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A STUDY OF 2-D TURBINE BLADE PROFILE TO MINIMIZE THE PRESSURE LOSS

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ABSTRACT

A two-dimensional axial-type turbine blade is optimized. Shape parameters are used to design a blade profile, and these shape parameters are employed as design variables for optimization. These consist of polynomial function for suction and pressure side, ellipse for leading edge and circle for trailing edge. As an objective function, the pitchwise averaged total pressure is selected at the 30% downstream of axial chord length from the trailing edge, which is the inlet location of next stage turbine blade. Aerodynamic, mechanical and geometry constraints are imposed to ensure that the optimized profile meets all engineering restrict conditions. Two-dimensional compressible flow analysis codes are applied and validated with the experimental results on the VKI turbine blade. A turbine blade profile for optimization is selected at the mean radius of turbine rotor using a heavy-duty gas turbine. 11 design variables are chosen for blade design. On the optimized turbine rotor, the total pressure reduction is 6%, which is same to the 0.6% total-to-total efficiency increase.

INTRODUCTION

The turbine efficiency is the most important factor on the performance of heavy duty gas turbines for power plants, air turbines, or turbo expanders etc.,. Losses in the turbine consist of mechanical losses due to rotating parts or bearings, tip clearance losses due to the tip gap, secondary flow losses due to curved passages, and profile losses due to the blade shape. More than 60% of total loss on one stage of turbine is generated by two latter loss mechanism. These losses could be reduced how to design a turbine blade profile. So, it needs to develop a new design technology for optimum turbine blade profile.

Blade profiles have been designed according to the inlet and exit conditions with considering the aerodynamic characteristics such as incidence or deviation angle as well as the structural characteristics. Many methods to design a blade profile have been developed such as a method using multi-polynomial [**1**], method using shape parameters [**2**], and inverse method [**3**] etc.,. Even blades have been designed according to the design point by any one among several methods, it is not known the relationship between the blade profile and its efficiency.

3-D turbine blade profile is usually designed by stacking 2-D blade profiles which are designed according to the flow conditions at several radial locations, therefore; the optimization of 2-D blade profile gives lots of effect to the efficiency of 3-D turbine blade. It was tried by using Bezier curves [**4**]. In this study, shape parameters, which can modify the blade profile directly, are applied to optimize a 2-D blade profile. This method has advantages such as it can be figured out the relationship between a blade profile and its efficiency directly because the blade profile is controlled by shape parameters.

It is selected the pitchwise area averaged total pressure at 30% axial chord downstream from the trailing edge as an objective function. It is maximized without losing blade loading and blade sectional area. This is same to minimize the total pressure loss in the passage.

NOMENCLATURE

A	blade sectional area
Cl	blade loading coefficient
Cp	pressure coefficient
Cx, Ct	axial and tangential chord
d_{te}	straight section of trailing edge
h	enthalpy
o	throat
P	pressure
Px, Py	peak point of pressure surface
P_t*	total pressure

Re	Reynolds number
r_{le}, r_{te}	leading and trailing edge radius
Sx, Sy	peak point of suction surface
v	absolute velocity
w	relative velocity
Y	total pressure loss coefficient
β_{in}, β_{out}	inlet and exit blade angle
ε_{up}, ε_{up}	inlet upper and lower wedge angle
η_{ux}, η_{uy}	half of major and minor axis of upper ellipse
η_{lx}, η_{ly}	half of major and minor axis of lower ellipse
φ	turning angle on pressure surface
κ, λ	leading and trailing edge turning angle
r	density
x	enthalpy loss coefficient
ζ	unguided turning angle

Subscripts

0	initial value
t	stagnation
∞	value at reference location
1,2,3	nozzle inlet, nozzle exit, rotor exit

SELECTION OF DESIGN VARIABLES

Optimum Design Variables

Shape parameters which are able to design general axial-type turbine blades are induced from a seriously twisted turbine blade. Fig. 1 show these parameters, which was good enough to design general axial type turbine blades [**5**].

Using many shape parameters could express the blade profile well and has a flexibility to control the profile locally, but it requires much computational time for optimization and makes the method of blade design complex. 11 shape parameters are chosen without reducing the accuracy of blade design basis on the shape parameters which have strong effect [**6**]. Shape functions for optimization do not need because the chosen design variables can change the shape of blade profile directly.

