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A STREAMLINE CURVATURE METHOD OF ANALYZING AXISYMMETRIC AXIAL, --ETC(U)
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A STREAMLINE CURVATURE METHOD OF ANALYZING AXISYMMETRIC AXIAL, MIXED AND RADIAL FLOW TURBOMACHINERY

M. W. McBride

Technical Memorandum
File No. TM 77-219
21 July 1977
Contract No. N00017-73-C-1418

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REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER: TM-77-219	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle): A STREAMLINE CURVATURE METHOD OF ANALYZING AXISYMMETRIC AXIAL, MIXED AND RADIAL FLOW TURBOMACHINERY.		5. TYPE OF REPORT & PERIOD COVERED: Technical Memorandum
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s): M. W. McBride		8. CONTRACT OR GRANT NUMBER(s): N00017-73-C-1418
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Applied Research Laboratory, State College, PA 16801		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS: Naval Sea Systems Command, Washington, DC 20362		12. REPORT DATE: 21 July 1977
		13. NUMBER OF PAGES: 21
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): (12) 23p.		15. SECURITY CLASS. (of this report): UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release. Distribution unlimited.
Per NAVSEA - September 8, 1977.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

turbomachinery
flow analysis
streamline curvature method
modeling
axial flow
radial flow

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A Streamline Curvature Method of through flow analysis for axisymmetric axial, mixed and radial flow turbomachinery has been implemented. The method can be used for either the direct or indirect problem and can solve open flow problems such as flow through a propeller on a body of revolution as well as the more familiar duct flow problems. Computational refinements allow modeling of blade row chordwise and spanwise loading and blockage distributions and selection of streamline locations. Several examples of the abilities of the analysis are presented.

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To: B. R. Parkin, R. E. Henderson, W. S. Gearhart, M. J. Pierzga, A. L. Treaster, F. E. Smith, J. R. Ross, M. L. Billet, ARL Files (4), Water Tunnel Files

Information: (Abstract Only) J. C. Johnson, N. A. Abourezk, F. P. Finlon, C. L. Key, M. T. Pigott, R. F. Custard, F. S. Archibald, E. P. Bruce, R. F. Davis, J. J. Eisenhuth, G. B. Gurney, W. R. Hall, G. H. Hoffman, J. W. Holl, G. C. Lauchle, J. L. Lumley, B. E. Robbins, G. L. Sanders, D. R. Stinebring, D. E. Thompson, A. Yocum

From: M. W. McBride

Subject: A Streamline Curvature Method of Analyzi[illegible]symmetric Axial, Mixed and Radial Flow Turbomachinery

References: See page 15.

Abstract: A Streamline Curvature Method of through flow analysis for axisymmetric axial, mixed and radial flow turbomachinery has been implemented. The method can be used for either the direct or indirect problem and can solve open flow problems such as flow through a propeller on a body of revolution as well as the more familiar duct flow problems. Computational refinements allow modeling of blade row chordwise and spanwise loading and blockage distributions and selection of streamline locations. Several examples of the abilities of the analysis are presented.

Acknowledgment: The work reported in this paper was sponsored by the Naval Sea Systems Command, Code SEA 0351. Special thanks is given to M. J. Pierzga who assisted in developing the direct flow analysis.

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Nomenclature

α	angle between a streamline and a reference line
β	dummy variable of integration in η direction
η	ordinate parallel to reference line
k	streamline curvature = (radius of curvature $R_k)^{-1}$
n	ordinate normal to a streamline
P	static pressure
ϕ	angle between streamline and axis of rotation
V_m	meridional velocity = $(u^2 + v^2)^{1/2}$
ρ	fluid density
r	ordinate along a radial line
s	ordinate parallel to a streamline
U	rotor rotational velocity
u	velocity parallel to axis of rotation
v	velocity normal to axis of rotation
R	radius
V_θ	tangential velocity
V_∞	total velocity at reference conditions
X	ordinate parallel to axis of rotation
Y	ordinate normal to axis of rotation
ξ	coordinate location on η axis

Subscripts

x	denotes partial differentiation W.R.T. X
y	denotes partial differentiation W.R.T. Y
i	denotes inner boundary
o	denotes outer boundary

