Design of Centrifugal Compressor-1

Session delivered by:

Prof. Q. H. Nagpurwala

Session Objectives

To introduce the delegates to

- the procedure for aerodynamic design of centrifugal compressors
- the methods for obtaining impeller geometry
- the procedure for the design of vaned and vaneless diffusers
- Effect of geometric parameters on centrifugal compressor performance

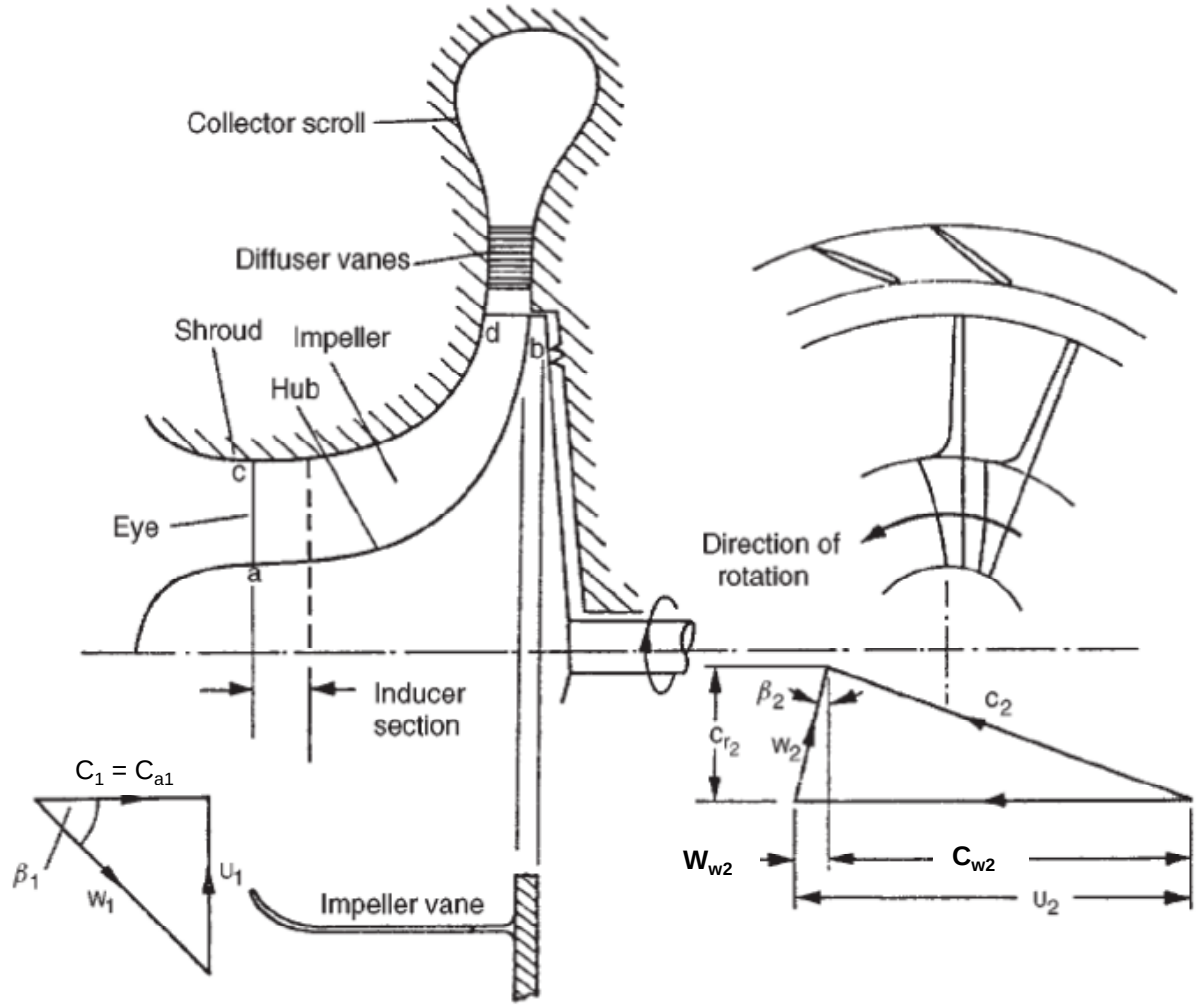
Nomenclature

C	Absolute velocity
n	Number of vanes
p	pressure
N	Rotational speed
r	Radius
T	Temperature
U	Impeller speed at tip
U_e	Impeller speed at mean radius of eye
V, W	Relative velocity
β	Relative flow angle
σ	Slip factor
ψ	Power input factor
ω	Angular velocity

Suffixes

a, x	Axial component
a	Ambient
r	Radial component
w, θ	Whirl component

Centrifugal Compressor



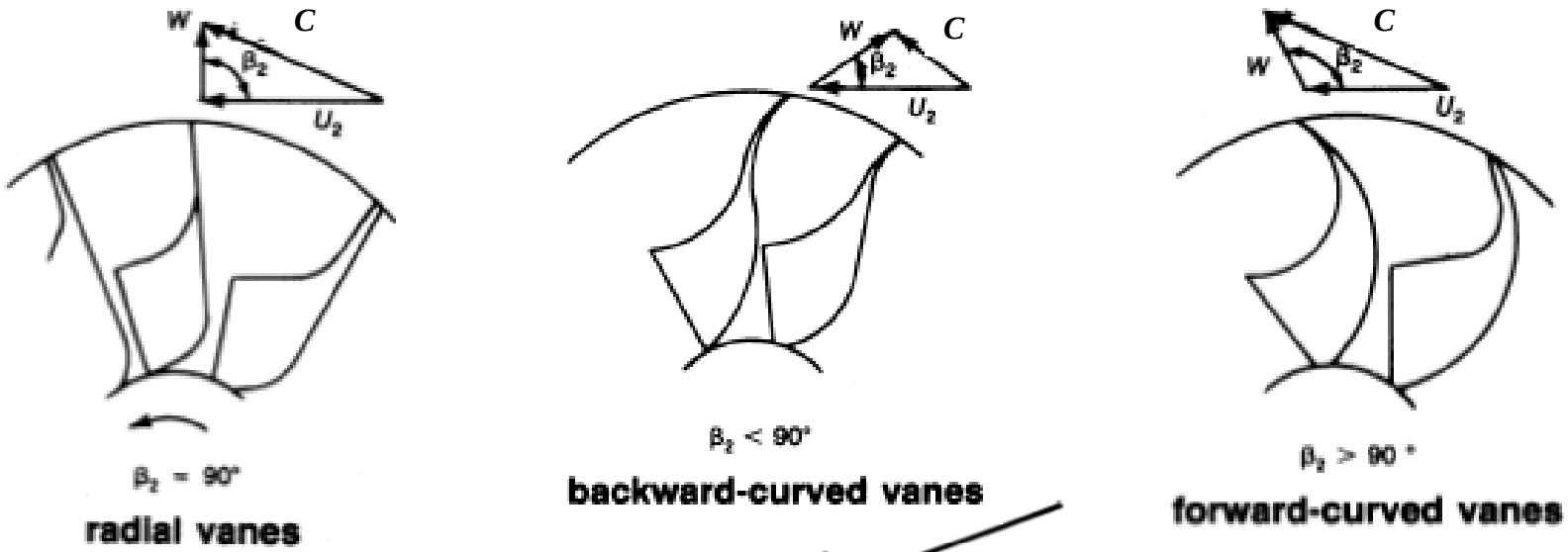
Centrifugal compressor stage and velocity diagrams at impeller entry and exit

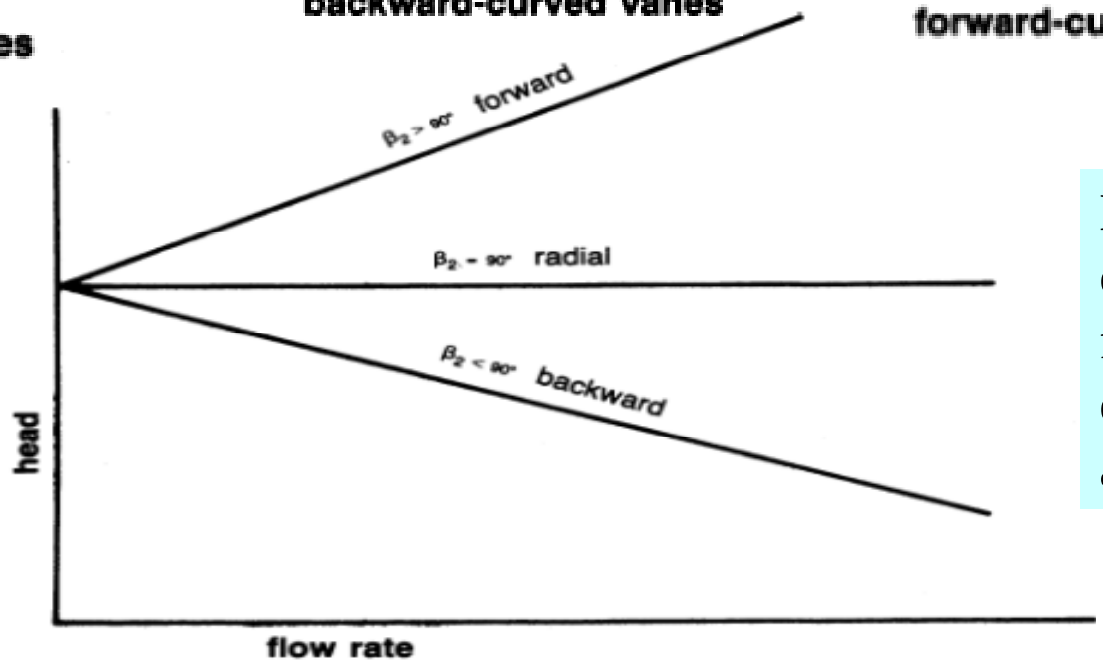
Centrifugal Impeller with Vaned Diffuser

Without Splitter Blades

With Splitter Blades

Types of Impellers

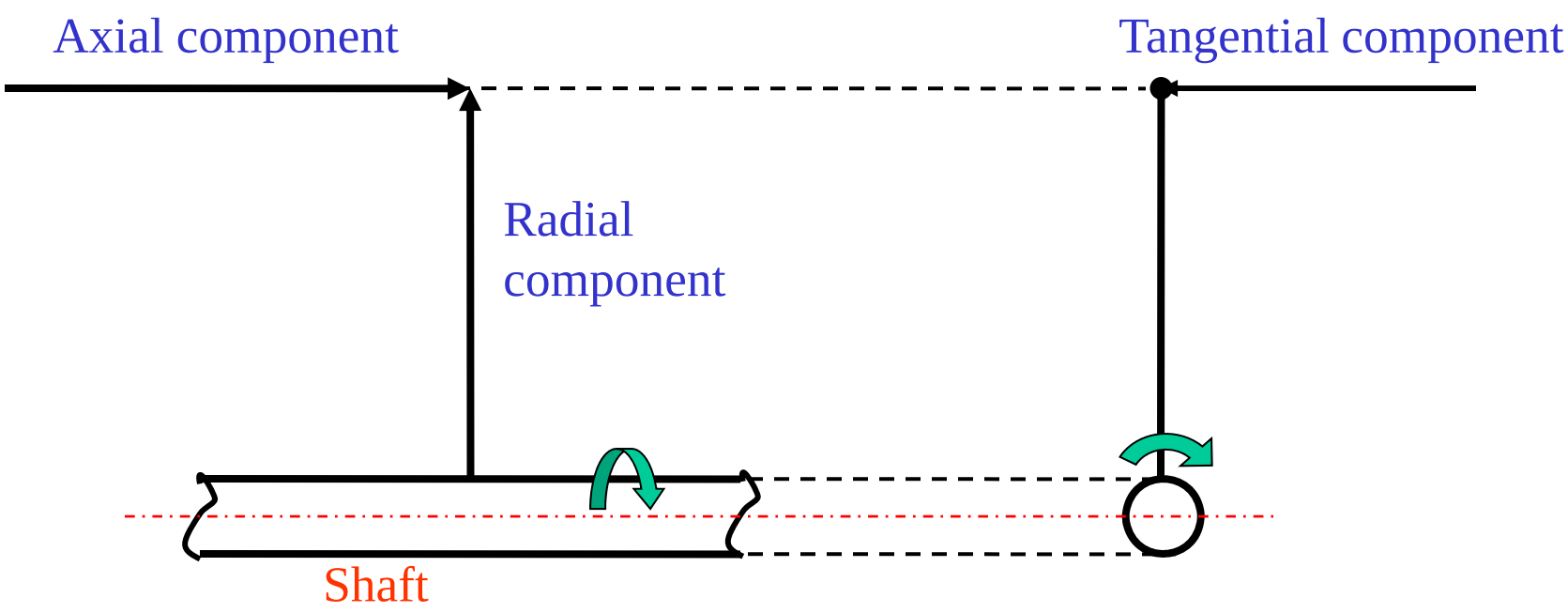




Head – flow characteristics for various outlet blade angles

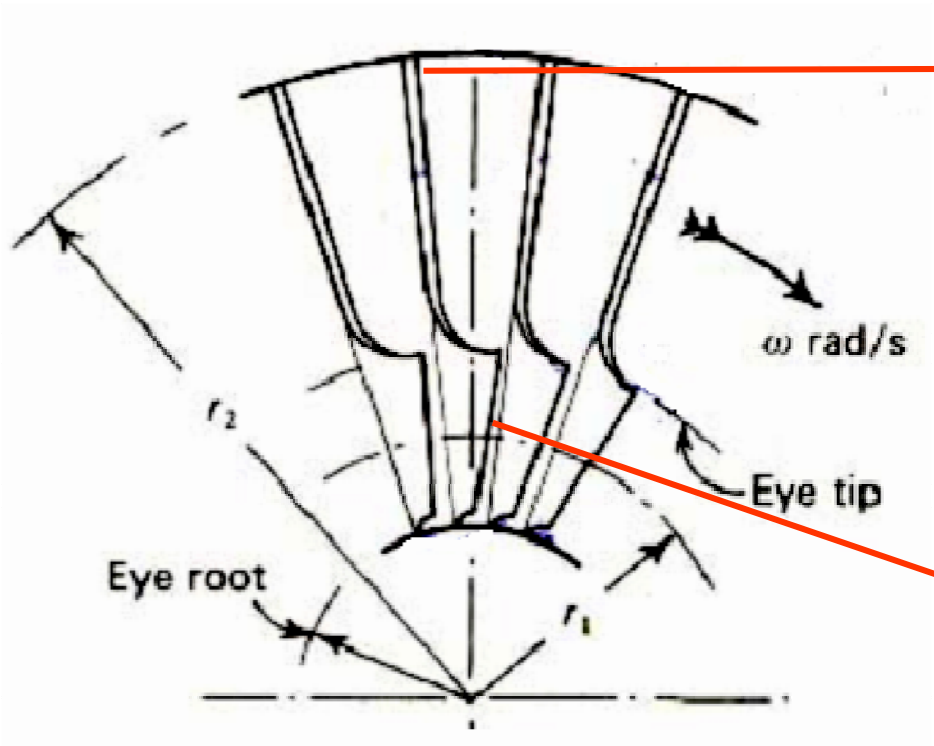
Components of Fluid Forces on Blades / Vanes

