PRECEDING PAGE BLANK NOT FILMED

Boundary Layers in Centrifugal Compressors

ROBERT C. DEAN, JR.

Creare Incorporated
Hanover, New Hampshire

The objective of this paper is to demonstrate the utility of boundary-layer theory in the design of centrifugal compressors. Boundary-layer development in the diffuser entry region is shown to be important to stage efficiency. The result of an earnest attempt to analyze this boundary layer with the best tools available is displayed. Acceptable prediction accuracy was not achieved. The inaccuracy of boundary-layer analysis in this case would result in stage efficiency prediction as much as four points low. Fluid dynamic reasons for analysis failure are discussed with support from flow data. Empirical correlations used today to circumnavigate the weakness of the theory are illustrated.

In centrifugal compressors, there are two key boundary-layer situations which powerfully influence stage efficiency. They are diffusion in the impeller of the inlet relative velocity and diffusion (pressure recovery) in the diffuser (stator).

Figures 1a and 1b illustrate the influence on stage efficiency of these two diffusion processes. The impeller diffusion is measured by the ratio DR of the inlet relative velocity over the relative velocity at the separation point in the impeller. After this point, no further relative diffusion is possible.

The diffuser (stator) diffusion is measured by its pressure recovery coefficient C_p, which is the static pressure rise from impeller tip to the stage outlet plenum divided by the difference between stagnation and static pressure at impeller tip.

Figures 1a and 1b have been constructed from our detailed centrifugal analysis methods, which agree well with the actual test points spotted on the curves. One can see that, in order to reach the ultimate performance discussed below, improvement in DR and C_p will be about equally important and for both pressure ratio 3 and 10 stages.

Impeller diffusion involves complex flow situations that are very inadequately understood today. These include boundary-layer separation

and free-shear flow under the influence of Coriolis forces, separated flow in a rotating coordinate system, transonic flow over the blades and tip leakage, and secondary flow. Johnston (ref. 1) has surveyed the slight knowledge of the Coriolis effects.

Today so little is known fundamentally about most of these impeller flow situations that little can be said positively about boundary-layer analysis in the impeller. We choose, therefore, to discuss here the region between the impeller tip and diffuser throat. Boundary-layer growth there has a direct and serious influence on diffuser recovery.

Before proceeding to the fluid dynamics, it is valuable to sketch the context of the problem, in order to put the specific matter in proper focus.
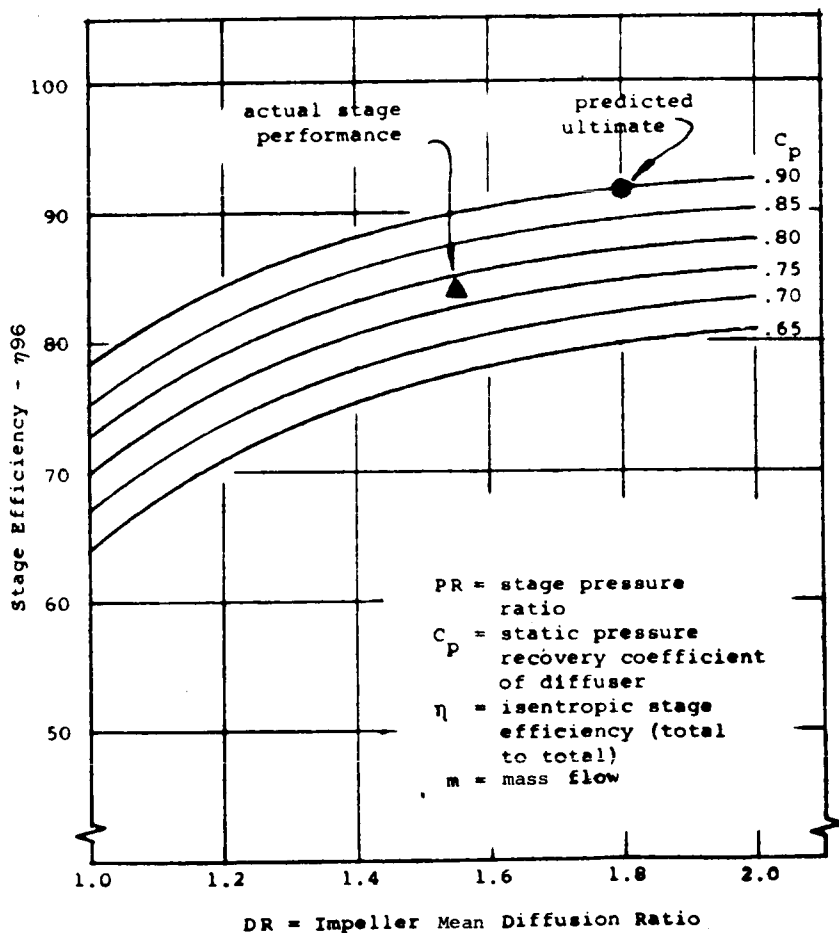

FIGURE 1a.—*Effect of impeller mean diffusion ratio on stage efficiency, for P.R.=3, m=3 lb/sec.*

CONTEXT

For years, the centrifugal compressor has been out of favor in many circles for use in high-performance gas turbines; but, within the past 10 years, the military small gas turbine has advanced so much in specific power that its dropping flow rate per unit power again favors the centrifugal compressor. Today, axicentrifugal machines with two to six axial stages ahead of a pressure ratio 3 to 6 centrifugal stage are common. Within 5 years, new engine designs may incorporate only a single centrifugal stage operating at pressure ratio 10 to 15 or with one or two axial
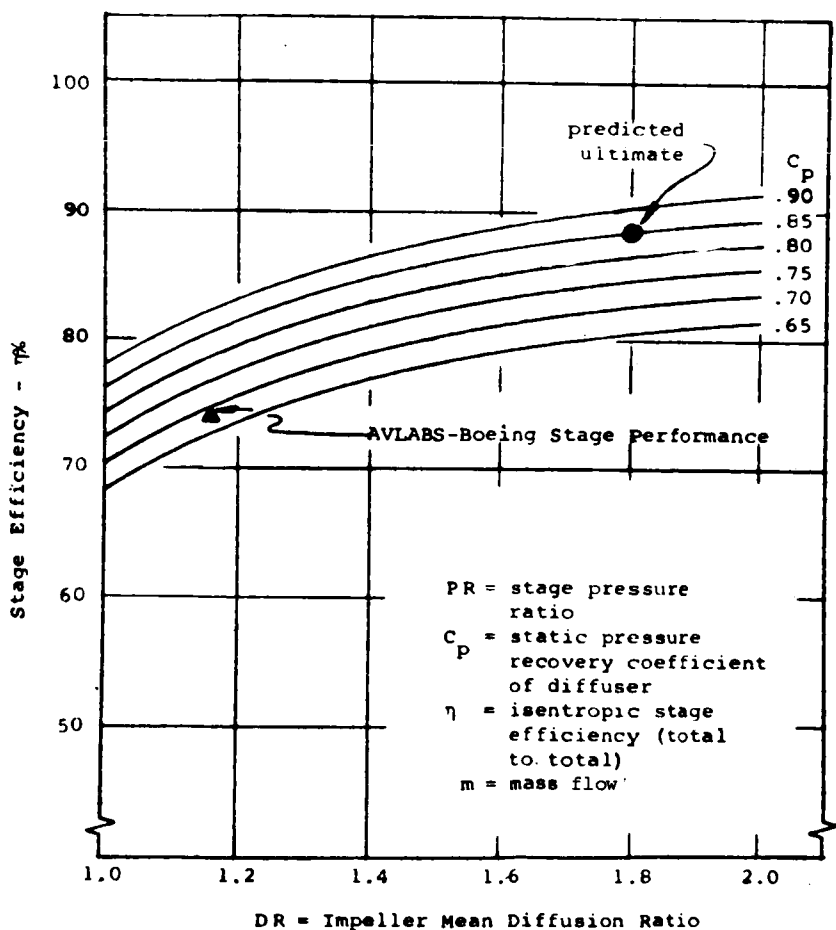

FIGURE 1b.—*Effect of impeller mean diffusion ratio on stage efficiency, for P.R.=10, m=2 lb/sec.*

precompression stages giving overall compressor pressure ratios of $20X$ to $30X$.

By 1980, small gas turbine engines should be demonstrated with centrifugal compressors of 12 to 15 pressure ratio and 80- to 85-percent total-to-static stage efficiency. A forecast comparing 1970 and 1980 engines is shown below.

Expected Performance of Demonstration Engines

	1970	1980
pr	15–20	20–30
pr$_{\text{cent}}$	3–6	12–15
η_{cent}	80–83	80–85
TIT	2000–2200	2500–3000
sfc	0.5	0.35
swt	0.3	0.2
SHP	200–250	250–300

where pr = pressure ratio
      TIT = turbine inlet temperature, °F
      η_{cent} = centrifugal compressor total-to-static adiabatic efficiency
      sfc = engine specific fuel consumption, lbm/hp–hr
      swt = engine specific weight, lbm/hp
      SHP = specific power = engine hp/air-flow rate, hp/lbm/sec

These prospects have focused considerable technical attention on the small centrifugal compressor. Of prime interest has been the helicopter engine. The U.S. Army Aviation Materiel Laboratories (AVLABS) lately has been the principal sponsor of research and advanced concept demonstration.

Recently, Dean, Wright, and Runstadler (ref. 2) have critically reviewed contemporary fluid design methods and sought means to reach ultimate performance. This paper is based on that work.

IMPORTANCE OF THE DIFFUSER

Figure 2 presents the state of the art for small centrifugal compressors and shows our forecast of ultimate performance (ref. 3). The forecast is not a guess, but is based on detailed calculations and performance attained with other related flow devices.

Figures 1a and 1b show the importance of diffuser performance in reaching the ultimate performance of machinery. Figure 3 displays the
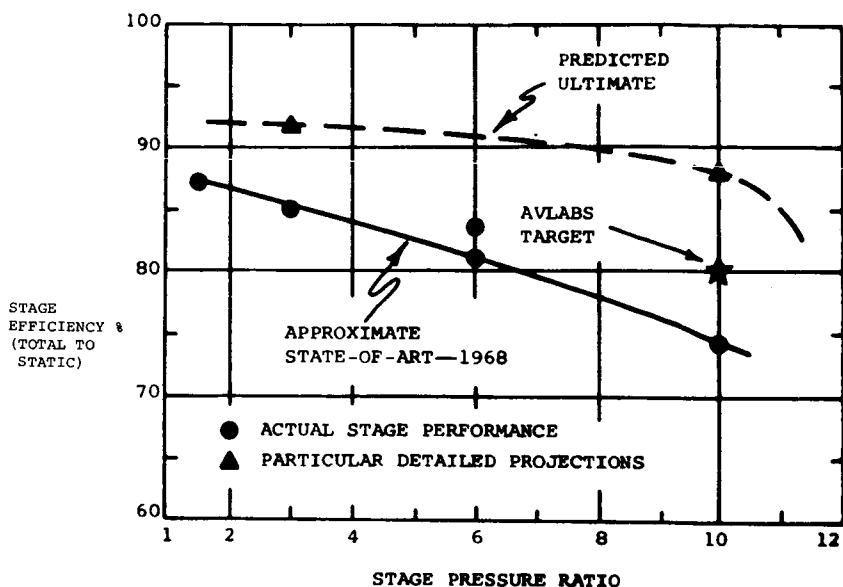
FIGURE 2.—*State-of-art centrifugal compressor performance and predicted ultimate (from ref. 3).*

state of the art of diffuser performance and our forecast of ultimate recovery, again based on detailed calculations and related flow data.

