TURBOMACHINERY AERODYNAMICS

Lect 34

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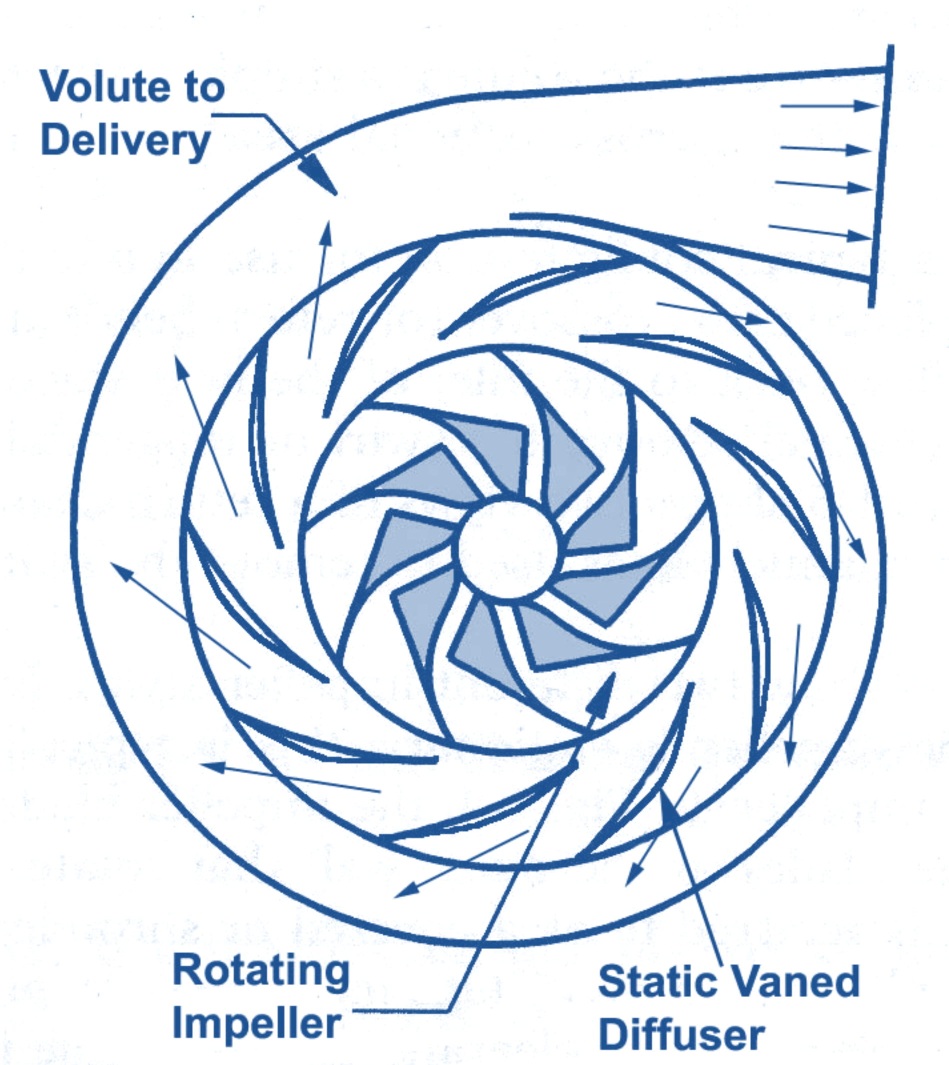
Centrifugal Compressors

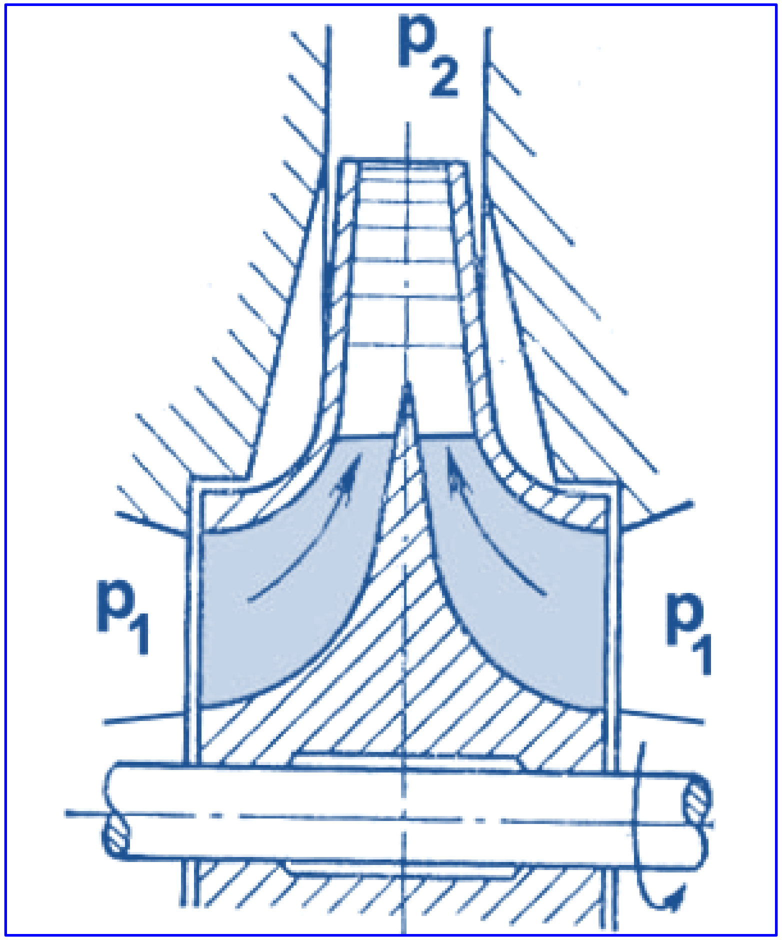
Design of Centrifugal Compressor elements – Impellers, Vanes etc.