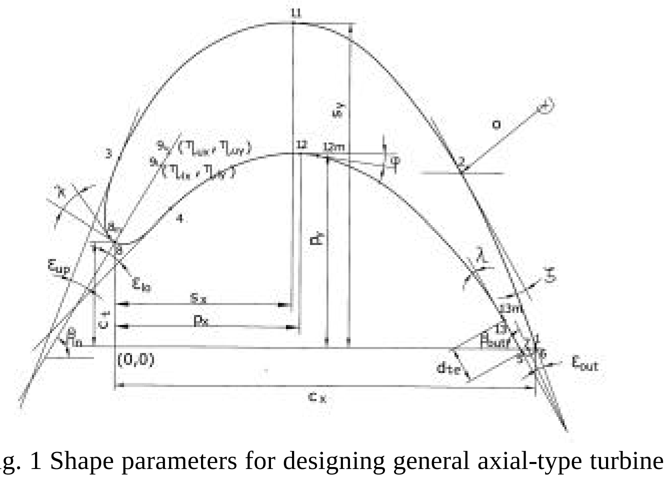


Fig. 1 Shape parameters for designing general axial-type turbine blades

Table 1 Shape parameters and design variables for blade optimization

Shape parameters	Design variables
ζ	q_2, q_1
ε_{up}	q_3, pt3 (x_3, y_3)
ε_{lo}	q_4, pt4 (x_4, y_4)
ε_{out}	q_5, q_1
η_{ux}, η_{uy}	q_3, pt3 (x_3, y_3)
η_{lx}, η_{ly}	q_4, pt4 (x_4, y_4)
o	q_2, pt2 (y_2)
c_t	pt8 (y_8)

To make a blade profile by using 11 design variables listed in table 1, ellipse from the leading edge(pt8) to pt3 on the suction surface, 3rd order polynomial from pt3 to the throat(pt2), and circle from the throat to the trailing edge(pt1) are employed. On the pressure surface, ellipse from the leading edge to pt4, 3rd order polynomial from pt4 to pt5, and circle at the trailing edge(pt5-pt1) are employed.

Fig. 2 shows redesigned blade profiles using design variables by the solid line. It shows some deviations between the original and redesigned blade profile at the fore part of suction surface. Standard deviation(σ) between the original and redesigned blade profile is calculated. σ is 0.45% on the suction surface and 0.17% on the pressure surface, so totally less than 0.33%. It is enough to start optimization because its shape is changed in the optimization process

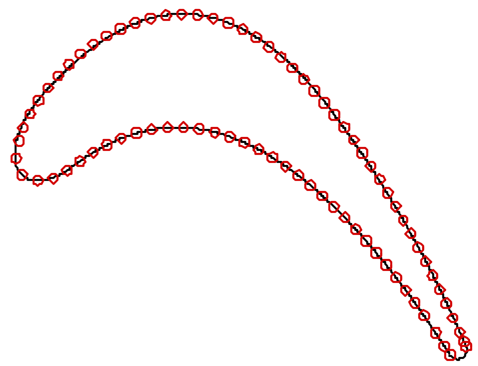


Fig. 2 Comparison of redesigned blade profiles using design variables

OPTIMIZATION AND FLOW ANALYSIS ALGORITHM

Objective Function and Constraints

We can choose the blade loading as an objective function on the turbine blade, but the increment of blade loading should accompany with the increment of blade area due to larger torsional stress. In this study, total pressure loss instead of blade loading is chosen as an objective function. Minimizing the total pressure loss is equivalent to maximize the pitchwise area averaged total pressure at 30% axial chord, which is approximately axial gap between blade to blade, downstream from the trailing edge. Constant total pressure is maintained at the inlet during optimization.

$$\text{Maximize ; obj} = P_t(\boldsymbol{X}) \text{ at } x=1.3Cx \qquad (1)$$
$$A \geq A_0$$
$$Cl \geq Cl_0$$

Constraints are applied to the initial blade sectional area and blade loading, and these values should not decrease than initial values in the optimization process. Another constraint comes from geometry conditions. Ellipse is applied to the shape of leading edge, and its shape depends on the inlet wedge angle. To use an ellipse as a blade profile, maximum and minimum wedge angles should be restricted not to generate a wiggled profile.

The initial blade sectional area and blade loading coefficient are 0.2133 and 1.467, respectively. The initial total pressure coefficient, which is the pitchwise area averaged at 30% axial chord downstream from the trailing edge and normalized by $\boldsymbol{r}_\infty U_\infty^2$, is 2.9037.

