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A NOVEL QUASI-3D DESIGN METHOD FOR CENTRIFUGAL COMPRESSOR IMPELLER ON THE BLADE-TO-BLADE PLANE

Farzad Poursadegh
PhD. Student
poorsadegh@mech.sharif.edu

Ali Hajilouy-Benisi
Associate Professor
hajilouy@sharif.edu

Mahdi Nili-Ahmadabadi*
Assistant Professor
m.nili@cc.iut.ac.ir

School of Mech. Eng., Center of Excellence in Energy Conversion, Sharif Univ. of Tech.
*Mechanical Engineering Department, Isfahan Univ. of Tech.

ABSTRACT

In this research, a novel quasi-3D design method is developed for the centrifugal compressor impeller on the blade-to-blade plane. In this method, an iterative inverse design method called Ball-Spine Algorithm (BSA) is incorporated into the quasi-3D analysis code solving the Euler equations on the blade-to-blade and meridional planes at each shape modification step. In design procedure, the difference between the target and current pressure distribution along the suction or pressure sides of the impeller causes the blade-to-blade profile to be changed and the target pressure distribution to be satisfied.

In order to validate the quasi-3D analysis code, the centrifugal compressor of a gas turbine is investigated numerically using a full 3D Navier-Stokes analysis code. The meridional and blade-to-blade planes pressure distributions obtained from quasi-3D and 3D analysis codes are compared showing good agreement between them. Furthermore, the pressure ratio and efficiency of the centrifugal compressor is obtained by some experiments in which the flow parameters at the compressor inlet and outlet are measured. Comparison of 3D analysis results with the experimental results shows good agreements.

Finally, the current pressure distribution along the pressure side at 50% span is smoothed and considered as the target pressure distribution. The quasi-3D design procedure converges to a new profile after 400 modification steps. The designed impeller is numerically analyzed showing the flow pattern of the impeller is improved and the total to static efficiency of impeller increases by 0.64 percent and the total pressure ratio increased by 3.38 percent.

1 INTRODUCTION

In order to improve the performance of the modern highly loaded turbomachinery components, the designer needs a good understanding of the flow field in the blade passage. It is possible to design aerodynamically efficient blades by taking into account the important three-dimensional viscous effects in the designing process. For example one can design blade with smooth blade loading to reduce the shock losses or reduce the tendency of flow to separate by limiting the adverse pressure gradient [1].

These improvements in blade design can be achieved by using an inverse method which essentially generates geometries to perform the prescribed conditions.

One of the most important classifications of inverse methods is according to the approximations used by the methods. These methods could be either quasi 3-D or full 3-D design, inviscid or viscous flow [1]. The axisymmetric methods can be used with limited success in defining blade detailed geometry and controlling blade surface flow field properties [2]. Because of this uncertainty in controlling the properties on blade surfaces the focal point in development of inverse methods has been the blade-to-blade geometry definition and flow characteristics. In this kind of inverse method naturally there are two design quantities for characterizing the performance: either the flow properties on both sides of blade or the properties on one side (usually suction side) and the blade thickness distribution [1]. Many researchers used various strategies of design based on the flow properties distribution on the blade sides, such as, Zhengming [3] used velocity, Demeulenaere [4] and Giles [5] used static pressure and Leonard [6] used Mach number distribution. In all of these inverse methods, satisfying the flow tangency condition at the blade boundaries during the iteration causes the modification of geometry. Another approach used for modifying the blade geometry uses a purely mathematical algorithm, minimizing an error function that expresses the difference between the prescribed and the actual distributions obtained by solving the flow field in analysis mode which is known as optimization problem. This approach guarantees the improvement of the objective function [7, 8 and 9]. Thus in contrast to the inverse method where the designers experience in choosing the right pressure or velocity distribution is of key importance in obtaining a good design, the optimization method will always guarantee a better design but the physical sense of inverse methods and being temporally efficient, have made these methods more attractive and efficient [10, 11].

In this investigation, a quasi-3D code solving Euler equations on the both meridional and blade-to-blade planes of a centrifugal compressor is incorporated in BSA in order to design of blade angular profile.

Eliminating adverse pressure gradient on the suction and pressure sides of the blade of centrifugal impeller can be done by modifying the angular profile and reducing the tendency of flow to separate, it will cause the compressor efficiency to be improved. Having modified the pressure distribution along the pressure side of blade at mean radius, the blade angular profile is changed to satisfy the modified pressure distributions. The compressor with modified angular profile is numerically analyzed by a fully 3D viscose code and its performance from numerical analysis is compared with that of the existing compressor. Also, in order to validate the fully 3D and quasi-3D code, some experimental investigations are carried out at Gas Turbine Laboratory of Sharif University of Technology.