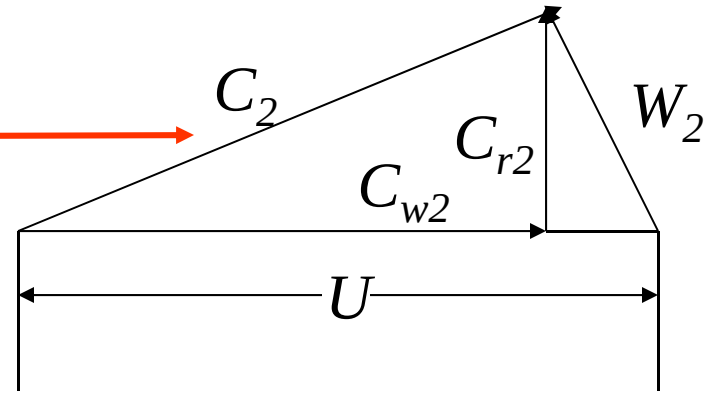
Fluid particles flowing through a rotor experience forces in **axial, radial** and **tangential** directions.



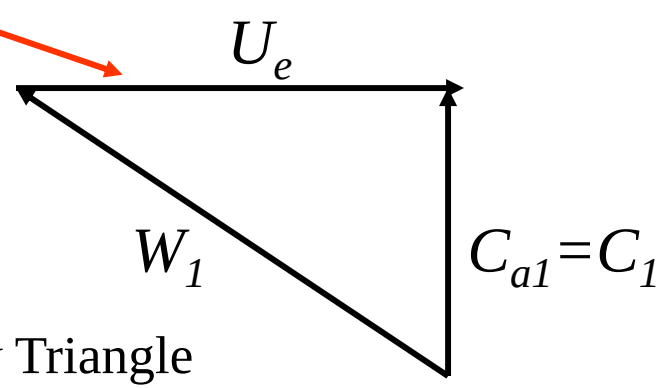
Velocity Triangles

Approximate blade shapes can be obtained from velocity triangles.





Outlet velocity triangle



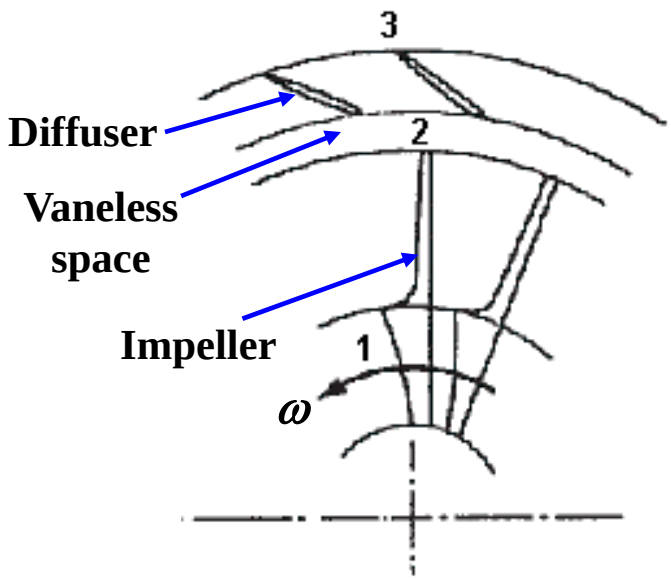
Inlet Velocity Triangle

Design Formulae

From Euler turbine equation of energy exchange,

Specific work, $\dot{W} = \left(U_2 C_{w2} - U_e C_{w1}\right) = h_{03} - h_{01}$

$$= c_p \left(T_{03} - T_{01}\right)$$



If the flow at inlet to the impeller is axial, then $C_{w1} = 0$, and

$$\dot{W} = U_2 C_{w2} = c_p \left(T_{03} - T_{01}\right)$$

Design Formulae

Slip Factor $\sigma = \dfrac{C_{w2}}{C'_{w2}}$, $\sigma = \dfrac{C_{w2}}{U}$ for radial impeller

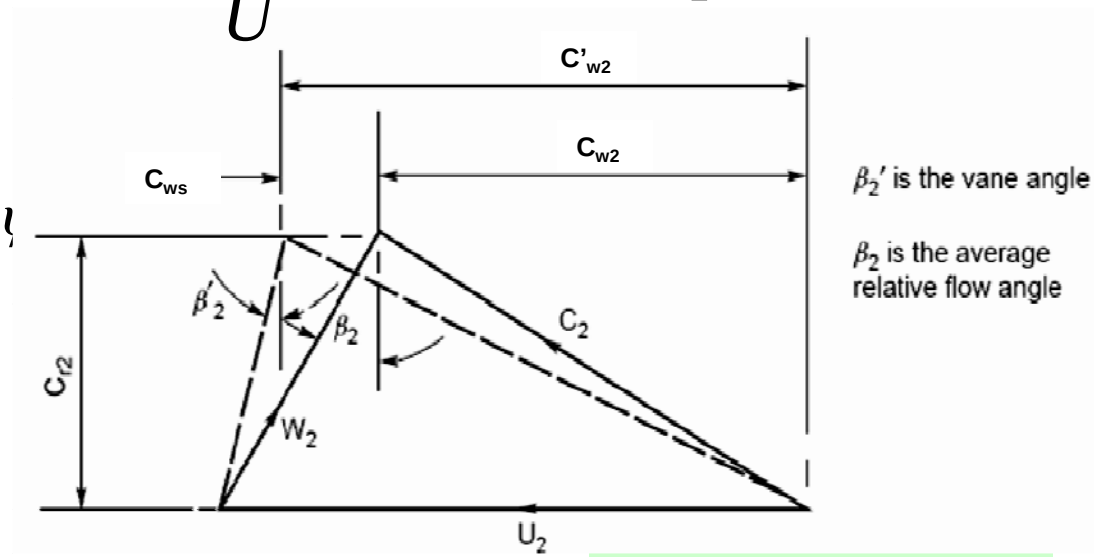
Then, $\dot{W} = \sigma U^2$

Introducing power input factor, ψ

$$\dot{W} = \psi \sigma U^2$$

and temperature rise

$$T_{03} - T_{01} = \frac{\psi \sigma U^2}{c_p}$$



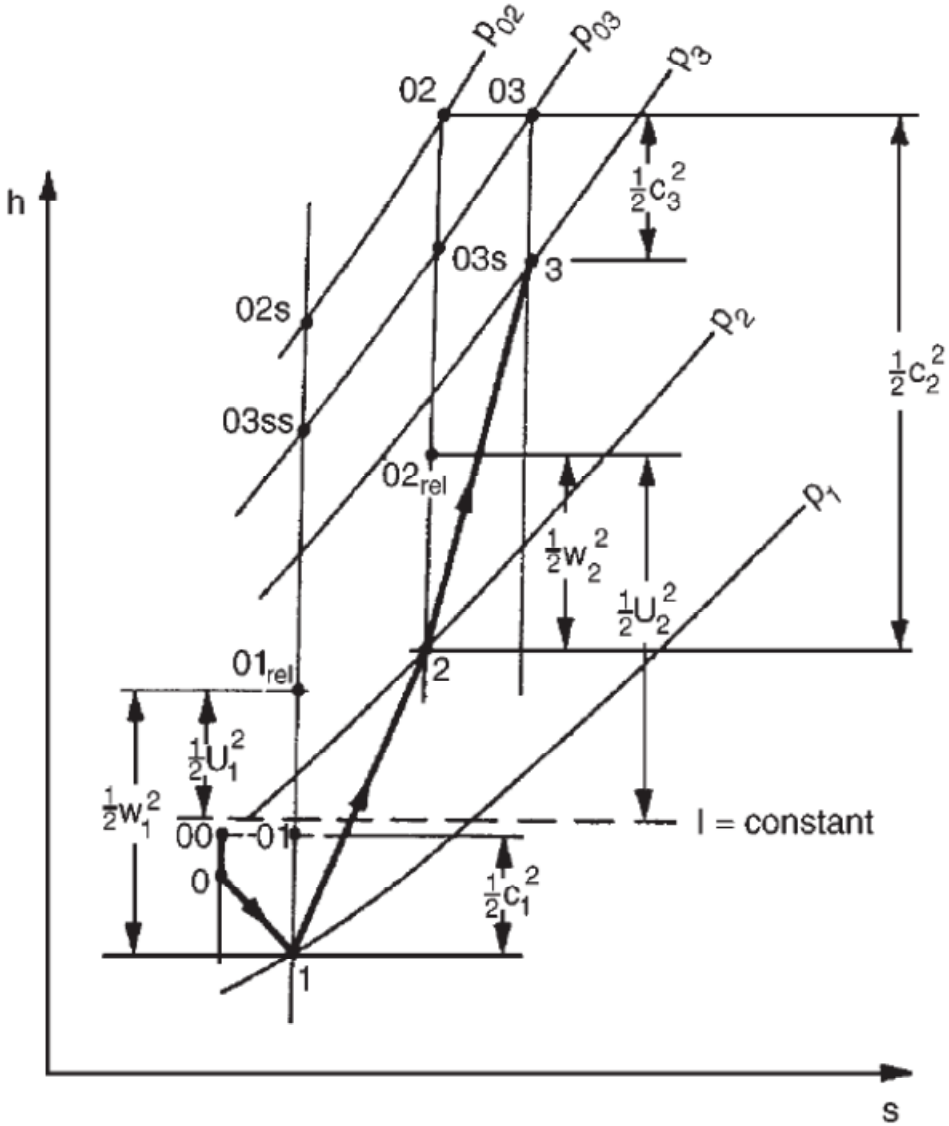
Velocity triangle at impeller exit

$$C_\theta \equiv C_w$$

$$\frac{p_{03}}{p_{01}} = \left(\frac{T'_{03}}{T_{01}}\right)^{\gamma/(\gamma-1)} = \left[1 + \frac{\eta_c (T_{03} - T_{01})}{T_{01}}\right]^{\gamma/(\gamma-1)}$$

$$= \left[1 + \frac{\eta_c \psi \sigma U^2}{c_p T_{01}}\right]^{\gamma/(\gamma-1)}$$

Compression Process on h-s Diagram



The fluid is accelerated from velocity C_0 to velocity C_1 and the static pressure falls from p_0 to p_1.

Since the stagnation enthalpy is constant in steady, adiabatic flow without shaft work then $h_{00} = h_{01}$

or $$h_0 + \frac{1}{2}C_0^2 = h_1 + \frac{1}{2}C_1^2$$

Design Procedure

- Assume rotational speed, tip speed and air entry velocity
- Determine the pressure ratio of the compressor and the power required to drive it, assuming that the velocity of the air at inlet is axial,
- Calculate the inlet angle of the impeller vanes at the root and tip radii of the eye assuming that the axial inlet velocity is constant across the eye
- Estimate the number of vanes
- Estimate the axial depth of the impeller channels at the periphery of the impeller
- Estimate the inlet angle of the diffuser vanes, and
- the throat width of the diffuser passages, which are assumed to be of constant depth

Design Specifications

The following data are suggested as a basis for the design of a single-sided centrifugal compressor:

Power input factor, ψ	1.04
Slip factor, σ	0.9
Rotational speed, N	290 rev/s
Overall diameter of impeller	0.5 m
Eye tip diameter	0.3 m
Eye root diameter	0.15 m
Air mass flow, m	9 kg/s
Inlet stagnation temperature, T_{01}	295 K
Inlet stagnation pressure, p_{01}	1.1 bar
Isentropic efficiency, η_c	0.78

Design Requirements

Requirements are

(a) To determine the pressure ratio of the compressor and the power required to drive it assuming that the velocity of the air at inlet is axial

(b) To calculate the inlet angle of the impeller vanes at the root and tip radii of the eye, assuming that the axial inlet velocity is constant across the eye

(c) To estimate the axial depth of the impeller channels at the periphery of the impeller

Pressure Ratio and Power

(a) Impeller tip speed $U = \pi * 0.5 * 290 = 455.5$ m/s

Temperature equivalent of the work done on unit mass flow of air is

$$T_{03} - T_{01} = \frac{\psi \sigma U^2}{c_p} = \frac{1.04 * 0.9 * 455.5^2}{1.005 * 10^3} = 193K$$

$$\frac{p_{03}}{p_{01}} = \left[1 + \frac{\eta_c (T_{03} - T_{01})}{T_{01}}\right]^{\gamma/\gamma-1} = \left(1 + \frac{0.78 * 193}{295}\right)^{3.5} = 4.23$$