η	denotes a function of η
ξ	denotes a property of a particular streamline corresponding to a value of ξ on η
1	blade row inlet
2	blade row exit

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Introduction

The Streamline Curvature Method (SCM) has been developed over many years to solve various classes of axisymmetric and quasi-three dimensional turbomachine through flow problems. The method relies on the ability to define accurately the streamlines on a meridional plane and to determine radial and convective accelerations based on their geometry. In the past, the SCM has been successfully applied to axial flow pumps and compressor/ turbines, this being performed with the equations written in orthogonal or intrinsic coordinate systems. Except for the cumbersome quasi-three dimensional analysis (the direct problem), approximations are often made which limit the accuracy of the analysis, particularly with regard to the blade rows. Blade rows are usually treated as thick actuator disks with chordwise loading, blockage, and radial body forces ignored.

Computationally, many of these programs are inefficient and limited in the types of cases they can handle. This paper documents a computer analysis which addresses the problems described. The program is capable of modeling axial, mixed and radial flows with multiple blade rows and can solve an approximately direct as well as the indirect problem. Blade to blade effects are incorporated as a circumferentially averaged radial body force. Particular attention to stability of convergence makes this analysis suitable for problems which are usually very difficult to solve.

In order to improve the usefulness of the SCM, equations were written such that reference stations may take an arbitrary path through the machine. This allows the complete modeling of the leading and trailing edges of the blade rows. The option of intrablade computing stations models the blade loading and blockage distribution, and radial body forces. The use of specialized curve fitting routines allows radial as well as mixed flow and axial flows to be analyzed. The stability of the program allows very high station aspect ratios to be used. Pseudo-streamlines and the ability to define streamline locations improves the accuracy and speed of the program and allows concentration of data in regions of interest.

To date, the program has been used to analyze axial flow and radial flow pumps and has performed the direct analysis of an open propeller in an infinite medium, all with good success. Examples of these cases will be presented in this report. Documentation of the actual program will appear in a later report.

Development of the Equations of Motion

In the axisymmetric inviscid analysis, Euler's momentum equation is recast into a radial equilibrium equation. Conservation of mass, total energy and angular momentum complete the analysis. The equations must be solved by successive approximations, with necessary data and derivatives taken from the iteratively approximated velocity field and streamline geometry.

Let x and y be rectangular coordinates in a meridional plane as illustrated in Figure 1. Euler's equations in these rectangular coordinates, for a two-dimensional incompressible flow are:

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = - \frac{1}{\rho} \frac{\partial P}{\partial y}$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = - \frac{1}{\rho} \frac{\partial P}{\partial x} \quad . \qquad (1)$$

Figures (1) and (2) represent the meridional plane of the solution and geometric quantities used in the following equations are shown on these figures.

The pressure gradient between points a and b, Figure 2, is represented by:

$$\left.\frac{dP}{d\eta}\right|_a^b = \left.\frac{\partial P}{\partial s} ds\right|_a^b + \left.\frac{\partial P}{\partial n} dn\right|_a^b \quad . \qquad (2)$$

To [illegible] the pressure gradient along η requires the partial derivatives in [illegible] be determined.

In the streamwise direction one has

$$\frac{\partial P}{\partial s} = \frac{\partial P}{\partial y} \sin \phi + \frac{\partial P}{\partial x} \cos \phi \quad . \qquad (3)$$

Combining (1) and (3) we have

$$- \frac{1}{\rho} \frac{\partial P}{\partial s} = [u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y}] \sin \phi + [u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y}] \cos \phi \quad . \qquad (4)$$

Note that

and

$$v = V_m \sin \phi$$

$$u = V_m \cos \phi \quad . \qquad (5)$$

The following derivatives are determined by the chain rule:

$$\frac{\partial v}{\partial x} = V_m \cos\phi\, \phi_x + \sin\phi\, V_{m_x}$$

$$\frac{\partial u}{\partial x} = -V_m \sin\phi\, \phi_x + \cos\phi\, V_{m_x}$$

$$\frac{\partial v}{\partial y} = V_m \cos\phi\, \phi_y + \sin\phi\, V_{m_y} \tag{6}$$

$$\frac{\partial u}{\partial y} = -V_m \sin\phi\, \phi_y + \cos\phi\, V_{m_y} \quad .$$