Optimization Algorithm

Optimization is a procedure to find design variables($\boldsymbol{X}$), which make the objective function to be optimized (maximization, minimization, or target), without violating given constraints. In this study, VisualDOC[**7**], which is developed as a commercial code by Vanderplaats, is used for blade optimization. MMFD(Modified Method of Feasible Direction) among many methods is applied with constraints.

Flow Analysis Scheme

Continuity equation, momentum equations, equation of state, and energy equation are used for compressible viscous flow analysis. For turbulent flow analysis, two-equation extended k-ε turbulence model [**8**] is applied with standard wall function. The difference between the two-equation standard k-ε turbulence model and extended k-ε turbulence model occurs on the dissipation rate equation. The extended k-ε turbulent model includes two time scales to allow the dissipation rate to respond to the mean strain more effectively than that of the standard k-ε turbulent model [**9**]. Computed results with the extended k-ε turbulent model were better than those with the standard k-ε turbulent model for complex turbulent flow problems [**8,9**].

The governing equations are transformed to the generalized coordinates and differentiated within the finite control volume. A 2^{nd} order central differencing scheme is applied to the convective terms with adaptive 2^{nd} and 4^{th} order dissipation terms. The viscous and source terms of the governing equations are discretized by 2^{nd} order central differencing scheme. A 2^{nd} order upwind scheme is employed for all scalar transport equations including turbulence modeling equations. A pressure based predictor/multi-corrector solution procedure is employed to ensure divergence free flowfield solutions at the end of each time marching step. A time centered Crank-Nicholson scheme is used for the temporal discretization. To solve the system of linear algebraic equations, an iterative ADI method is employed.

In the computational region, exit is selected at the far downstream not to be affected by the disturbance of passage flow and velocities at the exit are compensated to be constant mass flow rate. Computational blocks are inserted to avoid grid skewness and distortion at the leading and trailing edge. Inserted computational blocks improve the smoothness and orthogonality of grids. Fig. 3 shows the grids developed with multi-blocks on the turbine blade.

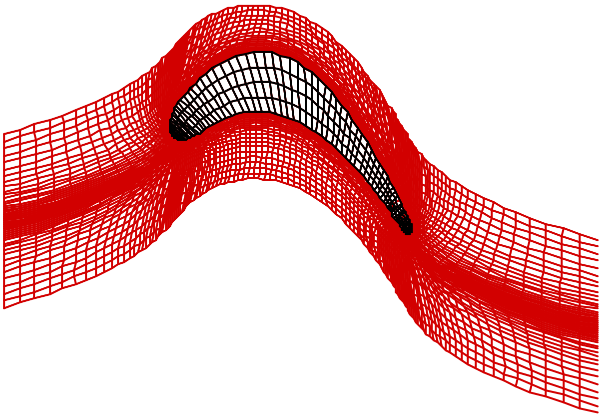


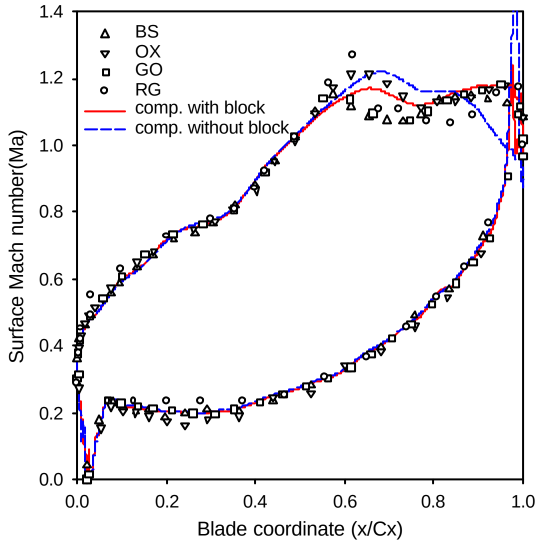
Fig. 3 Computational grids generated with multi-blocks within a turbine passage