A study of this set of graphs proves that diffuser recovery must advance quite a lot before the ultimate centrifugal stage will be realized. A 1-point gain in C_p (e.g., 0.75 to 0.76) is worth about $\frac{1}{3}$-point gain in compressor efficiency (for pressure ratio$=10$). In turn, a $\frac{1}{3}$-point gain in compressor efficiency gives about a 1-percent reduction in fuel consumption. So, even small gains in diffuser performance lead to attractive gains in engine performance.

The sizable gains needed in the diffuser must be won from a difficult flow device with very shallow passages (aspect ratio $\frac{1}{10}$ to $\frac{1}{20}$), large viscous effects, shocks, and unsteady and three-dimensional transonic flow.

DIFFUSER FLOW MODEL

Before we consider the entry boundary layer, it is necessary to appreciate the rudiments of diffuser flow and the model we shall use to predict it.

An analytical model is essential in this complex flow situation before any real understanding can be had. No significant models existed in the
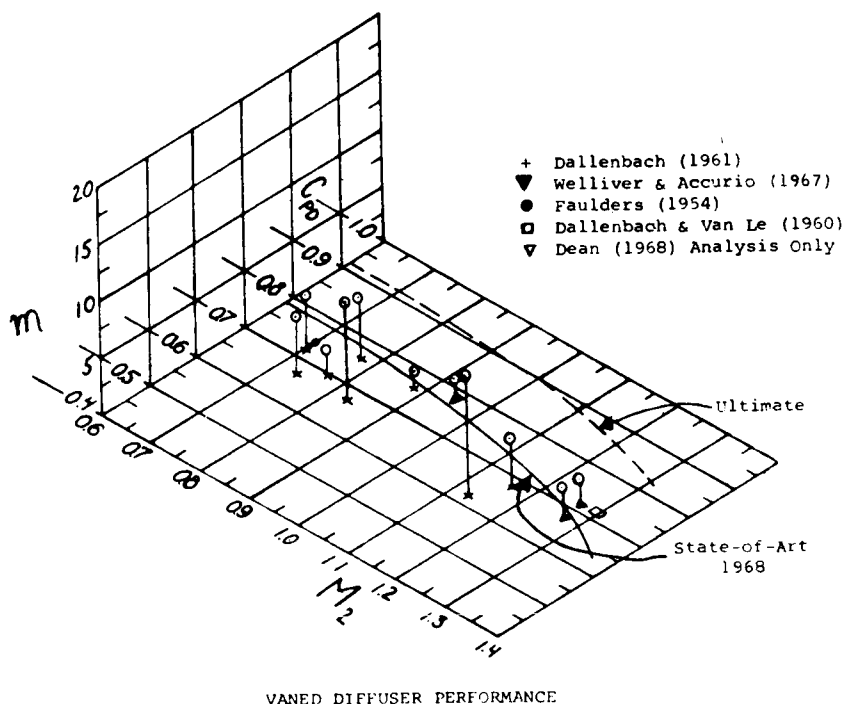
FIGURE 3.—*State-of-art centrifugal compressor diffuser performance and projected ultimate (from ref. 3).*

literature before Welliver and Acurio (ref. 4) published the model we use here. They also published the first extensive set of high-quality data on the centrifugal diffuser; this alone was a major contribution. Dean, Wrignt and Runstadler (ref. 2) used these data in their analysis.

The Welliver and Acurio diffuser data have a number of striking features. First, there is a shock at channel diffuser entry, as many schlieren pictures like figure 4 and wall pressure data revealed. This shock appeared when the impeller tip discharge Mach number rose just above 1.0. Secondly, there was always a remarkable pattern in the isobaric plots made from many taps on both walls.

Note in figure 5 that the isobarics roughly parallel the swirling streamlines in the vaneless and semivaneless spaces up to just ahead of the entry shock. This means that little supersonic diffusion occurred.[1] Indeed, only 10 percent of the diffuser's pressure recovery was gained up to the shock.

----

[1] With a much larger vane leading-edge radius ratio (to impeller tip radius), supersonic diffusion will occur to subsonic values and without an entry shock.
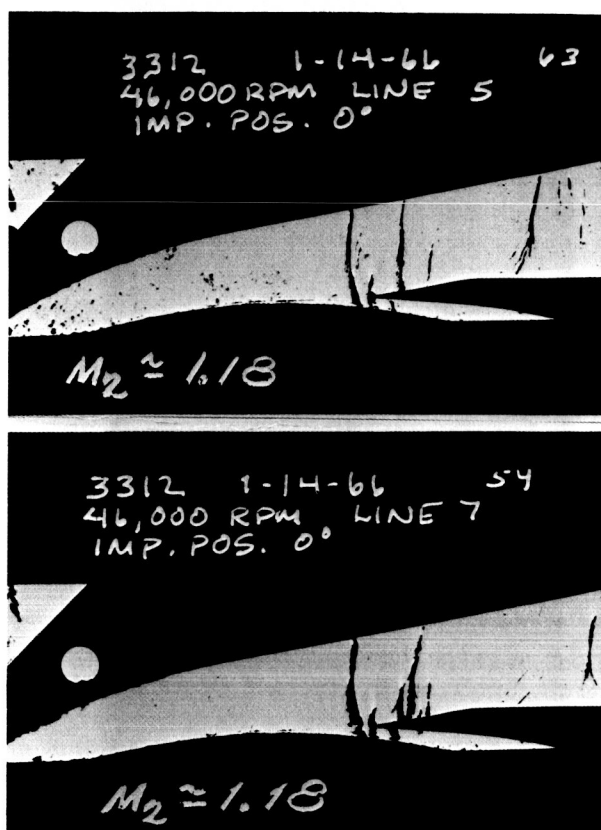
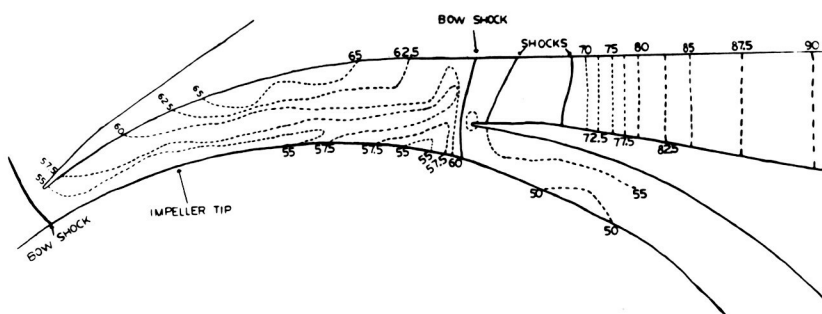FIGURE 4.—*Schlieren photographs of high-Mach-number centrifugal compressor diffuser flow (from ref. 4).*

FIGURE 5.—*Isobarics measured in high-Mach-number flow of centrifugal compressor vane-island diffuser (numbers on contours are pressures, psi) (from ref. 4).*

Just ahead of the shock there is a zone of very rapid adjustment. Here the flow Mach number decreased or increased suddenly to produce a normal-like shock of proper strength to match the subsonic throat conditions downstream. This zone of rapid adjustment was accompanied by sharp streamline curvature near the vane leading edges. The shock typically gave 30 percent of the overall pressure recovery.

Behind the shock, the flow pattern becomes one-dimensional immediately. Note, in figure 5, that the isobarics are perpendicular to the passage centerline even within a fraction of a throat width downstream of the channel entry. The flow pattern changes in less than a throat width from a swirling flow to a straight flow; this seems remarkable to us. This characteristic is not a supersonic phenomenon; subsonic relaxation solutions produce the same pattern, without the shock, of course. Apparently, this flow pattern is a dominant characteristic of the peculiar geometry of such compressor diffusers.

In the diverging channel diffuser, the flow is so one-dimensional that one immediately thinks of modeling this region as a simple isolated straight diffuser. This is what was done in the model.

The channel diffuser gave typically 55 percent of the overall pressure recovery. At the end of the channel, the flow was dumped into a collector in Welliver and Acurio's case. No pressure recovery occurred because of the large dump area ratio.

This diffuser flow was modeled for analysis as shown in figure 6. First, there is a region of mixing of the impeller's (r,θ) plane distorted discharge flow. This region was assumed to be of zero radial extent, in accord with the theoretical results in Johnston and Dean (ref. 5). The mixed-out flow properties are computed by a compressible version of the Dean and Senoo (ref. 6) theory.[2]

No meridional mixing is assumed. To date, backflow into the impeller, which has been observed in the data, has not been specifically incorporated into the diffuser model.

After the thin (r,θ) plane mixing region, the flow is assumed to proceed with constant stagnation temperature and pressure on the passage centerstream surface right through the shock (with a shock Δp_o correction) to the diffuser throat. The centerline flow Mach number is assumed to be constant up to just ahead of the shock when it suddenly changes to give a shock strength that satisfies throat conditions. The shock is assumed to be thin and "normal".[3]

Flow centerline properties after the shock are taken equal to throat centerline properties unless the throat is of finite length. In that case, the

---

[2] Found in reference 2, Appendix II.
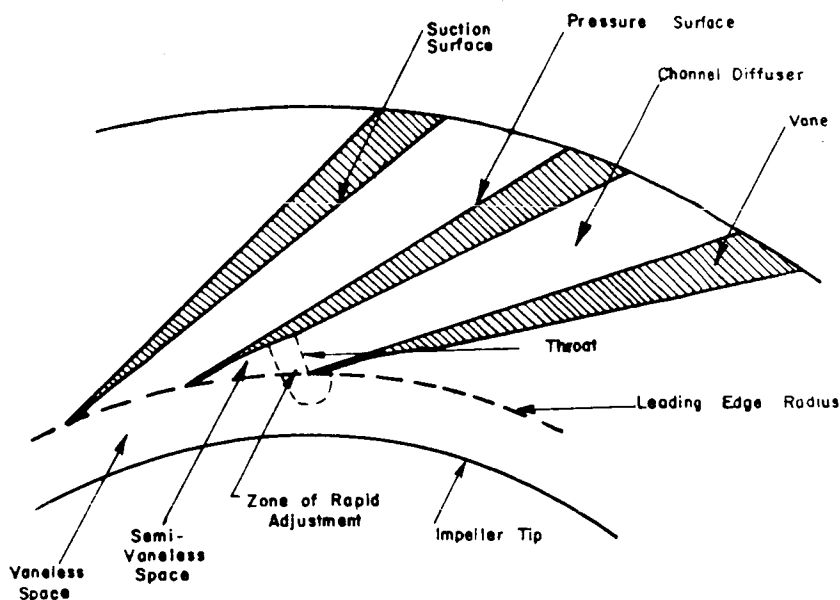[3] I.e., one-dimensional, normal shock theory.

FIGURE 6.—*Principal flow regions in high-Mach-number vane-island diffuser flow (from ref. 2).*

one-dimensional duct flow relations with wall friction (e.g., Shapiro, ref. 7) are used to calculate the centerline Mach number and other flow properties at the entrance to the diverging channel.

