THEORETICAL AND EXPERIMENTAL INVESTIGATIONS ON DOLPHIN TURBINE BLADE PROFILE

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ABSTRACT

An important and one of the most labor-intensive stages in the designing of turbine stages is their profiling. The effect of the blade shape on the aerodynamic performance has become an important turbomachinery design consideration. In order to assess this effect, an improved flow cascade design has been developed. So, an experimental and theoretical research program concerning the aerodynamic effect of blade shape on the flow characteristics through turbine blade channels was carried out in this work. A time marching technique has been used for the design and analysis of dolphin turbine blade profile. A fair agreement between the predictions of the time marching technique and the measurements of the experimental work for dolphin turbine blade profile validates it as an economic and practical approach to axial flow turbines.

The results of this particular configuration can be used to give an idea about the dolphin blade losses and inlet flow angle and also give some guidance for future research work in this area. The results present the influence of the Mach number on the pressure distributions, velocity distribution and on the overall loss coefficient produced by a dolphin turbine blade profile. Also the results indicate the effect of dolphin configuration on secondary losses. The dolphin configuration leads to a decrease in secondary losses by about 8.7 % than the normal profile at 0.6 exit Mach number. The exit Mach number was varied from 0.2 to 1.2 in numerical analysis and from 0.2 to 0.6 in the experimental work.

Keywords: Turbine Blade Cascade, Blade Configuration, Blade Losses, Numerical Simulation, Turbomachinery.

INTRODUCTION

Recent years have seen intensive development of computational fluid dynamics and increasing application to problems in turbomachinery. The reasons for this are two-fold. Firstly, the ever-increasing cost of high quality experimentation means that it is very cost-effective to study a variety of configurations by numerical simulation and select just a few promising designs for rig testing. Secondly, the numerical simulation of the flowfield can be used to provide real insight and understanding leading to improved design concepts.

One of the main steps in the aerodynamic design of turbomachines is the determi-nation of the profile shape of the blades starting from the desired inlet and outlet conditions. Besides the experimental approach, different numerical approaches are possible depending on the flow model assumed: the analysis and design can be fully three-dimensional or the different blade sections can be investigated separately based on a two-dimensional flow in an approximately axisymmetric stream surface, the blade-to-blade surface.

Most applications made use of the two-dimensional blade-to-blade approximation. The equations may be solved in either finite difference or finite volume form. Specialized numerical techniques are needed to ensure stability of the integration of the equations through time until a steady state solution is reached. Among the first attempts to solve these problems is the integral method, first proposed by McDonald [1] and then modified by Denton [2]. This approach is a finite-element one in overcoming the difficulties associated with complex geometeries together with the inherent simplicity of finite differences. In the finite volume form of the Denton's method [2], the equations are regarded as equations for conservation of mass, energy, and momentum applied to a set of interlocking control volumes formed by a grid in the physical plane. When solved in this way it is easier to ensure conservation of mass and momentum than in the differential approach but similar numerical schemes are necessary to ensure stability. Time marching solutions of the Euler equations are now very widely used for calculation of flow through turbomachinery blade rows [3]. The resulting method is simpler, faster, and more accurate. Boiko and Kozhevnikov [4] studied theoretically the potential cascade flow of ideal incompressible fluid. The aim of their proposed study was to design an optimal profile of the cascade from the viewpoint of minimum profile losses with assigned geometrical characteristics, which ensures design parameters of the flow at the outlet and meets the strength requirements. Examples, of the improved approach to the streamline curvature method are given in [5] and [6]. The transonic cascade flow is calculated with an efficient and flexible Galerkin finite element method applied to the full potential equation in artificial compressibility [7]. While automated procedures for correcting the shape of the blades, based on a correlation between the velocity of flow and the curvature of the blade that ensures weaker shock waves and shorter diffuser sections, are discussed in Reference [8]. A new straight cascade code was developed. The 2-D or quasi-2-D/quasi-3-D forms of the Euler equation are solved, [9]. The results of an experimental study on the effect of trailing edge cutting off on the aerodynamic performance of lightly cambered S-profiles are reported in Reference 10. Experiments were carried out for both forward and revised flow conditions. The profile chord is cut by 3, 6, and 9 percent of the chord at the sharp trailing edge end and the performances of these profiles are compared. References 11 and 12 deals with the effect of surface curvature distribution on turbine cascade performance. The influence of the leading edge geometry on blade profile losses are discussed in Reference 13.

In the present work a time marching technique and experimental research program has been used for turbomachinery cascades for the design and analysis of dolphin turbine blade profile concerning the aerodynamic effect of blade shape on the flow in turbine bladings. The Mach number was varied from 0.2 to 1.2 in numerical analysis and from 0.2 to 0.6 in the experimental work. The result presents the influences of the exit Mach number on the pressure distributions, velocity distributions and on the overall loss coefficient produced by a dolphin turbine blade profile. The results also indicate the effect of dolphin configuration on secondary losses and the effect of the degree of expansion on the total losses. The results of this particular configuration can be used to give an idea about the order of magnitude of the dolphin blade on both the losses and inlet flow angle and give some guidance for future research work in this area.

THE NUMERICAL METHOD

Computational methods are becoming important tools for the development of advanced turbomachine components and may be used to screen new designs before running costly experiments. Both the time marching and the streamline curvature methods solve the flow in turbomachinery. The throughflow design problem is essentially one of determining the velocity distribution from annulus geometry and thermodynamic properties such as total temperature and total pressure. This gives the velocity distribution needed to choose the blading profile. The development problem is one of determining the losses in order to decide on the necessary blade profile. In this work, the time marching technique is used to analyze the flow through Dolphin turbine blade cascade.

Time Marching Method

Time-marching solutions of the Euler equations are now widely used for calculation of flow through turbomachinery blade rows [3]. The basic principle of time marching is to start with a guessed flow distribution and integrate the time-dependent equations of motion and energy forward with time until a steady-state solution is obtained.

