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Ideas and Methods of Turbomachinery Aerodynamics: A Historical View

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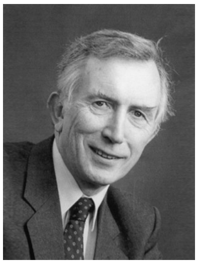
Most accounts of the history of turbomachinery look at the machines that were produced. This paper looks at the underlying ideas behind the designs and the methods that were used. It will be seen that in this history the inventiveness of engine designers far outstripped the capabilities of these methods and, further, that this gap spurred the advancement of new ideas. Thus, although analyses were unable to capture many features of the flows, this has not precluded aeroengine turbomachinery from being successfully developed using methods that were far from a complete description. The paper concludes with a look at two different ways in which engineers have dealt with some turbomachinery aerodynamic issues that seemed to offer major difficulties at the time.

I. Introduction

ALTHOUGH it is sometimes claimed that engineering is synonymous with applied science, the turbine, the gas turbine, and the jet engine furnish ample evidence that this is not true. More specifically, the engineering of these devices has run far ahead of the knowledge base that one could reasonably consider science. The goal of this paper is to illustrate the way in which engineers have successfully addressed the design of jet engine turbomachinery even though they were much removed from complete understanding of the processes taking place, noting that even now it is not possible to calculate all of the details of the actual flows. The authors have their background in compressor aerodynamics and the paper reflects this. Because the threads that link the topics are the models and methods for describing the flow in turbomachines, the paper has applications to turbines as well.

To put the scientific problem in context, even a cursory look at the workings of a multistage compressor or turbine conveys the difficulty of an exact flow description. The adjacent moving and stationary blade rows typically have different numbers of airfoils, so that each pair of rows has a continuous variation in (combined) geometry as the rotor moves, with a different variation for each blade row pair. In addition, the movement of one row relative to the other makes the process unsteady. Inward along the span of the blades, the geometry is predominantly two-dimensional, in what is, for convenience, often referred to as the blade-to-blade plane; however, there is also a variation in blade shape in the radial direction. The geometry and flowfield properties, therefore, are spatially three-dimensional. Combined with the unsteadiness there are, thus, four dimensions to the flow. These four-dimensional motions are now starting to be analyzed, although, as will be discussed, what



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this means in terms of the understanding of the flow is not yet resolved.

There have been several review papers describing the history of jet engine technology, including advances that have been made in turbomachinery.[1–9] These have generally highlighted the changes in the attributes and capabilities of the devices. This paper is different in emphasis and focuses rather on the evolution of concepts and tools (analytical, numerical, and experimental), including the manner in which these were used to enable successful designs. The majority of the discussion is concerned with the ideas and the calculation methods, rather than the experimental techniques because, when viewed as an enabler for aeroengine turbomachine development, the latter have changed relatively little compared to the former. This will be revisited in a later section.

There are three main objectives of the paper. One objective is to describe the progression of the identifiable new concepts and tools that have been used to enhance the ability to grapple with turbomachinery aerodynamics. Our view, and the way the paper is structured, is that there are a few seminal ideas that have allowed major advances. A second objective is to provide the contextual background, that is, the major differences between then-existing powers of description and flow features of interest. The third objective is to present some reflections on the problems as seen then and now, in the sense of contrasting what difficulties were real and what were imagined.

II. One-Dimensional Treatment

An early, and striking, simplification adopted when turbomachinery aerodynamics were involved was consideration of the flow in a stator in a stationary frame of reference (that is, one fixed to the stator) while consideration was also given to the flow in a rotor in the frame of reference that rotates with the rotor. It was assumed that effects of the next row are seen in a time-averaged manner, so that conditions are taken to be axisymmetric upstream and downstream of each row and so that the flow was, thus, steady in both the stator and the rotor. Velocity vector relationships, usually referred to in this context as velocity triangles, could then be drawn to transfer the flow from stator outlet to rotor inlet and rotor outlet to stator inlet. This approach has been so successful, and the assumptions so ingrained, that many engineers forget that they invoke assumptions of circumferential uniformity between the blade rows.

The use of velocity triangles and the change in the frame of reference can be traced back to the early steam turbines,[10] as indicated by Fig. 1. The approach is still used because the flow between the rows is treated as axisymmetric in present design processes. An important result, known as the Euler turbine equation, can be derived by combination of the steady flow energy equation for adiabatic flow with the expression for the torque exerted on the fluid in the annulus.

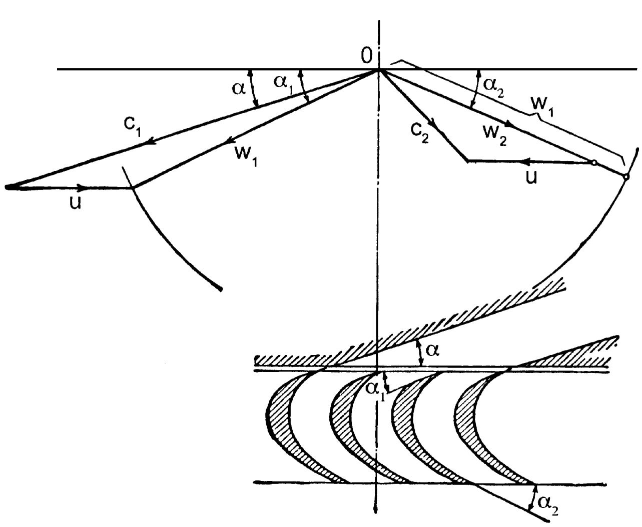

Fig. 1 Turbine velocity triangles at blade row inlet and outlet from Stodola[10] in 1904.

This yields a one-dimensional expression relating the difference in thermodynamic and kinematic properties across a blade row as

$$\Delta h_t = \Delta(U v_\theta) \tag{1}$$

In Eq. (1), h_t is the stagnation (or total) enthalpy, U is the blade speed, and v_θ is the tangential component of velocity. Equation (1) states that the change in stagnation enthalpy per unit mass (and, hence, the shaft work per unit mass) is equal to the change in blade speed multiplied by tangential velocity. Because U is zero for a stationary blade row, the stagnation enthalpy is constant across a stator.

Regarding the flow as steady in both the stationary and the rotating frame of reference (the latter often referred to as the relative frame of reference) and the use of velocity triangles to move from the stationary to the rotating frame, is now a standard part of turbomachinery aerodynamics. It appears in undergraduate courses, in research papers, and in industry. The extent to which this idea is remarkable is often overlooked, because to raise the stagnation enthalpy (as in a compressor) or decrease it (as in a turbine) requires that there must be a variation in static pressure with respect to time.[11] In other words, that the absolute flow must be unsteady and nonuniform. The equation describing the change in stagnation enthalpy for a fluid particle shows this:

$$\frac{\mathrm{D}h_t}{\mathrm{D}t} = \frac{1}{\rho}\frac{\partial p}{\partial t} \tag{2}$$

In Eq. (2), the term $\mathrm{D}h_t/\mathrm{D}t$ denotes the rate of change of stagnation enthalpy for a fluid particle and $\partial p/\partial t$ is the time rate of static pressure at a fixed point in the flow; the latter is positive for a compressor and negative for a turbine, with a commensurate change in the stagnation enthalpy as the particle moves through the blade row.

Early approaches to turbomachinery flow were built on the ideas of contemporary hydraulics, and so the blades (for example for the early steam turbines) were simply straight lines and circular arcs.[12] The variation in the radial direction was frequently ignored, even though the blade speed, needed in the velocity triangles, is proportional to the radius. In total, therefore, the flow was approximated to one dimension and viewed basically as a steady channel with losses.

The one-dimensional approximation was satisfactory for steam turbines generating electricity and propelling ships; given that the standard of comparison was reciprocating engines, this is perhaps not altogether surprising. Difficulties arose, however, when attempts were made to build axial compressors. The efficiencies realized were less than 60% and, in hindsight, it is evident that the blades were seriously stalled. In England it was Griffith who is credited with seeing that the low efficiency was the consequence of the unrealistic amount of turning the blades were intended to produce.[13] He had noted that propellers, which have little turning, had much higher efficiency, and he designed an axial compressor stage with blades giving only small deflection. The measured efficiency was about 90% (though the accuracy of the measurements was not high), which demonstrated that efficient compression was possible in an axial machine. Note that at that time (late 1920s), boundary-layer theory was still relatively new, and many practitioners of fluids engineering were not familiar with the ideas.