As we shall show shortly, boundary-layer blockage in the diffuser throat is a key variable. In the model, blockage was to be predicted by two-dimensional, compressible, turbulent boundary-layer theory starting at zero thickness at the impeller tip.[4] This paper is concerned primarily with the success of this last feature of the model. Before we review that, we must appreciate how sensitive performance predictions and design optimization are to the precision of the boundary-layer theory results. According to the flow model, the only element affected is the channel diffuser.[5]

CHANNEL DIFFUSER CHARACTERISTICS

Once the flow model was put together, Welliver and Acurio proceeded to test it. Channel diffuser data was produced by Runstadler (ref. 14) on

---

[4] See reference 2 for further details and discussion of the assumptions.

[5] Boundary-layer growth has a nil continuity effect in the semibounded space ahead of the channel entry.

simple laboratory diffusers which correlated very well with data from the
Boeing compressors as shown in figure 7. This early data showed that
throat blockage B_4 was the most powerful variable governing maximum
attainable C_p (with complete flexibility in choosing channel diffuser
L/W and 2θ). Throat Mach number was a weak variable until well into
the supercritical range (throat centerline Mach number $= 1.0$), when the
channel shock[6] separated the diverging passage.

The success of the model for the channel diffusers caused AVLABS to
commission us to produce a full range of transonic and subsonic data for
simple, flat symmetrical diffusing passages. This work is reported by
Runstadler (ref. 8). With 2300 tests, he covered a wide range of Mach
number, Reynolds number, throat boundary-layer blockage, throat aspect
ratio, length-over-throat-width ratio and divergence angle. Typical maps
are shown in figures 8a, 8b, and 8c. The agreement of the laboratory and
compressor diffuser data is shown in figures 9a, 9b, and 9c. In general, the
compressor diffuser performed a little better than the laboratory diffuser.
We consider the agreement to be good despite the large uncertainty band
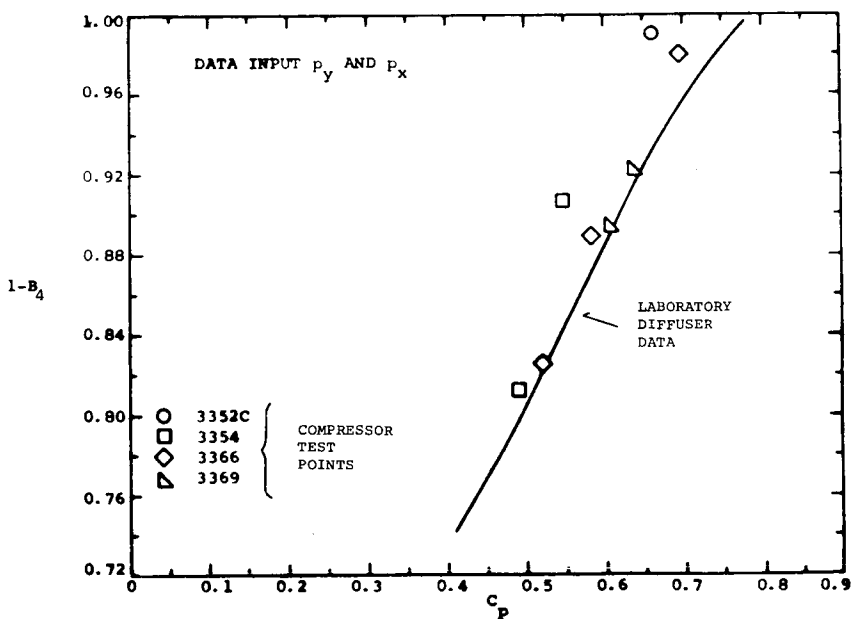of the compressor data.

FIGURE 7.—*Comparison of pressure recovery for compressor channel with laboratory diffuser data (from ref. 2).*

[6] A second shock which appears as the back pressure is lowered. The entry shock remained fixed in position over the full compressor flow range.
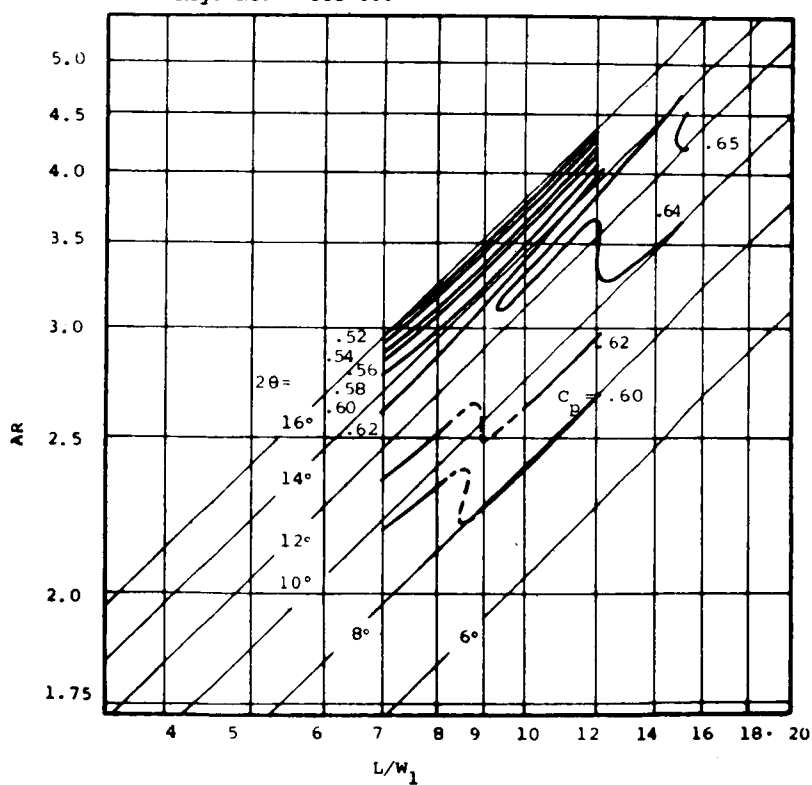
AS = 0.25
M   = 1.0
B   = 0.10
Rey. No. = 588 000

FIGURE 8a.—*Straight-channel diffuser performance (from ref. 8).*
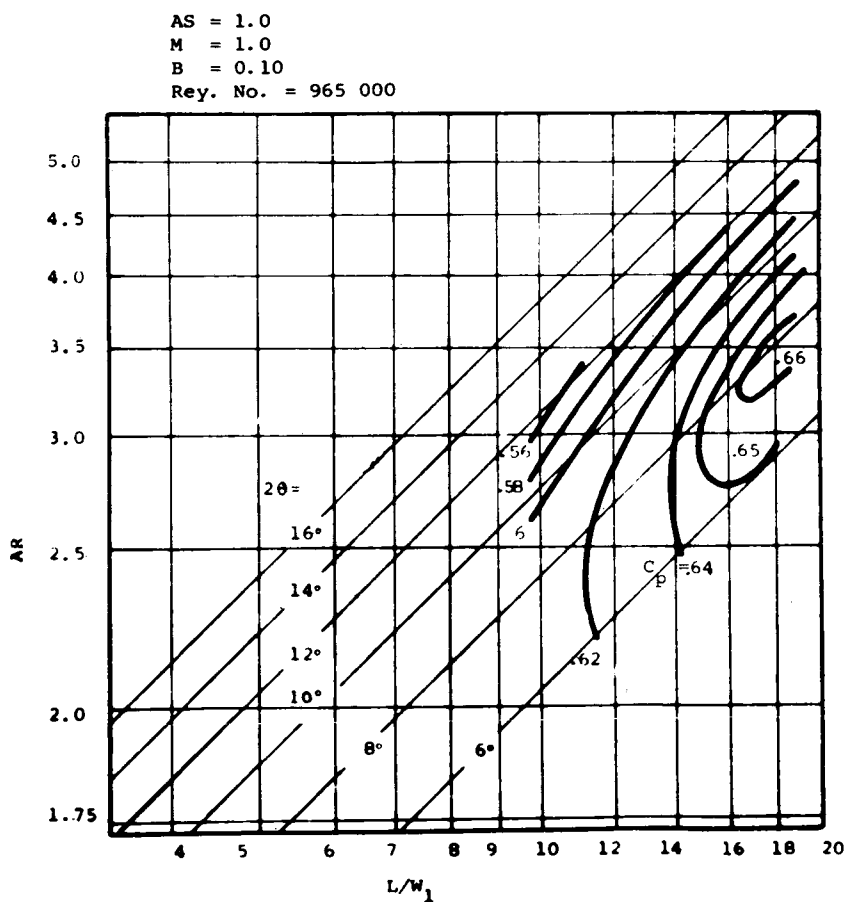
AS = 1.0
M   = 1.0
B   = 0.10
Rey. No. = 965 000

FIGURE 8b.—*Straight-channel diffuser performance (from ref. 8)*.

AS = 5.0
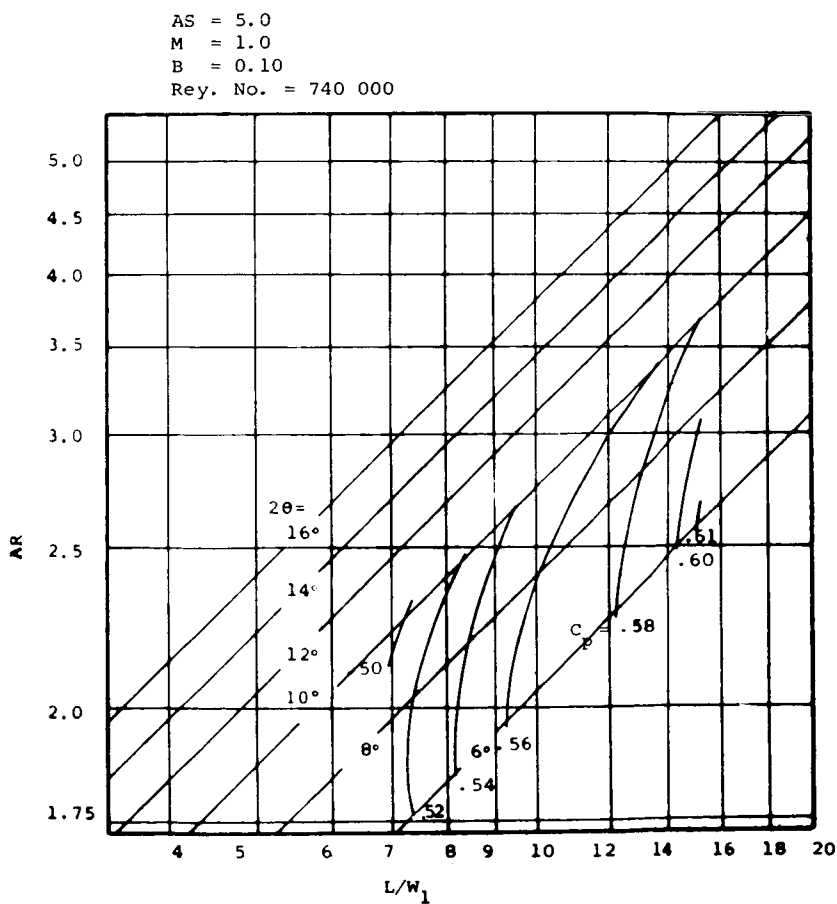M   = 1.0
B   = 0.10
Rey. No. = 740 000

