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WILLIAM G. STELTZ*, DAVID H. EVANS*, WILLIAM F. STAHL**

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The Aerodynamic Design of High Performance Low Pressure Steam Turbines

Introduction

Significant advances in turbomachinery aerodynamic technology have been achieved in the past decade or so with particular successes noted in the area of low pressure steam turbine development. A significant factor in these advanced developments is due in no small measure to the availability and utilization of high speed digital computers in the solution of the system of turbomachinery aero-thermodynamic equations. The coupling of the solution of this complex equation system with the necessary verification as derived from experimental testing programs is the subject of this paper.

The development of the equation system is presented in essentially summary form resulting in a quasi three-dimensional approach to the aerodynamic design of turbomachinery. The solution of these equations is indicated by utilization of the digital computer, and a unique solution technique is employed for a particular solution of these general equations.

The verification of this overall design approach is demonstrated by the design, construction, and test of a research low pressure steam turbine consisting of three stages. The performance evaluation of this test vehicle was accomplished by means of overall thermodynamic instrumentation and by means of a radial traverse immediately downstream of the last rotor blade. Comparisons are made of these test data with the design flow conditions obtained from solutions of both the complete set of aero-thermodynamic equations and that corresponding to the particular solution referred to above.

1. The general aero-thermodynamic relationships

The development of the system of aero-thermodynamic equations is traced from basic considerations of the several conservation equations to working equations which are then generally solved by numerical techniques. Pioneering work in the development of this

* Steam Turbine Division, Westinghouse Electric Corporation, Philadelphia.
** Gas Turbine Systems Division, Westinghouse Electric Corporation, Philadelphia.

type of equation systems was performed by Wu [1] of NACA in the early 1950's. Subsequent refinement and adaptation of his fundamental approach has been made by several workers in recent years.

The general three dimensional representation of the conservation equations of mass, momentum, and energy can be written as follows for application to this adiabatic system:

– conservation of mass

$$\frac{\partial \rho}{\partial t}+\nabla \cdot \rho \mathbf{V}=0, \tag{1}$$

– conservation of momentum

$$\frac{\partial \mathbf{V}}{\partial t}+\mathbf{V} \cdot \nabla \mathbf{V}=-\frac{1}{\rho} \nabla p, \tag{2}$$

– conservation of energy

$$\nabla h=T \nabla s+\frac{1}{\rho} \nabla p. \tag{3}$$

From thermodynamic principles relationships between the properties of a fluid may be described by equations of state, such that

$$p=p(\rho, T) \tag{4}$$

and

$$h=h(p, T). \tag{5}$$

For steam, these functions are presented in the form of the steam tables [2].

As the increase in system entropy is a function of the several internal loss mechanisms inherent in the turbomachine, it is necessary that definitive known (or assumed) relationships be employed for its evaluation. In the design process this need can usually be met, while the performance analysis of a given geometry usually introduces loss considerations more difficult to completely evaluate.

Assuming the flow to be steady and axi-symmetric, equations (1) to (5) can be solved given the proper boundary conditions for the variables $\mathbf{V}$, ρ, p, T, h and s.

With the above assumptions the governing equations can be written in cylindrical coordinates as:

$$\frac{\partial}{\partial r}(\rho r b V_r)+\frac{\partial}{\partial z}(\rho r b V_z)=0, \tag{6}$$

$$V_r \frac{\partial}{\partial r}(r V_\theta)+V_z \frac{\partial}{\partial z}(r V_\theta)=0, \tag{7}$$

$$V_r \frac{\partial V_z}{\partial r}+V_z \frac{\partial V_z}{\partial z}=-\frac{1}{\rho} \frac{\partial p}{\partial z}, \tag{8}$$

$$V_r \frac{\partial V_r}{\partial_r}+V_z \frac{\partial V_r}{\partial z}-\frac{V_\theta^2}{r}=-\frac{1}{\rho} \frac{\partial p}{\partial r}, \tag{9}$$

$$\frac{1}{\rho}\,dp = dh_t - T\,ds - \frac{1}{2}\,d(V_\theta^2 + V_z^2 + V_r^2), \tag{10}$$

where

$$h_t = h + \tfrac{1}{2}(V_\theta^2 + V_z^2 + V_r^2) \tag{11}$$

and b has been introduced in the continuity equation to account for blade blockage.

With the entropy variation known as a functional relationship of the aerodynamic design parameters, equations (4) to (11) can be solved for the (now) eight unknowns, V_r, V_θ, V_z, ρ, p, T, h and h_t.