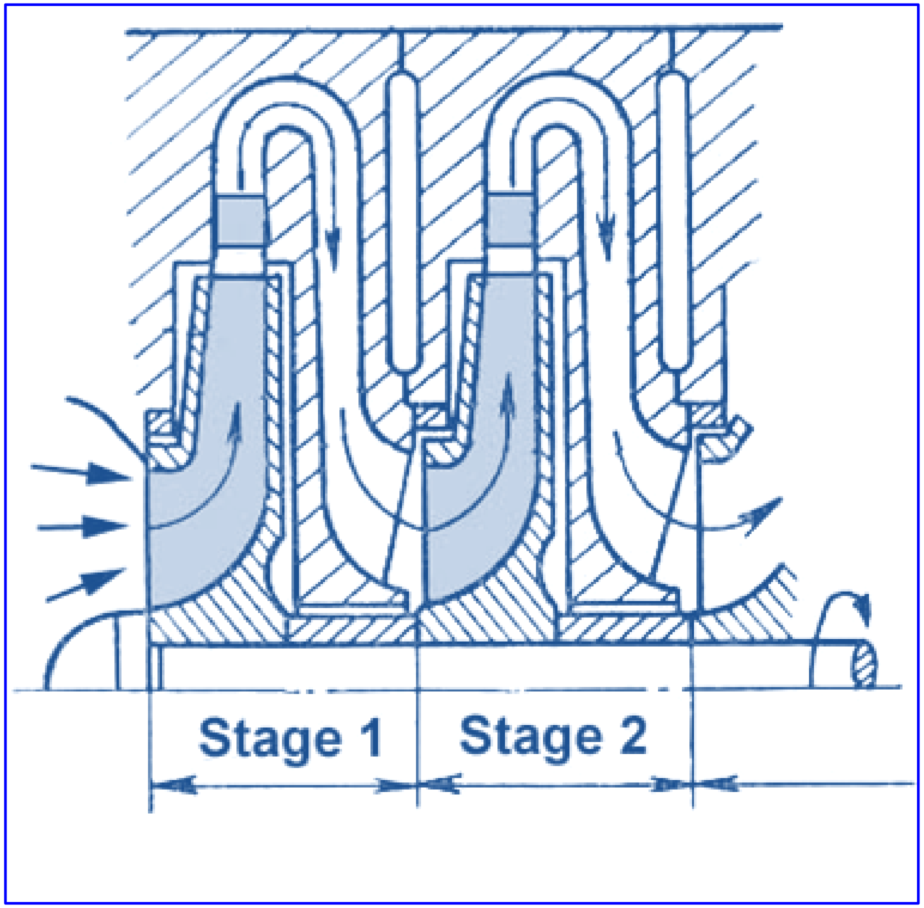
a) Impeller
b) Diffuser Vanes
c) Vaneless diffuser
e) Inlet Guide vanes
f) Volute

Other Design Possibilities:

a) Double-sided impeller :



b) Multi-staged compressor



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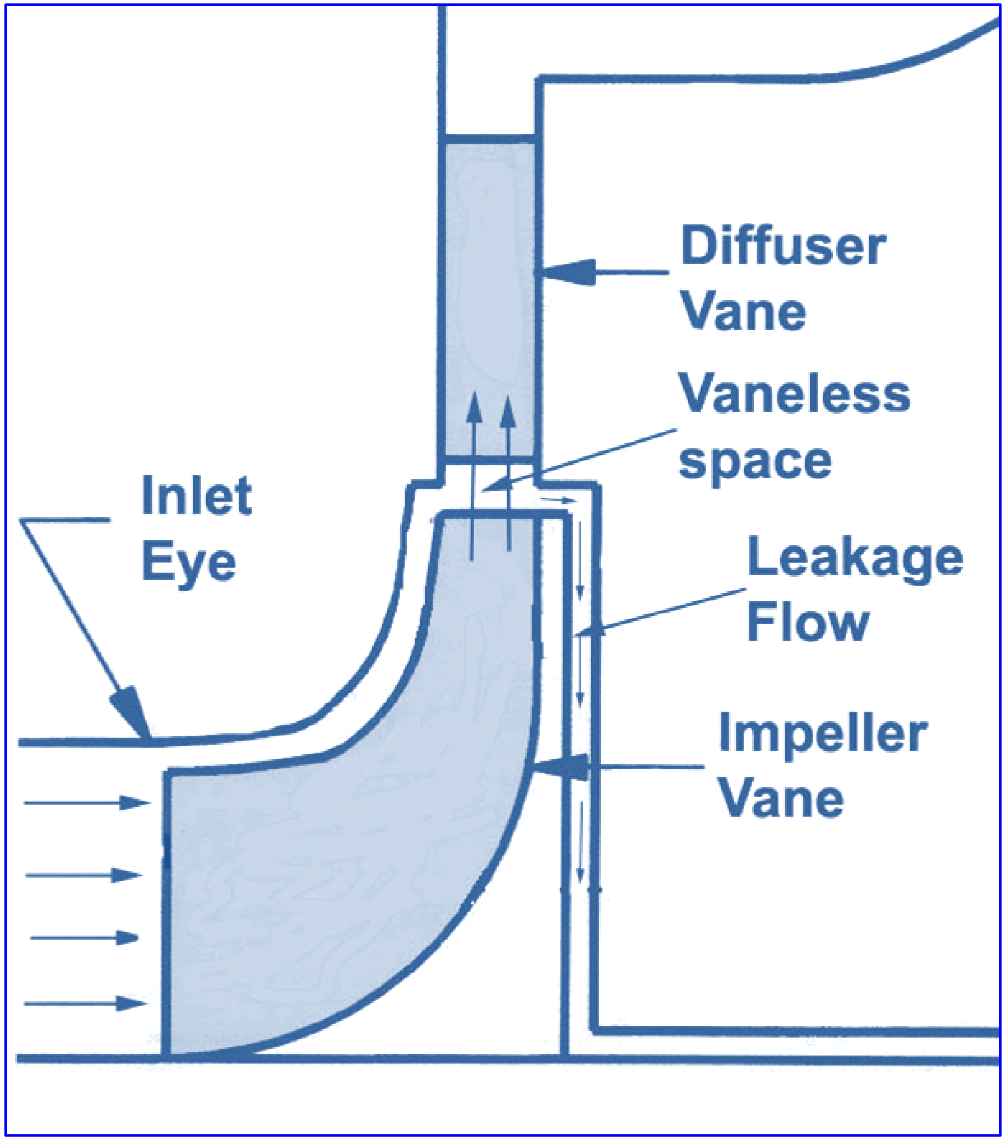

