Institute for
Thermal Turbomaschinery
and Machine Dynamics

Graz University of Technology
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Design of Thermal Turbomachinery

Lecture at the
Department of Aerospace Engineering
Middle East Technical University
Ankara, April 2008

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>> MIDDLE EAST TECHNICAL UNIVERSITY

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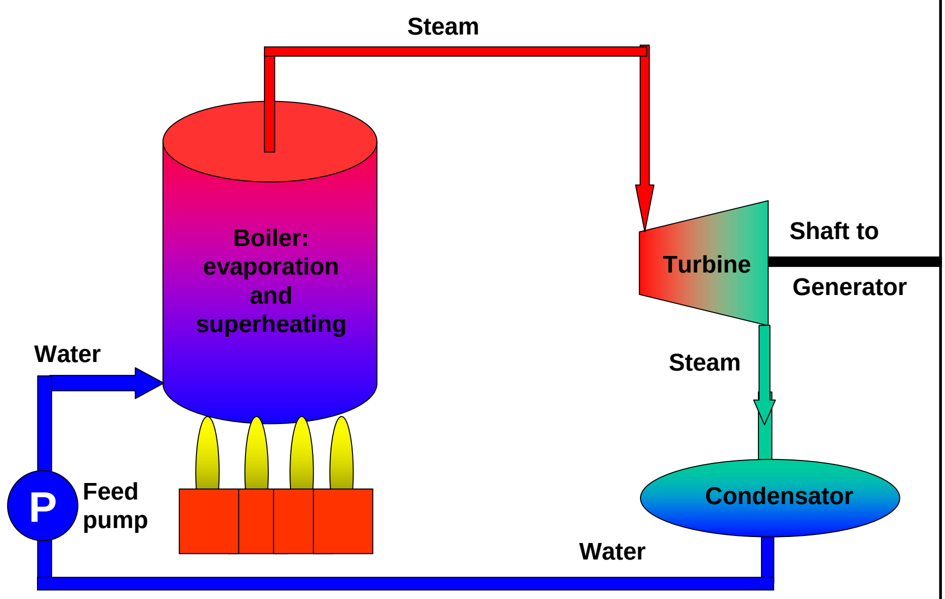
Definition of thermal Turbomachinery

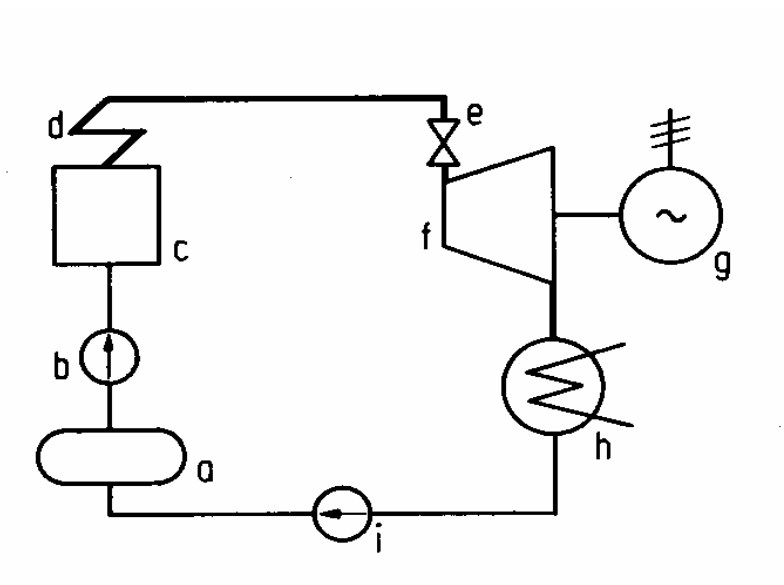
- Thermal turbomachinery work with compressible fluids
  ↔ hydraulic machinery, ventilators
- Only rotating motion of the rotor and continuous flow and work process ↔ combustion engines
- Energy conversion is based on fluid flow:
  Stationary and rotating blades are used to transfer potential into kinetic energy
- Due to high rotational speed contact-free sealings are used
  ↔ piston ring in combustion engines
- Thus working range: large volume flows, not too high pressures

Steam process



Source: Heitmeir, LN TU Graz Thermische Turbomaschinen

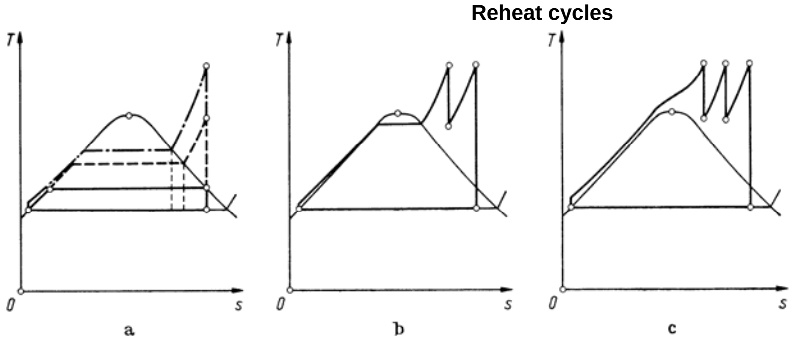
Steam process



Temperature – entropy – diagram

Idealized process without losses

Reheat cycles



Peak pressure: 170 (– 300 bar, supercritical)

Condensation pressure: 0.02 – 0.1 bar (condensation)

