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A DUAL FLOW HIGH MACH NUMBER CENTRIFUGAL COMPRESSOR FOR A SMALL GAS TURBINE APU

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ABSTRACT

Small gas turbine auxiliary power units (APU's) of conventional load compressor type wherein the gas generator or core module directly drives a separate centrifugal load compressor are installed in aircraft and helicopters to supply both compressor air for main engine starting and air conditioning combined with shaft power to drive an electric generator.

This paper describes the test development of a dual flow centrifugal compressor (DFC) where the impeller flow was split into two streams, the inner (hub) stream supplying compressed air to the gas generator core module, and the outer (DFB) bleed stream delivering a compressed air to the aircraft pneumatic power system.

DFC development rig testing revealed that the hub or core stream satisfied compressor design requirements but that the DFB stream flowpath demonstrated unstable characteristics with decreasing efficiency as test speeds were increased.

At the time of the development program in the early 1990's convergence difficulties were encountered with CFD attempts to corroborate the test results, and thus pinpoint plausible explanations, as a consequence a renewed upgraded 2010 CFD analysis of the dual flow compressor is presented herein confirming the test performance characteristics of both flow streams and the fundamental reason for poor DFB performance as excessive diffusion at high relative Mach numbers.

NOMENCLATURE

A	Area
b	Impeller exit blade height
Beta	Blade angle
Cp	Diffuser Pressure Recovery = (Pe-p2)/ (P2-p2)
C	Absolute velocity
CFS	Inlet Volume flow
D	Diameter
De Haller Number	= W2/W1rms
Etac	Compressor efficiency
H	Head
IGV	Inlet Guide Vane
M	Mach number
N	RPM
Ns	Specific Speed (dimensionless form) $= \omega \sqrt{CFS} / (g\,Had)^{0.75}$
P	Total Pressure
Q	Flow Function = $W \sqrt{T} / AP$
q	Work factor = $\Delta H / U_2^{\,2}$
Rc	Pressure Ratio = Pe/P1
SL	Sea Level
T	Total Temperature
U	Tangential Velocity
W	Airflow or Relative velocity
$W\sqrt{T}/P$	Normalized flow function
θ	Inlet Prewhirl (with rotation)
β_2	Backsweep Angle
Δ	Difference
ω	Angular velocity

Subscripts

ad	Adiabatic
1	Impeller Inlet
2	Impeller Tip
c	Compressor or Corrected
e	Scroll exit total

Note all angles are with respect to the meridional plane.

1. INTRODUCTION

A dual or split flow centrifugal compressor (DFC) was previously developed in the 1970's by Meshew and Swenski [1] as a simplified "Load Compressor" for a pneumatic supply small gas turbine APU. This concept integrated both the load (bleed) compressor and gas generator compressor into a single impeller.

Lower pressure bleed air was split off the impeller shroud at an intermediate radius matching the desired bleed supply pressure, while the higher pressure APU gas generator core flow was further compressed centrifuging out to a (larger) impeller tip radius. This DFC concept is depicted schematically on Figure 1, illustrating the compactness of the design approach and clearly intrinsic weight savings so crucial for airborne applications.

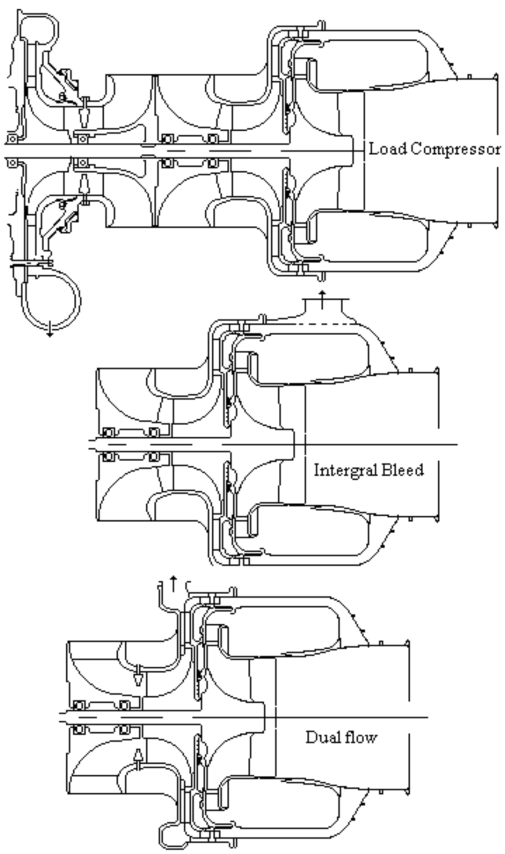

Figure 1. Pneumatic Air Supply Options

Potential merits of DFC type APU relative to the existing load compressor and integral bleed configurations are listed as follows :

- Independent choice of optimum bleed and APU core pressure ratios.
- Higher power to volume ratio.
- Lower cost, reduced part count, higher reliability.
- Improved starting characteristics.

The test development efforts of Meshew and Swenski [1] revealed low efficiencies for the flow split off from the impeller outer shroud which was presumed to stem from high flow recirculation losses in the impeller exit vaneless space. Several years later it was conceived by the author that the use of inducer throat shroud bleed [2], and or, [3] inlet guide vane (IGV) modulation could possibly resolve the performance problems encountered in reference [1].

It was decided therefore to conduct a compressor test research program to demonstrate the effect of both inducer shroud bleed and IGV regulation on a small dual flow compressor rig.