FIGURE 8c.—*Straight-channel diffuser performance (from ref. 8).*

FIGURE 9a.—*Comparison between Boeing-AVLABS channel diffuser data from compressors with Creare-AVLABS straight-channel diffuser data. Pressure recovery C_p versus throat blockage B_4 and Mach number M_4 (from ref. 2).*

FIGURE 9b.—*Comparison between Boeing-AVLABS channel diffuser data from compressor with Creare-AVLABS straight-channel diffuser data. Pressure recovery* C_p *versus throat blockage* B_4 *and Mach number* M_4 *(from ref. 2).*
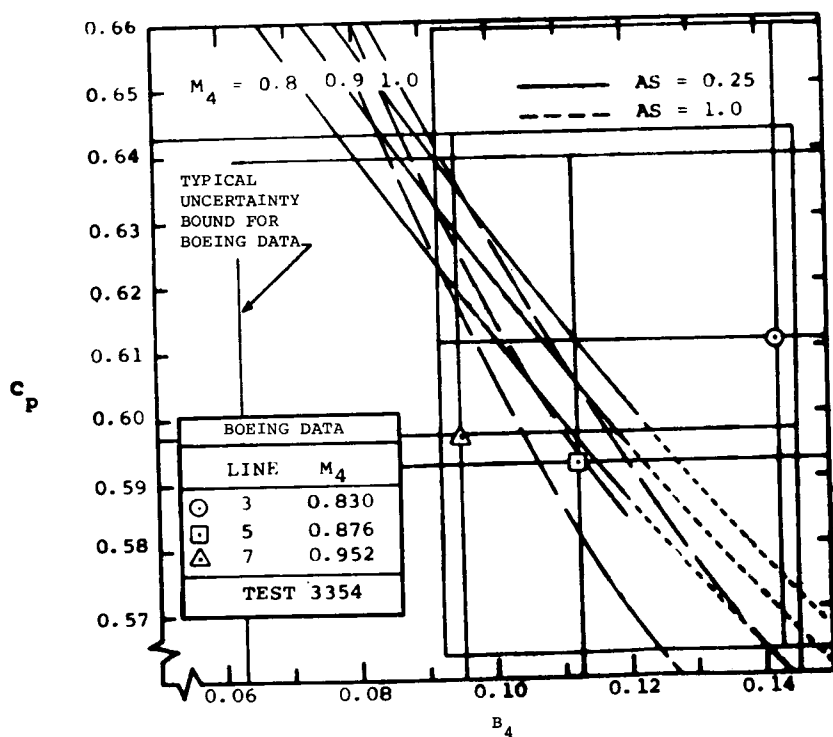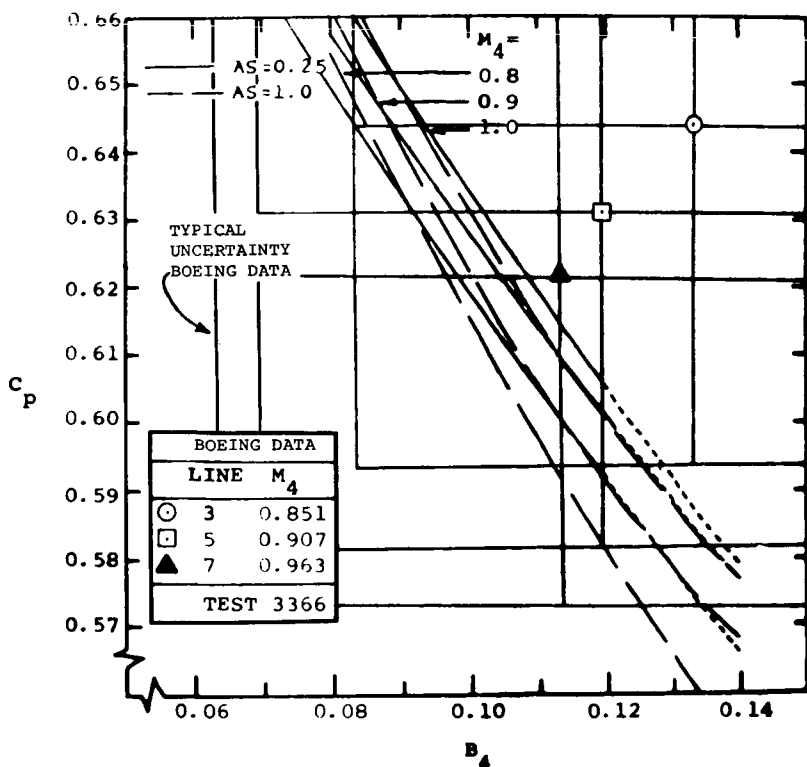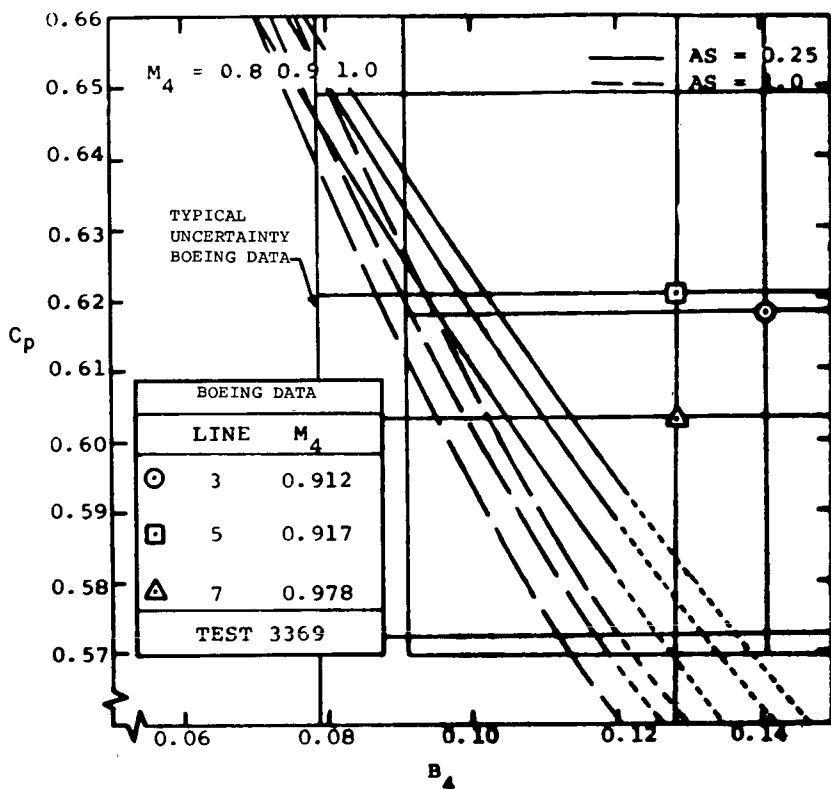
FIGURE 9c.—*Comparison between Boeing-AVLABS channel diffuser data from compressors with Creare-AVLABS straight-channel diffuser data. Pressure recovery* C_p *versus throat blockage* B_4 *and Mach number* M_4 *(from ref. 2).*

The strongest characteristics of these maps are shown in figures 10a, 10b, and 10c. Throat blockage, B, and aspect ratio, AS, have the greatest effect on maximum attainable channel diffuser pressure recovery (the peak of the "hill" on the maps of figure 8). Mach number up to 1.0 had little effect, as can be seen in figure 10. Dean (ref. 3), Runstadler and Dean (ref. 9), Runstadler (ref. 8) and Dean, Wright, and Runstadler (ref. 2) discuss at length the implications of these empirical findings in centrifugal diffuser design and optimization.

The important aspect for our purposes here is the dependence of channel diffuser and overall diffuser performance on throat boundary-layer blockage. This variable is what the boundary-layer theory, applied from impeller tip to throat, attempts to predict. If that prediction is inaccurate, figure 10 shows the consequences. Remember that the channel diffuser gave 55 percent of the overall recovery and that 1-point variation in
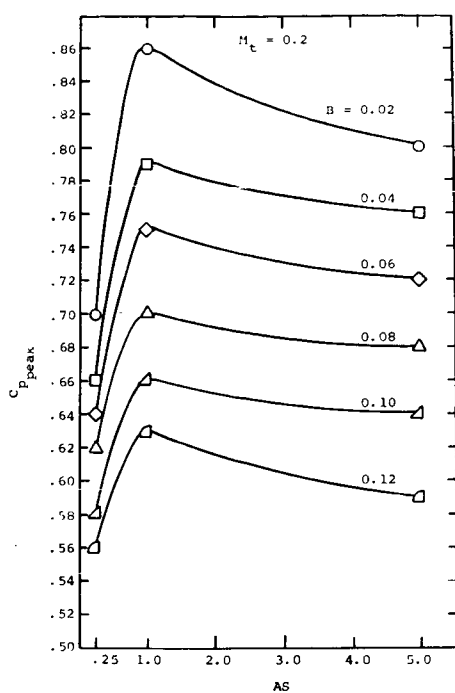
FIGURE 10a.—*Peak pressure recovery versus aspect ratio; Mach number=0.2 (from ref. 8).*
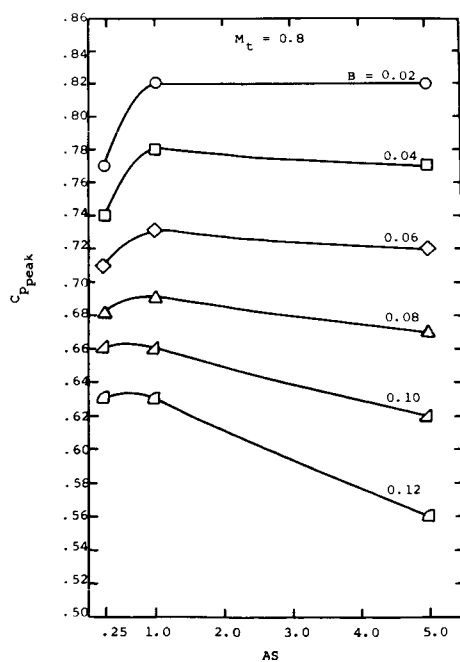
FIGURE 10b.—*Peak pressure recovery versus aspect ratio; Mach number=0.8 (from ref. 8).*
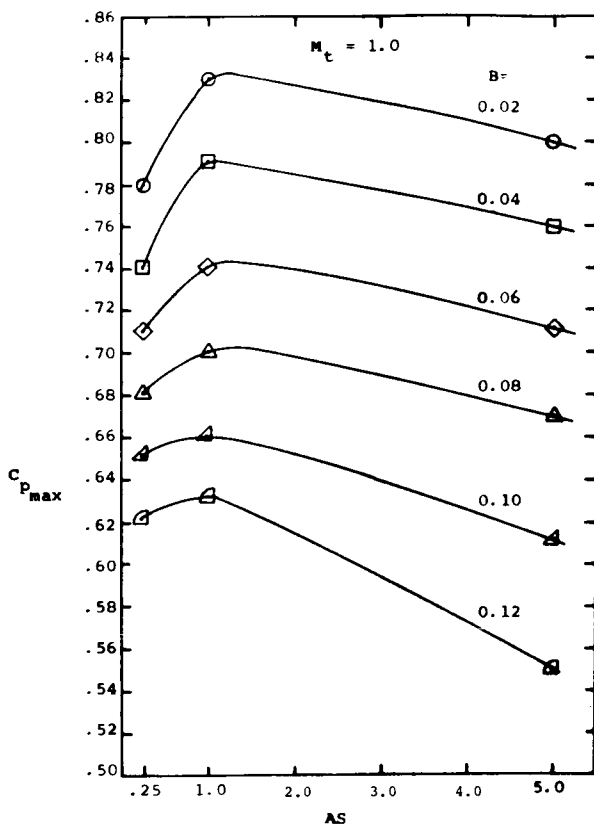
FIGURE 10c.—*Peak pressure recovery versus aspect ratio; Mach number = 1.0 (from ref. 8).*

overall C_p amounts to $\frac{1}{3}$-point in stage efficiency and about 1-percent variation in engine fuel consumption.

