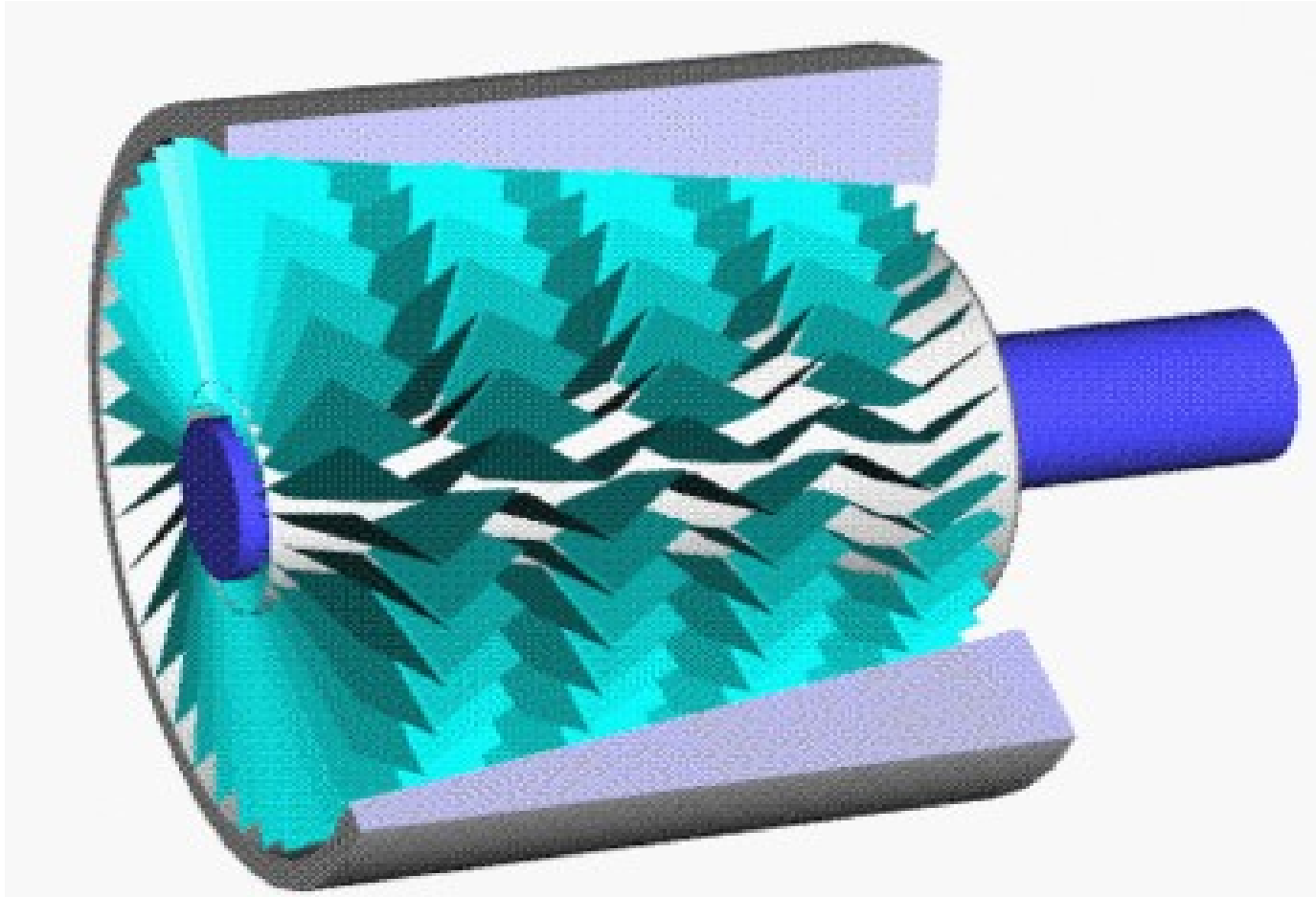


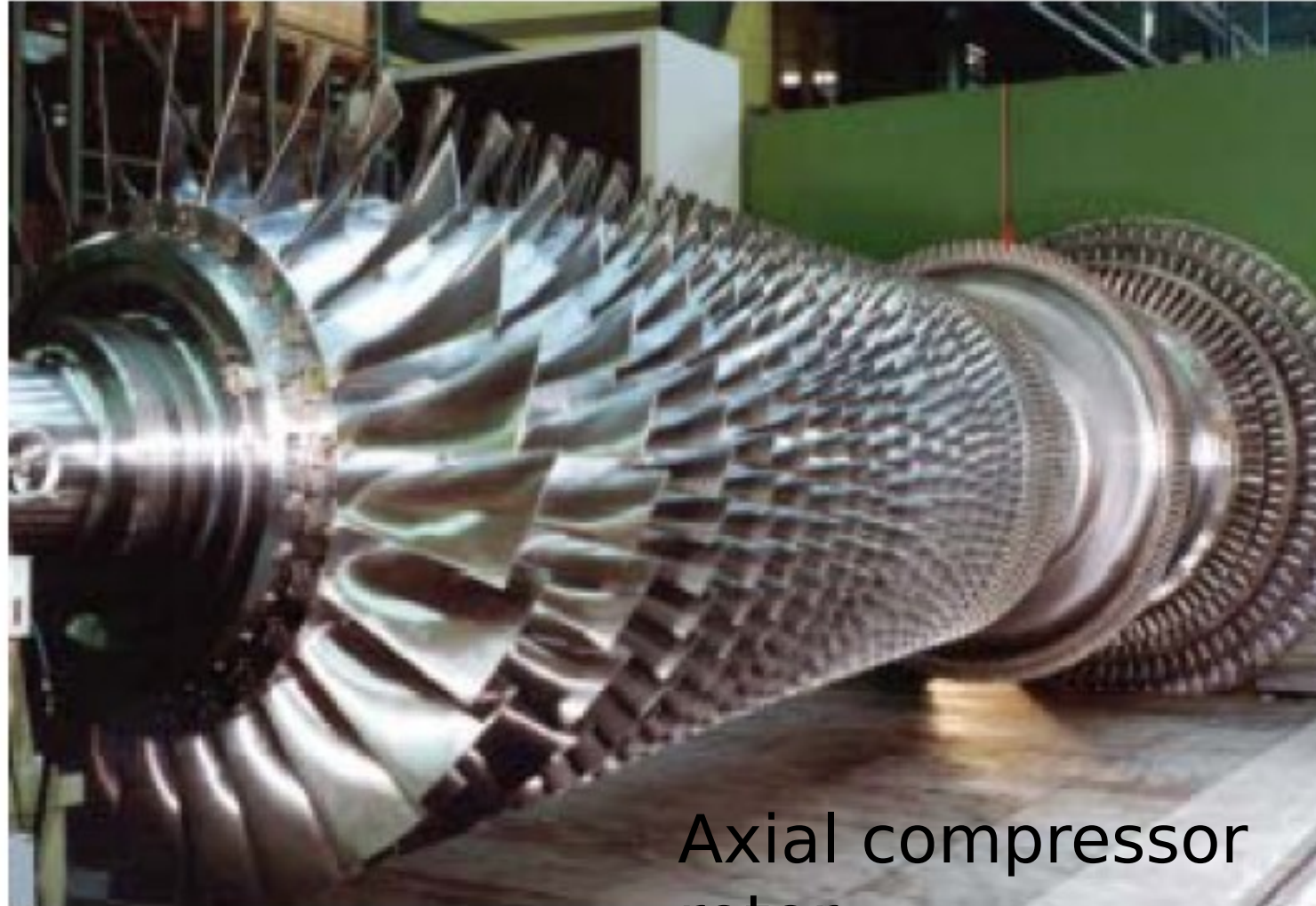
CHAPTER SIX



AXIAL COMPRESSORS

Contents

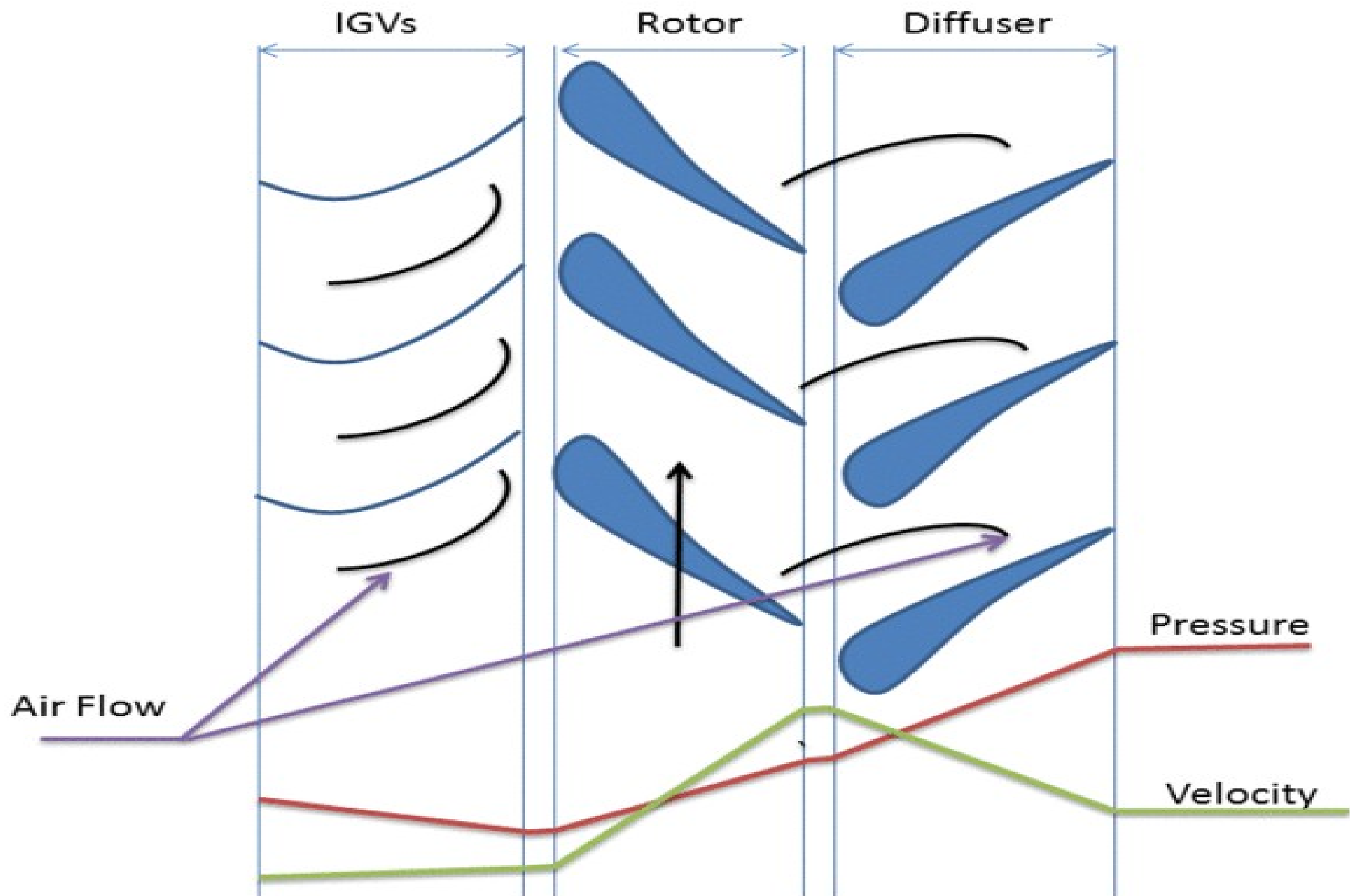
- *Working principle of axial compressor stage*
- *Enthalpy entropy diagram*
- *Work done factor*
- *Radial equilibrium the*
- *Surging and stalling*