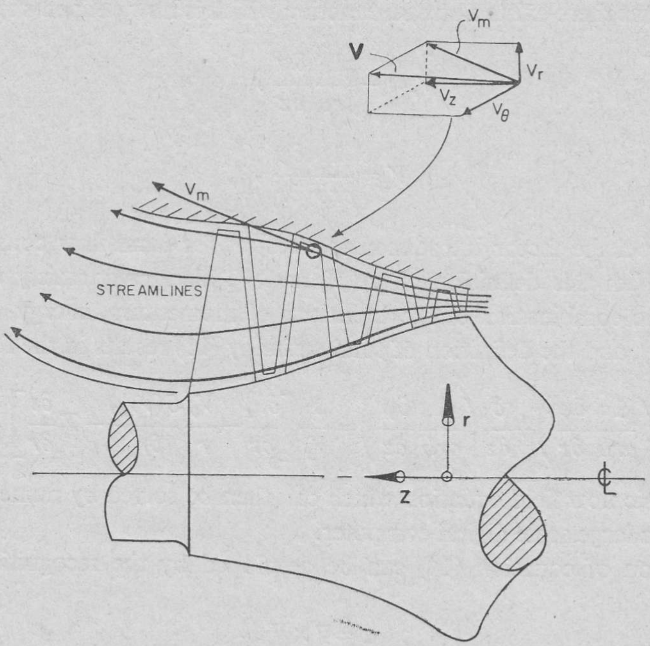

Fig. 1. Turbine blade path flow field

The meridional plane, defined as that plane passing through the turbomachine axis and containing the radial and axial coordinates, can be used to describe an additional representation of the flow process. The velocity V_m (Fig. 1) represents the stream tube meridional plane velocity with direction proportional to the velocity components V_r and V_z. If the changes in entropy and total enthalpy along the streamlines are known, or specified, it can then be shown that equations (8) and (9) are equivalent and that (7) is also satisfied, see for example [1 and 4]. Application of equation (7) to the rotating blade in effect describes Euler's turbine equation when equated to the blade force producing useful work. In this situation it is most convenient to choose equation (9), the commonly known radial equilibrium equation, as the relationship for continued evaluation.

At this point in the development it is necessary to identify the desired method of solution of the set of fundamental differential equations. Two commonly encountered techniques have been employed and are usually referred to as the streamline curvature and matrix solution techniques. Briefly, the streamline curvature technique is structured to evaluate meridional velocities and to trace streamlines in the flow field allowing the calculation of the streamline curvature itself. The matrix approach on the other hand utilizes a stream function satisfying the equation of continuity; the calculation procedure then determines the stream function throughout the flow field. The streamline curvature technique has been developed to a high degree of sophistication [3] and has been applied to axial flow compressor design as well as to axial flow turbine problems. The matrix method has also been successfully developed [4] and applied in a similar manner; particular note is made here with regard to low pressure steam turbine application. The method employed in this paper is that of the matrix approach which was used to design the research steam turbine.

It is asserted that there exists a stream function which is now defined such that

$$V_r = -\frac{1}{\rho r b}\frac{\partial \psi}{\partial z}, \tag{12}$$

$$V_z = \frac{1}{\rho r b}\frac{\partial \psi}{\partial r}, \tag{13}$$

where b is the blockage factor introduced to account for local flow constrictions in the bladed regions. With this definition of ψ, V_r and V_z identically satisfy the equation of continuity (6). The combination of the equations of momentum, energy, and continuity, (9), (10), (12), (13), and the definition of total enthalpy (11) results in the following:

$$\frac{\partial}{\partial r}\left(\frac{1}{\rho r b}\frac{\partial \psi}{\partial r}\right)+\frac{\partial}{\partial z}\left(\frac{1}{\rho r b}\frac{\partial \psi}{\partial z}\right)=\frac{1}{V_z}\left[\frac{\partial h_t}{\partial r}-\frac{V_\theta}{r}\frac{\partial (rV_\theta)}{\partial r}-T\frac{\partial s}{\partial r}\right]. \tag{14}$$

This is the basic flow field equation which can then be solved by numerical techniques via utilization of a large scale digital computer.

A simplification of equation (14) can be developed by the recognition of vorticity, defined as

$$\omega = \nabla \times \mathbf{V} \tag{15}$$

and its introduction into the final flow field equation. By specifying zero vorticity in the field, the flow field equation reduces to

$$\frac{\partial}{\partial r}\left(\frac{1}{\rho r b}\frac{\partial \psi}{\partial r}\right)+\frac{\partial}{\partial z}\left(\frac{1}{\rho r b}\frac{\partial \psi}{\partial z}\right)=0 \tag{16}$$

that is, the equation defining an irrotational flow field. In effect, the working equation, (14), is that describing the rotational turbomachine flow field and the numeric evaluation of the right-hand side defines the distribution of vorticity within the field. To take full advantage of this approach a meaningful correlation between vorticity and irreversibility is necessary, a matter not completely within grasp at the present time.

