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Vaneless Diffuser Flow with Extremely Distorted Inlet Profile

K. BAMMERT

M. RAUTENBERG

Chief,
Radial Compressor Dept.

W. WITTEKINDT*

Research Assistant

Institute for Turbomachinery
Hannover, Germany

The exit flow distribution of 90-deg centrifugal compressor impellers is distorted in peripheral direction as well as over the diffuser depth. First, there are the rotating jet-and-wake regions behind each impeller channel. Second, the flow has a completely distorted inlet profile from the front to the back wall of the diffuser. Measurements of static and total pressure, total temperature, and flow angle profiles along the whole diffuser length have been carried out in a high specific speed compressor. The results show great angle and total pressure differences in the first part of the diffuser and a backflow zone near the front wall. At the end of the diffuser, a separation zone was located. It is shown that by means of a one-dimensional calculation method, using a dissipation factor, the measured representative mean values of the diffuser flow can be determined very well in the regions where no separation occurs.

*Presently with Klein, Schanzlin & Becker, Saarbrüchen.

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NOMENCLATURE

a = speed of sound
b = diffuser width
c = absolute velocity
c_r = radial component of absolute velocity
c_u = peripheral component of absolute velocity
c_d = dissipation coefficient
c_f = wall friction coefficient
c_p = specific heat
k_c = friction factor
$\dot{m}_{red}$ = reduced mass flow
n = rotating speed (rpm)
p = static pressure
p_K = plenum chamber pressure
p_{tot} = total pressure
r = radius
s = isentropic
t = pitch
tot = total
u = circumferential velocity
y, z = coordinates
D = diffuser
D_2 = outer impeller diameter
$M = c/a = c/\sqrt{\varkappa \cdot RT}$ Mach number
$M_{u2} = u_2/a_K = u_2/\sqrt{\varkappa \cdot RT_K}$ circumferential Mach number
R = gas constant
Re = Reynolds number
T = static temperature
T_K = plenum chamber temperature
T_{tot} = total temperature
α = flow angle between absolute velocity and peripheral component
$\varkappa$ = isentropic exponent
$\lambda_D = r/r_2$ = diffuser radius ratio
ν = kinematical viscosity
ρ = density of the medium
$\tau = T/T_K$ = temperature ratio
τ_w = wall shear stress
π = pressure ratio

INTRODUCTION

The past 20 years have seen considerable progress in the endeavor to increase the pressure ratios and also the specific mass flow rates of the impellers of centrifugal compressors with radially ending blades. The pressure ratio could be increased by improved blade forms and higher circumferential velocities. Raising the specific mass flow rate also necessitates increasing the aspect ratio at the impeller inlet and the outlet width of the impeller. However, an impeller design, according to these requirements, has the effect that the high kinetic flow energy at the impeller outlet enters the diffuser rather irregularly in both peripheral and axial direction. The diffuser has then to convert the kinetic energy into static pressure with a minimum of loss despite these disturbances of the flow profile.

At the Institute for Turbomachinery of the University of Hannover, experimental work has been carried out in the past few years on highly loaded centrifugal compressors with high mass flow rates in order to gain a deeper insight into the process of energy conversion in the diffuser. A considerable part of this work was devoted to the flow immediately downstream of the impeller. The idea is to become acquainted with the mechanism of transition from non-steady flow conditions to a certain degree of steadiness of the flow. The most important results are given in References (1-3).[1] Furthermore, it was proved that it is not the momentum exchange which mainly influences the rise of static pressure and losses — as assumed by Dean and Senoo (4) — but the friction. In the area of non-steady flow, Fig. 1, which extends from a diffuser radius ratio $\lambda_D = 1$ to approximately 1.15, the wall friction accounts for only one part of the total losses. In fact, the major part of the total

[1] Underlined numbers in parentheses designate References at end of paper.