A small 4.35 inch (110mm), tip diameter impeller was chosen as a test demonstration vehicle. This impeller was scaled down from a larger existing production high specific speed impeller with shroud blade stock added for the dual flow path which was designed to split off 30% of the total inlet flow. Pertinent compressor design parameters are listed below.

Table 1. DFC Design Parameters, Nc 102,000 rpm

Flow streams	Core (DFC)	Bleed (DFB)
Flow (lb/sec, SL, 59 F)	1.35	0.55
Pressure ratio	5.6	4.0
Inlet Specific Speed (total)	0.89	0.66
Tip diameter D_2 (inch)	4.35	3.8
Inlet Shroud diameter (inch)	2.65*	3.08
Tip Width b_2 (inch)	0.20	0.12
Backsweep β_2 (°)	45	35
Inlet Prewhirl θ (°)	0	0
Inducer tip Rel Mach	1.35	1.42
Inducer Throat area (in^2)	3.8	3.8
Diffuser throat area (in^2)	1.05	0.6

* Selected quasi dividing streamline.

A photograph of a dual flow type test impeller prior to final back shroud machining is shown on Figure 2. The impeller had nine long blades and nine intermediate splitter blades with radial element blading and radial straight splitter blade leading edges. Inducer blade normal tip leading edge thickness was .012 inch.

Figure 2. DFC Impeller Prior to Final Backface Machining

2. TEST RIG DESCRIPTION

The compressor module of the test turbodrive rig is shown on Figure 3 and featured partial span IGV's with inducer shroud bleed adjacent to the inducer throat. This bleed was not re-circulated and was dumped to test cell ambient conditions.

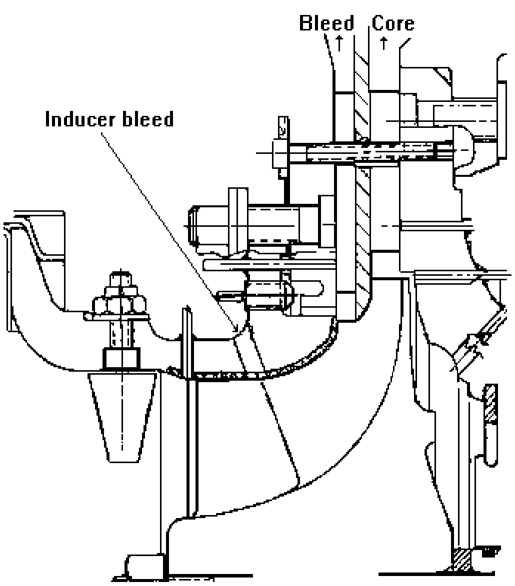

Figure 3. Turbodrive Test Rig

The compressor shroud was plasma sprayed with an abradable aluminum epoxy material in order to prevent possible destructive rubbing operation with tight clearances at the design speed of 100,000 rpm.

Compressor back pressure could be set by two separate valves for the core and shroud bleed flows. Both discharge scrolls are shown on Figure 4 and were designed with limited exit area to prevent operation into deep choke, additionally the bleed scroll incorporated a choked orifice plate sized to simulate the flow characteristics of an aircraft pneumatic secondary power and air conditioning duct system.

The DFC and DFB diffusers both had 17 wedge type vanes and were integrally machined as one component sandwiched in-between the turbo front and aft shrouds.

3. INSTRUMENTATION AND DATA REDUCTION

The compressor instrumentation listed on Table 2 was used to define the impeller, diffuser, and overall stage performances. Six static pressure taps were equally spaced along the outer stationary shroud, from the inducer inlet to the impeller tip. Due to the small impeller blade tip widths, tip instrumentation was confined to three static pressures equally spaced over one diffuser vane pitch.

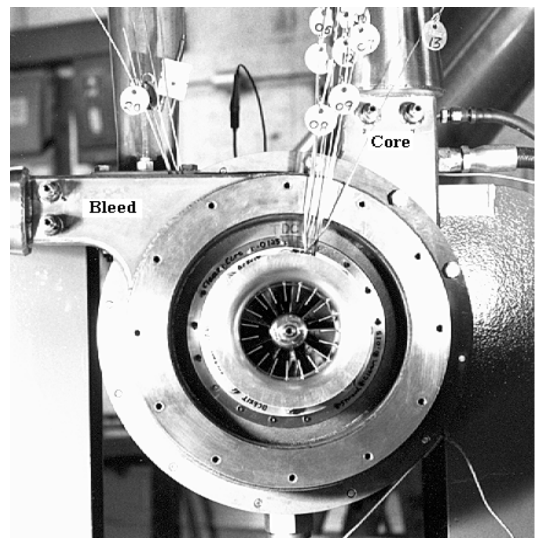

Figure 4. DFC Installed in Test Cell

Estimated uncertainty on stage test efficiency at design speed was +/- 0.5 % points under quiescent flow conditions.

Table 2. List of Instrumentation

Station	Temperature (R.T.D)	Static Pressure	Kiel probe Total Pressure
Inlet Venturi	1	3	1
Inducer Eye *		3	3
Impeller Tips	-	3	-
Scroll exits	3	3	3

Test measurements used in the calculation of the impeller performances were inlet total pressure and temperature, average tip static pressure, scroll exit total temperature, and airflow.