The quantities determined in (5) and (6) are substituted into Equation (4) and after reduction the result is:

$$-\frac{1}{\rho}\frac{\partial P}{\partial s} = V_m \cos\phi\, V_{m_x} + V_m \sin\phi\, V_{m_y} \quad . \tag{7}$$

Equation (7) reduces to

$$-\frac{1}{\rho}\frac{\partial P}{\partial s} = V_m \frac{\partial V_m}{\partial s} \quad , \tag{8}$$

which is seen to be the differential form of the steady Bernoulli's equation for an incompressible flow.

The streamwise normal component of Equation (2) is developed in a similar manner by noting that

$$\frac{\partial P}{\partial n} = \frac{\partial P}{\partial y}\cos\phi - \frac{\partial P}{\partial x}\sin\phi \quad , \tag{9}$$

and

$$-\frac{1}{\rho}\frac{\partial P}{\partial n} = [u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}]\cos\phi - [u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}]\sin\phi \quad . \tag{10}$$

After combining (5), (6) and (10), we obtain the following result:

$$-\frac{1}{\rho}\frac{\partial P}{\partial n} = V_m^{\ 2} \cos\phi \; \phi_x + V_m^{\ 2} \sin\phi \; \phi_y \tag{11}$$

which is equivalent to

$$-\frac{1}{\rho}\frac{\partial P}{\partial n} = V_m^{\ 2} \left\{ \frac{\partial x}{\partial s} \phi_x + \frac{\partial y}{\partial s} \phi_y \right\} \quad . \tag{12}$$

The quantity in the brackets in (12) is recognized to be equivalent to $\partial\phi/\partial s$, the curvature (k) of the streamline.

Finally, Equation (12) becomes

$$-\frac{1}{\rho}\frac{\partial P}{\partial n} = k \, V_m^{\ 2} \quad . \tag{13}$$

Combining Equations (2), (8) and (13), we determine the radial equilibrium equation for nonswirling flow to be

$$\frac{\partial P}{\partial \eta} = \frac{\partial P}{\partial s}\frac{\partial s}{\partial \eta} + \frac{\partial P}{\partial n}\frac{\partial n}{\partial \eta}$$

and finally,

$$-\frac{1}{\rho}\frac{\partial P}{\partial \eta} = k \, V_m^{\ 2} \sin\alpha + V_m \frac{\partial V_m}{\partial s} \cos\alpha \quad . \tag{14}$$

The pressure gradient due to swirl is determined in a similar manner and leads to the term,

$$-\frac{1}{\rho}\frac{\partial P}{\partial R} = \frac{1}{R} V_\theta^{\ 2} \quad , \tag{15}$$

where there are no circumferential derivatives, i.e., the flow is assumed to be uniform in the circumferential direction. The computational form of the radial equilibrium equation is

$$-\frac{1}{\rho}\frac{\partial P}{\partial \eta} = k\, V_m^{\,2} \sin\alpha + \frac{1}{R} V_\theta^{\,2} \sin(\phi+\alpha) + V_m \frac{\partial V_m}{\partial s} \cos\alpha \quad . \tag{16}$$

The three terms on the right hand side of Equation (16) represent the meridional curvature, the radial and the convective accelerations respectively. Integration of the equation will yield the static pressure difference between any two points in the flow field.

Application of the Continuity and Energy Equation

From Equation (16), the static pressure difference relative to some point, say η_i, in the flow, to another point, ξ, along the reference line may be found. To satisfy conservation of mass and energy across the reference line, an absolute value of static pressure must be found at the reference point, η_i. The following direct solution for this value is an improvement over other procedures which are iterative in nature.