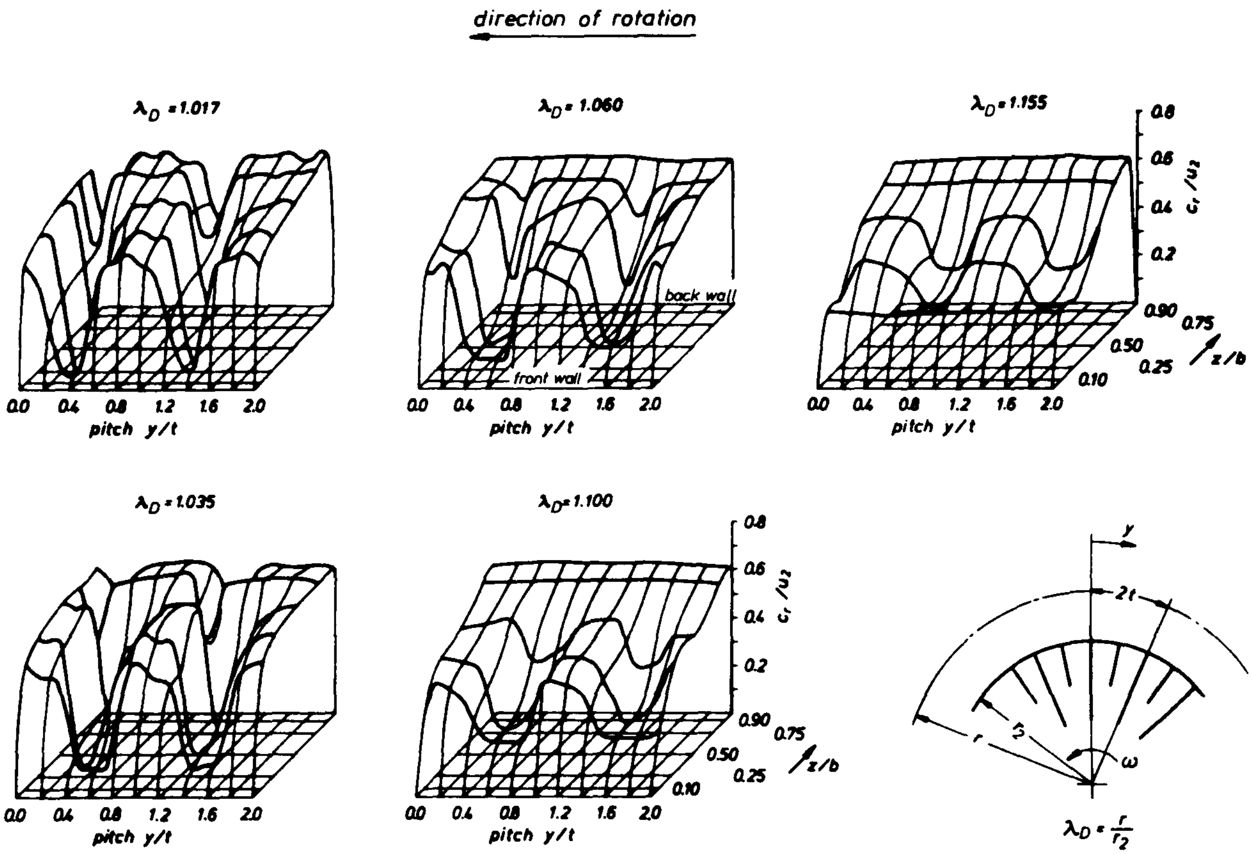

Fig. 1 Stereoscopic formation of the radial velocity
n_{red} = 14,000 rpm $\dot{m}_{red}$ = 6.30 kg/sec

pressure loss is due to the internal friction of the flow. This internal friction is the result of the turbulent apparent friction which occurs as a result of shear stress distribution over the diffuser width.