2 NOMENCLATURES

BSA	Ball Spine Algorithm
CP_rD	Current reduced Pressure Distribution
TP_rD	Target reduced Pressure Distribution
ΔP_i	Difference between TPD and CPD at each link
MVGE	Meridional velocity gradient equation
BVGE	Blade-to-Blade velocity gradient equation

3 QUASI-3D ANALYSIS

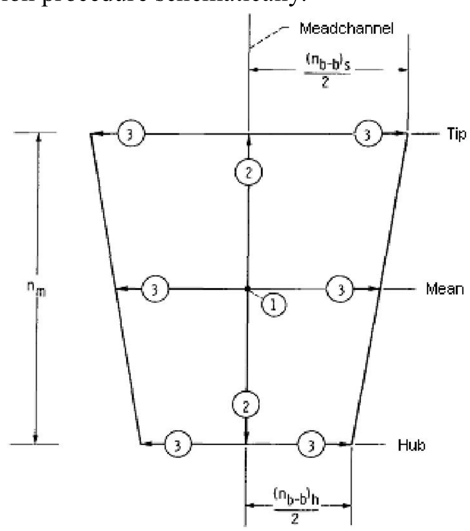
In this paper the quasi-3D analysis of the centrifugal compressor of a gas turbine is presented for its coupled meridional and blade-to-blade planes. For this purpose, it is necessary to specify the input parameters for the numerical code. The input quantities consist essentially of mass flow rate, rotational speed, number of blades, specific-heat ratio, inlet total temperature and density, gas constant, hub-to-shroud profile and mean blade shape [12].

Mass flow rate, rotational speed, inlet total temperature and density are obtained from experimental measurements at the compressor inlet in design condition. The code input parameters are specified in Table. 1.

Table1. Centrifugal compressor parameters

Parameter(unit)	Value
Operational condition	Design point
Number of impeller blades	25
Specific-heat ratio	1.4
Gas constant (J/(kg.K))	287
Inlet flow angle (°)	0.0
Inlet total temperature (°K)	322.2
Inlet total density (kg/m^3)	1.088

In order to determine the mean blade shape of the radial impeller, the angular coordinates of the mean blade shape is specified as a function of the axial coordinates, as shown in Fig.1. The impeller hub and shroud profiles are specified in Fig. 2. The solution procedure starts to discrete the velocity gradient equation for meridional plane with an initial guess based on 1-D flow assumption using stagnation conditions at the impeller inlet. Once meridional velocity components are specified, the numerical code starts to solve the blade-to-blade velocity gradient equation using meridional velocity distribution. Finally by integrating the mass flow rate over normal surfaces of the flow passage and modifying the initial guess to reach the computed mass flow rate equal to the input mass flow rate and satisfying mass continuity, after some iterations solution converges [12]. Figure 3 shows the quasi-3D solution procedure schematically.



(1) Estimate $W_{mid,mean}$
(2) Calculate W_{mid} from mean to hub and mean to tip from MVGE
(3) Calculate W from midchannel to suction and pressure surfaces at hub, mean and tip from BVGE
(4) Calculate total mass flow for channel cross section using continuity eq.
(5) Repeat steps 1 to 4 with new estimate for $W_{mid,mean}$ until either desired mass flow is obtained

Fig. 3 Calculation procedure chart [12]

Figure 4 shows the reduced pressure distribution along the suction and pressure sides of the blade at mean radius. As discussed in reference [11 and 13] the reduced pressure is the physical stress which is sensed from rotating passage wall as the static pressure is sensed from stationary passage wall. Therefore, the boundary layer thickness growth along the wall of rotating passage depends on the reduced pressure gradient. So, sharp adverse gradient of reduced pressure on the pressure or suction side of blade may cause the boundary layer thickness to grow abnormally.

In Fig.5 contours of static pressure are shown at the meridional and mean radius blade-to-blade planes. As expected, the static pressure increases through the impeller and from suction to pressure side of the passage. The contours of reduced pressure are shown in Fig. 6. Figure 7 shows the relative velocity contours, where the maximum relative velocity occurs at suction side of the blade near the shroud at the impeller inlet.

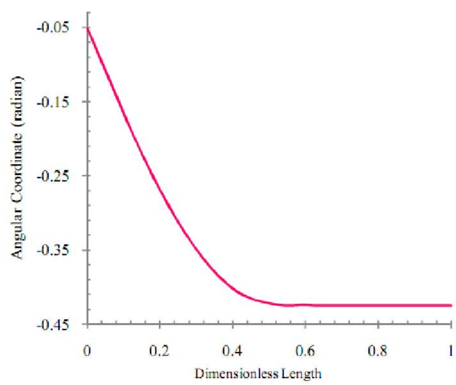


Fig.1 Mean blade shape of the impeller.

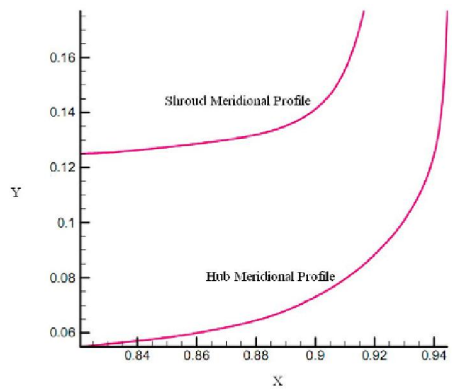


Fig.2 Hub and shroud profiles of the impeller.