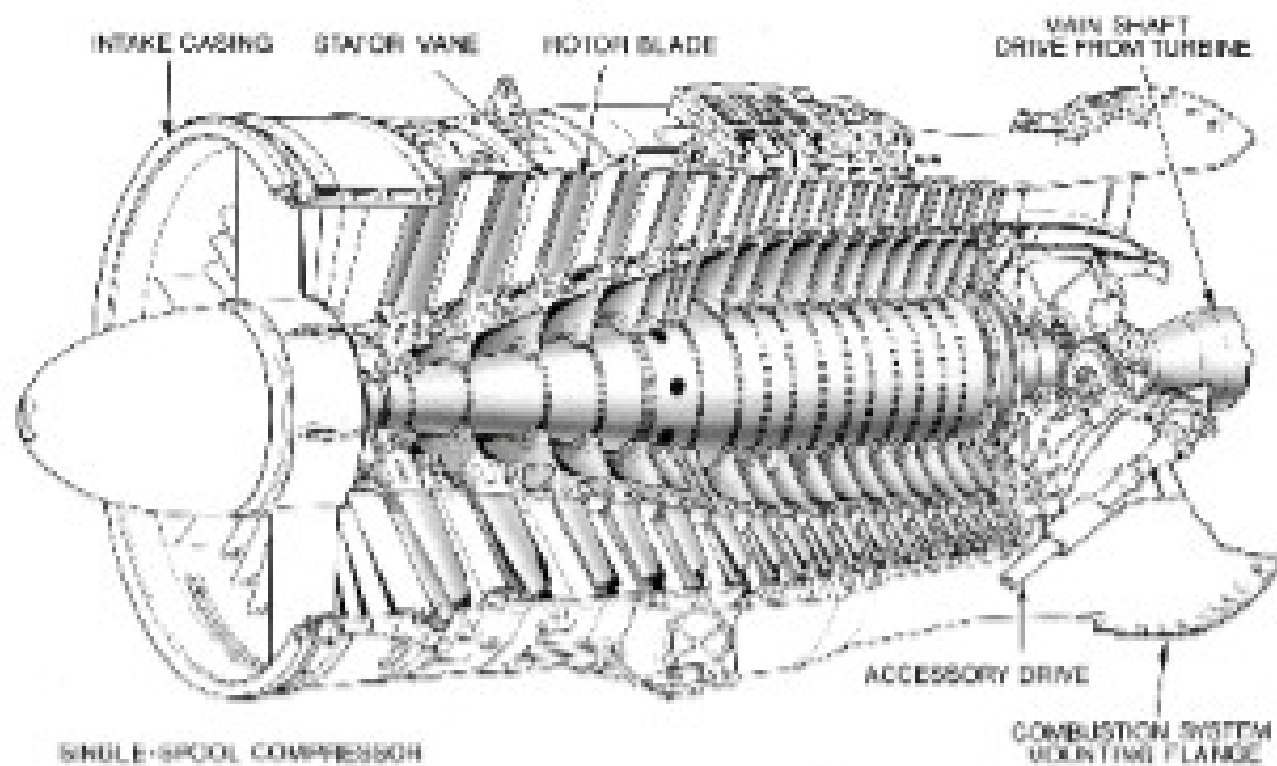
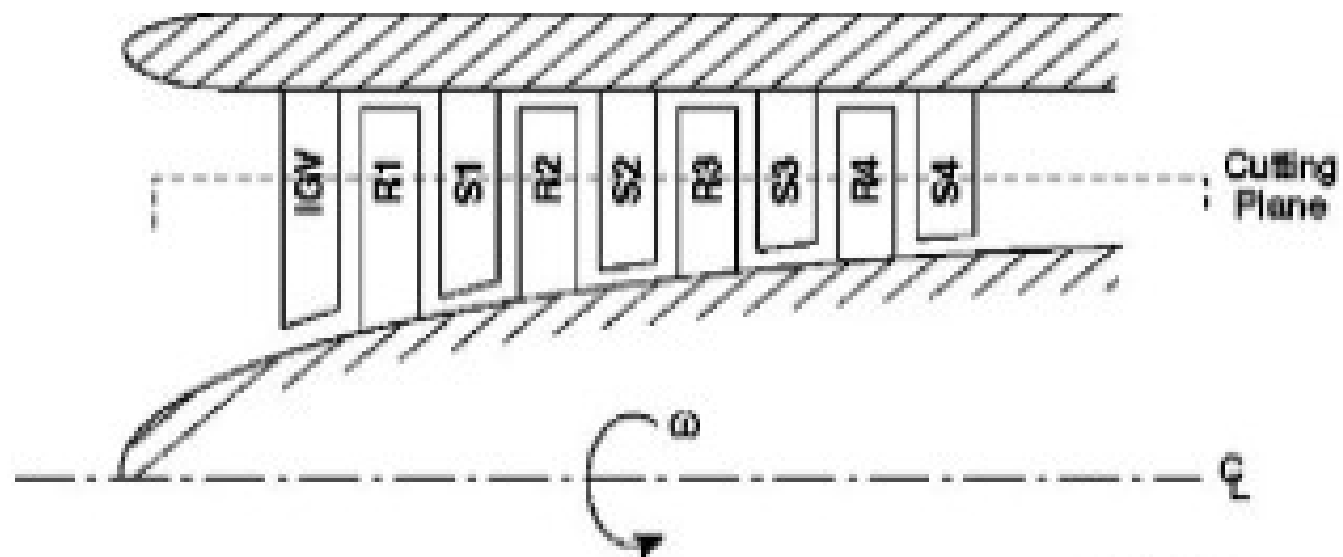