Other important issues to be designed

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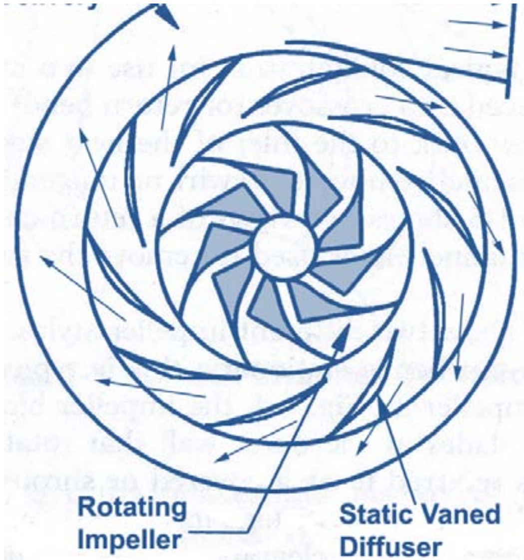
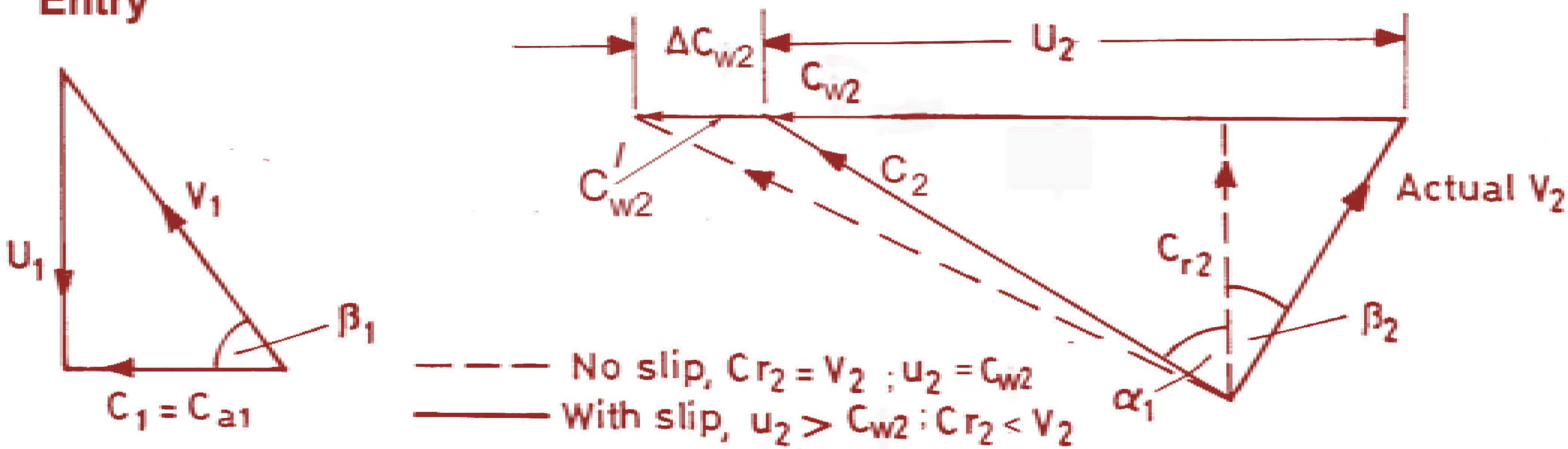


The velocity Diagrams

Impeller Entry

Impeller exit



In Axial Plane

In Radial Plane

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Slip factor

In a real compressor relative velocity vector V_2 is at angle β_2 because of non-radial exit from the impeller tip as the real viscous flow detaches near the tip from the impeller vane (trailing) surface

Stanitz formula, $$\sigma_s = 1 - \frac{0.63.\pi / N}{1 - \phi_2 . \tan\beta_2^*}$$

No dependence on backsweep

Where $\phi_2 = \frac{C_{r_2}}{U_2}$ & N = no. of blades

which, for a radial vane, $$\sigma_s = 1 - \frac{0.63.\pi}{N}$$

$\beta_2 < -45^0$; N > 8

Stodola Definition $$\sigma = 1 - \frac{(\pi/N)\cos\beta_2}{1 - (V_{r2}/U_2)\tan\beta_2}$$

$0^0 < \beta_2 < -60^0$

Wisner's definition $$\sigma = 1 - (\pi/N)\cos\beta_2$$

$\beta_2 > -45^0$; N > 20

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Forward Curved Vanes	Small Volume	High Pressure ratio	High speed High noise , Low Efficiency
Backward curved Vanes	Large Volume and size	Low to High Pr Ratio	High Efficiency, Low Noise
Radial Vanes	Medium Volume and Size	Medium to High Pr ratio	Good Efficiency
Radial Vaned CCs have been used in A/C engines for 50 years. Now, well designed backward curved vaned CCs are increasingly being used for higher efficiency.			