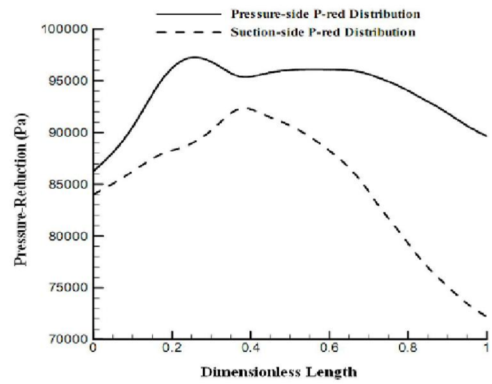


Fig.4 Reduced pressure distribution through blade surfaces at mean radius.

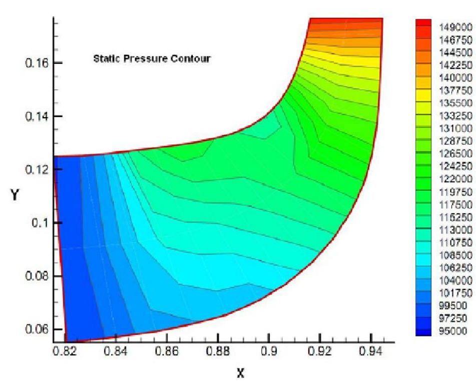


Fig.5-a. Contours of static pressure on the meridional plane.

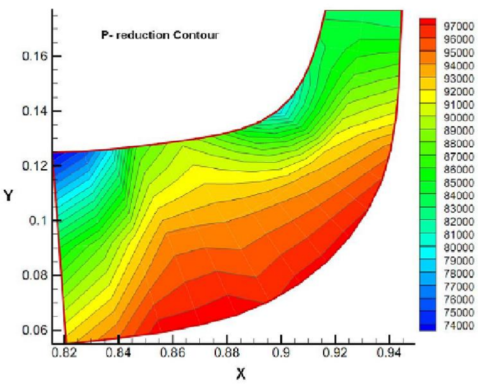


Fig.6-a. Contours of reduced pressure on the meridional plane.

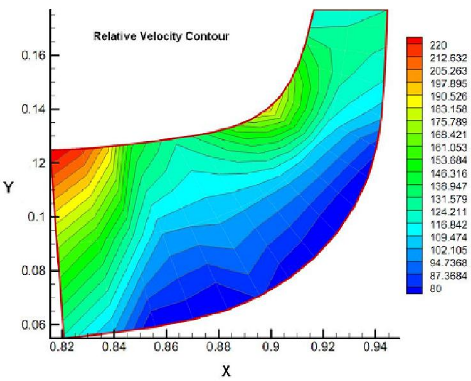


Fig.7-a. Contours of relative velocity on the meridional plane.

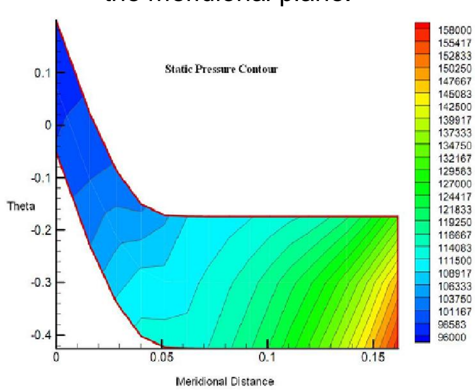


Fig.5-b. Contours of static pressure on the blade-to-blade plane at mean radius.

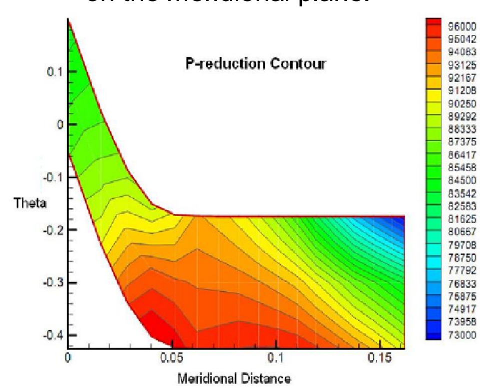


Fig.6-b. Contours of reduced pressure on the blade-to-blade plane at mean radius.

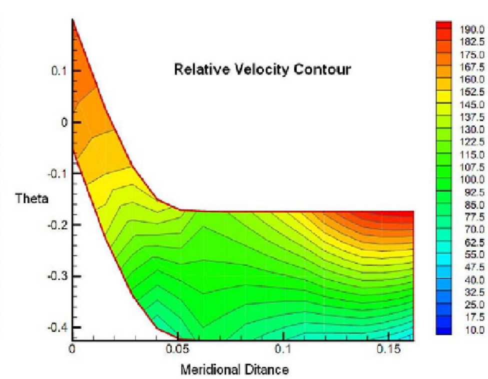


Fig.7-b. Contours of relative velocity on the blade-to-blade plane at mean radius.