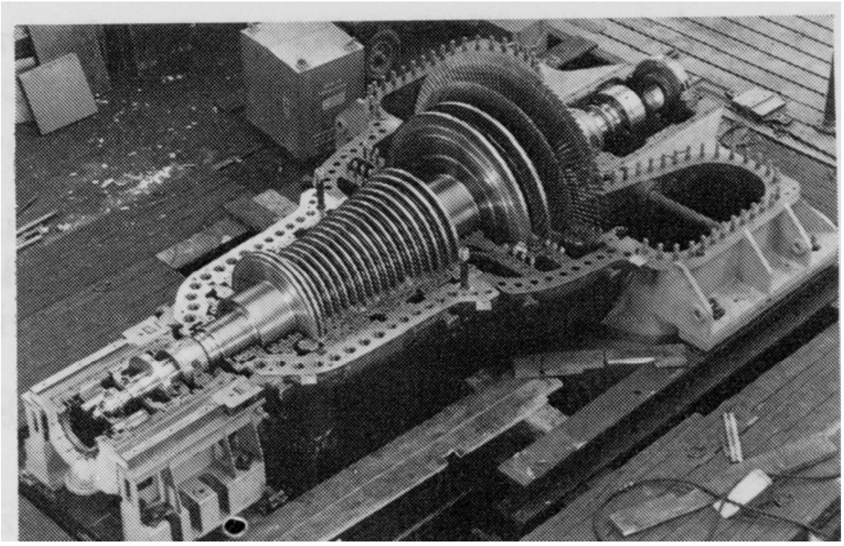
Peak temperature: 525 – 650°C

Power: 5 – 1000 MW

Efficiency: 40 % (aiming at 50 % till 2020)

Source:Traupel Thermische Turbomaschinen

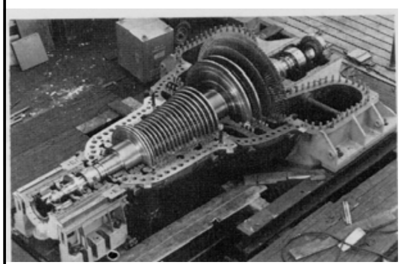
Steam turbine design

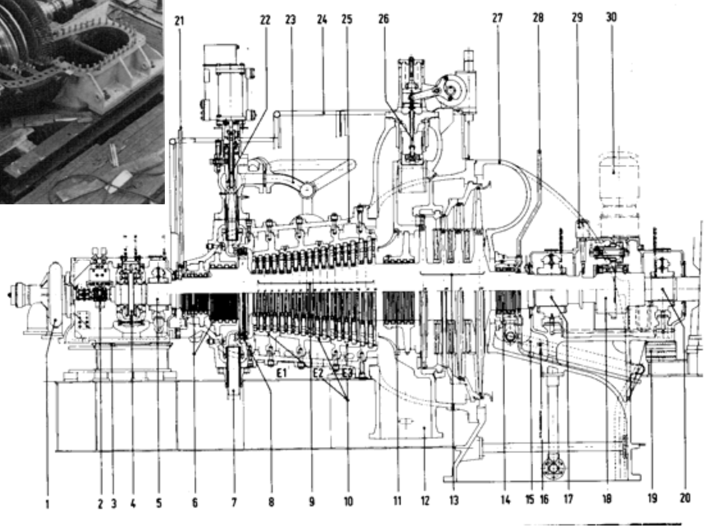


20/25 MW Extraction-Condensation turbine, live steam 70 bar, 525⁰C, extraction pressure 2,6 bar, Condensation pressure 0,032 bar

