Panel method prediction of flow through a torque converter turbine

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Abstract

The flow around the blades of a torque converter turbine, consisting of an axisymmetrical radial cascade, was investigated. The pressure distribution along the blade surfaces was determined experimentally, using a torque converter model equipped with pressure measuring stations on the blade surfaces of a stationary turbine. A two-dimensional potential flow panel method was developed for the analysis of incompressible flow through axisymmetrical radial cascades. The method was verified by application to a rectilinear cascade and through use of a conformal transformation. It yielded results in excellent correspondence with published data. For the turbine cascade the panel method predicted a blade surface pressure distribution which compared well enough with the experimental results to be considered as a blade profile design tool.

Nomenclature

α	Free spiral vortex parameter
β	Angle between profile surface tangent and global x-axis
Γ	Vortex strength in vortex–sink combination forming free spiral vortex
γ	Function representing local strength of vortex distribution; vortex strength; vortex strength in specific node (when used with index)
θ	Angle in polar coordinate system
Λ	Sink strength in vortex–sink combination forming free spiral vortex
ϕ	Velocity potential
A	Trailing edge tip; quantity defined in Equation (4) (when used with index)
B	Corner on upper profile surface near trailing edge; quantity defined in Equation (4) (when used with index)
C	Free spiral vortex parameter
C_{mom}	Moment coefficient
C_{p}	Coefficient of pressure
C_{ref}	Reference pressure coefficient
j	Index number of node
MAD	Mean absolute deviation measure of fit
N	Number of nodes distributed along profile surface
P	Pressure (static unless otherwise indicated)
r	Radial distance in polar coordinate system; radius
u	Vector component in x-axis direction; velocity component in x-axis direction
V	Flow velocity
v	Vector component in y-axis direction; velocity component in y-axis direction
x	Distance along horizontal axis in cartesian coordinate system
y	Distance along vertical axis in cartesian coordinate system
Z	Number of profiles in cascade

Subscripts

γ	Due to profile surface vortex distribution
dyn	Dynamic pressure $\left(\frac{1}{2}\rho V^2\right)$
F	Due to free spiral vortex
j	Index number of node
N	Index number of node N
n	Normal to profile surface
ref	Of reference flow
stag	Stagnation (total) pressure
stat	Static pressure
t	Tangential to profile surface
u	In x-axis direction
v	In y-axis direction
1	Index number of node
2	Index number of node
3	Index number of node

Superscript

*	Indicating local coordinate system of panel

Introduction

To aid in optimizing the operation and improving the efficiency of torque converters, various authors have established theoretical design and analysis tools for hydrodynamic drives. An example is the so-called one-dimensional hydrodynamic model which predicts the performance characteristics of a torque converter by using fluid properties, geometry and the operating conditions as input, to calculate the momentum flux over the members at specific operating points. This study takes a more basic approach by analysing a single element of the torque converter, which is seen as an entity operating under certain inlet conditions. The reasoning is that analysis of the performance of single elements eventually leads to better understanding of the whole torque converter. The ultimate objective is

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to establish a method whereby blade profile shapes may be optimized.

Specific objectives of this study are

- the experimental evaluation of the pressure distribution around a blade in a torque converter turbine;
- development of a theoretical two-dimensional potential flow analysis technique for calculating this pressure distribution;
- computer implementation of the potential flow analysis technique; and
- verification of this technique through comparison with published data and experimental results.

Background

The subject of this investigation is the second turbine of a commercial torque converter which is illustrated in Figures 1 and 2. The turbine is basically an axisymmetrical radial cascade with 20 identical blades, one of which is shown in Figure 3. This blade profile is defined by series of lines and arcs which are detailed in Table 1.

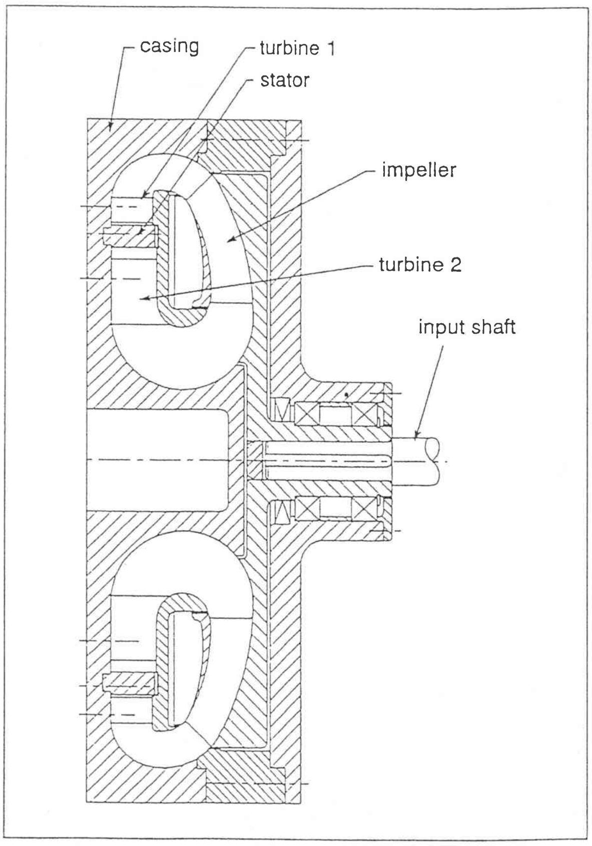

Figure 1 Cross-section of the torque converter model.