The development of the general equations is in effect an exercise reflecting the various mathematical manipulations and assumptions necessary to arrive at the final concise flow field equation. It is to be noted, however, that the judicious selection and application of the assumptions and the guiding knowledge as to the effect of the assumptions on the turbomachine application is necessarily based on extensive prior experience in this field.

The relative importance of any aspect of these development procedures must be finally established by the experimental verification of the design assumptions.

The solution of this system of flow field equations is best obtained through use of high speed digital computers, which solve the flow field relationships and include the variabilities of steam properties as represented by the steam table formulations [2].

2. A particular solution of the general equations

A particular solution of the general equations may be obtained by enforcing a design restriction on the flow field such that the condition of irrotationality is maintained. This assumption of zero vorticity resulted in the flow field equation (16). This equation may then be solved by a variety of techniques, it is the purpose of this section to describe one such technique and its application to the design of the research turbine.

The evaluation of the flow field's aero-thermodynamic properties can be obtained by application of the electrolytic tank analogy to this irrotational version of the turbomachine flow field equation. It has been shown [5] that two tank analogies exist both yielding the same fluid mechanic results, these are shown in Fig. 2. As it is easier to measure the voltage distribution in the electrolyte rather than the current distribution it is preferable to use analogy B in which equipotential lines correspond to the turbine meridional streamlines.

The pertinent electrolytic tank equation is

$$\frac{\partial}{\partial x}\left(H\frac{\partial E}{\partial x}\right)+\frac{\partial}{\partial y}\left(H\frac{\partial E}{\partial y}\right)=0, \tag{17}$$

where the depth (H) of electrolyte is proportional to $1/\rho rb$, where again b is introduced to account for blockage, and the correspondence of flow field velocities to the electrolytic tank electropotential distribution is

$$V_r \propto -\frac{1}{\rho rb}\frac{\partial E}{\partial x}, \tag{18}$$

$$V_z \propto \frac{1}{\rho rb}\frac{\partial E}{\partial y}. \tag{19}$$

Construction of the tank analog can be accomplished at low cost. The electrical system is also simple, with however, care being taken to provide a fine wire for sensing the potential distribution.

The method at first glance may seem to be a direct solution of the irrotational flow field, however, the model must be constructed with finite depths to faithfully represent the distribution of field densities; the radius and blockage are known everywhere. The technique in effect demands an estimate of the solution before a compatible result can be achieved. At this point in the design process the turbine stage duties and work distribution will have been specified. These will allow the determination of the total enthalpy and tangential velocity in the spaces between blade rows. As these are the strongest factors in defining the density distribution, their specification will allow a good initial estimate to be

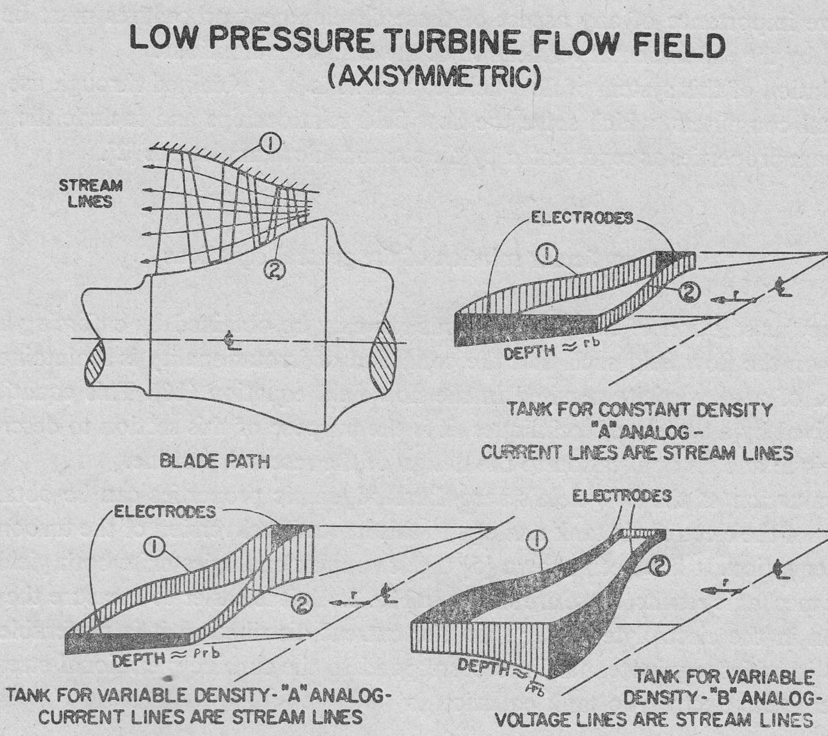

Fig. 2. The electrolytic analogy and turbomachinery application

made. An initial approximation of the density distribution can be made through use of the simplified radial equilibrium equation, that is, the simplification of equation (9) to

$$\frac{1}{\rho}\cdot\frac{\partial p}{\partial r}=\frac{V_\theta^2}{r}, \tag{20}$$

where the V_θ distribution is known. The resulting potential distribution then defines a refined distribution of density, and a subsequent tank bottom can be constructed. The process converges so rapidly that, often, a single model is sufficient to achieve a compatible flow field solution within the uncertainties of the experimental apparatus.