From an aerodynamic viewpoint, the compressor has been more problematic than the turbine. Many engineers in the 1930s and 1940s believed that the axial compressor was impractical because of the difficulties in devising a machine that could be matched from front to back. Many stages would be required to produce a large pressure ratio and if there were a small error in the first stage, the mismatch would be cumulative, thus creating a large error in the velocity into the last stage. There were also concerns about the boundary layers on the hub and casing; if the boundary layer on the blades themselves were close to separation, surely the boundary layers on the endwalls would separate. Despite the inability to calculate the endwall boundary layers, however, by 1939 the Brown Boveri Company of Switzerland had developed gas turbines with axial compressors. By the end of the war the Germans had jet engines with axial compressors in service, and the British had flown a jet engine with an axial compressor.[14]

As mentioned above, the use of velocity triangles and one-dimensional analyses continues to this day, particularly in preliminary design and concept screening exercises, though now the one-dimensional treatment would be at a specified spanwise positions with the appropriate blade speed and inlet and outlet velocities. The one-dimensional approach is used in conjunction with empirical correlations for properties such as allowable turning, static pressure rise (or fall), and loss in total pressure.

The one-dimensional approach has also been extremely useful in explaining and determining choking flow, which becomes critical as the flow velocities are increased. Blade row choke is not in itself harmful, provided that, when it occurs, it has been allowed for in the design. Blade row choking when not expected normally has serious consequences for efficiency.

The one-dimensional treatment also appears in another aspect of compressor technology: consideration of the limitations on compressor stability that result in rotating stall and surge. The treatment of these phenomena in turbomachines, which must explicitly include description of the unsteadiness, is at a more rudimentary level than is well-conditioned unstalled flow. As such, much useful information has been obtained with models in which the flow in a blade passage (or even in the whole annulus) is dealt with as one-dimensional.[15]

III. Two-Dimensional Treatment

By the 1930s, many steam turbines had aerofoil-shaped rotor and stator blades (in the blade-to-blade plane), although blade profiles formed from straight lines and circular arcs were also used. The axial compressors almost always used aerofoil shaped blades. Furthermore, in recognition of the spanwise variations in blade speed (a practice taken over from propeller design), twisted blades were used to keep the incidence, which was calculated from velocity triangles, small all along the span. (Incidence is the angle between the inlet flow direction and the direction in which the forward part of the blade is pointing.) This was a period of intense activity in Germany and Britain, with the approaches differing sharply.

In the 1930s Germany (in association with ETH in Zurich) had a more advanced knowledge base in aerodynamics, which they brought to bear in designing compressor blades. Analytical methods, based on thin airfoil theory, enabled them to use information from wing aerodynamics to design blades in cascade.[16] To use these methods, however, it was required that the camber of the blade (the difference in direction of the blade at entry and exit) was small. Given the desire to achieve high values of the quantity $\Delta h_t/U^2$, this implied that the blades should be highly staggered (that is, the blade should be highly inclined toward the tangential direction). The analytical approach also ran into difficulties when the relative Mach number approached unity and could not allow properly for the boundary-layer effects.

The British adopted a different, more empirical, approach to designing compressor and turbine blades. The starting point was the idea that the blades should be considered as passages, with the turbine treated as a nozzle and the compressor as a diffuser. For the compressor blades this meant that the pressure rise in a blade row would be limited to a fraction of the inlet dynamic pressure. Small cascade wind tunnels were built in which blades could be tested to see how they performed. (We now know that these tests had many unsatisfactory aspects, like very low Reynolds number and severe blockage on the endwalls.) These results provided a reasonable basis for design and, based on these tests, a set of guidelines were built up. These guidelines were expanded over the years and the correlation that relates allowable stagger, turning, and solidity (how close together blades should be) is usually referred to as Howell's correlation, whereas the correlation for flow deviation (the difference between outlet flow direction and blade outlet direction) is normally referred to as Carter's rule.[17] It is remarkable that these empirical rules, though modified, are still used to this day as a general check on a new design. The passage view of cascade behavior may seem obvious in hindsight, but other approaches continued into the 1960s (Ref. 18).

Early work on compressors put great emphasis on the blade profile shape used. The most widely used was the NACA-65 section, but the British C-sections were used extensively, as were the simple double circular arcs. Cascade tests at NACA, reported in NASA SP-36, showed that there was little difference in loss or operating range between stall and choke, though the choking was worst for the C4, better for the NACA-65, and best of all for the double circular arc. Although the tests showed that all of the profiles had similar loss, this did not provoke the realization that most details of the profile shape were unimportant. The inlet and outlet flow angles mattered, and so did the maximum thickness and the leading edge shape. However, apart from these, a wide range of profile thickness and camber distributions would be comparable in performance.

After World War II, NACA became deeply involved in compressor design. New correlations for blade row performance were adopted, based on improved cascade tests. The most important of these is the diffusion factor, which relates stagger, camber, and solidity to blade aerodynamic loading.[19] With b the blade chord, s the pitch, V the relative velocity, v_θ the tangential velocity, and 1 and 2 denoting the inlet and exit stations, the diffusion factor is defined as

$$D = 1 - V_2/V_1 + (s/2b)\left(\left|v_{\theta_2} - v_{\theta_1}\right| \Big/ V_1\right) \tag{3}$$

Equation (3) shows the two effects that limit the blade row pressure rise. The first two terms represent the one-dimensional diffusion and the third represents the effect of the cross-passage pressure gradient associated with the turning of the flow from inlet to outlet. Measurements showed that blade losses could be correlated as a function of the diffusion factor, with the losses rising rapidly when the value of this quantity exceeded roughly 0.6. The diffusion factor is still in wide use today as a preliminary design tool and as a guide during other stages of design.

Corresponding criteria were also developed for turbines. One of those that is still used to characterize blade row aerodynamic loading is that of Zweifel (see Refs. 20 and 21), defined as

$$Z = F_\theta \Big/ \left(p_{t_1} - p_2\right) b_x \tag{4}$$

where $F_\theta = \rho v_x s(v_{\theta 2} - v_{\theta 1})$ is the tangential force, $p_{t_1} - p_2$ the difference between inlet stagnation and exit static pressure (for ideal incompressible flow, just the dynamic pressure at blade exit), and b_x the axial chord. It was found that the optimum efficiency was obtained with values of Z, the Zweifel loading coefficient, near 0.8. From this, the designer could select the optimum pitch chord ratio for a given turning of the flow.

It was commonly found that the minimum loss for a compressor cascade rose as the Mach number increased, the changes becoming pronounced above an inlet Mach number of about 0.7. More serious for the designer of compressors, the range between choke (or negative incidence stall) at negative incidence and stall at positive incidence shrinks as Mach number rises. This is illustrated in Fig. 2,

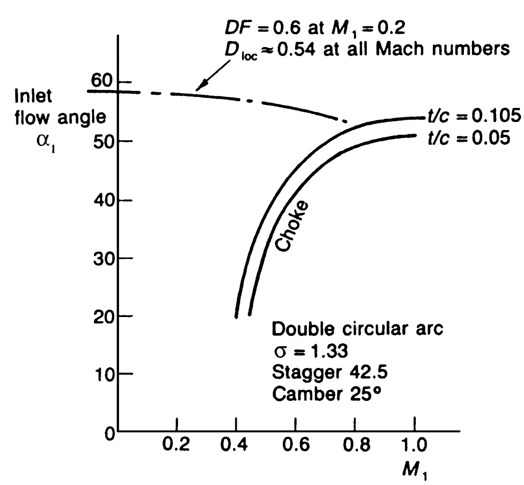

Fig. 2 Inlet flow angle vs inlet Mach number for cascades of double circular arc compressor blades, lines of choke at different thickness-chord ratio and line for constant local diffusion factor for the thicker blade.[17]

which shows the stall and choke limits for a given cascade of blades. The upper line is indicative of stall. (The condition for blade stall is not well defined, but it is often taken as condition at which loss is double the minimum.) The lower line is the choke boundary, which is a definite line that cannot be crossed. The usable range narrows sharply as the Mach number rises and can be as small as a two or three degrees at inlet Mach numbers near unity. This narrowing of useful range between choke and stall is one of the most important properties of compressor blades, with a large impact on all aspects of high-speed compressor design.