An online self calibrating electronic data acquisition system recorded all the test data. Flow continuity, impeller tip annulus area (with blockage allowance), and temperature rise were used to compute the mixed exit meanline impeller exit vector conditions.

Since many map combinations were possible dependent upon DFB and DFC throttle conditions plus DFB IGV settings compressor mapping was confined from a DFC maximum flow point simulating an APU no load operation, to the surge point. DFC surge was distinctly audible but the bleed compressor emitted unstable operation increasing in pulsations at rotational speeds above 90% design.

4. COMPRESSOR TEST PERFORMANCES

4.1 Core Compressor DFC

Compressor test performances for the core (DFC) and shroud bleed (DFB) flows are shown on Figures 5 and 6 in terms of normalized flow (W√T/P), versus pressure ratio and adiabatic efficiency, with percent design corrected speed as a parameter.

DFC adiabatic efficiency based on scroll exit total pressure at design corrected speed of 100%, with a surge margin of 5% was 77% with a pressure ratio of 5.63. Peak overall efficiency was 80.4 % at 95 % speed. The impeller and diffuser performances shown on Figure 6 at this condition were 86.4% with a pressure recovery (Cp) of 0.69, which is respectable considering the relevant impeller size and inducer tip Mach number of 1.35.

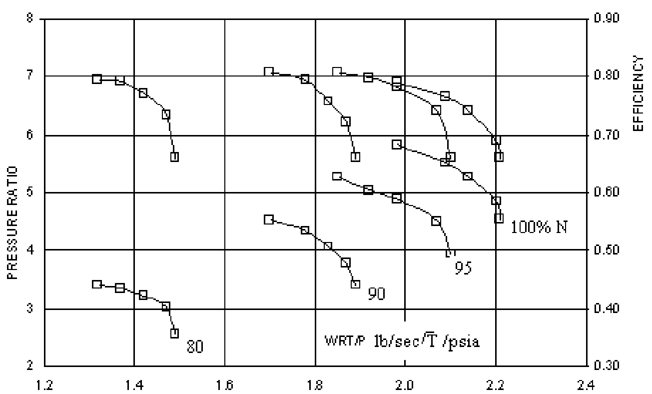

Figure 5. DFC Core Overall Performance

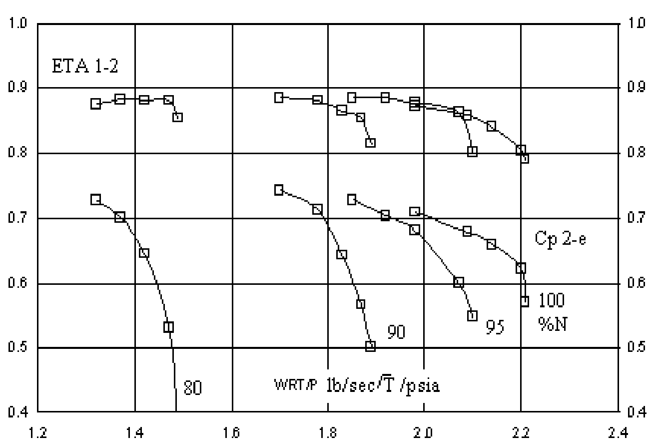

Figure 6. DFC Impeller and Diffuser Performances

The DFC test results therefore confirmed that the core performance characteristics were typical of gas turbine type high specific speed moderate pressure ratio centrifugal compressor.

4.2 Bleed Compressor DFB

Compressor test performances for the shroud bleed (DFB) flowpath are shown on Figures 7 and 8 in terms of normalized flow versus pressure ratio and adiabatic efficiency, with percent design corrected speed as a parameter. The average bleed flow was 30% of the total inlet flow, which is almost in proportion to the ratios of the diffuser throat areas and stage pressure ratios.

Peak DFB adiabatic efficiency based on scroll exit total pressure at design corrected speed of 100% was only 48% with a pressure ratio of 3.1, with a peak overall efficiency of 68 % at 60 % speed. This was a result of low performances for the impeller and the diffuser as shown on Figure 8. At 100% speed the peak impeller efficiency dropped to 70% with a low diffuser recovery of 0.37. This reduction in efficiency was accompanied by the sharp increase in compressor work factor q, ($\Delta H / U_2{}^2$) shown on Figure 9 which would normally be characteristic of flow reversal and recirculation.

This result was not entirely un-expected as reported compressor rig test performance calibrations [1],[2], with gas turbine type dual flow compressors had revealed low DFB efficiencies thereby prompting alternate approaches in this venture with the incorporation of part span inlet guide vanes and inducer shroud bleed, as a possible methods of both unloading the DFB shroud diffusion and curtailing inducer tip backflow.

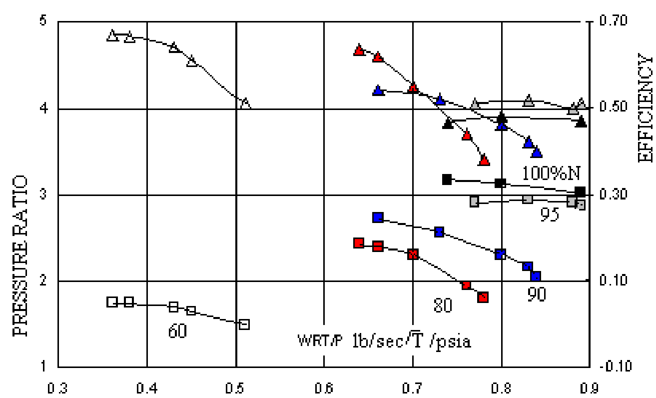

Figure 7. DFB Overall Performance

In analyzing the DFB tip vector conditions it was noted that an abnormally high tip blockage factor (0.8-0.7 order) was necessary to produce reasonable impeller slip factors typical of impellers with 18 blades and 35 degree backsweep.