3. The research low pressure steam turbine

As with any theoretical declaration, and with special regard to its practical implications, verification is required by experimental means. To verify the design approach in all its ramifications is of course an impossible task, but to yield meaningful practical conclusions the more important aspects of the approach can be evaluated. For this purpose a three stage research steam turbine was designed, constructed, and tested in a steam turbine development laboratory.

The overall turbine parameters of flow rate, pressure ratio, speed, and power were chosen to match the facilities design constraints which have been reported in reference [6]. A three stage turbine with last stage characteristics similar to production units set the major geometric dimensions and relationships such as hub/tip ratio and maximum tip speed. The hub/tip ratio of the first stage turned out to be low enough to require attention to the variation of its blade section orientation with diameter, resulting in a low pressure turbine design containing three stages of twisted and tapered blading. A deliberate effort was made to minimize the overall active blade path axial length. This induced a large stream tube divergence which in the middle stage resulted in a flow angle of 52 degrees with respect to the turbine axis.

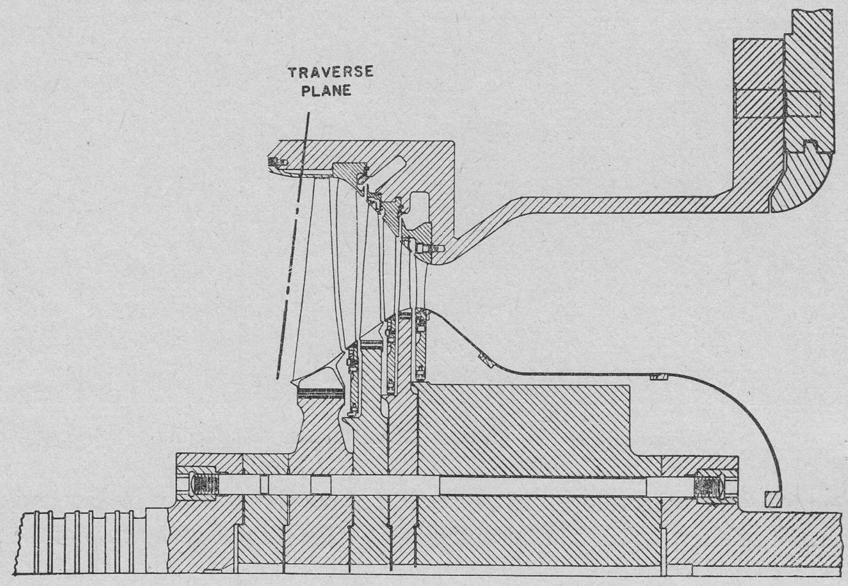

Fig. 3. The research low pressure steam turbine

A smooth blade path contour is apparent in Fig. 3 which shows a longitudinal section of the research turbine. In addition to the extreme flow divergence in the blade path other particular features are worth noting in the last stage. The rotor blade is smoothly contoured at the hub to accept a radial inflow and to exhaust nearly axially into the exhaust system diffuser. To be compatible with this concept the last stator tip region performs this function allowing the rotor tip to sweep a cylindrical path and to seal properly when axially displaced. The upstream rotor blades are sealed on tip shroud cylindrical surfaces in order to maintain the conical stream tube divergence in the blade path. In general, care was taken to minimize internal flow irreversibilities in order to more exactly establish the measured turbine performance as that of the pure blade path unencumbered as much as possible by parasitic losses.