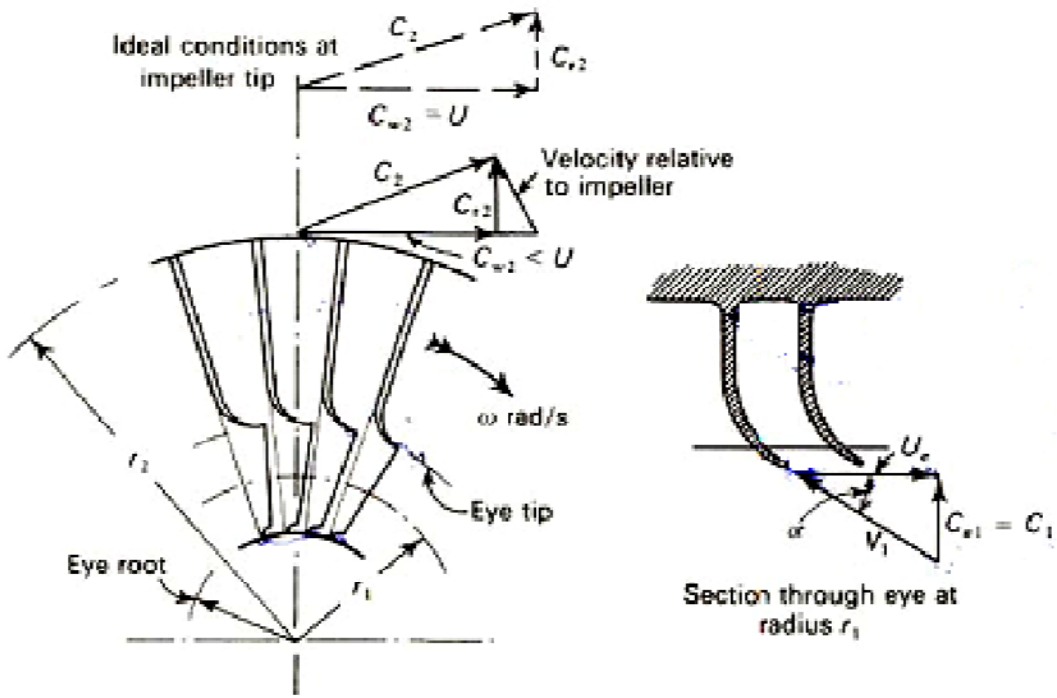
Power required = $m\, c_p (T_{03} - T_{01}) = 9 * 1.005 * 193 = 1746$ kW

Static Density at Inlet

(b) To find the inlet angle of the vanes it is necessary to determine the inlet velocity, which in this case is axial, i.e. $C_{a1} = C_1$.

C_{a1} must satisfy the continuity equation $m = \rho_1 A_1 C_{a1}$, where A_1 is the flow area at inlet.

Since the density ρ_1 depends upon C_1 and both are unknown, a trial and error process is required.



Design Procedure

- The iterative procedure is not critically dependent on the initial value assumed for the axial velocity, but clearly it is desirable to have some rational basis for obtaining an estimated value for starting the iteration.
- The simplest way of obtaining a reasonable estimate of the axial velocity is to calculate the density on the basis of the known stagnation temperature and pressure; in practice this will give a density that is too high and a velocity that is too low.
- Having obtained an initial estimate of the axial velocity. The density can be recalculated and hence the actual velocity from the continuity equation; if the assumed and calculated velocities do not agree it is necessary to iterate until agreement is reached.
- Note that it is normal to design for an axial velocity of about 150 m/s, thus providing a suitable compromise between high flow per unit frontal area and low frictional losses in the intake.

Iteration on C_{a1}

Annulus area of impeller eye, $A_1 = \dfrac{\pi(0.3^2 - 0.15^2)}{4} = 0.053 m^2$

Based on stagnation conditions

$$\rho_1 \cong \frac{p_{01}}{RT_{01}} = \frac{1.1*100}{0.287*295} = 1.30\, kg/m^3$$

$$C_{a1} = \frac{m}{\rho_1 A_1} = \frac{9}{1.30*0.053} = 131\, m/s$$

Since $C_1 = C_{a1}$, the equivalent dynamic temperature is

$$\frac{C^2{}_1}{2c_p} = \frac{131^2}{2*1.005*10^3} = \frac{1.31^2}{0.201} = 8.5K$$

$$T_1 = T_{01} - \frac{C^2{}_1}{2c_p} = 295 - 8.5 = 286.5K$$

$$p_1 = \frac{p_{01}}{(T_{01}/T_1)^{\gamma/(\gamma-1)}} = \frac{1.1}{(295/286.5)^{3.5}} = 0.992\, bar$$

$$\rho_1 = \frac{p_1}{RT_1} = \frac{0.992*100}{0.287*286.5} = 1.21 kg/m^3$$

Check on C_{a1}

$$C_{a1} = \frac{m}{\rho_1 A_1} = \frac{9}{1.21*0.053} = 140\, m/s$$

Iteration on C_{a1} (… contd.)

Final trial:

Try $C_{a1} = C_1 = 145$ m/s

Equivalent dynamic temperature is

$$\frac{C_1^2}{2c_p} = \frac{145^2}{2*1.005*10^3} = \frac{1.45^2}{0.201} = 10.5 \text{ K}$$

$$T_1 = T_{01} - \frac{C_1^2}{2c_p} = 295 - 10.5 = 284.5 \text{ K}$$

$$p_1 = \frac{p_{01}}{(T_{01}/T_1)^{\gamma/(\gamma-1)}} = \frac{1.1}{(295/284.5)^{3.5}} = 0.968 \text{ bar}$$

$$\rho_1 = \frac{m}{RT_1} = \frac{0.968*100}{0.287*284.5} = 1.185 \text{ kg/m}^3$$

Check on C_{a1}: $C_{a1} = \dfrac{m}{\rho_1 A_1} = \dfrac{9}{1.185*0.053} = 143$ m/s

Flow Angles at Inducer

- This is a good agreement and a further trial using C_{a1}= 143 m/s is unnecessary because a small change in C_{a1} has little effect upon ρ. For this reason it is more accurate to use the final value 143 m/s, rather than the mean of 145 m/s (the trial value) and 143 m/s. The vane angles can now be calculated as follows:

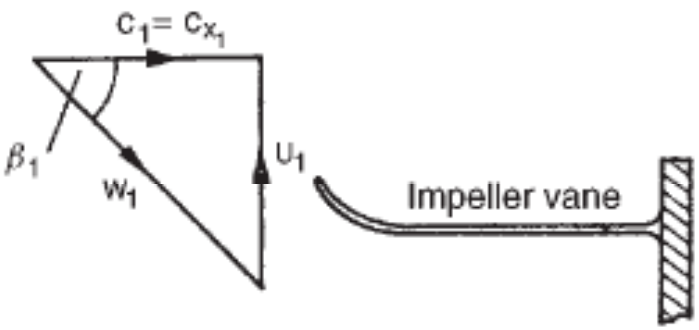
Peripheral speed at the impeller eye tip radius

$$= \pi * 0.3 * 290 = 273 \text{ m/s}$$

and at eye root radius = 136.5 m/s

$$\beta \text{ at root} = \tan^{-1}\left(143/136.5\right) = 46.33^\circ$$

$$\beta \text{ at tip} = \tan^{-1}\left(143/273\right) = 27.65^\circ$$



Radial Velocity and Static Density at Exit

(c) To calculate the required depth of the impeller channel at the periphery we must make some assumptions regarding both the radial component of velocity at the tip and the division of losses between the impeller and the diffuser so that the density can be evaluated. The radial component of velocity will be relatively small and can be chosen by the designer; a suitable value is obtained by making it approximately equal to the axial velocity at inlet to the eye.

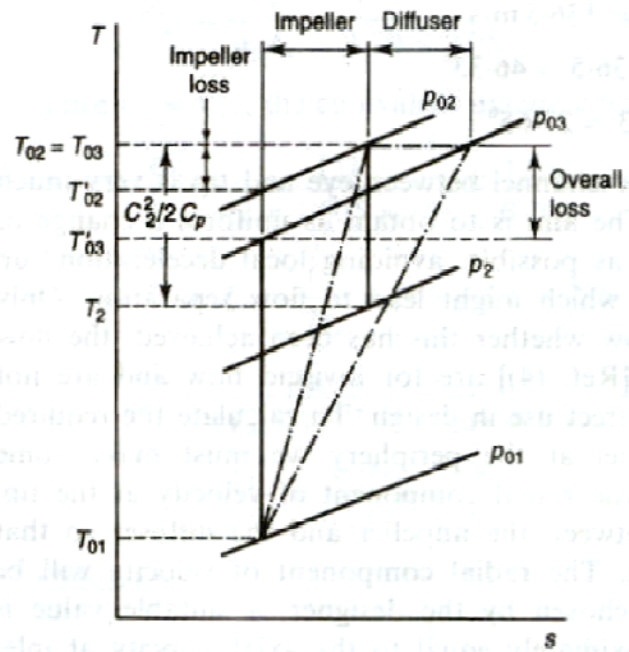
- To estimate the density at the impeller tip, the static pressure and temperature are found by calculating the absolute velocity at this point and using it in conjunction with the stagnation pressure which is calculated from the assumed loss up to this point.

Absolute Velocity at Exit

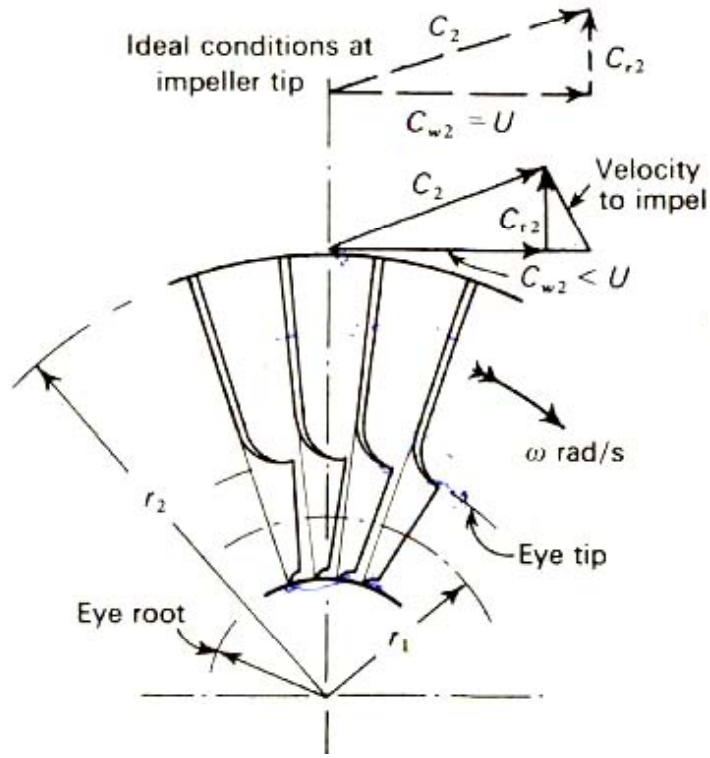
Making the choice $C_{r2} = C_{a1}$, we have $C_{r2} = 143$ m/s

$C_{w2} = \sigma\, U = 0.9*455.5 = 410$ m/s

$$\frac{C_2^2}{2c_p} = \frac{C_{r2}^2 + C_{w2}^2}{2c_p} = \frac{1.43^2 + 4.10^2}{0.201} = 93.8\,\mathrm{K}$$



Division of loss between impeller and diffuser



Distribution of Total Loss

Assuming that half the total loss, i.e. 0.5(1- η_c) = 0.11, occurs in the impeller, the effective efficiency of compression from p_{01} to p_{02} will be 0.89 so that,

$$\frac{p_{02}}{p_{01}} = \left(1 + \frac{0.89*193}{295}\right)^{3.5} = 1.582^{3.5}$$

Now, $(p_2/p_{02}) = (T_2/T_{02})^{3.5}$

and $T_{02} = T_{03} = 193+295 = 488$ K

so that $$T_2 = T_{02} - \frac{C_2^2}{2c_p} = 488 - 93.8 = 394.2 \text{ K}$$

$$\frac{p_2}{p_{02}} = \left(\frac{394.2}{488}\right)^{3.5}$$

Impeller Width at Exit

Since $(p_2/p_{01}) = (p_2/p_{02})\,(p_{02}/p_{01})$

$$\frac{p_2}{p_{01}} = \left(1.582 * \frac{394.2}{488}\right)^{3.5} = 2.35$$

$$p_2 = 2.35 * 1.1 = 2.58 bar$$

$$\rho_2 = \frac{p_2}{RT_2} = \frac{2.58 * 100}{0.287 * 394.2} = 2.28\ \text{kg/m}^3$$

The required flow area normal to the radial direction at the impeller tip is

$$A = \frac{m}{\rho_2 C_{r2}} = \frac{9}{2.28 * 143} = 0.0276\ \text{m}^2$$

Hence the depth of impeller channel at exit

$$\boxed{b = \frac{0.0276}{\pi * 0.5} = 0.0176\ \text{m or } 1.76\ \text{cm}}$$

(This result will be used when discussing the design of the diffuser in the next section)

Estimation of Number of Blades

The number of impeller vanes can be obtained from the correlations for slip factor

Stanitz correlation

$$\sigma_s = 1 - \frac{0.63\pi/Z}{1 - \phi_2 \tan \beta'_2}$$

Wiesner correlation

$$\sigma_s = 1 - \frac{\sqrt{\cos \beta_2}}{Z^{0.7}}$$

Stodola correlation

$$\sigma = 1 - \frac{(\pi/Z)\cos \beta'_2}{1 - \phi_2 \tan \beta'_2}$$

where $\phi_2 = c_{r2}/U_2$ and β'_2 is measured from radial direction.