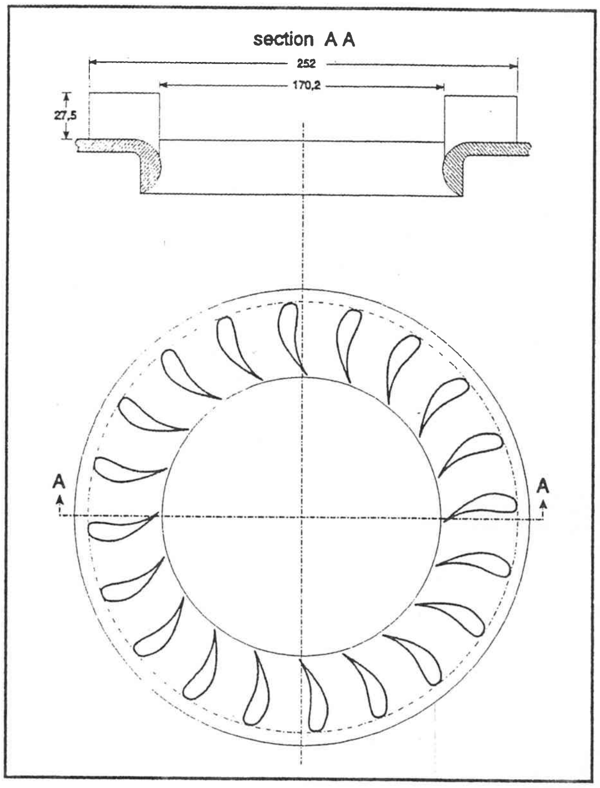

Figure 2 Plan and elevation of turbine.

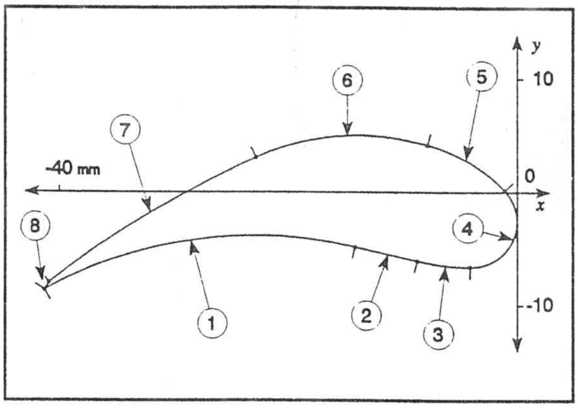

Figure 3 Turbine blade profile.

Experimental work

The torque converter turbine

A full-scale model of the torque converter with water as working fluid was used for the experiments. The model was equipped with two stationary turbines, separated by a stator and thus simulated the torque converter in the stalled mode only. This facilitated pressure measurements on the surfaces of the turbine blades, accomplished by means of holes, drilled in the surfaces of some blades and connected to a manometer, via a system of channels, tubes and valves. In total there were 23 holes on different blades,

Table 1 Detail of curves defining blade profile

Curve		Starting point		Arc centre (mm)		Arc radius
No.	Type	x	y	x	y	(mm)
1	arc	−41.293	−8.677	−22.703	−42.106	38.250
2	line	−13.991	−4.861			
3	arc	− 8.567	−6.065	− 3.869	13.509	20.130
4	arc	− 4.066	−6.620	− 4.020	− 2.600	4.020
5	arc	− 1.076	0.137	−11.427	− 9.423	14.090
6	arc	− 7.677	4.159	−13.673	−17.153	22.140
7	arc	−22.775	3.029	12.981	−75.801	86.560
8	line	−40.913	−8.065			

perpendicular to the blade surfaces and in the middle of the blade span. The blades were accurately NC machined to be identical and the pressure distributions over them are assumed identical. By measuring static pressures at the 23 holes, the pressure distribution over the blade surface was thus obtained. Over the rear 20% of the blade chord the profile was too thin to allow the drilling of holes and no pressures could be measured in this area.

In two-dimensional airfoil theory it is customary to define the pressure distribution along an airfoil surface by means of a dimensionless coefficient of pressure, expressing the pressure with reference to the static and dynamic pressures of the undisturbed stream. Since the 'undisturbed stream' velocity varies inversely with radius in a radial cascade flow, the pressure at a defined location in the model is used as reference. This location is termed the 'reference pressure station' and it was chosen to be at the cascade inlet radius, midway between two adjacent turbine blades. The dimensionless *reference pressure coefficient* is defined as:

$$C_{\text{ref}} = \frac{P_{\text{stat}} - P_{\text{ref stat}}}{P_{\text{ref dyn}}} \tag{1}$$

A 'Kiel' type probe (shrouded pitot tube) was used to measure the stagnation pressure at the reference pressure station. The dynamic pressure at the reference pressure station was assumed to be the difference between the stagnation pressure and a static pressure measured at the sidewall of the flow circuit, at the same radial position as the reference station, but midway between two other blades. The numerator of equation (1) is the difference between the reference static pressure and the static pressure registered at a blade surface pressure hole. These pressures were measured with a mercury manometer. As will shortly be shown, it is also important to find the flow angle at the reference pressure station. This 'reference flow angle' was measured using a 'wedge' probe in conjunction with two water manometers.