4 INVERSE DESIGN METHOD

4.1 Fundamentals of Inverse Design Method

Fundamentals and benefits of inverse design algorithm called Ball-Spine Algorithm (BSA) which is used in this research are thoroughly explained by reference [11]. In this method it is assumed that the walls of flow passage are composed of a set of virtual balls that can freely move along the specified directions called spines which are the normal lines connecting the balls with the same x position on two walls. Passing fluid flow through the flexible passage causes a pressure distribution to be applied to the inner side of passage wall. By applying a target pressure distribution to the outer side of each passage wall, the force due to the difference between the target and current pressure distribution at each point of the wall is applied to each virtual ball and causes its movement [11]. As the target shape is obtained, this pressure difference logically vanishes. In duct inverse design problems, it is essential that a characteristic length to be fixed. The direction of spines depends on what characteristic length should be fixed. Therefore, for different passages, the spines are differently defined. Another constraint for wall modification is that one point of each wall should be fixed. Typically, the starting point of each wall is fixed [11].

In the blade-to-blade plane of the centrifugal compressor, meridional length at each radius should be fixed [13]. In such geometry, the arbitrary straight lines from pressure to suction side of blade are the best choice for the spines. In Fig. 8, the schematic of ball displacement along the spine for the blade-to-blade plane is shown. Similar to the other iterative inverse design methods, the flow field should be analyzed at each shape modification step. In this research, the quasi-3D analysis code is used as flow solver at each shape modification step, due to low computational costs and also good accuracy in flow field predicting.

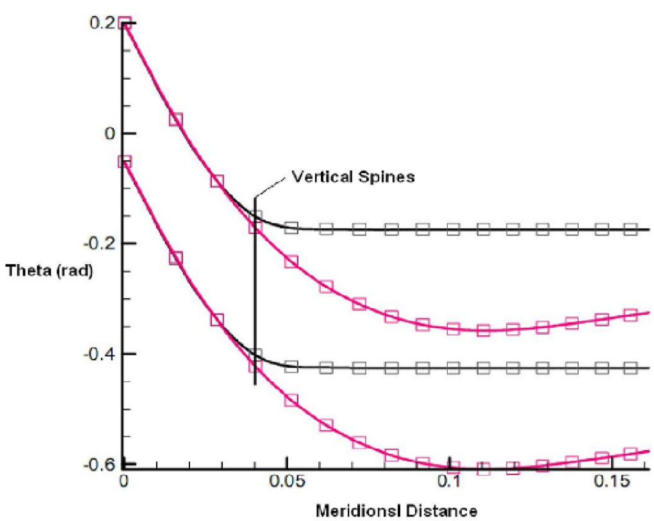


Fig.8 Schematic of ball displacement along spine

4.2 BSA Design Procedures

As discussed before BSA is typically incorporated in existing flow solution procedures. The computed pressure surfaces are normally obtained from partially converged numerical solutions of the flow equations. During the iterative design procedure, as the CP_rD approaches to the TP_rD, the force applied to the flexible wall gradually vanishes and at the final steps, the subsequent solutions of the flexible wall equations yield no changes in the passage surface coordinates.

5 COMPRESSOR EXPERIMENTAL FACILITIES

The measuring parameters and stations of experimental arrangement for the centrifugal compressor are shown in Fig. 9. The compressor consists of 25 radial element blades and 24 radial diffuser vanes. The measurement process is performed at four measurement stations included impeller inlet, shroud wall, impeller outlet and diffuser outlet. The rotational speed and mass flow rate are measured by a magnetic pick-up and a bell mouth at the gas turbine inlet, respectively. Overall, 11 static pressure taps, 6 stagnation pressure probes and 8 thermocouples are used for measuring the thermodynamic parameters of the fluid at different positions. The variation of total to total isentropic efficiency and total pressure ratio versus the normalized rotational speed or normalized mass parameter are extracted from the experiments.

In reference [11] validation of 3D numerical simulation results by the experimental performance results are thoroughly explained.

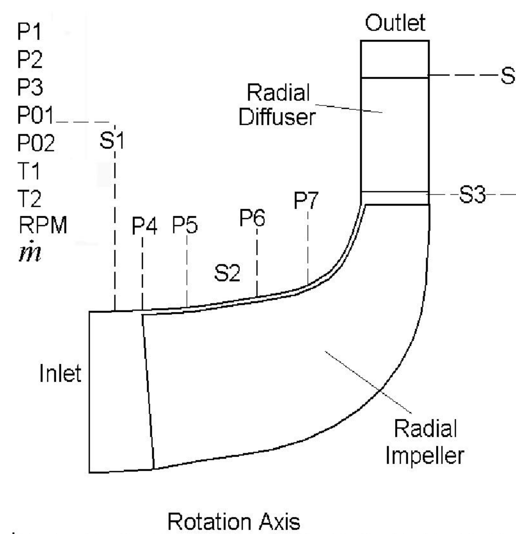


Fig.9 measuring parameters and stations of the centrifugal compressor of the gas turbine [11]

As discussed in section 3, the quasi-3D solver used in this research is based on inviscid flow regime and cannot estimate the performance. So the experimental pressure measurements at impeller shroud are used for validation of quasi-3D simulation results. Figure 10 shows good agreement between experimental and numerical predicted pressure distribution.