FLOW DOWNSTREAM OF THE IMPELLER EXIT

On the centrifugal compressor described in Reference (5), measurements were taken applying a technique with a high frequency sensitivity. In Fig. 1, the stereoscopic formation of the radial component of the absolute velocity in the entrance part of the diffuser is plotted against the rotor blade pitch, t, and the diffuser width, b, with y and z as coordinates. It can be clearly seen that at $\lambda_D = 1.06$, there has already been a full equalization of the radial velocity in peripheral direction at least at the back wall. At $\lambda_D = 1.155$, this equalization can be seen also at the front wall; only in the middle of the diffuser are there still differences.

If, by application of the Dean/Senoo method (4), the static and total pressures are calculated and plotted against λ_D, one can see, as shown in Fig. 2, that the static pressure rises excessively and that the total pressure does not decrease in conformity with the measurements. Measurements and calculation can only be brought into conformity with each other, as clearly shown in Fig. 2, when, in a flow calculation with wall friction, a much increased dissipation coefficient as compared with the wall friction coefficient is applied which depends on the radius ratio, λ_D. The dissipation coefficient, c_d, consists of a wall friction component, c_f, and an internal friction component, c', which is added to c_f (2); hence,

$$c_d = c_f + c' \tag{1}$$

with c_f for air determined according to Mager (6) as

$$c_f = 2 \cdot 0.0296 \cdot Re^{-0.2} \left(1 + \frac{\varkappa - 1}{2} M^2\right)^{-0.45} \tag{2}$$

The component, c', is defined in the range $1 \leq \lambda_D \leq 1.16$, with a decay function

$$c' = c'_0 \,(13.5 - 17.88\,\lambda_D + 5.38\,\lambda_D^2) \tag{3}$$

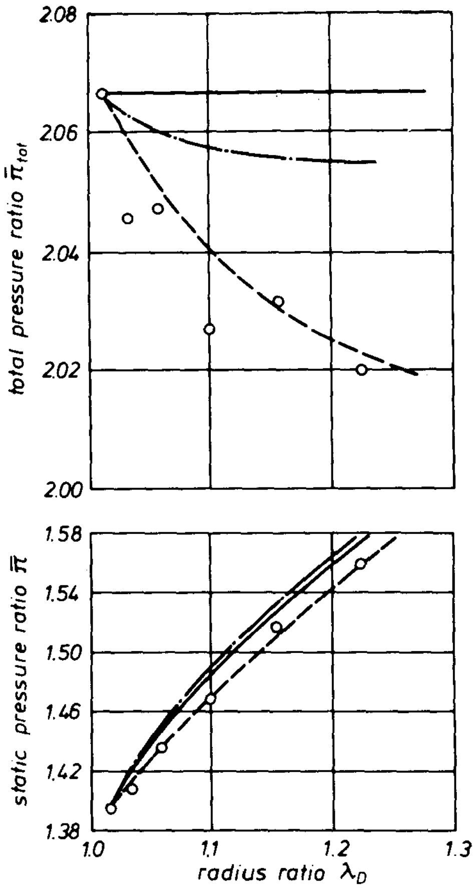

Fig. 2 Static and total pressure ratio within the mixing zone

O experiment; ——— with momentum transfer; ——— incompressibility, frictionless; — — — compressibility, with friction