BOUNDARY-LAYER ANALYSIS AND RESULTS

In order to predict throat blockage, we attempted a wide variety of theoretical attacks using two- and three-dimensional turbulent boundary-layer theory and shock/boundary-layer interaction theory. Both Mellor (Princeton) and Johnston (Stanford) concluded that three-dimensional theory was too weak to handle the complex flows discussed here. So we used the method of Englert (ref. 10) which appeared to be the most competent available for transonic flow on the basis of the Stanford Conference (1968) results.

The measured pressure distributions of Welliver and Acurio were employed as input. Because initial conditions at impeller tip were very uncertain, the starting boundary-layer thickness and shape factor were varied over wide ranges. The details will not be repeated here; they are fully presented in reference 2.

After due consideration of the use of various shock/boundary-layer interaction theories, all of which have been developed for external flow, they were abandoned. This was because of the shallow passage and thick boundary layers in the compressor. Instead, the Englert calculations were pressed right on through the measured shock pressure rise to the throat.

The results of all of this calculation are discouraging. They are shown in figure 11. Note that the "best" boundary-layer theory results gave throat blockage values for various tests that were about *two times* higher than the measured data as reduced both by Boeing and by us. This error is of grave consequence. Note, in figure 10, that a theoretical blockage of 0.20, compared to the measured value of 0.10, would give about 20 points (estimated) lower channel diffuser recovery, which would amount to a reduction in stage efficiency of about 4 points. One probably would not

Calculation Scheme	3354--7**	3366--5	3369--5
From Boundary Layer Calculation	0.2	0.23	0.27
Boeing Best Estimate Using $P_{o_{4CL}}$, P_y, Data	0.096	0.111	0.126
From $P_{o_{4CL}}$, P_4, Data	0.095	0.119	0.128

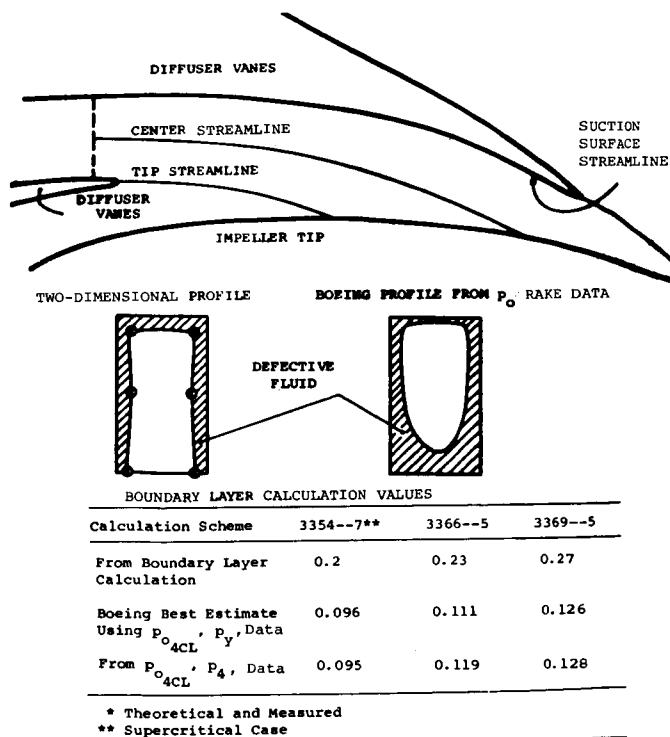* Theoretical and Measured
** Supercritical Case

FIGURE 11.—*Throat blockage comparison (from ref. 2).*

even build the compressor if one believed the boundary-layer theory results.

We conclude that the two-dimensional boundary-layer theory is unsatisfactory for this work. Why does it fail?

THE FLUID MECHANICS OF THE DIFFUSER ENTRY REGION

In order to understand why the theory fails, we must understand the flow. That is very difficult today for many reasons.

First, there is little good data. Even the "good" data is highly suspect for reasons considered in detail in reference 2. The principal reasons are the narrow passages (order 0.2 inches deep in the Boeing compressor), instrument distortion of the flow, gross unsteadiness, high oscillation frequencies (blade passing frequency≃20 KC), distorted flow from the impeller, three-dimensional shear flow, strong mixing, shocks, transonic flow, etc. Proven stagnation temperature measurement errors were on the order of 20° F. Backflow was suggested by a compendium of all the data, but could not be resolved by the instruments. The best modern instrument practice was used by Welliver and Acurio, with uncertain data results.
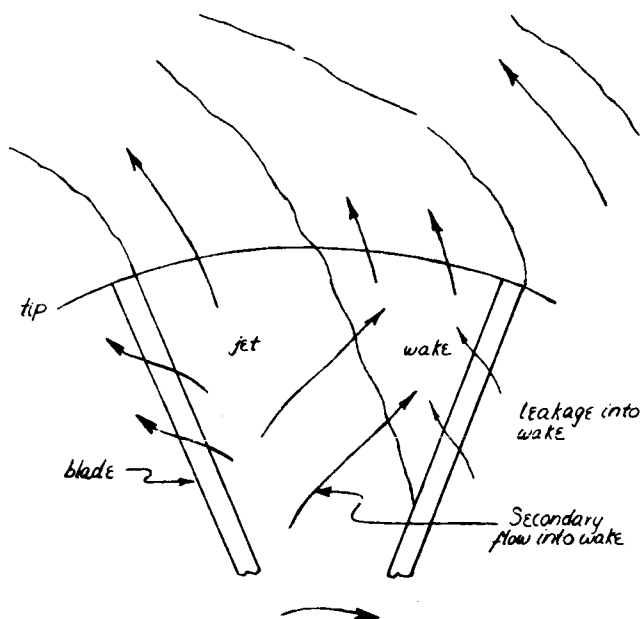

FIGURE 12.—*Wake-jet flow pattern at exit of centrifugal impeller.*

Despite the paucity of data, there are certain flow characteristics that we can deduce. The full diagnosis is displayed in reference 2. Here, we will repeat only the essential conclusions.

The flow from the Boeing impeller was separated grossly in the blade-to-blade plane. The wake width, figure 12, was about 75 percent and the jet velocity about 900 fps.[7] This discharge pattern leads to a very unsteady absolute flow as shown by hot-wire traces in figure 13 from another, low-speed, and much less separated impeller.

The weakness of the radial outflow in the wake region encourages back-flow off the diffuser sidewalls as suggested in figure 14. Such backflow has
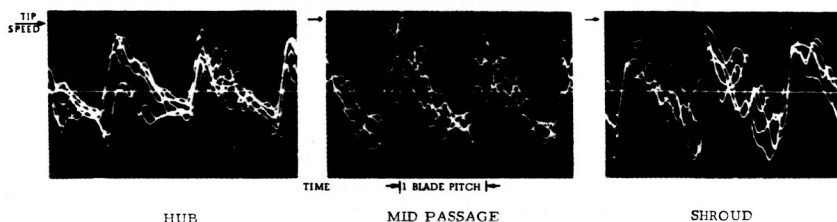

FIGURE 13.—*Typical radial hot-wire anemometer traces for a particular impeller passage measured at impeller tip (multiple traces superimposed) illustrating relative unsteady flow. (Approximately three channels are shown).*
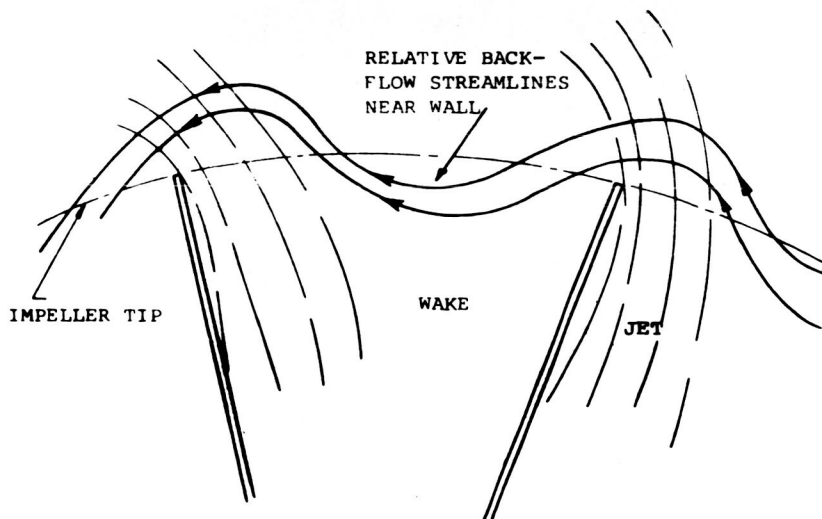
FIGURE 14.—*Schematic of backflow pattern in rotor space.*

[7] Tip speed approximately 2000 fps.