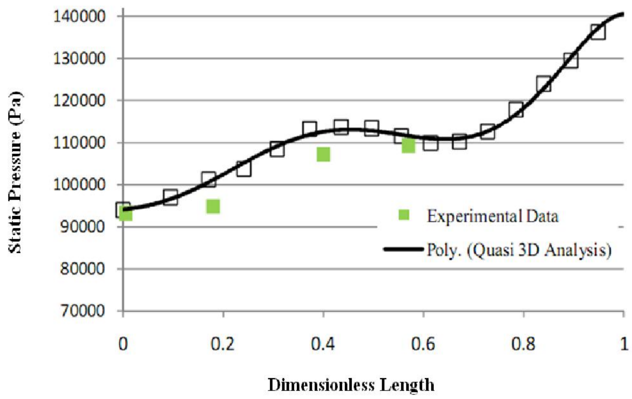


Fig.10 Pressure distribution results of the quasi-3D analysis and experimental measurements

6 FULLY 3D ANALYSIS OF THE COMPRESSOR

Blade angular profile modification is the final aim of this research. The quasi-3D analysis code, which is incorporated in the inverse design algorithm, is based on inviscid flow regime, therefore cannot estimate the performance. Thus, a fully three-dimensional viscous code is used to analyze the flow field of compressor and to validate the quasi-3D analysis results. Also, it can be used for performance prediction of the current and modified compressor.

6.1 Numerical Method and Boundary Conditions

In this analysis, structural elements are used for grid generation of impeller and diffuser. The finer grids are used for zones having sharp gradients such as adjacent to the walls and blades [11, 13]. The Reynolds-Averaged Navier-Stokes equations (RANS) which describe the conservation of mass, momentum and energy are solved by means of a finite volume method. The discretization of equations is done via a coupled implicit method in which the energy, momentum and mass equations are solved together. The Reynolds stress terms in momentum transport equations are resolved using the shear-stress transport (SST) turbulence model, developed to blend the robust and accurate formulation of the k-ω model in the near-wall region with the free-stream independence of the k-ε model in the far field [11].

Using the mixing plane interface model, computational domain is divided into stationary and moving zones and utilizes relative motion between the various zones to send calculated values between zones. To complete the model in rotating zones, the coriolis and centrifugal accelerations are added to the momentum equations. The mixing plane model on the other hand has the advantage that only one pitch of the impeller and diffuser has to be modeled [11]. The boundary conditions for this analysis include mass flow rate and total temperature at the inlet, average static pressure at the outlet, no slip condition for stationary walls, zero relative velocity respect to the rotating zone for rotating walls, mixing plane for interface between the impeller and diffuser and periodic boundary condition [11]. It is noticeable that the quantities of the boundary conditions are set from experimental measurements at the inlet and outlet of the compressor while the compressor works as a part of the gas turbine in its operation line.

In this analysis after checking for mesh independency, a number of 377608 elements is used to insure mesh independency.

6.2 Validations and Results

The 3D numerical analysis is accomplished in 0.73 nominal rotational speed. In this section, the validation of 3D results is accomplished with experimental results and then it is used to validate quasi-3D inviscid code results.

In Fig. 11 the comparison between 3D and quasi-3D pressure distribution at the shroud wall with experimental measurements is presented, which is shown good agreements between experimental and numerical results.

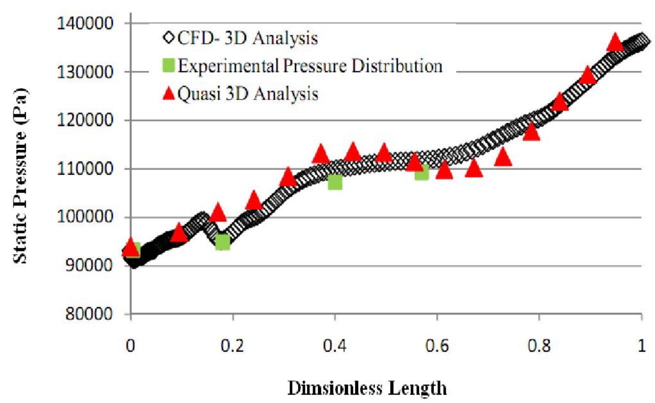


Fig.11 Pressure distribution results of the 3D, quasi-3D analysis and experimental measurements

In Fig. 12, the hub pressure distribution of the quasi-3D analysis is compared with that of the 3D numerical analysis. Although in the quasi-3D analysis the viscous and 3D flow field effects and clearance and blade thickness are ignored, its results well agree with the 3D simulation results. In Figs. 13 and 14 the static pressure distributions through blade surfaces of the quasi-3D analysis are compared with that of full 3D analysis. As shown in these figures, in spite of some discrepancies, there are good agreements between results trend. The deviations can be related to the flow incidence at leading edge and streamline slip at trailing edge.