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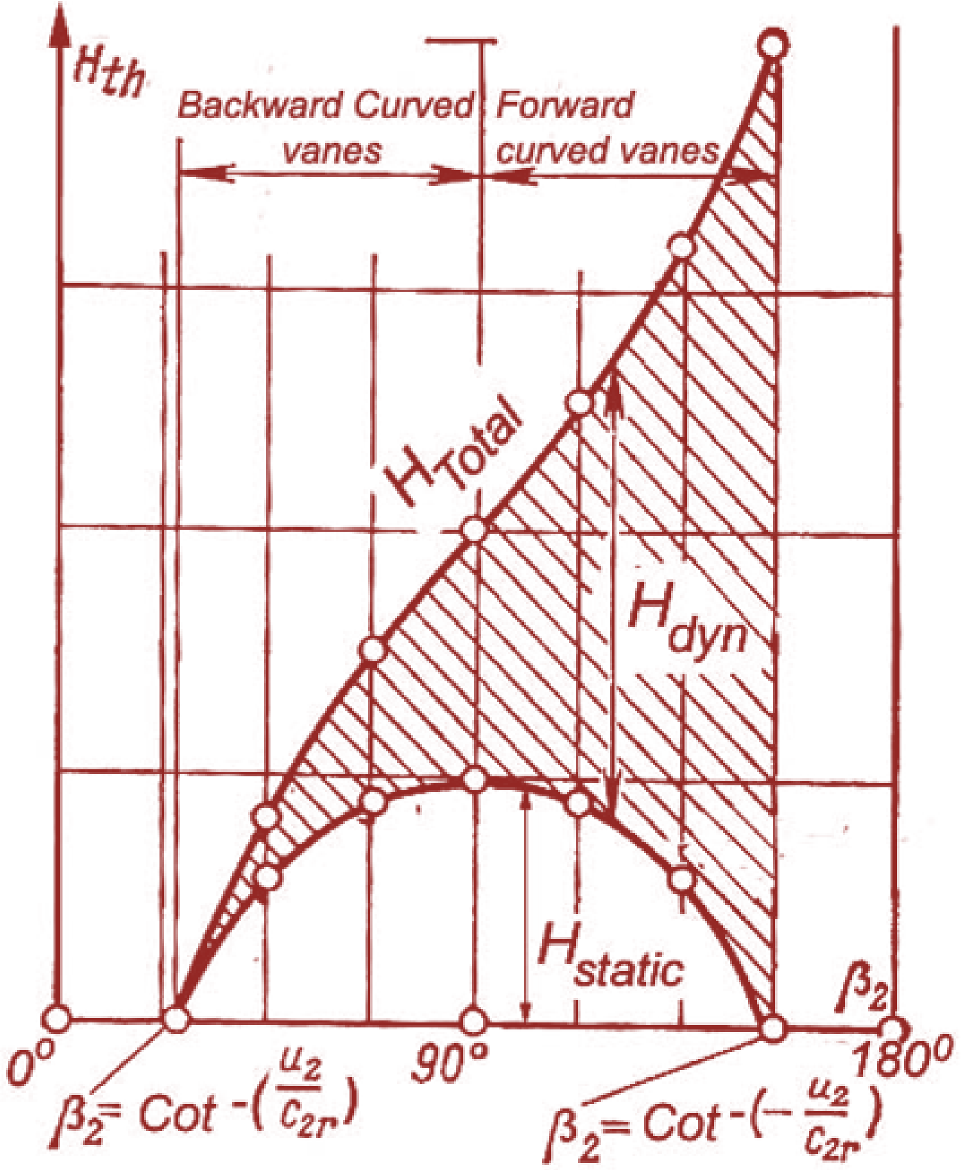
In highly forward and highly backward curved (β_2 >-60^0) impellers slip factor looses its meaning



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At the compr. entry face

$$\tan\beta_1 = \frac{C_{a1}}{U_1}$$

$U_1 = \omega.r_{eye}$ where r_{eye} varies from the root to the tip of the eye

Thus for a high speed compressor (or large sized) β_1 shall vary hugely from root to tip of the eye.

Under off-design operations, at any radius, incidence, i_r

$$i_r = \left(\beta_1 - \beta_1^*\right)_r$$

To be decided by designer

High positive incidence i (≥ +5^0) may precipitate early flow separation inside the impeller vane passage, even near the eye, specially if high diffusion (*i.e. high adverse pressure gradient*) is being attempted inside the impeller vanes.

At the <u>exit plane of the impeller</u>, the exiting flow deviates from the trailing edge and lag behind in rotational mode. This is often referred to as the *lag or deviation angle.*

[illegible]$_{av}$ [illegible]$_2$ [illegible]$_2^*$

which is an average at the passage exit, and β_2* is the impeller vane exit angle set by design

<u>Diffusion Limit :</u>

An upper limit of realistic diffusion limit $V_2/V_1 \approx 0.6$

In rotating diffuser $V_2/V_1 < 0.6$

In Impeller design, $\rho_1 A_1 / \rho_2 A_2 > 2.0$

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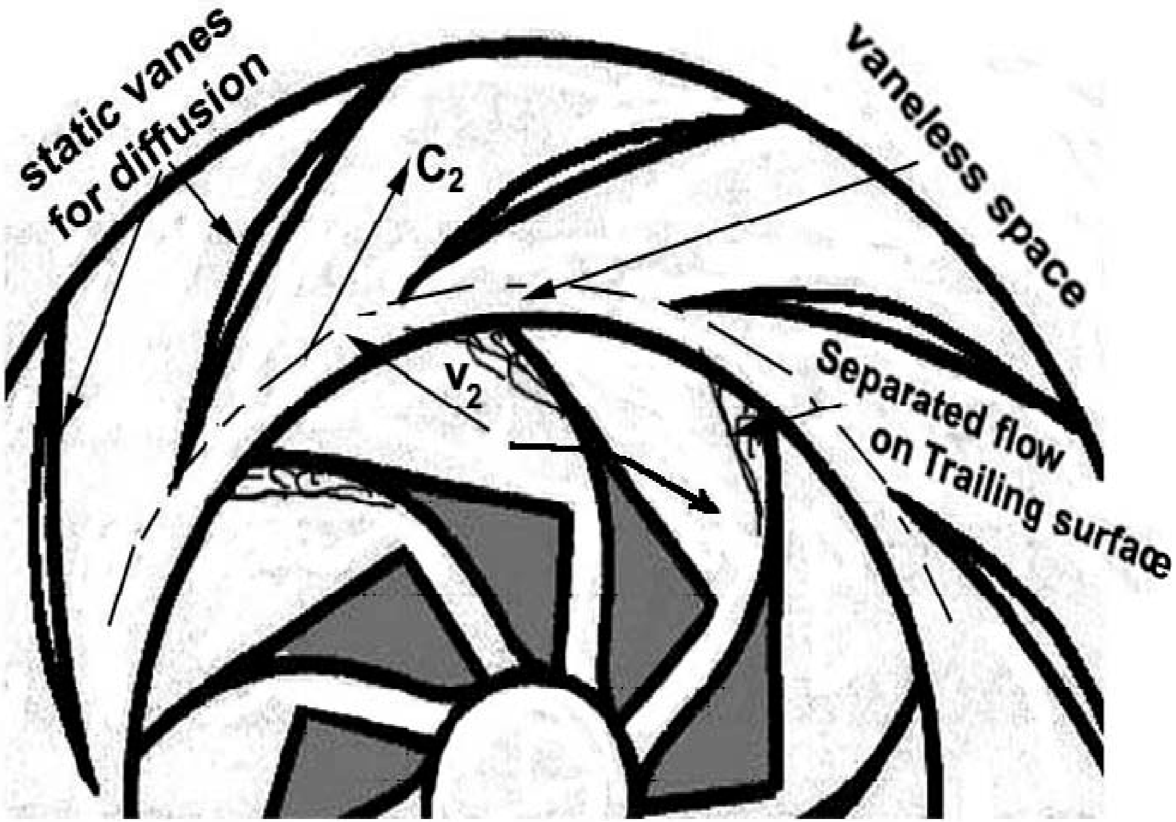
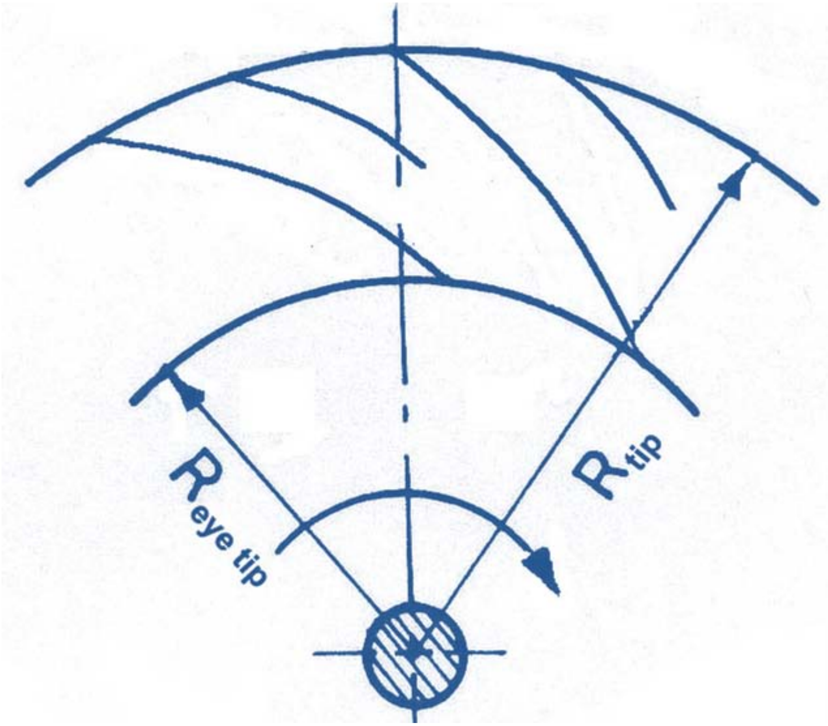
Design of the vaneless space

