Lecture 11
Compressors

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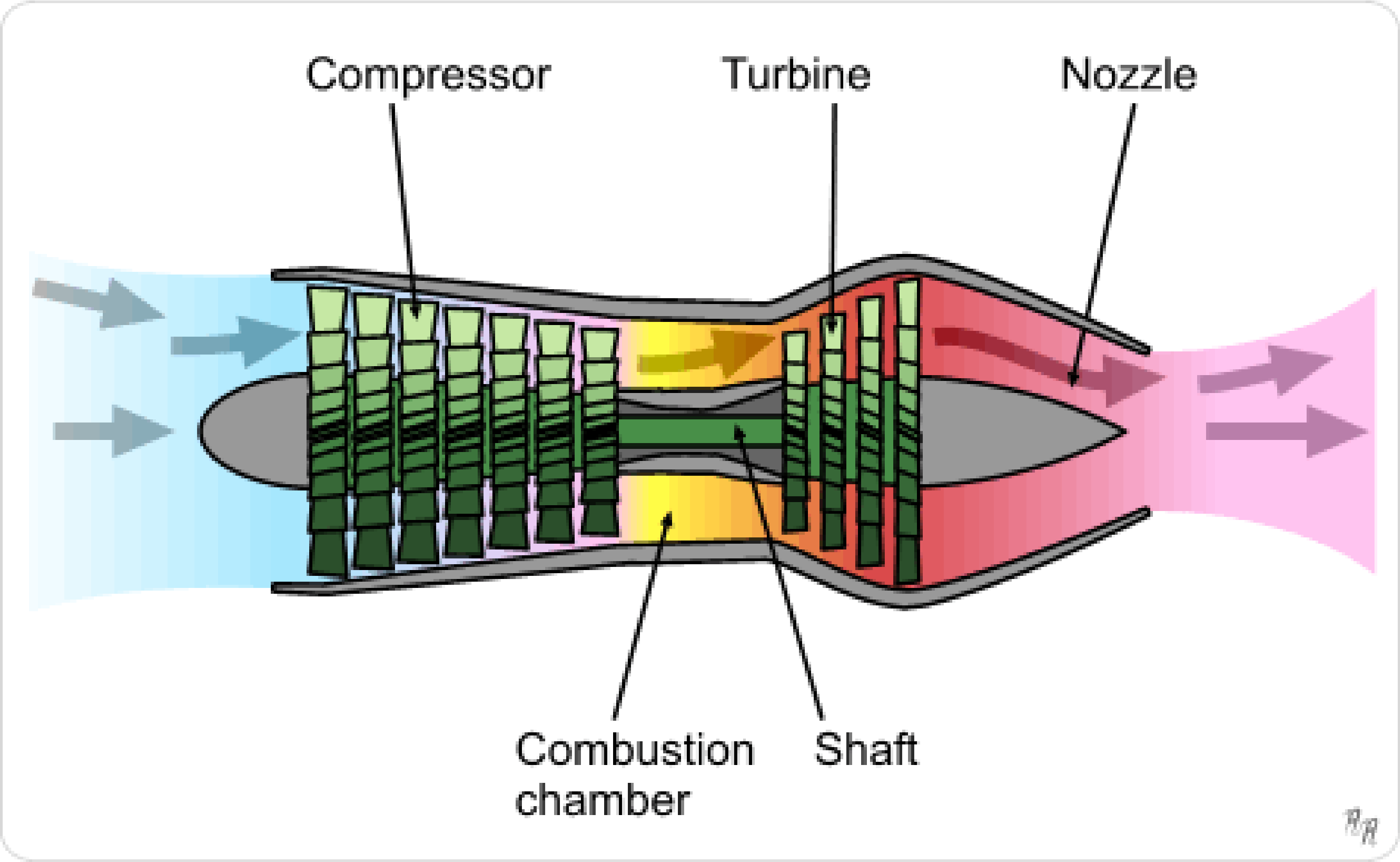
Turbojet with Axial Compresor



Turbomachinery – Axial Compressors

Centrifugal compressors are not commonly employed in turbojets!