In the present case, $\beta'_2 = 0$

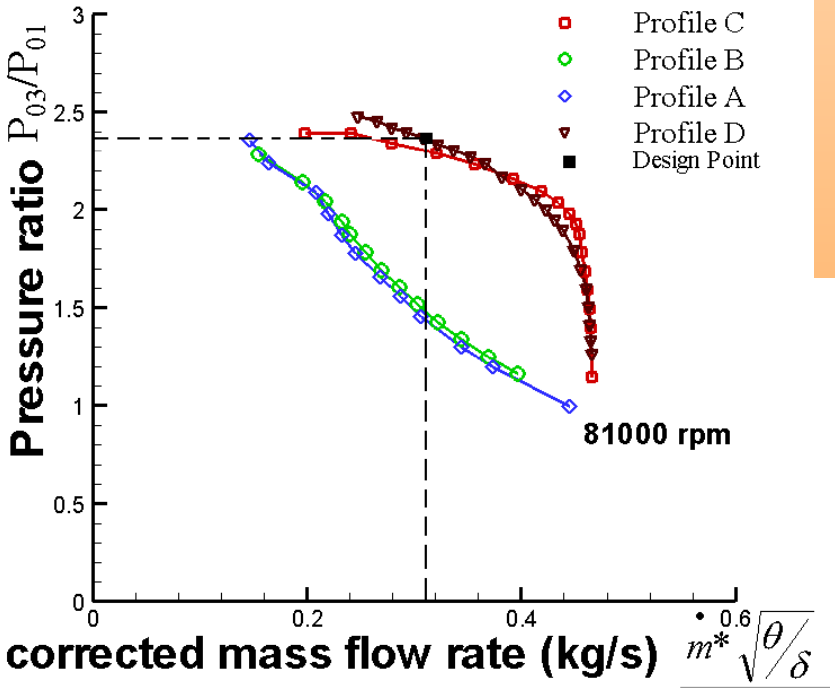
Hence, Stanitz correlation reduces to $\sigma_s = 1 - \frac{0.63\pi}{Z}$

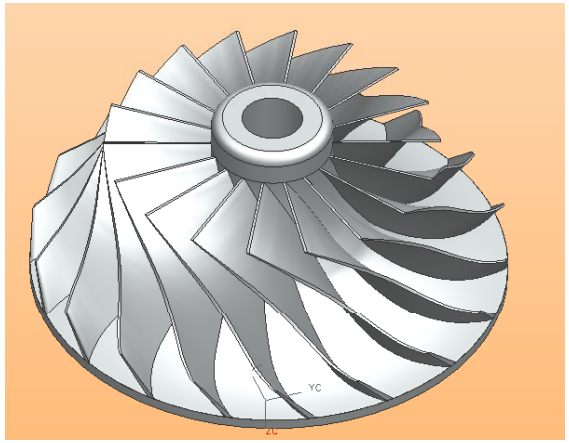
For $\sigma_s = 0.9$, number of blades, $\boldsymbol{Z} = 19.79 \simeq \mathbf{20}$

Generation of Blade Profile

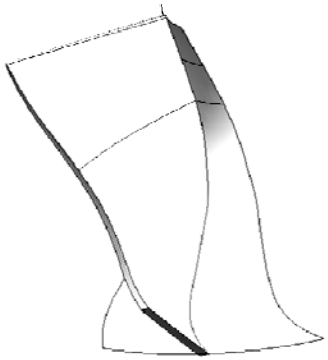
Impeller vane profiles can be generated

- by using softwares, like BLADEGEN
- by trial and error process, using CAD and CFD tools

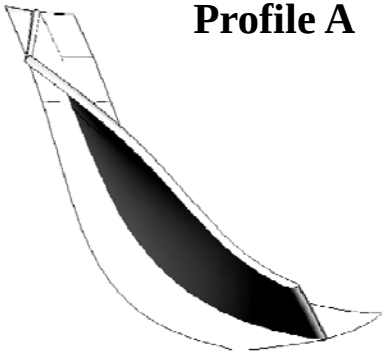




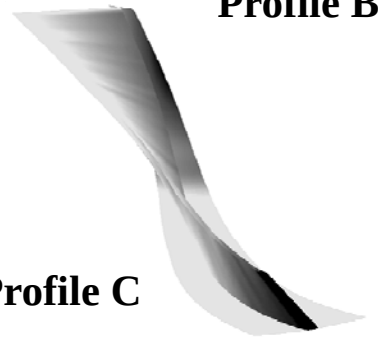
Final Profile D



Profile A



Profile B

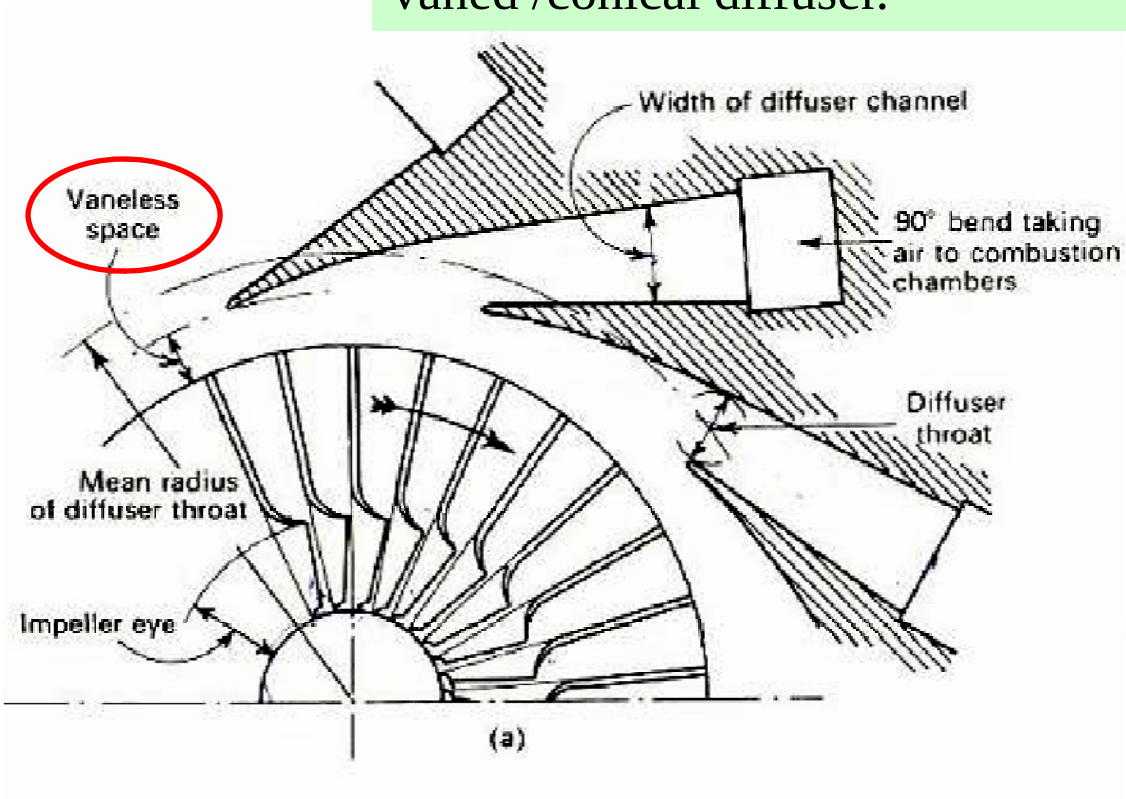


Profile C

Types of Diffusers

- Vaneless diffuser
- Conical diffuser

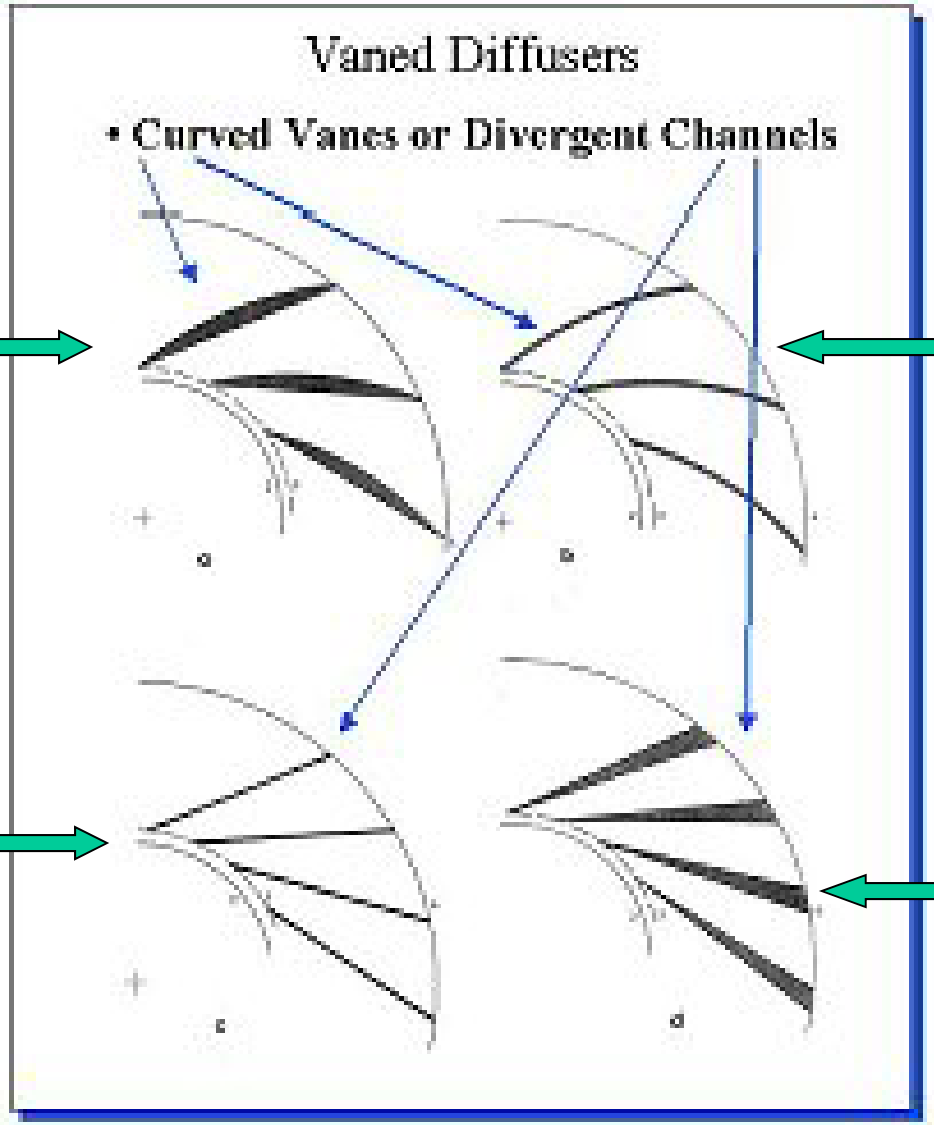
Angular momentum is conserved in the vaneless space; i.e. $r.C_w$ = constant. C_r also varies in the vaneless space because of the change in radius. Hence, estimation of C_w and C_r will yield the absolute velocity and flow angle at inlet to the vaned /conical diffuser.