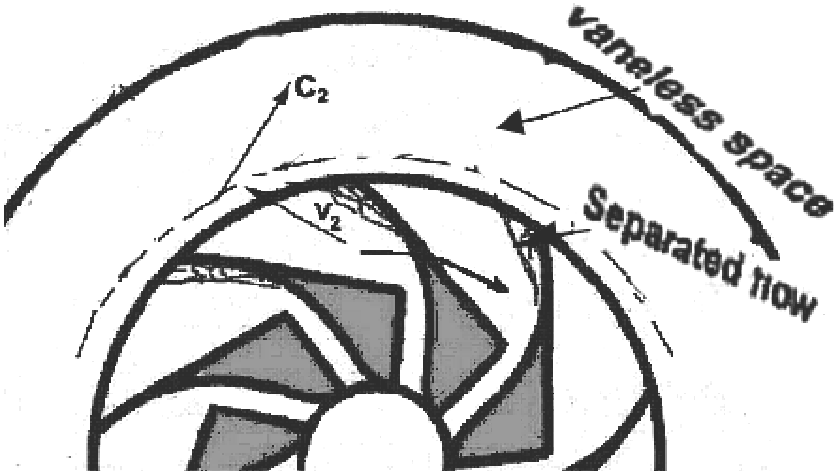


- Vaneless space is often used to decelerate impeller exit flow from supersonic to subsonic speed
- A completely vaneless diffuser is lighter , has broader mass flow operating range but has a lower efficiency

Reduction in deviation angle at the impeller exit under off-design operating conditions is to be designed in to the impeller and the vane designs.



Backward curved vanes + splitter



Vaneless diffuser

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The general relationship for Compressor Pressure ratio is given by

$$\pi_{0C} = \frac{p_{03}}{p_{01}} = \left[1 + \frac{\psi.(\sigma_s.U_2^2 - U_1.C_{w1}).\eta_{0c}.(\gamma-1)}{a_{01}^2}\right]^{\frac{\gamma}{\gamma-1}}$$

- **Theoretical energy density (H_{th}) transfer is highest with forward curved vanes, in which most of the energy would be available in kinetic form, H_{dyn} at the impeller exit.**
- **While a radial impeller gives almost 50-50 split of static (H_{static}) and dynamic heads (H_{dyn}) at the impeller exit, the backward curved vanes give high static pressure development in the impeller.**
- **Pre-swirl ($\alpha_1 > 0$) reduces the work done by compressor**