A continuity equation may be written for every station in the problem:

$$\rho \int_{\eta_i}^{\eta_o} V_m(\eta)\; r(\eta)\; \sin(\alpha_\eta)\; d\eta = \text{const} \quad . \tag{17}$$

In words, the mass flow across a reference line regardless of its path between two streamlines, is constant, assuming an incompressible fluid. The energy equation may be written for a particular point, ξ, on the reference line (along a streamline) as

$$\left\{ P_{\infty_\xi} + \frac{1}{2}\rho\, V_{\infty_\xi}^{\,2} - \frac{1}{2}\rho\, V_{\theta_\xi}^{\,2} - P_{\eta_i} - \int_{\eta_i}^{\xi} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \eta} \right) d\eta \right\} = \frac{1}{2}\rho\, V_{m_\xi}^{\,2} \quad . \tag{18}$$

The first two terms on the left hand side represent the total energy (static plus dynamic) available at the point, ξ. The term P_{∞_ξ} contains all head loss and rotor energy changes between the reference condition and the station of interest. The third term is derived from the conservation of angular momentum and it is the rotational kinetic energy. The quantity RV_θ is assumed constant in the absence of losses or momentum changes due to a rotor or stator. The sum of the fourth and fifth terms is the static pressure at ξ. The term in the integral is the static pressure difference from Equation (16).

Equations (17) and (18) may be combined and integrated:

$$\int_{\eta_i}^{\eta_o} \left\{ P_{\infty_\eta} + \frac{1}{2}\rho V_{\infty_\eta}{}^2 - \frac{1}{2}\rho V_{\theta_\eta}{}^2 - P_{\eta_i} - \int_{\eta_i}^{\eta} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \beta} \right) d\beta \right\}^{1/2} r_\eta \sin(\alpha_\eta)\, d\eta =$$

$$\int_{\eta_i}^{\eta_o} \frac{1}{2}\rho V_{m_\eta} r_\eta \sin(\alpha_\eta)\, d\eta \quad . \tag{19}$$

Because P_{η_i} is a constant, Equation (19) may be rearranged to give

$$\int_{\eta_i}^{\eta_o} P_{\eta_i} W^2 d\eta = \int_{\eta_i}^{\eta_o} \left\{ P_{\infty_\eta} + \frac{1}{2}\rho V_{\infty_\eta}{}^2 - \frac{1}{2}\rho V_{\theta_\eta}{}^2 - \left\{ \int_{\eta_i}^{\eta} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \beta} \right) d\beta \right\} - \right.$$

$$\left. \frac{1}{2}\rho V_{m_\eta}{}^2 \right\} W^2 d\eta \tag{20}$$

where,

$$W = r_\eta \sin(\alpha_\eta) \quad ,$$

and finally;

$$P_{\eta_i} = \frac{\displaystyle\int_{\eta_i}^{\eta_o} \left\{ P_{\infty_\eta} + \frac{1}{2}\rho \left(V_{\infty_\eta}{}^2 - V_{\theta_\eta}{}^2\right) - \frac{1}{2}\rho V_{m_\eta}{}^2 - \int_{\eta_i}^{\eta} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \beta} \right) d\beta \right\} W^2 d_\eta}{\displaystyle\int_{\eta_i}^{\eta_o} W^2 d\eta} \quad . \tag{21}$$

The static pressure anywhere along a reference line is then:

$$P_\xi = P_{\eta_i} + \int_{\eta_i}^{\xi} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \eta}\right) d\eta \quad . \tag{22}$$

The velocity profile which satisfies angular momentum, continuity and total energy is given by rearranging (18):

$$V_{m\xi} = \frac{2}{\rho}\left\{P_{\infty_\xi} + \frac{1}{2}\rho(V_{\infty_\xi}^{\ 2} - V_{\theta_\xi}^{\ 2}) - P_{\eta_i} - \int_{\eta_i}^{\xi} \left(-\frac{1}{\rho}\frac{\partial P}{\partial \eta}\right) d\eta\right\}^{1/2} \quad . \tag{23}$$

The flow field is solved by marching downstream to each station in turn and integrating Equation (16). The static pressure is then obtained from Equation (22). The improved velocity profiles are generated by Equation (23) until the changes in the profiles are small between two successive passes.