Types of Diffusers

Aerofoil shape vanes

Uniform thickness curved vanes

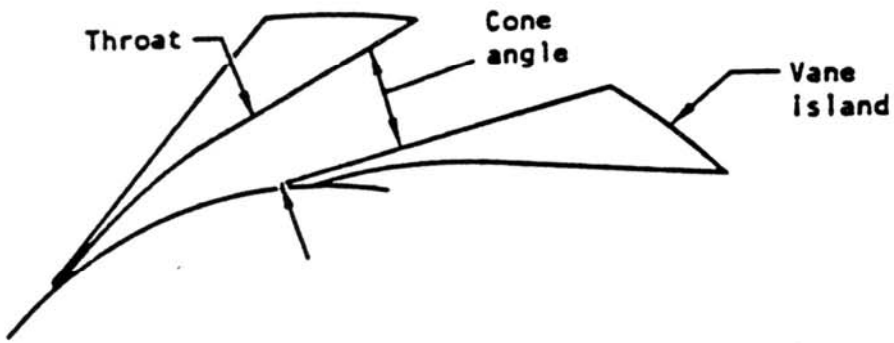


Uniform thickness straight vanes

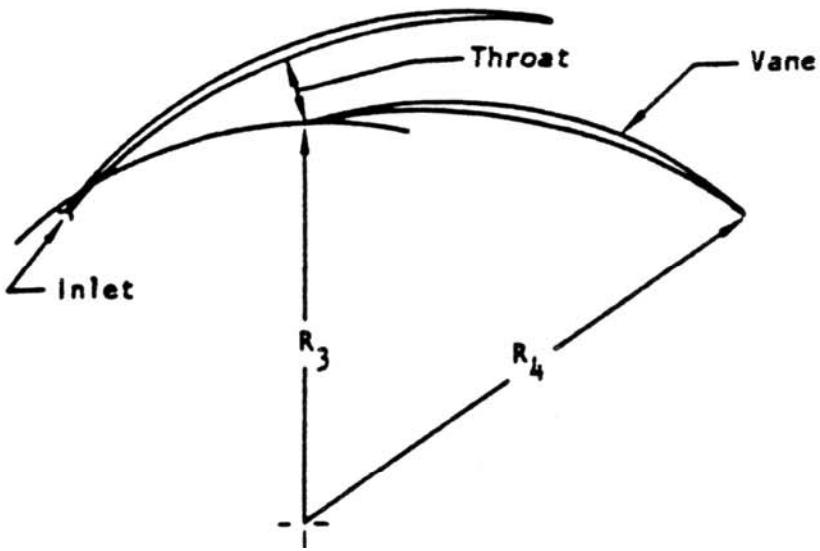
Wedge shape vanes

Types of Diffusers

- Straight wedge type
- Aerofoil shaped vanes



(a) Straight wedge type



(b) Aerofoil shaped vanes

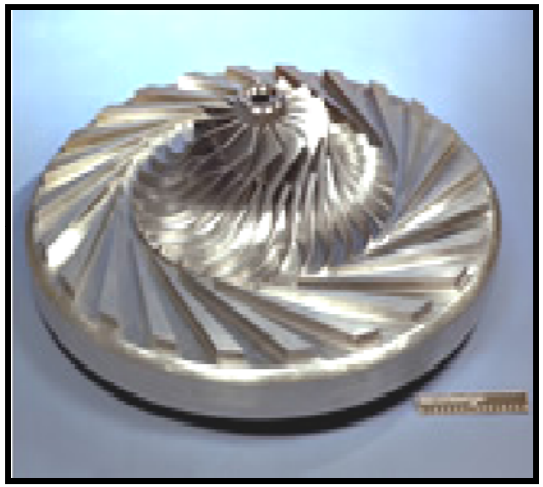
Types of Impellers and Diffusers

Impellers

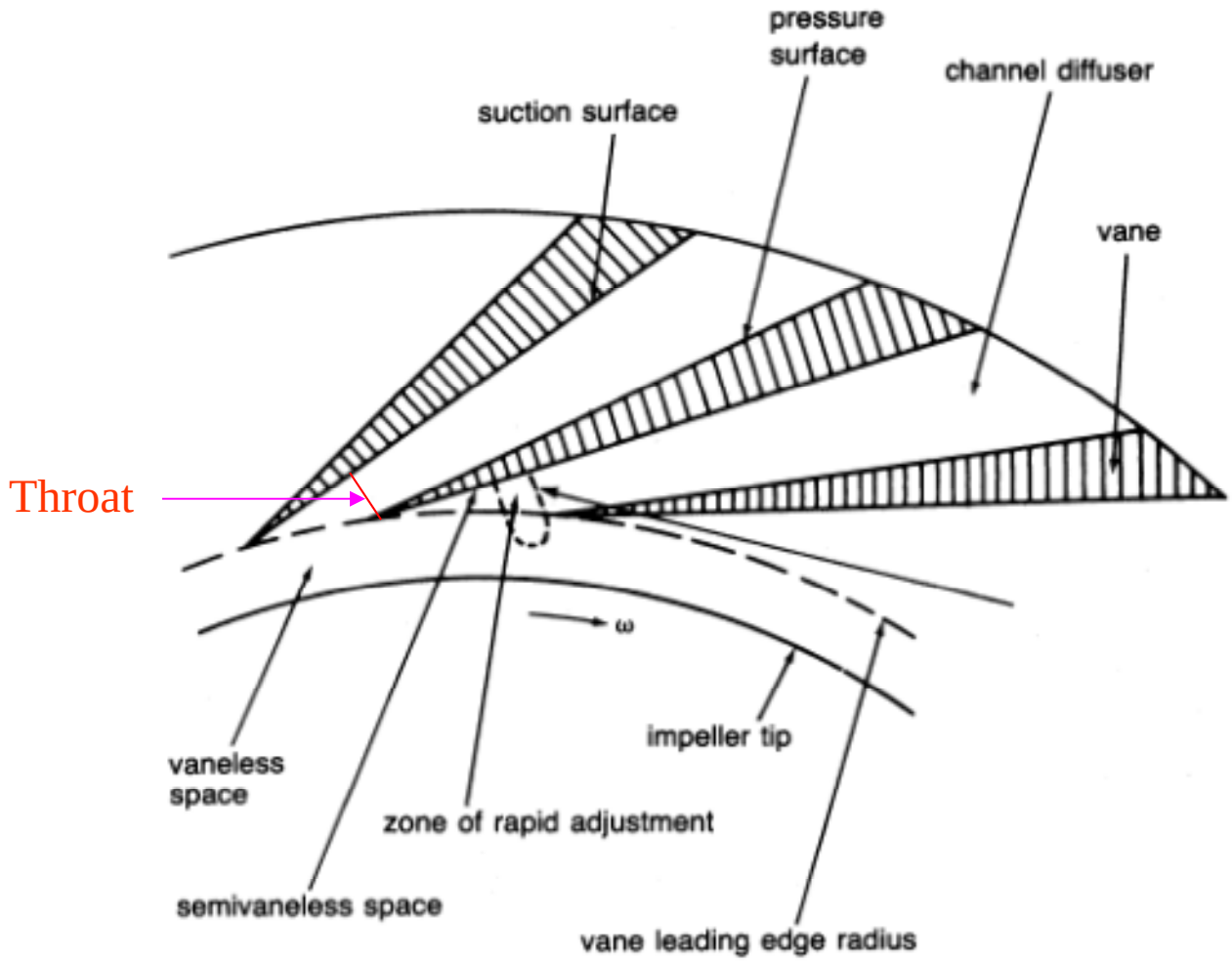
- **Radially ending**
- **Backswept**
- **Pre-swirl**
- **Above w/splitter blades**

Diffusers

- **Vaneless**
- **Vaned**
  - **Radial**
  - **Wedge**
- **Discrete-passage**



Typical Diffuser Geometry



Flow regions of vaned diffuser

Diffuser Design

Radial width of vane less space	= 5 cm
Approximate mean radius of diffuser throat	= 0.33 m
Depth of diffuser passages	= 1.76 cm
Number of diffuser vanes	= 12

<u>It is required to determine</u>:

(a) the inlet angle of the diffuser vanes, and

(b) the throat width of the diffuser passages, which are assumed to be of constant depth.

For simplicity, it will be assumed that the additional friction loss in the short distance between impeller tip and diffuser throat is small and therefore the 50 % of the overall loss can be considered to have occurred up to the diffuser throat. <u>For convenience, suffix 2 will be used to denote any plane in the flow after the impeller tip, the context making it clear which plane is under consideration.</u>

Iteration on C_{r2} at Diffuser Inlet

(a) Consider conditions at the radius of the diffuser vane leading edge, i.e. at r_2 = 0.25+0.05 = 0.3m. Since in the vaneless space $C_w r = constant$ for constant angular momentum,

$$C_{w2} = 410 * \frac{0.25}{0.30} = 342 m/s$$

The radial component of velocity can be found by trial and error. The iteration may be started by assuming that the temperature equivalent of the resultant velocity is that corresponding to the whirl velocity, but only the final trial is given here.

Try C_{r2} = 97 m/s

$$\frac{C_2^{\ 2}}{2c_p} = \frac{3.42^2 + 0.97^2}{0.201} = 62.9\,\mathrm{K}$$

Ignoring any additional loss between the impeller tip and the diffuser vane leading edges at 0.3m radius, the stagnation pressure will be that calculated for the impeller tip, i.e.

$$\frac{p_{02}}{p_{01}} = 1.582^{3.5}$$

Static Density at Diffuser Inlet

Proceeding as before we have

$$T_2 = 488 - 62.9 = 425.1\,\mathrm{K}$$

$$\frac{p_2}{p_{02}} = \left(\frac{425.1}{488}\right)^{3.5}$$

$$\frac{p_2}{p_{01}} = \left(1.582 * \frac{425.1}{488}\right)^{3.5} = 3.07$$

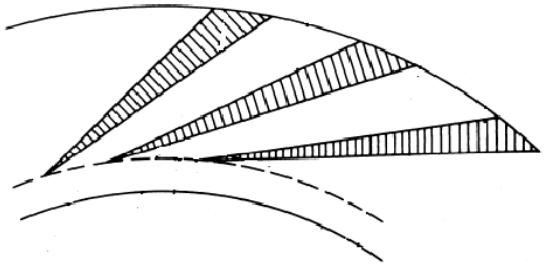
$$p_2 = 3.07 * 1.1 = 3.38 \text{ bar}$$

$$\rho_2 = \frac{3.38 * 100}{0.287 * 425.1} = 2.77 \text{ kg/m}^3$$

Diffuser Inlet Angle

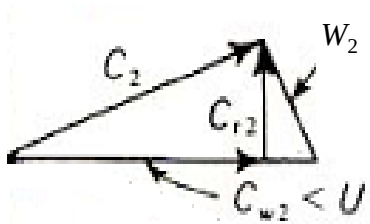
Area of cross-section of flow in radial direction

$$= 2\,\pi * 0.3 * 0.0176 = 0.0332\ \text{m}^2$$



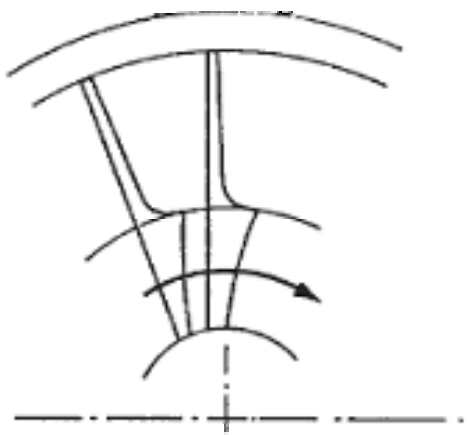
Check on C_{r2}:

$$C_{r2} = \frac{9}{2.77 * 0.0332} = 97.9\ \text{m/}s$$



Taking C_{r2} as 97.9 m/s, the angle of the diffuser vane leading edge for zero incidence should be

$$\tan^{-1}\left(C_{r2} \big/ C_{w2} \right) = \tan^{-1} \frac{97.9}{342} = 16^\circ$$



Static Density at Diffuser Throat

(b) The throat width of the diffuser channels may be found by a similar calculation for the flow at the assumed throat radius of 0.33 m

$$C_{w2} = 410 * \frac{0.25}{0.33} = 311\,\text{m/s}$$

Try $C_{r2} = 83\,\text{m/s}$

$$\frac{C^2{}_2}{2C_p} = \frac{3.11^2 + 0.83^2}{0.201} = 51.5\,\text{K}$$

$$T_2 = 488 - 51.5 = 436.5\,\text{K}$$

$$\frac{p_2}{p_{01}} = \left(1.582 * \frac{436.5}{488}\right)^{3.5} = 3.37$$

$$p_2 = 3.37 * 1.1 = 3.71\,\text{bar}$$

$$\rho_2 = \frac{3.71 * 100}{0.287 * 436.5} = 2.96\,\text{kg/m}^3$$

Diffuser Angle at Throat

As a first approximation, we may neglect the thickness of the diffuser vanes so that the area of flow in the radial direction

$$= 2 * \pi * 0.33 * 0.0176 = 0.0365 \text{ m}^2$$

Check on C_{r2} :

$$C_{r2} = \frac{9}{2.96 * 0.0365} = 83.3 \text{ m/s}$$

$$\text{Direction of flow} = \tan^{-1}\frac{83.3}{311} = 15^\circ$$

Diffuser Throat Width

Now, $m = \rho_2 A_{r2} C_{r2} = \rho_2 A_2 C_2$, or $A_2 = {A_{r2} C_{r2}}/{C_2}$,

Hence, flow area in the direction of resultant velocity, i.e. total throat area of the diffuser passages is

$$0.0365 \sin 15^\circ = 0.00945 \text{ m}^2$$

Therefore, with 12 diffuser vanes, the width of the throat in each passage of depth 0.0176 m is,

$$\frac{0.00945}{12 * 0.0176} = 0.0440 \text{ m} \quad \text{or} \quad 4.40 \text{ cm}$$

Inducer Relative Mach Number on Ground

Consider inlet Mach number at the tip radius of the impeller eye

Inlet velocity	=	143 m/s (axial)
Eye tip speed	=	273 m/s
Relative velocity at tip	=	$\sqrt{\left(143^2 + 273^2\right)} = 308\ \text{m/s}$
Velocity of sound	=	$\sqrt{\left(1.4 * 0.287 * 284.5 * 10^3\right)} = 338\ \text{m/s}$
Maximum Mach number at inlet	=	308/338 = 0.91

This would not be considered satisfactory even if it were actually the maximum value likely to be reached. But, if the compressor is part of an aircraft engine required to operate at an altitude of 11000 m, where the atmospheric temperature is only about 217 K, we must calculate the Mach number under these conditions.

Inducer Relative Mach Number in Flight

Since there will be a temperature rise due to the ram effect in the intake when the aircraft has a forward speed, the effect of drop in atmospheric temperature will not be quite so great as might be expected. We will take 90 m/s (324 km/hr) as being the minimum speed likely to be reached at high altitude.

Temperature equivalent of forward speed	= 4K
Inlet stagnation temperature	= 217 + 4 = 221 K
Temperature equivalent of axial inlet velocity from the first example	= 10.5 K
Inlet static temperature at altitude	= 210.5 K
Inlet Mach number at altitude	$= 0.91\left(\frac{284.5}{210.5}\right)^{\frac{1}{2}} = 1.06$

This is clearly too high and we will find the Mach number when an inlet prewhirl of 30° is introduced.

Effect of Inlet Prewhirl

In this case the absolute inlet velocity will be slightly higher than before, so that the inlet static temperature will be slightly lower. A new value for the axial velocity must be found by the usual trial and error process. Reverting to the original sea-level static case:

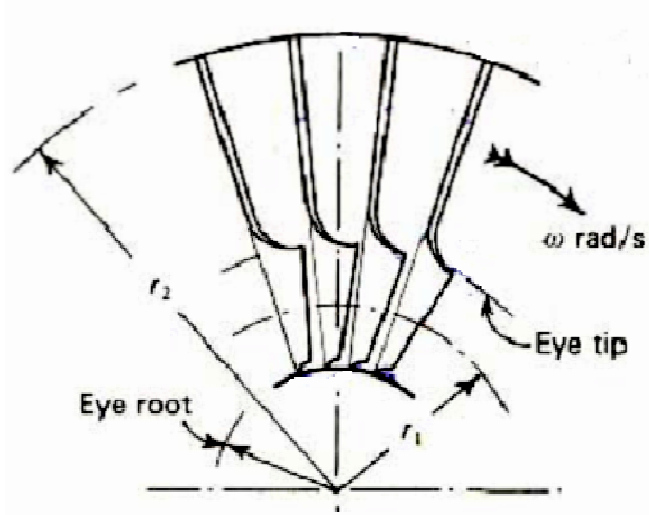
Try C_{a1} =150 m/s

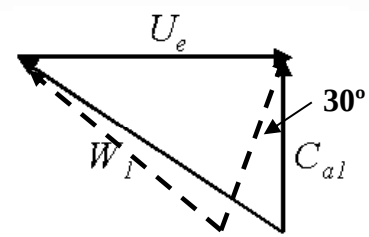
C_1 = 150/cos 30 = 173.2 m/s

Temperature equivalent of C_1 = 14.9 K

T_1 = 295-14.9 = 280.1K

p_1 = 0.918 bar and $\rho_1 = 1.14 kg/m^3$





Check on C_{a1} = $\dfrac{9}{1.14*0.053} = 149 m/s$

Whirl velocity at inlet, C_{w1} = 149 tan 30

= 86 m/s

Maximum relative velocity = $\sqrt{149^2 + (273-86)^2}$

= 239 m/s

Hence maximum inlet Mach number at T_{01}= 295 K is $\dfrac{239}{\sqrt{(1.4*0.287*280.1*10^3)}} = 0.71$

Effect of Prewhirl on Pressure Ratio

Under altitude conditions this would rise to little more than 0.8. Hence, a prewhirl of 30° can be regarded as adequate.