been seen by many observers. For instance, Faulders (ref. 11) and Johnston (ref. 1) observed it plainly in a static diffuser rig as shown in figures 15 and 16. In the Boeing case, time-average wall traces, shown in figure 17 by oil streaks in the schlieren windows, suggest mild backflow. We feel the time-average streaks are meaningless because the backflow is probably transient in stator coordinates as the wakes move past and travel through the strong pressure fields of the diffuser vanes (fig. 18). There may be important acoustic wave and resonance phenomena acting
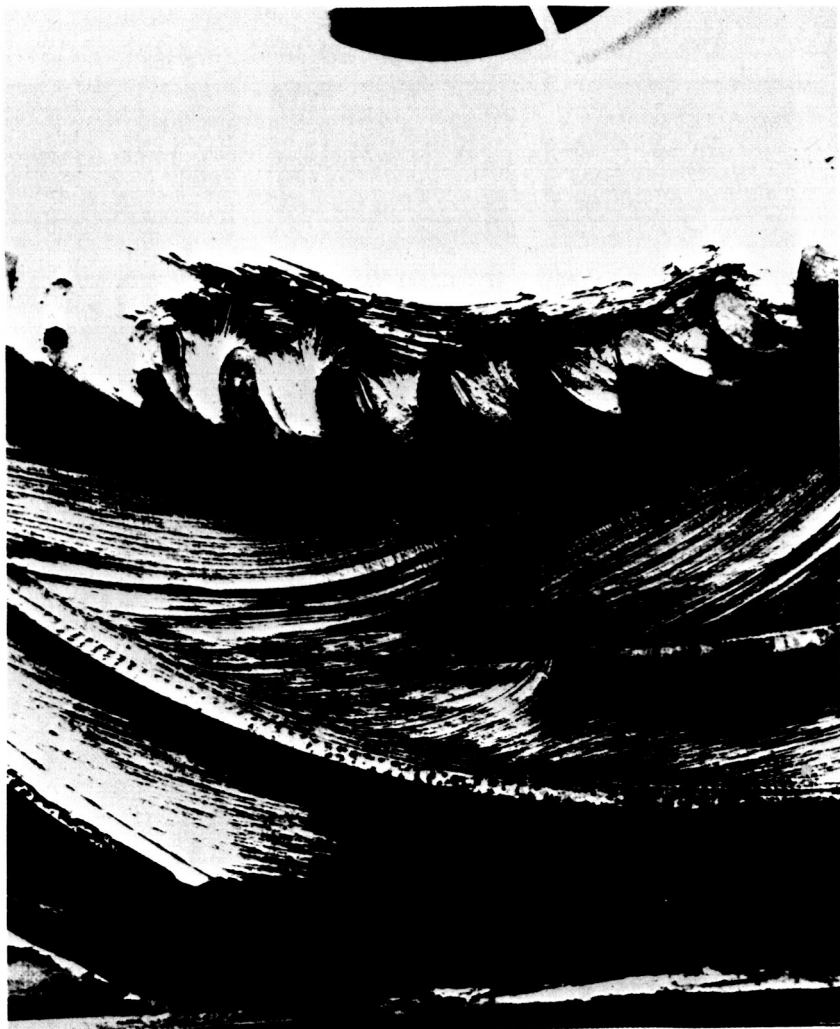
FIGURE 15.—*Limiting wall streamline traces from vaned diffuser flow (from ref. 15).*
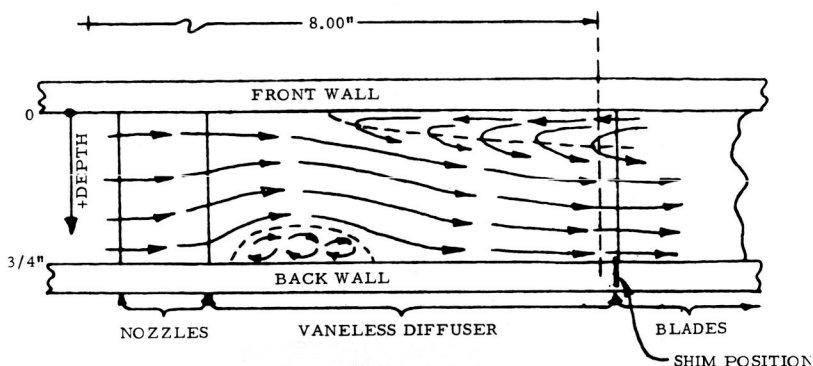
FIGURE 16.—*Flow in vaneless region of vaned diffuser (from ref. 15).*
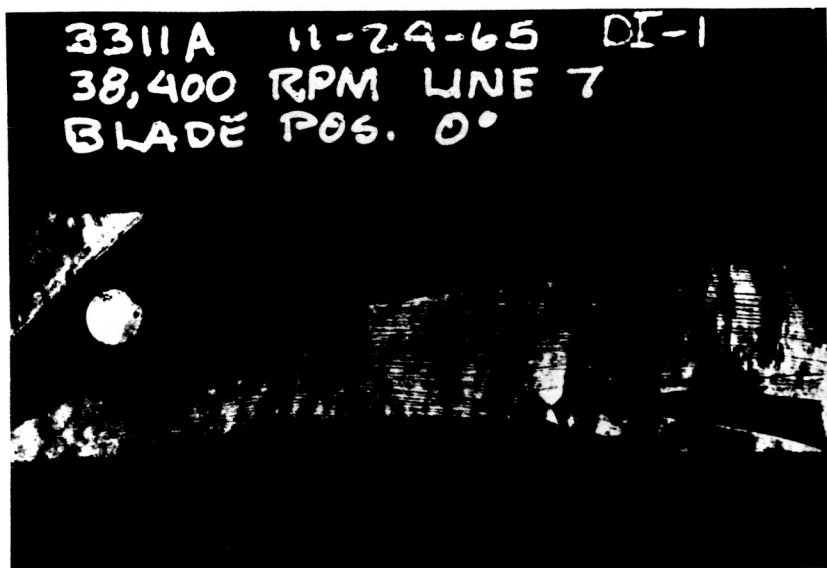
FIGURE 17.—*Oil traces in vaneless region of vaned diffuser flow (from ref. 2).*

in the impeller and diffuser channels. It should be mentioned that the
diffuser shock pattern was unaffected by the impeller blade position
relative to the diffuser vanes (see ref. 4), but, plainly, the flow conditions
at the impeller tip are very unsteady.

We conclude that the boundary layer on the diffuser sidewalls was
flowing backwards at certain times in an unsteady, three-dimensional way.
Perhaps this backflow, and consequent mixing with and energizing of the
return flow by the outflow, can account for the large prediction error of
the two-dimensional boundary-layer theory.
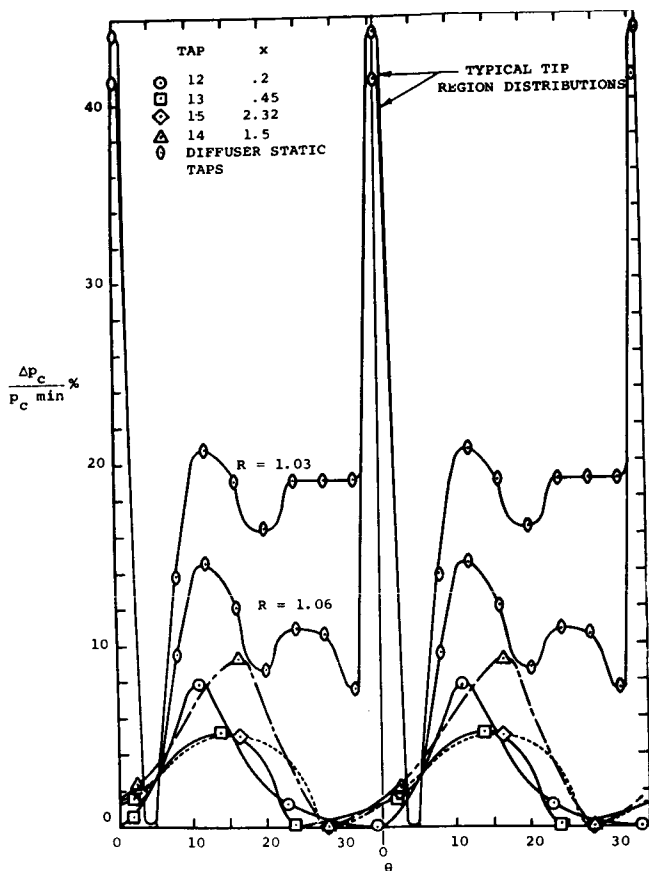
FIGURE 18.—*Tangential variation of cover pressure (from ref. 2).*

In addition to this, there is strong three-dimensional mixing occurring in the entry region. This is driven by the (r,θ) plane mixing of the impeller discharge jet-wake pattern. The fluid near the sidewalls will move at a lower tangential velocity than the core flow. Thus the (r,θ) plane distortion results in an (x,θ) plane distortion and shear, shown schematically in figure 19 for the (r,θ) plane and figure 20 for the (x,θ) plane. We think this (x,θ) distortion leads to the rollup of eddies which cause strong momentum transport perpendicular to the diffuser sidewalls. The consequence of this action should be a loss in core stagnation pressure, while the profile should be flattened at the same time. Then p_o would be lowered in the throat along with a blockage decrease. We could not prove a p_o loss on the centerline in the entry region, although the data diagnosis is quite uncertain. This question is examined in detail in reference 2.
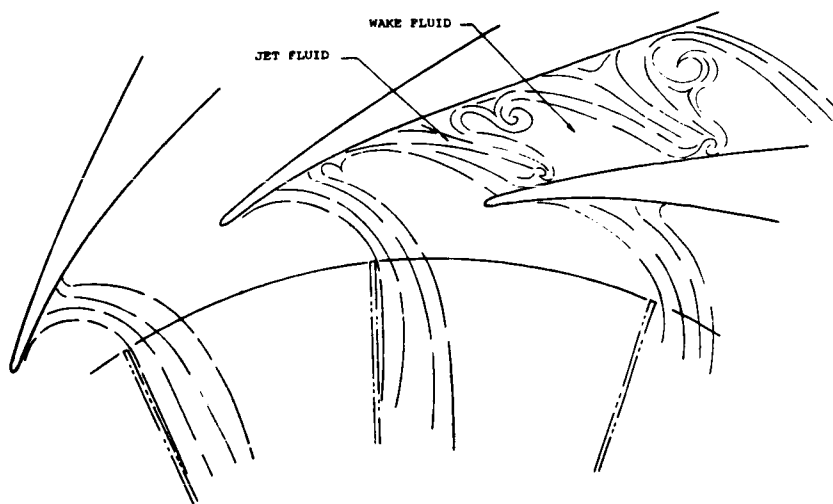
FIGURE 19.—*Schematic of the progression through the diffuser of wakes and jets from the impeller (from ref. 2).*

FIGURE 20.—*Schematic of vortex roll-up on interfaces between jet and wake (from ref. 2).*

According to Bradshaw (ref. 12) the very strong turbulence and mixing and gross unsteadiness of the free stream probably obviates the validity of the usual boundary-layer theory. This too may be a cause of the prediction inaccuracy.

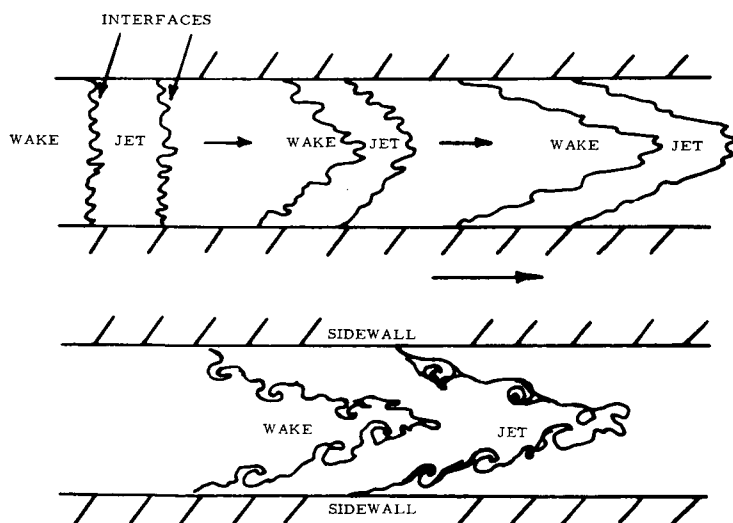Finally, the entry shock may not affect the boundary-layer thickness as a simple boundary-layer theory would claim. In deducing the possible shock effect on blockage, we note from Welliver and Acurio that the shock produced approximately the simple normal-shock pressure rise. That the data are somewhat uncertain on this point must be noted, however. The principal difficulty was that the taps were not spaced closely enough to detect always the minimum pressure at the foot of the shock in the zone of rapid adjustment. Yet, despite the uncertainties, there is enough data to conclude that the shock upstream Mach number and the pressure rise across the sometimes spread-out shock correspond to simple normal-shock theory. This is the same behavior observed by Neumann and Lustwerk (ref. 13; see Shapiro, Vols. I and II) for very spread-out shocks in constant-area ducts.