The Governing Equations

The inviscid flows to be considered are governed by the Euler equations. Figure 1 shows the details of the finit element. For unsteady two- dimensional flows through blade channels, these equations can be written in the following conservation forms for a control volume, ΔV, over a time step, Δt, to give

Continuity $\Delta\rho = \Sigma_n (\rho \mathbf{V}.\overline{\mathbf{dA}})\, \Delta t/\Delta V$ (1)

x-momentum

$\Delta(\rho V_x) = \Sigma_n (P dA_x + \rho\, V_x \mathbf{V}.\overline{\mathbf{dA}})\, \Delta t/\Delta V$ (2)

y- momentum

$\Delta(\rho V_y) = \Sigma_n (P dA_y + \rho\, V_y \mathbf{V}.\overline{\mathbf{dA}})\, \Delta t/\Delta V$ (3)

Energy $\Delta(\rho E) = \Sigma_n (\rho H \mathbf{V}.\overline{\mathbf{dA}})\, \Delta t/\Delta V$ (4)

where $\overline{\mathbf{dA}}$ is a vector representing the area of the face of the element in the direction of the inwards normal to the face and the summations are over the n faces of the element, (see Figure 1).

These equations must be solved in conjunction with the perfect gas relationships.

$H = C_p T + 0.5 V^2$ (5)

where H is the stagnation enthalpy, and V is the velocity. In two dimensional flow, it is usual to replace the energy equation by the assumption H = constant. The system is completed by the equation of state,

$P = \rho RT = \rho R [(H - 0.5 V^2) / C_p]$ (6)

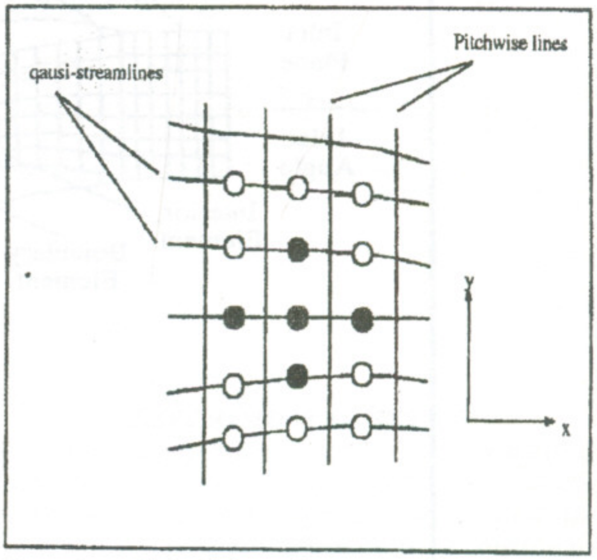

Figure 1 Finite Volume element

Finite Volume

The finite volume elements used for the scheme are shown in Figure 1. It is formed by a series of quasi-streamlines which are evenly spaced in the *Y*-direction and by pitchwise lines which need not be evenly spaced in the *X*-direction. The control volumes overlap in the pitchwise direction and calculating points are located at the center of each element. The quasi-streamlines upstream and downstream of the passage are extended approximately one chord with the flow inlet and outlet directions. Cusps are placed at the leading and trailing edges of the blade to prevent discontinuities in the grid.

Periodicity is applied over the bounding quasi-streamlines including cusp as shown in Figure 2.

