TURBOMACHINERY AERODYNAMICS

Lect - 8

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Axial Flow Compressor

3-D Blade Design Laws

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From Radial Equilibrium Condition and using some simplifying flow conditions (constant H, C_a, ρ along the radius) we get :

$C_w \cdot r$ = constant.

This condition is commonly known as *Free Vortex Design Law*

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The simple Radial Equilibrium may be used to explain some of the basic characteristics of an axial compressor

- Radial equilibrium requires that in a medium (<1.0) to low ($\ll 1.0$) hub/tip radius ratio in a rotor blade, change of whirl component (ΔC_w or ΔV_w) must be very large near the hub (root) compared to that near the casing (tip)

- Radial equilibrium, thus, requires that flow turning at hub $\Delta\beta$ must be much larger at hub than at the tip.

- Hence hub airfoil must be of much higher camber than that of the tip airfoil

- Whirl component downstream of the rotor (C_{w2} or V_{w2}) is higher than the upstream

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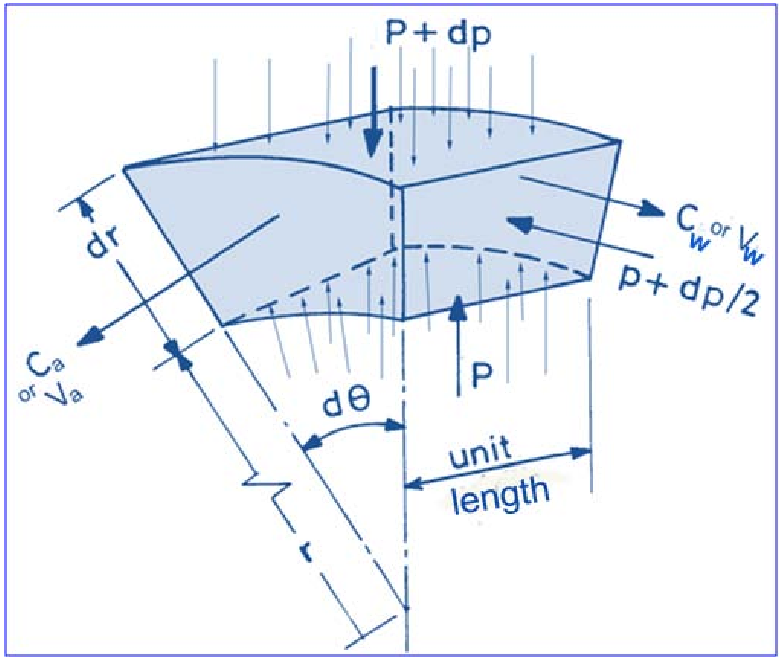
- The Radial static pressure gradient *dp/dr* will be greater downstream of the rotor than upstream



- Static pressure rise across the blade root will be lesser than that across the rotor tip

- Thus degree of reaction, R_x across the root will be much less compared to that at the tip