Given "normal-shock" behavior, we expect to be able to apply the simple normal-shock theory in the core of the flow. Across the shock, the mass flux per unit area ρC is constant. Therefore, for the compressor diffuser entry shock, continuity asserts that the area of the core flow must be constant. Because the passage geometrical area does not change much across the shock, then the boundary-layer flow area must be constant, too, or since

$$\rho C (A - \delta^*) = \text{constant}$$

$$\frac{\partial \rho C}{\rho C} + \frac{\partial A - \partial \delta^*}{A - \delta^*} = 0$$

with

$$\partial \rho C = \partial A = 0$$

Then

$$\partial \delta^* = 0$$

This result says that the throat blockage should equal the blockage *ahead* of the entry shock. Also it says that increasing shock strength will not increase blockage, contrary to the assumptions in the Welliver and Acurio flow model. They claimed (with our support) that surge occurs because the entry shock strengthens with decreasing mass flow, making the throat blockage so high that the channel diffuser characteristic assumes an unstable positive slope $(\partial C_p / \partial m)$.

Actually, measured shock strength and throat blockage do rise with decreasing flow and surge does seem to occur when the measured diffuser $\partial C_p / \partial m$ goes positive. So whether the shock does or does not materially increase δ^* is not plain from the evidence. Perhaps the real case lies between a duct shock and a shock on an external surface. The partial confinement by the vane of the region in which the shock lies might be responsible for intermediate behavior.

Table I presents throat blockage calculated from the computed displacement thickness of the entry boundary layers at a station just *ahead* of the measured shock location. The agreement with the data is quite good for the three cases computed.

The shock/boundary-layer interaction argument presented above may resolve the discrepancy between boundary-layer theory and measurements. At this point we have too little reliable data to be sure. If this is indeed the answer to the dilemma, then the Welliver and Acurio surge model is incorrect in format. However, the diffusion from impeller tip to channel throat does rise as the flow is reduced (because throat Mach number drops) so δ^* and blockage will rise even if the shock is ignored.

Perhaps the true resolution of the difficulty with the entry boundary-layer prediction lies in a combination of some or all of the effects discussed above. All would tend to reduce blockage below that predicted by two-dimensional theory, but the sum effect we cannot calculate.

CORRELATIONS

At this impasse the designer still must design and as accurately as possible ($\pm\frac{1}{2}$-point uncertainty in stage efficiency prediction is desirable), so we have resorted to correlation of throat blockage versus diffusion from impeller tip to throat (fig. 21). The amount of good data available for making this correlation is woefully inadequate but it does give a better prediction than theory. We believe, but cannot prove, that the correlation can be scaled with streamline length $Re_y^{1/5}$.

TABLE I.—*Diffuser Throat Blockage (Measured Versus Calculated) With Two-Dimensional Turbulent Boundary-Layer Theory, Illustrating the Effect of Entry Shock on Calculated Blockage*

	Boeing test number		
	3354	3366	3369
Boundary layer calculation without shock____	0.108	0.098	0.120
Data from figure 11_____	0.095	0.118	0.128
Boundary layer calculation including shock___	0.2	0.23	0.27
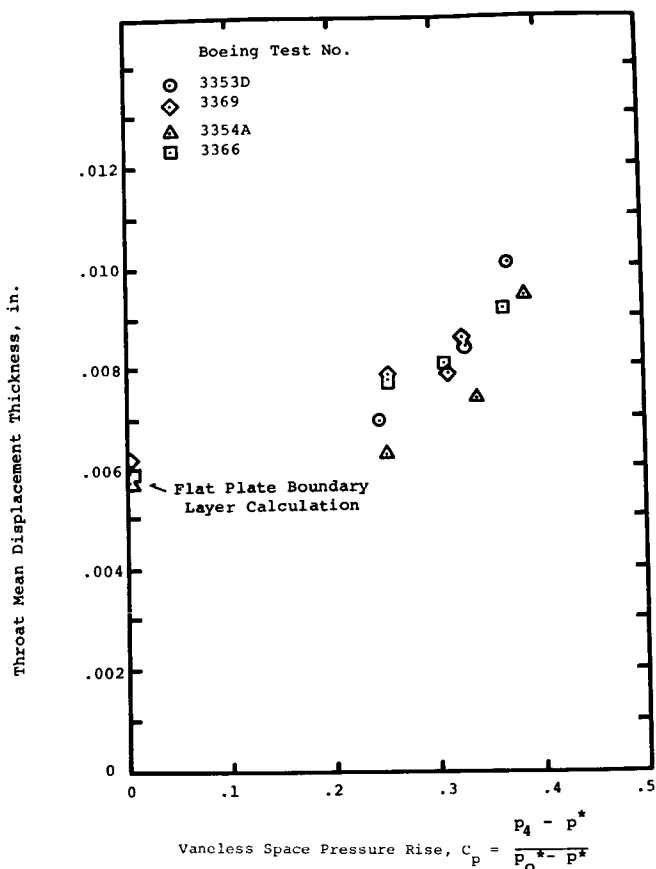
FIGURE 21.—*Correlation of throat blockage versus diffusion from impeller tip for the channel diffuser throat.*

CONCLUSION

We have now laid out our dirty linen and confessed we cannot make boundary-layer theory work for us for a vital prediction in the centrifugal compressor. Yet we are certain that achieving the ultimate compressor performance forecast in figure 2 will require powerful fluid dynamic tools. We do not think that boundary-layer theory will grow in competence to help much here in the next 10 years. The fact that the simple diffuser has defied analysis and that the shock in the duct problem has not even been explained experimentally discourages much hope of analyzing theoretically such a very complex flow as in the diffuser entry; that is,

within the next decade. Yet we are convinced that the small centrifugal compressor will be pushing its ultimate limits by 1980, even without much more than qualitative assistance from boundary-layer theory.

LIST OF SYMBOLS

A	Flow area (normal to mean velocity vector, specifically defined)
AR	Channel diffuser area ratio
AS	Channel diffuser throat aspect ratio
B	Boundary-layer blockage,

$$B \equiv 1 - \frac{A_{\text{effective}}}{A_{\text{geometrical}}}$$

C	Absolute velocity (relative to a Newtonian frame; e.g., compressor casing)
C_p	Pressure recovery coefficient,

$$C_p \equiv \frac{p_{\text{exit}} - p_{\text{ref}}}{\cdot (p_o - p)_{\text{ref}}}$$

where measuring and reference states and stations must be specifically defined

C_{pD} Pressure recovery coefficient,

$$C_{pD} \equiv \frac{p_{\text{coll}} - p^*}{p_o{}^* - p^*}$$

where $* \equiv$ mixed-out state (must be specifically defined)

DR Diffusion ratio,

$$DR = V_1/V_{\text{sep}}$$

h	Static enthalpy/unit mass
h_o	Stagnation enthalpy/unit mass
L	Diffuser centerline length (from throat to exit plane)
M	Mach number
m	Mass flow rate
p	Static pressure
pr	Pressure ratio,

$$pr \equiv p_{\text{coll}}/p_{oi}$$

R	Radius ratio, r/r_2
r	Radius from impeller centerline
V	Relative velocity (in coordinate system rotating steadily in Newtonian space)

W	Diffuser throat width (in principal plane of divergence)
W_x	Shaft work per unit mass of fluid
x	Axial coordinate
δ^*	Boundary-layer displacement thickness
η	Efficiency (total to static),

$$\eta = \frac{h_{o_s} - h_{o_i}}{W_x}$$

θ	Tangential angular coordinate
2θ	Diffuser divergence angle
ρ	Density

Subscripts

1, 2, 3, 4	Stations in the stage
CL	Centerline
c	Cover
coll	Collector station (receiving volume after diffuser)
i	Inlet or impeller
o	Stagnation
s	Indicates that process follows an isentropic path
sep	Flow separation value
t	Tip or throat
x	Upstream of shock (e.g., M_x) or axial component
y	Downstream of shock (e.g., M_y)

REFERENCES

1. JOHNSTON, J. P., *The Effects of Rotation on Boundary Layers in Turbomachine Rotors*. Report MD–24, Dept. Mech. Eng., Stanford U., May 1970.
2. DEAN, R. C., JR., D. D. WRIGHT, AND P. W. RUNSTADLER, JR., *Fluid Mechanics Analysis of High-Pressure-Ratio Centrifugal Compressor Data*. Report 69–76, U.S. Army Aviation Materiel Laboratories, February 1970.
3. DEAN, R. C., JR., *The Unresolved Fluid Dynamics of Centrifugal Compressors*. To be published by ASME.
4. WELLIVER, A. D., AND J. ACURIO, *Final Report on AVLABS Contract DAA–177–AMC–173(T)*. U.S. Army Aviation Materiel Laboratories, Reports 67–47 and 67–30, Vols. I and II, published September 1967 and August 1967, respectively.
5. JOHNSTON, J. P., AND R. C. DEAN, JR., *Losses in Vaneless Diffusers of Centrifugal Compressors and Pumps*. ASME Paper 65–FE–1, *J. Basic Eng.*, 1965.
6. DEAN, R. C. JR., AND Y. SENOO, Rotating Wakes in Vaneless Diffuser. *Trans. ASME, J. Basic Eng.*, September 1960.
7. SHAPIRO, A. H., *The Dynamics and Thermodynamics of Compressible Fluid Flow, Vols. I and II*. The Ronald Press Co. (New York), 1953.
8. RUNSTADLER, P. W., JR., *Pressure Recovery Performance of Straight-Channel Single-Plane Divergence Diffusers at High Mach Numbers*. U.S. Army Aviation Materiel Laboratories, Report 69–56, October 1969.

9. RUNSTADLER, P. W., JR., AND R. C. DEAN, JR., Straight Channel Diffuser Performance at High Inlet Mach Numbers. *Trans. ASME, J. Basic Eng.*, Ser. D, Vol. 91, September 1969.

10. ENGLERT, G. W., *Estimation of Compressible Boundary Layer Growth Over Insulated Surfaces With Pressure Gradient*. NACA Tech. Note 4022, June 1951.

11. FAULDERS, C. R., *An Aerodynamic Investigation of Vaned Diffusers for Centrifugal Compressors*. M.I.T., Gas Turbine Lab, January 1954.

12. BRADSHAW, P., *Prediction of Turbulent Shear Layers in Turbomachinery*. Presented at the International Symposium on Fluid Mechanics and Design of Turbomachinery, Penn State U., August–September 1970.

13. NEUMANN, E. P., AND F. LUSTWERK, Supersonic Diffusers for Wind Tunnels. *J. Appl. Mech.*, Vol. 16, No. 2, 1949.

14. RUNSTADLER, P. W., JR., *Straight Diffuser Performance at High Inlet Mach Numbers*. Creare TN–60, September 1966.

15. JOHNSTON, J. P., Experimental Data on Three-Dimensional Flow in a Centrifugal Compressor Diffuser. Gas Turbine Lab, M.I.T., December 1954.

DISCUSSION

H. LINHARDT (Airco Cryogenics): I find your enthusiasm for the high-pressure-ratio compressor very interesting; however, I do believe, as far as industrial application is concerned and also in some aircraft applications, a new material has to be invented before you can draw 15-to-1 pressure ratio because of the high tip speed you're talking about. The other problem with the high-pressure-ratio compressor is the small performance range between the choke and surge, and I do not believe that there is any reasonable application for such a device.