Axial compressor
rotor

Working Principle

- A compressor is a machine which increases the static pressure of a fluid.
- An axial flow compressor is usually designed to have multi – stages due to a small pressure ratio that can be attained in a single stage.
- Inlet guide vanes are provided at the inlet of an axial flow compressor which directs and accelerates the flow towards the rotor.
- A stage in an axial compressor is comprised of a rotor blade followed by a stator blade and the inlet guide vane is independent of a stage.
- The rotor increases the stagnation pressure of the fluid, which gives rise to both static pressure and velocity of the fluid. Further static pressure increase is brought by diffusing the fluid through the stator (diffuser) blades.





Comparisons

- Comparison between axial flow turbines and compressors.
 - Low turning of the flow in axial compressors than in axial turbines.
 - Small pressure ratio per a single stage in axial compressors than axial flow turbines.

Why?

- Aerodynamic losses are intensified in compressor flows than turbine flows. This is due to that flow through nozzles are aerodynamically more efficient than flow through diffusers.
- An axial flow turbine can be reversed to work as a compressor but the efficiency is disappointing.

Usually a very high rotational speed is required in axial compressors since the velocity triangles demands the blade velocity to be large to get the desired pressure ratio, compared to an axial turbine.

In turbines the limiting condition is imposed by rotor blade stresses, but in compressors stage performance is limited by Mach number consideration.