The work at NACA during the mid-1950s characterizes what was probably the high point of the development of semi-empirical models for two-dimensional blade row description. Work on this aspect did continue, mainly inside companies, but the main advances have come through the two-dimensional, and later three-dimensional, computational procedures to be described hereafter.

Most conventional profiles of compressor blades have a tendency to separate the suction surface boundary layer before the trailing edge, and with the conventional family of compressor blades there is little to be gained in trying to design the shapes numerically. A major alteration in thinking, enabled by the development of computational procedures for transonic potential flow, was the use of prescribed velocity distribution blades-that is the selection of a pressure distribution along the suction surface that would not cause the boundary layer to separate and would allow the calculations to proceed. Since the 1980s, aeroengine compressor blades have had profiles of the this type,[22] which produce lower losses by tailoring the pressure gradient on the suction surface to avoid separation by relaxation of the pressure gradient as the boundary-layer thickens. Virtually all turbine blades are tailored in the sense of being designed in relation to the boundary-layer behavior.

IV. Start of Three-Dimensional Thinking

A. Overall Flow Pattern in the Annulus

The most primitive treatment of the change in flow quantities in the spanwise direction was to allow for variation in blade speed when the blade stagger angles are set, given a uniform axial velocity. In the 1930s there was (at least in England) little recognition that the swirling flow between blade rows implied a radial pressure difference from hub to tip, although this is taught today to undergraduates. A biography of Whittle (see Ref. 23) has a chapter entitled "The Affair of the Vortex Blading" that recounts how a disagreement between Whittle and engineers at the company building his engine, BTH Ltd., led to the realization that the latter viewed the turbine nozzle exit flow as consisting of a row of discrete, essentially independent, jets at constant pressure. Hence, they did not account for any radial pressure difference and the exit pressures, velocities, and blade angles were not correct. To demonstrate that the exit velocity field was more nearly an axisymmetric swirling flow (with a consequent radial pressure gradient), Whittle carried out experiments with a sheet-metal nozzle ring; these put the disagreements to rest.

The basic reasoning for the existence of the radial pressure variation stems from the approximation that centripetal acceleration in a swirling flow is balanced by a radial pressure gradient; such a balance is generally referred to in the turbomachinery community as simple radial equilibrium. It is represented by a reduced form of the radial momentum equation as

$$\frac{\mathrm{d}p}{\mathrm{d}r} = \frac{\rho v_\theta^2}{r} \tag{5}$$

where r is the radius about the centerline of the turbomachine and v_θ is the tangential velocity. Simple radial equilibrium, as stated, is an approximation because it assumes negligible curvature of the streamlines in other directions and, hence, negligible radial accelerations except for the centripetal component. For a turbomachine with cylindrical hub and casing, simple radial equilibrium also implies that the radial shifts in streamline position, when viewed in the meridional plane [the plane composed of the axial (x) and radial (r) directions] take place within the blade rows, with the streamlines following a constant radius path between the blades rows (Fig. 3).

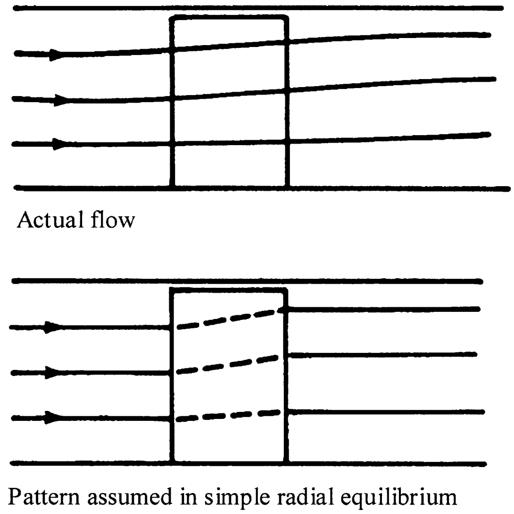

Fig. 3 Conceptual view of streamlines in cylindrical annulus for simple radial equilibrium assumption: upper line, casing and lower line, hub.

Equations (1) and (5) can be combined to give expressions for the stagnation pressure and stagnation temperature variations that illustrate other aspects of early turbomachinery design features. It was common to design for an approximately uniform distribution of stagnation pressure rise. If losses are uniform in the radial direction, this corresponds to uniform stagnation temperature rise. To maintain uniform stagnation temperature through the machine implies that the work in (for a compressor) or work out (for a turbine) is uniform radially. The Euler equation shows that if the work is uniform in the radial direction then, with Ω as the angular velocity of the shaft and r_1 and r_2 the inlet and outlet radii of a streamtube,

$$\Delta(U v_\theta) = \Omega\left(r_2 v_{\theta_2} - r_1 v_{\theta_1}\right) = \text{constant} \tag{6}$$

One way in which Eq. (6) can be satisfied is if the circumferential velocities obey the relation

$$v_{\theta_1}\big/r_1 = \text{constant}, \qquad v_{\theta_2}\big/r_2 = \text{constant} \tag{7}$$

in other words, if the flow upstream and downstream of the rotor has a free-vortex distribution of tangential velocity. [The requirement for uniform work is that the *difference* in $r v_\theta$ is uniform; this can be achieved[24,25] with tangential velocity distributions other than those in Eq. (7).]

For many years, the free vortex was viewed as the preferred distribution of whirl velocity. One reason for this was that, with uniform losses, the axial velocity is radially uniform. Designs that deviated from free vortex also made the calculations more complicated. Another apparent advantage comes from consideration of the circulation around the blades. If the flow is free vortex upstream and downstream, the circulation about each blade is radially uniform and there is no downstream trailing vorticity (analogous to wing tip vortices that are associated with an induced drag). Today, the use of simple radial equilibrium is restricted to teaching and to "back of the envelope" calculations to consider or check the impact of changes to the flow. However, for more than 20 years, this represented the cutting edge of turbomachinery; a lot of hard work with slide rules, then with calculators, and finally with computers went into doing the designs based on this. Figure 4 shows a chronology of compressor calculation methods used at General Electric (GE),[7] along with the engines and compressors that were designed with these methods.

A problem with the free-vortex design was that the losses were not uniform, and either more work had to be put in to compensate for this where losses were high, normally near the endwalls, or the axial velocity would fall. Another problem was the high relative Mach numbers at the rotor tips, leading to high loss with the blade profiles then in use. The restriction to free vortex also meant that the work exchange in the outer sections of the blades was limited. In spite of these disadvantages, and even though free-vortex design

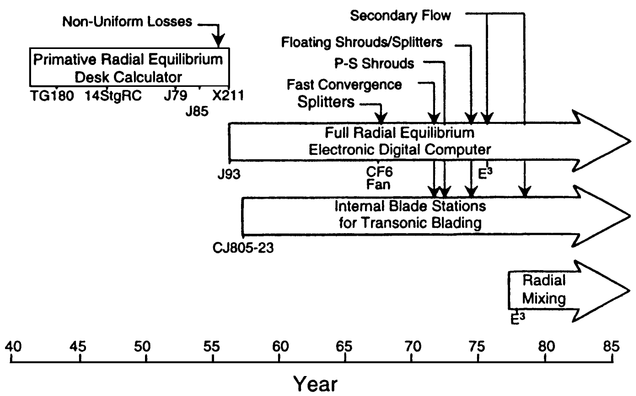

Fig. 4 Evolution of compressor vector diagram calculation methods at GEAE.[7]

led to compressor blades with considerable twist[25] and to negative reactions at the hub of turbines (so that the static pressure actually increased from inlet to outlet of the turbine rotor), this type of flow characterized a number of early engines. For example, Dring and Heiser[26] discuss the free-vortex design of early aeroengine turbines. Smith[7] describes the free-vortex design of the J85 compressor (although free-vortex design was unusual for GE even at that time, for the aforementioned reasons). Streamline curvature procedures, to be described below, allowed computations of flow with arbitrary swirl distributions and nonuniform axial velocity, so that designers for both turbines and compressors were able to break away from the limitations of free-vortex thinking.