RESULTS AND DISCUSSIONS

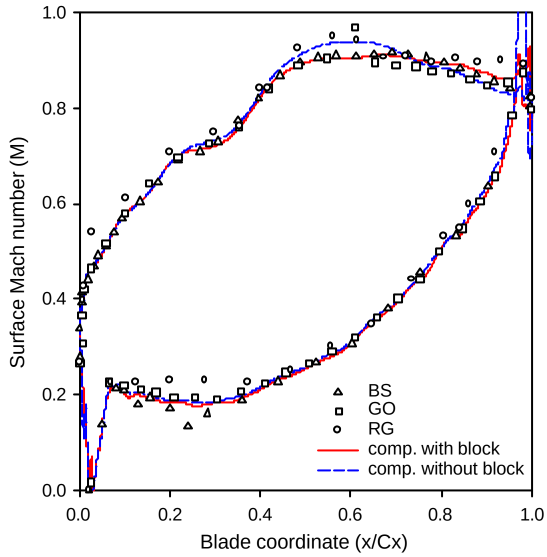
VKI Turbine Blade

Flow structures on the VKI turbine blade [**10**] are calculated in order to validate the flow analysis code. Flow conditions set equally to the experimental conditions, and inlet Mach number is 0.265. As a first case, the transonic flow is calculated. The exit Mach number is 0.97. Reynolds number and temperature at the inlet are 8.6×10^5 and 293K, respectively. 85x24 grids are employed. In a transonic flow condition, shock wave occurs at the throat and expansion wave occurs at the trailing edge. The computed results also show the shock and expansion wave as shown in experimental results from the density contours clearly.

Fig. 4 shows the surface Mach number, which is obtained from the relationship between the surface static pressure and total pressure. In the figures, computed results show that employing multi-blocks gives more stable and accurate computed results due to improve the orthogonality and smoothness of grids. Fig. 4(b) shows the surface Mach number compared with those tested in pure subsonic flow conditions. Reynolds number and exit Mach number are 7.8×10^5 and 0.78, which are same to the experiment conditions. The computed results agree well with the experimental results. As shown in Fig. 4(a), the results computed with multi-blocks are better than those with single computational block. From these results, the multi-blocks are employed in the computational region for the following turbine blade optimization.



(a) Transonic flow, Ma_{ex}=0.97 and Re=8.6x10^5



(b) Subsonic flow, Ma_{ex}=0.78 and Re=7.8x10^5

Fig. 4 Comparison of surface Mach numbers computed with multi-block and single block grid on the VKI turbine blade with the transonic or subsonic flow condition

Blade on a Heavy Duty Gas Turbine

A turbine rotor using on a heavy-duty gas turbine is selected and 2-D blade profile at the mean radius of turbine rotor is chosen for optimization. In the optimization process, boundary conditions at the inlet and exit set to operating conditions on the design point [**11**] which is shown in Table 2. Inlet Mach number is 0.5462 and Reynolds number is 1.74x10^6. In order to decide the number of grids without grid dependence, it is checked whether the surface pressure coefficient is converged or not according to the various grid number. The surface pressure coefficient is not changed even grids are increased more than 81x41. 101x51 grids are employed for sufficient grid independence. It is necessary 3800 iterations to obtain to 0.5x10^{-5} residual which is selected as the convergence condition. In this condition, the variation of objective function is less than 0.1x10^{-3} even reducing the residual by more iteration.

Table 2 Operating and flow condition at the inlet and outlet to optimize

Contents	Inlet	Outlet
Pitch (Pitch/Cx)	0.75	0.75
Velocity (m/sec)	U_∞=393.66	
Static temperature (K)	T_∞=1392.01	T_{out}=1366.33
Static pressure (Pa)	P_∞=741918	P_{out}=626509
Flow angle (degree)	b_{in}=61.28	b_{out}=61.84
Density (kg/m^3)	r_∞=1.86	

Axial chord of turbine blade is fixed in the optimization process. If the axial chord is not fixed, an objective function can not be decided as well as gas turbine has to modify seriously due to axial length change. So, it is allowed only the tangential chord length (Ct), which is shown in Fig. 1. It needs 45 iterations to reach the minimized total pressure loss in the passage without violating the given constraints. The blade sectional area and blade loading has no gain. Total pressure loss as an objective function is decreased to 6%.