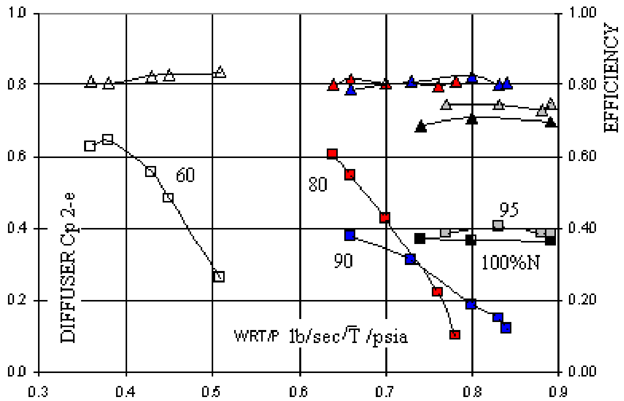

Figure 8. DFB Impeller and Diffuser Performances

It is germane however to record that the complete DFB performance was often unstable and that the performance data presented represents a selective average of the fluctuating test measurements particularly above 90% corrected speed where the inducer tip relative Mach numbers exceeded 1.2, and the impeller relative velocity de Haller number [5] W2/W1rms decreased as shown plotted on Figure 10. The next test phase was thus initiated to experimentally investigate the effects of partial IGV's and inducer shroud bleed.

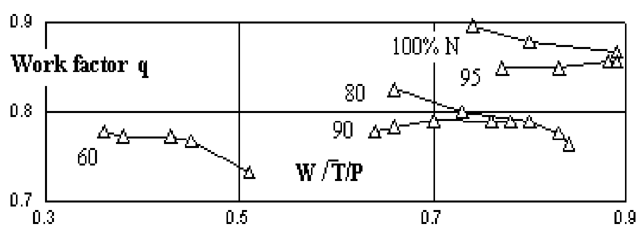

Figure 9. DFB Work Factor Variation

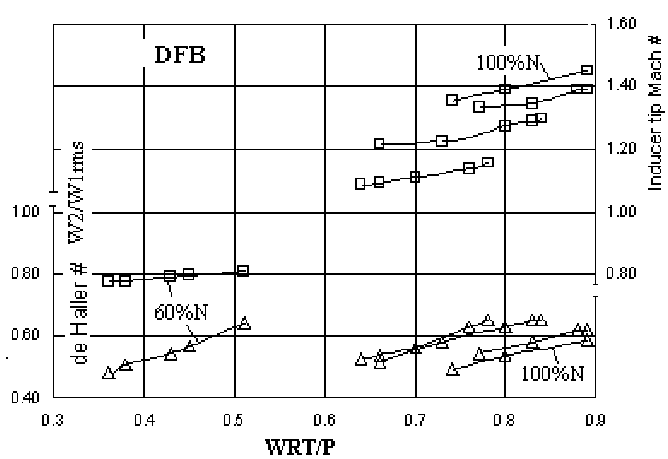

Figure 10. DFB Mach and de Haller Number

4.3 Tests with IGV's and Inducer Shroud Bleed

As a consequence of the increased DFB instability characteristics above 90% speed the most meaningful comparison of IGV and inducer bleed effects to be made was at 90% corrected speed, which is shown on Fig11, revealing that neither inducer shroud bleed or IGV set at 20 deg (with rotation) offered any significant improvement in DFB performance . This was a disappointment since the major program goal had been to develop a viable dual flow compressor offering both weight and manufacturing cost savings when installed in a small gas turbine APU.

Further inspection of the test results shown on Figure 11 shows that the core peak compressor efficiency again remained essentially unchanged and that 20 deg stagger of the partial IGV's actually decreased the core flow and pressure ratio, which was possibly due to the extension of the partial IGV slightly below the selected inducer quasi dividing streamline. As regards the effect of inducer shroud bleed it may be significant that bleed slightly decreased the core pressure ratio since if the shroud flow was separating bleed would normally be expected to act opposite by curtailing the amount of separation.

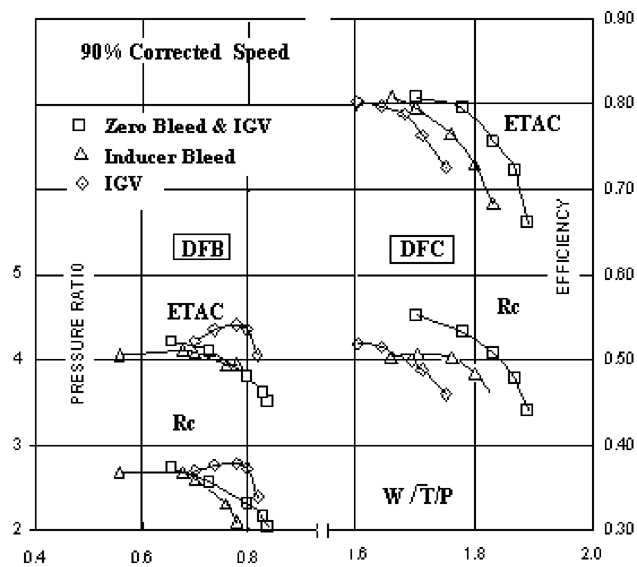

Figure 11. Effect of IGV and Bleed

The test program terminated with two additional tests, one with a tandem diffuser, and the next with a semi-vaneless diffuser, the intention being to extend coverage of the DFB impeller performance not so much as the modified diffuser performances.