A. D. WELLIVER (Boeing Co.): Dr. Dean made a comment that he would try to design the impellers so they separate, and I would like to clarify that point just a little bit. It's the one point which I have run into over the years that people seem to get more confused over than any other. It has something to do with the fact that many of the people who have designed centrifugals have designed for relatively low pressure ratio and we, on the other hand, were striving for a centrifugal of a fairly high pressure ratio. We designed some centrifugals and had them running at a pressure ratio of 6 and 87 percent total-to-static efficiency with reasonably good range (15 percent). But the flow models that we used seem to run out at this pressure ratio with respect to prediction of static pressure rise and performance that we actually measured in the compressor. It was for this reason that we started looking for a better flow model as we pushed to higher pressure ratio. Now, as far as the separation concept goes, I believe that our experience at both low and high pressure ratio is that if you have a lower pressure ratio design, you can afford to have little or no separation. You must design for the static pressure rise that you might get from one of the potential flow concepts or potential flow programs and you cannot tolerate having large flow separation. As you go to higher pressure ratio, you will find you're allowed to separate earlier in the machine and the mixing loss, which is really a loss of the relative pressure, doesn't have such a dire effect on the machine, and so pressure ratios of say at least 10 to 1 at 80 percent might still be quite feasible and we're getting closer all the time.

Now the other point I wanted to make is quite practical to the compressor designer. What happens when you move across the speed line? It's all well and good to design a compressor that has one point but, as H. Linhardt pointed out, if you have zero surge margin you can't fit that in

inches. Now one of the things that we have learned is that the vaneless space (the area right ahead of the shock system) appears to be what I would call a great adjustor of the flow and, in many cases, the streamlines actually curve far up towards the upper wall and then come down almost in front of the diffuser vane. The amount of curvature varies as you move from the maximum air flow towards the surge line. Actually, as you get closer and closer to the surge line this flow straightens out and finally you go into surge.

G. F. WISLICENUS (Arizona): I like optimism. We all know that in the incompressible field we built centrifugal pumps a long time ago that were 90-percent efficient. To some extent I share the optimism, but I do not understand why we have to live with separation in the impeller. I do not think we had separation in the impeller of the centrifugal pump—not major separation. I cannot understand the author's optimism about the future efficiencies unless he can avoid major separation in the impeller. I do not as yet understand why he feels that the violent unsteady flow which he would get would be helpful and, incidentally, I believe you can avoid the separation in the compressor.

J. L. DUSSOURD (Ingersoll-Rand Research): It seems to me we've missed one important word here in this question of separating and non-separating impellers. That's the simple word "specific speed." If you have an impeller which has a very low specific speed, it simply means that the amount of kinetic energy which is tied in with the V^2 coming into the impeller is smaller compared to the total amount of work which the impeller is putting into the stream. This means that if you don't diffuse this kinetic energy very efficiently, you're not really hurting the performance of the impeller too much. On high-pressure-ratio machines with very high tip Mach number, we are forced to have a comparatively low inducer-to-tip-diameter ratio and therefore a low specific speed. Otherwise your inducer Mach number becomes extremely high. So you can afford, with this kind of a machine, to have a relatively inefficient recovery of the V^2. If you have a high-specific-speed impeller, which is more common perhaps at low pressure ratio, you cannot afford to do that.

DEAN (author): The vigorous discussion of this paper when presented was appreciated; it illuminates the depth and breadth of concern today for the centrifugal compressor.

The following comments are in reply to verbal discussions offered at the meeting.

H. Linhardt commented that strength of materials limitations would prevent the attainment of 15:1 pressure ratio in a single centrifugal stage. I agree that sufficiently strong materials are not available today. However, pressure ratio 12:1 has been achieved at 2200 fps. In the paper, I have called for 15:1 pressure ratio after 10 years or more development.

This prognostication does not seem unrealistic when only 300 fps must be added to tip speed in order to generate 15:1 pressure ratio. With the emergence of new composite materials, fiber reinforcement, control of microstructure from forging powder metallurgy preforms, etc., the needed gain does not seem improbable.

Linhardt also commented that high-pressure-ratio centrifugals were unattractive because of the small range between choke and surge. In reply, I point out that the Boeing experience showed a much broader range (on the order of 10 percent) than had been expected previously. As was mentioned in the paper, we attribute compressor surge to diffuser flow instability. At the moment, there is no understanding of diffuser and stage stability. I have never seen any adequate experimental work on this subject. Appropriate theoretical work is just starting (e.g., Ehrich, ref. D–1). With this lack of attention and knowledge, how can one expect good range or damn the machine with the stigma of poor range?

Our research suggests that the diffuser's throat conditions control stability and surge. We are developing means for controlling surge, but it is too early to report with any certainty. However, the early results encourage the prediction that high-pressure-ratio, high-efficiency centrifugal compressors with excellent range will be developed within 10 years.

Mr. Linhardt also commented on the virtues of the United Aircraft of Canada "pipe" diffuser versus the two-dimensional type used by Boeing and employed for illustration in this paper. We are not championing any particular type of diffuser (other than a well-designed one). However, it is still not plain that the "pipe" diffuser has any distinct advantage over the two-dimensional channel diffuser. No one, including United Aircraft of Canada, has offered hard evidence of superiority. The first order of business today is to get the general diffuser geometry close to an optimum configuration regardless of the details of the diffuser type. Guiding principles for that have been discussed by Dean (ref. D–2) and Dean, Wright and Runstadler (ref. D–3).

By his discussion, A. D. Welliver of the Boeing Company helped to clarify the controversial question of designing a centrifugal compressor with a separated impeller. This matter has been widely misinterpreted by people who do not read carefully what we have said on the matter. So here again we shall repeat. *It is not true, ipso facto, that an optimum centrifugal stage will have an unseparated impeller.*

Under certain conditions, the optimum design proves to be unseparated but, in many other cases, the optimum design proves to have a certain degree of separation at the impeller tip. As we have pointed out repeatedly (e.g., Johnston and Dean, ref. D–4) the degree of separation should not exceed about 40 percent of the passage area because the consequent mixing loss begins to soar with further increases in wake width.

Below 40 percent, the losses due to separation usually are not very serious.

Dr. Wislicenus commented that a separated machine could never be an optimum design. He is stating this as a matter of faith. We do not subscribe to this faith for the centrifugal compressor and, I might point out, some supersonic axial compressor designers do not either. Likewise, if one examines many successful transonic axials, one can only conclude that the blading is heavily separated at the blade row exit. In all these cases, the designers strove for optimum performance.

These situations are much like the simple two-dimensional diffuser. For it, many workers (e.g., Kline, et al., at Stanford) have established that optimum pressure recovery occurs in a stalling diffuser. If one insisted, along with Dr. Wislicenus, that the diffuser should be unseparated, one could not achieve maximum static pressure recovery. On the other hand, if one's objective is to design a diffuser for minimum stagnation pressure loss, then an unseparated design would be optimum.

There are plenty of instances in the design of fluid dynamic systems where separation has been purposely incorporated in order to achieve optimum performance. E. S. Taylor (MIT) distinguished these by saying they are "separated, but not stalled." The centrifugal compressor happens to be a member of this set, in spite of the intuitive repulsion for this set evidenced by most experts.

While it was not a subject of this paper, it is interesting to mention that the paper and motion picture presented by Dr. Johnston at this conference demonstrate in the open literature for the first time a major reason why separation of the centrifugal compressor impeller does not lead to major losses.

When Dr. Johnston and I were together at the Ingersoll-Rand Company circa 1958, we observed through the use of flow visualization with milling yellow aniline dye that the boundary layer on the suction surface of a centrifugal compressor passage was highly stabilized by Coriolis forces. On the other hand, the boundary layer on the driving or pressure side was destabilized. In the last sequence in Johnston's motion picture shown here, bursts of turbulence off the driving face of the passage were seen plainly. We observed the same thing in 1958.

We also observed that the bounding surface between the through-flow jet in the separated impeller and the wake region was unusually quiescent under the influence of the same Coriolis acceleration. Most telling was the occasional observation of a Karman vortex street running from the separation point along the jet-wake interface nearly to the tip of the impeller. If there were turbulent mixing there, as one would expect in stationary coordinates, the pattern of the Karman vortex street would never persist for such a long distance (order 20 times the street width).

Theoretical reasoning about the effect of Coriolis acceleration and these empirical observations have led us to the conclusion that mixing losses due to separation *within* the centrifugal impeller are greatly attenuated by the Coriolis acceleration. Thus one can have separated flow inside the impeller with a relatively small internal loss penalty compared to what would be suffered in the absence of Coriolis acceleration (e.g., axial turbomachine or stationary coordinates). The internal flow in impellers has been discussed at length in Dean (ref. D–2) and Dean, Wright, and Runstadler (ref. D–3).

Dr. J. L. Dussourd questioned our emphasis on impeller diffusion. Of course, we agree with him that there are cases where the relative kinetic energy of the inlet flow is very small compared to the work addition. If this kinetic energy were lost by irreversible mixing, not much influence on stage efficiency would accrue. On the other hand, there are many practical cases where this is not so. We have analyzed a few of them in the paper and have shown in figure 1 the gains which can be made by improving internal diffusion in the impeller. These figures plainly show that internal diffusion is significant for these cases. Perhaps Dr. Dussourd wants to challenge our calculations; but if he accepts their validity, then he must accept the conclusion.

In many cases of high-performance compressors, where overall size is important, the inducer tip relative Mach number is pushed up toward or even beyond 1.0. Centrifugal stages have been designed up to 1.4. Even for a pressure-ratio-10 machine, such as the Boeing compressor which had a relative Mach number of 0.84, internal diffusion was important in order to prevent severe impeller discharge mixing losses. The jet relative velocity leaving the impeller was calculated to be about 700 fps. Losing that much kinetic energy, even at a tip speed of 2000 fps, leads to several points loss in stage efficiency.

Obviously, specific speed is not the important parameter in determining whether internal diffusion is important or not. Rather, it is the ratio of inducer tip relative velocity over impeller tip speed, all squared. This parameter represents the potential loss in kinetic energy compared to the work input of the impeller (when slip factor is close to 1.0). For a design where this parameter is only a few percent, impeller diffusion is not important. However, in a case like the Boeing design (with even a low inducer tip relative Mach number compared to others), the ratio was on the order of 0.20 and impeller internal diffusion was important. See Dean, Wright, and Runstadler (ref. D–3) for extensive discussion of this point and a detailed examination of the Boeing RF–2 data.

Unfortunately, none of the discussors really talked about the main subject of the paper, which was boundary-layer behavior in the diffuser entry region of centrifugal compressors. However, their comments did bring out important background information, which was not presented

due to a lack of time. For those who are deeply interested, we suggest consulting our other recent works mentioned in the references here and those of the paper.

REFERENCES

D-1. EHRICH, F. F., Aerodynamic Stability of Branched Diffusers. Paper 70–GT–27, ASME Gas Turbine Conference, (Brussels), May 24–28, 1970.

D-2. DEAN, R. C., JR., The Unresolved Fluid Dynamics of The Centrifugal Compressor. *Advanced Centrifugal Compressors*, ASME publication, 1971.

D-3. DEAN, R. C., JR., D. D. WRIGHT, AND P. W. RUNSTADLER, JR., *Fluid Mechanics Analysis of High-Pressure-Ratio Centrifugal Compressor Data.* Technical Report 69–76, USAVLABS.

D-4. JOHNSTON, J. P., AND R. C. DEAN, JR., Losses in Vaneless Diffusers of Centrifugal Compressors and Pumps. ASME Paper 65–FE–1, *J. Basic Eng.*, May 1965.