Typical velocity triangle for an axial compressor

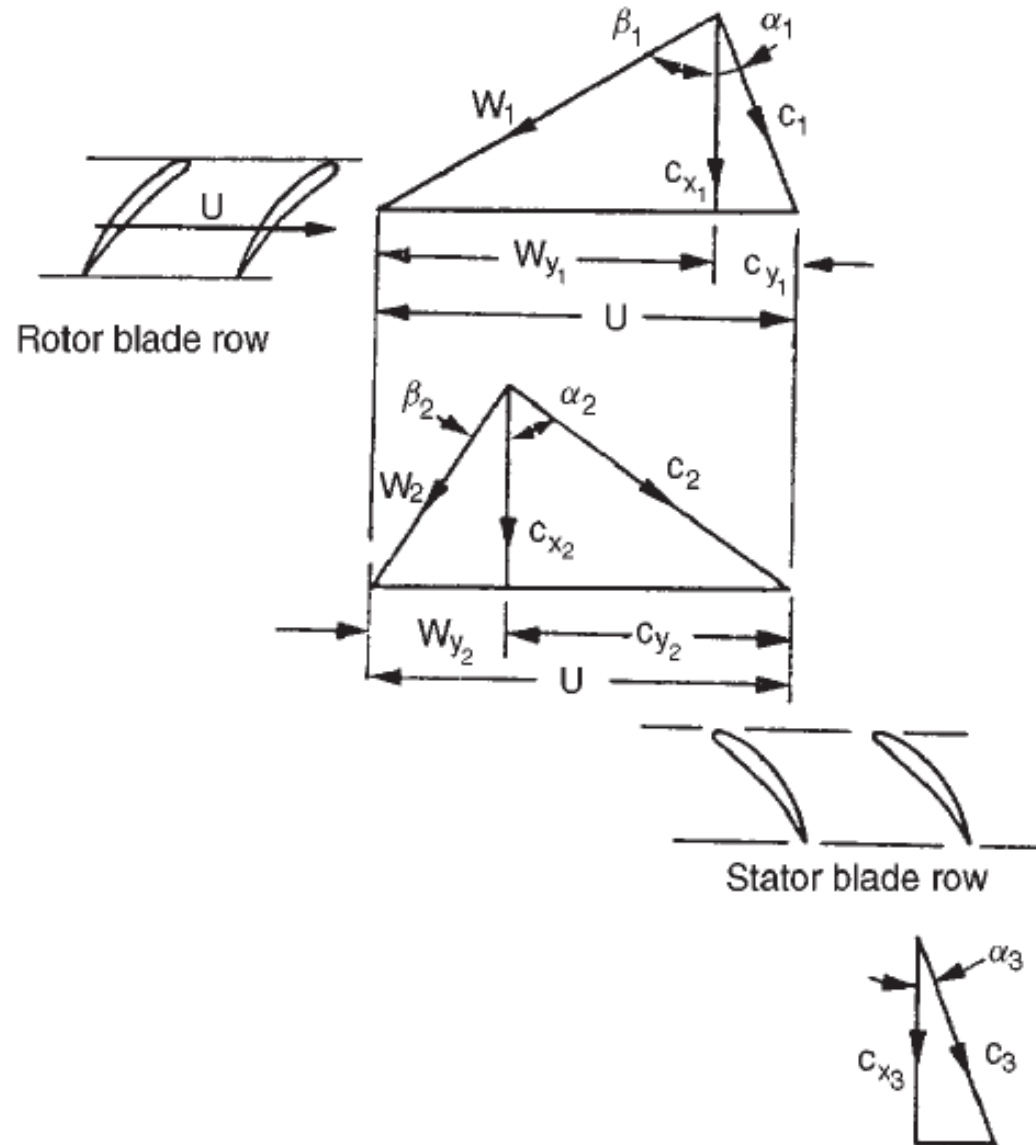


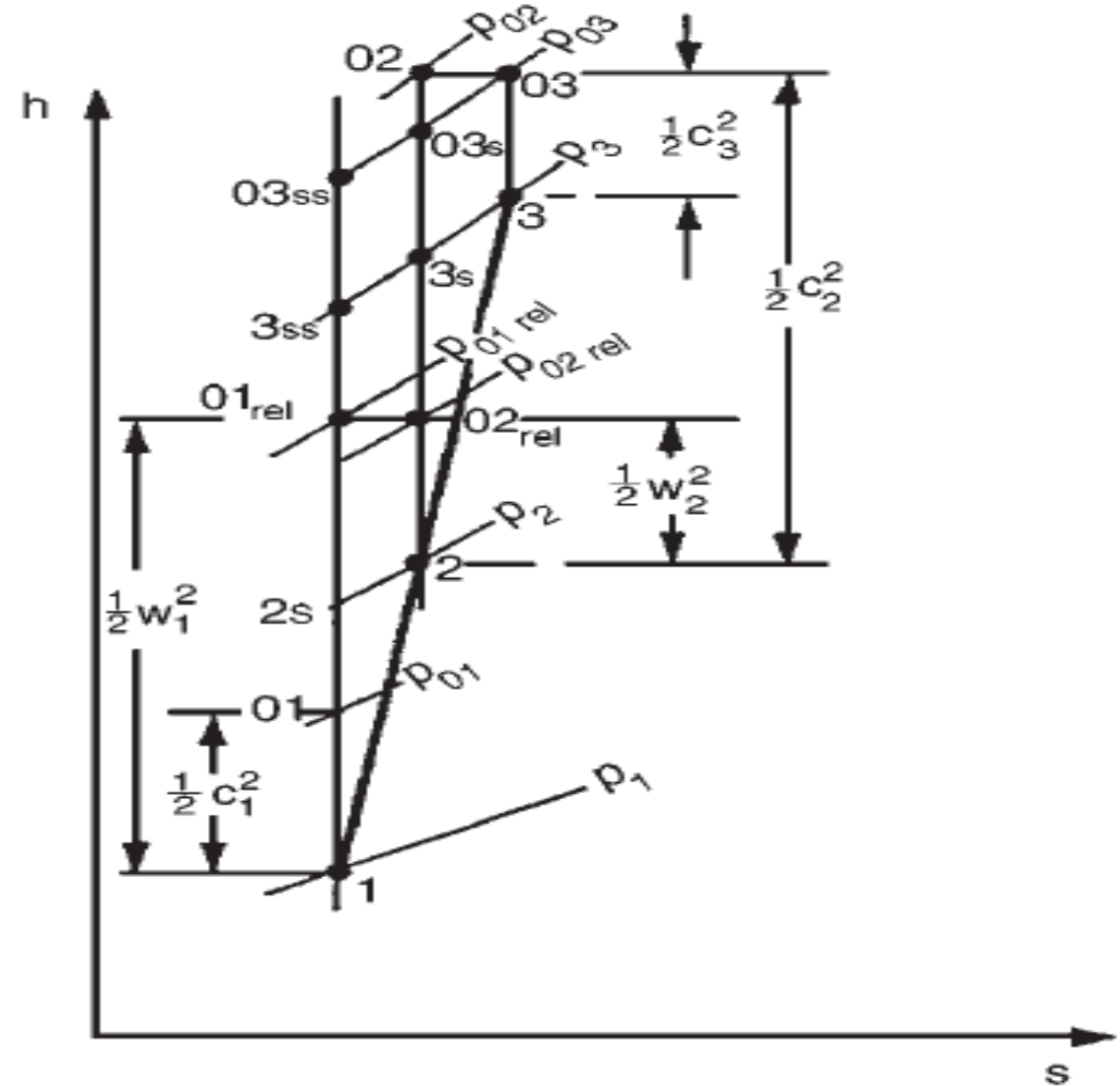
FIG. 5.2. Velocity diagrams for a compressor stage.