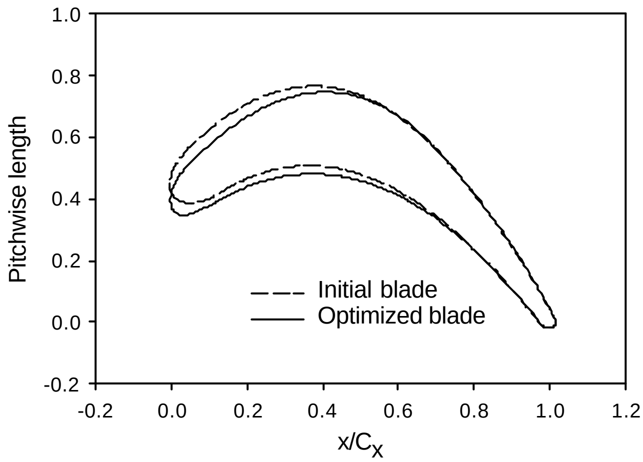
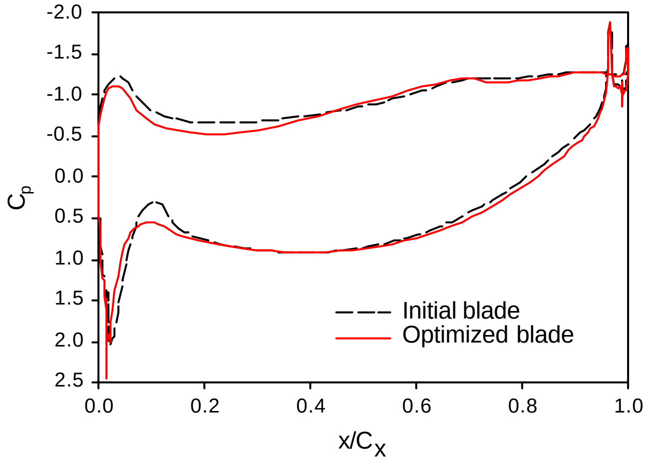
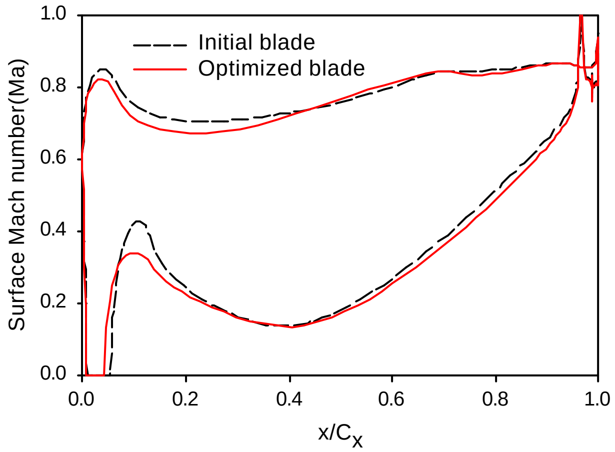


Fig. 5 Comparison of initial blade profile with optimized blade profiles

Fig. 5 shows the difference between the initial blade and optimized blade. The difference in the rear part of blade is negligible, but the fore part moves downward circum-ferentially. From the optimization, the tangential chord length at the leading edge is decreased and the size of fore part on the blade is a little reduced. The reduced size of fore part on the blade prevents strong static pressure drop generated by fast turning flow around the leading edge, and makes the static pressure drop smoothly along the passage.



(a) Surface pressure coefficient



(b) Surface Mach number

Fig. 6 Surface pressure coefficient and surface Mach number on the initial and optimized blade

Fig. 6(a) shows the surface pressure coefficient, and the phenomena mentioned previous paragraph are shown clearly. The small static pressure drop on the fore part of the blade makes the change of surface Mach number weak. Fig. 6(b) shows that the surface Mach number is reversely changed to the static pressure change shown in Fig. 6(a). However, the surface pressure and Mach number are related with the flow area change along the passage directly.

Area of turbine passage is calculated by the size of circle between the pressure and suction surface. Fig. 7 shows the radius of circle within the optimized blade. This figure shows that the area within the passage is expanded to the maximum area, and then reduced to the throat after passing to the peak point of suction surface smoothly. The area on the rear part of the blade is not changed. However, the changing rate of area on the optimized blade is small. This causes the reduction of total pressure loss.

