Vortex Shedding in High-Speed Compressor Blade Wakes

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The wakes of highly loaded compressor blades are generally considered to be turbulent flows. Recent work has suggested that the blade wakes are dominated by a vortex streetlike structure. The experimental evidence supporting the wake vortex structure is reviewed. This structure is shown to redistribute thermal energy within the flowfield. The effect of the wake structure on conventional aerodynamic measurements of compressor performance is noted. A two-dimensional, time-accurate, viscous numerical simulation of the flow exhibits both vortex shedding in the wake and a lower-frequency flow instability that modulates the shedding. The numerical results are shown to agree quite well with the measurements from transonic compressor rotors.

Introduction

ALTHOUGH bluff bodies and low-speed airfoils have long been known to shed vortices into their wakes, the wakes of high-speed, high-Reynolds-number turbomachinery blading have generally been considered to be turbulent and unstructured. Usually, the wake is described only in terms of a time-averaged velocity profile. This is consistent with the view that, assuming uniform inflow and expecting turbulence, the flow in the frame moving with the compressor rotor is uniform and that variations observed in the stationary frame are primarily due to blade-to-blade geometric differences. This view, however, is erroneous in detail with practical import for compressor design.

There is a considerable body of experimental data that shows that blunt trailing-edge airfoils typical to turbines shed vortex streets.[1-3] Recent work has shown that a similar phenomenon may occur in sharp trailing-edge transonic compressors as well.[4,5]

If we assume for the moment that high-speed compressor blade wakes can consist of a vortex streetlike structure, then a number of questions immediately become relevant. These include: Do all blade wakes contain vortices? Why are the vortices rarely observed experimentally? How are the vortex streets formed and what is their structure? What is the practical importance of the wake structure to the compressor designer? We will address the last question first so as to provide a groundwork for the discussions that follow.

In the context of high-efficiency, high-performance transonic compressors and fans, the influence of the wake temporal and spatial structure for a given time-averaged velocity defect can be surprisingly large. The wake structure can influence the compressor aerodynamics, noise, and structural integrity. Vortex shedding decreases the blade row efficiency (higher base drag and shock loss, increased unsteadiness) and induces artifacts in standard measurement techniques. An increased level of unsteadiness into following blade rows can increase the loss there and alter the mass flow characteristics, independently of whether that blade row is itself shedding vortices. The interac-

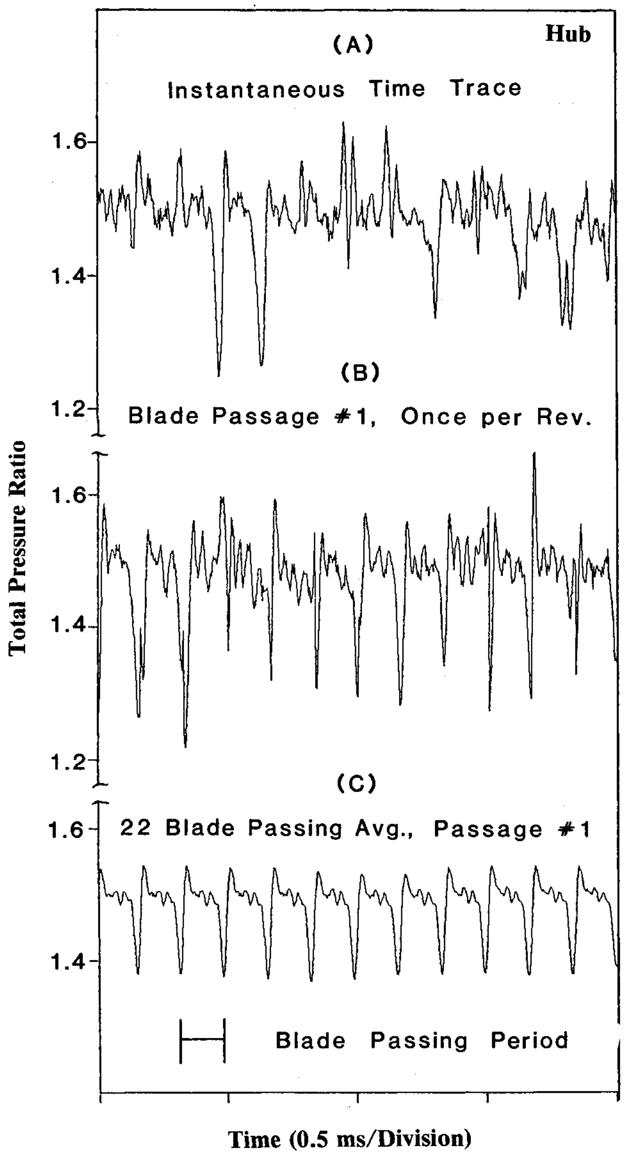

Fig. 1 Transonic compressor rotor exit total pressure near hub: a) instantaneous measurement, b) an individual blade passage as seen once per revolution, and c) an ensemble average of that individual passage.[6]

Received Feb. 13, 1987; revision received Aug. 20, 1987.

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tion of the shed vortex street with subsequent blade rows is also a source of noise. Obviously, the unsteady loading due to the vortex streets and related phenomena can have deleterious effects on the structural integrity of a compressor when the aerodynamic excitation frequencies coincide with those that are structurally important. The blade wake structure also influences compressor design and analysis in that it represents a physical phenomenon not usually modeled (either because the analysis is inviscid or steady state). Thus, the design intent may not represent a physically realizable system.

This paper is an extension of previous work in which the presence of vortex streets was inferred from experimental measurements and their effect on the blade shock system was discussed.[4,6] Here, we review that work, discuss the effects of vortex streets on the temperature distribution within the flow, note the influence of the wake structure on conventional aerodynamic performance measurements, present the results of an ab initio numerical simulation showing similar behavior, and suggest areas for further investigation.