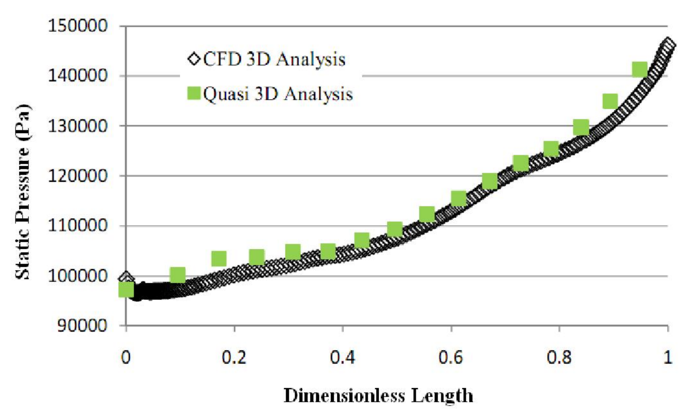


Fig.12 Pressure distribution on the hub due to the quasi 3D and 3D analysis

Because the losses due to the viscous and 3D flow field inside the impeller are not considered in the quasi-3D analysis, this deviation at the inlet and outlet regions can be justified.

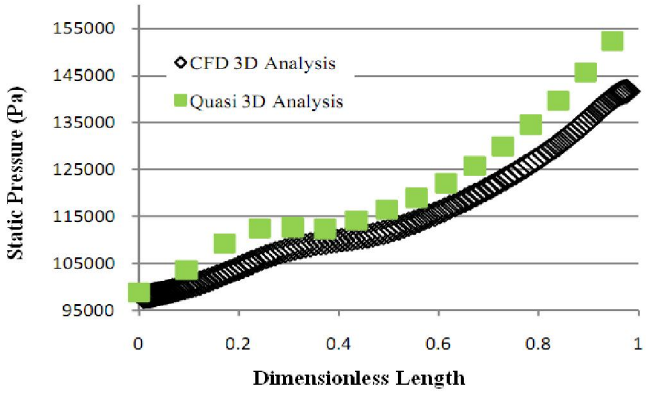


Fig.13 Pressure distribution on the blade pressure side at mean radius due to the quasi 3D and 3D analysis

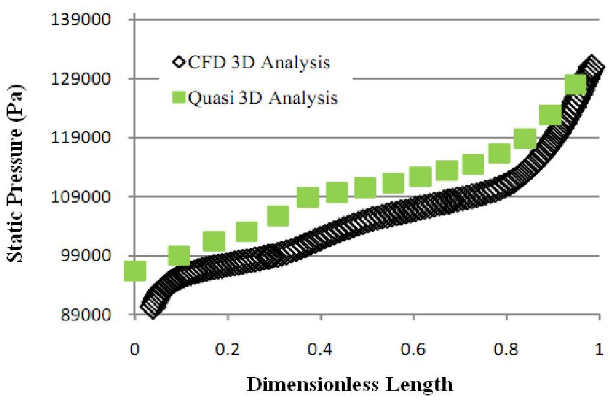


Fig.14 Pressure distribution on the blade suction side at mean radius due to the quasi 3D and 3D analysis

7 BLADE ANGULAR PROFILE MODIFICATION

7.1 Inverse Design Validation Test Case

After validating the quasi-3D analysis results, the BSA design algorithm is incorporated in the quasi-3D code to modify the blade angular profile by modifying the reduced pressure distribution along the blade sides. Firstly, it is necessary to validate the inverse design method, for this. The reduced pressure distribution along the pressure side in Fig. 15 is considered as our target reduced pressure distribution for the surface shape design. The method should converge to the shape shown in Fig.1 from an initial arbitrary shape. Initiating from an arbitrary pressure distribution, the design process is converged after 100 modification steps. Figure 16 shows the modification procedure result from the initial guess to the target shape.

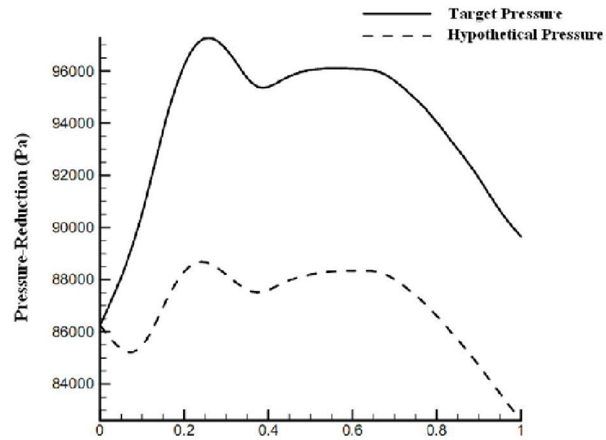


Fig.15 Target and Current Pressure Distribution for Validation Test Case

The iterations are stopped after the residuals are reduced by 3 orders of magnitude in which the residuals are defined as difference between the current and the target reduced pressures. After each geometry modification step, the quasi-3D analysis code is run until the residuals of the analysis code are reduced by 3 orders of magnitude. These convergence specifications for the design algorithm and analysis code are enough to confirm the required convergence so that the difference between calculated and target shape cannot be recognized. Any excessive decrement of the residuals just increases the computational cost and time [11].