4.4 Tests with DFB Diffuser Modifications

The design channel diffuser was modified as shown on Figure 12, first with the tandem and second with the semi–vaneless configuration where the first row vanes were removed.

The two diffuser changes were made in-situ without diss-assembly of the DFC module and rotating assembly

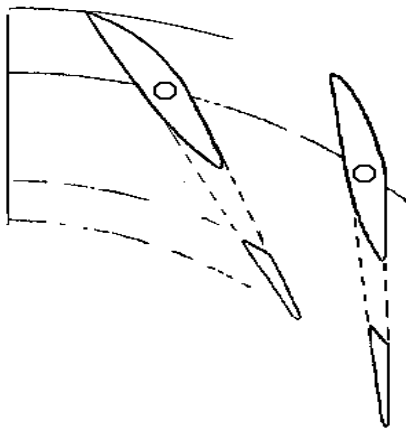

Figure 12. DFB Diffuser Modifications

Tests with the tandem diffuser showed lower performance than the design diffuser with only a minor increase in choked flow, however tests at 90% N with the semi-vaneless diffuser, Figure 13, did show an increase in choke flow split from 33% to 40%. The surprising result was that the DFB efficiency increased slightly while the DFC efficiency fell 5% points apparently as the different flow split moved its impeller further towards choke.

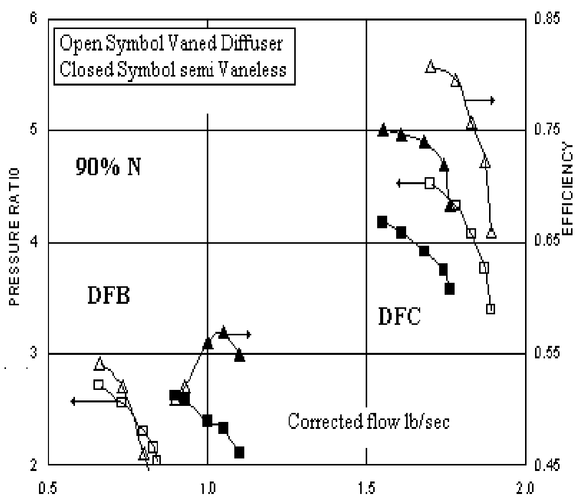

Figure 13. Test results with Semi-Vaneless Diffuser

A CFD analysis of the DFC compressor had been attempted during the design phase (circa early 1990's) but failed due to convergence problems. The impeller design is however mathematically quite challenging since the shroud curvatures are particularly tight combined with DFB inducer tip Mach numbers over 1.4 at design speed.

One analogy to the DFB flowpath would be a high hub/tip diameter ratio centrifugal compressor [4] the efficiencies of which are particularly sensitive to increasing inducer tip Mach number when combined with high shroud curvature.

Since CFD technology has improved substantially over the last decade a renewed computational analysis was commenced with the intent of pinpointing more specific reasons for the DFC performance problems more so than suspected excessive diffusion along the shroud at the high inlet Mach numbers.

It was at this time that it was also conceived that the DFB flow hysteresis may have been stabilized by divorcing the two streams with separate rotating shrouds as patented by Bornemisa [6]. Such a conceptual divided flow impeller is portrayed on Figure 14 proposed for either bleed air or turbofan bypass flow applications, exclusion of the DFC shroud would be a more practical concept from an integral digital machining aspect...

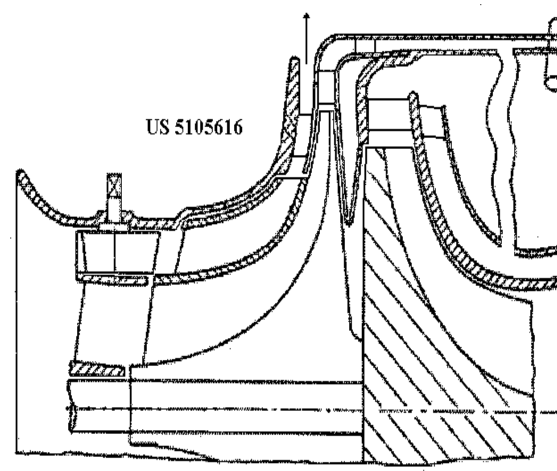

Figure 14. Conceptual Divided Flow Impeller

5. 2010 CFD ANALYSIS

The CFD model was constructed for a single impeller main blade passage along with a single vane passage for each of the two diffusers. Figures 15a and 15b show views of the grid, which contained 491,648 cells. The model was constructed by combining 12 structured grid blocks. The impeller flow path was comprised of six blocks while each diffuser contained two blocks. The inducer bleed slots were also included in the model, although the bleed case was not run. The impeller flow path was connected to the diffusers using mixing planes. Mixing planes were also utilized at the diffuser exits as a convenient method to determine mixed-out stage performance. The model was analyzed using ADPAC [7], a general, multi-block CFD code developed by NASA-GRC. The Spalart-Allmaras [8] turbulence model with wall functions were chosen.