Computational Procedure

Certain data are necessary to start the iterative computation cycle. Data specifying the design, i.e., the geometry, blade location, loading and thickness, and reference fluid conditions are input. Initial one-dimensional approximations of all the velocity profiles and the streamline pattern are made. From the initial guess for the velocity profiles and streamline geometry, derivatives necessary for the terms in Equation (16) can be determined. This equation is integrated and the pressure difference as a function of η is saved. This information is used to solve Equation (21) for P_{η_i}. Once solved, Equation (23) is used to generate improved velocity profiles. The improved velocity profiles are integrated to give the mass flow as a function of η and by specifying percentages of the total mass flow, new streamline locations are determined. This data is fed back into the program and the cycle is repeated until convergence is met.

All of the major program variables such as streamline location, radius of curvature at every point, velocities and other derivatives are severly damped against their previous values to prevent instabilities from growing during the computations. As a result of this treatment, problems with the ratio of radial distance to axial spacing (station aspect ratio) that are large can be solved. This feature is important when problems such as propellers in an approximately infinite medium are to be analyzed. The computational aspects of the SCM analysis will be fully reported in a later document, along with a complete problem solution.

Effects of Rotors and Stators

In an inviscid solution, the effect of a rotor or stator is to change the angular momentum of the fluid passing through a blade row and in the case of a rotor to change the total pressure, the term $(P_{\infty\eta} + 1/2\rho\, V_{\infty_\eta}^{\ 2})$ in Equation (21). In both cases the term $1/2\rho\, V_{\theta_\eta}^{\ 2}$ is changed.

A rotor changes the total head in proportion to the change in angular momentum of the fluid passing through the rotor. The total pressure is increased (or in the case of a turbine, decreased) in accordance with the equation,

$$\Delta P_{\infty_\eta} = \frac{1}{2}\,\rho(U_2\,V_{\theta 2} - U_1\,V_{\theta 1}) \quad , \tag{24}$$

where V_θ is positive in the direction of rotation.

Losses

Frictional and secondary flow losses play an important role in determining the performance of any fluid handling machine. Our ability to predict these losses theoretically for a generalized turbomachine in nonexistant, therefore correlational data must be used if the effects are to be included in the analysis. To obtain these data, a machine must be analyzed for inviscid flow. Comparison with experimental results then indicates the magnitude and distribution of the losses encountered. These data can be incorportated into the analysis as a distributed total pressure loss and the program 'tuned' for a particular machine. It must then be assumed that the losses are similar for other machines of the same general type. Data exist for a variety of pumpjet configurations and have been used successfully in predicting the performance of new machines.

Examples

In this section we present several examples of the uses of the Streamline Curvature Method as described in this report. The first example is of a counterrotating set of open propellers on a body of revolution. The outer boundary streamline is defined by a potential flow around the body shape. The effect of the rotors on this streamline are small enough to be neglected.

The plotted streamlines for this configuration are shown in Figures (3) and (4) demonstrate the more significant results. First, there is a streamline contraction through the rotors as the flow is accelerated near the body. The velocity profiles exhibit the characteristic bulge or jet behind the rotor and in this case the average jet velocity is about 1.6 times the free stream velocity. The plot also demonstrates the ability of the program to use curved reference stations and to solve problems with high station aspect ratios (in this case, AR=25). The analysis is able to determine either the tip radius of the propellers for a given mass flow rate, or, given the tip radius as in the direct solution, to determine the mass flow and powering requirements for the configuration.

The second example is of a Francis-type turbine with wicket gates and a simulated inlet volute. This particular problem demonstrates the ability of the analysis to describe axial, mixed and purely radial flows. A streamline plot is presented in Figure 5.

The third example is the direct solution of the flow through an open propeller. In this case only the angularity distribution in the rotor exit plane is specified, rather than the usual tangential velocity distribution. A tangential velocity distribution is iteratively determined that satisfies the angularity while at the same time all other equations of motion are satisfied. Figure (6) shows the theoretical velocity profile obtained from this procedure and corresponding experimental data for one typical case.

Summary

A method of through-flow analysis for turbomachinery has been developed which has proven successful for a variety of types, including axial, radial and mixed flow machines. The equations of motion are solved by a different approach than is commonly used, in that a differential equation for the pressure gradient rather than the velocity gradient is used. Additionally, the energy and continuity equations are satisfied by a direct integration rather than by iterative approximations.