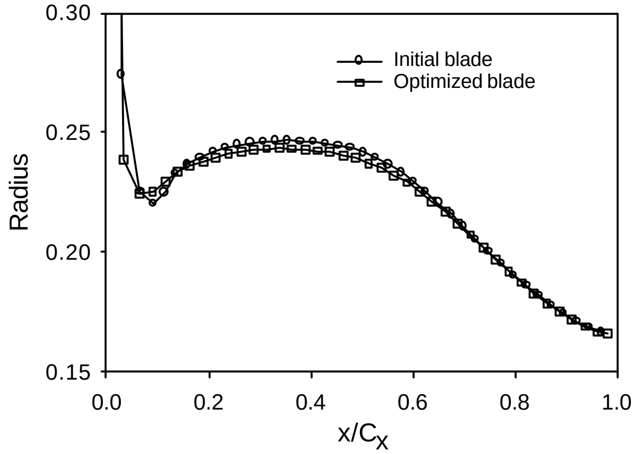


Fig. 7 Radius of circles between the pressure and suction surface within a passage to compare with the blade passage area

The reduction of total pressure loss at the exit is directly related with the efficiency increase. However, the efficiency is calculated from the enthalpy change. It needs the relationship between enthalpy loss coefficient and total pressure loss coefficient. In the incompressible flow, the total pressure loss coefficient could be simplified to $Y_N \cong x_N$. The relationship between Y_R and x_R is obtained by the same method as done on the nozzle. Total-to-total efficiency is calculated using enthalpy loss coefficient from equation(2) defined by Horlock [**12**] as follows;

$$h_{t-t} = \frac{1}{1+(x_R w_3^2 + x_N v_2^2)(1+\frac{g-1}{2} Ma_3^2)/2(h_{t1}-h_{t3})} \quad (2)$$

P_t^* is defined as $P_t/(r_\infty U_\infty^2)$, and it is related with Y_N as follows;

$$Y_N = \frac{P_{t1}-P_{t2}}{\frac{1}{2} r_\infty U_\infty^2} = 2(P_{t1}^* - P_{t2}^*) \quad (3)$$

In the computation, Y_N is simplified to x_N because Mach number is less than 1 even the flow is compressible. Before optimization, x_N is 0.346, but x_N is changed to 0.326 after optimization with constraints. That is 6% reduction of total pressure loss. In order to apply these values to the equation(2), the equation part of denominator assumes to the O(0.1) with assumption of general turbine efficiency. 6% reduction of total pressure loss is same to the 0.6% total-to total efficiency increase with considering the enthalpy change due to the decrease of total pressure loss. This is obtained on only one stage of turbine. That effect is increased with the number of turbine stages.

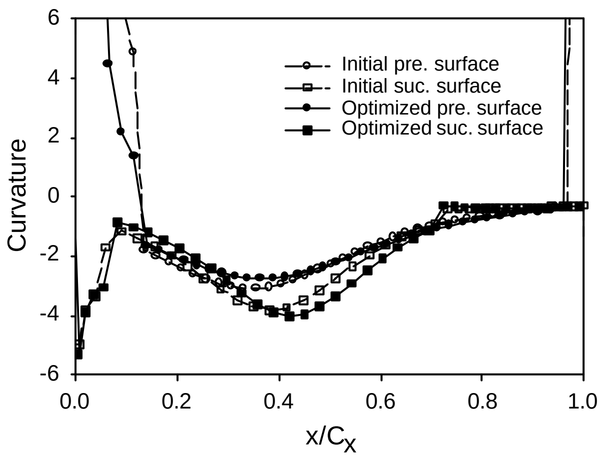


Fig. 8 Curvatures on the pressure and suction surface of blades

Fig. 8 shows the curvature of blade surface. The quick change of blade curvature causes the flow separation and pressure loss [**13**]. It is important to design blade profile smooth. Constant curvature is generated on the rear part of suction surface because the circle is applied to that area. The changing rate of curvature on the optimized blade is smooth. Even the changing rate of curvature on the optimized blade is increased on the suction surface compared with the initial one, it is not steep but smoothly changed in a whole region.

CONCLUSIONS

Blade is designed using shape parameters, and optimized. The axial chord of blade is fixed, and 11 design variables are employed for optimization. It is selected as an objective function the pitchwise area averaged total pressure at the 30% axial chord downstream from trailing edge, which is the inlet location of next turbine blade. Without reducing initial blade sectional area and blade loading, 6% of total pressure loss in the passage is reduced. This is same to 0.6% increase of total-to-total efficiency on the one stage of turbine. The efficiency is increased with the number of turbine stage.

The size of fore part on the optimized turbine blade is a little reduced and the circumferential length of the leading edge is decreased. The optimized blade shape is not changed from the initial one in the rear part of blade i.e., from the throat to the trailing edge. The changing rate of blade passage area on the optimized blade is small compared with that on the initial one. This causes the reduction of profile loss and total pressure loss.

In the actual application, The optimization of 3-D blade is necessary. The method using shape parameters will expand to 3-D blade.

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