Review of Experimental Observations

Flow-visualization studies have demonstrated the existence of vortex streets in the wakes of transonic flat-plate and turbine airfoil cascades[1–3] but, excepting the special case of acoustic resonance enhanced shedding,[7,8] shedding in high-speed compressor airfoils had not been conclusively reported. Hot-wire measurements in the wakes of controlled diffusion airfoil cascades yielded ambiguous results,[9] with one configuration showing evidence of periodicity in the wake while another did not. In the context of hydrofoils, Blake[10] suggested that all bodies shed but may do so in a discontinuous fashion, i.e., shed in bursts. Lack of phase coherence between the bursts would then tend to obscure the shedding when examined with spectral analysis techniques. This is consistent with the observation that the least ambiguous information has been instantaneous flow visualizations.

Time-resolved measurements in the outflow of a transonic compressor rotor operating near the compressor's peak efficiency points revealed high-frequency (three to four times blade passing) total pressure and temperature fluctuations of substantial amplitude in the core flow between the wakes, as well as large fluctuations in the wake strength of any particular blade[6] (Fig. 1). Similar observations were made in three different single-stage transonic compressors in three different test facilities using a variety of high-frequency response probe types.[12] The machines differed in design intent (both commercial fan and high-speed military designs were included) but all operated at high rotor efficiency (above 90%). The outflow of all three rotors demonstrated similar types and degrees of rotor relative unsteadiness, the unsteadiness being maximum near the design or maximum efficiency operating point. This indicates that rotor relative unsteadiness is a common phenomenon in transonic compressors.

A laser anemometer physically measures one or more components of the instantaneous velocity at a point in space. Because of the relatively high level of turbulence common to turbomachinery flowfields, the instantaneous velocity measurements are usually averaged and the result presented as the average velocity at that point (Fig. 2a). The velocity statistics at the point can also be examined however (although this is not commonly done) and can be presented in terms of a probability density distribution (pdd), a histogram showing the number of

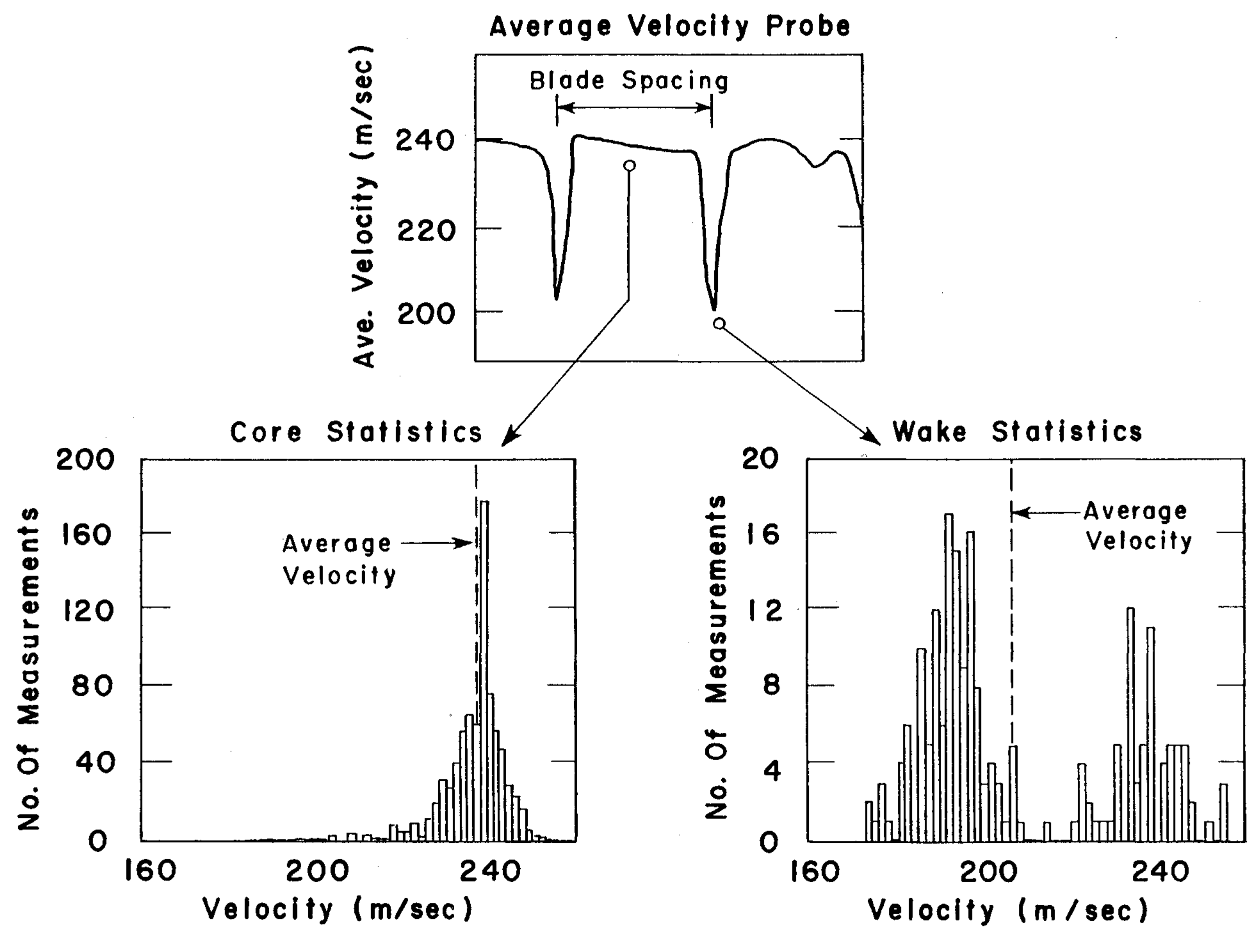

Fig. 2 Laser anemometer measurement of the outflow from a transonic compressor rotor: a) average velocity profile, b) histogram of velocity statistics in core flow showing Gaussian profile characteristic of turbulence, and c) velocity statistics in wake showing bimodal distribution attributed to vortex street.[4]