B. Flow Pattern in the Endwall Regions

The difficulties that stem from flow in the endwall regions have an impact that goes considerably beyond the axisymmetric design problem just described. It is worthwhile to discuss turbomachinery endwall flows in a bit more depth because the issues inherent in their description are reflected in the broader problem of turbomachinery flows as a whole. Both computations and experiments show that the three-dimensional motions created have appreciable velocities normal to the endwall and, hence, pressure fields that are not simply that of the freestream. Thus, although initial views of endwall behavior drew heavily on boundary-layer ideas (which were well developed and successful in other areas of fluids engineering) the resulting methodology was not, in our view, very useful.

One important aspect of the problem, referred to as secondary flow, was reported in an investigation of cross passage flow found near the ends of the vanes at the corner of a wind tunnel.[27] The subject was generalized and applied to fluid machinery, notably by Hawthorne,[28,29] Smith,[30] and Lakshminarayana and Horlock.[31] The phenomenon can be qualitatively understood when the pressure and velocity interactions of a postulated inviscid *primary* flowfield are considered-that is, a flow defined by a mean streamwise direction, with nonuniform streamwise velocity. (A complementary explanation in terms of the differential convection of vortex lines has also been given.[17,32]) In the situation of interest here, the velocity nonuniformity is due to the entering endwall boundary layer.

Across a blade passage there is a pressure gradient between pressure and suction surfaces sufficient to deflect the flow in the freestream. If the freestream velocity is V and the average pressure gradient across the pitch of the blades is $\partial p/\partial n$, where n is the direction parallel to the endwall and normal to the streamlines, the normal component of the Euler equation is

$$\frac{\partial p}{\partial n} = \rho \frac{V^2}{r_C} \tag{8}$$

where r_C is the local streamline radius of curvature, and the flow is regarded as approximately two-dimensional and parallel to the endwall. For the endwall boundary-layer flow the velocity, denoted by v_{BL}, is lower than in the freestream. However, because the flow in the boundary layer is also nearly parallel to the endwall, the normal pressure gradient is, to good approximation, unchanged. In the boundary layer, therefore,

$$\frac{\partial p}{\partial n} = \rho \frac{v_{\mathrm{BL}}^2}{r_{C_{\mathrm{BL}}}} \tag{9}$$

where $r_{C_{\mathrm{BL}}}$ represents the radius of curvature for fluid in the boundary layer.

From Eqs. (8) and (9), the radius of curvature for streamlines in the boundary layer is

$$r_{C_{\mathrm{BL}}} = r_C (v_{\mathrm{BL}}/V)^2 \tag{10}$$

Because v_{BL} is smaller than V, the radius of curvature of streamlines in the boundary layer decreases relative to the radius in the freestream. In other words, the low velocity flow tends to overturn relative to the flow outside the boundary layer, acquiring a velocity component normal to the primary or freestream direction. With blades designed for freestream conditions, flow in the endwall boundary layer is overturned and leaves at an angle other than that intended. A classical secondary flow approach to treatment of these three-dimensional flows is encapsulated in this description, namely, the calculation of a small perturbation to a known primary flow. Even with this linearized approach, however, implementation of the theory becomes complicated for realistic configurations. (Secondary flow theory also implies, from two-dimensional continuity arguments, that the freestream is underturned. However, the changes in flow direction scale with the ratio of the boundary-layer thickness to the depth of the freestream region and are thus much smaller than in the boundary layers.)

Secondary flow ideas have been extremely useful for qualitative (and even quantitative) descriptions of some turbomachinery problems. Examples are the radial transport of hot fluid in a turbine,[33] in which a secondary flow treatment complemented three-dimensional computations, and mixing in compressors,[34] in which secondary flow considerations provided a means for scaling physical effects. Nevertheless, two main aspects mitigated against its use as a general design tool.

First, in turbines, the passage turning is large (typically more than 90 deg, compared to 20–40 deg for a compressor). Consequently, there are large cross-passage pressure gradients in turbine blade rows that imply that the secondary flow from pressure to suction side of the passage is sufficiently strong that nonlinear effects are important. For example, the cross-passage flows roll up into discrete passage vortices rather than "gently filling the entire passage."[35] An example of the result of such motion is given in Fig. 5, which shows measured contours of stagnation pressure at the inlet and exit of a turbine cascade.[36] The inlet contours are nearly parallel to the endwall, showing the incoming boundary layer. At exit, however, the low stagnation pressure is concentrated in a roughly circular region on the suction side of the passage. These flow structures, which are not captured by secondary flow analyses (but are captured by current three-dimensional computations) are important for loss, for the flow angle presented to the next blade row and for the movement of high-temperature fluid, which can cause local burning on the airfoil or platform.

The second problem with the use of secondary-flow theory is that in blade rows with tip clearance there is a strong influence of the tip clearance flow on the structure of the crossflow in the endwall region. Indeed, for a compressor tip, where the turning is generally small (less than 30 deg) it is not too strong a statement to say that the flow pattern is dominated by the clearance leakage. Evidence of the degree in which there was a gap in the ability to predict compressor endwall flows is that the correlative approach of Smith[37] was regarded as the state of the art for the behavior of endwalls in multistage compressors for more than two decades. In this treatment, a control volume analysis, combined with basic assumptions about the flow in a repeating stage environment, enabled a large amount of multistage compressor data to be organized in a way that could be used to estimate endwall blockage and loss. The structure of the flow in the endwall, much less the detailed behavior, was unknown

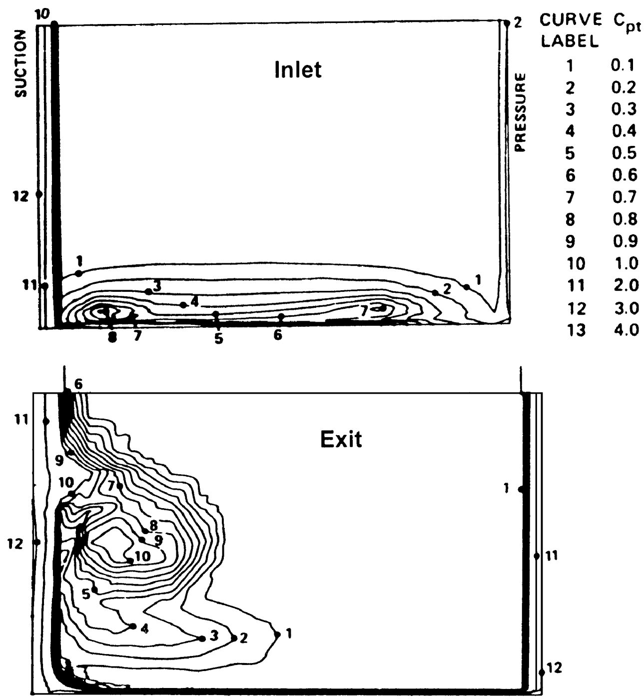

Fig. 5 Measured inlet and exit stagnation pressures in turbine cascade.[36]

(and not described) but the method substituted appropriate empirical information to provide the design parameters that were needed.

For the turbine, a complementary result was that the strong passage vortices seen in experiments also could not be captured by computational procedures until the late 1980s. Semi-empirical treatments of turbine endwall loss, suitable for the design process, were, thus, developed.[38,39]

The availability of three-dimensional computational flow procedures makes it possible to attack endwall flow issues in a much less empirical manner, and computations of tip clearance flows in multirow turbomachines are now common. As stated in the initial paragraph of the paper, however, long before these methods appeared, high-performance turbomachinery was being successfully developed.