The measurements above were taken with the torque converter impeller running at speeds of 185, 260 and 290 rpm. The reference flow angle, as defined in Figure 4, was found to be 54.3° in all cases. The values for C_{ref} were calculated for each impeller speed, resulting in the distributions shown in Figure 5. The pressure distributions obtained at different speeds are nearly similar, affirming the repeatability of the blade pressure measurements and proving the validity of the reference pressure coefficient derived.

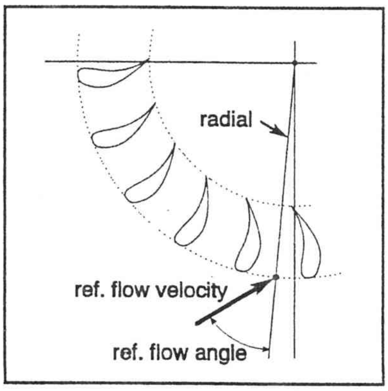

Figure 4 Reference flow.

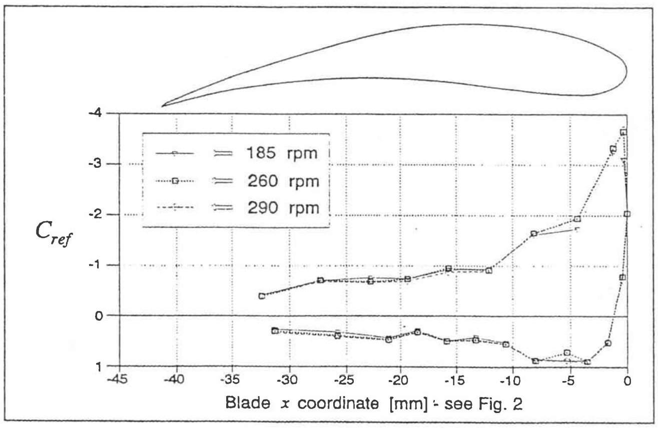

Figure 5 Experimental pressure coefficient distributions.

Potential flow analysis

Assumptions

In this analysis the flow around the torque converter blades is assumed to be incompressible, two-dimensional, inviscid and irrotational. The first assumption (incompressibility) is reasonable. Although the turbine blade aspect ratio is only 0.67, the second assumption (two-dimensionality) is tempting since it greatly simplifies the analysis, and the flow circuit curvature through the turbine is small. However, Reynaud,[1] who experimented on the same torque converter model, measured velocity differences of up to 25% and flow differences of up to 12° across the flow passage upstream of the second turbine blade row. Hence this assumption inevitably sacrifices some accuracy. The inviscid fluid assumption essentially leads to the neglect of the boundary layer on the blade surfaces. Using various techniques of flow visualisation, Reynaud concluded that little or no flow separation occurred in the turbine and, according to Hess & Smith [2] the inviscid flow assumption is thereby justified.

The panel method

According to Lakshminarayana [3] the panel method is the most suitable potential flow analysis method if the interest lies primarily in the pressure distribution. Therefore this method was chosen for the potential flow analysis. The panel method developed here will now be briefly outlined, the full details of the method and related mathematics having been given by Venter.[4]

Nodes and panels

A number of nodes are chosen on each blade surface, starting and ending at the trailing edge tip and proceeding in a clockwise direction. The section of surface between succeding nodes forms the panels and on each panel a local (x^*, y^*) coordinate system is defined with the x*-axis connecting the initial and end nodes of the panel and the y*-axis projecting outwards from the initial node. The blade surface between succeeding nodes is approximated by a cubic polynomial defined in the local coordinate system. In this way nodes and panels are distributed over all blades in the cascade, these distributions all being identical and with the nodes more densely spaced at the leading and trailing edges. A total of N nodes are chosen over the Z blades.

Free spiral vortex

A 'free stream' is commonly defined during the analysis of the flow around single profiles or in rectilinear cascades. The equivalent for the radial cascade is a free spiral vortex which follows from the conformal transformation mapping a Cartesian coordinate system onto a polar system as detailed for example by Wislicenus [5] and Scholz.[6] The free spiral vortex is centred at the cascade centre and consists of the superposition of a vortex (strength Γ) and a sink (strength Λ). With $\alpha = \arctan\left(\frac{\Gamma}{\Lambda}\right)$ and $C = \sqrt{\Gamma^2 + \Lambda^2}$, the flow velocity potential of the free spiral vortex at a point (r, θ) is then (from Anderson [7])

$$\phi_F = -\frac{C}{2\pi}\left(\theta \sin\alpha + \ln r \cos\alpha\right). \tag{2}$$

By differentiation the flow velocities in the x and y directions are

$$\begin{aligned} u_F &= -\tfrac{C}{2\pi r}\cos(\alpha+\theta) \text{ and} \\ v_F &= -\tfrac{C}{2\pi r}\sin(\alpha+\theta)\,. \end{aligned} \tag{3}$$

Distribution of singularities and their induced flow velocities

Distributions of flow singularities are set up over all panels on all blades. Only vortices are used as singularities. The strength of the vortex distribution varies linearly over each panel and its value in the j-th node is denoted by γ_j. It may be shown that the total flow velocities in the x and y directions due to the vortex distributions on all panels on all blades can be expressed as

$$\begin{aligned} u_\gamma &= \textstyle\sum_{j=1}^{N} \gamma_j A_j \text{ and} \\ v_\gamma &= \textstyle\sum_{j=1}^{N} \gamma_j B_j\,, \end{aligned} \tag{4}$$

where A_j and B_j are quantities that depend only on the geometry of the blades and the cascade and can be calculated explicitly at any field point (x, y). Thus the only unknowns involved in the flow velocities are the vortex strengths γ_j in the nodes.