Typical axial compressor designs are multistage

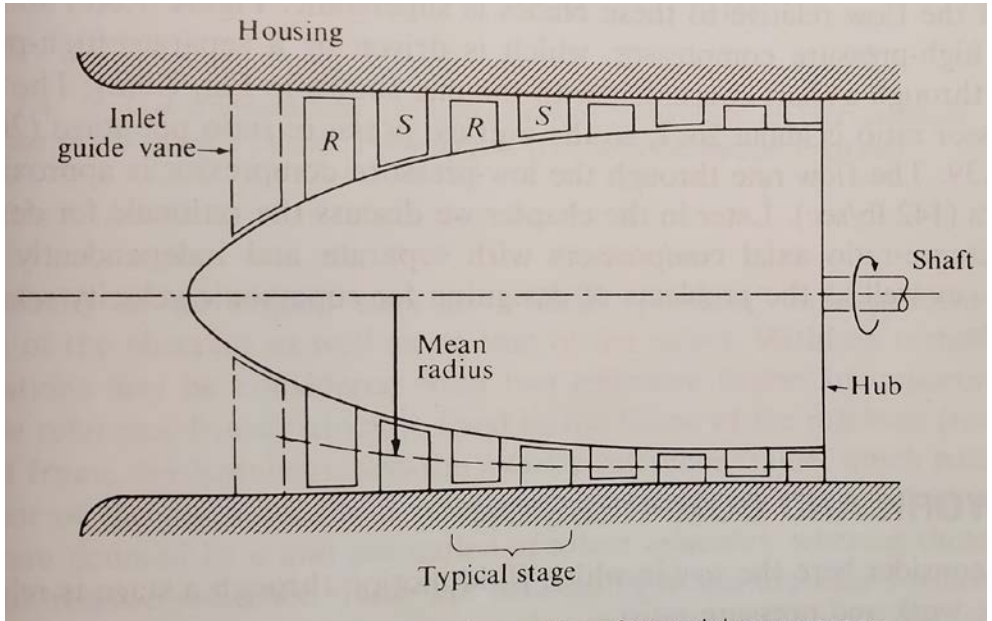


FIGURE 7.4 Schematic of a section of an axial compressor.

Axial compressor section from Hill & Peterson [1]

R: Rotor or rotating blade connected to the hub

S: Stator or stationary blade connected to the housing

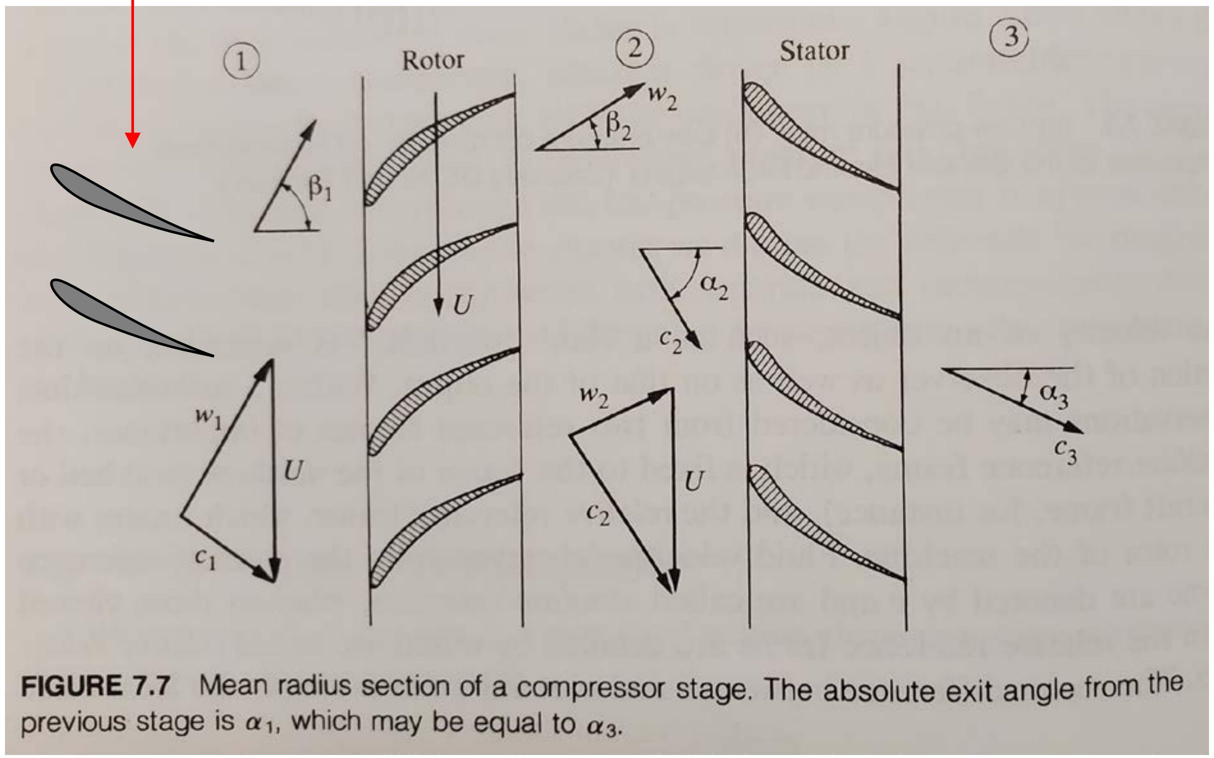
One pair of rotor blades and stator blades/vanes makes up a single stage. The flow first passes through the rotor, then passes through the stator.