Refinements to the numerical method permit more accurate modeling of the streamlines, allowing mixed and radial flows to be analyzed. Multiple blade rows with intrablade computing stations allows modeling of the blade spanwise and chordwise loading distributions and the blade thickness distribution. Selection of streamline location and density improves accuracy in regions of interest.

Particular attention to the stability of the numerical procedure allows very high station aspect ratios to be used and allows problems such as propellers in an unbounded medium to be analyzed, as well as the standard internal flow problems.

Friction losses and secondary flow effects may be included in an empirical fashion when a particular machine type is prescribed.

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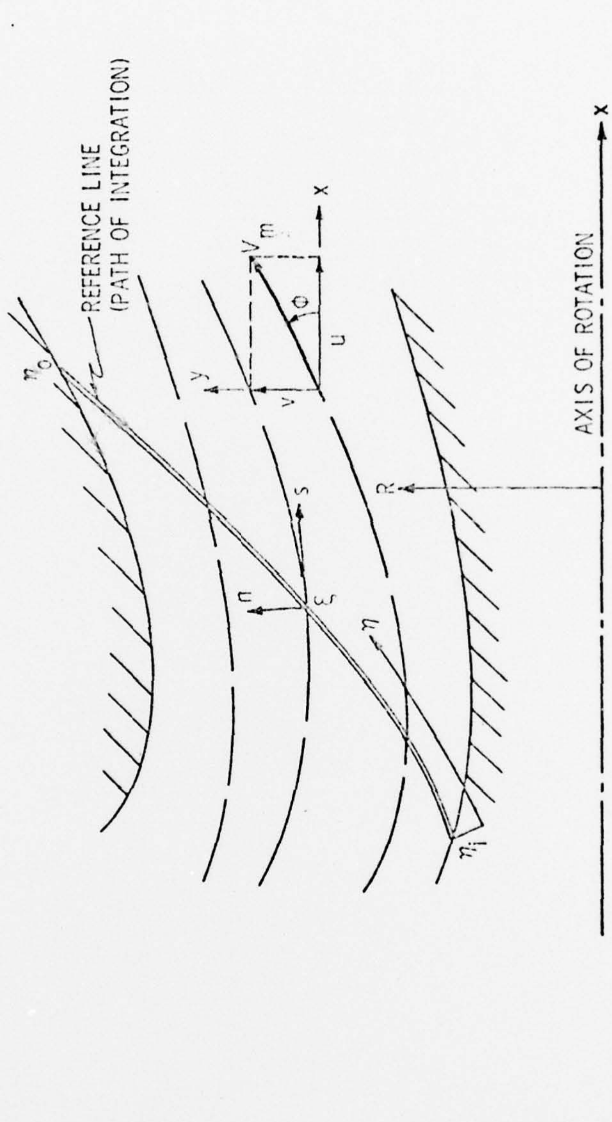

Figure 1 - General Reference Station Parameters (Meridional Plane)