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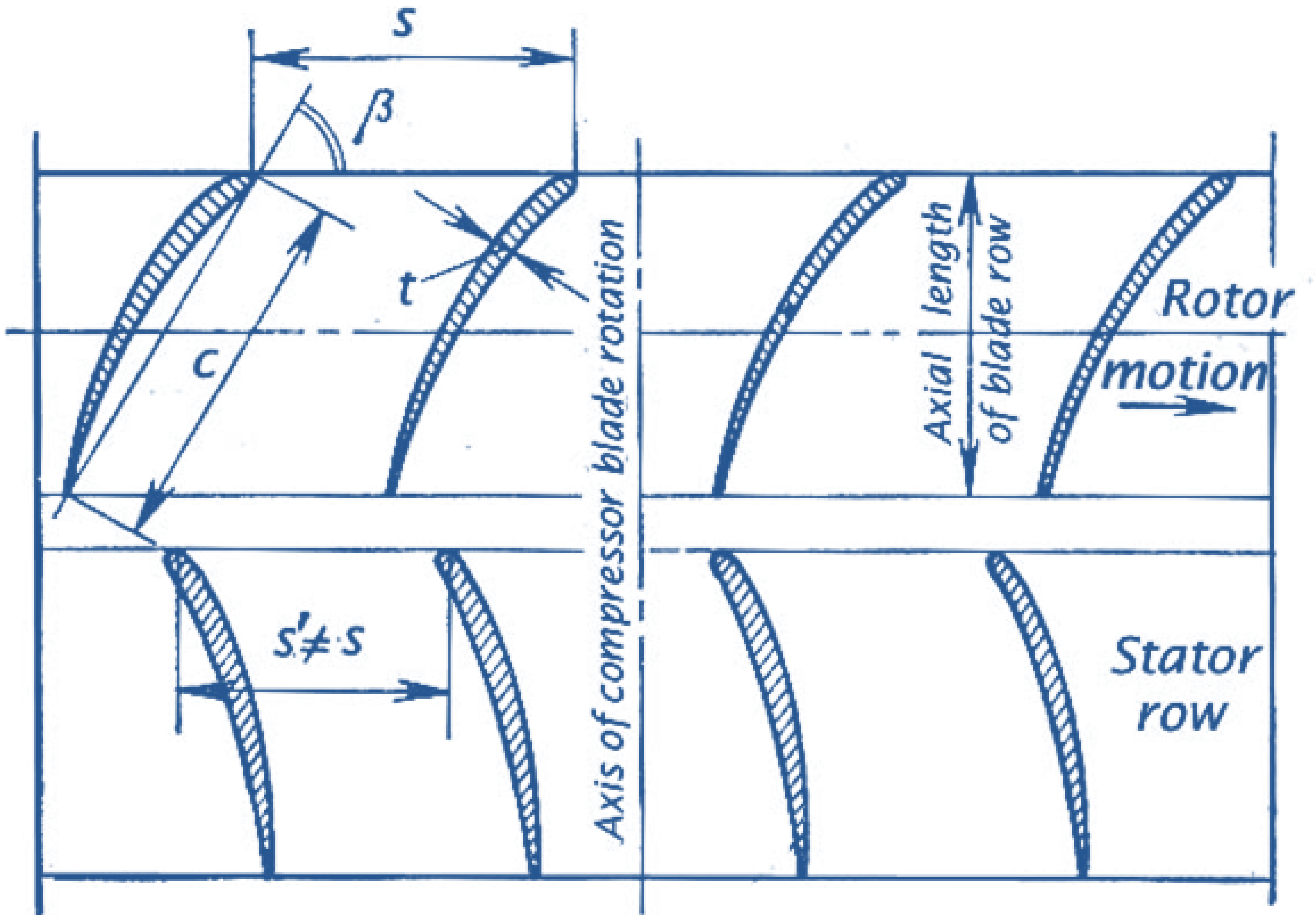
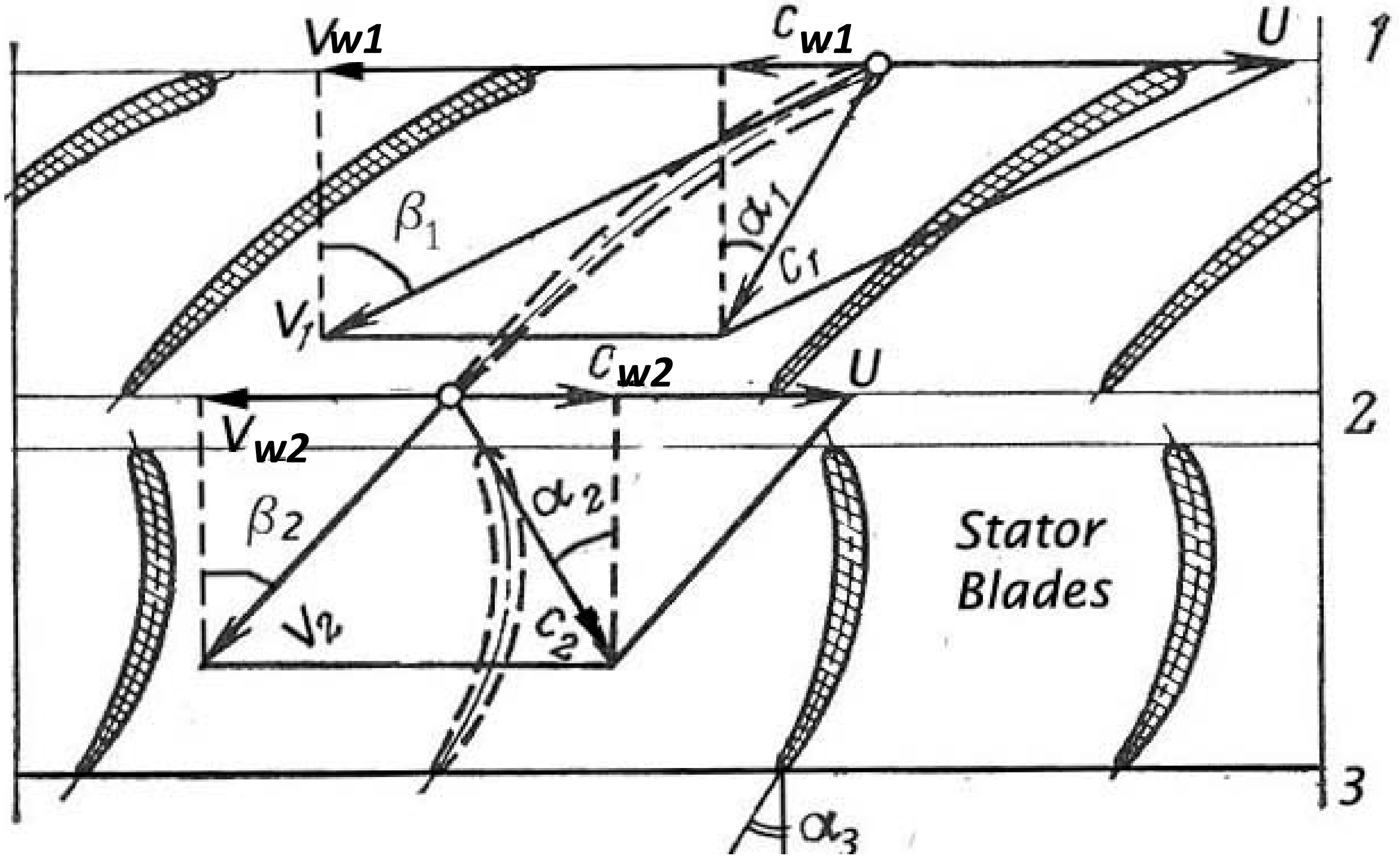