Note that the cross-sectional area decreases as temperature, pressure, and density increase. There is not substantial acceleration of flow through the compressor.

Note that axial compressors can be made with: 1) constant housing diameter, 2) constant hub diameter, 3) constant blade length, or 4) all three properties varying from stage to stage!

Single Compressor Stage

There may be inlet guide vanes at the beginning or previous stage stator



Compressor stage from Hill & Peterson [1]

c_1 has typically been turned by previous stator or inlet guide vanes. Without inlet guide vanes, c_1 has not been turned for the first stage.

Velocities are tangential or azimuthal in ↕ direction and axial in ↔ direction

This is a cross-section at constant radius, for example, with a cylindrical surface "rolled" into a plane. We see the cross-section of each blade.

Single Compressor Stage

$\vec{U}$ = rotor velocity (tangential) at the particular radius
(The radial velocity component is not portrayed in the graphic)

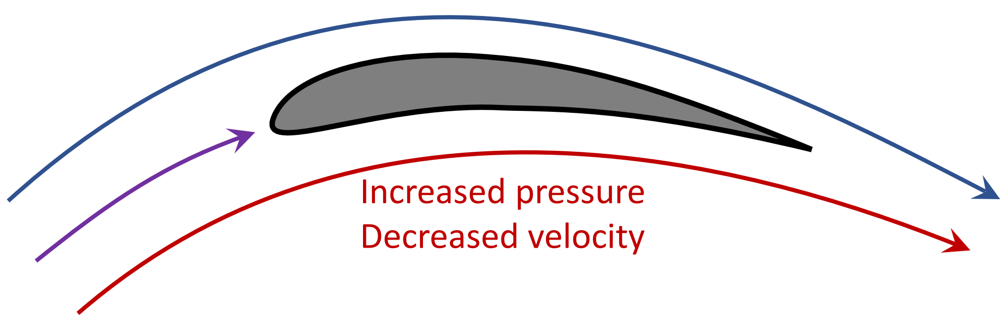
$\vec{c}$ = velocity relative to the housing and stator (tangential and axial component)

$\vec{w}$ = velocity relative to the rotor (tangential and axial component)
$\vec{w} = \vec{c} - \vec{U}$

$\vec{c} = \vec{U} + \vec{w}$ Note: $\vec{U}$ is only tangential but $\vec{c}$ and $\vec{w}$ may have other components!

Each blade is an airfoil that is cambered so that it turns the flow. It has high pressure on the concave surface (side) and low pressure over the convex surface (side). Our vector represents an average value in the passage between two successive blades.