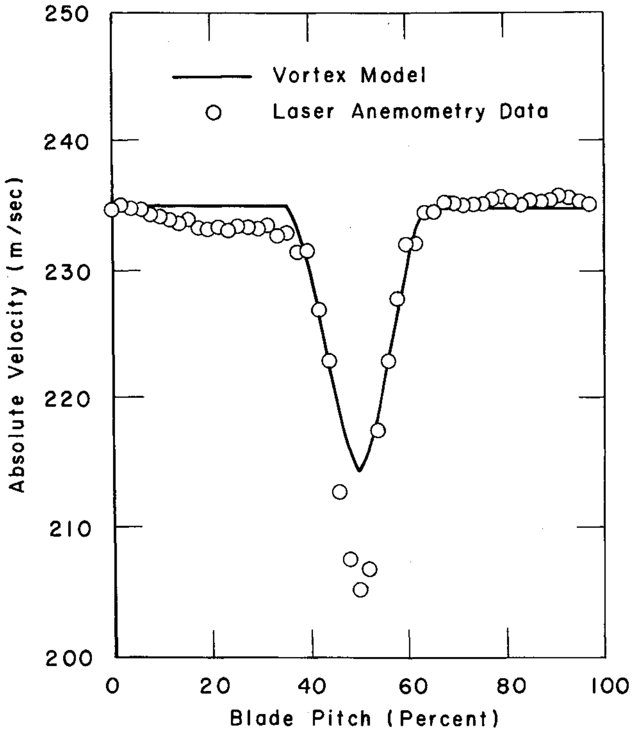

Fig. 3 Schematic representation of the vortex street geometry used in the model. The box delineates the region illustrated in Figs. 6–8.

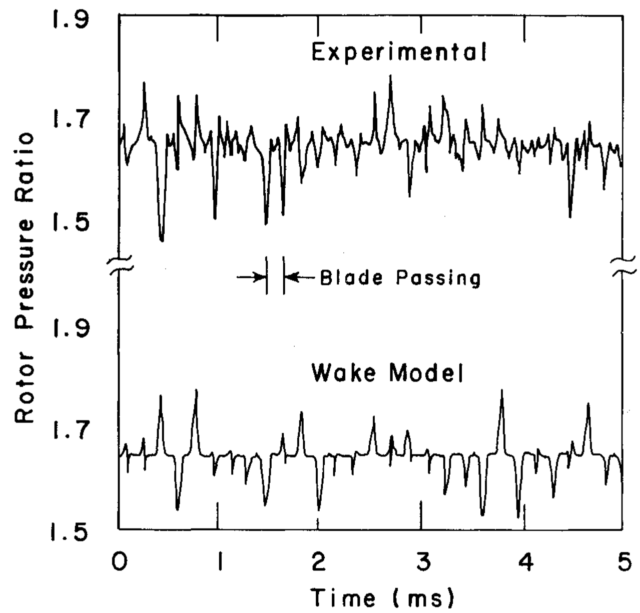

Fig. 4 Comparison of mean absolute velocity measured by a laser anemometer in a rotor blade passage with that from the average of the vortex street model.[4]

observations made at each value of velocity. For a turbulent flow, the shape of this histogram should be Gaussian. Laser anemometer measurements in the vicinity of the passage shock have shown a bimodal velocity distribution indicative of a shock moving to either side of the measurement point.[11] These observations were explained in terms of small (0.5% of axial chord) motion of the passage shock about its mean position. Vortex shedding in the blade wake was advanced as a possible driver of this motion.[6]

Subsequent laser anemometer measurements in the rotor wakes revealed that the velocity distributions were bimodal there as well, i.e., two velocities were just as likely with few measurements observed at the "average" velocity as they would be normally derived from such data[4] (see Fig. 2). Simultaneous time-resolved temperature and pressure probe measurements in the rotor outflow showed unsteadiness relative to the rotor at two time scales. One was the high-frequency disturbances mentioned earlier. The second appeared primarily as a modulation of the wake flow with frequency components on the order of one-half to three times the shaft rotational speed. This modulation could not be explained in terms of blade-to-blade geometric differences since the fluctuations were primarily aperiodic with rotor rotation (and, thus, not locked to the geometry) and were several times the magnitude of the periodic disturbances. A striking feature of the disturbances themselves was the large total temperature fluctuations in the wakes – ±3-5% of the mean total temperature (i.e., 10-20°C) – indicating local regions of intense cooling and heating. The instantaneous adiabatic efficiency calculated from these measurements showed a concomitant variation with some local regions appearing as over 100%. These temporal fluctuations and laser anemometer measurements were explained as resulting from a vortex street structure in the blade wakes. The over 100% efficiency observations have yet to be explained.

Descriptions of the Blade Wake and Vortex Street

In the previous work, the wake was modeled as two staggered rectilinear rows of Rankine vortices of opposite sign in a uniform freestream.[12] The vortices consist of an inner region with a forced-vortex core and an outer region following the irrotational flowfield of a classic von Kármán vortex street. This model was fit to the laser anemometer data so that the time-averaged velocity profile and the statistical distribution of velocities (the pdd) in the wake would match. The vortex size and strength were adjusted to fit the velocity profile while the ratio of the streamwise vortex spacing (A) to the distance between vortex rows (H) determined the velocity statistics (see Fig. 3).

As can be seen in Fig. 4, the model fits the data relatively well, which is a consistency check. The model prediction of a shedding frequency of 16 ± 2 kHz is quite close to the 14–15 kHz inferred from the core flow pressure fluctations. Furthermore, the model readily explains the high level of fluctuations observed in the wake with the high-frequency response probes. This is an artifact of sampling caused by the random position of the vortices in the wakes as the compressor revolves past the probe location. Figure 5 compares the absolute frame rotor outflow total pressure fluctuations predicted by the model with measurements.

While the vortex street model in Ref. 12 does a good job of explaining many of the experimental observations, it is a static model, describing the wake state only at a particular axial station. Thus, it contains no information on the vortex formation process, its evolution, or decay. At this time, we know of no published analytical model describing this process in detail.

Dynamic Energy Redistribution

One of the more interesting implications of identifying the structure of the wake of a transonic compressor blade as containing a vortex street is that the unsteady pressure field in the blade relative frame can redistribute thermal energy. The vortex street propagates at a velocity different from that of the mean flow and entrains fluid into and through the wake. As the entrained fluid passes through the pressure field of the vortex, its total enthalpy can be changed. In essence, the vortex street can be thought of as an array of tiny turbomachines.