n_{red} = 14,000 rpm $\dot{m}_{red}$ 6.30 kg/sec

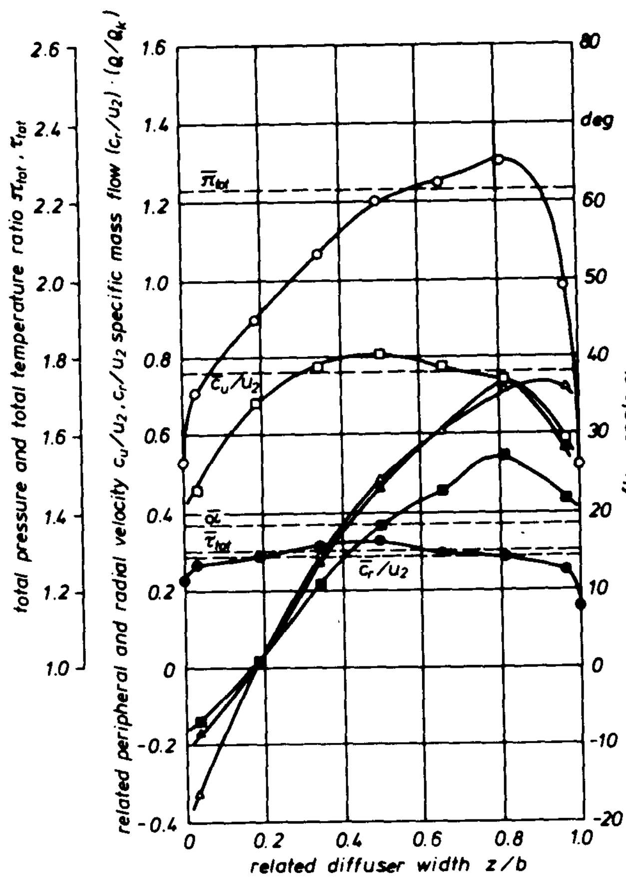

Fig. 3 Patterns of flow quantities within the diffuser

O π_{tot} ⊚ τ_{tot} □ c_u/u_2 ■ c_r/u_2

Δ α ▲ (c_r/u_2) • (ρ/ρ_K) λ_D = 1.15

n_{red} = 20,500 rpm $\dot{m}_{red}$ = 5.134 kg/sec

and c_0' is, according to the tests,

$$c_0' = 0.107 - 0.226\ M_{u2} + 0.129\ M_{u2}^2 \qquad (4)$$

in the range $0.8 \leq M_{u2} \leq 1.4$.

It can be seen from Fig. 1 that also at λ_D = 1.155, there is still a difference between the radial velocities on the front wall and back wall of the diffuser. This gives reason to believe that the steady flow at the diffuser inlet is considerably disturbed.

FLOW IN THE DIFFUSER IN THE AREA DOWNSTREAM OF THE MIXING ZONE

The centrifugal compressor test rig used here is equipped with an impeller of D_2 = 290 mm which has a nominal speed, n = 20,500 rpm. At this speed, the compressor reaches a mass flow in its design point of $\dot{m}_{red}$ = 5.134 kg/sec.

Using miniature probes, the diffuser of equal area with curved front wall and even back wall and an inlet width of 35.6 mm was scanned at seven radii (7). Across the diffuser width, the total pressure, total temperature, and flow angle were measured. These measurements were taken with Kiel probes, window-type temperature probes, and two-hole cobra probes. Furthermore, the static pressures were measured along the whole diffuser front and back wall. For calculation of the velocity in the diffuser from the measured values, it was assumed that the static pressure develops linearly between front and back wall.

As a representative example of the measurements, Fig. 3 shows the patterns of the following flow quantities:

(a) $\pi_{tot} = p_{tot}/p_K$ directly measured

(b) $\tau_{tot} = T_{tot}/T_K$ directly measured

(c) α directly measured

(d) $c_r/u_2 = \sin\alpha/u_2 \sqrt{2 \cdot c_p \cdot T_{tot} \cdot (1-T/T_{tot})}$