A 50% reaction stage blading



A high reaction stage bladings

- If one looks at a stage consisting of a rotor and a stator, the radial equilibrium would also impact the flow across the stator

- Stator blade rows reduce the whirl component

- Downstream of stator radial pressure gradient *dp/dr* will be much lower than upstream of the stator

- Static pressure rise, Δp across the stator at hub would be higher than at the tip. This may lead to high blade loading and even flow separation at stator hub

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In view of the simplifications and the constraints of the free vortex design law, a generalized vortex law may be written as

$$C_w \, . \, r^n = \text{constant}$$

Where, n =1 is gives us back the free vortex law.

Normally, –1> n >2.

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- when $0.75 < n \leq 1.0$ it yields *near-free vortex* or *relaxed-free-vortex* designs in which the blade sections are slightly overloaded with respect to free vortex blade loading.

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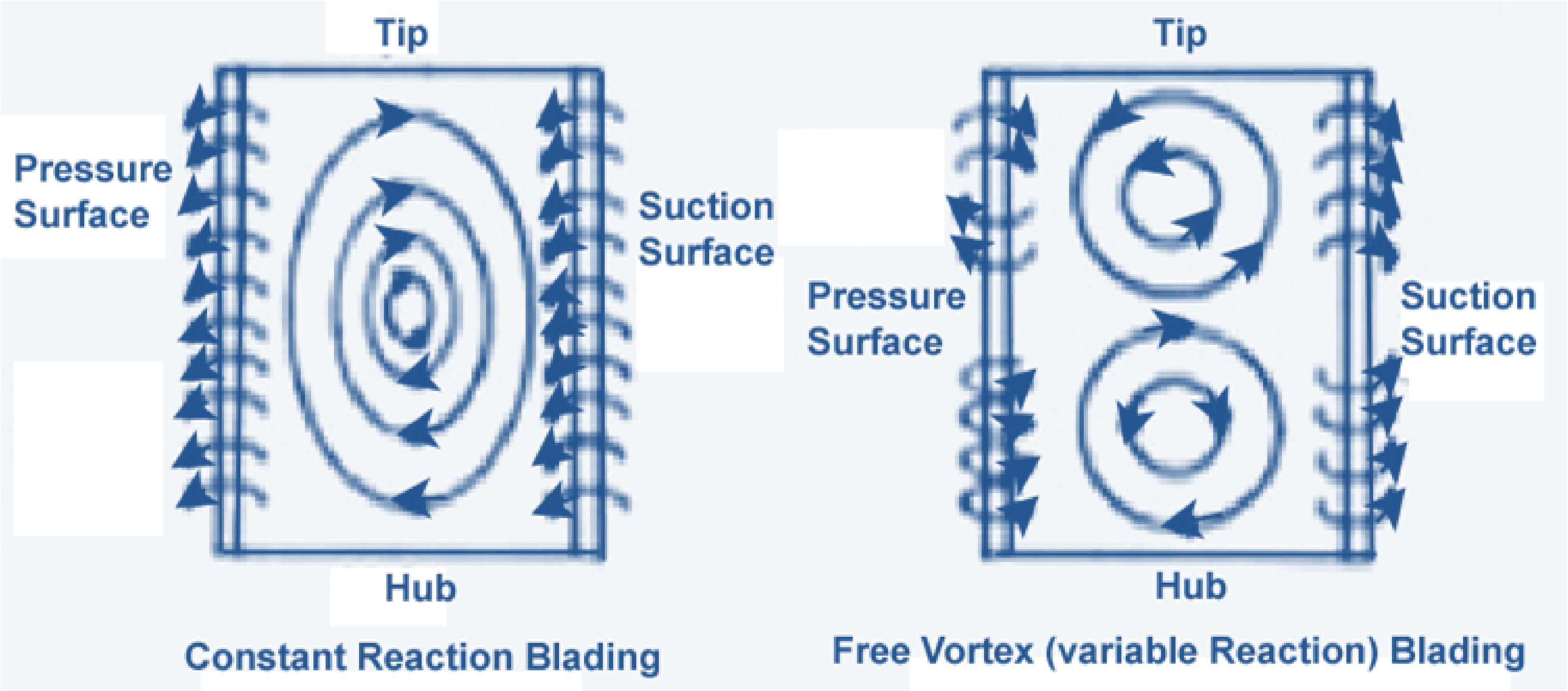
- when $n>1$ the blades are underloaded w.r.t. FVD law;
- $n = -1$ is rarely used - often known as the forced vortex design
- $n = 0$ is known as the Exponential design law and often is used to arrive at constant degree of reaction blade designs