To show the effect of 30° prewhirl on the pressure ratio, we will take the worst case and assume that the prewhirl is constant over the eye of the impeller.

Speed of impeller eye at mean radius, $U_e = \dfrac{237+136.5}{2} = 204.8\,\text{m/s}$

Actual temperature rise $= \dfrac{\psi}{c_p}\left(\sigma U^2 - C_{w1}U_e\right)$

$$= \frac{1.04\left(0.9*455.5^2 - 86*204.8\right)}{1.005*10^3} = 175\,\text{K}$$

$$\frac{p_{03}}{p_{01}} = \left[1+\frac{0.78*175}{295}\right]^{3.5} = 3.79$$

This pressure ratio may be compared with the original value of 4.23 obtained with no prewhirl. It is sometimes advantageous to use adjustable inlet guide vanes to improve the performance under off-design conditions.

Mach Number at Impeller Exit

We next consider the relevant Mach numbers in the diffuser. The maximum value will occur at the entry to the diffuser, that is, at the impeller tip. Once again the values calculated in the previous example will be used.

The temperature equivalent of the resultant velocity of the air leaving the impeller was found to be

$$\frac{C_2^2}{2C_p} = 93.8K$$

and hence

$$C_2 = 434\text{m/s}$$

T_2 was found to be 394.2 K and thus the Mach number at the impeller tip equals

$$\frac{434}{\sqrt{\left(1.4*0.287*394.2*10^3\right)}} = 1.09$$

Mach Number at Diffuser Inlet

Now consider the leading edge of the diffuser vanes. The whirl velocity was found to be 342 m/s and the radial component 97.9 m/s. The resultant velocity at this radius is therefore 356 m/s. The static temperatures was 425.1 K at this radius so that the Mach number is

$$\frac{356}{\sqrt{1.4*0.287*425.1*10^3}} = 0.86$$

In the particular design under consideration, the Mach number is 1.09 at the impeller tip and 0.86 at the leading edges of the diffuser vanes. It has been found that as long as the radial velocity component is subsonic, Mach numbers greater than unity can be used at the impeller tip without loss of efficiency. It appears that supersonic diffusion can occur without the formation of shock waves if it is carried out at constant angular momentum with vortex motion in the vaneless space. But the Mach number at the leading edge of the diffuser vanes is rather high and it would probably be advisable to increase the radial width of the vaneless space or the depth of the diffuser to reduce the velocity at this radius.

Comments

- **High Mach numbers at the leading edges of the diffuser vanes are undesirable, not only because of the danger of the shock losses, but because they imply high air speeds and relatively large pressures at the stagnation points where the air is brought to rest locally at the leading edges of the vanes. This causes a circumferential variation in static pressure, which is transmitted upstream in a radial direction through the vaneless space to excite the impeller tip. Although the variation would considerably reduce by the time it reaches the impeller, it may well be large enough to excite the impeller vanes and cause mechanical failure due to vibrational fatigue cracks in the vanes. This will occur when the exciting frequency, which depends on the rotational speed and relative number of impeller and diffuser vanes, is of the same order as one of the natural frequencies of the impeller vanes. To reduce the likelihood of this, care is taken to see that the number of vanes in the impeller is not an exact multiple of the number in the diffuser; it is common practice to use a prime number for the impeller vanes and an even number for the diffuser vanes.**

- **The reason for the vaneless space will now be apparent: the dangers of both, shock losses and excessive circumferential variation in static pressure, would be considerably increased if the leading edges of the diffuser vanes were too near the impeller tip where the Mach numbers are very high.**

Further Work

With these dimensions

1. Create a geometric model of the Centrifugal compressor using a CAD tool.
2. Export the model to a CFD tool and carry out the fluid flow analysis
3. Using an FEM tool, carry out the structural dynamic analysis choosing appropriate material properties.
4. Create a shaft and choose bearings and carryout rotor dynamic analysis
5. Generate manufacturing drawings
6. Generate compressor operating characteristics using a suitable post processor.

Design Guidelines

- Relative velocity ratio, $W_2/W_{sh1} > 0.75$ → de Haller number
- Exit absolute flow angle, $\alpha_2 = 60^o - 65^o$
- Sweep angle for backward swept impeller = 30º
- Ratio of inducer tip diameter to impeller outer diameter, $d_{sh1}/d_2 = 0.4 - 0.55$
- Number of impeller blades by Wiesner's correlation, $\sigma_s = 1 - \frac{\sqrt{\cos\beta_2}}{Z^{0.7}}$
- Inducer hub-tip diameter ratio = 0.5 – 0.6

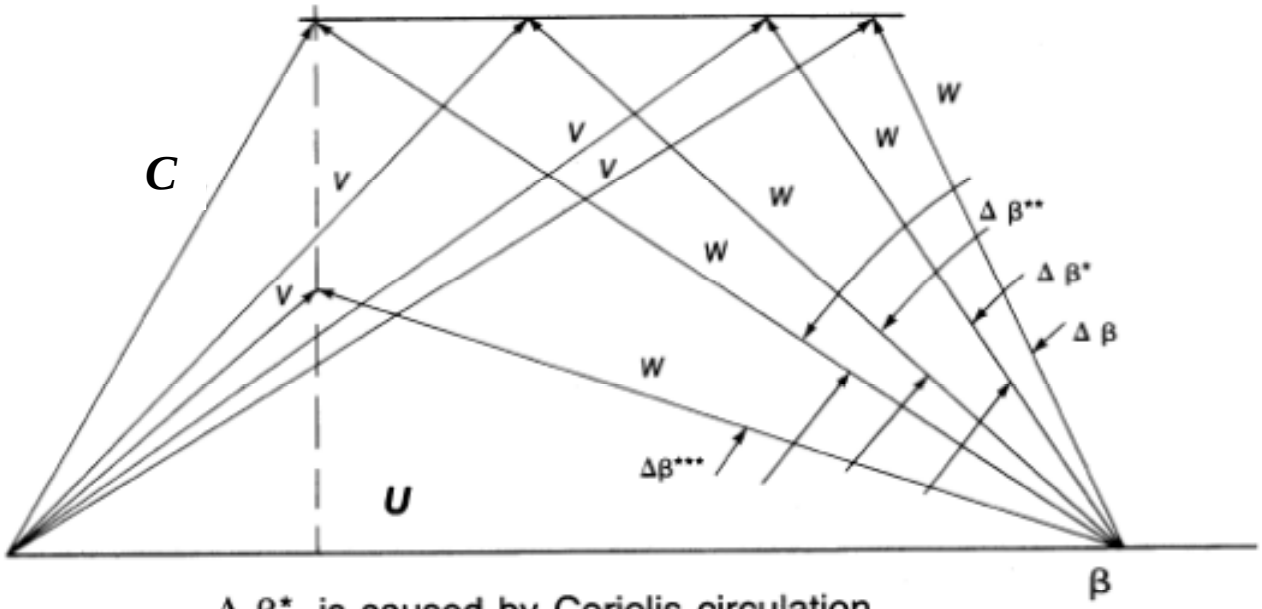
Effect of Geometric Parameters

Performance of the centrifugal compressor is affected by the following geometric parameters:

- Impeller vane thickness
- Splitter blades
- Trailing end skew
- Exit trim
- Inducer leading edge sweep
- Vaned shroud at inducer
- Impeller with integral shroud
- Inlet guide vanes

Impeller Vane Thickness

Because of manufacturing problems and physical necessity, impeller vanes have definite thickness. When fluid leaves the impeller, the vanes no longer contain the flow, and the meridional velocity decreases. Hence, both the relative and absolute velocities also decrease with consequent change in exit flow angle.

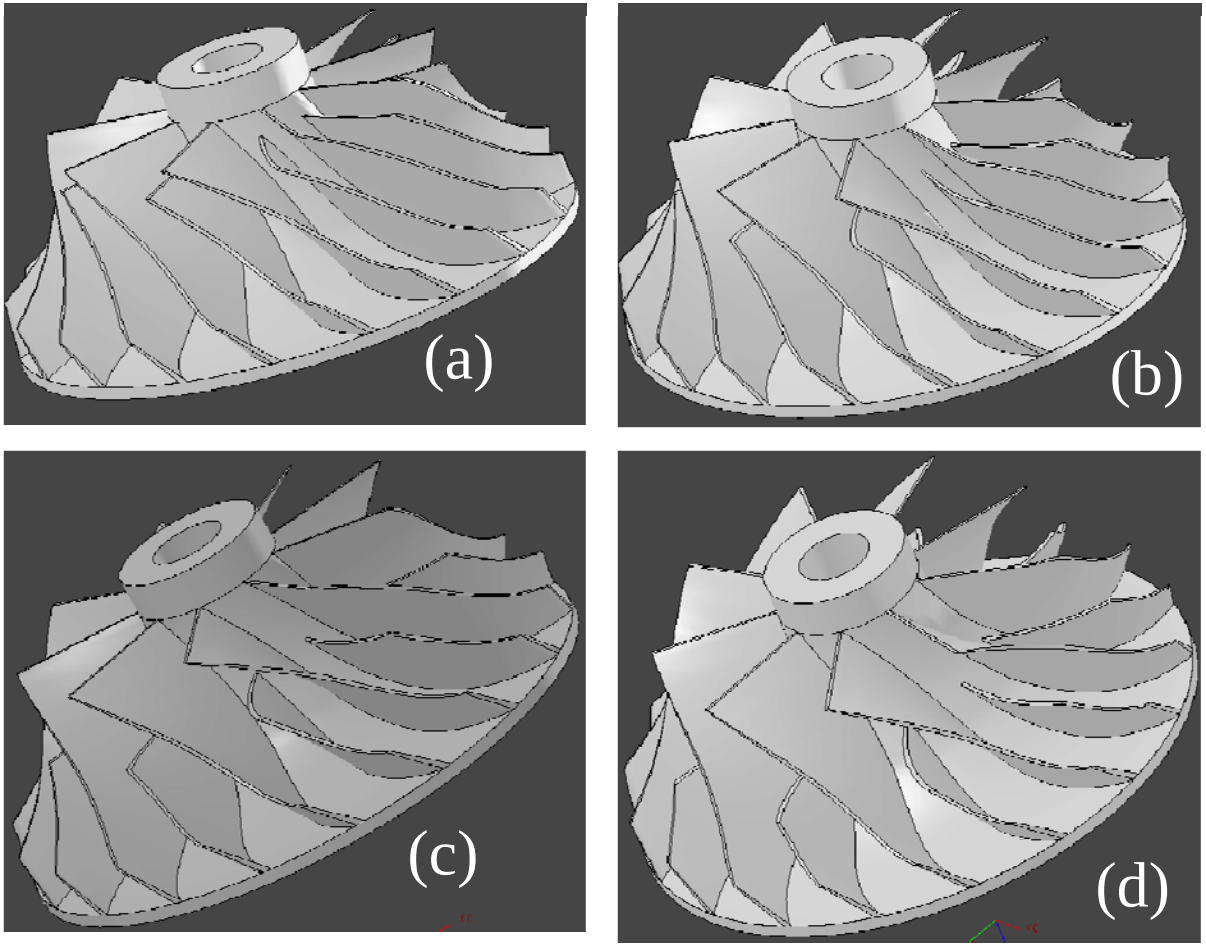


Δ β* is caused by Coriolis circulation

Δ β** is caused by boundary-layer effects

Δ β*** is caused by the blade thickness

Splitter Blades



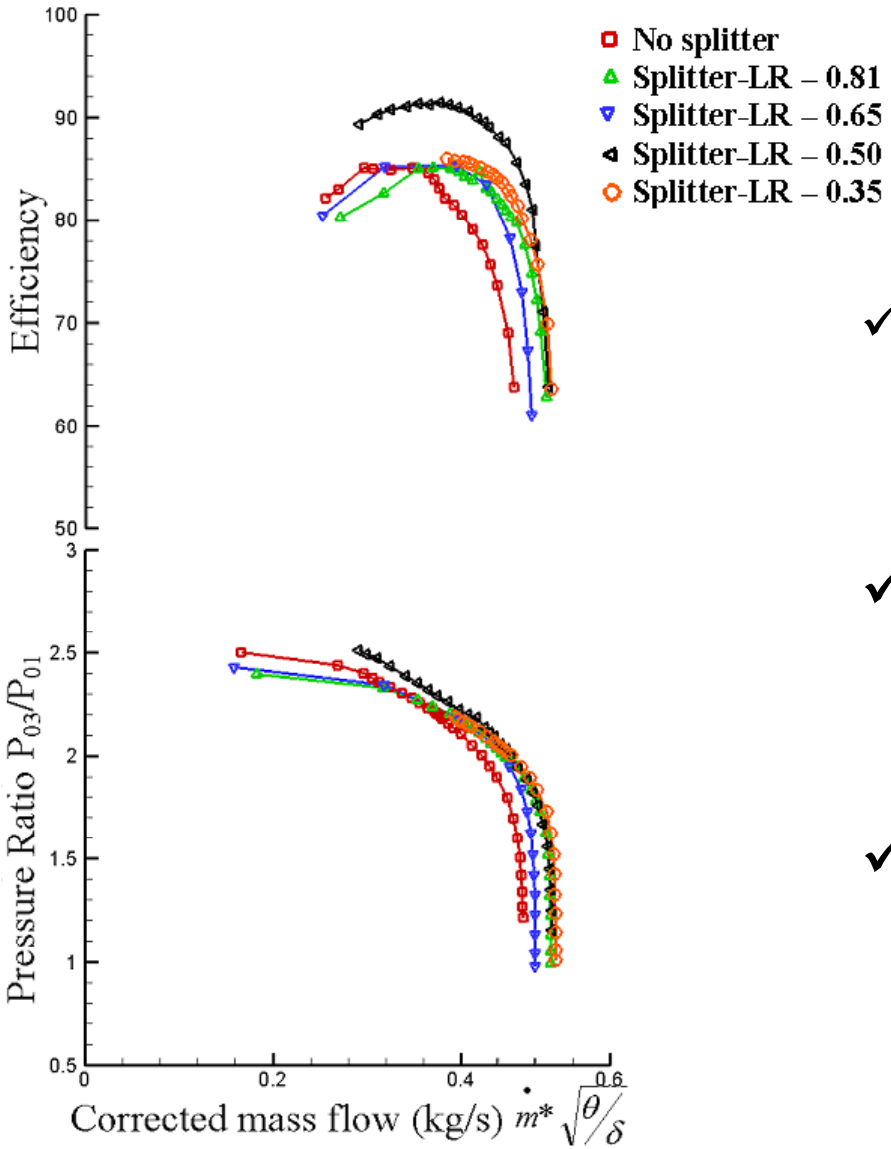
(a) LR- 0.81 (b) LR-0.65 (c) LR-0.50 (d) LR-0.35

