UNSHROUDED CENTRIFUGAL TURBOPUMP IMPELLER DESIGN METHODOLOGY

George H. Prueger, Morgan Williams, Wei-chung Chen, John Paris
Boeing / Rocketdyne Propulsion and Power

Robert Williams, Eric Stewart
NASA – MSFC

ABSTRACT

Turbopump weight continues to be a dominant parameter in the trade space for reduction of engine weight. Space Shuttle Main Engine weight distribution indicates that the turbomachinery make up approximately 30% of the total engine weight. Weight reduction can be achieved through the reduction of envelope of the turbopump. Reduction in envelope relates to an increase in turbopump speed and an increase in impeller head coefficient. Speed can be increased until suction performance limits are achieved on the pump or due to alternate constraints the turbine or bearings limit speed. Once the speed of the turbopump is set the impeller tip speed sets the minimum head coefficient of the machine. To reduce impeller diameter the head coefficient must be increased. A significant limitation with increasing head coefficient is that the slope of the head-flow characteristic is affected and this can limit engine throttling range.

Unshrouded impellers offer a design option for increased turbopump speed without increasing the impeller head coefficient. However, there are several issues with regard to using an unshrouded impeller: there is a pump performance penalty due to the front open face recirculation flow, there is a potential pump axial thrust problem from the unbalanced front open face and the back shroud face, and since test data is very limited for this configuration, there is uncertainty in the magnitude and phase of the rotordynamic forces due to the front impeller passage. The purpose of the paper is to discuss the design of an unshrouded impeller and to examine the hydrodynamic performance, axial thrust, and rotordynamic performance. The design methodology will also be discussed. This work will help provide some guidelines for unshrouded impeller design.

INTRODUCTION

Unshrouded impellers are used commonly in compressors and some industrial turbopumps. In rocket engine applications unshrouded impellers are successful employed on the Pratt & Whitney RL-10 upper stage engine. The current impetus to unshrouded impellers is the ability to increase impeller tip speed limits, which in some turbopump designs limit the operating speed of the machine. This would limit the speed at which the turbopump could operate and consequently set the lower bound for turbopump weight. The use of shrouded impellers in rocket turbopumps is based on the need to maintain performance levels at all required operating points in the design. The performance of an unshrouded impeller degrades as the tip clearance is increased, reference 1. This affects both the discharge pressure capability and the efficiency of the machine. High discharge pressure, cryogenic, turbopumps typically have substantial variation in impeller tip clearance from assembly, to chill, to operation. This is due to differences in materials between housing and rotor materials, as well as deflections in the housings due to pressure loads. The application of advanced computational fluid dynamic tools to design impellers which are less sensitive to tip clearance is one of the goals of the NRA8-21 Unshrouded High Performance Impeller Technology Project. Johannes Lauer, et. al, reference 2, conducted an experimental study on compressor impellers to ascertain what the design parameter drivers were for sensitivity to tip clearance. The study was not conclusive probably due to the variation in design parameters investigated, but lead to some insight into potential mechanisms for tip clearance sensitivity. The tools will also be used to predict axial thrust and rotordynamic coefficients of an unshrouded impeller.

DESIGN METHODOLOGY

The design speed was previously set by a conceptual evaluation of the tip speed capability of an unshrouded titanium impeller. The selection of head coefficient of 0.53 was selected to success at achieving wide operating range. Thus, the diameter was calculated to be 15.75 inches. Table 1 lists the design parameters.

Table 1: Impeller Design Point Parameters

Parameter	Value
Pump Speed, RPM	32,000
Impeller Tip Diameter, Inch	15.75
Impeller Tip Speed, Feet/sec	2200
Impeller Head Coefficient	0.53

Rocketdyne's LOSSISOLATION program was used to define the blade angles required to achieve the required head. Rocketdyne's centrifugal detail geometry through analysis tool, eTANGO, was used to develop the impeller contours, blade definition, initial pressure loading, and grids for subsequent