Source: Dietzel Dampfturbinen STM 7, S. 45

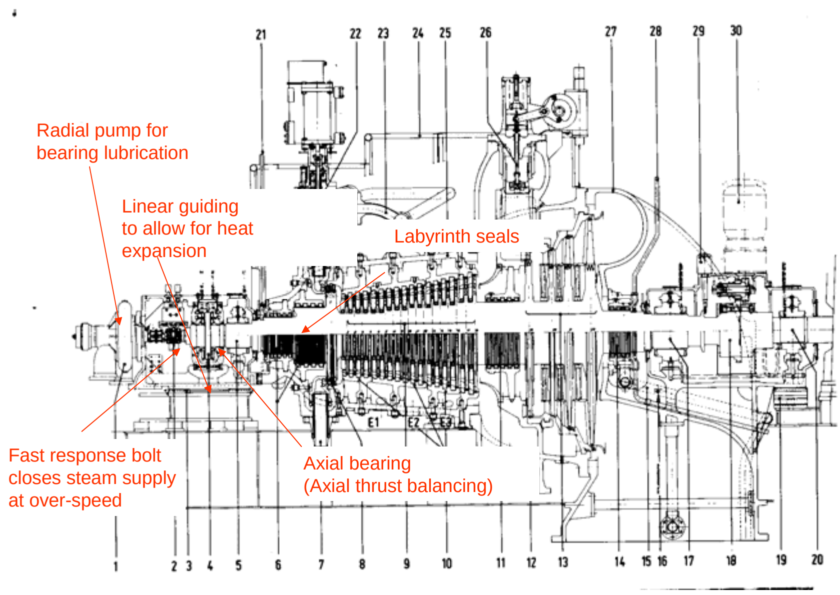
Steam turbine design





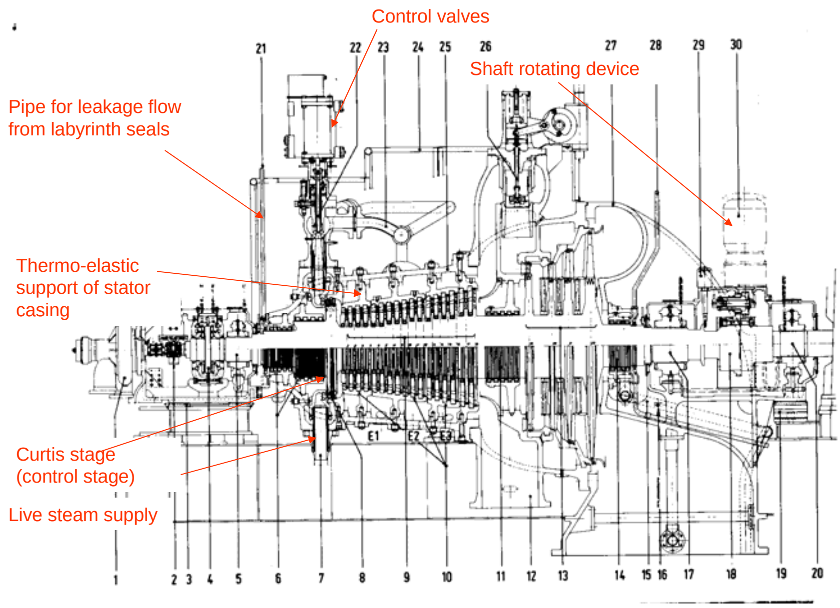
Source: Dietzel Dampfturbinen STM 7, S. 45

Steam turbine design



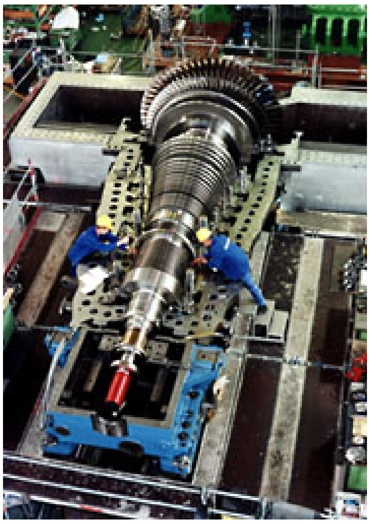
Source: Dietzel Dampfturbinen STM 7, S. 45

Steam turbine design



Source: Dietzel Dampfturbinen STM 7, S. 45

Siemens N-class steam turbine



Siemens N-type turbines are used for normal to medium live steam conditions (100 bars, 500 °C). This type of turbine has a reaction-type design that has proven itself many times over. N-type turbines can be either condensing or backpressure machines. The standard configuration allows up to two controlled extractions and/or several uncontrolled extraction points.

Source: Siemens

Steam turbine design

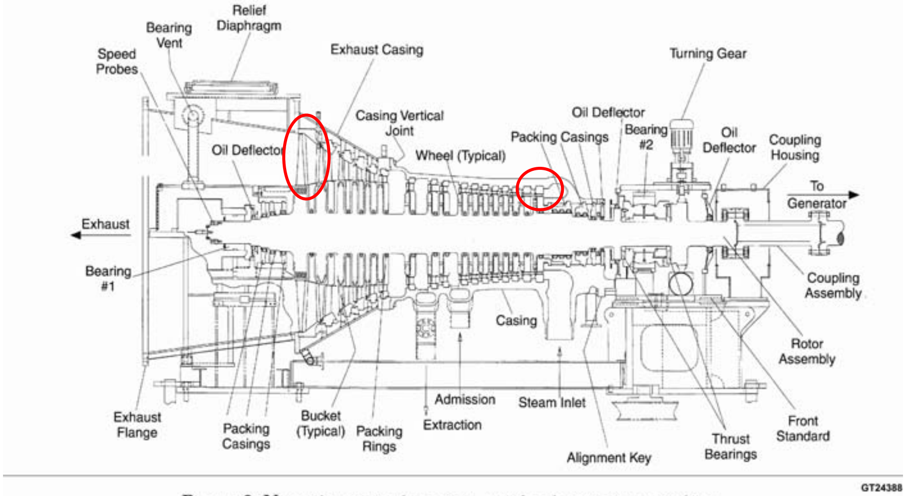
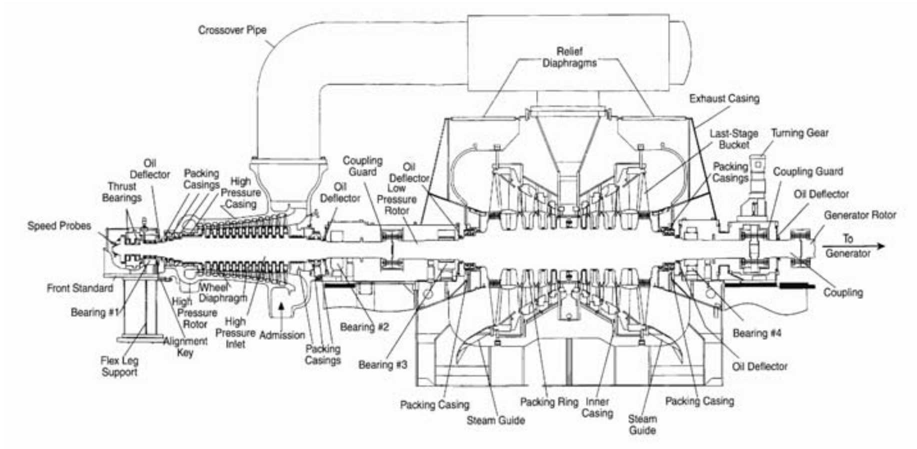