On the upstream end a large plenum provides a low velocity region capable of delivering a circumferentially uniform steam supply to the test turbine. The exit diffuser ducts the exhaust steam into a collecting hood and then into the facility condenser. Fig. 4,

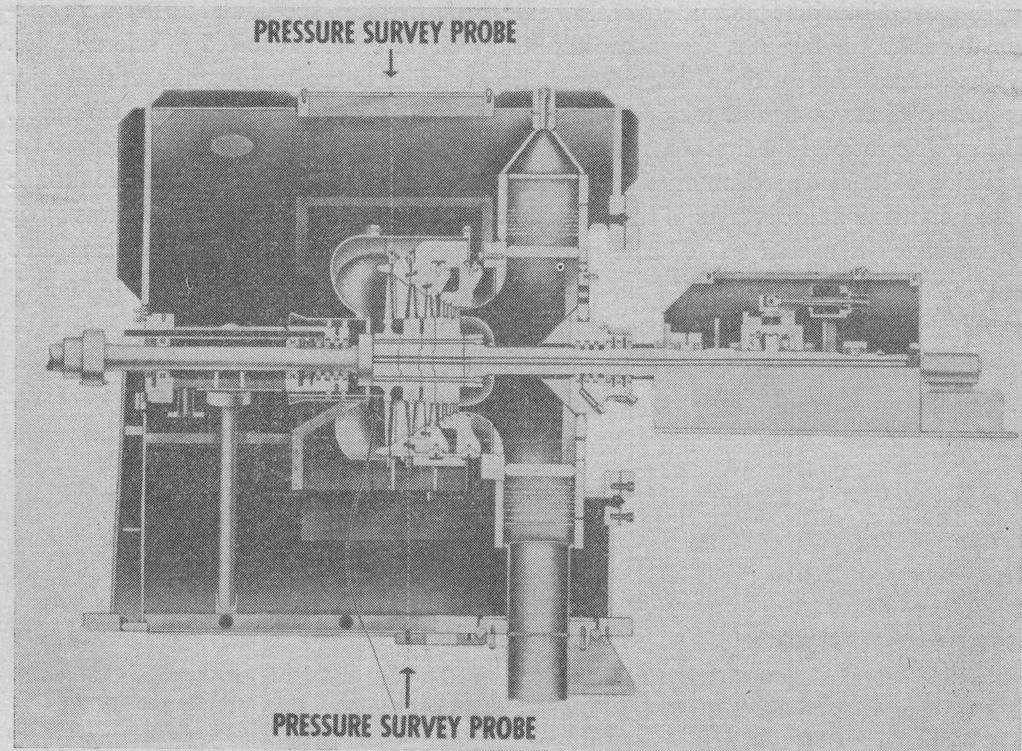

Fig. 4. The low pressure steam turbine test facility

taken from [6] – presents a longitudinal section of the low pressure facility and the relative placement of a test turbine; in this figure a six-stage turbine is shown, not the research turbine.

The aerodynamic design of the research turbine was determined by means of the irrotational design technique. The electrolytic tank approach provided an initial flow field solution which was later verified by a digital computer solution. Subsequent rotational solutions were achieved by means of an advanced version of the digital computer program and when compared to the turbine test data showed closer agreement with the measured quantities.

4. A comparison of test results to design parameters

Following the design of the three stage research turbine, an extensive test program was undertaken to evaluate its aerodynamic performance over a range of operating conditions. The test facility, being quite flexible in its operating characteristics, could impress a range of exhaust pressures on the exhaust system nearly independent of the condensing heat load requirements of the turbine mass flow. Inlet conditions could also be varied to subject the turbine inlet to a range of steam temperatures and pressures, and hence the turbine to differing mass flows, pressure ratios, and power output.