Single Compressor Stage



Lift is produced by the pressure differences on the two sides of the airfoil

In our case, the lift force times the tangential velocity is a power term – the airfoil is doing work on the flow

A torque τ is required to change the angular momentum of the flow through the rotor. The change in angular momentum per unit mass is:

$$r\left(c_{2_t} - c_{1_t}\right) = r(w_{2_t} - w_{1_t})$$ Where r is the radial position

Consider $d\dot{m}(r)$ to be the mass flow rate through the annular cross-section at radius r of thickness dr. Then, the rate of change of angular momentum of that mass flow rate is:

$$d\tau = (d\dot{m})r\left(c_{2_t} - c_{1_t}\right)$$ Which is torque

Single Compressor Stage

Torque $d\tau$ time angular velocity Ω gives power $\boldsymbol{P}$

$$d\boldsymbol{P} = (d\dot{m})r\left(c_{2_t} - c_{1_t}\right)\Omega$$

Since $U = \Omega r$, $d\boldsymbol{P} = (d\dot{m})U\left(c_{2_t} - c_{1_t}\right)$

Using average values for U, c_1, c_2 over all radii of interest, we obtain the total power for the rotor blade:

$$\boldsymbol{P} = \dot{m}U\left(c_{2_t} - c_{1_t}\right) = \dot{m}U\left(w_{2_t} - w_{1_t}\right)$$

The work per unit mass $\boldsymbol{P}/\dot{m} = U\left(c_{2_t} - c_{1_t}\right)$

Also: $\boldsymbol{P}/\dot{m} = h_2^\circ - h_1^\circ$ (assuming adiabatic flow and determined in the non-rotating frame)

So: $h_2^\circ - h_1^\circ = U\left(c_{2_t} - c_{1_t}\right)$

Single Compressor Stage

The flow through the stator may be assumed to be adiabatic – also no work is done

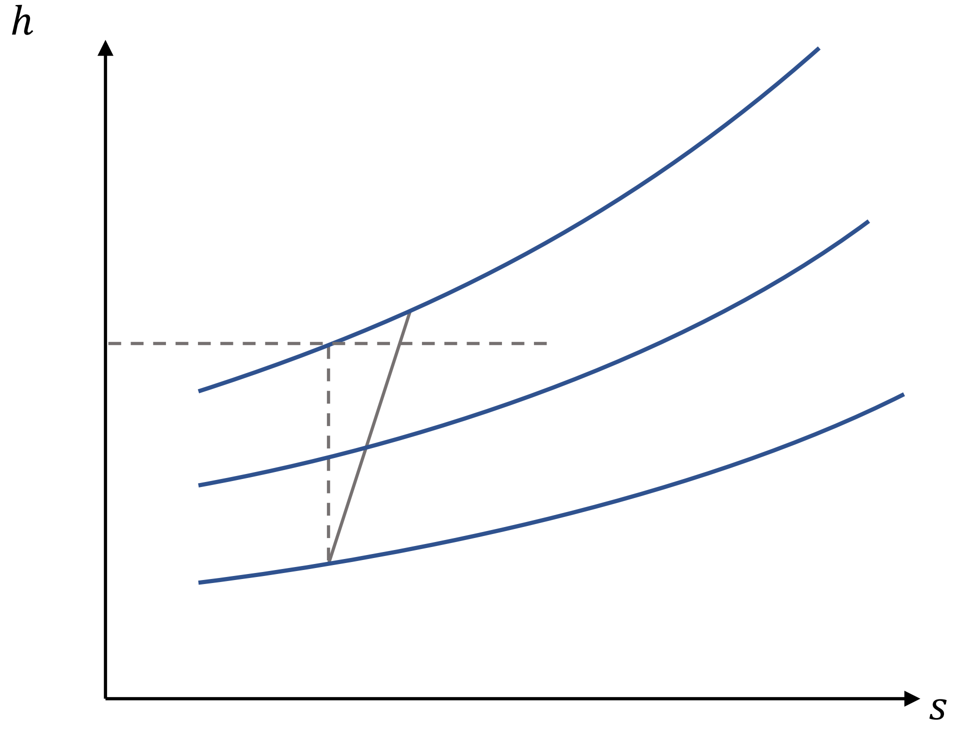
$h_3^\circ = h_2^\circ$ Stagnation values here are first determined in a non-rotating frame of reference, but then add $U^2/2$

So: $U\left(c_{2_t} - c_{1_t}\right) = h_3^\circ - h_1^\circ$ Is the total change across the stage

It follows that a large turning of flow through the rotor leads to a large work output – there will be limitations on the amount of turning that is possible, however!