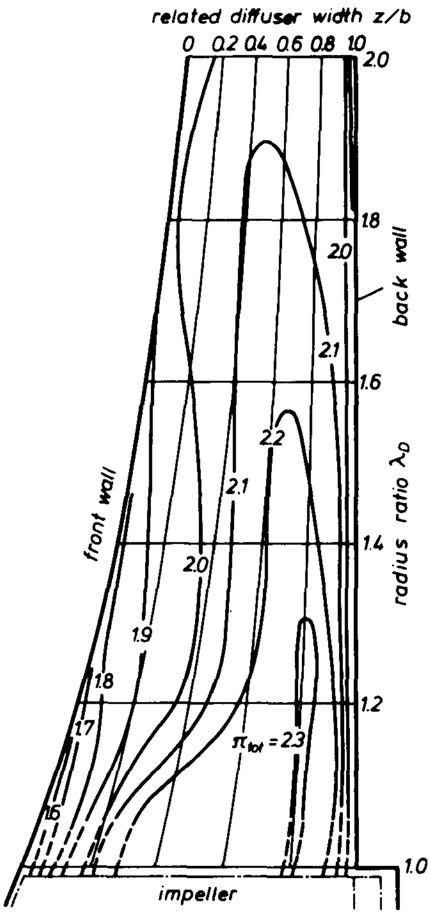

Fig. 4 Isobars of total pressure ratio
n_{red} = 20,500 rpm $\dot{m}_{red}$ = 5.134 kg/sec

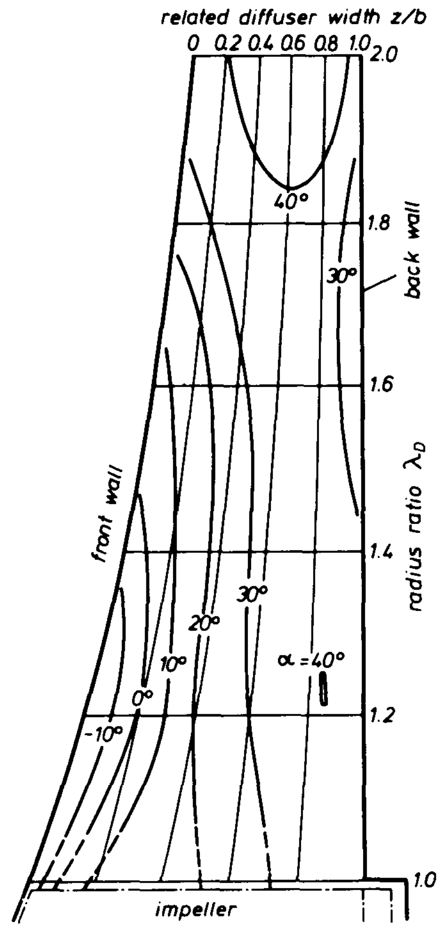

Fig. 5 Flow angle α
n_{red} = 20,500 rpm $\dot{m}_{red}$ = 5.134 kg/sec

calculated from measurements

(e) $c_u/u_2 = c_r/u_2 \cdot \text{ctg } \alpha$
calculated from measurements

(f) $c_r/u_2 \cdot \rho/\rho_K$
calculated from measurements.

It can be seen that the directly measured quantities, π_{tot} and α, increase considerably from the front wall to the back wall. This, of course, leads particularly to a rise in c_r/u_2. The total pressure shows a difference of more than 30 percent between the extreme values. However, the curve of the flow angle is the most striking one. It shows a value of almost -20 deg at the front wall, at a width ratio, z/b = 0.18, the angle is 0 deg, and toward the back wall, it rises to 38 deg. The total difference is almost 60 deg. At the front wall, there is still a dead water zone at λ_D = 1.15, which is caused by the flow already separated in the impeller. This is particularly clear from the specific mass flow $(c_r/u_2) \cdot (\rho/\rho_K)$. The mass flow is concentrated toward the back wall of the diffuser.

Distribution of Flow Quantities in the Diffuser

Fig. 4 shows a section through the vaneless diffuser, with the isobars of the total pressure ratio π_{tot} for λ_D = 1 to 2. The area from λ_D = 1 to 1.06 is marked by broken lines as no measurements were possible there. The isobars clearly show the steep decrease in total pressure in the lower part near the front wall, where also the zone of backflow is situated. At λ_D = 1.6, the total pressure isobar, π_{tot} = 2.1, turns from the back wall to the diffuser middle, which is the result of a much thickened boundary layer.