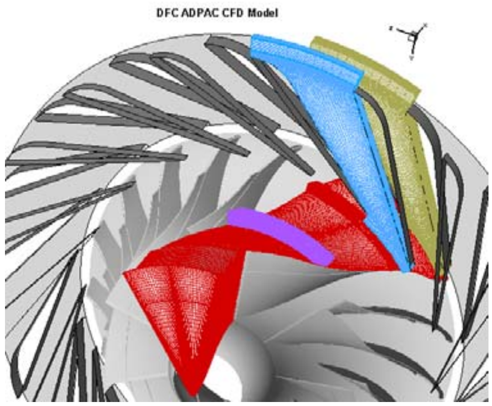

Figure 15a. DFC Compressor CFD Grid

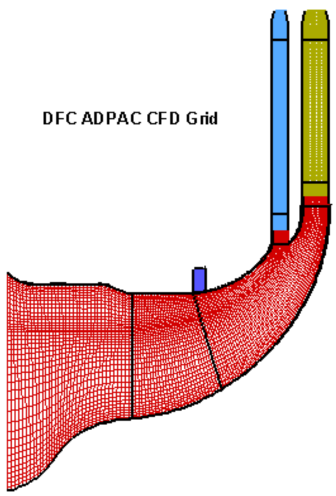

Figure 15b. DFC Compressor CFD Grid, Meridional View

The model was analyzed for two speed lines: 90 and 100%. To determine the DFB performance map, the DFC exit pressure ratio was fixed close to the design point value and the pressure ratio of the DFB varied. Likewise for the DFC map, the DFB exit pressure ratio was fixed. This analysis method simulated the test operation. Figure 16 shows the predicted CFD pressure ratio and efficiency maps compared to test. In general, the CFD analysis predicted performance trends similar to the test with very good DFC performance and poor DFB efficiency. The model appears to over-predict the DFC choked flow and under-predict the DFB choked flow. These inaccuracies may be attributed to differences in the respective back pressures for the CFD runs compared to test. Unfortunately, there was not enough detailed test data available to sort out the differences in choked flow.

Another noticeable difference between the CFD predicted performance and the test data is the higher predicted DFB efficiency at 90% speed. As shown, the CFD model is predicting a DFB efficiency approximately 10 points higher than test. It is suspected that this difference is caused by inaccuracies in the CFD tip clearance modeling. At 100% speed, the DFB efficiency was approximately seven points higher than test. Table 3 shows a performance comparison summary of representative points between design, test, and CFD.

Table 3. DFB and DFC Performance Summary at 100% Speed and Flow

	DFB			DFC		
	Design	**Test**	**CFD**	**Design**	**Test**	**CFD**
Pressure Ratio	3.65	3.12	3.03	5.10	5.27	5.45
Efficiency (%)	74.8	48.0	54.9	78.1	74.0	80.8
Flow Split (%)	33	27	22	67	73	78

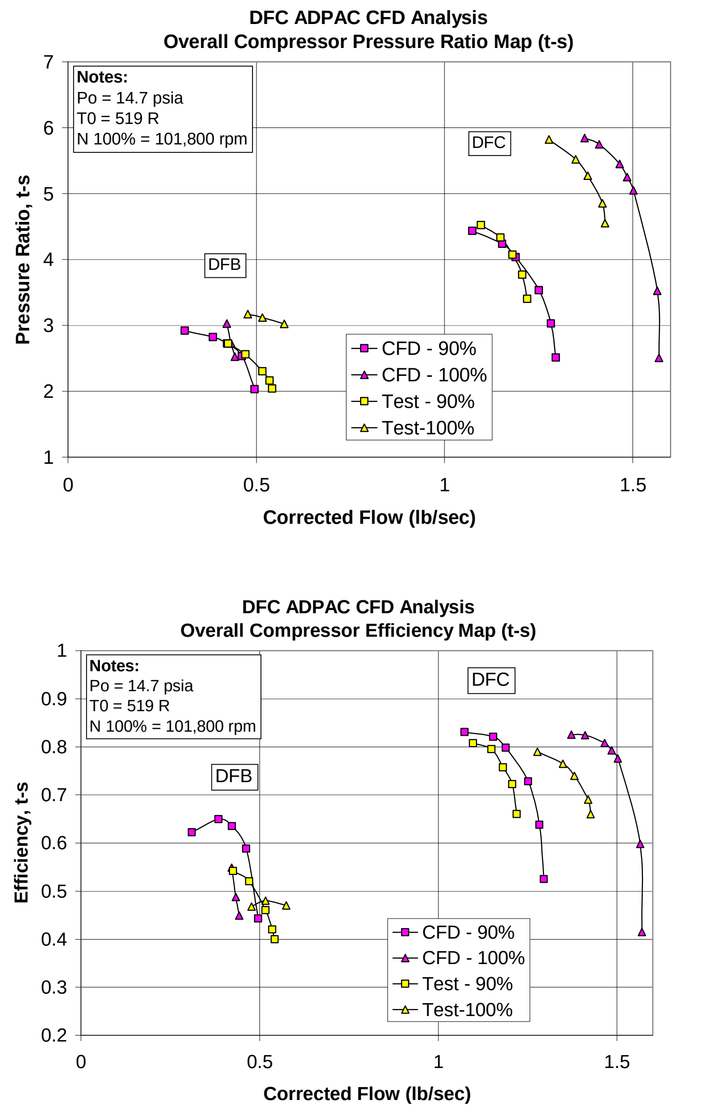

Figure 16. DFB and DFC CFD Predicted Performance Map Compared to Test, Pressure Ratio (top) and Efficiency (bottom)

It was encouraging that the CFD model predicted similar performance trends to the test data, but the real value of the analysis was revealed in the flow field plots. Figures 17 and 18 show flow field plots for the peak DFB efficiency point.