Control points and equations

At any point in the flow field the flow velocities induced by the free spiral vortex and the vortices distributed on the panels on all the blades, are given by

$$\begin{aligned} u &= u_F + u_\gamma \text{ and} \\ v &= v_F + v_\gamma\,. \end{aligned} \tag{5}$$

From (3) and (4) and Figure 6 the velocity normal to the blade surface is then

$$\begin{aligned} V_n(x,y) &= -u\sin\beta + v\cos\beta \\ &= \tfrac{C}{2\pi r}\left[\cos(\alpha+\theta)\sin\beta - \sin(\alpha+\theta)\cos\beta\right] \\ &\quad + \textstyle\sum_{j=1}^{N}\gamma_j\left[A_j\sin\beta + B_j\cos\beta\right] \end{aligned} \tag{6}$$

The velocity tangential to this surface is

$$\begin{aligned} V_t(x,y) &= u\cos\beta + v\sin\beta \\ &= \tfrac{C}{2\pi r}\left[-\cos(\alpha+\theta)\cos\beta - \sin(\alpha+\theta)\sin\beta\right] \\ &\quad + \textstyle\sum_{j=1}^{N}\gamma_j\left[A_j\cos\beta + B_j\sin\beta\right]\,. \end{aligned} \tag{7}$$

The unknown vortex distributions must now be chosen so as to make the blade surfaces streamlines of the flow. It can be proved that this will be achieved if one of the following equivalent conditions is satisfied: the normal velocity at all points on the boundary must be zero; or the tangential velocity at any point just outside the blade surface must be equal to the vortex strength at the surface; or the tangential velocity just inside the blade surface must be zero (see Anderson [7]). These conditions are termed the streamline boundary conditions.

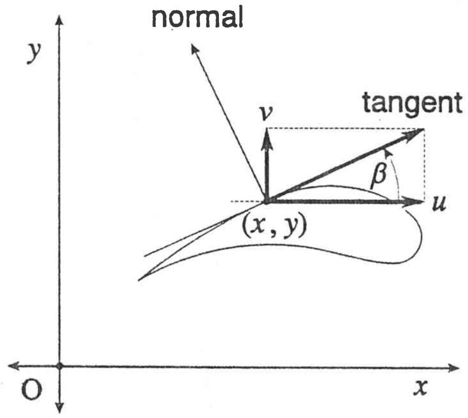

Figure 6 The normal and the tangent to the profile.

Applying the last condition to (7) leads to the linear equation

$$\begin{aligned} &\textstyle\sum_{j=1}^{N} \gamma_j \left[A_j \cos\beta + B_j \sin\beta\right] \\ &= \tfrac{C}{2\pi r}\left[\cos(\alpha+\theta)\cos\beta + \sin(\alpha+\theta)\sin\beta\right] \end{aligned} \tag{8}$$

to be satisfied by the unknown γ_j's. Multiple 'control' points (x, y) are chosen on the blade surface and this equation is applied at infinitesimally small distances away from the control points towards the inside of the blade profile. Thereby enough equations may be generated from which to solve for $\gamma_1, \gamma_2, ..., \gamma_N$. Here the blade surface nodes themselves are chosen as control points. Since the distribution of panels and vortices are identical on the different blades, it is only necessary to set up and solve linear equations on one blade. The blade chosen for this purpose is called the 'control profile'.

As proven by, for example, Moran [8] and Anderson [7] the streamline boundary flow condition is not adequate to allow a unique solution for a blade that is generating lift. Another condition must be applied namely the Kutta condition which specifies the circulation generated by a blade. As shown by Moran [8] this is done by making the vortex strength in the trailing edge tip equal to zero and stipulating that the flow on the upper and lower surfaces of the blade join smoothly at the trailing edge tip and continues along the extension of the bisector of the trailing edge angle, away from the profile (see Figure 7). For a profile with sharp trailing edge the outlet flow angle is hereby fixed to the angle of the trailing edge bisector.

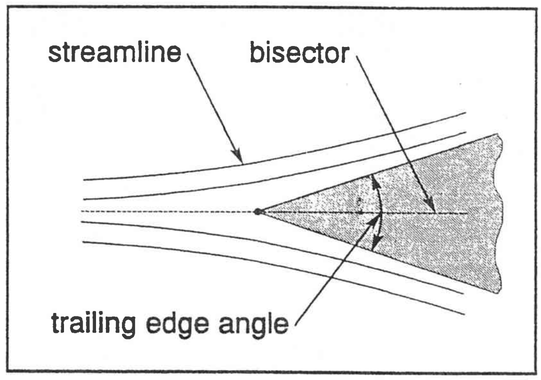

Figure 7 Trailing edge angle bisector.

Computer implementation of the panel method

A set of computer programs were written to apply the panel method to the torque converter turbine. The accuracy of the method and its implementation were verified by comparison to published data.