The isoclines of the flow angle, α, are shown in Fig. 5. The dead water zone at the front wall is limited by the line, α = 0 deg. It extends as far as $\lambda_D > 1.4$ and reaches its maximum thickness at λ_D = 1.2 with 20 percent of the diffuser width. At the back wall, at $\lambda_D > 1.5$, the thickening of the boundary layer can be seen as in the isobar diagram. Above λ_D = 1.8, the boundary layer narrows the free cross section still further.

Fig. 6 shows the lines of constant specific

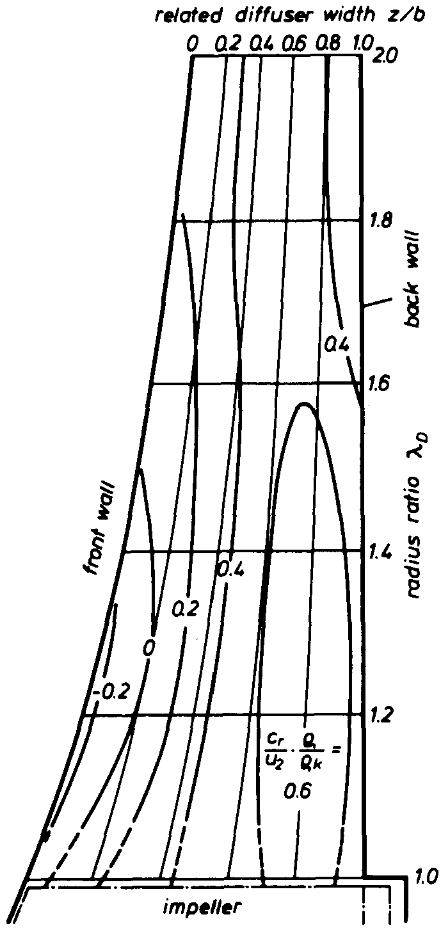

Fig. 6 Specific mass flow, $c_r/u_2 \cdot \rho/\rho_k$
n_{red} = 20,500 rpm $\dot{m}_{red}$ = 5.134 kg/sec

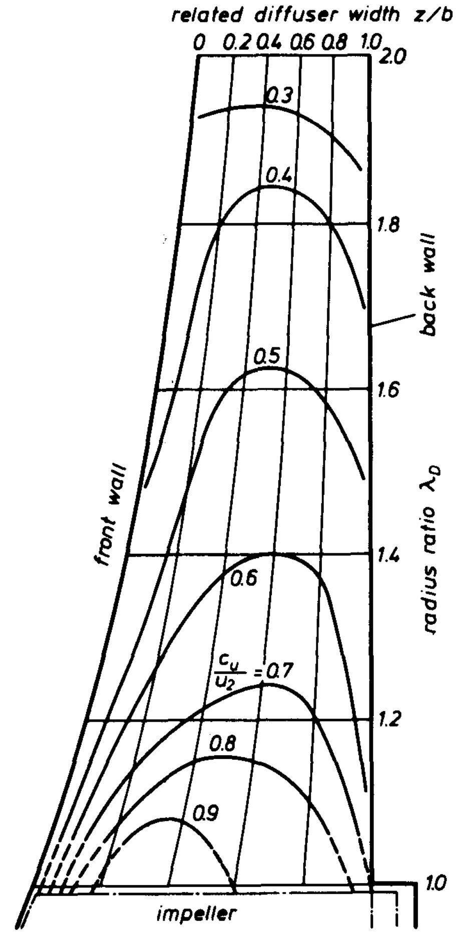

Fig. 7 Related peripheral velocity, c_u/u_2
n_{red} = 20,500 rpm $\dot{m}_{red}$ = 5.134 kg/sec

mass flow. Also, one can clearly see the effects of the thickening boundary layer and dead water zone. The results are in conformity with the measurements of Johnston (8), where it was shown that the medium leaving the impeller first flows to the back wall and then to the front wall, whereupon it moves toward the back wall again. On its way from λ_D = 1 to 2, the flow makes an oscillatory movement between the walls of the diffuser. In the areas of flow adhesion to a wall, a dead water zone or a much thickened boundary layer is formed at the opposite wall.