The dynamics of this problem have been addressed by McCune[13] with a straighforward unsteady energy equation analysis. He shows that the pressure field fluctuations from the vortices change the total temperature distribution in the fluid and that the amplitude of this effect scales with the square of the freestream Mach number. At Mach 1, the vortices are predicted to produce a temperature fluctuation of 3–5% of the freestream total temperature, quite close to the observed value. This scaling with Mach number explains why the phenomenon was first noticed in a high-speed machine (tip relative Mach number of 2.2).

Kurosaka et al.[15] treat the problem in terms of the general Eckert-Weise effect,[14] in which the recovery factor at the rear of a shedding right circular cylinder can be negative and the wake centerline cooled to below the freestream inflow value, showing that this is the result of vortex street behavior. A key point is that this cooling by the vortices is observed with steady-state instrumentation, i.e., the average centerline temperature is depressed. There must, of course, be a concomitant increase in total temperature in the surrounding flow outside the wake. Kurosaka et al. also demonstrated that the vortex shedding and temperature separation can be significantly enhanced with acoustic feedback, essentially extending acoustic resonance work to include energy separation.[7,8]

The intense, localized hot and cold spots generated by the vortex street explain the large, abrupt temperature fluctuations observed in the time-resolved total temperature measurements of the compressor wakes.

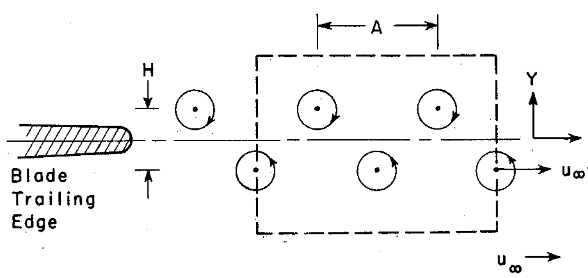

Fig. 5 Comparison of the time-resolved rotor exit absolute total pressure with that predicted by the vortex street model.[12]

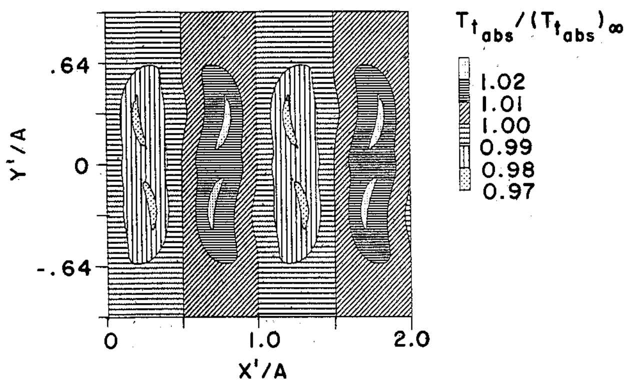

Fig. 6a Contour plot of absolute total temperature contours predicted by the wake model.

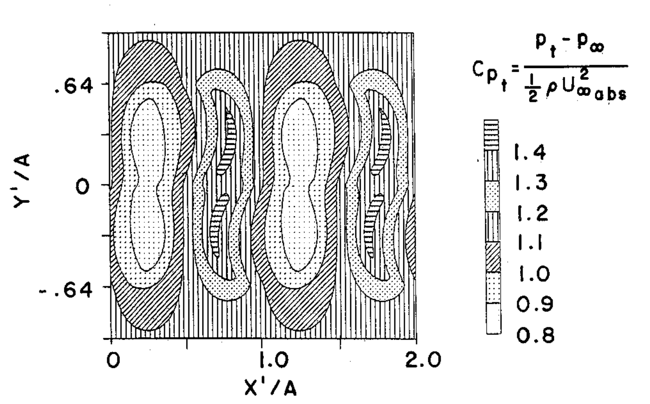

Fig. 6b Contour plot of absolute total pressure contours predicted by the wake model.

Wake Structure Effects on Apparent Compressor Efficiency

An earlier work on high-frequency passage shock motion driven by vortex shedding identified three mechanisms by which the loss in the compressor could increase.[6] The first is the increase in entropy rise across a shock wave undergoing small periodic axial motion compared to that for a steady shock at the same average approach Mach number. This change was quite small, amounting to only a 0.1–0.2% decrease of adiabatic efficiency in the rotor studied. The second loss source was the small-scale nonuniformities in total pressure generated by the oscillating shock wave. In this case, it was assumed that, due to the small spatial extent of the perturbation (one-eighth ~ one-quarter chord), the enthalpy would not be recovered as pressure rise in the diffusion process in the stator but rather appear as a mixing loss downstream. This loss was calculated to be of the same order as the wake mixing loss, about 1% (10% of the total measured stage loss). The third loss mechanism is the amplification of the spatial nonuniformities in the stage outflow, discussed earlier, by the shock system in a following transonic stage. This loss was estimated to be two to three times as great as that in the first stage, i.e., 2–3% of stage adiabatic efficiency.

In this section, we wish to discuss not additional thermodynamic loss mechanisms (i.e., production of entropy) but rather apparent losses, artifacts induced in the measurement processes and their interpretation by the periodic nature of the wake structure. If not properly accounted for, these artifacts can result in an erroneous estimate of compressor performance.

Before doing so, however, we wish to point out a simple curiosity. Adiabatic efficiency is often used to express turbomachine loss (entropy production). In an unsteady, transonic flow, however, the change in adiabatic efficiency need not be congruent with the entropy production, as it must in a steady flow.