Figure 6. Nonreheat, single-casing, axial exhaust steam turbine

Source: GE Steam turbines for STAG Combined Cycle PPlt GER3582e.pdf

Steam turbine design

High pressure turbine and two-flow condensation turbine



Source: GE Steam turbines for STAG Combined Cycle PPlt GER3582e.pdf

Steam turbine in a nuclear plant

1200 MW turbo set with 6-flow condensation turbine

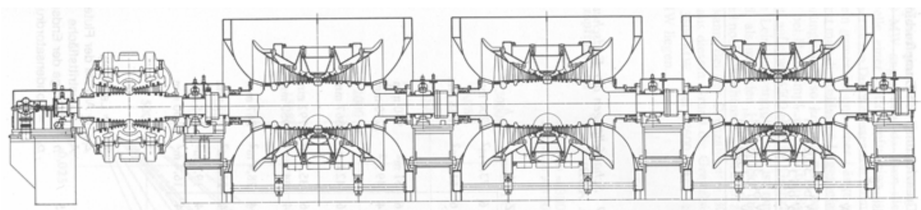
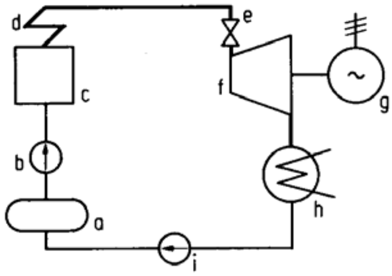


Bild 8.18 1200-MW-Dampfturbosatz eines Kernkraftwerkes (nach Fa. Siemens AG)

Source: Siemens

Last Stage of a Condensation Turbine



High pressure turbine with inner and outer casing

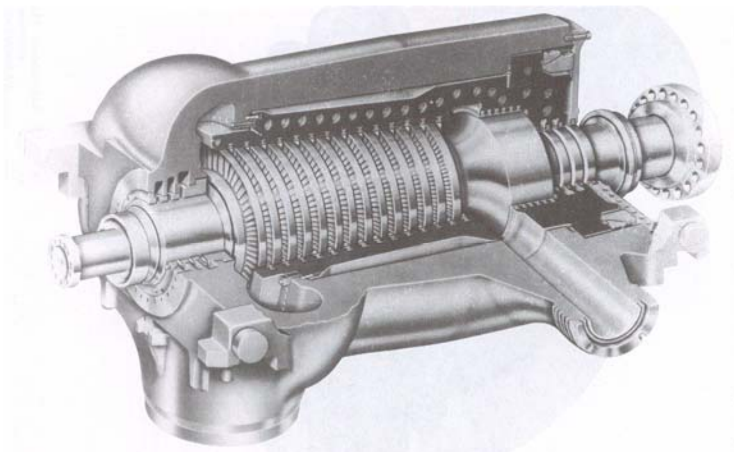
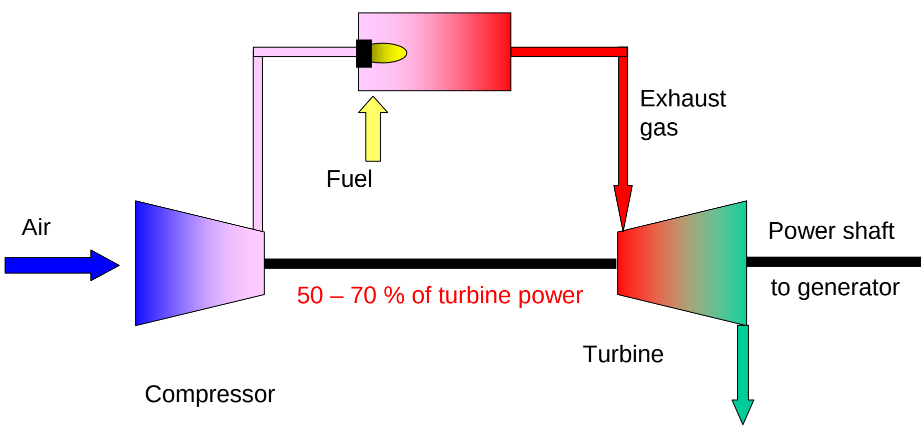


Bild 8.13 Hochdruckteil einer mehrgehäusigen Turbine (nach Fa. Siemens AG)

Source: Siemens

Basic Principle of a Gas Turbine



Pressure ratio: usually about 15, but up to 40 and more

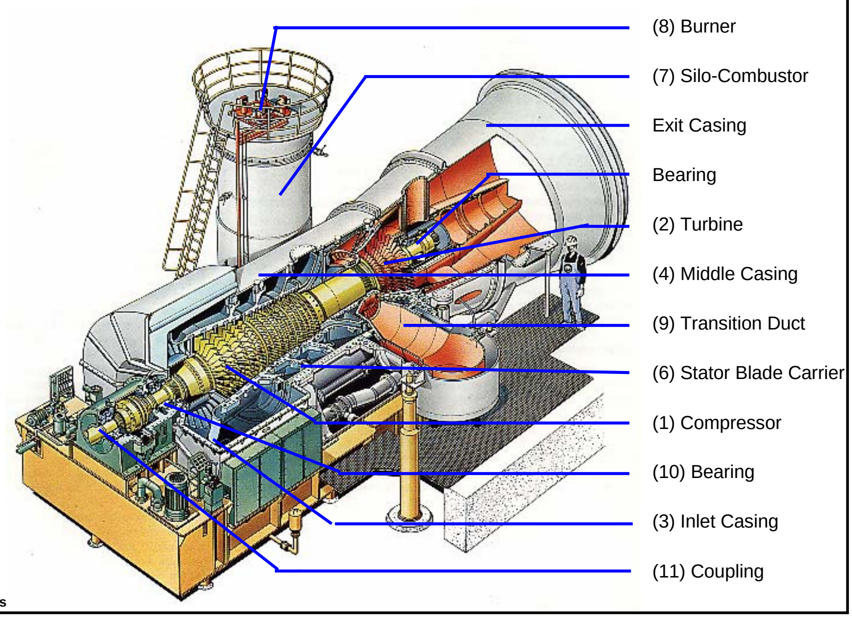
Turbine inlet temperature (TIT): 900° - 1700°C