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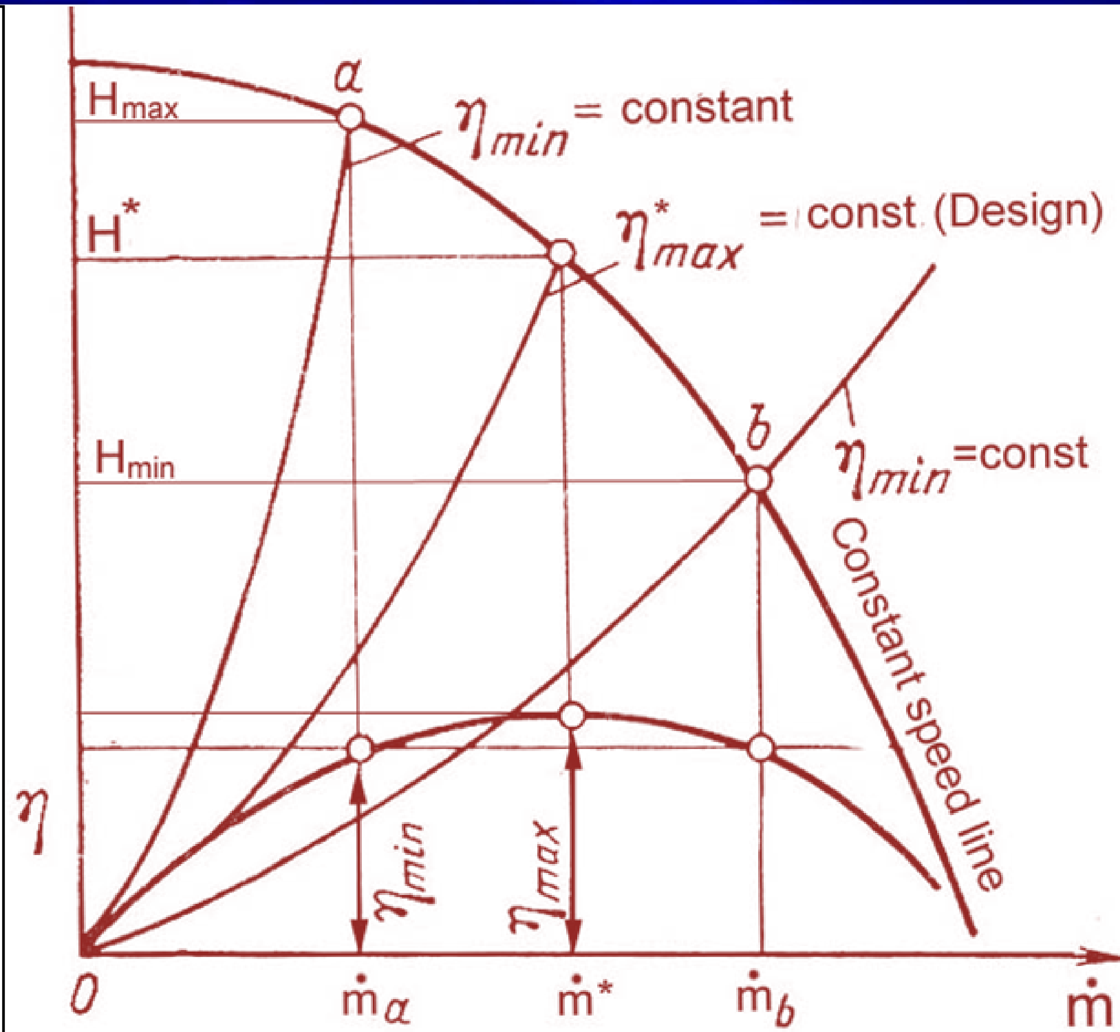
The theoretically obtained points to the right of ***b*** are considered choked, i.e. the compressor cannot process greater mass flows. The compressor is said to go in to stall at $\dot{m}_a$, this happens when high pressure rise is attempted at low mass flow



With Mass Flow Control only

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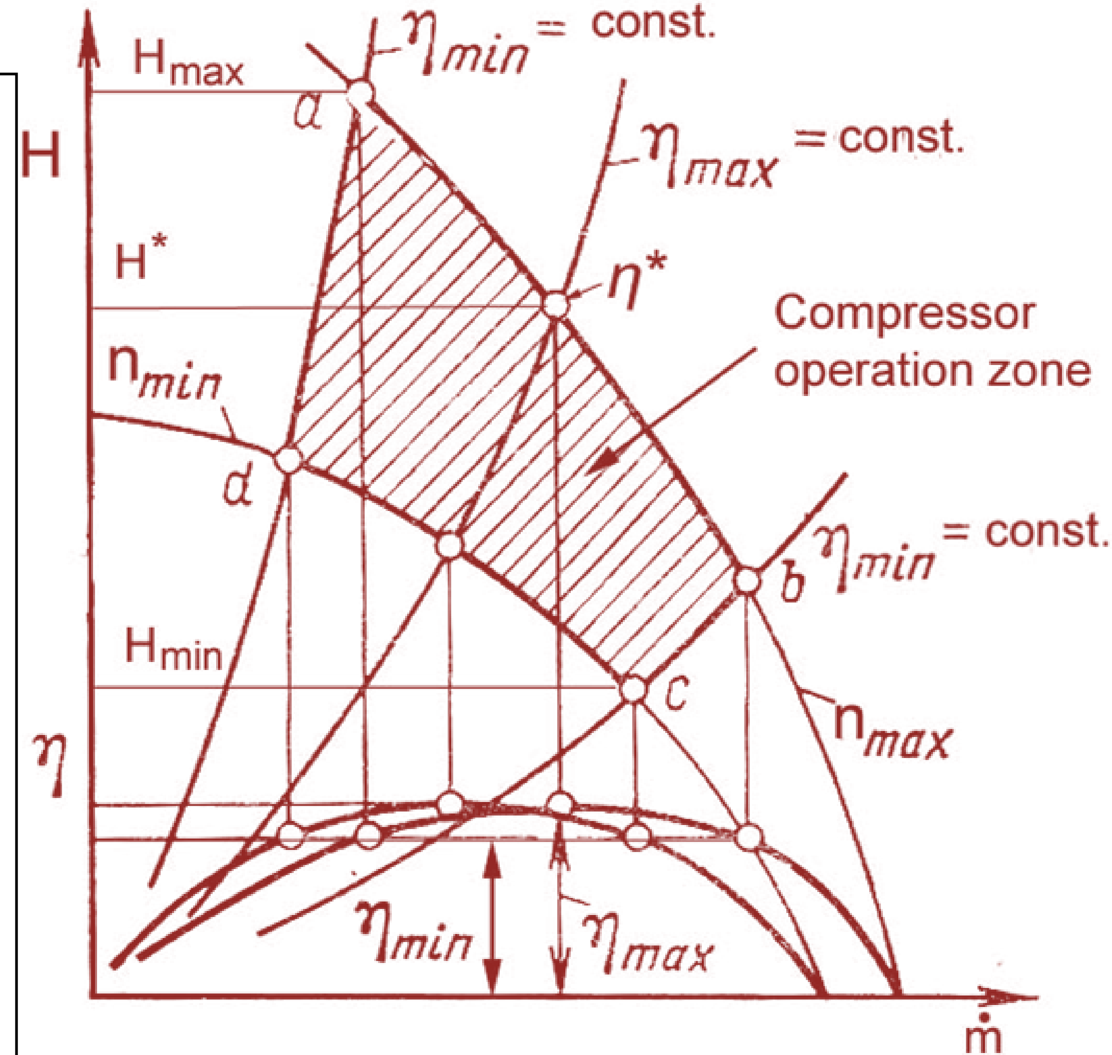
- In aircraft engines, rotating speed is variable during actual running.
- Thus the zone of operation is bounded between the points *a*,*b*,*c* and *d* .
- The η_{min} lines and the speed lines, n_{max} and n_{min}, define the boundaries (shaded area) of operation.