V. Throughflow and Streamline Curvature

Simple radial equilibrium provides a good description when the blade span is small compared to the tip radius, for example, less than 20%. However, for compressors or turbines for which the hub-to-tip radius ratio is small, for example, the fan on the front of a jet engine or the late stages of a low-pressure (LP) turbine, the assumptions have less validity. More important, many of these machines have annulus walls that slope in or out at angles that are sufficiently large that radial accelerations other than centripetal cannot be neglected. An approach to axisymmetric flow was therefore needed that could account for streamline curvature in the meridional (radial r and axial x) plane.

The initial approach to include this effect was the actuator disk model, an idea for providing a representation of the effect of a blade row, which is reported to go back to the momentum theory of propellers of Rankine in 1865 (see Ref. 40). The actuator disk can be taken to be normal to the axial direction, so that the r and θ coordinates are in the plane of the disk.[41,42] The model also includes the assumptions that the radial velocity and the product of axial velocity and density are continuous across the actuator disk (the former because the disk exerts no radial force, the latter because the mass flow in an axisymmetric stream tube must be continuous). The tangential velocities, $v_{\theta 1}$ and $v_{\theta 2}$, are normally specified as the design duty of the blade row, and these are different across the disk.

The algebraic complexity associated with actuator disk analyses is large, especially when the hub and casing are not cylindrical (that is, the walls are not constant radius) and when the flow is compressible (the Mach numbers are high enough for the density to vary).[43] Although it absorbed a large amount of academic attention, the approach did not have a large impact on axisymmetric approaches to turbomachinery design in industry because methods better suited to numerical solution, described hereafter, became available. The overall concept, however, has been applied successfully in two other areas connected with turbomachinery aerodynamics. The first of these is for descriptions of asymmetric and unsteady flows in turbomachines, such as inlet distortion, rotating stall and rotor whirl, where the approximation has sufficient fidelity to show, in a relatively simple manner, important features of the flowfield. [For example, the most recent American Society of Mechanical Engineers (ASME) Melville Award paper[44] and ASME International Gas Turbine Institute (IGTI) Turbomachinery Committee best paper,[45] both of which addressed problems associated with asymmetric flows in turbomachines, used actuator disk ideas as the framework for the analytical approach.] The second is in conjunction with three-dimensional computations to provide good representations for boundary conditions upstream and downstream of the region of investigation with little computational effort.

Another approach beyond simple radial equilibrium was developed by Wu at NACA[46] in the late 1940s and early 1950s. He considered the flow on two sets of orthogonal stream surfaces, the S1 and S2 surfaces. At inlet S1 is a constant radius surface that warps as it passes through the blades, so that it is no longer purely constant radius at the exit. Warping also occurs for the S2 surface, which enters the machine as a meridional (x, r) surface. The equations on the S1 and S2 are coupled, and could (in principle) be solved together, and so the treatment is basically exact. At the time of this work, computers were inadequate to perform calculations of this type. When computers did become available it was decided that it was more useful to work on the meridional surface and the blade-to-blade surface, not allowing warp and twist to occur. Although this was an approximation (because the coupling terms were omitted), it was not the largest of the errors encountered. Later, when full three-dimensional methods became available, it was no longer necessary to adopt the stream surface approach. Although Wu's method was never applied as originally envisioned, its legacy remains in that the meridional plane is often referred to as the S2 surface and the blade-to-blade axisymmetric surface as S1.

The approach that had most impact is the throughflow method.[47] In this axisymmetric procedure the flow is solved separately on the meridional (x, r) plane and on the blade-to-blade surface at a number of radii along the span. On the blade-to-blade surfaces the flow is specified from correlations or obtained from two-dimensional calculations. The equations used in both of these computations are simplified; in the meridional plane it is assumed that the flow is axisymmetric, whereas in the blade-to-blade surface the meridional velocities are ignored, though the convergence or divergence of the meridional streamlines must be included.[48] The two types of surfaces used in design procedures are given in Fig. 6. The equations on the meridional plane and on the blade-to-blade surface

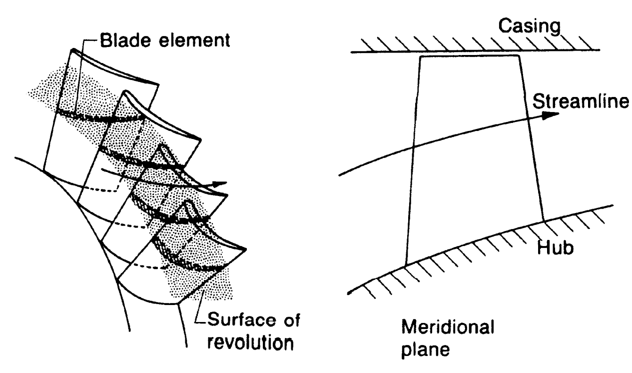

Fig. 6 Description of the turbomachinery flow in terms of two interacting two-dimensional flows: a) blade-to-blade surfaces and b) axisymmetric analysis of flow in a meridional (radial, axial coordinates) plane.

are solved separately. An iterative process between the two surfaces is possible, though rarely has this been done beyond one or two iterations.

Various methods exist for solution of the throughflow on the meridional plane, but the best known and most widely used is the streamline curvature method.[49] The process is based on the axisymmetric radial component of the momentum equation (which includes the effects from the curvature of the streamlines in the meridional plane). The solution procedure is to guess the meridional streamline pattern, to find from this the curvatures, and then to include the consequent pressure gradients in the equation. The method is very fast and generally robust.

The throughflow calculation is still a vital tool in turbomachinery design and analysis of results. Designers know it is not exact or complete, but there is a large body of experience that provides strong guidelines for what to believe and where to be cautious. In addition, one now has the ability to examine the conclusions using three-dimensional computations. The throughflow methods used in companies also have extensive associated empiricism, not only for the local blade row performance (losses, choking, and deviations) but also for such aspects as real-gas effects, spanwise mixing, leakage flow from seals, etc. The role of throughflow calculations, which were made useful for design by the computers of the late 1950s, is shown for GE engine compressors in Fig. 4.

Throughflow calculations also played a key role for turbines, in line with the comments about the move from free-vortex flow. As an example, Dorman et al.[50] report on the design of a so-called controlled vortex turbine in which the work done at the root was reduced (less turning) and the work at the tip was increased. In that paper (in which the only referenced publication is that to the streamline curvature method of Novak[49]) there is explicit mention of the degree to which this type of design was enabled by the advance in available design tools.

Although throughflow calculations were developed to account for meridional streamline curvature, there are few examples to show the strength of this effect in typical circumstances. As suggested by Cumpsty,[17] the effect may often be small in turbine and compressor stages, and, with hindsight, the greatest benefit over the simple radial equilibrium approach was the use of a rapid and systematic computational method.

VI. Three-Dimensional Methods

A design system that requires manual iteration between blade-to-blade and meridional surfaces is neither easy to use, nor suited to error-free operation. Secondary flows, referred to earlier in connection with the blade-to-blade flows, are difficult to accommodate in two-dimensional methods. There was thus great interest in the development of fully three-dimensional calculations for turbomachinery.

The two-dimensional blade-to-blade calculations that existed at the initial stages of this development were based on solution of potential flow or Euler equations, with a boundary-layer calculation added to give loss and blockage. The throughflow methods are inherently inviscid (though allowance for effects of spanwise mixing and estimated losses has been included). It was, therefore, perhaps natural (aside from the any considerations of computational capability) that the first three-dimensional methods should be inviscid and solve the Euler equations. Although this could not include computation of viscous loss, it could, in principle, furnish the flow pattern from which features of the flow could be deduced. Provided that the flow was made to leave the trailing edge in the correct direction (imposing an appropriate Kutta condition) this worked reasonably well for turbines, but was less satisfactory for compressors in which the freestream flow can be strongly affected by the growth of boundary layers and the tendency for them to separate. Furthermore, even if the boundary layers do not separate in a two-dimensional manner, the three-dimensional separations in corners can cause blockage, which reduces the effective flow area and, thereby, changes the pressure rise in a manner unacceptable for design.