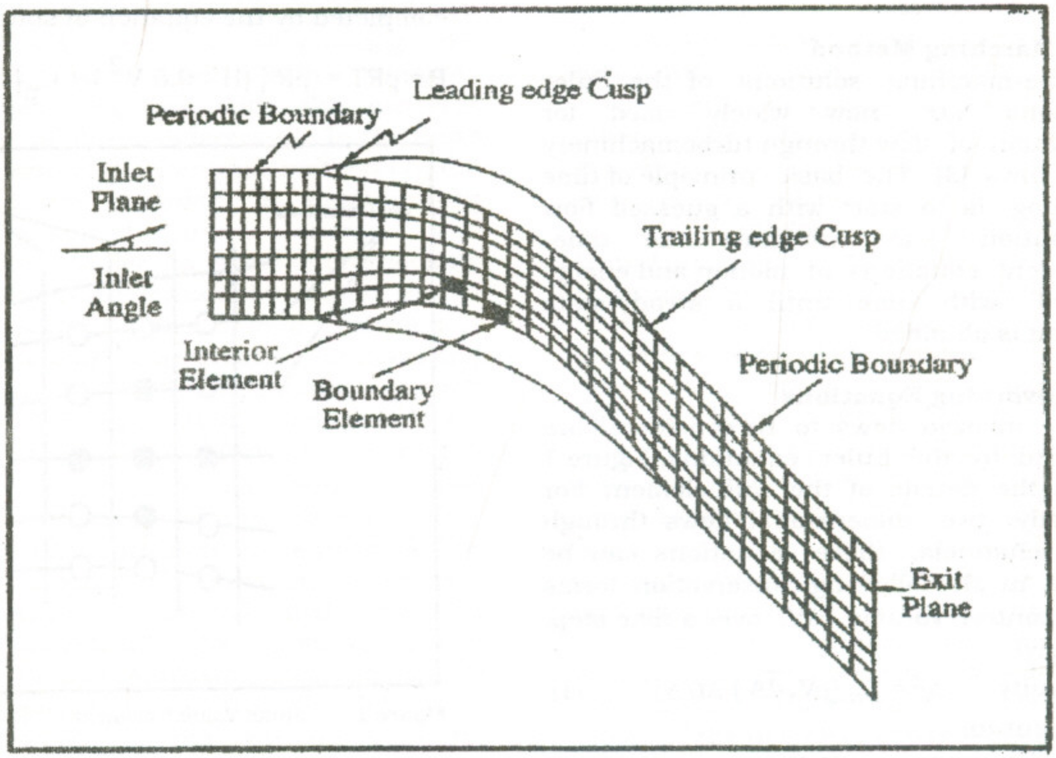

Figure 2 Cascade geometry and computational grid

Boundary Conditions and Numerical Procedure

The boundary conditions determine the nature of the flow and by keeping them constant, a steady state solution is obtained. The boundary conditions applied at the downstream boundary are specified as a uniform static pressure on the last pitchwise line and a condition of zero velocity gradient along the quasi-streamlines. At the upstream boundary the stagnation pressure and temperature and flow direction are specified and there is assumed to be no pressure gradient along the quasi-streamlines. The static pressure on the first pitchwise line is taken to be the same as that calculated on the same quasi-streamlines at the second pitchwise line. This static pressure is used in conjunction with an assumption of isentropic flow from the stagnation conditions to calculate the density and velocity.

The procedure adopted is to update the density at all grid points then use the new density in conjunction with the old velocities to obtain the pressure at each grid point, equation (5). The periodicity condition is applied on the pressure term of the streamline boundaries outside the blade passage. For cascade analysis, the periodicity condition is applied by averaging the mass and momentum fluxes on the streamlines boundaries outside the blade passage. Finally, the new pressures together with the old densities and velocities are used to update the momentum fluxes ρV_x and ρV_y by applying a momentum balance Over the control volumes faces. At the end of each time step, the slip condition is applied on the solid boundaries, where the velocity is aligned with the blade surface and then the momentum fluxes are corrected accordingly. The analysis starts from an initial guess of the flow pattern and solve the unsteady continuity, momentum and energy equations for the evolution of the flow in time until a steady state solution is reached. More details of the numerical method are found in Reference 3.

The inviscid flow solution together with the momentum integral equation of boundary layer can be used to obtain the friction loss coefficient. The momentum thickness on the blade surfaces can be evaluated from the following equation, [14].

$$\frac{\delta^{**}}{t} = \frac{0.036}{R_e^{\,0.2}}\left[\int_0^{s/t}\left(\frac{w}{w_2}\right)^{3.86}\frac{ds}{t}\right]^{0.8} \quad (7)$$

The friction loss coefficient and the trailing edge loss are given by

$$\zeta_f = \frac{2\left(\delta_{ss}^{**} + \delta^{**}ps\right)}{t\sin\alpha_2} \quad (8)$$

$$\zeta_e = K\,\frac{d_2}{t\,\sin\,\alpha_2} \quad (9)$$

where K is a proportionality factor and equals to 0.2.

In a more general analysis of losses it is suggested by Chen and Dixon [15] that, the following secondary loss correlation could be used in turbine blade loss predictions. The new correlation of τ_s including loading, aspect ratio, and inlet boundary layer velocity profile parameters is proposed, i.e.,

$$\tau_s^{*} = \tau_1 + K_1\,Z\,\frac{\cos\alpha_2}{\cos\alpha_1}\left(\frac{C}{h}\right) + \frac{2\delta_2/h}{(3m+1)\left(\frac{1}{2m}+\frac{1}{2}-\frac{\delta_2}{h}\right)} \quad (10)$$

where K_1 = 0.0055, m = 0.1428, δ_2/h = 0.379 (l/h) Re $^{-0.2}$,

$\tau_1 = Y_1 \cos^2\alpha_2 / \cos^2\alpha_1$, $Z = [\,2\,(\tan\alpha_1 - \tan\alpha_2)]^2 \cos^2\alpha_2 / \cos\alpha_m$ and

$$Y_1 = \frac{2\frac{\delta_1^{*}}{h}}{1.42\left(0.5-\frac{\delta_1^{*}}{h}\right)} \quad (11)$$

EXPERIMENTAL SET-UP

The arrangement of the experimental set-up is shown in Figure 3. The blade cascade consists of seven blades, and the blade profiles were fabricated according to the calculated results. The main geometrical parameters of the blade N (90-22) which is taken from Reference 14, are given in Table 1.

Table 1 Blade Geometrical parameters

c	$\bar{l}$	$\bar{t}$	α_0	α_1
45	1	0.75	90	22

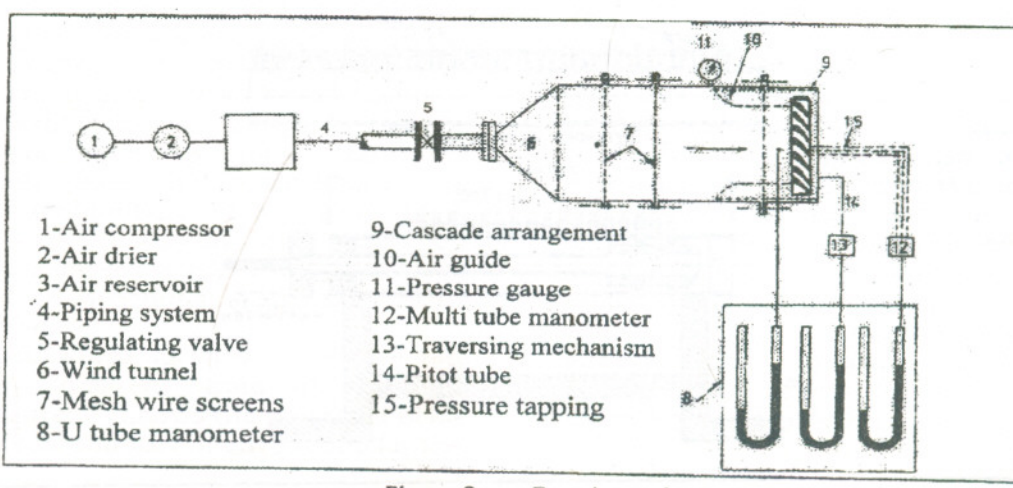

Figure 3 Experimental set up

A dolphin shape was made on the blade beginning from the leading edge to 9 percent of the blade chord. The idea is to introduce local changes without substantially affecting the remaining part of the profile. Four cases of the dolphin shape profiles shown in Figure 4, were studied and the performances of these profiles were compared.