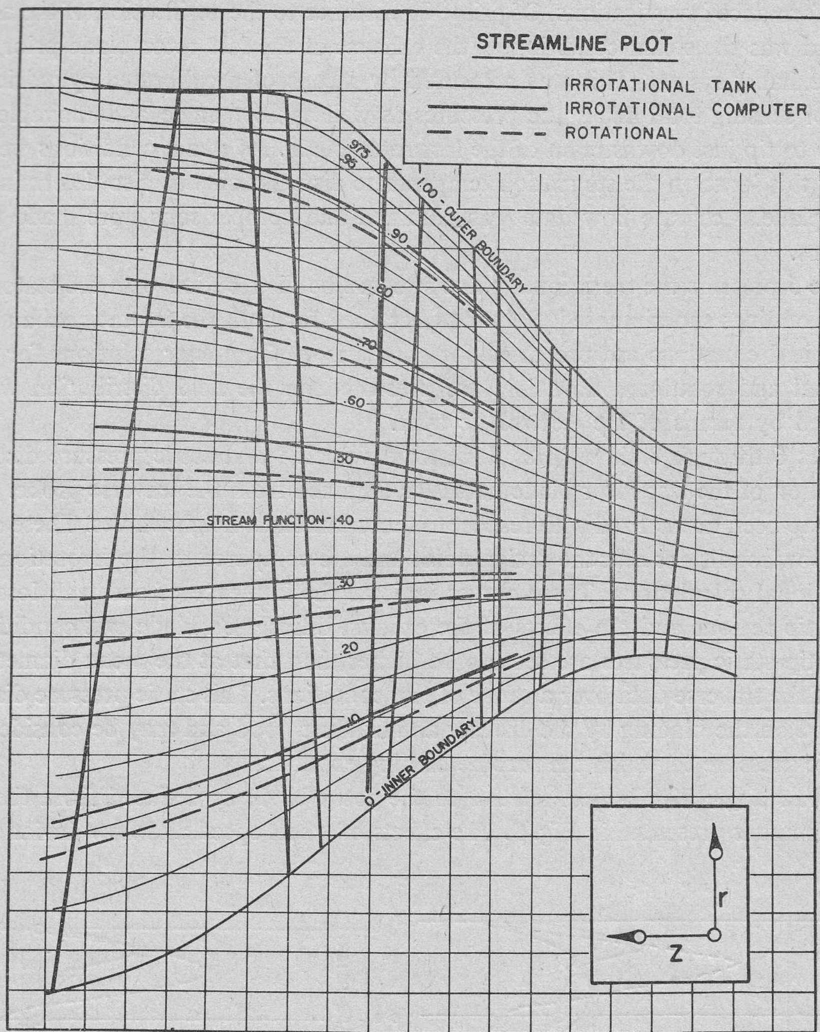

Fig. 5. Meridional plane streamline plot

A prime parameter in low pressure steam turbine operation is the pressure ratio the last stage is subjected to, or in other terms, this pressure ratio defines the turbine exhaust volumetric flow as well as its leaving absolute and axial Mach numbers. Field conditions impose a wide range of exhaust volumetric flow with a less radical variation in upstream pressure levels and attendant mass flow. The test program was structured to investigate the effect of these several variables. The moisture level as such was eliminated as a variable in the desire to obtain tested data unencumbered by these additional uncertainties.

The turbine's overall performance was determined by those primary measurements of mass flow rate, brake power absorption, steam inlet conditions, and exhaust pressure. Measurements of secondary importance determined the system losses, such as shaft bearing losses and miscellaneous shaft leakages. No interstage extractions were taken during these performance investigations. Of prime importance to the verification of the aerodynamic design was the steam condition at the exhaust of the last rotor blade prior to entry into the exhaust system. This was measured by means of a calibrated cylindrical probe capable of sensing total and static pressures as well as flow angle and movable radially from hub to tip just downstream of the last rotor blade. As steam conditions were superheated at this location the stagnation temperature distribution was also determined at the traverse plane. Accurate flow data was obtained and comparisons were made to design predictions.

Figure 5 presents the meridional plane stream line distribution with the stator and rotor blade outlines superimposed. Solutions obtained for both the irrotational and rotational flow field equations are presented for comparison. Computer solutions for both the irrotational and rotational flow field are presented on the field distribution which was determined by means of the electrolytic tank.

Figures 6 through 8 show mass flow, axial velocity, and static pressure distributions as a function of the last rotor blade height as obtained from the traverse probe. All parameters have been normalized with respect to the mean blade height values. These data were taken at flow conditions near the design point and are compared to the irrotational design and rotational calculations. These curves indicate the influence of end wall losses in the hub and tip regions and the effects of the exhaust diffuser on blade exit conditions. The hub and tip static pressures are seen to be higher than that at the mean diameter and is typical of the traverse data over a range of test conditions. This static pressure distribution is imposed on the blading by the diffuser and exhaust hood and may be considered to be that static pressure to which the turbine blading expands.

The flow angle distribution relative to the rotor blade as deduced from the traverse measurements is presented in Fig. 9. The effects of wheel speed and flow deviation due to