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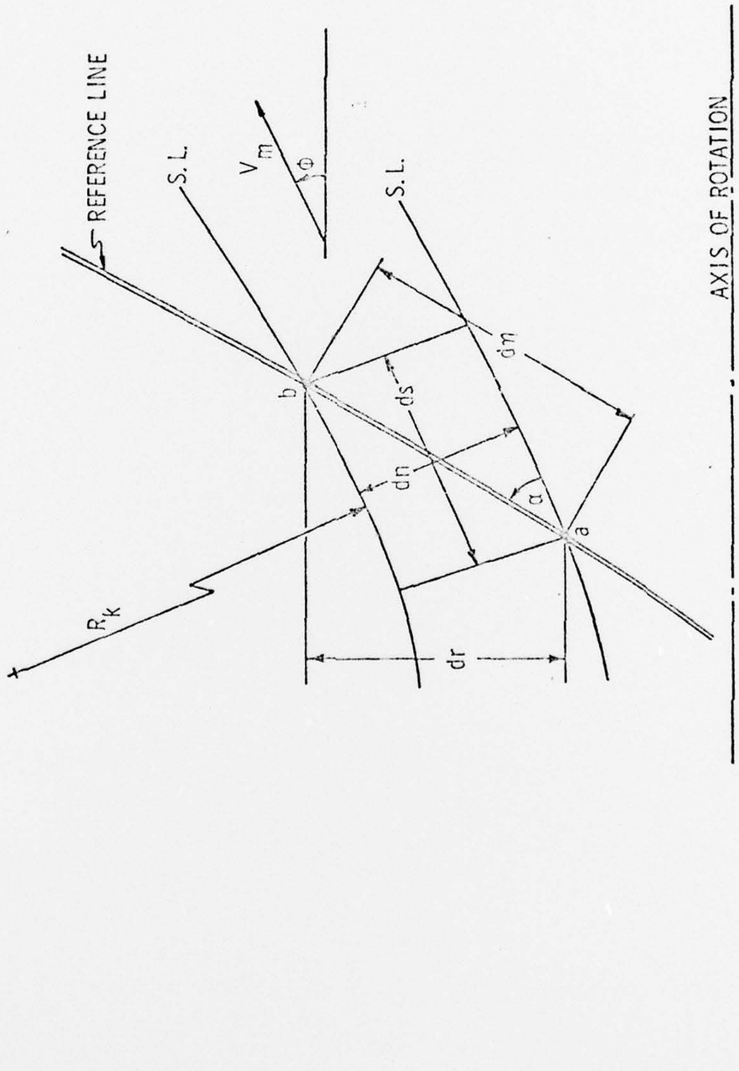

Figure 2 - Differential Streamline Element

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Figure 3 - Overall View of the Streamlines Through
a Counterrotating Open Propeller Set

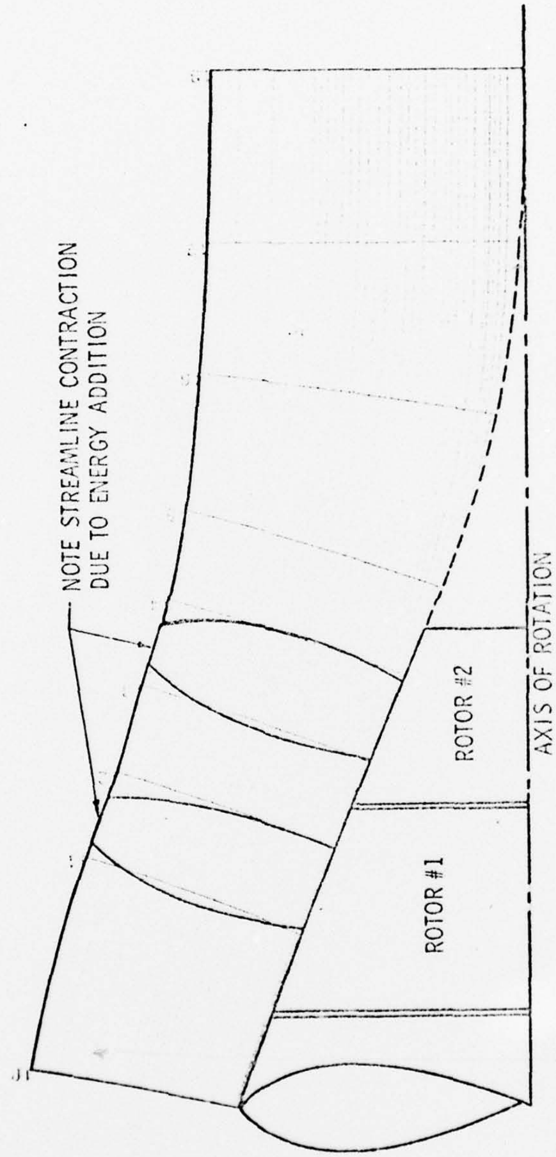

Figure 4 - Detail View of the Streamlines Through a Counterrotating Open Propeller Set