Turbine exit temperature (TET): 400° - 600°C

Power: 100 kW – 300 MW

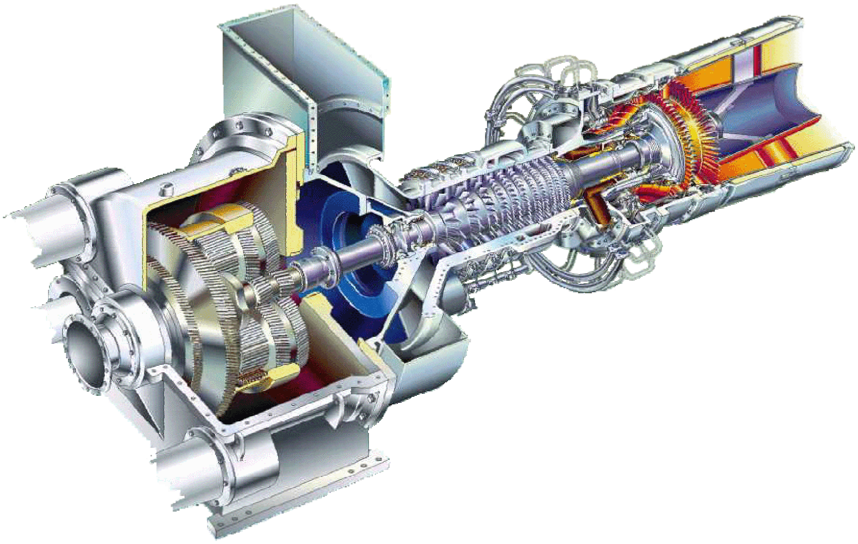
Source: Heitmeir LN TU Graz

Stationary Gas Turbine

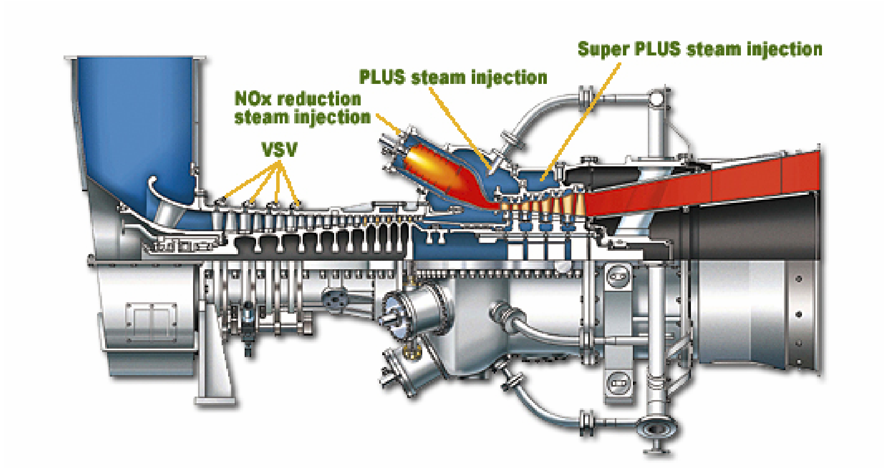


Source: Siemens

Stationary Gas Turbine



Kawasaki STIG M7A-91ST Gas Turbine



Source: Kawasaki

Radial Compressor

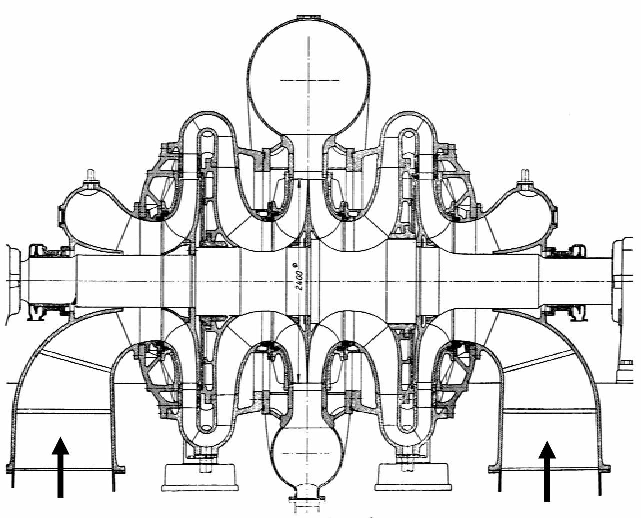


Abb. 7.16. Zweiflutige, zweistufige Speicherpumpe. $H = 170 - 190$ m, $\dot{V} = 14{,}5$ m³/s, $n = 333$ U/min, $P_{max} = 27600$ kW. Geschweißtes Spiralgehäuse, Laufräder der ersten Stufe aus Chromstahlguß, die der zweiten Stufe, Saugkrümmer und übriges Gehäuse aus Stahlguß (Pumpspeicherwerk Reisach-Rabenleite, Voith)

Source: Pfleiderer, Strömungsmaschinen, S 310

Description of Fluid

Ideal gas:

$$p \cdot v = R \cdot T$$

Real gas:

Enthalpy-entropy-diagram

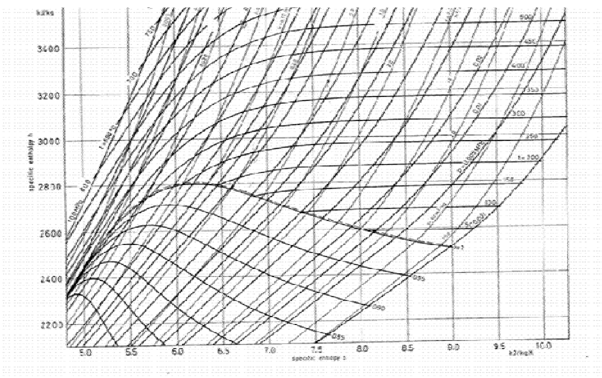
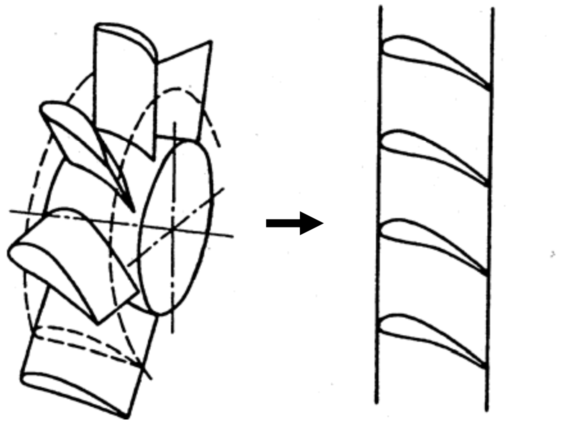