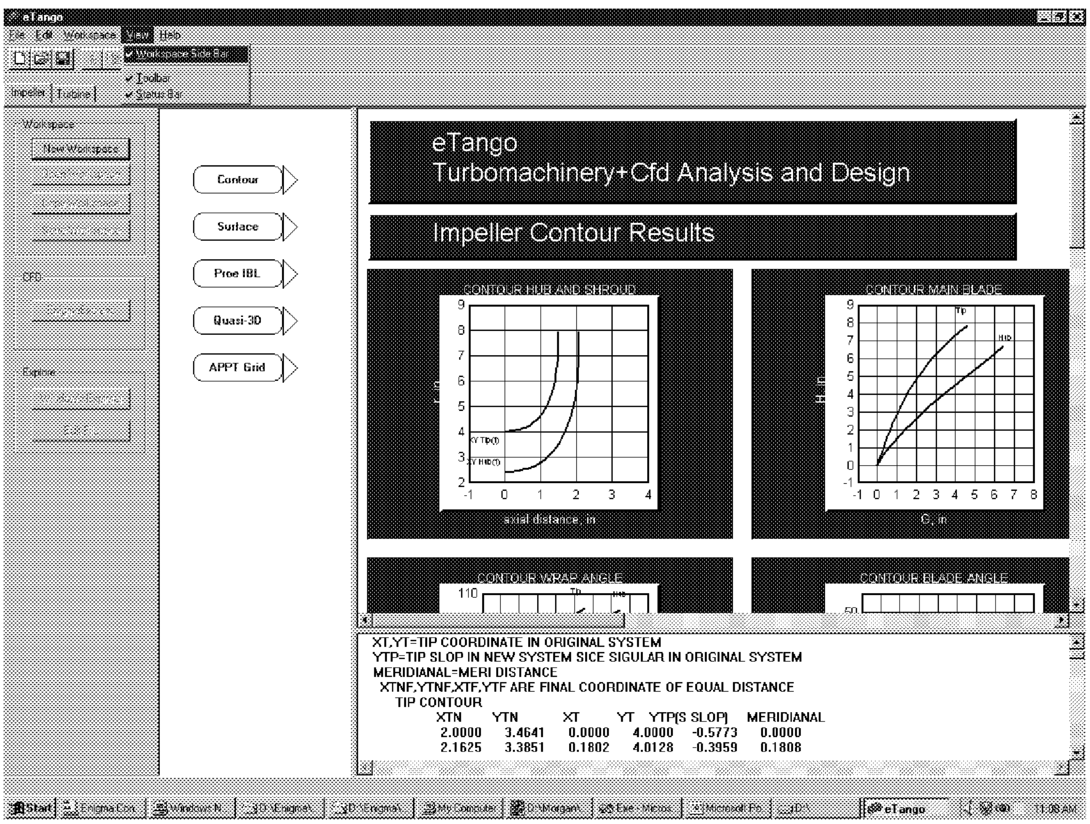

Figure 1: eTANGO Design / Analysis Interface

computational fluid dynamic (CFD) analysis of the geometry. Since Rocketdyne's typical impellers are shrouded, eTANGO was upgraded to incorporate tip clearance regions for the CFD analysis.

Figure 1 shows the eTANGO environment with the contours and blade description for a 6+6 impeller. The use of this design tool allowed the rapid generation of all the impeller designs required completing the ongoing trades study. The interface is intuitive and allows the design engineer to interactively make changes to the design variables and see the impact on the pressure loading. There is direct output from this tool to a Pro/ENGINEER generic model for rapid generation of the impeller solid model shown in figure 2, for a 6+6 impeller.

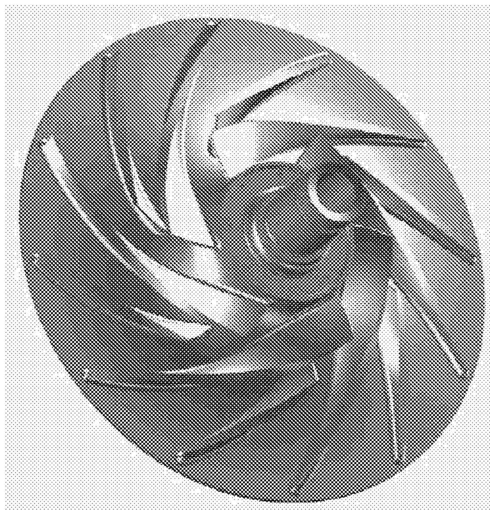

Figure 2: Generic Pro/ENGINEER Solid Model

DESIGN TRADES

Decreased performance sensitivity to tip clearance is a necessity to allow for incorporation of unshrouded impeller technology into rocket engine turbopumps. Based on literature review and tip clearance modeling assumptions, it was decided that the primary design parameters of interest are:

1. Blade solidity
2. Blade number
3. Blade wrap
4. Axial length
5. Diffusion factor
6. Cant angle
7. B_2-width
8. Exit blade angle
9. Head coefficient

Further review of these parameters indicated that three were fixed due to engine balance constraints or need to minimize changes to the tester. These are:

1. Head coefficient
2. Axial length (shroud contour)
3. B_2-width

With the above two parameters fixed, blade solidity, blade wrap, diffusion factor, and exit blade angle are all varied with change in blade number. This leaves blade number, and cant angle as the remaining parameters to study. Cant angle is most likely a second order affect on performance and was eliminated from the study. Although, cant could have a significant impact on structural design to meet increased tip speed.

The design parameter, which was held for further study, was the blade number. The following blade numbers were selected for further evaluation: 5+5, 6+6, and 8+8. Table 2 documents the final design parameters for each design.

Table 2 : Impeller Trade Study Design Parameters

Parameter	Blade Number		
	5+5	**6+6**	**8+8**
Head Coefficient	0.53	0.53	0.53
Exit Flow Coefficient	0.128	0.118	0.117
Diffusion Factor	0.80	0.60	0.43
Inlet Blade Angle, Degrees @ RMS	22	22	22
Inlet Blade Height, Inch	1.6	1.6	1.6
Tip Diameter, Inch	15.8	15.8	15.8
B_2-Width, Inch	0.58	0.58	0.58
Exit Blade Angle, Degrees	74	49	38
Total Blade Wrap, Degrees	52	98	120
Axial Length, Inch	2.08	2.08	2.08
W_2 / W_1 (Relative Velocity Ratio)	0.88	0.90	0.90

The impeller grid distribution is shown in table 3, with a typical grid shown in the meridional and blade-to-blade planes in figure 3.

Table 3: Impeller Grid Distribution

Zone ID	Nodes (Meridional x Radial x Blade-to-Blade)
1	7 x 11 x 33
2	11 x 11 x 29
3	17 x 11 x 13
4	17 x 11 x 13
5	5 x 29 x 33
6	23 x 11 x 33
7	33 x 5 x 33