Application to a rectilinear cascade

The method was first modified by changing the geometry of the cascade and the character of the free stream so that it could be applied to rectilinear cascades. The program was then applied to several cases for which published C_p distributions are available, making comparisons and verifications possible. Figure 8 shows the C_p distribution predicted by the program for the rectilinear Gostelow cascade using cusped blades at a stagger angle of $-37.5°$, a pitch to chord ratio of 0.99 and an inlet angle of $-53.5°$ (Gostelow [9]). This cascade was also used as a test case by several other authors (e.g. Tanaka *et al.* [10] and McFarland [11]). The Kutta condition was applied as for the torque converter turbine blade. To calculate α, the iteration method suggested by McFarland [11] was used. The C_p distribution was calculated using the inlet velocity as reference, as was done in the original paper of Gostelow.[9] The agreement between the analytical C_p distribution given by Gostelow and that calculated here is excellent.

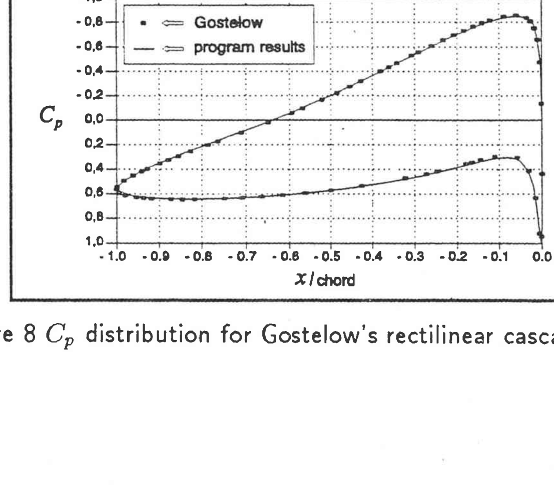

Figure 8 C_p distribution for Gostelow's rectilinear cascade.

Conformal transformation

Having established the accuracy of the analysis for the case of a rectilinear cascade, it is possible to evaluate the accuracy of the analysis for a radial cascade using a conformal transformation which maps a rectilinear cascade onto a radial cascade (Wislicenus [5] or Gostelow [9]). The program for rectilinear cascades was first applied to the Gostelow cascade. By integrating the tangential flow velocity on the profile surface, from the trailing edge in a clockwise direction around a profile in the Gostelow cascade, the flow potential on the profile was calculated in the chosen control points. The Gostelow cascade was transformed to an axisymmetrical radial cascade by means of a conformal transformation and the new locations of the control points were determined. The program for axisymmetrical radial cascades was applied to this cascade and the profile surface potentials calculated in the new control points. The flow potential is invariant under a conformal transformation and consequently the potential values obtained from both programs should correspond. However, since the programs used different conventions regarding their free streams, these potential values differed by a constant scaling factor. To remove this factor the two sets of potential values were normalized by dividing each by its potential value at the leading edge (which was approximately the largest value obtained). These normalized potential values are shown in Figure 9, plotted against the original coordinates of the (unstaggered, untransformed) Gostelow profile. It is clear that the agreement is excellent.

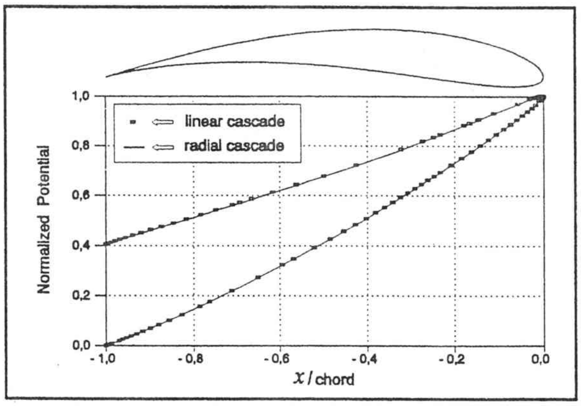

Figure 9 Potential values for linear and radial Gostelow cascades.

It has now been shown that the linear cascade program gives results in accordance with published data. Furthermore, the radial and linear cascade programs agree if applied to conformally related cascades. This leads to the conclusion that the radial cascade program is also substantially correct.

Application to the torque converter turbine

The method was then applied to the radial turbine cascade using the modified blade trailing edges discussed below.

Turbine trailing edge conditions

The trailing edge of the torque converter turbine blade is shown in Figure 10. The shape of this trailing edge creates a problem with the application of the Kutta condition. To simply apply the condition as detailed above would mean the specification of an outlet flow angle which is clearly larger than wat could realistically be expected from the cascade. Also, if the upper trailing edge corner B is not treated in a special way, inviscid flow theory would predict an infinite flow velocity around this corner. Different solutions to this problem, in the form of slight modifications to the blade profile, were investigated.

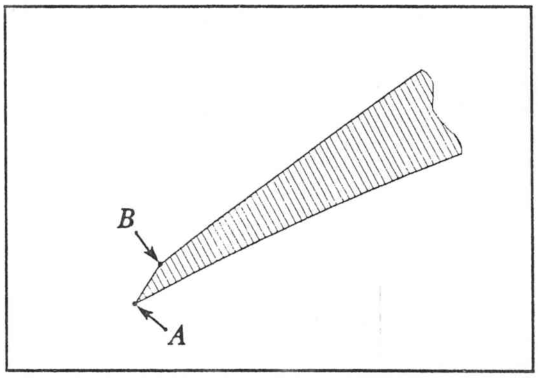

Figure 10 Turbine blade trailing edge.