Fig. 8. Mollier h, s-Diagram

Bild 8. Mollier h, s-Diagramm

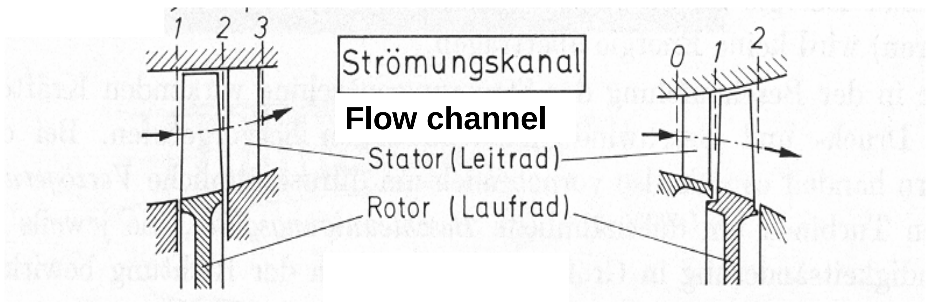
Cascade



Transfer of 3D geometry to 2D linear cascade (mid section)

Source: Niehuis LN Strömungsmaschinen

Turbomachinery Stage – Meridional View

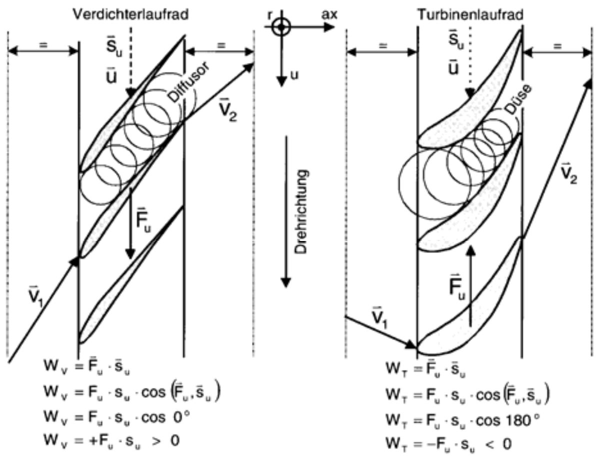


Compressor
- decreasing channel height

Turbine
- increasing channel height

Source: Münzberg

Turbine vs. Compressor Flow



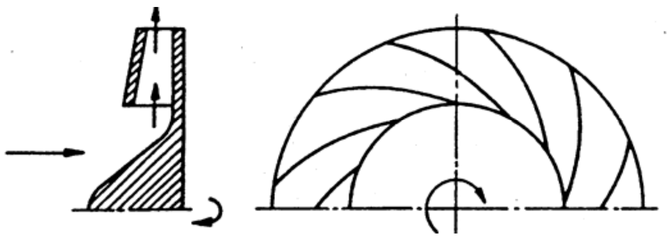
Zur Vorzeichenvereinbarung bei Verdichtern und Turbinen

Compressor
- decelerating flow
(diffuser)

Turbine
- accelerating flow
(nozzle)

Source: Flugzeugtriebwerke, Willy J.G. Bräunling,

Radial Machine

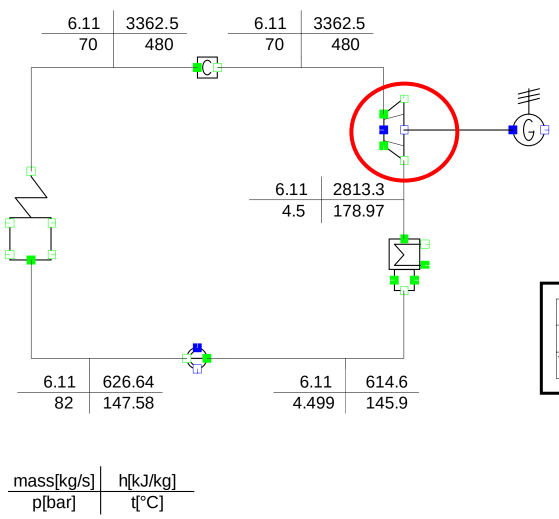


Linear cascade of a radial machine

Source: Niehuis LN Strömungsmaschinen

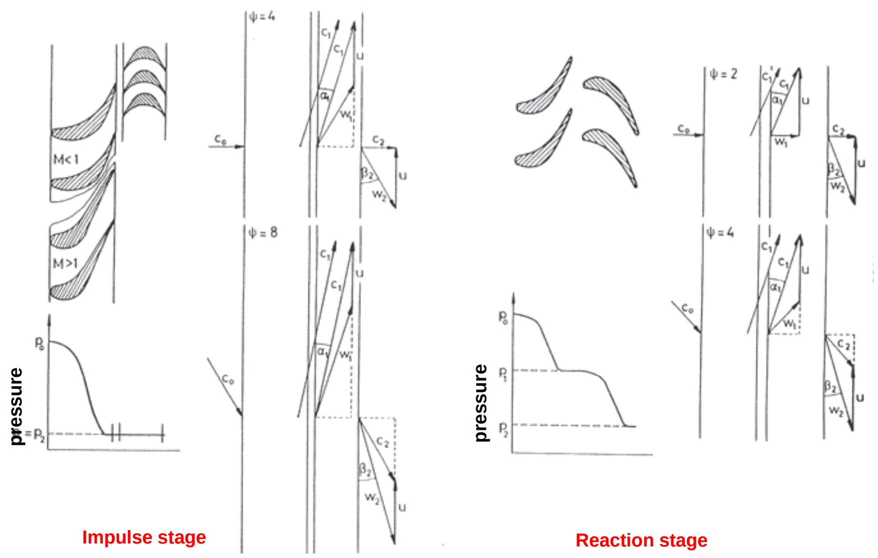
From Thermodynamic Layout to a Turbine Cascade