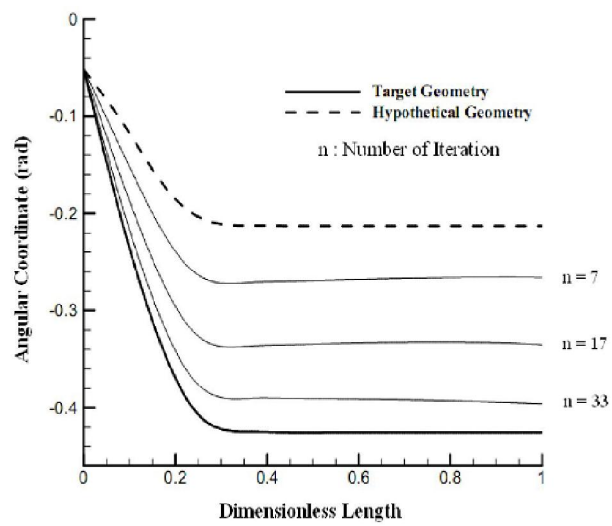


Fig.16 Wall shape modification procedure

7.2 BLADE ANGULAR PROFILE MODIFICATION

After validating the inverse design method, the angular profile of the current impeller at mean radius is obtained by modifying the reduced pressure distribution along the pressure side of the blade as shown in Figs. 17 and 18.

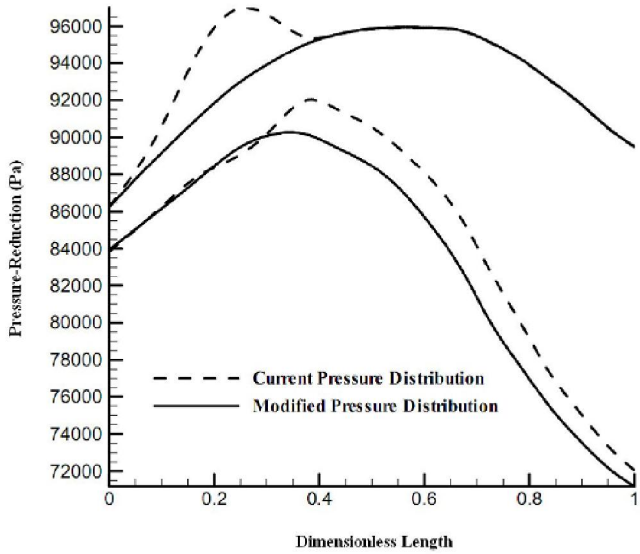


Fig.17 Current and modified pressure distribution

In Fig.18, after modifying the current reduced pressure distribution, the corresponding angular profile after 400 shape modifications is obtained. In this shape modification, sharp adverse pressure gradient along the pressure side is removed but, the inlet and outlet pressures on this side are not changed. It means that in this paper it is aimed to modify the pressure distribution which causes efficiency modification. In Figs. 19, 20 the flow field contours of the modified geometry are compared with that of the current geometry.