The blade trailing edge was slightly modified by first smoothing out corner B with an arc and then by extending the two arcs that make up the upper and lower blade surfaces near ;the trailing edge, until they meet. Both trailing edges are illustrated in Figure 11. Although this results in a thin, sharp trailing edge to which the Kutta condition can be applied as explained above, the modifications are arbitrary and the resulting specified circulation may not be realistic. To solve this the outlet flow angle of the cascade must be established by either calculation or measurement. Due to practical considerations regarding the torque converter model, the latter proved impossible and the angle had to be estimated by calculation.

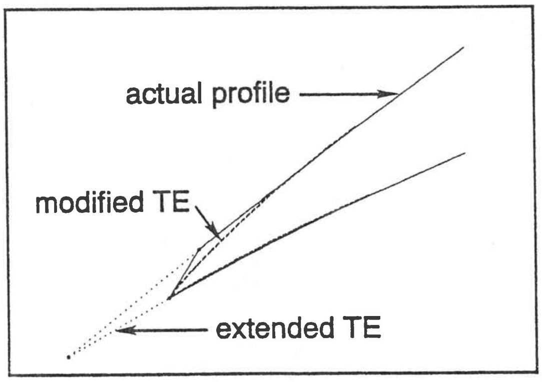

Figure 11 Extended and modified trailing edges.

Dixon [12] describes a geometrical method for the estimation of the outlet flow angle of a linear cascade. This method may be modified for application to a radial cascade using a logarithmic spiral instead of Dixon's straight line, as illustrated in Figure 12. The logarithmic spiral connects the trailing edge of one profile with the suction surface of the next, intersecting the latter orthogonally. Only one such logarithmic spiral exists. Applied to the torque converter cascade, this method yielded a radial outlet flow angle of 29.59°.

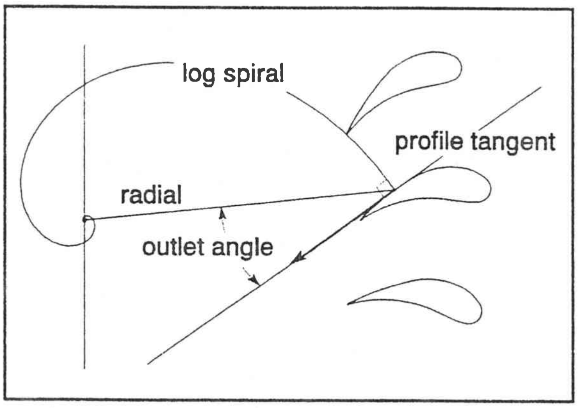

Figure 12 Estimated outlet angle.

The trailing edge was adjusted such that the radial angle of the trailing edge bisector equalled this estimated outlet flow angle. The modification used two arcs, which joined the upper and lower curves of the profile surfaces smoothly and met in a sharp, tip with a bisector at the required angle (see Figure 13). Care was taken to keep the extension as short as possible to avoid inaccuracies. The Kutta condition was then applied to the adjusted blade trailing edge.

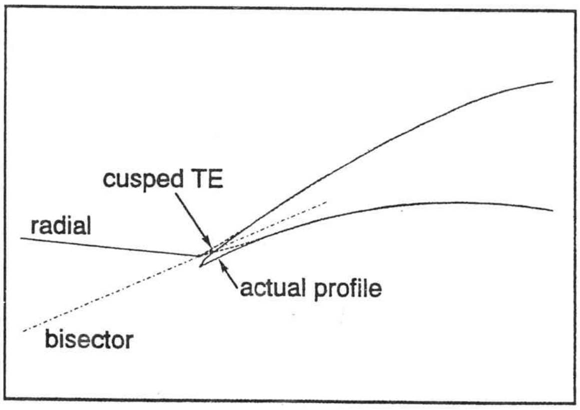

Figure 13 Adjusted trailing edge.

To solve the system of equations the free spiral vortex parameters C and α must be known. It can be proved that C acts only as a constant of proportionality of the flow velocities and that it cancels out of any ratio of one velocity to another. As will shortly be shown the pressure distribution around the blade is expressed by means of such a ratio and consequently any convenient value may be used for C (e.g. $C = 1$). The α parameter has a different influence. If α is known and the γ_j's have been determined, the flow velocity anywhere in the flow field can be calculated using (5). By calculating the angle of the flow at the location of the reference pressure station and changing the value of α until this angle corresponds to the reference flow angle measured experimentally, the α necessary for similarity to the experimental situation can be established. The pressure distribution calculated at this α should then correspond to the experimental pressure distribution.

Calculation of the pressure coefficient

In accordance with the initial assumptions it may be assumed that Bernoulli's equation holds for the flow through the stationary turbine. Then the stagnation pressure will be constant throughout the flow field and (1) may be rewritten as

$$C_{\text{ref}} = 1 - \left(\frac{V_t}{V_{\text{ref}}}\right)^2 . \qquad (9)$$

This gives the pressure coefficient in terms of the blade surface velocity and the velocity at the reference pressure station and allows the calculation of this coefficient by applying (7) at points on the blade surface and using the magnitude of the velocity given by (5). Instead of using (7) for V_t, it may also be equated to the vortex strength solution at the blade surface points (which follows from the streamline boundary condition and is much simpler but does not hold for the trailing edge node).