Figure 17 shows relative Mach number contours and streamlines for meridional views on either side of the splitter blade. Figure 17a shows a large area of flow separation along the shroud associated by the inducer shock. Although this secondary zone is somewhat large, it is typical of transonic centrifugal compressors. Figure 17b shows the mid passage flow field between the main blade pressure side and splitter blade. The shroud secondary zone in this plot initiates from a secondary shock downstream of the splitter leading edge. This secondary shock is not typical of centrifugal impellers operating at the peak efficiency point and is most likely contributing to the poor DFB performance.

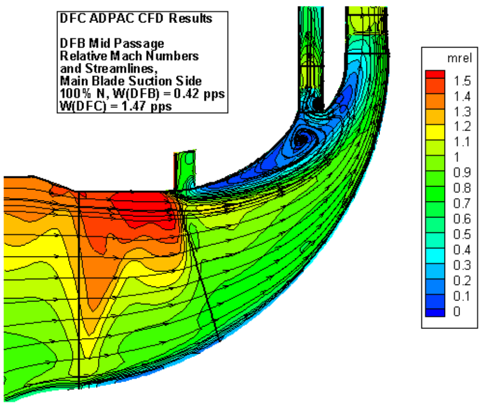

Figure 17a. Impeller Mid Passage Relative Mach Number Contours and Streamlines, Main Blade Suction Side, 100% Speed

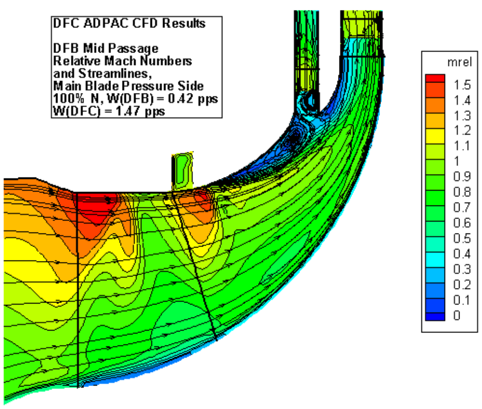

Figure 17b. Impeller Mid Passage Relative Mach Number Contours and Streamlines, Main Blade Pressure Side, 100% Speed

Figure 18 shows two relative Mach number contour plots along slices corresponding to the mid span of the DFB and DFC flow paths. When comparing Figure 18a with the two plots shown in Figure 17, it is apparent that nearly the entire DFB flow path is comprised of low-momentum secondary flow. Observably, this large amount of separated flow is the probable culprit of the low DFB performance. The DFC flow field, on the other hand, shows a flow pattern, which is typical of well performing centrifugal compressors with only a minimal amount of secondary flow.

Figure 18a and 18b also show the mid span diffuser predicted flow patterns. These flow patterns can be compared with test lampblack flow traces on the diffuser walls reproduced on Figure 19. These lampblack traces taken at 90% speed prior to heavy instability partially confirm the CFD predicted diffuser flow patterns, especially the apparent heavy separation in the DFB covered channel area.

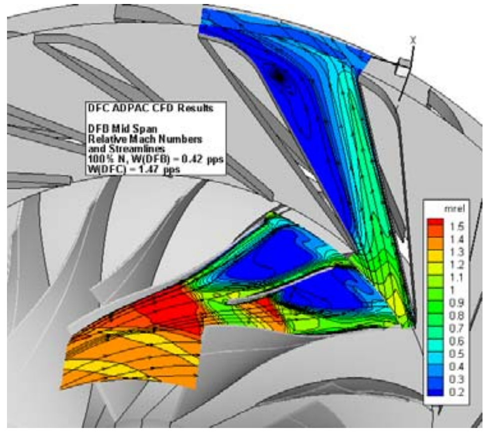

Figure 18a. DFB Mid Span Relative Mach Number Contours and Streamlines, 100% Speed

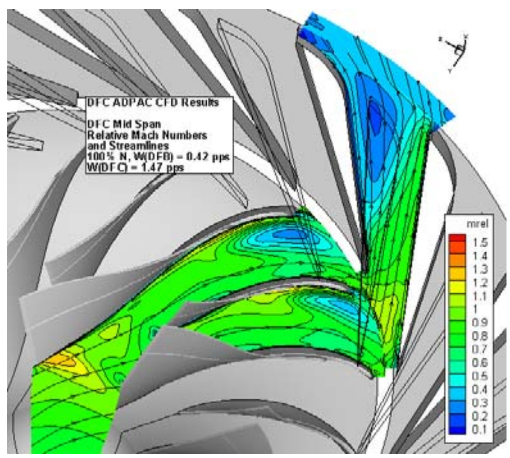

Figure 18b. DFC Mid Span Relative Mach Number Contours and Streamlines, 100% Speed