Steam cycle with back-pressure turbine (simplified):



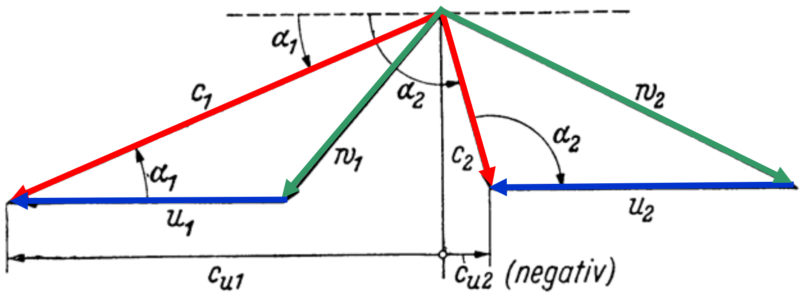
Power	3.13	MW
Efficiency	16.83	%
Turbine Dh_is	669.68	kJ/kg

Velocity Triangles of a Turbine Stage



Source: Jericha LN TU Graz

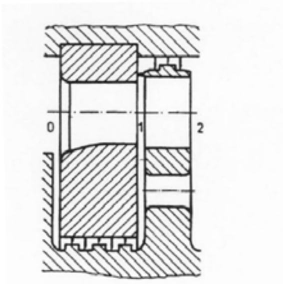
Velocity Triangle

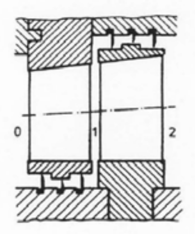


$$\vec{c} = \vec{w} + \vec{u}$$

Source: Traupel

Impulse Stage vs. Reaction Stage





Impulse Stage

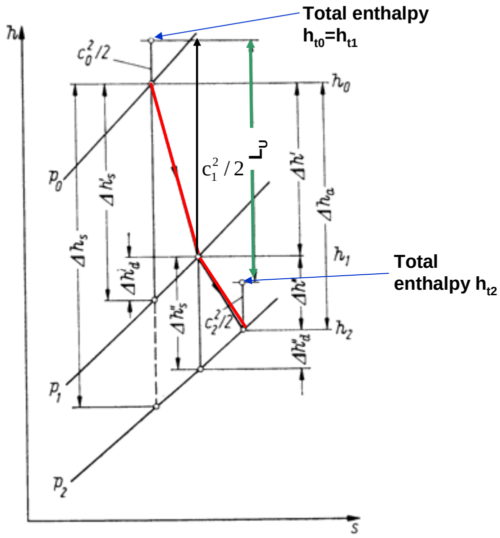
- Higher turning, thus lower efficiency
- Higher enthalpy drop
- Rotor pressure difference small so partial admission possible
- Smaller leakage loss (sealing at smaller radius)
- Smaller axial thrust

Reaction Stage

- Better stator and rotor efficiency
- Lower enthalpy drop
- only full admission
- Higher leakage loss
- Higher axial thrust, thus thrust balancing necessary

Source: Jericha LN TU Graz

h-s-Diagram of a Turbine Stage



Specific work: $$(h_2 + \frac{c_2^2}{2}) - (h_0 + \frac{c_0^2}{2}) = -L_u$$

Source: Jericha LN TU Graz

Stator Layout

$$h_1 + \frac{c_1^2}{2} = h_0 + \frac{c_0^2}{2}$$

$$h_0 - h_1 = \Delta h'$$

$$\frac{c_1^2}{2} = \Delta h' + \frac{c_0^2}{2}$$

for the isentropic expansion:‘

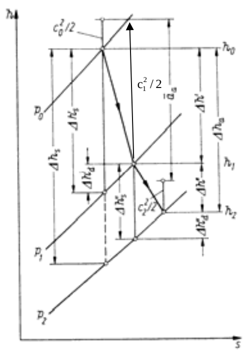
$$\frac{c_{1s}^2}{2} = \Delta h_s' + \frac{c_0^2}{2}$$

with a stator efficiency η‘

$$\boxed{\frac{c_1^2}{2} = \eta'(\Delta h_s' + \frac{c_0^2}{2})}$$

the absolute velocity c_1 at stator exit can be obtained

Using the continuity at stator exit the axial velocity component and thus the stator exit angle are obtained



Source: Jericha LN TU Graz

Rotor Layout

$$h_2 + \frac{w_2^2}{2} = h_1 + \frac{w_1^2}{2} + \frac{u_2^2 - u_1^2}{2}$$

$$h_1 - h_2 = \Delta h''$$

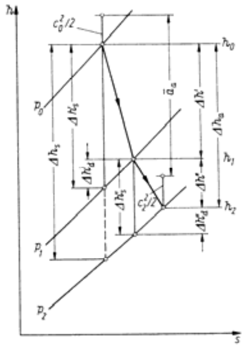
$$\frac{w_2^2}{2} = \Delta h'' + \frac{w_1^2 + u_2^2 - u_1^2}{2}$$

for the isentropic expansion:‘

$$\frac{w_{2s}^2}{2} = \Delta h_s'' + \frac{w_1^2 + u_2^2 - u_1^2}{2}$$

with a rotor efficiency η‘‘

$$\frac{w_2^2}{2} = \eta''\left(\Delta h_s'' + \frac{w_1^2 + u_2^2 - u_1^2}{2}\right)$$



the relative velocity w_2 at rotor exit can be obtained

Using the continuity at rotor exit the axial velocity component and thus the flow angle are obtained