Practical stage pressure ratio and efficiency in axial compressors

Type of Application	Type of Flow	Inlet Relative Velocity Mach Number	Pressure Ratio per Stage	Efficiency per Stage
Industrial	Subsonic	0.4-0.8	1.05-1.2	88%-92%
Aerospace	Transonic	0.7-1.1	1.15-1.6	80%-85%
Research	Supersonic	1.05-2.5	1.8-2.2	75%-85%

Specific work and isentropic efficiency

- Specific work
- Isentropic efficiency
- For constant pressure process
(From 03 to 03s and 03 to 03ss)
- The change in temperature is on a fraction
of the absolute temperature,



- For adiabatic compression

$$h_3 - h_2 = \frac{1}{2}(c_2^2 - c_3^2)$$

$$= [(p_{02} - p_2) - (p_{03} - p_3)]/\rho$$

For

$$h_{3s} - h_2 = (p_3 - p_2)/\rho$$

Incompressible flow approximation for a single stage

$$h_3 - h_{3s} = (p_{02} - p_{03})/\rho = (1/\rho)\Delta p_{0\text{stator}}$$

$$h_2 - h_{2s} = (p_{01\text{rel}} - p_{02\text{rel}})/\rho = (1/\rho)\Delta p_{0\text{rotor}}$$

Similarly for the rotor

$$\eta_{ts} = \frac{h_{3ss} - h_{01}}{h_{03} - h_{01}}$$

- Degree of reaction

For incompressible and reversible process (), then

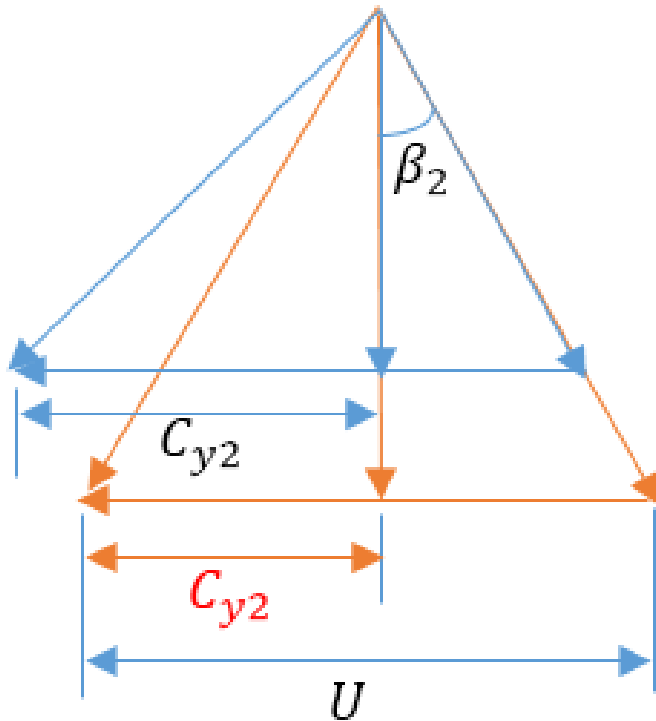
- Since an axial compressor performance is greatly affected by the pressure gradient across the blades, degree of reaction becomes one of important parameters that determine the efficiency.
- In a 50% reaction the rise in static enthalpy and pressure are equally shared by the rotor and stator blades. This ideally gives the best performance reaction wise. This was shown by the result of experimental studies undertaken by Howell.
- Howell reported that with a limit of 0.7 for the allowable Mach number, the efficiency and temperature rise are at a maximum with 50% reaction.

Blade loading

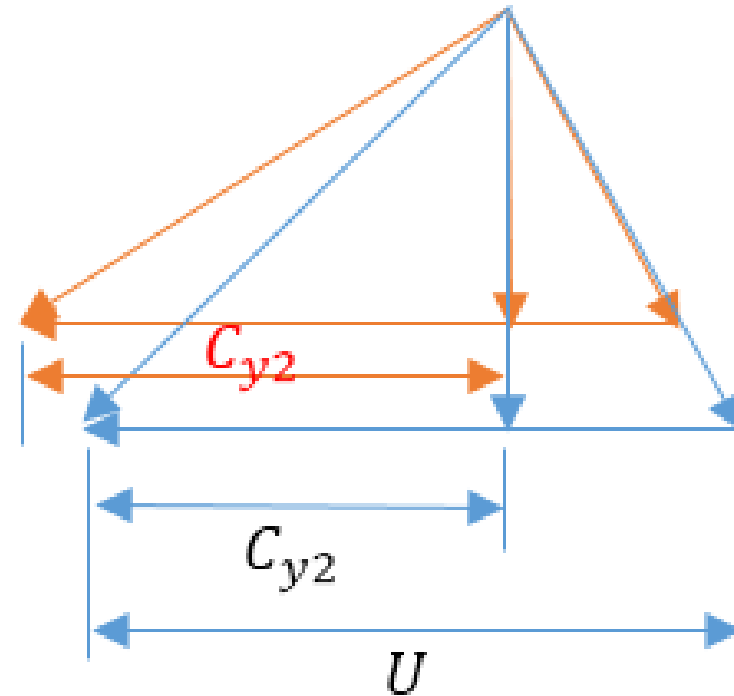
- Equivalent term for blade loading coefficient in axial flow turbines. It is an important parameter in optimizing the efficiency of an axial compressor stage.
- Since a compressor blade is sensitive to aerodynamic effect it will be helpful to define work done factor using aerodynamic parameters.
- For maximum blade efficiency, the optimum mean blade angle is

Work done factor

- Boundary layer development on the annulus wall makes a velocity profile that increases the axial velocity at the mean and decreases it the blade tip and hub. This effect is taken in to account by the work done factor, .