Single Compressor Stage

Pressure is increasing across the stage. The entropy may be increasing as well!



Since pressure is increasing, we have an adverse (positive) pressure gradient similar to a diffuser. Separation can occur if the pressure increase is too great!

Single Compressor Stage

Typically, the design rule is that $C_p < 0.6$

Where $$C_p = \frac{P_2 - P_1}{\frac{1}{2}\rho_1 w_1^2}$$

This applies strictly for $M \ll 1$

Via Bernoulli's equation: $$P_2 - P_1 = \frac{1}{2}\rho\left(w_1^2 - w_2^2\right)$$ (Assuming $\rho_2 \approx \rho_1$ here)

So that: $$C_p = \left(1 - \left(\frac{w_2}{w_1}\right)^2\right) \leq 0.6$$

This limits the reduction in the magnitude of the velocity or, effectively, the amount of the turning!

There are also Mach number limitations because shock losses can occur, especially near the tips!

Single Compressor Stage

$$C_p = \frac{P_2 - P_1}{\frac{1}{2}\rho_1 w_1^2} \quad \Rightarrow \quad \frac{P_2}{P_1} = 1 + \frac{C_p}{2}\frac{\rho_1 w_1^2}{P_1}$$

$$\frac{P_2}{P_1} = 1 + \frac{C_p}{2}\gamma\frac{w_1^2}{a_1^2} = 1 + \frac{C_p}{2}\gamma M_1^2$$

If we limit $M_1 \leq 0.8$ to avoid compressibility and shock problems, $C_p \leq 0.6$ and $\gamma = 1.4$, then:

$$\frac{P_2}{P_1} \leq 1 + \frac{(0.6)}{2}(1.4)(0.8)^2 = 1.27$$

If we also limit $\frac{P_3}{P_2} \leq 1.27$

Then: $\frac{P_3}{P_1} \leq 1.6$

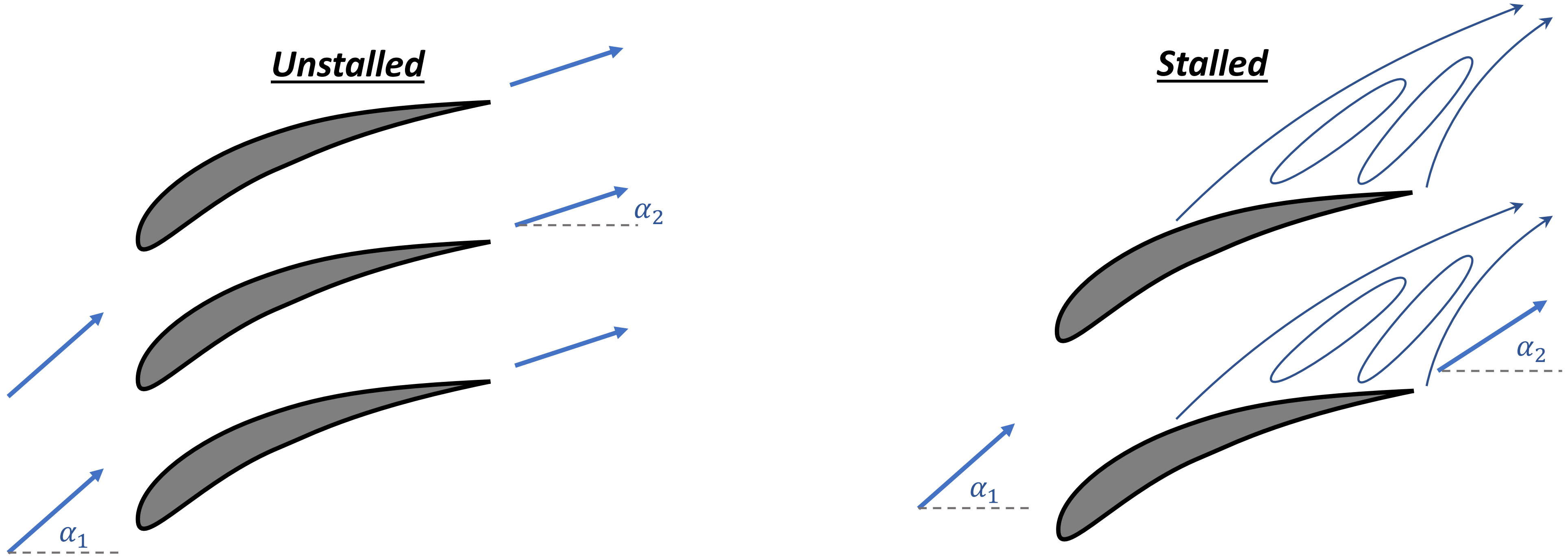
Single Compressor Stage

Most compressor stages will operate well below this limit!

Note that turbine stages only go as high as 3 because pressure is decreasing – no adverse pressure gradient. Typically, there are more stages in a compressor than a turbine.

Stall

Stall: Separated flow results in "stall" which has stagnation pressure losses and increased drag on the blades!

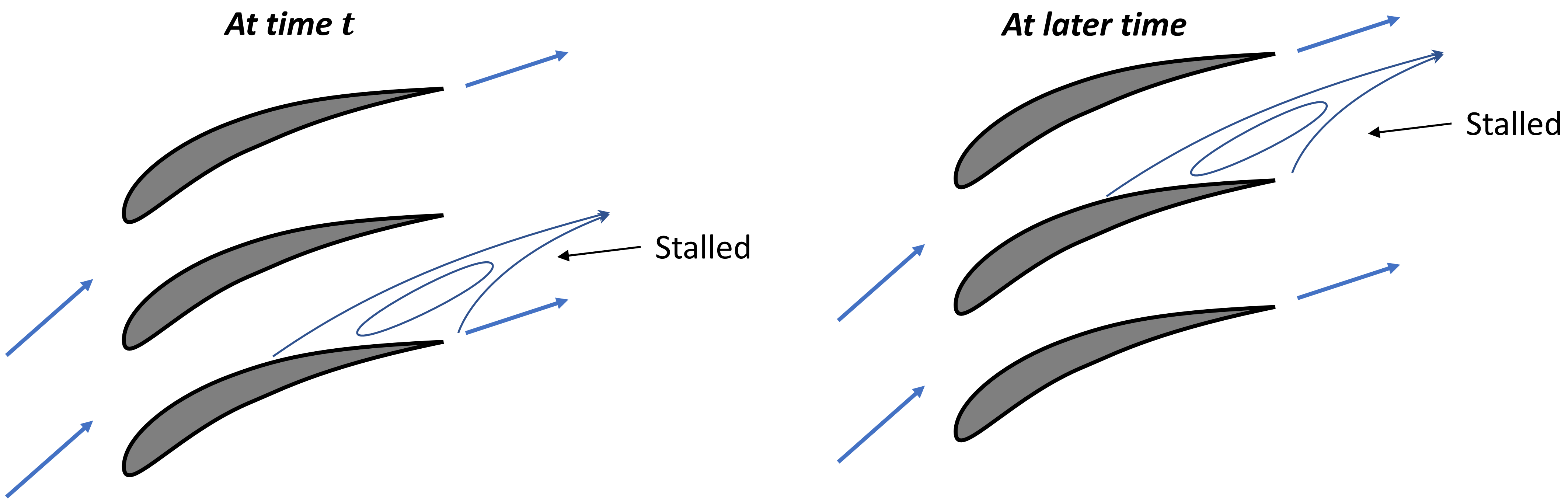


Unstalled: Streamlines follow the blade contours. Within a certain range, α_2 is insensitive to increases in α_1

Stalled: The turbulent mixing in the wake implies large stagnation pressure loss and large drag. More work and less pressure increase! For large α_1, the exit angle α_2 does become sensitive to increases in α_1, resulting in stall!

Stall

Rotating Stall: The stall usually occurs on a particular blade first. The separated flow causes some blockage of the passage forming a greater fraction of the incoming flow to turn up to the next blade passage.