To illustrate this, we will define the local adiabatic efficiency, η, in the usual manner relating the total pressure, P_t, and total temperature, T_t, ratios in the absolute (laboratory) frame as follows:

$$\eta = \frac{(P_{t2}/P_{t1})^{\gamma - 1/\gamma}_{\text{abs}} - 1}{(T_{t2}/T_{t1})_{\text{abs}} - 1} \tag{1}$$

where the subscripts 1 and 2 denote stations upstream and downstream of the rotor blade. Station 1 is assumed to have uniform conditions of total temperature and pressure, as would be the case with the first stage of a compressor. Station 2

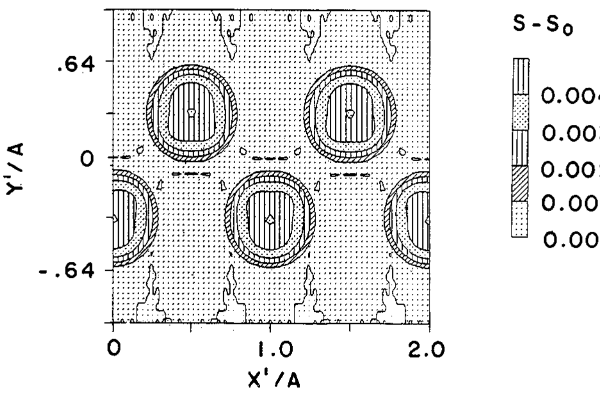

Fig. 7 Contours of entropy rise from wake model. Note that the entropy is essentially constant outside the vortex cores.

consists of a freestream region and a wake region, with the blade wakes represented by vortex streets as modeled in Ref 12. The modeled wake total pressure and temperature contours are shown in Fig. 6. (The elongated shape of the contours reflects the work done on the fluid at the vortex periphery.) The change in entropy S from station 1 to station 2 may be expressed as

$$\exp[-(S_2 - S_1)/c_p] = \frac{(P_{t2}/P_{t1})_{\text{abs}}^{\gamma-1/\gamma}}{(T_{t2}/T_{t1})_{\text{abs}}} \tag{2}$$

This can be seen in Fig. 7 for a flow with the compressor blade wake vortex model. The freestream stagnation temperature ratio is 1.175. As expected, the entropy variation is zero in the regions outside the vortex cores (since the model assumes that all the entropy change in the wakes is in the vortex cores themselves).

Equation (1) for the efficiency may now be rewritten by eliminating the stagnation pressure ratio using Eq. (2).

$$\eta = \frac{(T_{t2}/T_{t1})_{\text{abs}} \exp[-(S_2 - S_1)/c_p] - 1}{(T_{t2}/T_{t1})_{\text{abs}} - 1} \tag{3}$$

Equation (3) shows that, in a flow of uniform total temperature, the adiabatic efficiency is a unique function of the entropy change. Conversely, if there is no change in entropy from station 1 to station 2, then the efficiency is exactly 1.0, regardless of the variations in total temperature. However, if between stations 1 and 2 there is some loss mechanism (such as a normal shock wave) so that $S_2 > S_1$ in the freestream region, then the efficiency will be less than 1.0 and, in fact, will vary through the flow as the total temperature varies. In the case of a transonic compressor with a vortex street wake, the efficiency will vary even outside the vortex cores in the wake (which contain all of the entropy change) where the flow is otherwise modeled as irrotational and inviscid, i.e., the value of $(S_2 - S_1)$ is constant (as shown in Fig. 7). Since the flow in the absolute frame is unsteady (with or without vortex shedding) and the gradient of entropy is zero outside the cores, the total temperature must vary according to Crocco's theorem, which may be written as follows:

$$T\nabla S = \nabla(c_p T_t) - \boldsymbol{q}(\nabla \times \boldsymbol{q}) + \frac{\partial \boldsymbol{q}}{\partial t} \tag{4}$$

In the regions outside the vortex cores this reduces to

$$c_p \nabla T_t = \frac{-\partial \boldsymbol{q}}{\partial t} \tag{5}$$

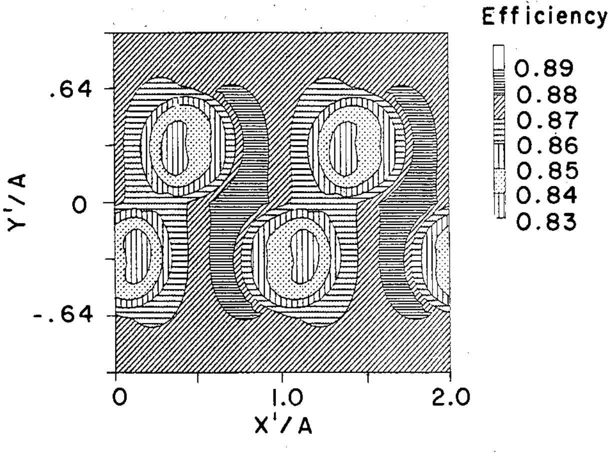

Fig. 8 Contours of adiabatic efficiency from the wake model. Note that these regions are not congruent with the entropy change in Fig. 7 and that there are regions of adiabatic efficiency indicated as above that of the freestream.

Equation (5) implies that the total pressure must vary as the total temperature changes so as to keep the entropy $(S_2 - S_1)$ constant. In any case, since $(S_2 - S_1)$ is constant and T_{t2}/T_{t1} varies throughout the vortex street, the efficiency will vary as well, even though the entropy is constant! This can be seen by comparing Figs. 7 and 8. The efficiency is not congruent with the entropy change. Because of the energy separation mechanisms discussed in the previous section, the total temperature in local regions can be greater or less than that of the freestream. Thus, there are regions in which the efficiency is greater than the freestream value. It should be pointed out with reference to Eq. (3) that, as long as the entropy increases, a variation in total temperature cannot result in an adiabatic efficiency above 1.

It is important to note that this difference between entropy rise and efficiency requires only that the flow have a change in entropy between the measurement stations (e.g., a shock wave) and a spatial variation in total temperature. Unsteadiness plays a part only in that, in the absence of heat conduction and viscous stresses,[14] a variation in total temperature can be produced only by unsteady pressure forces. The unsteadiness is required in the absolute frame, not the blade relative frame, and, thus, a blade relative unsteady wake is not a prerequisite. Rather, this effect is common to all transonic and supersonic turbomachines. Since the standard derivation of adiabatic efficiency, which is presented in Eqs. (1–3), is based on steady flow concepts, this discrepancy should not be surprising. Other than suggesting that adiabatic efficiency can be a misleading measure of turbomachine performance, the practical significance of this observation has yet to be enunciated, but the authors find it interesting.