Compressor blade



Compressor blade tip and hub

Stage and Total pressure rise

- The stage isentropic efficiency is given by

$$\eta_s = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} = \frac{(T_{02s}/T_{01}) - 1}{\Delta T_{0s}/T_{01}} = \frac{(P_{02}/P_{01})^{\frac{\gamma-1}{\gamma}} - 1}{\Delta T_{0s}/T_{01}} = \frac{r_{Ps}^{\frac{\gamma-1}{\gamma}} - 1}{\Delta T_{0s}/T_{01}}$$

- Therefore, the stage pressure ratio is

$$r_{Ps} = \left(1 + \frac{\eta_s \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$$

- For multi – stages of an axial compressor

- Constant stage temperature rise $T_{0n} = T_{01} + N\Delta T_{0s}$ $\frac{T_{0n}}{T_{01}} = 1 + \frac{N\Delta T_{0s}}{T_{01}}$

$$\frac{P_{02}}{P_{01}} = \left(\frac{T_{0n}}{T_{01}}\right)^{\frac{1}{\eta_P} \left(\frac{\gamma}{\gamma-1}\right)} = \left(1 + \frac{N\Delta T_{0s}}{T_{01}}\right)^{\frac{1}{\eta_P} \left(\frac{\gamma}{\gamma-1}\right)} \quad r_{PO} = \left(1 + \frac{N\Delta T_{0s}}{T_{01}}\right)^{\frac{1}{\eta_P} \left(\frac{\gamma}{\gamma-1}\right)}$$

- Constant stage pressure rise

$$r_{PO} = r_{Ps}^N$$

Illustrative Example An axial flow compressor has a tip diameter of 0.95 m and a hub diameter of 0.85 m. The absolute velocity of air makes an angle of 28° measured from the axial direction and relative velocity angle is 56° . The absolute velocity outlet angle is 56° and the relative velocity outlet angle is 28° . The rotor rotates at 5000 rpm and the density of air is 1.2 kg/m^3 . Determine:

1. The axial velocity.
2. The mass flow rate.
3. The power required.

Solution:

1. Rotor speed is given by:

$$U = \frac{\pi DN}{60} = \frac{\pi(0.95)(5000)}{60} = 249 \text{ m/s}$$

Blade speed at the hub:

$$U_h = \frac{\pi D_h N}{60} = \frac{\pi(0.85)(5000)}{60} = 223 \text{ m/s}$$

From the inlet velocity triangle (Fig. 5.13),

$$\tan \alpha_1 = \frac{C_{w1}}{C_a} \quad \text{and} \quad \tan \beta_1 = \frac{(U - C_{w1})}{C_a}$$

Adding the above two equations:

$$\frac{U}{C_a} = \tan \alpha_1 + \tan \beta_1$$

or:

$$U = C_a(\tan 28^\circ + \tan 56^\circ) = C_a(2.0146)$$

Therefore, $C_a = 123.6 \text{ m/s}$ (constant at all radii)

2. The mass flow rate:

$$\begin{aligned} \dot{m} &= \pi(r_t^2 - r_h^2)\rho C_a \\ &= \pi(0.475^2 - 0.425^2)(1.2)(123.6) = 20.98 \text{ kg/s} \end{aligned}$$

3. The power required per unit kg for compression is:

$$\begin{aligned} W_c &= \lambda U C_a (\tan \beta_1 - \tan \beta_2) \\ &= (1)(249)(123.6)(\tan 56^\circ - \tan 28^\circ)10^{-3} \\ &= (249)(123.6)(1.483 - 0.53) \\ &= 29.268 \text{ kJ/kg} \end{aligned}$$

EXAMPLE A multistage axial compressor is required for compressing air at 293 K, through a pressure ratio of 5 to 1. Each stage is to be 50% reaction and the mean blade speed 275 m/s, flow coefficient 0.5, and stage loading factor 0.3, are taken, for simplicity, as constant for all stages. Determine the flow angles and the number of stages required if the stage efficiency is 88.8%. Take $C_p = 1.005 \text{ kJ/(kg}^\circ\text{C)}$ and $\gamma = 1.4$ for air.

Solution.

the stage load factor can be written as

$$\psi = \phi(\tan \beta_1 - \tan \beta_2).$$

the reaction is

$$R = \frac{\phi}{2}(\tan \beta_1 + \tan \beta_2).$$

Solving for $\tan \beta_1$ and $\tan \beta_2$ gives

$$\tan \beta_1 = (R + \psi/2)/\phi \text{ and } \tan \beta_2 = (R - \psi/2)/\phi.$$