For assessment of the diffuser as a whole, the distribution of the peripheral component of the absolute velocity is of importance. The angular momentum changes according to the wall friction in the following manner:

$$\frac{d(r \cdot c_u)}{dr} = -c_f \frac{r \cdot c_u}{b \cdot \cos\alpha} \tag{5}$$

For the wall shear stress

$$\tau_w = c_f \frac{\rho}{2} c^2 \tag{6}$$

it is assumed that an infinitesimally small ring element of the flow exerts friction at the walls like a solid body. Despite this simplification, an attempt was made to determine whether with this simple setup it is also possible to calculate a flow in the diffuser which is so unevenly distributed. It can be shown that with equations (1) to (6), the zone up to λ_D = 1.15 already allows a satisfactory calculation.

Fig. 7 shows the line, c_u/u_2, in the entire diffuser. In the dead water zone at the front wall, the angular momentum decreases rapidly. This is due to the increased friction losses in the dead water vortex. In the middle of the diffuser, the peripheral component decreases uniformly. Only at $\lambda_D > 1.8$ is there a more rapid decrease of c_u over the entire cross section, which indicates increased losses.

Mean Values of the Flow in the Diffuser

From the measured flow quantities, repre-

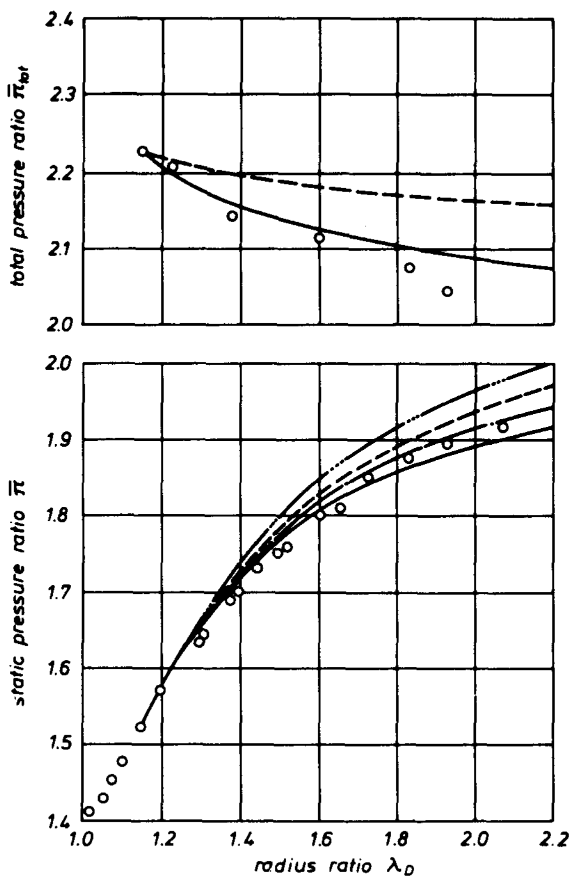

Fig. 8 Comparison of calculated and measured mean total and static pressure within the diffuser ($\lambda_D \geq 1.15$)
——·—— incompressibility, frictionless;
——-—— compressibility, frictionless;
—— —— $k_c = 0.0296$; ———— k_c modified;
O experiment
$n_{red} = 20{,}500$ rpm $\dot{m}_{red} = 5.134$ kg/sec