In order to determine the pressure distributions on the pressure and suction surfaces of the blade profile, ten pressure taps set on each side at the mid height of the two central blades and are connected by rubber tubes through selector mechanism to a multi U-tube manometer. The selector mechanism shown in Figure shown 5, was used to measure the static pressure at both sides of the blade profile. This mechanism consists of a hollow cylinder which has 20 holes along its length. From measured pressure distribution, the pressure coefficient, Cp, was calculated as:

$$C_p = \frac{P - P_\infty}{\frac{1}{2}\rho U^2} \qquad (12)$$

The flow parameters were measured using a three hole pressure probe. To control and adjust the position of the three-hole probe along the test section, a three dimensional traverse mechanism was used.

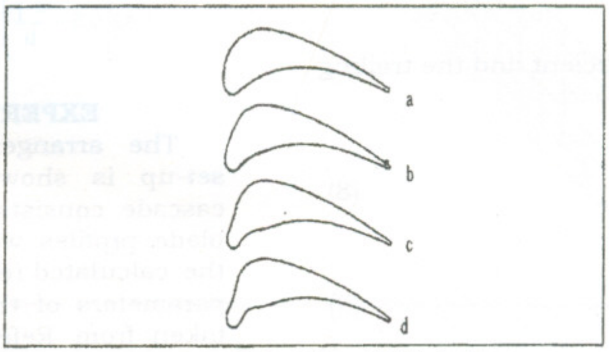

Figure 4 Dolphin shape

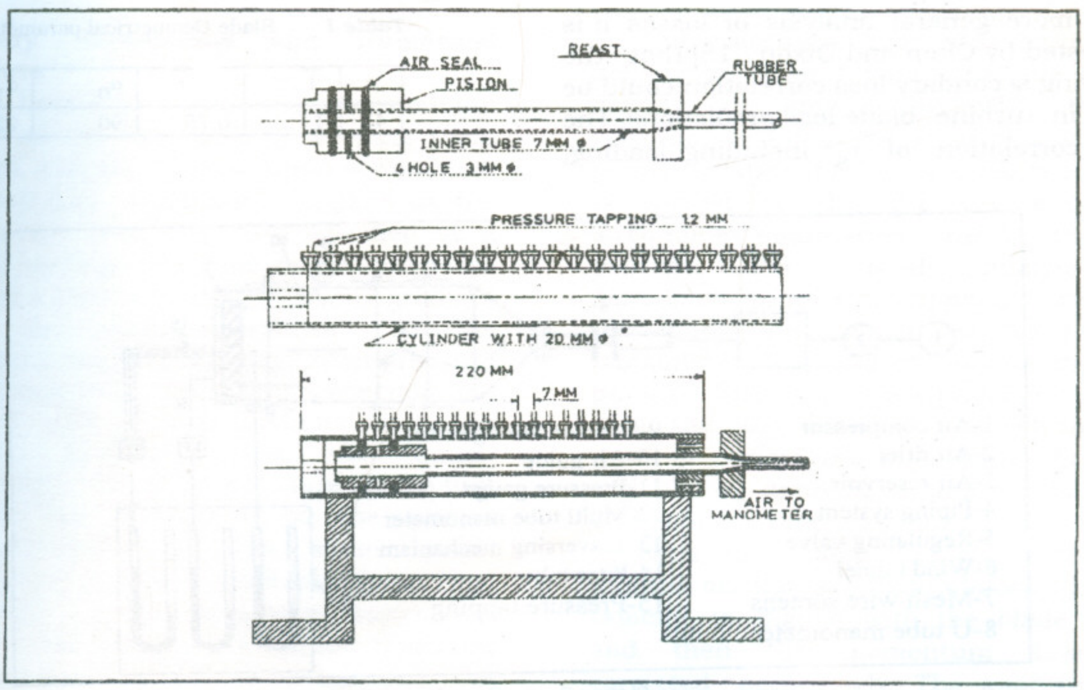

Figure 5 Selector mechanism

The cascade loss coefficient is obtained by plotting the average loss coefficient against the blade height of tested blade cascade. The secondary losses are determined by subtracting the measured midspan averaged loss coefficient from the passage averaged loss coefficient.

The inlet endwall boundary layer profile was determined by traversing a total pressure probe 200 mm upstream of the leading edges of the middle blade. The integral parameters of the inlet boundary layer are presented in Table 3.

Table 3 Inlet boundary layer data

$\bar{\delta}$	$\bar{\delta}^*$	$\bar{\delta}^{**}$	H
0.138	0.0174	0.0135	1.288

The experiments showed that the error in the mean velocity measurements by the three-hole probe is about ± 1% at the maximum calibration velocity at M = 0.6. The error in the static pressure measurements is about ± 0.5 %. The uncertainty in the experimental results were calculated using Kline and Mcclintock technique [16].

RESULTS AND DISCUSSION

The aerodynamic shape optimization of gas turbine blading is tested theoretically and experimentally in order to evaluate its efficiency and performance. The design problem aims at reshaping the leading edge of the baseline profile N(90-22) in order to minimize the losses. A certain preferred velocity distribution along the contour of the blade, ensuring minimum friction and cascade losses, is the basis for calculations. A real modification of the blade cascade which meets these requirements may differ substantially from the optimal one. The optimum combination of free geometrical parameters of the cascade ensuring a low level of blade losses is selected with account taken of restrictions on the blade configuration.