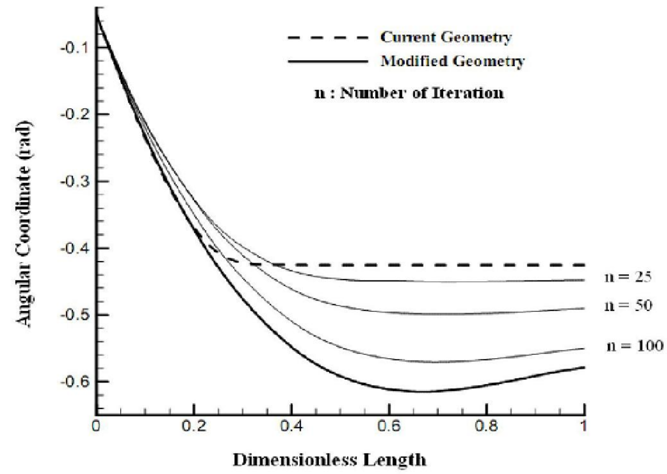
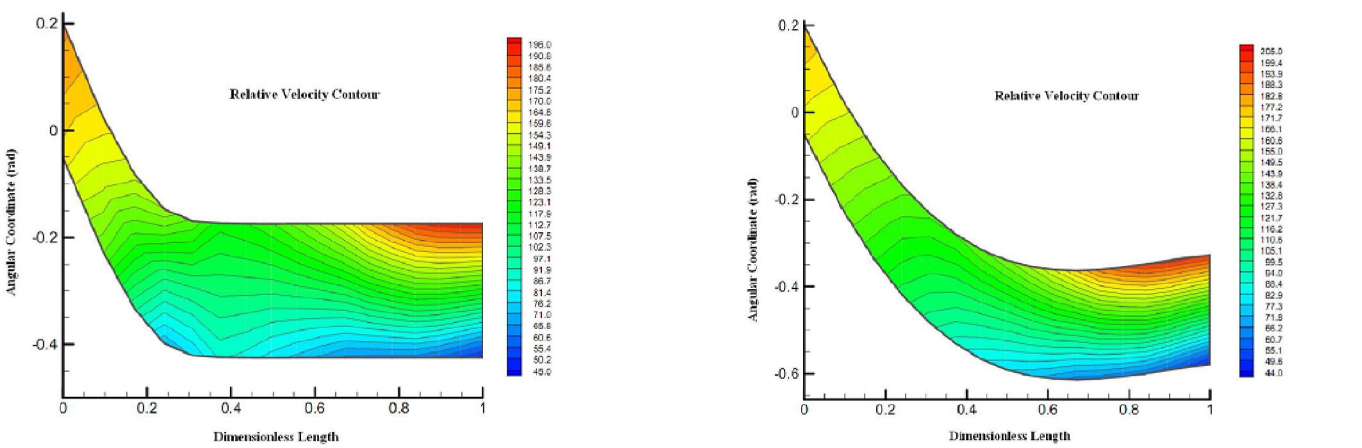


Fig.18 Current and modified blade angular profile

Figure 19 shows that the relative velocity decrement at the pressure side of the blade is improved by this shape modification. Figure 20 shows the static pressure contour on the blade-to-blade plane of the impeller, which is improved.

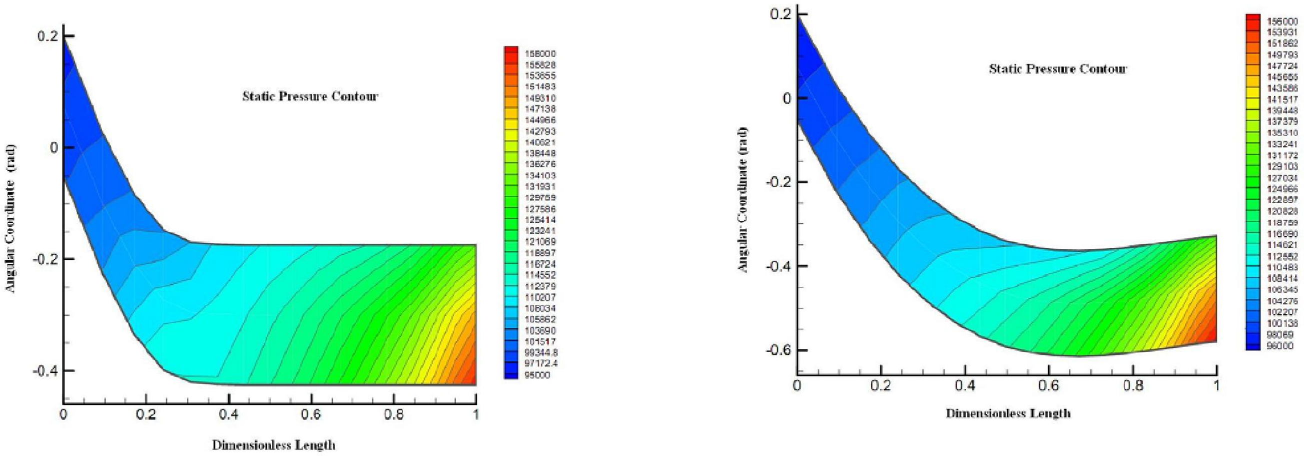
The 3D numerical analysis of the modified geometry with the current diffuser shows that the total to static efficiency is improved by 0.64 percent and total pressure ratio is increased by 3.38 percent. Increase in total pressure ratio is because of forward sweep shape of the blade which is proportionate to the modified pressure distribution. It is noticeable that, there is no difference between grid generation of the modified impeller and that of the current impeller in the 3D numerical analysis.



b) Modified Geometry

a) Existing Geometry

Fig.19 Relative velocity contour on the blade-to-blade plane of the impeller.



a) Existing Geometry b) Modified Geometry

Fig.20 Static pressure contour on the blade-to-blade plane of the impeller.

8 CONCLUSIONS

In this research, the BSA design procedure is incorporated in a quasi-3D analysis code for designing the angular profile of the centrifugal compressor impeller. For the convergence of the BSA in a rotating zone, the difference between the CP_rD and TP_rD should be applied to the flexible wall at each shape modification step. By eliminating useless positive pressure gradient on the pressure side of the blade as a consideration for TP_rD, the design procedure converges to the angular profile improving the compressor efficiency by 0.64 percent and total pressure ratio by 3.38 percent.

9 ACKNOWLEDGMENTS

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