Calculating β_1 and β_2 and observing for $R = 0.5$ that the velocity diagram is symmetrical,

$$\beta_1 = \alpha_2 = 52.45^\circ \text{ and } \beta_2 = \alpha_1 = 35^\circ.$$

Writing the stage load factor as $\psi = C_p \Delta T_0 / U^2$, then the stage stagnation temperature rise is

$$\Delta T_0 = \psi U^2 / C_p = 0.3 \times 275^2 / 1005 = 22.5^\circ\text{C}.$$

It is reasonable to take the stage efficiency as equal to the polytropic efficiency since the stage temperature rise of an axial compressor is small. Denoting compressor inlet and outlet conditions by subscripts I and II respectively then,

$$\frac{T_{0II}}{T_{0I}} = 1 + \frac{N \Delta T_0}{T_{0I}} = \left(\frac{p_{0II}}{p_{0I}} \right)^{(\gamma-1)/\eta_p \gamma},$$

where N is the required number of stages. Thus

$$N = \frac{T_{0I}}{\Delta T_0} \left[\left(\frac{p_{0II}}{p_{0I}} \right)^{(\gamma-1)/\eta_p \gamma} - 1 \right] = \frac{293}{22.5} [5^{1/3.11} - 1] = 8.86.$$

A suitable number of stages is therefore 9.

The overall efficiency is

$$\begin{aligned} \eta_{tt} &= \left[\left(\frac{p_{0II}}{p_{0I}} \right)^{(\gamma-1)/\gamma} - 1 \right] / \left[\left(\frac{p_{0II}}{p_{0I}} \right)^{(\gamma-1)/\eta_p \gamma} - 1 \right] \\ &= [5^{1/3.5} - 1] / [5^{1/3.11} - 1] = 86.3\%. \end{aligned}$$

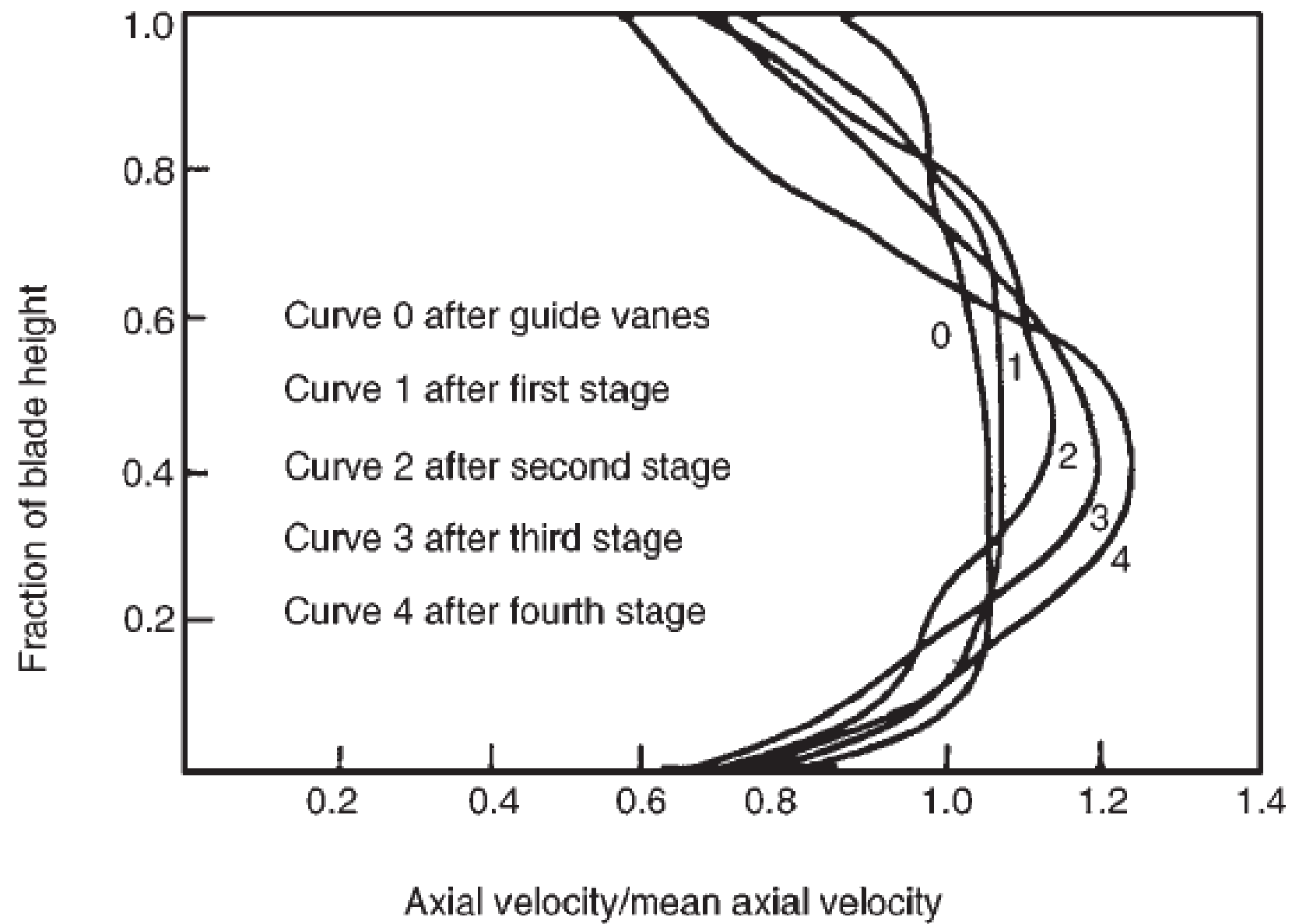


FIG. 5.7. Axial velocity profiles in a compressor (Howell 1945). (Courtesy of the Institution of Mechanical Engineers.)