The last shape, as shown in Figure 4, is appreciably different from the baseline profile. Figure 6 shows the predicted blade surface Mach number distributions for the different blade configurations using the time marching technique (T. M. T). It is seen from this figure that, the flow on the suction side turned to the accelerated flow as it approaches the trailing edge. This may be due to the change in the effective flow area, which influences the velocity distribution on the suction side than on the pressure side. It is also noticed from these figures that, the value and position of the maximum Mach number are different for the tested blade configurations. For configurations (a) and (c), the maximum Mach number occurs at a distance of 0.5 times the chord downstream of the leading edge on the suction surface. For the other configurations, the maximum Mach number occurs at about 0.4 times the blade chord. This means that, the Mach number distribution is affected by the shape of the leading edge and so by the shape of the blade channel. There is a remarkable observation that can be seen from Figure 6. The Mach number on the pressure surface near to the leading edge of configuration (c) is shown to be higher than that on the suction surface at the same location. This may be due to the shape of the leading edge of configuration (c) which causes the boundary layer to separate and formation of vortices.

Figure 7 indicates the variation of the predicted cascade total loss with exit Mach number for different degrees of expansions (F) of blade channels. The expansion degree of the blade channel is defined as the ratio between the inlet channel width to the exit width, $F = a_1/a_2$. Four different values of the degree of expansion (F) are tested, namely F = 0.45, 0.48, 0.58 and 0.62. Generally, it can be noticed that, as the exit Mach number increases, the total losses are decreased. It is also observed from this figure that, the configuration (c) which has a degree of expansion of 0.48 gives the lowest total losses. This means that the degree of expansion affects strongly the cascade total losses. This is due to the streamlined shape of the leading edge (dolphin shape) which affects the flow characteristics on the suction and pressure surfaces and consequently affects the pattern of the boundary layer growth and its separation.