Wake Structure Effects on Measurements

The structure of the compressor blade wakes can make a difference in the aerodynamic efficiency as commonly measured with aerodynamic probes in the rotor outflow. To illustrate this, we will consider two transonic compressor rotors with the same geometry, true mass-averaged total pressure and temperature rise, and, therefore, the same mixed-out adiabatic efficiency. One blade will be assumed to have a classic turbulent wake while the other's wake will consist of a vortex street with the same average velocity defect. (Here we use the term turbulent rather loosely, meaning random, thus ignoring any turbulent structure that may exist.) In other words, the wakes as measured by conventional laser anemometry techniques would appear identical. Conventional, low-frequency response probes are then used to accurately measure the true time average of the total temperature and pressure at the rotor outflow. (Since this is a two-dimensional analysis, the area-averaged quantities measured by probes are actually line averages.) The probes are assumed to be free from dynamic effects. We will take the average flow conditions at the rotor blade row exit plane to be those measured in a transonic rotor,[4] Table 1. Note that the total temperature is higher in the flow outside the wakes in the vortex street case compared to that in the turbulent wake case. This is required since the vortex street depresses the wake centerline temperature, but the total energy flux out of the blade row must be the same in both cases.

Table 2 shows the resultant average flow condition calculated with the two wake states. Results are given both close to

Table 1 Rotor exit conditions assumed for mixing calculation

Freestream	Vortex wake	Turbulent wake
Total pressure ratio	1.646	1.646
Total temperature ratio	1.175	1.170
Absolute exit flow angle, deg	45	45
Wake width/passage width	0.260	0.260

Table 2 Rotor outflow average laboratory frame conditions

	Vortex wake	Turbulent wake
Near rotor (130% axial chord):		
Mass-averaged total pressure ratio	1.644	1.642
Mass-averaged total temperature ratio	1.175	1.175
Mass-averaged adiabatic efficiency	**0.873**	**0.871**
Probe-indicated total pressure ratio	1.644	1.641
Probe-indicated total temperature ratio	1.175	1.177
Probe-indicated adiabatic efficiency	**0.872**	**0.859**
Far downstream (mixed out):		
Total pressure ratio	1.648	1.645
Total temperature ratio	1.177	1.176
Adiabatic efficiency	**0.866**	**0.867**

the rotor (the measurements were made at 130% of the axial chord), where the wake structures are distinct, and far downstream, where the wakes have totally mixed out. The mass-average quantities agree at both stations of course. The time averages close to the rotor do not, however. The time average of the vortex street wake case is quite close to the mass average (probably by coincidence), but the turbulent wake case efficiency is measured at over 1% low. (The difference between time or area and mass averages is well known. The point is that the low-frequency response probes can measure the time average only.) Far downstream, the mass-averaged efficiency has decreased by 0.5% (which is why probe stations are normally placed as far downstream as possible).

If the wake structure were always the same, the difference in measured efficiencies due to the structure could be mixed in with empirical probe calibration factors and thus be calibrated out (in theory). If, however, a small change in blade design or turbomachine operating point were to alter the wake structure alone, there would be an erroneous change in measured aerodynamic efficiency, an error of over 1% for the stage studied here. In other words, a change in the wake structure can appear as a change in stage efficiency, even though the momentum and thermal energy flux through the machine (the pressure and temperature rise) have not changed. This error would disappear if the measurements were made with sufficient time resolution to resolve the wake structure.

Thus, wake structure can introduce artifacts in the measurement of compressor performance, introducing apparent changes not representative of the state of the fluid exiting the machine. Because the magnitude of the temperature separation in the vortex street scales with the square of the Mach number, this effect should be most important for transonic and supersonic turbomachines. The argument should hold true for turbine flows as well, but this has not been verified.

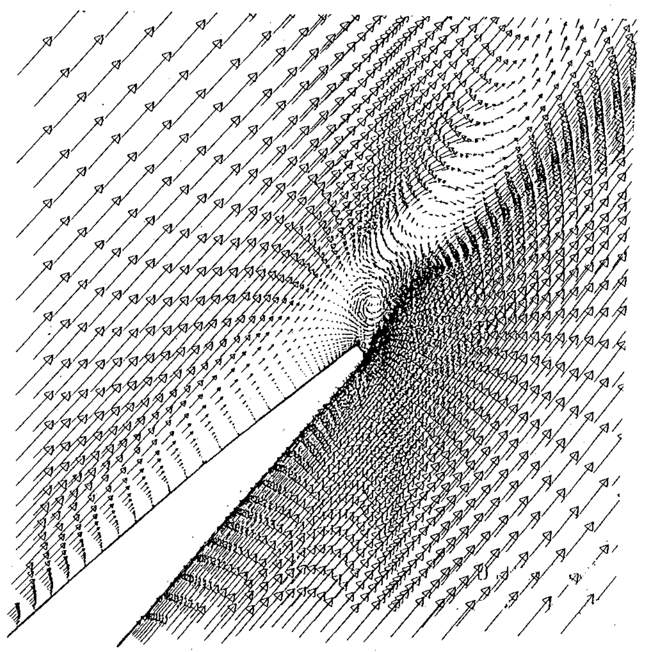

Fig. 9 Instantaneous vector plot (arrow orientation is velocity direction, length is velocity magnitude) of the trailing-edge region of a transonic airfoil calculated with a 2-D time-accurate Navier-Stokes code. Note the separated region on the suction surface and the vortices shed into the wake.

Numerical Simulation of Compressor Blade Wake Shedding

To further investigate the compressor wake structure, the flow through a single blade passage was simulated with a computational fluid dynamic (CFD) technique. A two-dimensional, time-accurate, Reynolds-averaged, explicit Navier-Stokes calculation was done with a relatively fine grid and a very small time step to ensure good spatial and temporal resolution. The midspan airfoil geometry of the transonic rotor measured in Ref. 4 was used. To accommodate supersonic inflow limitations in the code, the inlet relative Mach number was reduced from the 1.17 of the measurement condition to 0.92 for the calculation. Inlet total conditions were adjusted to keep the Reynolds number based on axial chord at 1.2×10^6. The inlet flow at the upstream boundary is specified as uniform. Thus, there is no external excitation or periodicity, only that generated by the flow itself. More details on the calculation can be found in Ref. 16.

