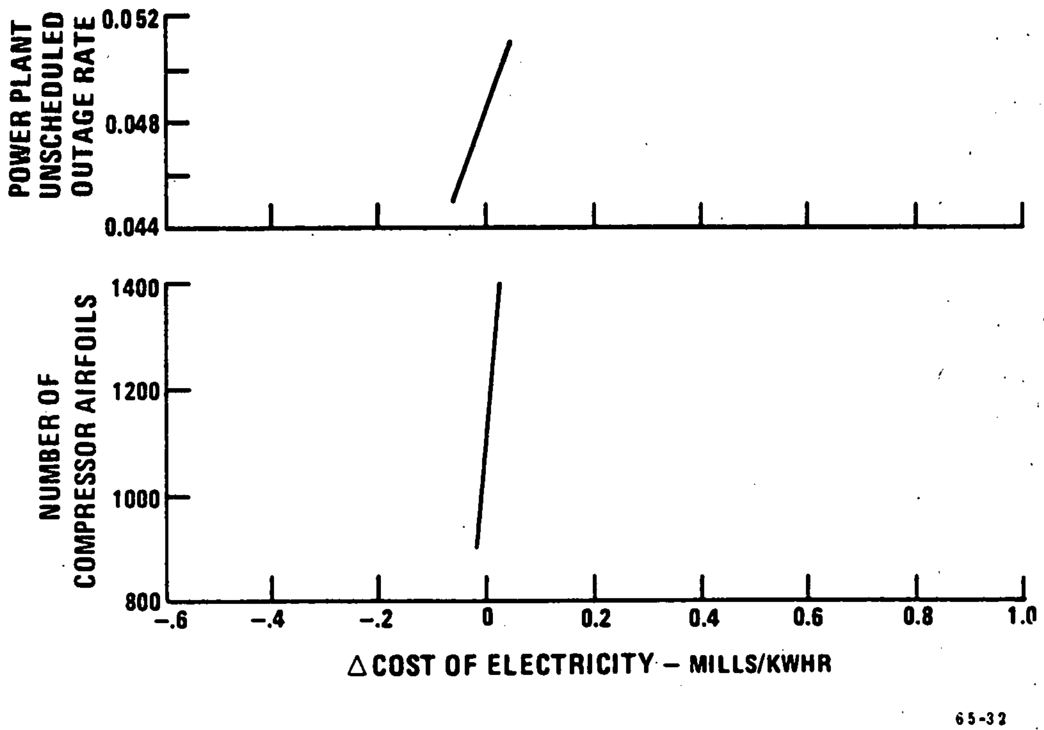

Figure 5-3. Number of Compressor Airfoils and Power Plant Unscheduled Outage Rate vs. Δ Cost of Electricity

Table 5-3 shows the parameters used in the COE calculations for the Reliable Engine and HTF designs.

TABLE 5-3
PARAMETERS FOR COST OF ELECTRICITY CALCULATIONS

	Baseline Reliable Engine	High Through Flow Engine
Compressor Efficiency	0.888	0.8835
Turbine Efficiency	0.925	0.925
No. of Compressor stages	9	9
No. of Compressor airfoils	1117	1304
No. of turbine stages	3	3
No. of turbine airfoils	336	336
Power Plant Utility Characteristic	1000	1000
Equivalent Unscheduled Outage Rate	0.0482	0.0480
COE Mills/kWh	90.128	90.369
Δ COE Mills/kWh	-0-	+0.261

The baseline (HRGT) COE was calculated to be 90.128 mills per kilowatt-hour. The final HTF preliminary design COE was calculated to be 90.369 mills per kilowatt-hour.

Reference 1:

Smith, R.M., "Cost of Electricity Computer Program for High Reliability Gas Turbine Engine," PSD memo R.M. Smith to P.J. Farris, July 5, 1979.

Section 6
CONCLUSIONS AND RECOMMENDATIONS

The HTF compressor/diffuser configuration defined in this program differs from the Baseline Reliable Engine Compressor in reliability (0.54% increase) and cost of electricity (0.29% increase). The COE is primarily impacted by the lower efficiency of the HTF (0.45% lower). Conclusions as to the desirability of the HTF compressor based on a direct comparison with the Reliable Engine Compressor are difficult to reach for two reasons. 1) the calculated performance, reliability and COE are very similar and (2) the baseline compressor is a conceptual design which lacks the analytical depth of the HTF design. The designs should be refined to a similar degree of detail for valid comparisons to be made. Considering the preliminary nature of these studies, the two configurations can be considered to be essentially equivalent.

We conclude that the HTF could be considered an excellent alternative to the Reliable Engine Compressor because it would have lower stage loadings, and consequently lower aerodynamic risk, while utilizing the desirable features of low aspect ratio and minimum number of stages. The HTF configuration would involve no more development than would the Reliable Engine Compressor.

It is recommended that the HTF compressor design be re-examined for use in the Reliable Engine when the current Reliable Engine Compressor design has reached an equivalent detail of analysis. Testing of the Department of Energy low aspect ratio compressor (Contract No. DE-AC-05-76R05035) conducted subsequent to the preliminary design analysis described herein has demonstrated excellent efficiency at the high loading levels targeted for the Reliable Engine baseline compressor conceptual design. Specifically, the axial compressor exceeded 89% efficiency at approximately 6.5 pressure ratio and 100% speed. Blade maximum vibratory stresses throughout the tests were a modest 15 KSI or about half the maximum allowable level. Surge margin substantiation for the Department fo Energy low aspect ratio compressor is in progress. Additional optimization of the HTF compressor with respect to increasing the selected loading limit as demonstrated in the Department of Energy compressor is therefore recommended during any final design analysis that may be conducted to further evaluate the reliability and COE of the high-through-flow compressor.

It is important to reiterate that the object of this study has been to determine the potential benefits of applying advanced compressor design techniques to a highly reliable industrial gas turbine design. An HTF compressor as described above would require a full development program to achieve the calculated efficiencies. No effort has been made to estimate the cost of such a development effort or include it in the COE studies.

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