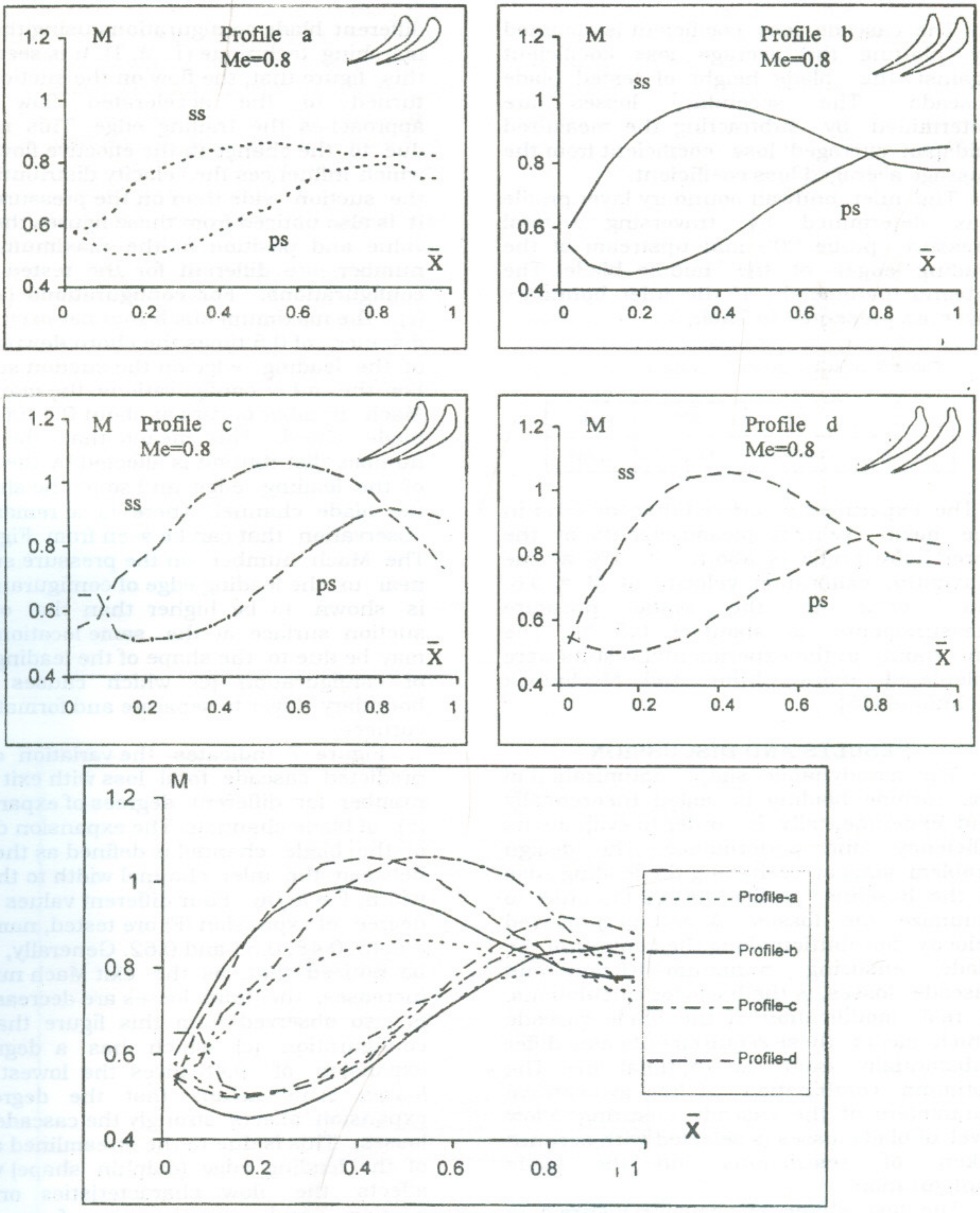

Figure 6 Mach number distribution along suction and pressure surface Me=0.8

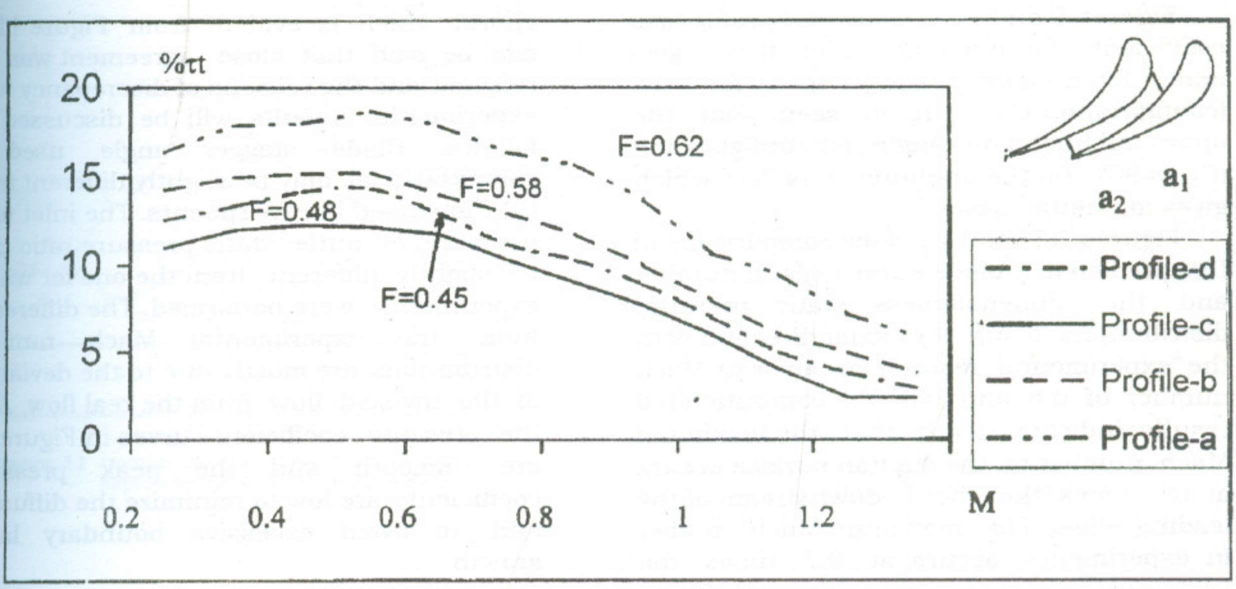

Figure 7 Effect of expansion degree on the total loss for different exit mach number

Figure 8 shows the predicted profile loss coefficient for different blade configurations with exit Mach numbers. From this figure, it can be seen that the profile loss coefficient increases with the decrease of exit Mach number. This is due to the suction surface of a blade being more prone to boundary layer separation. The separation depends besides the blade profile on factors like the degree of turbulence and Mach number. Also, an increase of M is accompanied with a decrease in flow density, which causes a decease in the boundary layer thickness. All these effects lead to a decrease in the profile loss coefficient. It can be observed from predicted results that, configuration (C) gives minimum losses and this configuration was fabricated and investigated experimentally and the results were compared with the theoretical results.

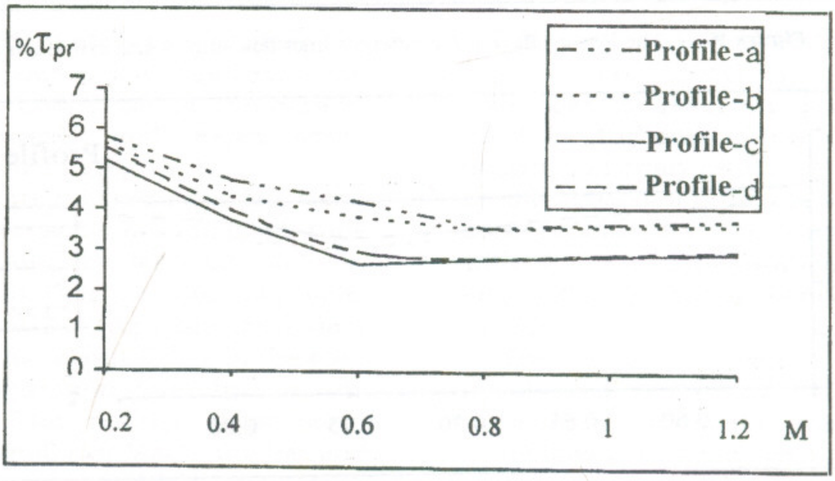

Figure 8 The loss coefficient for different blade configuration at different exit Mach number

Figures 9 and 10 indicate the profile loss coefficient for different inlet flow angles and Pitch-chord ratio ($\bar{t}$) for the (configuration-C). It is seen that the optimum inlet flow angle for configuration (C) is 90° and the optimum $\bar{t}$ is 0.75 which gives minimum losses.

Figures 11 and 12 show comparisons of the predicted blade surface Mach number and the dimensionless static pressure distributions along the axial direction with the experimental results for an exit Mach number of 0.8 and 0.6. The computational results indicate clearly that the maximum Mach number on the suction surface occurs at 0.5 times the chord downstream of the leading edge. The maximum Mach number in experiments occurs at 0.7 times the chord, which is evident from Figure 11. It can be said that close agreement was not reached and the reasons of discrepancy with experimental results will be discussed as follows. Blade stagger angle used in computational may be slightly different from the one used in experiments. The inlet total pressure to outlet static pressure ratio may be slightly different from the one for which experiments were performed. The difference from the experimental Mach number distributions are mostly due to the deviation of the inviscid flow from the real flow. Also the pressure coefficient curves in Figure 12 are smooth and the peak pressure coefficients are low to minimize the diffusion and to avoid excessive boundary layer growth.