The large increase in computational capability, however, has allowed viscous effects to be included, and the standard method now used is solution of the steady, three-dimensional Navier–Stokes equations. The form used is the so-called Reynolds averaged Navier–Stokes equations (RANS) in which the turbulence is included only in an average of the product of the turbulence quantities (the average denoted by brackets, for example, $\langle u'_x u'_\theta \rangle$) which give rise to effective stresses. All of these methods rely on models to represent the effect of turbulence on the effective viscosity; the models vary in complexity, but ultimately make use of some empirical information. A more recent development is so-called large eddy simulation methods that calculate (rather than model) the large-scale random motions so that the empiricism is confined to a description of the small-scale turbulence only.[51] Computer memory and speed capabilities currently preclude these for multiblade calculations.

Although computations are done on a multistage basis, the use of circumferentially uniform boundary conditions upstream and downstream of an individual blade row is currently common for design. It is important to include the radial variation appropriate to the upstream and downstream components, and this can be represented either with body forces or with an actuator disk. When a circumferential average of the real nonuniform flow is carried out it introduces extra nonlinear terms that appear as stresses in the equations of motion. These "deterministic stresses" are similar in algebraic expression to the Reynolds stresses from turbulence, but they are conceptually different because they are set by the steady blade flow in the upstream or downstream blade row and can be computed. The deterministic stresses have been described thoroughly by Adamczyk,[52] who has found that they are generally small, except when the blade rows have large regions of separation.

Aeroengine development has, until recently, had available only (at best) comparatively simple methods to deal with unsteady interactions between components or between blade rows. There are situations when the unsteady flow needs to be defined: important examples include problems of forced vibration (when the flow from one blade row excites another), flutter (a self-excited oscillation), and noise (the pressure waves propagating out of the turbomachine).

Now three-dimensional unsteady computations are growing in use. These have been used, for example, to track the movement of hot spots through a turbine or to examine the coupling between a fan rotor and the distortion in the engine inlet. One aspect of the modeling used in such situations concerns the blade number. Compressors and turbines seldom have the numbers of rotor blades and stator blades that are simple multiples of one another, and they are more likely to be, for example, 47 stators and 73 rotors. To carry out calculations with this combination of numbers requires that a full annulus be considered. If the numbers are altered to 48 stators and 72 rotors, however, the pattern repeats every two stators and the size of the calculation reduces accordingly; there is no evidence to show that such changes alter the flow pattern significantly.

At present unsteady RANS equations are used to describe unsteady behavior in compressors and turbines. It is often the case that the unsteady part forms a small perturbation to the mean flow, so if the steady flow is found with a complete nonlinear method, the unsteady flow can be found from a computation that is linearized about the steady flow. This normally brings large reductions in computational effort and opens up more complete treatments of problems with the finite resources available.

VII. Experimental Methods

A wide range of experimental methods has been developed for providing information about turbomachinery aerodynamics. Optical methods, such as laser anemometry and particle image velocimetry, allow measurement of flow quantities where solid probes cannot be used, for example, in the blade passage of high-speed rotors. Capacitance probes are used to measure the clearance over rotor tips while the machine is running. Tip timing can be used to find the untwist of the blades (a result of aerodynamic loading and the effect of centrifugal stress) and to find the amplitude, frequency, and phase of blade vibration. High-frequency (dc to, for example, 100 kHz) transducers allow the pressures associated with blade vibration, rotor–stator interaction, stall and surge to be measured. For low-speed research, hot wires and hot films are reliable methods of measuring velocity; in high-speed flows, hot films can provide measures of the skin friction. Tracer gas, with small, sensitive and fast-responding detectors can be used to study mixing and flow leakages. In an attempt to define the major contributions of this broad array of techniques, it is helpful to cast the discussion in terms of the engine development process and then in terms of aeroengine turbomachinery research.

A. Experimental Methods in Aeroengine Turbomachinery Development

Most measurements made in engines and in high-speed compressors and turbines resemble, in principle, those available 50 years ago. Mass flow is measured with an air meter. Stagnation temperature rise is normally found from local measurements with thermocouples (mounted on rakes or on stator blades), although resistance thermometers may be used at inlet to and outlet from the machine. Torque meters are also used to measure power. Stagnation pressures are measured with Kiel probes, either on rakes or on stator blades; static pressures are measured on the hub and casing walls.

What is different is that pressures are now measured with transducers (rather than mercury manometers), temperatures are found from thermocouples or resistance thermometers with electronic logging devices, and the information is recorded and processed digitally. Although more data are recorded more quickly and reliably, the general nature of the data is the same. The processing of these data with a throughflow program to turn pressures and temperatures into velocities, Mach numbers, flow directions and losses, required for understanding of the performance, is subject to the assumptions and approximations mentioned in the earlier discussion of the throughflow calculations. There are thus empirical inputs connected with the asymmetries that are embedded in such processing. This is true for both compressors and turbines, although there are additional issues with cooled turbines, for which the assumption of adiabatic flow along streamlines is unrealistic.

The preceding comments notwithstanding, we can mention three aspects in which experimental methods for turbomachinery aerodynamics have altered over the past several decades. One is that a modern test of a high-speed compressor or turbine in a rig, and even an engine, can include transducers to measure tip clearance and tip timing in an engine, allowing much closer connection between the aerodynamics and the mechanical configuration than in the past. A second is that the expense of rigs, coupled with the capabilities of computations, have resulted in a trend of fewer rig tests before going to the engine; sometimes (especially for small engines) going directly from design to engine. Third is that although most use of optical methods in high-speed stages have been in fans or single stages, these tests have provided information about such basic quantities as shock location. For multistage machines there are more problems in gaining optical access, which requires a clean window in the wall, and more difficulties with seeding the flow with particles fine enough to follow the flow, resulting in increased running time and cost.

B. Experimental Methods in Aeroengine Turbomachinery Research

Some of the changes in experimental methodology have already been described in connection with cascade testing. Cascade facilities are still in use at a number of research institutions. The nature of their use, however, has changed fundamentally. Rather than being the main route to performance information, they find most applicability in providing information about different phenomena (for example, clearance losses and flow structure) or for proof of concept experiments. The same is generally true of low-speed compressors and turbines, although the General Electric Aircraft Engines large (5-ft-diam) low-speed facilities have been an integral part of their design process (for which they have extensive experience in adapting results to high-speed regimes).

Low-speed experiments are comparatively inexpensive and have helped, and still help, greatly in exploration of underlying effects and processes (for example, progression of wakes and clearance flows through succeeding blade rows, details of boundary-layer transition processes, different routes to rotating stall, and mechanisms of rotor casing treatment). One of the things that makes their use so fruitful is the ease of applicability of hot-wire and optical instrumentation.

Effects that depend on compressible flow phenomena (for example, shock location and strength and shock-induced transition) require high-speed facilities. To run such a compressor or turbine facility in steady-state operation requires large amounts of power and normally a team of people. Considerable ingenuity has therefore been used to develop short-duration facilities for study of aerodynamics and heat transfer. For compressors, the main benefit is reduced power. For turbines, there is an additional benefit because heat transfer is much easier to measure as a transient, rather than in steady, conditions. When local surface temperature is monitored during a transient, it is possible to find the heat transfer rate and the heat transfer coefficient.

In all of these unsteady systems there is scope in which to have the right temperature ratios between metal and gas, or mainstream gas and cooling air, to simulate the real turbine. Normally all of the data must be taken in tenths (or less) of a second and a necessary advance has been the development of appropriate instrumentation and data acquisition procedures.

VIII. Some Real and Imaginary Problems with Turbomachinery

In conclusion, it seems appropriate to reflect on some of the issues that appeared at the time to be major hurdles. A concern with the methodology has formed the core of the early part of the paper, but what about the overall idea of a turbomachine? Under what conditions will the aerodynamics "work" and how well?

A. Turbines

For well over 100 years it has been known that axial turbines would work, in the sense that the pressure falls by a roughly predictable amount and net power is produced. Even 100 years ago steam turbines produced efficiencies of close to 70% with blades which, in hindsight, are seen to be far from aerodynamically good. (Parsons delivered a 75-kW steam turbine to the Cambridge power station in 1895. It was donated to the Cambridge University Engineering Department in 1912, where it was used by students until 1947. The measured efficiency was 67.5%.) Presently, a shortfall relative to expectation of more than a percentage point in efficiency would be considered disappointing.