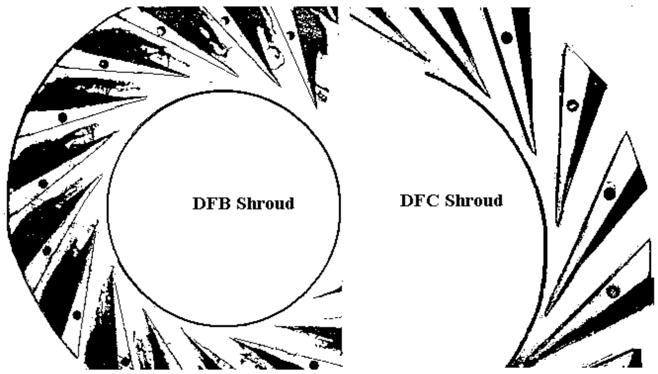

Figure 19. Diffuser Lampblack Flow Traces

The CFD analysis confirmed the speculation that the poor DFB performance was caused by shock initiated flow separation. The CFD analysis also showed that a secondary shock downstream of the splitter leading edge was heavily contributing to the poor performance. The DFB test performance was worse than the analysis most likely because the CFD model did not completely capture the detrimental effects of the relatively large tip clearance and high shroud curvature.

Further interpretation of the flow patterns shown in Figures 17 and 18 reveal that the DFB flow path exhibits characteristics similar to a choked flow condition, although the majority of the flow is exiting through the DFC diffuser. The inability to separate the inducer flow characteristics between the DFB and DFC flow paths appears to be an inherent deficiency in the design. This conclusion provides further argument that the performance could be improved by using the proposed divided flow impeller shown in Figure 14.

Although the CFD analysis did confirm the poor performance observed during testing, the fact that reasonable DFB performance was predicted at lower speeds offers some hope that the dual flow compressor with shared impeller flow path might be viable. For example, an additional DFB analysis was run at 100% speed line with the DFC peak pressure ratio set almost at the surge condition. For this combination with increased core back pressure, the DFB efficiency increased dramatically up to 65%. The secondary shock pattern in the splitter passage disappeared as well. Figure 20 shows the much improved DFB mid span flow pattern for this case (with DFC close to surge).

Although this condition would not have been tested due to the proximity to the surge point, it provides some evidence that reasonable bleed performance may be possible with a shared impeller flow path. Impeller design modifications may offer substantial improvement, particularly a reduction in the shroud curvature and a tighter tip clearance. Certainly, this design concept would benefit greatly when applied to a compressor with a more modest inducer relative Mach number (closer to sonic condition).

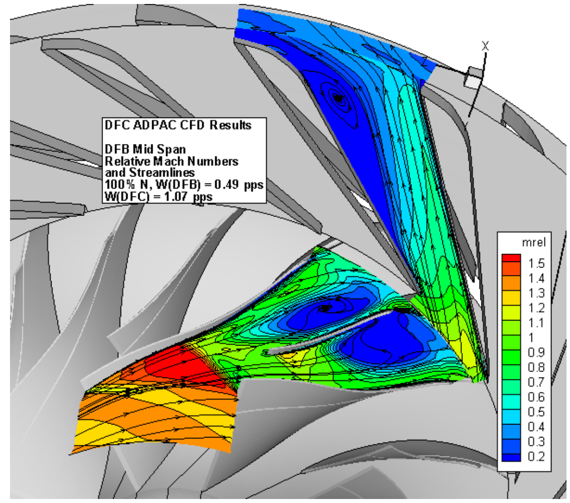

Figure 20. DFB Mid Span Relative Mach Number Contours and Streamlines, 100% Speed, with DFC Close to Surge

6. CONCLUSIONS

The design and testing described of this dual flow compressor was performed in the early 1990's and was pursued with the intent of obtaining a significant reduction in weight and cost of small gas turbine airborne auxiliary power units supplying both pneumatic and electric power, with the capability of independent APU core and bleed pressure ratio selection allowing core cycle and the aircraft pneumatic system design optimization.

It was recognized at the start of this test demonstration that dual flow compressors had been tested previously [1] exhibiting DFB performance deficiencies and flow instability, yet no satisfactory specific precipitating cause had been clearly identified. It was presumed that these recorded DFB characteristics and flow perturbations had been a result semi-stalled hysteresis operation along the inducer shroud to the impeller tip. Accordingly it was conceived that the incorporation of part span inlet guide vanes and inducer shroud bleed, might unload the shroud diffusion and curtail inducer tip stalling and backflow. Although IGV modulation was unsuccessful in that regard, modulation could be required to minimize DFB drag torque during start in an actual APU application.

These test experiments demonstrated that these two devices did not in this instance cure the DFB performance shortcomings and that the sharp increase in DFB compressor work factor q, and lower efficiency above 90% speed were still likely propagated by excessive diffusion triggering flow reversal and recirculation along the high Mach number inducer shroud.

The selected moderately high specific impeller design as a test vehicle was also an additional challenge in that the shroud curvatures were intrinsically tight combined with inducer tip relative Mach numbers over 1.4 at design speed. Choice of a lower specific speed could have relaxed the diffusion level but would have sacrificed the attributes of the intended APU application size and weight.

It was revealing that the DFC core efficiency remained high whilst the DFB was so low just as though both streams were completely divorced patterning the subsequent the CFD analysis.

The significance of this paper is that although the design performance goal for the DFB compressor flowpath were not satisfied by any means, had the current 2010 computational fluid dynamics technology been previously available a more successful dual flow compressor test demonstration might very well have been completed.

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