With Speed Control and Flow Control

- If more control variables are available it may be possible to extend the zone of operation of the compressor. All possible means of extending these boundaries further are being explored.
- Variable geometry (stagger) Inlet and exit (diffuser) guide vanes to be explored

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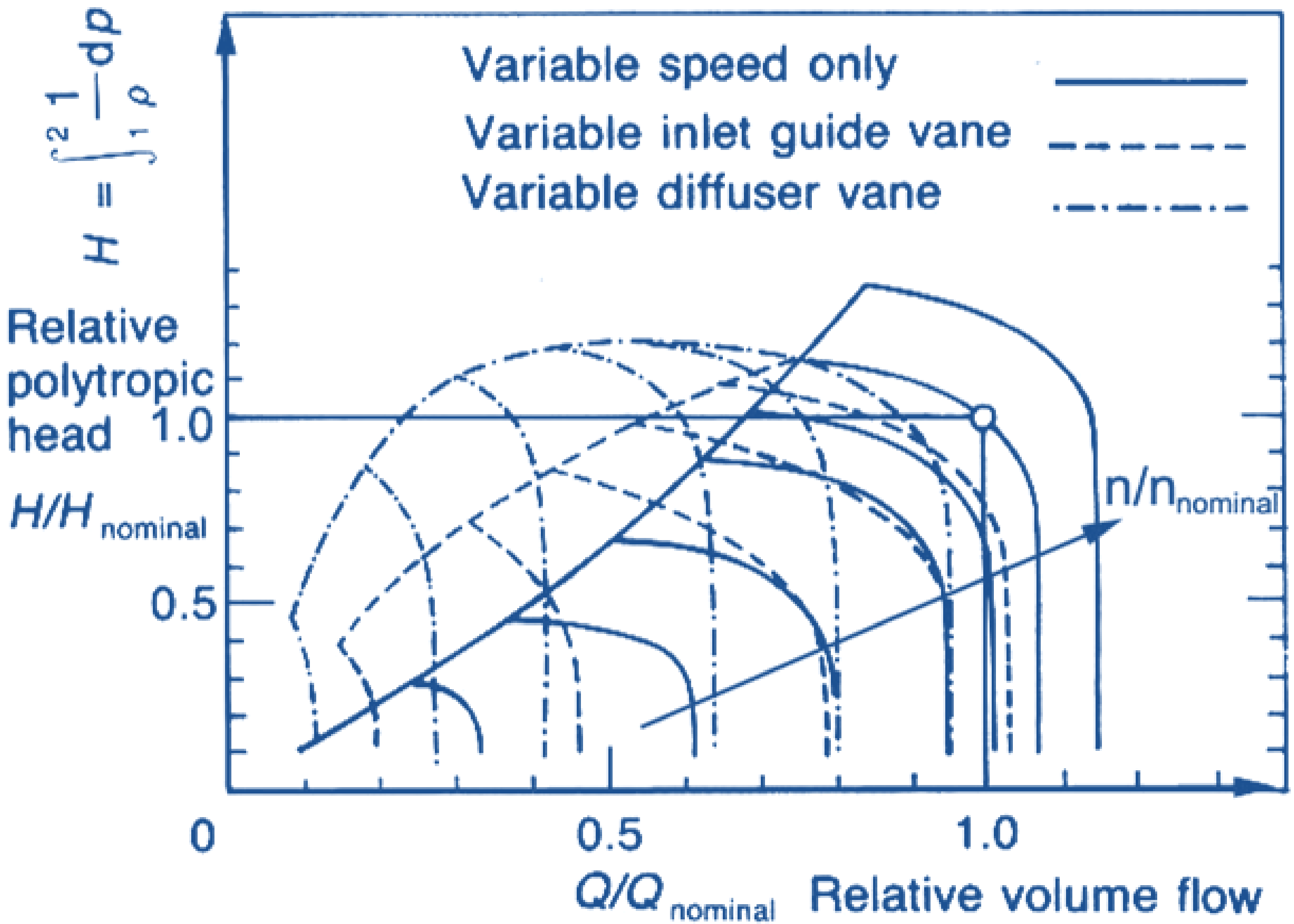
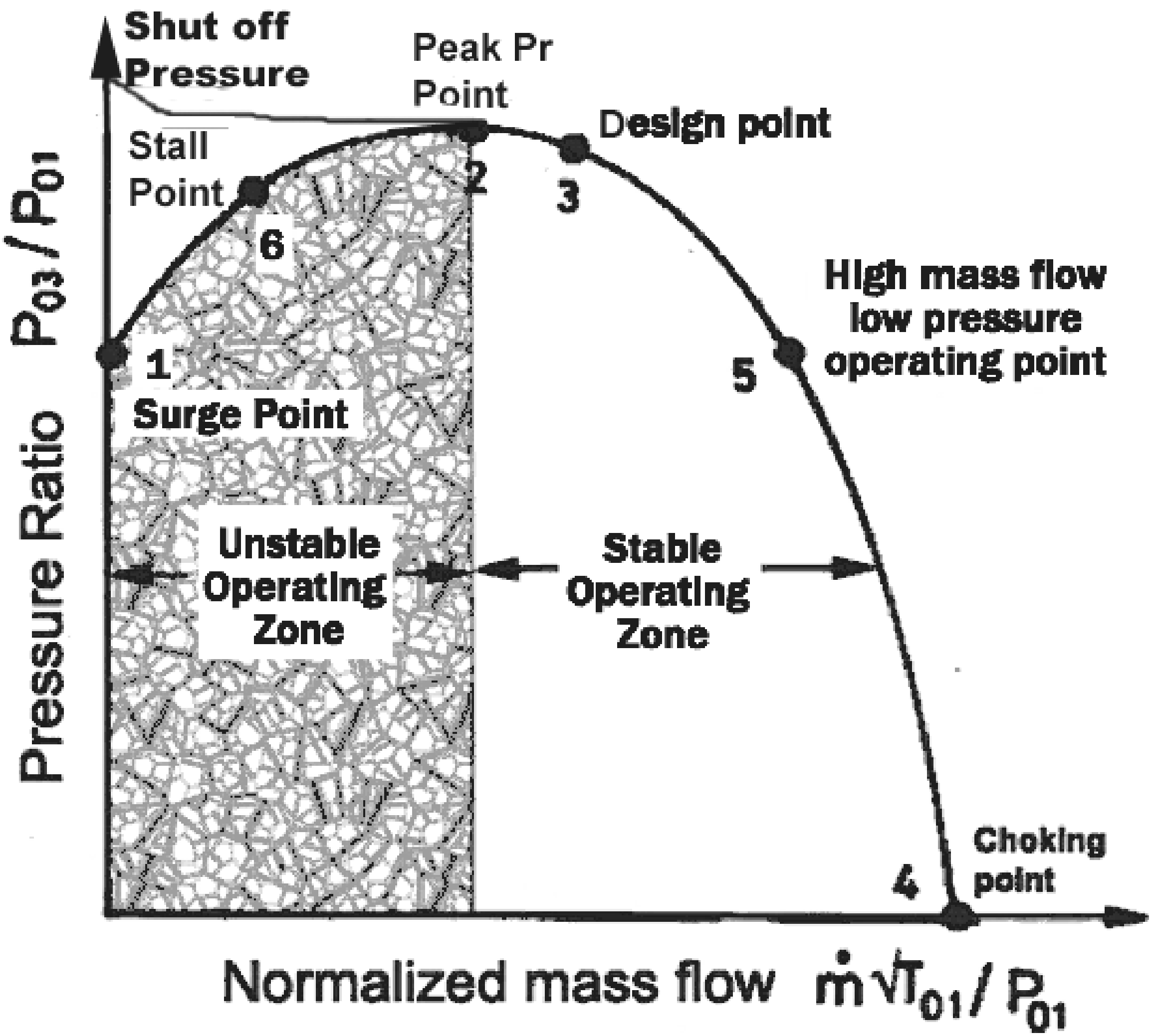