The calculated distribution of C_{ref} around the profile may now be compared to that obtained experimentally. To quantify this comparison the means of the absolute deviations between experimental C_{ref}'s and their calculated equivalents is used. This parameter is referred to as the *MAD* (mean absolute deviation) measure of fit. The C_{ref} distribution can also be integrated over the radial projection of the profile. By incorporating into this integral multiplication by the distance of every incremental area from the cascade centre, a measure of the moment around the cascade centre (or torque) generated by a single blade, is obtained. This *moment coefficient* is denoted by C_{mom}

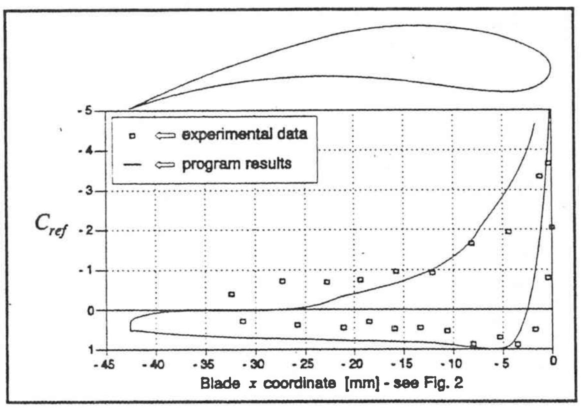

Figure 14 Reference point method results for extended trailing edge (Figure 11).

Table 2 Results for extended (Figure 11) and adjusted (Figure 13) trailing edges

Parameter	Reference point method		Optimal C_{ref} fit method	
	Extended TE	Adjusted TE	Extended TE	Adjusted TE
flow angle at reference station	−54.30°	−54.30°	−35.54°	−36.16°
α	36.02°	28.54°	36.24°	28.80°
MAD measure of fit	0.942	0.950	0.284	0.253
C_{mom} (exp) [mm^2]	−5985	−5985	−5985	−5985
C_{mom} (80%) [mm^2]	−6903	−6695	−8141	−7746
C_{mom} (100%) [mm^2]	−7379	−6935	−8998	−8128
total torque [Nm]	37.69	35.42	45.96	41.52
predicted upstream flow ∠	−68.08°	−67.99°	−50.85°	−51.48°
estimated stator outlet ∠	−55.60°	−55.60°	−55.60°	−55.60°
predicted turbine outlet ∠	33.62°	27.23°	33.90°	27.58°
estimated turbine outlet ∠	30.91°	29.59°	30.91°	29.59°

and may be calculated from the experimental and the panel method predicted C_{ref} distributions, and the results can be compared in further assessment of the accuracy of the analysis. The torque output of the turbine could also be measured experimentally and compared to an output torque calculated from the theoretical moment coefficient.

Since the extended trailing edge in Figure 11 and the adjusted trailing edge in Figure 13 yielded the most realistic results, only these will be discussed. The results for these blades are summarized in Table 2 and in Figures 14 to 17. The meaning of the table entries and the figure captions will shortly become clear.

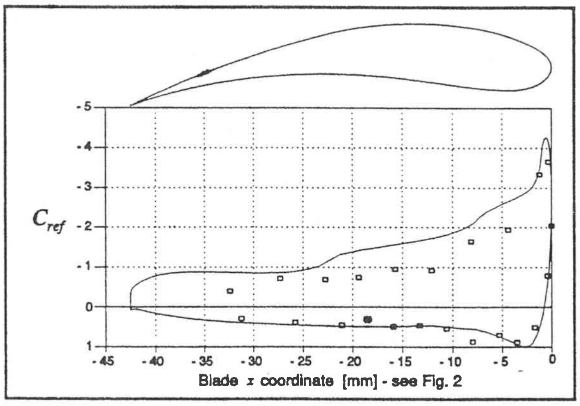

Figure 15 Optimal C_{ref} fit method results for extended trailing edge.

Extended trailing edge blade

Figure 14 shows the **pressure distribution** obtained for the extended trailing edge at the α which makes the calculated flow angle at the reference pressure station equal to the measured reference flow angle. This method for determining α is termed the ***reference point method*** and according to Table 2 it yields an α of 36.02. The MAD measure of fit is 0.942 and from Figure 14 it is clear that the predicted distribution is not a very good approximation of the experimental distribution over the full profile. It was decided to find the best possible correspondence that the panel method could yield by trying different values of α. This method for determining α is referred to as the optimal C_{ref} fit method and it results in $\alpha = 36.24°$, a minimum MAD of 0.284 and the C_{ref} distribution of Figure 15. The flow angle at the reference point now changes to −35.54°. The improvement in agreement between predicted C_{ref} and experimental C_{ref} distributions due to the optimal C_{ref} fit method, is reflected clearly in the drop in the MAD value. The fit to the experimental data is excellent on the pressure surface and the leading edge of the blade but on the suction surface the predicted C_{ref} values are generally too negative. In part this may be caused by the effect discussed previously whereby the flow deflection specified by the Kutta condition is higher than that of the actual cascade.