By the time the gas turbine was being considered for the jet engine there had been many years of experience of turbines with steam. Whittle, who was cautious enough to avoid using an axial compressor, was ready to use an axial turbine. There was no doubt whether the turbine would work, but there were doubts about its level of efficiency and its mechanical integrity.

The question of integrity has two parts, both of which have aerodynamic aspects, namely, withstanding both the high temperatures and high cycle fatigue. The initial response to high temperature was to find better materials, though in Germany cooled turbine blades were used in service during World War II. Nickel-based alloy blades were used in place of steel and forgings for rotor blades gave way to castings, then directionally solidified casting and finally single-crystal castings. Current blades also use thermal barrier coatings, that is layers of nonmetallic material deposited on the varies, blades, hub and shroud, which have low thermal conductivity compared to metal and can withstand higher temperature before being damaged.

Though the improvements in materials for turbines are impressive, the major increase in allowable turbine entry temperature is due to cooling.[4,8] Turbine cooling, which can be regarded as another aspect of turbomachinery aerodynamics, allows operation at rotor inlet temperatures more than 300°C above the incipient melting temperature of nickel-based superalloys.[8] There has been extensive work to improve the effectiveness of the cooling while retaining high efficiency of the turbine stage, and it may be remarked that once cooling is used it is no longer entirely straightforward to define the efficiency of the turbine, much less to measure it accurately and unambiguously. The difficulty of cooling the nozzle has been increasing not only because of the push for higher temperature, but

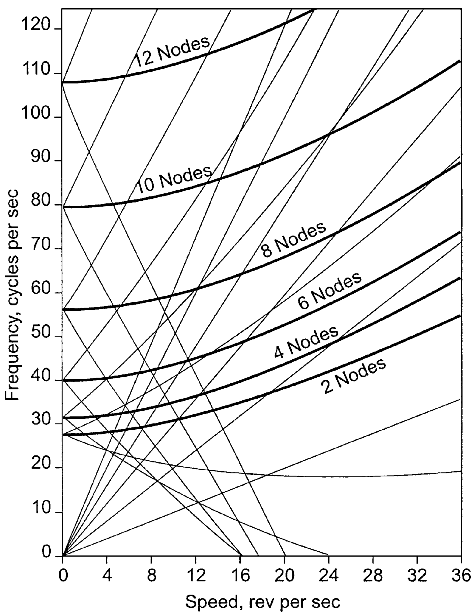

Fig. 7 Campbell's diagram[55] of 1924 showing the intersection of forcing frequencies and resonant frequencies of vibration modes in turbine wheels.

also because of the requirement to keep NOx levels down. To do this implies that more air is needed to cool the combustion products near the fuel injectors and less is available to cool the first-stage nozzle. Turbine heat transfer is a huge field, and it is inappropriate to enter into details here; the topic has been covered in detail in an IGTI Scholar lecture by Dunn.[53]

High cycle fatigue has been a major issue for turbomachinery since the days of the steam turbines. There are two different aerodynamic causes: forced vibration and flutter.[54] Forced vibration occurs because of the periodic effect of one blade row or other component (struts or inlet distortion, etc.) moving relative to another. (If a rotor runs downstream of a row of N inlet guide vanes, there will be an excitation at a frequency N times the angular velocity of the shaft.) In investigating this problem for steam turbines, Campbell[55] devised the eponymous Campbell diagram, which is still much in use as a way to portray potential situations of high-amplitude blade vibratory motion. Figure 7, which shows an original figure from the 1924 publication, illustrates the application. Resonances, and hence potential vibration problems, occur when a natural frequency of a blade (appearing in Fig. 7 as a line that is roughly horizontal) coincides with an excitation frequency, appearing on the Campbell diagram as a line through the origin and proportional to rotational speed. Low-order excitation tends to be more of a problem than high-order excitation and for turbines a common cause of aerodynamic forcing is discrete combustor cans or injectors.

To avoid frequency coincidences in the running range the natural frequency of the blade can be increased by an increase in stiffness of the blade or reduction of the mass of the blade near the tip. The stiffness can be raised by an increase in the camber (which may compromise aerodynamic performance) or by the use tip or part-span shrouds or lacing wires. (Shrouds are circumferential wing-like struts that are integral to each blade. The shrouds from any one blade make contact with the shrouds of adjacent blades, greatly increasing the overall stiffness of the assembly.) Blade mass is sometimes reduced by cutting back the trailing edge of the blade near the tip and many older compressor blades show this fix, evidence that a problem had been incurred late in the development of the machine.

Flutter is a self-excited vibration in which a negative aerodynamic damping response to small-amplitude oscillatory blade motions causes mechanical energy to be fed into the oscillations, thus increasing the amplitude. Flutter tends to occur in turbines or compressors when the blade span is large in relation to chord (high aspect ratio). It is therefore a potential problem for the low-pressure blades in steam turbines (where the fix is often lacing wires), for low-pressure turbines in gas turbines and for fans. Flutter shows up in a Campbell diagram as a region of high vibration that barely alters in frequency as the rotational speed increases.

Until recently, calculation of the amplitude of forced or self-excited vibration (or even the onset of the latter) was beyond the capability of unsteady methods. It has been emphasised that a complete description of the flow requires three spatial dimensions and variation with respect to time. Only in the last few years it has become possible to predict these with sufficient confidence that the occurrence can be foreseen and appropriate action taken to mitigate it. The calculations are still at the edge of what can be considered feasible for design, and some restriction in the range of cases examined is necessary. A recent paper by Simpson et al.,[56] which describes the forced vibration and flutter prediction for a multistage compressor, gives a view of the status of these unsteady aerodynamic issues.

B. Radial Compressors

It is not an exaggeration to state that the radial compressor always produces a pressure rise. If the compressor is sufficiently throttled, or if the mismatching between inlet and diffuser is sufficiently large, stall and surge can occur. However, if the design is reasonable, a single-stage radial compressor can be relied on to produce a pressure rise that is sufficient for useful operation of a gas turbine engine. It was for this reason that Whittle and von Ohain both adopted radial compressors in their pioneering engines. Although the radial compressor reliably produces a pressure rise, and the pressure ratio for a single stage can be around 10:1 in some gas turbines, the efficiency is generally lower than that for axial compressors. For very small compressors, however, the efficiency for a radial machine may be better than an axial machine. This is because axial blades can become too small to manufacture economically to the required precision and, in addition, the lower Reynolds number lowers the efficiency of axial blades.

The history and evolution of radial compressors, and the difficulties in their design, has recently been described by Krain.[9]

C. Axial Compressors

Axial compressors with many stages existed when Whittle and von Ohain were designing their engines, but there were not adequate design processes. There were also underlying physical reasons to expect that the performance of a multistage axial compressor would be poor. Only in the last few years has the understanding of compressor aerodynamics reached the point of being able to explain why such anxieties were misplaced or overemphasised.

There were three principal problems for the multistage axial compressor as viewed from the perspective of the early 1940s.

1) The first is matching of stages front to back at off-design conditions. The density rises along the length of an axial compressor as the pressure rises and to keep the axial velocity at an acceptable value, the flow area is reduced progressively from front to rear of the compressor. This, of course, is what gives rise to the shorter blades at the rear compared to the front. The ratio of outlet to inlet area is normally selected for the design conditions, which corresponds to high rotational speed, giving substantial pressure and density rise. At reduced-speed operation, when the pressure rise is smaller than at design, the flow area at the rear will be too small compared to the area at the front. (This is particularly serious during starting when the pressure ratio is not much greater than unity.) As a result, the rear stages of the compressor will tend to choke and the front stages will tend to stall. The degree to which this was seen as a difficulty is described by Carter,[57] writing more than a decade later, who stated, "By far the most important problem facing the designer of an axial compressor nowadays is the matching of the individual stages and the effect of matching on the surge behavior." The onset of instabilities (rotating stall and surge) in compressors and compression systems can still not be predicted in an *a priori* manner. As a gauge of the state of the art, treatment of these in the design process basically relies on empirical correlations, many of which date back several decades. (Discussion of these important instabilities is outside of the scope of the paper, but a recent review of the unsteady phenomena of interest is given by Paduano et al.[58])

2) The second is separation of the flow about the blades because of the adverse pressure gradient. The adverse pressure gradient increases as the incidence is increased and for a two-dimensional cascade of blades only a few (for example, less than five) degrees separate minimum loss from blade stall. Near the endwalls, the incidence has to rise, as can be seen from velocity triangles, and the flow, therefore, tends to stall at the endwalls.