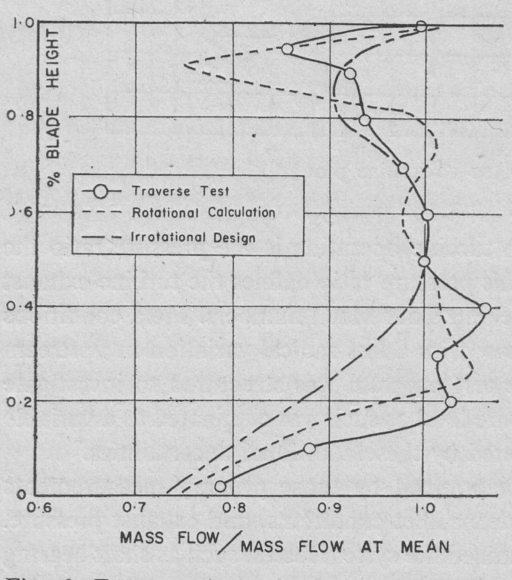

Fig. 6. Traverse plane mass flow distribution

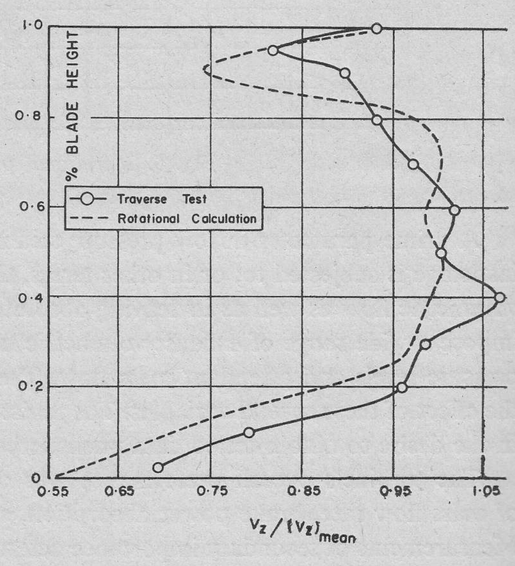

Fig. 7. Traverse plane velocity distribution

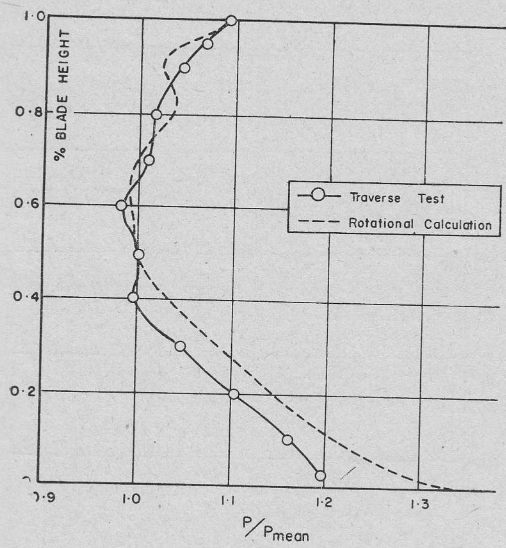

Fig. 8. Traverse plane static pressure distribution

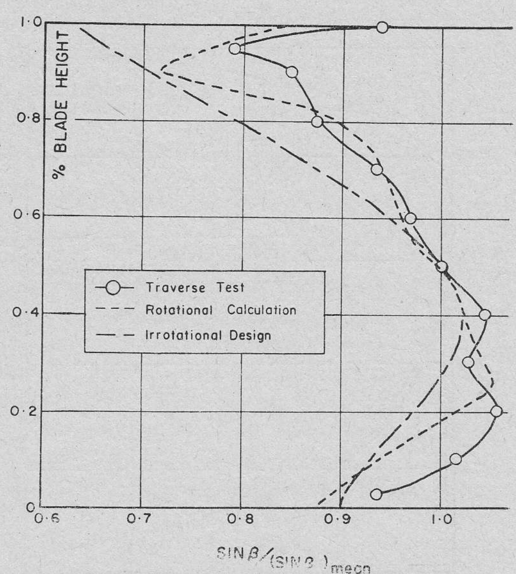

Fig. 9. Rotor blade relative flow angle

local supersonic expansion have been taken into account for this plot. It can be seen that deviations were experienced in the tip region starting approximately at the 70% blade height location with still additional variation at the tip 5% of the blade. This latter effect is due primarily to the leakage of steam across the blade tip clearance. The major portion of the remaining deviation can be explained by the untwisting of this blade due to centrifugal forces impressed at the high angular velocities of the research turbine.

These comparisons of test data to that determined by the rotational flow solution reflect better estimates of blade section and secondary flow losses which are included and continually reviewed during the iterative solution process. Agreement is good between test and calculations for the axial velocities and static pressures with again the same characteristic of higher pressures at the hub and tip. The mass flow distributions for the test, rotational, and irrotational calculations show a tested mass flow distribution higher than the irrotational calculation for the lower portion of the blade, this was typical for most of the test points. The agreement between test data and the rotational calculation is better in the hub region.

Figure 10 shows the total temperature variation as measured at the last rotating blade exit, the cumulative effects of end wall losses and leakages are clearly demonstrated. The temperature distribution for the rotational flow calculation is also shown.