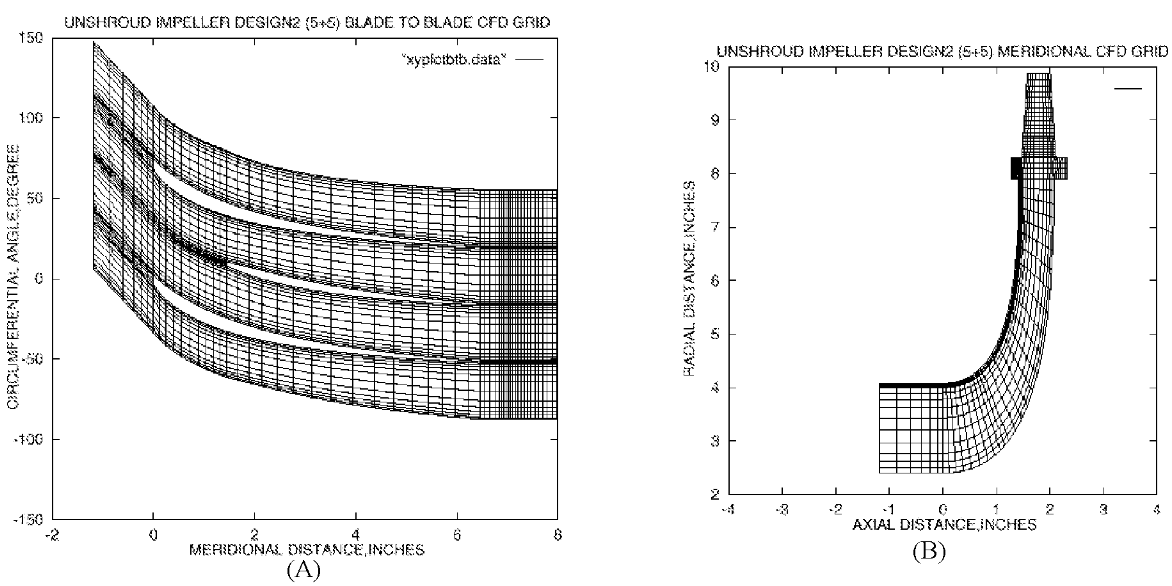

Figure 3: (A) Blade to blade plane, (B) Meridional Plane

PRELIMINARY RESULTS

The first impeller analyzed was a 7+7 configuration. Although this configuration is not part of the trade study space, the results are indicative of what to expect in terms of tip clearance impact. Table 4 lists the performance variables evaluated. Figure 4 shows the locations at which head and efficiency were calculated. These preliminary CFD results are consists with the J-2 Oxidizer pump open face and shrouded impeller test results, reference 3. The test report shows the impeller efficiency drops about 10 points between shrouded impeller and open face with axial clearance of 10% impeller discharge vane height.

Table 4 : Results of Shrouded and Unshrouded 7+7 Impeller Designs

	7+7 Unshrouded Impeller	7+7 Shrouded Impeller
Model Flow Rate, GPM	20,295	19,667
Euler Head (A-B), ft	128,486	141,379
Actual Head (A-B), ft	110,362	137,642
Efficiency (A-B)	0.86	0.974
Static Pressure Rise (A-B), psi	1975	2601
Euler Head(C-D), ft	141,014	153,169
Actual Head(C-D), ft	113,252	141,333
Efficiency (C-D)	0.8	0.92
Static Pressure Rise (C-D), psi	1976	2633
Flow Split (suction / pressure)	52% / 48%	51% / 49%
Leakage Flow, %	5.5%	N/A

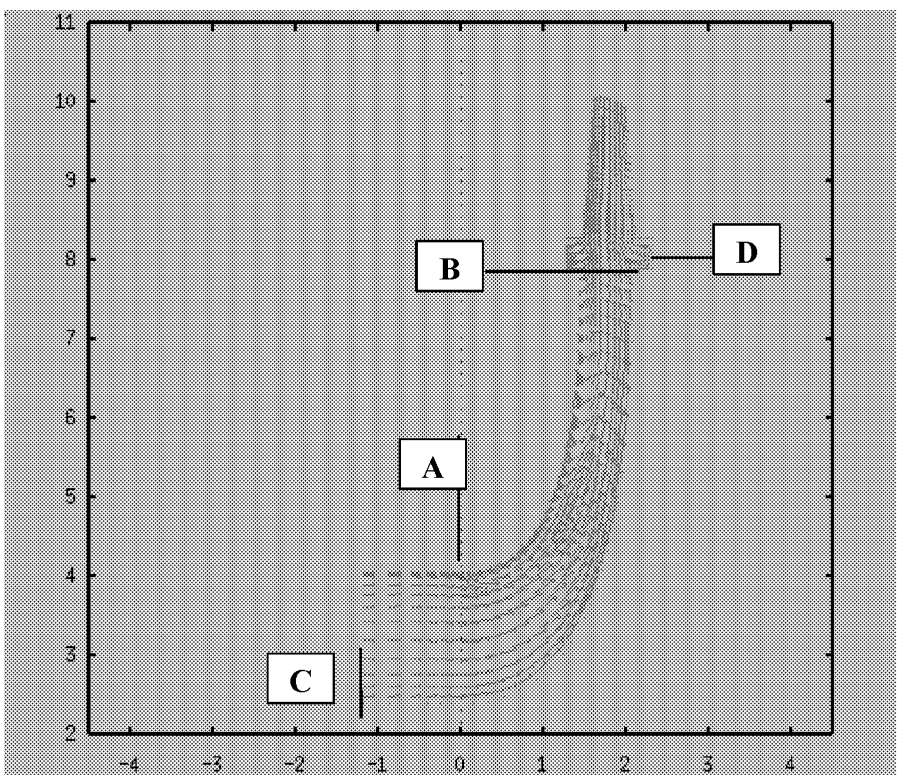

Figure 4 : Performance Calculation Planes

ROTORDYNAMIC ASSESSMENT

Stable turbomachinery operation depends on the damping of the rotor motion. Currently, rotordynamic stability parameters are estimated by using bulk flow theories and small perturbation (quasi-steady) assumptions. A well-established experience base with unshrouded impeller rotordynamic coefficients does not exist.