The primary observation to be made from this ab initio calculation is that vortices are shed into the blade wake. Figure 9 is a plot of instantaneous velocity vectors in the blade relative frame near the trailing edge. It shows the vortex structure in the wake as well as a separated region on the suction surface. (The vector length denotes the magnitude of the local flow velocity, the orientation the flow direction.) Only two vortices are readily seen here due to the change in vortex core translational velocity as the vortices propagate downstream. This has the effect of blurring the flow structure in any frame other than that moving with the cores. A similar problem in visualizing the vortices was encountered in the modeling effort.[12] In both cases, the ready identification of the vortex structure is strongly dependent on the frame of reference chosen, making the vortex street fairly elusive and hard to see.

The shedding periodicity shows up quite clearly in the calculated trailing-edge static pressure, Fig. 10. Also of interest is the lower frequency present, which modulates both the amplitude and the frequency of the vortex shedding. This is quite similar to low-frequency modulation observed in the experimental measurements.[12] In the simulation, the lower frequency correlates with the motion of the separation point along the suction surface and with axial motion of the passage shock (the two are

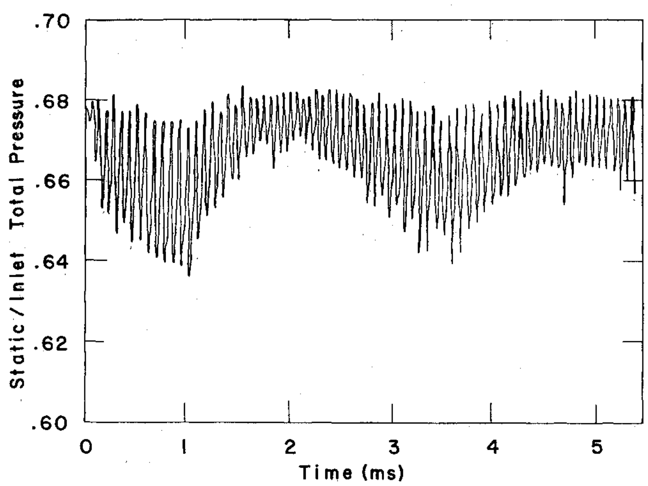

Fig. 10 Static pressure history at trailing edge of a transonic airfoil. Note the periodicity due to vortex shedding and the modulation of that shedding by a lower-frequency disturbance.

not in phase), Fig. 11. There is also a 30% fluctuation in blade moment at this frequency, which is low enough ($\sim$ 350 Hz) to be of concern to the structural designer. The exact cause of this low-frequency movement of the separation point is not yet clear but appears quite similar to instabilities observed in high-speed diffusers.

The modulation of the vortex shedding frequency by the low-frequency oscillation was considerable, a factor of 2. The strength of the vortices varied inversely with the shedding frequency. The shedding frequency range from the CFD calculation is compared with those inferred from the laser anemometer ppd's and from the core flow-shock motion fluctuations in Fig. 12. These three estimates are completely independent and show quite good agreement. The large fluctuations in frequency shown in the numerical simulation tend to explain the difficulty encountered in extracting a single, unambiguous frequency estimate from the experimental measurements. Fluctuations of this magnitude may also blur the bimodal anemometer histograms.

The blade relative total pressure in the numerical simulation, as would be measured with a fixed laboratory frame probe as the rotor passes, is compared with measurements[12] in Fig. 13. The qualitative agreement is excellent. The calculation clearly captures the wake modulation evident in the measurements.

The high-frequency jitter of the passage shock at the shedding frequency inferred in Ref. 6 is not observed in the calculation. However, since the predicted shock motion is no more than one grid cell size and the numerics spread the shock over five grid points, this is not surprising. A calculation with a much finer grid size would be required in order to address this problem properly.

Overall, the numerical, CFD simulation agrees extremely well with the experimental observations and the analytical model. The one area in which the numerical simulation does not add information is the decay of the wake structure as it is convected downstream since the numerical damping overwhelms most physical dissipation mechanisms.

Discussion and Conclusions

Considerable effort has traditionally been spent on establishing the proper parameter with which to correlate vortex shedding. A Strouhal number based on trailing-edge thickness is commonly used, especially for blunt trailing-edge bodies.[2,3,10] For compressor blades, the wake displacement thickness has been suggested as a more realistic correlation.[6] Since the wakes of compressor blades are relatively thick compared to the trailing edge, the difference in frequency between the correlations using the different parameters is considerable, a factor of 4. For

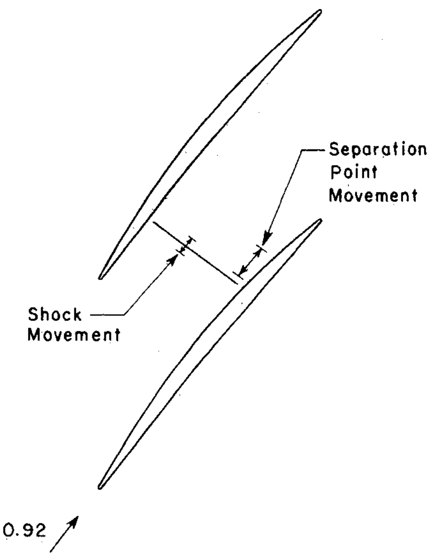

Fig. 11 Numerical simulation showing a correlation between the passage shock motion and the separation point movement.