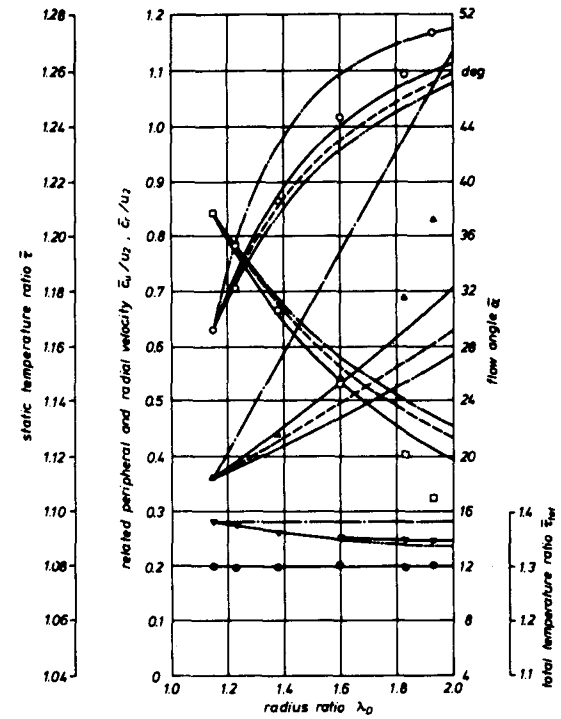

Fig. 9 Comparison of calculated and measured mean flow quantities within the diffuser
O $\bar{\tau}$; ⊗ $\bar{\tau}_{tot}$; □ $\bar{c}_u/u_2$; ∇ $\bar{c}_r/u_2$; ∇ $\bar{\alpha}$;
——·—— incompressibility, frictionless;
——-—— compressibility, frictionless;
—— —— $k_c = 0.0296$; ———— k_c modified
$n_{red} = 20{,}500$ rpm $\dot{m}_{red} = 5.134$ kg/sec

sentative mean values with respect to energy and mass flow are calculated. Fig. 8 shows the mean values of the total and the static pressure ratio and Fig. 9 those of the total and static temperature ratio, of the flow angle, and of the components of the absolute velocity. In addition, these figures show curves for $\lambda_D \geq 1.15$, which are caluclated for the following cases:

(a) Incompressible, frictionless
(b) Compressible, frictionless
(c) With friction according to equation (2)
(d) Modified with $c_d > c_f$.

Both figures show that only the "modified calculation" according to (d) is satisfactory. In contrast to the area up to $\lambda_D = 1.15$, no dissipation coefficient was used here which depends on the diffuser radius and is included in equation (1) as an additive term, c'. Instead for simplification, c' was set to 0 and the constant value $k_c = 0.0296$ in equation (2) was changed.

With $k_c > 0.0296$, one achieves adaption to higher losses and the wall friction coefficient, c_f, changes to the dissipation coefficient. It is surprising that such a simplified method allows a representative calculation of an entirely non-uniform diffuser flow. This method only fails to some extent in the area, $\lambda_D > 1.8$, i.e., where the boundary layer is much thickened. Also, further points in the characteristic field were calculated, and it was shown (7) that without boundary layer separation at the back wall, this calculation is well in conformity with the measurements in the whole diffuser.

These measurements showed that for calculation of the mean values of even a greatly disturbed diffuser flow, a simple wall friction method is sufficient if k_c is increased approximately by the factor 3 to 4. This provides a simple method of calculating the final state of the flow in the diffuser and, hence, the inlet data for the stator blades or a spiral.

SUMMARY

For a vaneless diffuser of a highly loaded centrifugal compressor, it was shown that with a simple wall friction setup, it is possible to calculate the representative mean values of the flow. For the zone immediately downstream of the impeller, it is necessary to choose a relation for the dissipation coefficient which depends on the radius, while for the subsequent zone, the simplified form is sufficient. A prerequisite for this is that the diffuser width is so large that the boundary layers of the front and back wall cannot grow together.

ACKNOWLEDGMENTS

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