A necessary facet of the design process is the realistic assessment of blade section losses in combination with those due to secondary flow and leakage effects. These effects as such manifest themselves throughout the design process and influence the definition of the blading inlet and exit angle distributions. The overall performance is of course directly affected by their magnitude. Figure 11 presents a plot of the overall aerodynamic performance of the research turbine normalized by a predicted expansion line efficiency and presented as a function of average turbine exit axial Mach number. The efficiencies

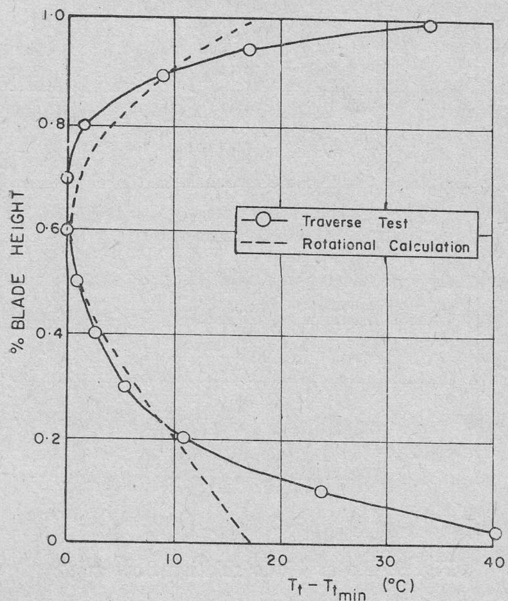

Fig. 10. Traverse plane total temperature distribution

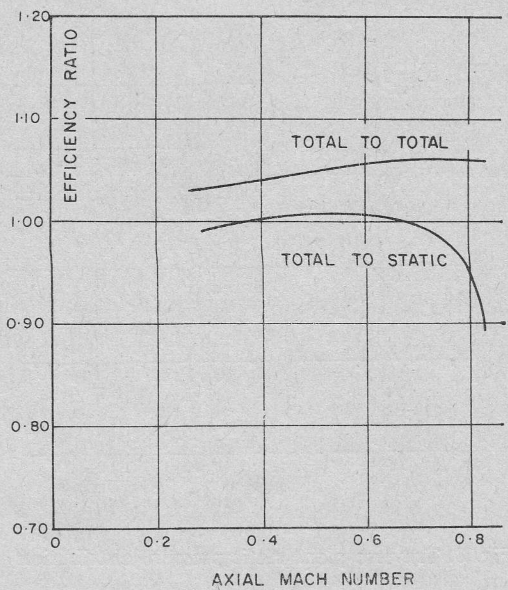

Fig. 11. Overall performance of the research turbine

are taken from the turbine inlet total conditions to the mass flow weighted values of exit total and static pressures. The total to static efficiency can be seen to peak slightly above the predicted level at an axial Mach number of approximately 0.55 with the total to total efficiency continuing to increase and to peak at approximately 0.8 axial Mach number. The difference between these efficiency definitions is of course due only to the kinetic energy of the exhaust stream.

5. Conclusions

The development of a quasi three-dimensional turbomachinery flow field design process has been traced from fundamental considerations to a system of design equations. Application of this design procedure was made to a three stage low pressure research steam turbine featuring significant streamline divergence and attendant radial flow. The turbine was erected and tested in a low pressure steam turbine test facility with particular emphasis placed on the accurate determination of its overall performance and turbine exit flow conditions.

The turbine performance on an overall basis was quite close to the predicted level as obtained by simplified predictive techniques. The subsequent detailed evaluation of the traverse data indicated areas which deviated from the chosen design approach. These variations were more completely assessed by means of an advanced flow field technique which yielded calculated values more closely aligned with the turbine test data.

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Projektowanie wysokosprawnych, niskociśnieniowych turbin parowych metodami aerodynamicznymi

Streszczenie

W pracy przedstawiono sposób projektowania niskoprężnych turbin parowych uwzględniający podstawowe zależności teoretyczne, wyniki badań aerodynamicznych oraz wyniki badań przeprowadzanych na eksperymentalnej turbinie parowej, służącej do weryfikacji metod projektowania.

Opracowano i porównano alternatywne techniki rozwiązywania równań dla pola przepływu. W opisie turbiny doświadczalnej szczególną uwagę zwrócono na stosowane rozwiązania mechaniczne i system oprzyrządowania. Dalej opisano wyniki badań charakterystyk turbiny. Wyniki przeprowadzonych prób porównano z warunkami założonymi w projekcie, zarówno w odniesieniu do ogólnych osiągów turbiny, jak i danych z badań parametrów w kierunku promieniowym.

Аэродинамическое проектирование паровых турбин НД больших мощностей

Резюме

В работе представлен подход к аэродинамическому проектированию паровых турбин НД, от фундаментальных теоретических рассуждений и прикладных исследований, до проектирования, испытания и анализа результатов исследований на экспериментальной паровой турбине, специально запроектированной для проверки этих современных методов проектирования. Обработаны и сравнены с собой альтернативные техники решения уравнений течения. В описании экспериментальной турбины особое внимание уделяется сообразному решению механической части и измерительной системе. Затем описываются результаты исследований характеристики турбины. Полученные результаты сравниваются с расчетными предположениями, относительно мощности и распределения параметров течения по радиусу.