To help understand the unshrouded impeller's rotordynamic performance, Enigma's computational rotordynamic methodology was applied to the unshrouded impeller. This method directly simulates the rotor whirling motion (no quasi-steady assumptions) and can be, in principle, applied to large eccentricity whirl problems.

For Navier-Stokes based rotordynamic calculations, the impeller shaft/hub moves with an imposed whirling harmonic motion, figure 5, and the flow equations are integrated time-accurately until reaction force time periodicity is observed. The fluid reaction force vector time history is calculated; the force history can then be post-processed and decomposed into normal and tangential components. Because of the direct simulation of the moving hub, the flow model must consist of the complete three-dimensional geometry (full 360 degrees in circumference). A similar approach to access the rotordynamic fluid forces on seals has been fully described in reference 4.

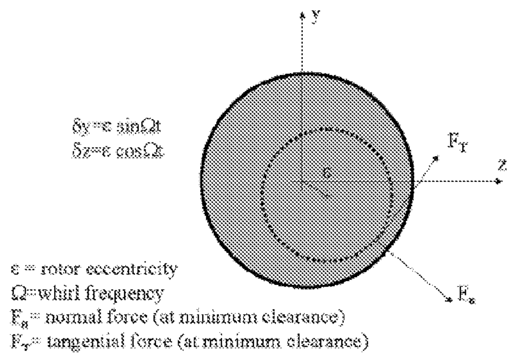

Figure 5: Whirling impeller rotor (hub)

Four whirl cases were computed: forward and backward synchronous, and forward and backward super synchronous. The effect of whirl ratio on the housing fluid forces are shown in figure 6. Using this calculation methodology rotordynamic coefficients can be supplied to the rotordynamics community to evaluate the impact on stability of the machine at all required operating points.

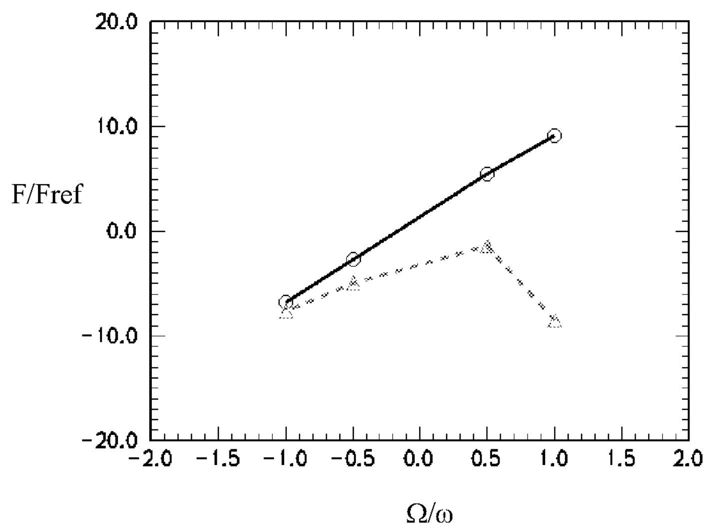

Figure 6: Computed normal and tangential forces

CONCLUSIONS

Performance degradation due to impeller tip clearance is well documented in literature. Two methods are available to combat this issue. Maintain tight tip clearances at all operating points or design an impeller with tip clearance insensitivity. The former is difficult to achieve in a high pressure turbopump due to housing deflections and material growth and shrink due operating speeds and cryogenic fluid temperatures. Literature review indicated that blade number variations could decrease tip clearance sensitivity. A trade study has been undertaken to evaluate blade number impact on performance with varying tip clearance. Rotordynamic assessment of turbopump stability is of great concern for rocket engine turbopumps. This is due to the inability to provide high levels of damping in the system at will. A method has been described for evaluating these forces with unshrouded impellers.

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