Stall propagates or rotates

Starting

After compressor is operating, density increases substantially with flow direction. Before steady operation, low density exists through compressor. In order to maintain desired value of mass flow, velocity increases too much in later stages or the mass flow much be at reduced levels.

Generally a combination occurs and axial velocity is low in the first stages and higher than design in the latter stages during starting.

Starting

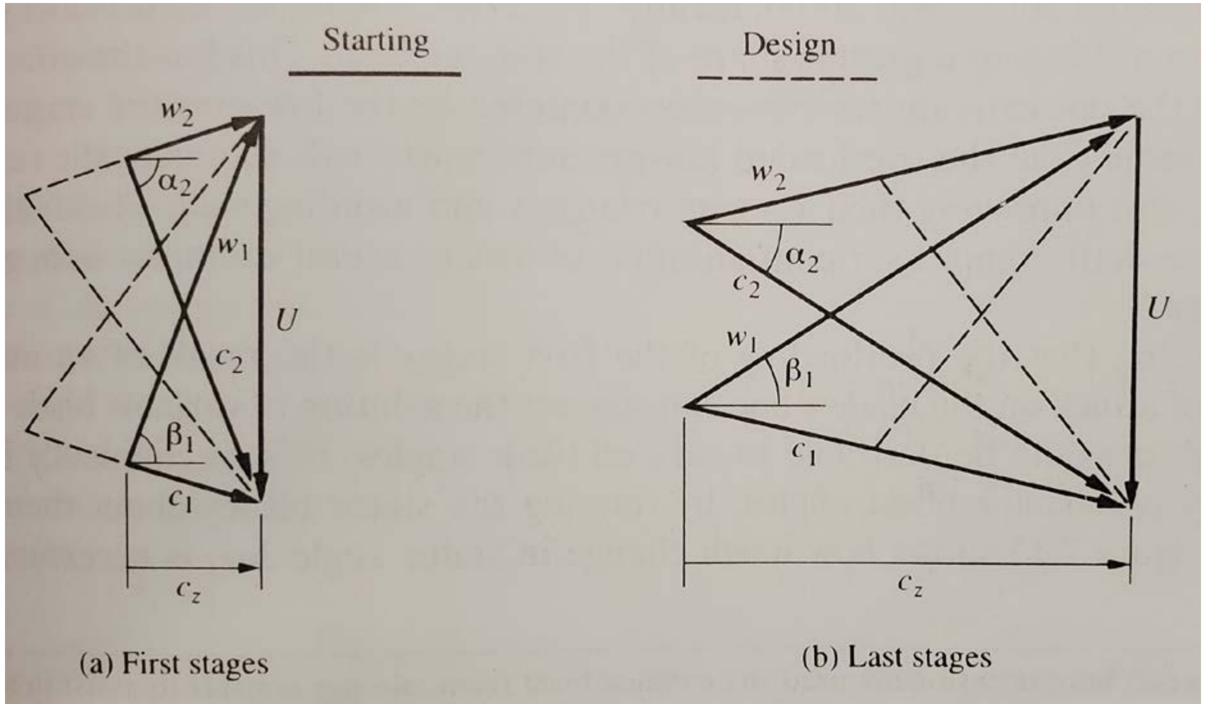


FIGURE 7.12 Effect of axial velocity change.

Effects of axial velocity change from Hill & Peterson [1]

First Stages: Component of velocity in the axial direction is reduced at positions 1 & 2. Reduced velocity compared to design – the angle is greater, which means separation can occur. Also the load on the stages is increased.

Last Stages: Increased velocity compared to design angle of incoming flow is reduced, decreasing the load on the last stages of blades.

Possible Corrections:

1. Release some air midway bypassing the last stages of compression.
2. Have two parts of compression driven at different speeds by different turbines. The low pressure stage runs slower, while the higher pressure stage runs faster.
3. Variable blade angles on the stator blades.

References

[1] Hill, Philip G., and Carl R. Peterson. *Mechanics and Thermodynamics of Propulsion*. Reading, Mass: Addison-Wesley Longman, 1992.