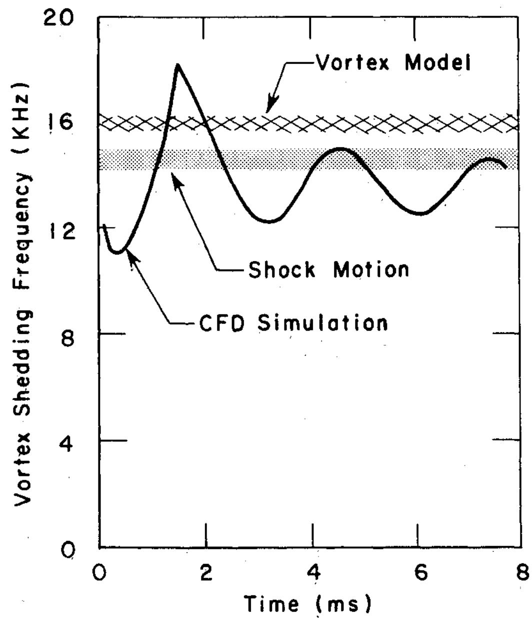

Fig. 12 Comparison of the shedding frequency predicted by the CFD simulation with those inferred from measurements.

the blade section studied here, a frequency of 15 kHz is predicted using the wake thickness, matching both the experimental and numerical results.

The modulation of the frequency and strength of the vortex shedding observed in the CFD calculation is extremely important for the interpretation of experimental measurements. The presence of the modulation considerably complicates the practical problem of vortex detection in rotating machinery. The impact on laser anemometry techniques and spectral analysis methods needs to be quantitatively assessed.

Examination of the details of the numerical simulation as well as classical vortex shedding analysis[17] suggests that the process may be considerably more complex than can be represented by a simple Strouhal number. Many shedding processes may compete. In particular, it is observed in the calculation that the vortex shedding strength varies directly with the length of the separation zone on the suction surface, thus suggesting that an unseparated airfoil may not exhibit this type of strong shedding behavior. A more complete discussion of vortex formation and evolution will be the subject of a later paper.

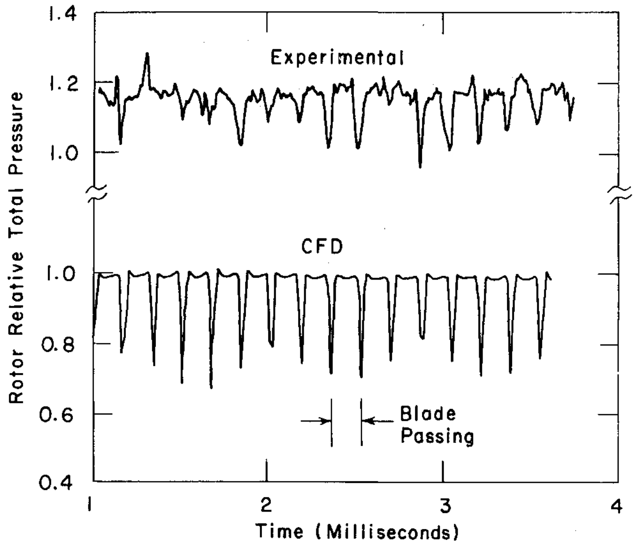

Fig. 13 Comparison of the rotor relative total pressure time history measured in a transonic compressor with that predicted by the CFD simulation.

This work has tended to treat the compressor blade and vortex street in isolation when, in reality, it is part of a very complex environment. Acoustic feedback has been shown to both enhance shedding and alter the frequency.[8,14,20] There are many mechanisms involving interblade row interactions for forcing the shedding as well. The possibility of phase locking between blades is also evident.[18] Clearly, much work can be done in this area.

Laser anemometer measurements of the turbulent kinetic energy within the stator passages of a transonic compressor stage have shown a structure consistent with a vortex street wake (i.e., periodic islands of high kinetic energy in the wake).[19] A laser anemometer measures the stationary frame ensemble-averaged flowfield. Thus, the vortices must be shed at a fixed phase relative to the stator in order to appear in the data. This implies that there is an additional mechanism by which the measurement of time-averaged performance using closely coupled instrumentation (in the stator leading edge, for example) may give misleading results. The assumption that the flowfield is well sampled would no longer be valid. The stage geometry is such that a stationary frame probe would sample along a vertical line in the flow maps given in Figs. 6–8. Phase locking implies that the patterns evident in the figures are stationary in the flowwise direction with respect to the measurement station. Thus, the probe will not be uniformly surveying the flowfield. The quantitative importance of this effect has yet to be determined.

Another area needing more investigation is the three-dimensional wake structure in real turbomachines. Although it is often dangerous to generalize from two to three dimensions, we will point out that, if the vortices have considerable spanwise extent, they can serve as an extremely powerful mechanism for the radial transport of fluid—complicating the evaluation of the radial work and efficiency distributions. Work with cylinder shedding has shown that external forcing (acoustic in this case) can enhance the spanwise coherence of the shed vortex street.[20]

A change in measured performance due to the wake structure is a result of the difference between common probe (generally time) averages and true mass averages. Thus, in addition to the vortex wake structure discussed in this paper, other processes that redistribute thermal energy in the flow (and, thus, change the difference between the averages) can have similar effects. Local heat transfer, either within the flow or to the blades, may be important in this regard. Furthermore, the dynamic behavior of aerodynamic probes can introduce artifacts into the performance measurements that could either amplify or reduce the effects noted earlier, depending upon the exact structure of the flow.

In this paper, we have made the following observations concerning the structure of compressor blade flow and wakes:

1) The wakes of high-Reynolds-number, transonic compressor blades can consist of shed vortex streets. This is confirmed by measurement, modeling, and numerical simulation.

2) The vortex shedding scales with wake thickness.

3) The shedding frequency and strength are sensitive to the environment.

4) The vortex street can depress the average wake temperature.

5) The importance of the wake structure increases with the square of the freestream Mach number.

6) Wake structure can influence the measurement of compressor efficiency.

7) Changes in wake structure can be mistaken for changes in compressor performance.

8) The wake structure can drive the blade shock system, inducing loss.

9) The wake structure can be influenced by other fluid dynamic instabilities present in a blade passage, which themselves may be important for performance and structural dynamic reasons.

10) Caution is required when interpreting steady-state solutions of inherently unsteady flowfields.

Overall, vortex shedding in transonic compressor blade wakes can have significant influence on compressor behavior.

Acknowledgements

This work was supported in part by the U.S. Air Force Office of Scientific Research and the NASA Lewis Research Center. The authors would like to thank Mr. R. Haimes for assistance with the numerical simulation (he wrote the code), and Professor E. M. Greitzer for many useful discussions.

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