Table 2 also shows the **moment coefficient** calculated from the experimental data over the forward 80% of the blade chord (C_{mom}(exp)), that calculated from the panel method results over the forward 80% of the blade chord (C_{mom}(80%)) and that calculated from the panel method results over the entire blade (C_{mom}(100%)). Thus C_{mom}(exp) and C_{mom}(80%) are directly comparable. The reference point method yields a result closer to the experimental value than that of the optimal C_{ref} fit method. However, inspection of Figure 14 indicates that this could be due to a fortuitous cancellation of errors in the predicted pressure distribution rather than a closer approximation of the experimental pressure distribution. The numerically higher C_{mom} value resulting from the optimal C_{ref} fit method is due to the low C_{ref}'s predicted on the suction surface as discussed earlier. The 'total torque' was calculated using $C_{\mathrm{mom}}(100\%)$ and can be compared to the experimental torque of 30.3 Nm, cited by Reynaud.[1] The discrepancy between the predicted and experimental C_{mom} values discussed above, was the major contributor to this difference.

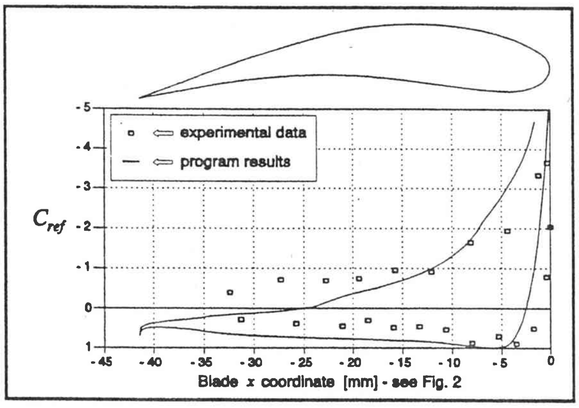

Figure 16 Reference point method results for adjusted trailing edge (Figure 13).

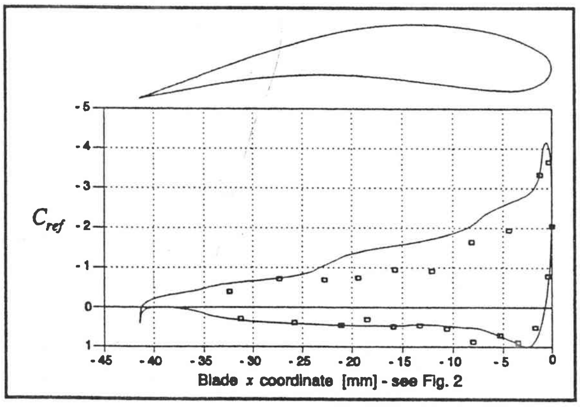

Figure 17 Optimal C_{ref} fit method results for adjusted trailing edge.

Four different flow angles are shown in Table 2. The first angle is defined as follows: if a field point is moved radially outward to infinity and the flow angle at this point calculated by means of the panel method, this flow angle tends to a limit which is indipendent of the coordinates of the field point. This limiting angle is referred to as the 'predicted upstream flow angle'. The program predicts a flow angle that is already within 2° of its limiting value at a radius 15 mm greater than the radius of the cascade leading edge. In the torque converter this upstream flow angle is considered to be the inlet flow angle of the turbine and as such it should be close to the outlet flow angle of the stator cascade, which precedes the turbine in the flow path.

Using the geometrical method of Figure 12 the stator outlet flow angle has been estimated and is referred to in Table 2 as the 'estimated stator outlet angle'. The best correspondence between this angle and the predicted upstream flow angle is obtained using the optimal C_{ref} fit method. The 'predicted turbine outlet angle' in Table 2 is the flow angle calculated by the panel method at a field point on the cascade trailing edge radius, midway between two successive trailing edges. This angle may be compared to the 'estimated turbine outlet angle' again obtained by the geometrical method.

Adjusted trailing edge blade

Figures 16 and 17 indicate that this trailing edge performed somewhat better than the simply extended trailing edge. The C_{ref}'s are significantly less negative over the rear third of the blade suction surface and for the optimal C_{ref} fit method there is a steady pressure recovery with the C_{ref}'s corresponding well to the experimental distribution over this area. For this method then the MAD fit is also the best and this trailing edge also performs better as far as the moment coefficients and the agreement between predictged and estimated flow angles are concerned.

Discussion

It was found that when the turbine blade trailing edge shape was modified to give an outlet flow angle in agreement with that expected from the cascade geometry, the inlet flow angle that would give the best fit to the experimental pressure distribution would agree to within 4° with the upstream stator outlet flow angle. The shape of the predicted pressure distribution then agrees very well with the experimental distribution except in the 10 to 60% chord region on the suction side. Taking into consideration the complex three-dimensional flow in the torque converter, especially at stall as in the current case, the agreement is as good as can be expected.

Conclusions

- A theoretical two-dimensional potential flow technique was developed for the analysis of incompressible flow through a torque converter turbine cascade.
- A set of computer programs implementing this technique was developed.
- The soundness of the technique and computer programs was verified by comparison with published results.
- The technique was applied to the torque converter turbine and yielded results in reasonable agreement with the experimental measurements.
- The agreement between predicted and experimental pressure distributions was good enough to warrant the use of the present method for the analysis of proposed turbine blade profile designs.

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