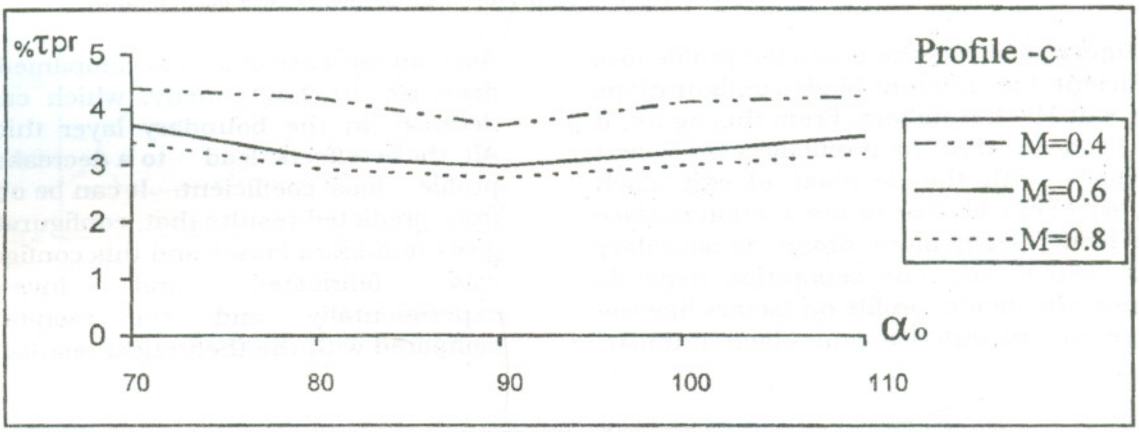

Figure 9 The loss coefficient for different inlet flow angles and exit Mach number

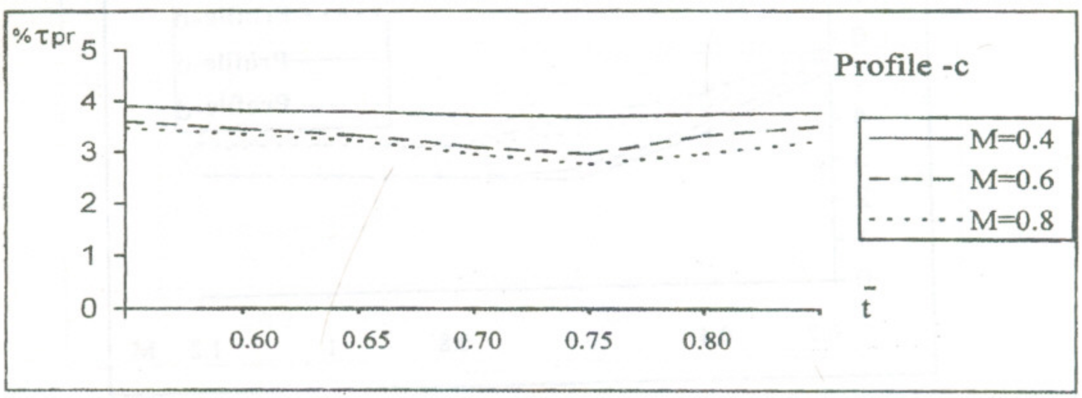

Figure 10 The loss coefficient for different $\bar{t}$ and exit Mach number

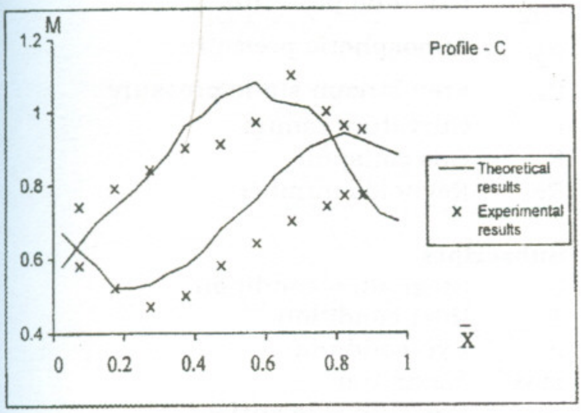

Figure 11 Comparison between theoretical and experimental velocity distribution

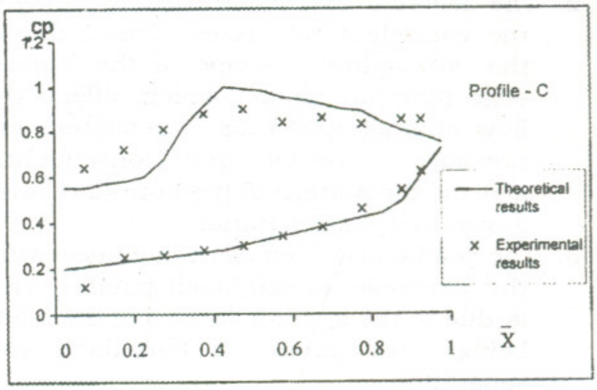

Figure 12 Comparison between theoretical and experimental velocity distribution

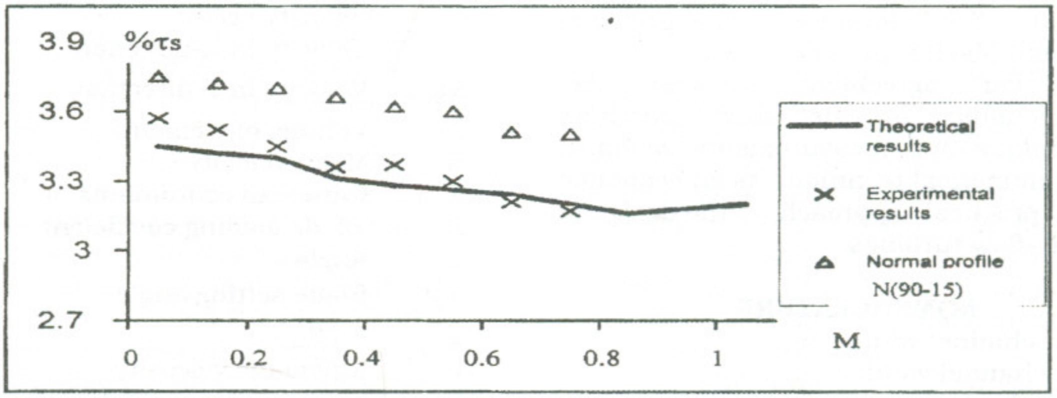

Figure 13 Variation of secondary losses with exit Mach number

Tests were conducted to establish the influence of the exit Mach number on the blade secondary losses and a comparison between theoretical and experimental results are presented in Figure 13.