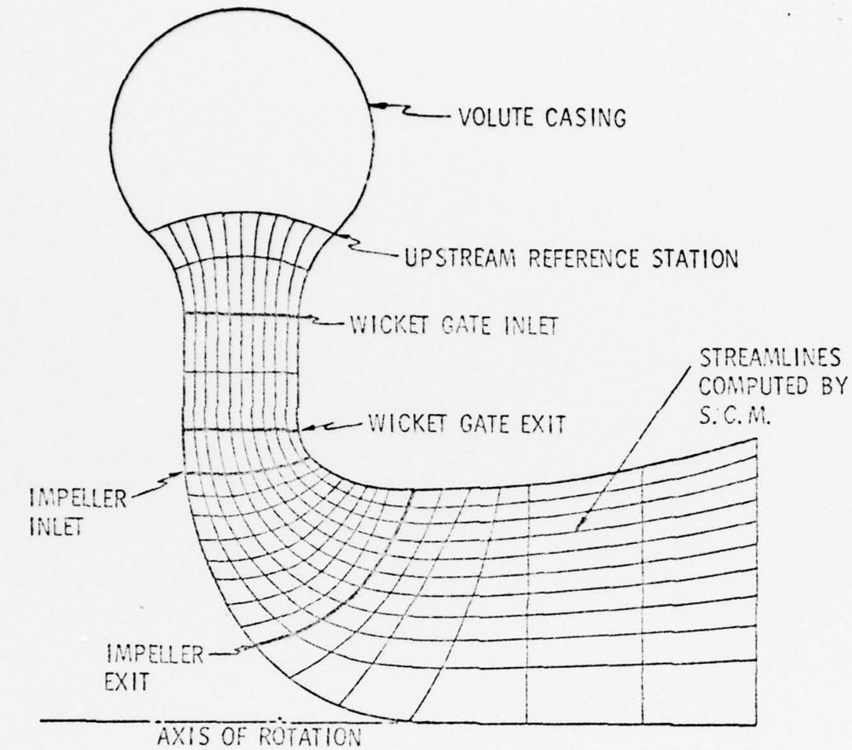

Figure 5 - Streamlines in a Francis Type Turbine

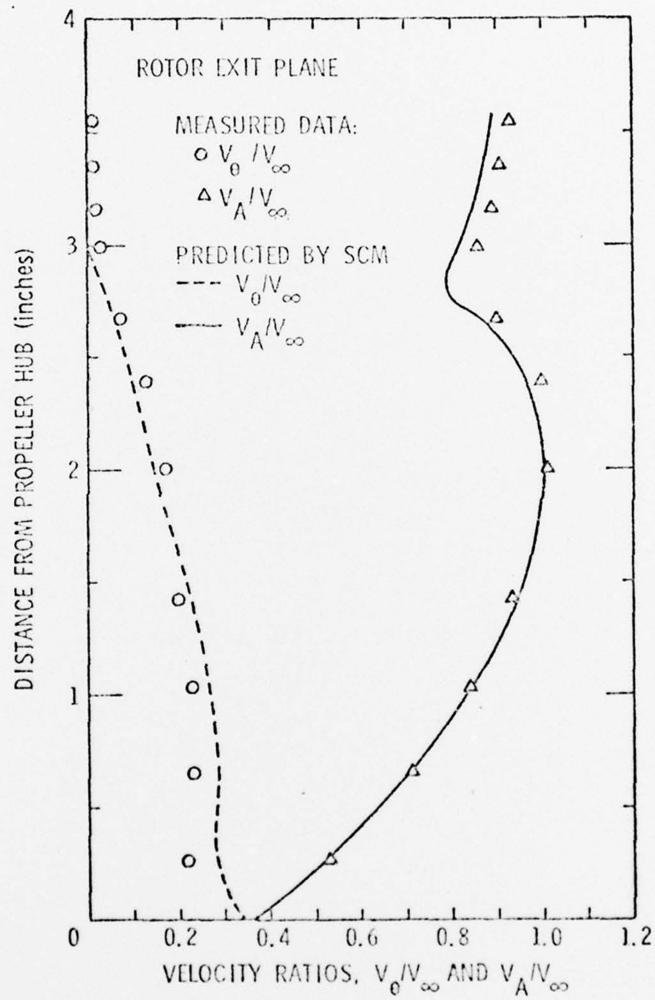

Figure 6 - Comparison of Theoretical and Experimental Velocities Behind an Open Propeller as Solved by the Direct (Specified Angularity) Method

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