Geometric models of impellers with different splitter to main blade length ratios (LR=0.81, 0.65, 0.50 and 0.35)

Compressor specifications

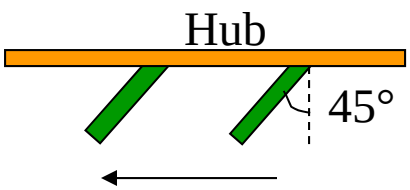
Inducer hub dia.	**22 mm**
Inducer tip dia.	**65 mm**
Impeller tip dia.	**87 mm**
Number of vanes	**18**
Rotational speed	**81000 rpm**
Inlet pressure	**0.98 bar**
Inlet temperature	**303 K**

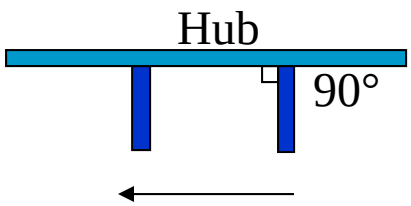
Splitter Blades

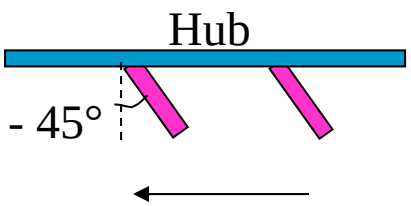


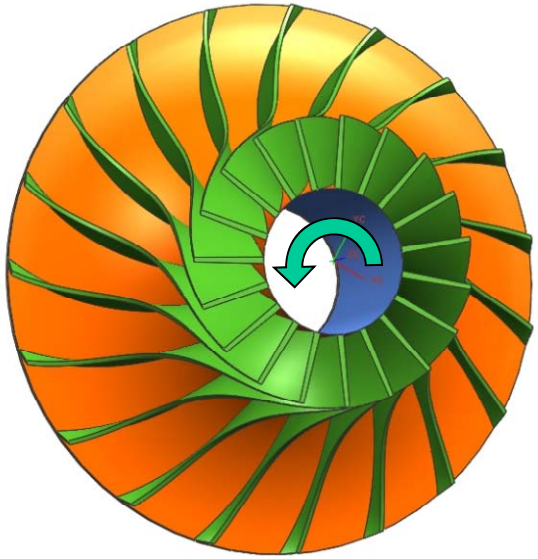
- ✓ The optimum length of splitter blade is half the length of the main blade.
- ✓ Maximum efficiency and pressure ratio at 0.50 splitter length.
- ✓ No significant improvement in performance when splitter length is reduced further.