3) Third, even at conditions for which the blades do not stall at the endwalls, the losses there will be higher than in the freestream. Consequently, there will be a buildup in entropy from one stage to another in the endwall region with high temperature (because of low local efficiency for the same pressure rise) or reverse in direction (going from back to front, with negative axial velocity because of flow separation).

All of these were real concerns, though evidence was emerging that the actual situation might be less dire than imagined. The Brown Boveri Company was selling multistage axial compressors as part of a land-based gas turbine; the British Royal Aircraft Establishment bought one in 1939 to experiment with matching and, as mentioned, by the end of World War II both the Germans and the British had engines with axial compressors flying. Nevertheless, the state of the art was such that shortly after the war, Rolls-Royce had enough trouble in developing the compressor for a new jet engine (the Avon) that they were ordered by their customer to use a compressor successfully designed by another British company, Armstrong–Siddley. Even in the 21st century there have been problems with axial compressors.

The principal method for avoidance of the problems associated with stage matching is to use variable stators. The management of problems 2 and 3 might, in contrast, be viewed as being left to "nature" and designed around. It is only in the last few years that the way "nature" works has been unravelled.

1. *Variable Geometry and Stage Matching*

Variable stators, which allow the stagger of the stators to be increased as the speed of the compressor falls, are virtually universal on axial compressors with overall pressure ratio in excess of about 6:1 on one spool. The flow area at the rear stage is normally sized relative to the inlet area, based on the density rise at design conditions (this being the condition at which efficiency is normally most important). When the density rise is less than this, the rear stages will tend to choke, because the flow area is too small relative to inlet flow area. The consequent restriction on mass flow means that the front stage or stages will tend to stall. When the stator vanes are moved so that the flow into the rotors has a tangential velocity component in the direction of rotation, the incidence onto the rotor is decreased, the stall can be removed, and the efficiency can be raised. When the efficiency is raised, the rear stages are able to pass more flow, and the underlying problem is, therefore, relieved too. Bleed is also sometimes arranged near the middle stages of an axial compressor; at low speeds the bleed valves can be opened, increasing the flow through the front stages. Bleed has two disadvantages: There is work done in compressing the air that is dumped, leading to a drop in compressor overall efficiency, and the bleed flow can be a source of noise. With several rows of variable stators, overall design pressure ratios on a single spool in excess of 20:1 have been achieved.

2. *Flow Separation Near Endwalls*

The three-dimensional flow behavior of blades appears to have been first addressed by Wadia and Beacher,[59] who noted that the pressure distribution about compressor blades near the endwalls was less severe than would be expected from the local incidence. They explained this as an effect of radial relief, envisaging radial, that is, spanwise, velocities producing an alteration of the local pressure distribution that was set up by the local incidence. The real explanation is not quite this and is, in a way, simpler. The high local incidence near the endwalls does not produce large pressure gradients in the way that it would if the high incidence were over the whole span. Instead, the pressure distribution around the blades is more of an average of the flow over the whole span, and it is dominated by the majority of the blade that is in the freestream, well away from the endwall. [The discussion by Cumpsty that accompanies the paper by Wadia and Beacher attempts to explain this.] In other words, the pressure distribution about the profile near the endwall needs to be determined by a three-dimensional method and not by two-dimensional methods. Two-dimensional reasoning, based on local incidence, leads to an unduly pessimistic conclusion.

3. *Losses Near the Endwalls*

Although the pressure distribution around the profiles is less severe than was imagined when people started to think about multistage axial compressors, the losses are higher near the endwalls. Smith[37] had shown that in a so-called repeating stage (a stage in which the axial velocity profile out of the stage was the same as that entering the stage) the pressure rise along the span must be uniform along the span. If so, and if the losses rise toward the endwall, the temperature must also rise toward the walls. This would be a serious problem toward the rear of a multistage compressor, for which the mean exit temperature at exit can now be in excess of 900 K. Measurements do not show that the temperature is much higher near the endwalls, but they do confirm that the pressure rise is virtually uniform along the span. Over a span of about a decade, experiments and computations showed that it was a combination of secondary flow in the blade passages (with consequent secondary flow and radial transport) and turbulent mixing that was responsible for the aforementioned temperature distribution.[34,60,61] The effect of such spanwise adjustment is now routinely included in axisymmetric throughflow calculations and is also captured by three-dimensional computations that calculate the secondary flows and model an appropriate level of turbulence.

In summary, the multistage compressor can be said to work for several reasons that could not have been imagined in the early days of its use. One is the invention of variable stators to cope with off-design conditions. More surprising is that, because the flow is three dimensional, the pressure gradients around the blade profiles near the endwalls are less severe than would be expected from two-dimensional reasoning; this means that strong two-dimensional separation, with large loss and increased deviation, does not occur. Even with this, multistage compressors would not be practical if there were not spanwise mixing processes to even out the temperatures attributable to the greater losses near the endwalls.

IX. Summary and Conclusions

The ambitions and demands of the gas turbine, and the steam turbine before it, have outstripped the scientific underpinning of the aerodynamics used to design them. The methods adopted to calculate the flow have, at least until recently, been gross approximations of the true behavior; in this sense, the drive to develop better descriptions of compressors and turbines has meant that "machines produce ideas just as surely as idea produce machines."[29] The most striking assumption, accepted so completely it is rarely questioned, is that of steady flow in the blade passages. Until about 1940, the radial variation in the flow was largely ignored; very often the treatment of the blade-to-blade flow was only one-dimensional. By the end of the World War II, the blade-to-blade flow was well understood, but the treatment was mainly based on correlations until the 1970s, when two-dimensional calculation methods became widespread.

The introduction of three-dimensional methods, based on numerical solution of the RANS equations, has had a large impact on all aspects of turbomachinery. The effects of three-dimensional methods are still being accommodated and understood. This is especially true for turbomachinery aerodynamic design in a true three-dimensional fashion, for example, as described by Denton and Xu.[62] The continuing increase in computer speed, which makes possible calculations that were prohibitively costly several years before, has been accompanied by a decline in the amount of testing carried out. The techniques for measuring flow in multistage machines have changed in concept much less than the power to compute them. Unsteady testing methods have been developed to enable low-cost experiments at actual Mach numbers and to facilitate heat transfer measurements. There is undoubted scope to integrate the results of these tests and the output from computational fluid dynamics (CFD) more closely, as well as for novel ways for in-depth interrogation of the flow features revealed by the latter.

The simplified methods used in the past to design and analyze turbomachinery required an appreciation of what was important and, within the approximations adopted, some level of understanding. Now some engineers are expected to use validated computer codes that they do not fully understand; more important, they may not appreciate the assumptions, approximations, or simplifications embedded in the methods. Many people are concerned at the aerodynamic performance risks that this poses and anxious that the education of engineers should equip them properly for the use of computer-based methods in their careers.

The availability of three-dimensional CFD has altered not only the design and analysis process, but the way that engineers approach the flow and try to understand it. This raises an interesting philosophical issue concerning the meaning and scope of understanding, namely, what does it mean to "understand" a flow in four dimensions? The flows that we can usefully describe as understood, in the sense of matching cause and effect, are normally in no more than two dimensions; for example, one-dimensional unsteady, two-dimensional steady and axisymmetric steady, with special examples in three dimensions. Is it plausible that true understanding of the more complicated flows may not be possible and that we must be content with the ability to predict the behavior?

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