The exact nature of the vortex formation depends of the blade design laws, blade geometry and the operating condition

Preliminary blade designs are also driven by the radial variation of the degree of reaction along the blade length

Three limiting possibilities are often started with:

Rx = 0%

Rx = 50%

Rx = 100%

50% reaction means diffusion and hence blade loading is equally shared by the rotor and stator

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- In case when the degree of reaction is much different from 50%, one of the blades, either rotor or stator is more loaded.

- The two limiting cases are 0% and 100% reaction split between the rotor and the stator.

- As we have seen the reaction may vary from the root to the tip (as in Free Vortex designs)

- Which means the split between the rotor and the stator vary from the root to the tip of a stage

- In Rx = 0% case entire diffusion happens in stator and rotor is used for flow energization. Such a rotor will not have any diffusion occurring in the rotor blade passage, and may be called an impulse rotor (as energy transfer happens due to flow turning). A supersonic rotor may have Rx=0%

- in Rx = 100% case entire diffusion happens in the rotor along with flow energization. The stator is used only for flow turning

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Compulsions of these choices are often present depending on whether one is designing a:

i) Small sized axial compressor
ii) Large sized axial fan (in a bypass turbofan)
iii) First stage of a multi-stage axial compressor
iv) Middle stage of a multi-stage compressor
v) End stage of a multi-stage compressor
vi) High hub/tip radius ratio stage
vii) Low hub/tip radius ratio stage

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Axial Distribution of the specific work (W_{th}) and efficiency (η_i) amongst the individual stages of a typical multi-stage compressor must be completed and are arrived at from early design choices :

	Initial Stages	Middle stages	Last stages
η	0.86	0.92	0.88
π	1.5-1.8	1.3-1.4	1.1-1.2
ΔT_0 °C	40-75	30-50	15-30

The radial distribution of these parameters are then taken up for each stage design

In the modern axial compressor designs the 3-D flow features inevitably present are:

- Radial variation of Mach number and Reynolds number
- Consequently radial variation of density and pressure gradients
- Radial variation of blade thickness due to Mach number variation
- Radial variation of work input
- Hub or casing geometry introducing radial flow
- Leakage at the tip and axial gaps
- Air bleed in an intermediate stage

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- Secondary flow development is a consequence of all the earlier points <u>and the operating point</u>
- Some non-uniformity in the inlet flow
- Combination of subsonic and supersonic flow
- Radial variation of all the design parameters is high for low hub/tip radius ratio stage, which have high aspect ratio (h/c >2.0)
- Radial variation of parameters is less in a high hub/tip ratio blade e.g. in the last stages, which have low aspect ratios (h/c<2.0)

More detailed discussion on Compressor Blade Design in a later lecture

Next Lecture ----

3-D flow in full mathematical form