Centrifugal Compressor characteristics with multiple controls

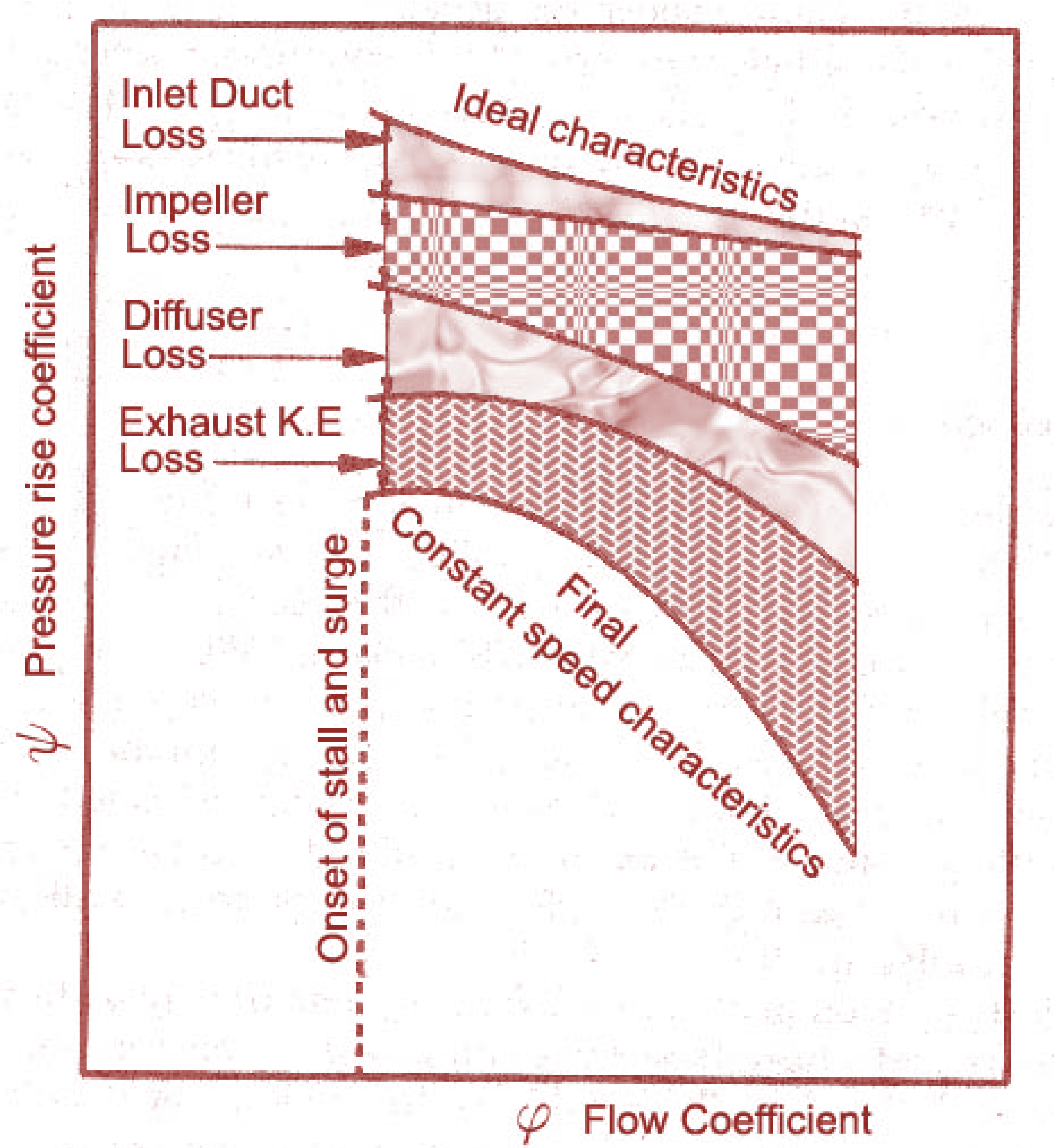


Stall and Surge control

•**Surging tends to originate in diffuser passages where frictional effects of the vane retard the flow.**

• **Flow reversal may vary from one blade passage to the next.**

• **The surging is reduced by making the number of diffuser vanes an odd number mis-match of the impeller vanes. In this way pressure fluctuations are more likely to be evened out over the annular vaneless circumference.**



Losses : Ideal and Real Characteristics

- **Most of the losses are still found by rigorous rig test.**
- **CFD gives good 1st cut estimation of loss analysis**

- Efficiency, η is borne out of loss analysis, whereas work done factor Ψ, is borne out of flow analysis as shown in the last slide. A value of σ_s is also arrived at by either CFD analysis or a first cut value by simple flow analysis.
- The flow parameters need averaging both at the compr. inlet (eye) along the vane height as well as the impeller exit along the depth of the vane.

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Next Lecture

Radial Turbines

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