Trailing End Skew

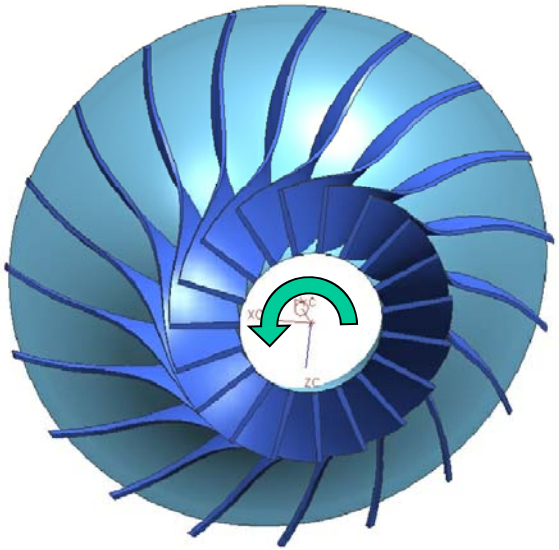








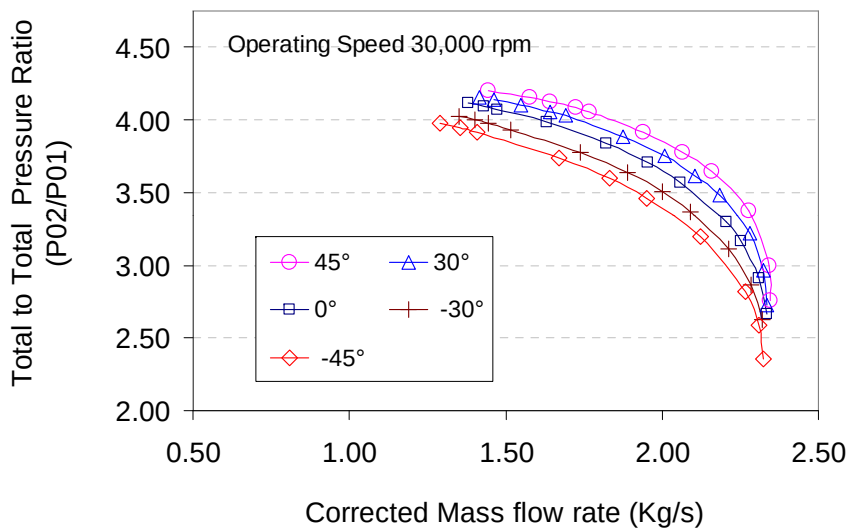
45° Skew

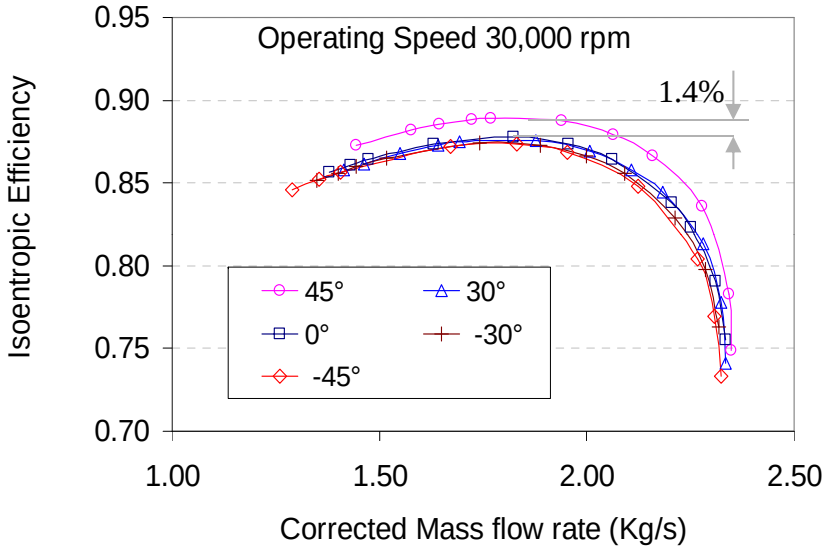


0° Skew

-45° Skew

Trailing End Skew

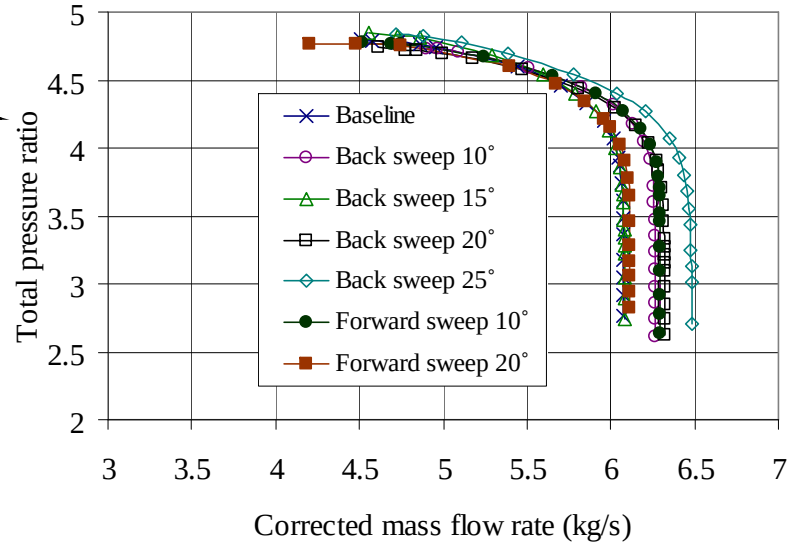


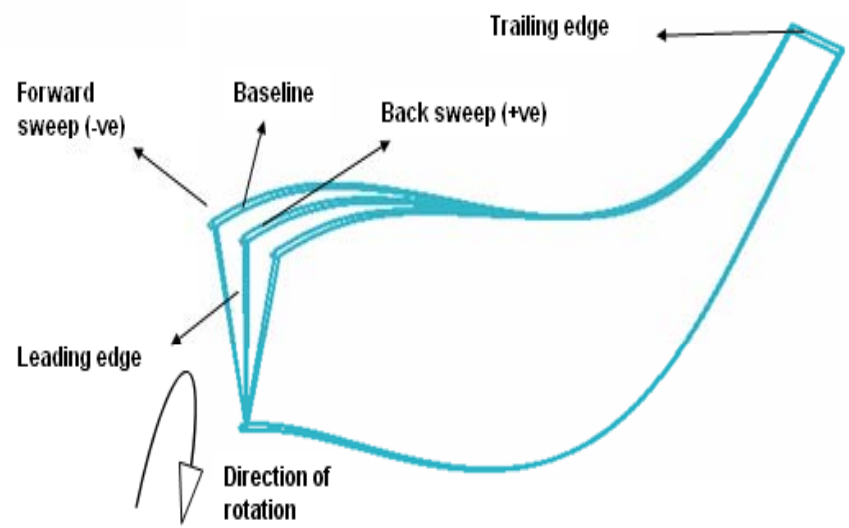


Effect of exit skew on impeller performance

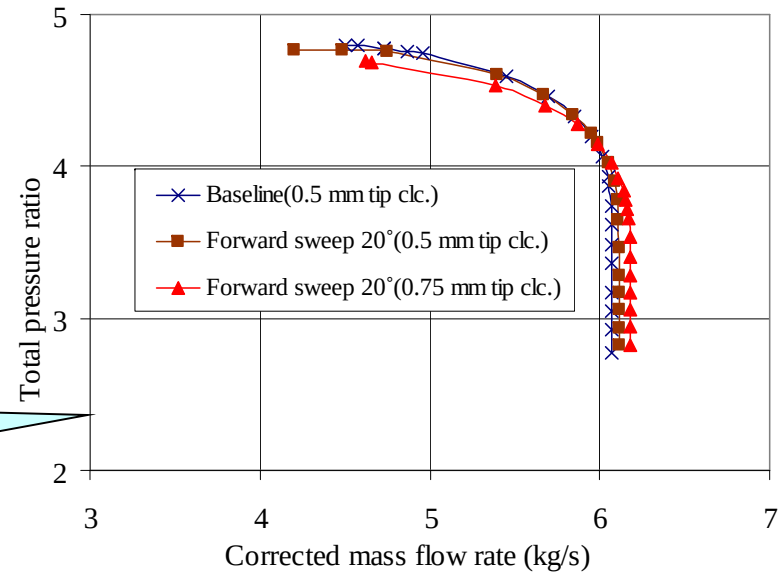
Leading End Skew

Stall margin has increased with 20º forward sweep



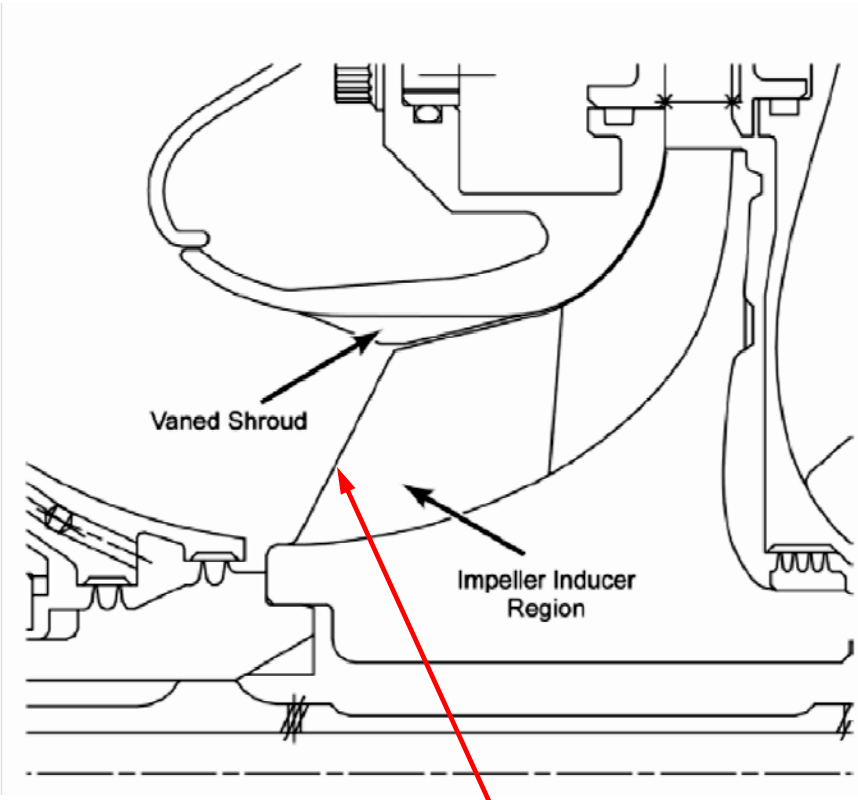


Definition of leading end sweep

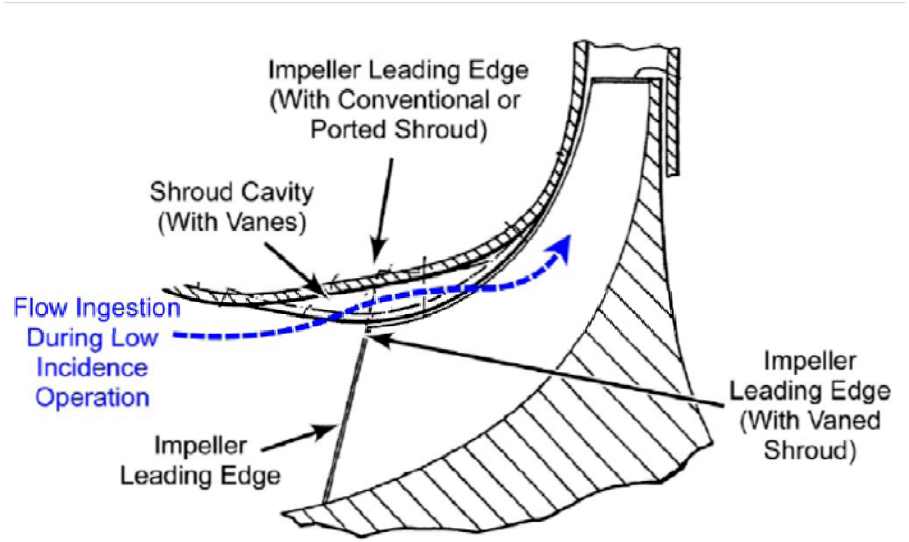


A higher tip clearance with forward sweep may reduce performance but the compressor production cost will be low

Vaned Shroud at Inducer

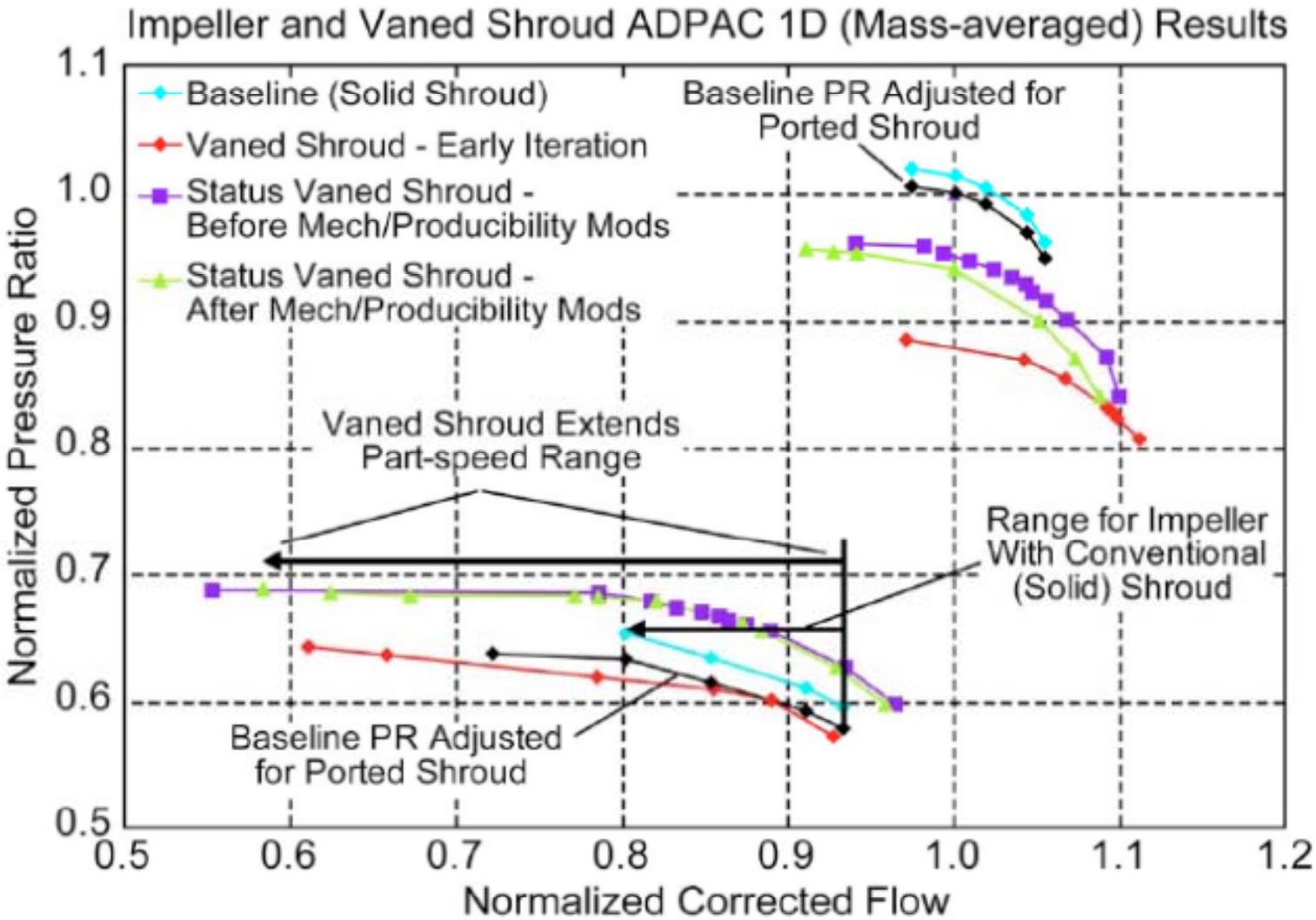


Inducer leading end sweep



Barton M.T. et al: ASME J. of Turbomachinery, Vol. 128, October 2006, pp 627-631

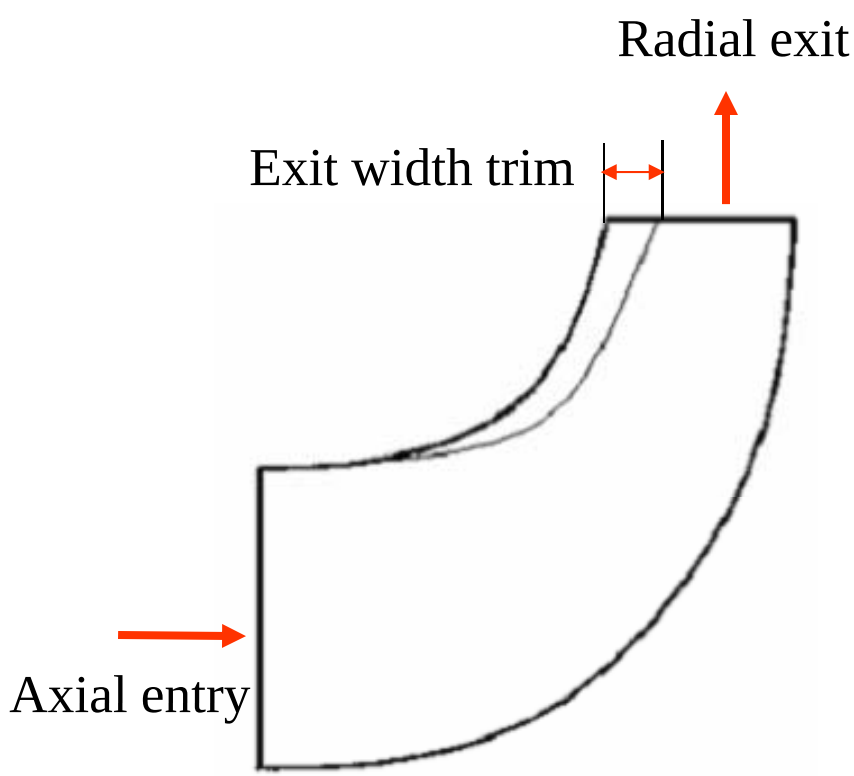
Vaned Shroud at Inducer



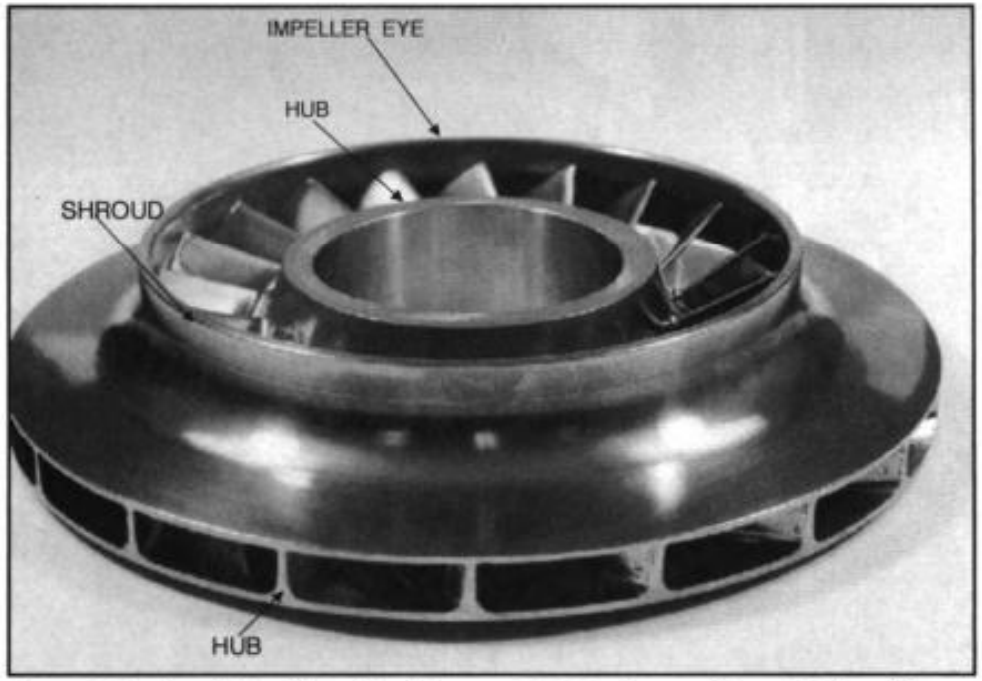
Barton M.T. et al: ASME J. of Turbomachinery, Vol. 128, October 2006, pp 627-631

Exit Trim

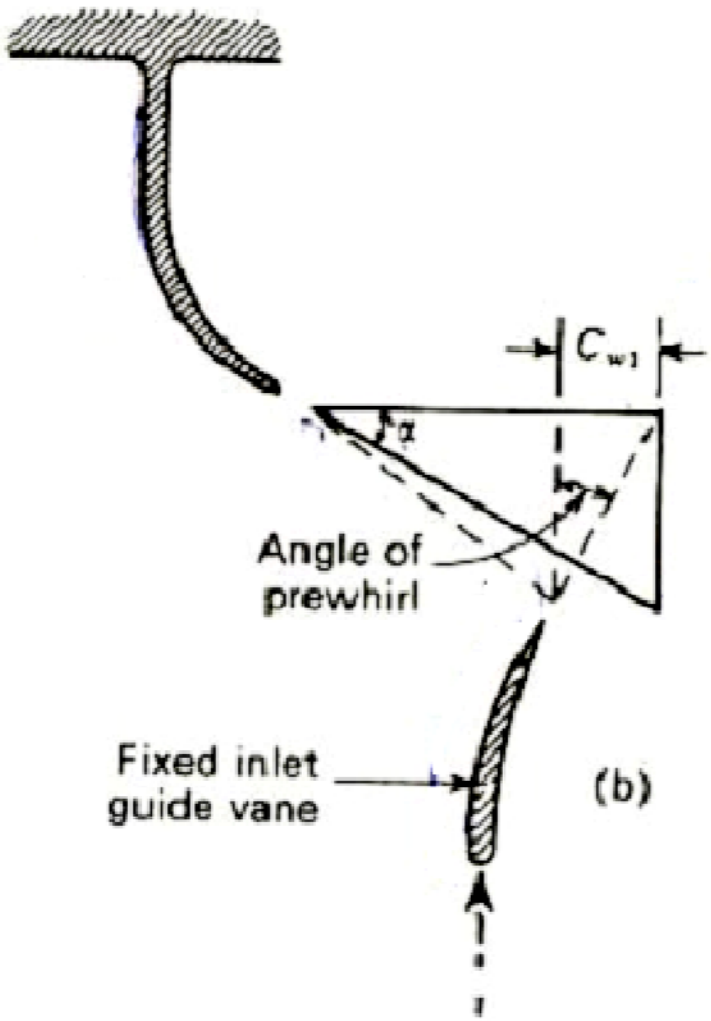
Altering the axial width of the impeller vanes at exit, known as **Exit Trim**, with respect to the baseline design, helps the manufacturer in producing design variants to meet different customer requirements without spending effort in carrying out an *ab initio* design for a new product within a certain range of specifications.



Impeller with Integral Shroud



Inlet Guide Vanes



- Inlet Guide Vanes (IGV) provide prewhirl to the flow entering the impeller.
- The inlet velocity triangle is altered.
- The work capacity of the compressor also changes due to introduction of whirl component of velocity.

$$W = U_2 C_{w2} \pm U_1 C_{w1}$$

- Introduction of IGVs is desired to improve the off-design performance of the compressor and to limit the inducer tip Mach number; but the friction losses will tend to increase.

Session Summary

This session has covered:

- Basic design methodology of centrifugal compressors.
- Calculation of aerodynamic and geometric parameters of the impeller and generation of velocity triangles.
- Calculation of diffuser parameters, especially the diffuser inlet angle and the throat width of diffuser passage.
- Effect of geometric parameters on compressor performance.