From the figure it is seen that the trend and behavior is the same but the magnitude of secondary loss is less for dolphin configuration than that for normal profile. For example at M2 = 0.6, the secondary losses decrease by about 8.7 % in the case of dolphin configuration than that in the normal profile. The results show some discrepancy at small exit Mach number and a fairly agreement at high exit Mach numbers. Also the results indicate that the dolphin configuration decreases the secondary losses.

CONCLUSIONS

A time marching technique and experimental research program has been used for turbomachinery cascades for the design and analysis of dolphin turbine blade profile. Based on the above studies on the aerodynamic characteristics of the profiles with different configurations of the leading edge, the following conclusions can be drawn.

1. The Mach number distribution is affected by the shape of the leading edge and hence by the shape of the blade channel. This may be due to the shape of the leading edge of configuration (c) which causes the boundary layer to separate and leads to the formation of vortices.

2. The degree of expansions affects strongly the cascade total losses. This is due to the streamlined shape of the leading edge (dolphin shape) which affects the flow characteristics on the suction and pressure surfaces and consequently affects the pattern of the boundary layer growth and its separation.
3. The profile loss coefficient increases with the decrease of exit Mach number. This is due to the suction surface of the blade being more prone to boundary layer separation.
4. Within the tested range, the optimum inlet angle for configuration (C) is 90° and the optimum $\bar{t}$ is 0.75 which gives minimum losses.
5. The dolphin configuration leads to a decrease in the secondary losses by about 8.7 % than the normal profile at 0.6 exit Mach number.
6. The fair agreement between the predictions of the time marching technique and measurements validates the numerical technique as an economic and practical approach to the design of axial -flow turbines.

NOMENCLATURE

a_1	inlet channel width
a_2	exit channel width
a^*	Critical velocity
A	Area vector of face of element
c	Chord
Cp	Pressure coefficient
c_p	Specific heat capacity at constant pressure
c_v	Specific heat capacity at constant volume
d	Diameter
E	Specific internal energy
F	Degree of expansion = a_1/a_2
G	Mass flow rate
H	Specific stagnation enthalpy or the shape factor of boundary layer
I	Streamwise grid point number
k	Specific heat ratio
h	Blade height
M	Mach number
p	Static pressure
p_{01}	Inlet total pressure
p_{02}	Exit total pressure
p_2	Atmospheric pressure
P_∞	Free-stream static pressure
r	Curvature radius
R	Gas constant
Re	Reynolds number

Subscripts

0	Stagnation condition
1	Inlet condition
2	Exit condition
max	Maximum
ps	Pressure side surface
ss	Suction side surface
s	Surface
S	length of the blade surface
t	Blade pitch or time
T	Temperature
V	Velocity vector
V_x	Velocity in X-direction
V_y	Velocity in Y-direction
ΔV	Volume of element
w	Mean velocity
X, Y	Cartesian coordinates
Z	Blade loading coefficient
α	Angle
α_y	Blade setting angle
ξ	P_2/P_1
ν	Kinematic viscosity
δ	boundary layer thickness
δ^*	Displacement thickness
δ^{**}	Momentum thickness
τ_{pr}	Profile loss
τ_s	Secondary loss
τ_t	Total loss
ζf	Friction loss coefficient
ζe	Trailing edge loss
ρ	Density
L.E	Leading edge
T.E	Trailing edge
U	Free-stream velocity

Dimensionless groups

$\bar{t} = t/c$ Pitch – chord ratio

$\bar{x} = x/c$ Axial distance- chord ratio

$\bar{l}$ Aspect ratio (l/c)

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دراسة نظرية وعملية على ريش تربينات ذات بروفيل شكل الدولفين

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ملخص البحث

يتناول البحث دراسة نظرية وعملية لدراسة خصائص السريان خلال ريش تربينات ذات بروفيل يأخذ شكل الدولفين وذلك للعمل على تقليل مفاقيد ريش التربينات. تمت الدراسة على اساس اختيار بروفيل عادى لريش التربينات تم اجراء تعديلات عليه على أن يصل شكل البروفيل الى شكل الدولفين. تقدم الدراسة النظرية نموذجا لدراسة خصائص السريان من حيث حساب توزيع السرعات والضغوط وكذلك لحساب المفاقيد على ريش التربينات مع امكانية تغيير رقم الماخ حيث تراوح رقم الماخ من ٢, الى ١,٢ . أما بالنسبة للدراسة المعملية فقد تضمنت تصنيع الجهاز العملى وكذلك تصنيع ريش التربينات وتم اجراء التجارب المعملية حيث تراوح رقم الماخ من ٢, الى ٦,.

بينت النتائج ان استخدام ريش تربينات ذات شكل الدولفين قد عمل على تقليل مفاقيد ريش التربينات وقد تم عمل مقارنة بين النتائج النظرية والنتائج المعملية حيث وجد ان هناك توافقا بينهما.

هذا البحث يقدم دراسة فى وسائل التحكم فى مفاقيد ريش التربينات والتى يصاحبها حدوث مناطق دوامية وانفصال للسريان مما يترتب علية حدوث مشاكل فى الانسياب خلال ريش التربينات. وهذا البحث يقدم اقتراحا بأستخدام ريش التربينات ذات شكل الدولفين مما يعمل على تقليل مفاقيد ريش التربينات.