Source: Jericha LN TU Graz

Specific Stage Work (Euler Equation)

$$\left(h_2 + \frac{c_2^2}{2}\right) - \left(h_0 + \frac{c_0^2}{2}\right) = -L_u$$

$$h_0 - h_2 = \Delta h' + \Delta h'' = \Delta h$$

$$L_u = \Delta h + \frac{c_0^2 - c_2^2}{2} = \Delta h' + \Delta h'' + \frac{c_0^2 - c_2^2}{2} = h_{t_0} - h_{t_2}$$

$$L_u = \frac{1}{2}\left(c_1^2 - c_2^2 + w_2^2 - w_1^2 + u_1^2 - u_2^2\right)$$

$$w_1^2 = c_1^2 + u_1^2 - 2u_1 c_1 \cos\alpha_1$$

$$w_2^2 = c_2^2 + u_2^2 - 2u_2 c_2 \cos\alpha_2$$

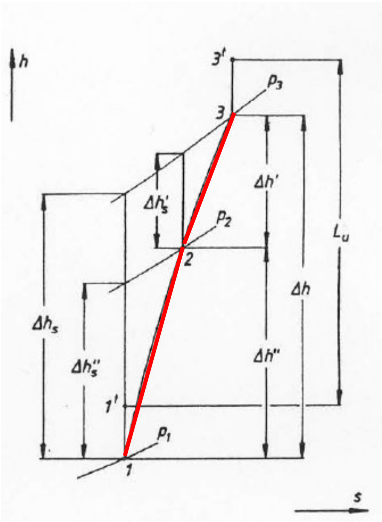
$$L_u = u_1 c_1 \cos\alpha_1 - u_2 c_2 \cos\alpha_2 = u_1 c_{u1} - u_2 c_{u2}$$

Axial flow at constant radius: $L_u = u \Delta c_u$

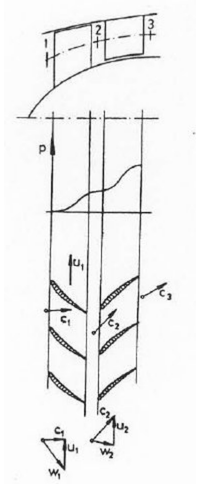
High circumferential speed (large diameter, high rotational speed) leads to high specific work!!!

Source: Jericha LN TU Graz

Layout for a compressor stage



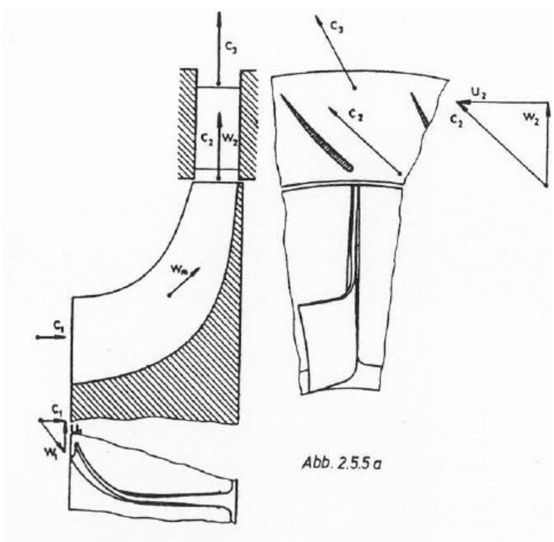
Axial compressor



Source: Jericha LN TU Graz

Layout for a compressor stage

Radial compressor



Source: Jericha LN TU Graz

Efficiency Definition

Circumferential Efficiency, valid at mid section:

Circumferential efficiency without exit loss:

$$\eta_u = \frac{L_u}{\Delta h_s + \frac{c_0^2 - c_2^2}{2}}$$

Circumferential efficiency with exit loss:

$$\eta_u^* = \frac{L_u}{\Delta h_s + \frac{c_0^2}{2}}$$

Isentropic circumferential efficiency:

$$\eta_{su} = \frac{\Delta h}{\Delta h_s}$$

Source: Jericha LN TU Graz

Inner Efficiency

Inner work: $L_i = L_u - \Sigma \Delta L = L_u - \Delta L'_{sp} - \Delta L''_{sp} - \Delta L_R - \Delta L_V$

with

$\Delta L'_{sp}$	**Stator leakage loss**
$\Delta L''_{sp}$	**Rotor leakage loss**
ΔL_R	**Wheel friction**
ΔL_V	**Ventilation loss**

Inner efficiency without exit loss: $\eta_i = \frac{L_i}{\Delta h_s + \frac{c_0^2 - c_2^2}{2}}$

Inner efficiency with exit loss: $\eta_i^* = \frac{L_i}{\Delta h_s + \frac{c_0^2}{2}}$

Isentropic inner efficiency: $\eta_{si} = \frac{\Delta h_i}{\Delta h_s}$ $\qquad \Delta h_i = \Delta h - \Sigma \Delta L$

Source: Jericha LN TU Graz

Total Efficiency

$$\eta = \eta_i \cdot \eta_l \cdot \eta_m$$

$\eta =$ Total efficiency

$\eta_i =$ Inner efficiency considering all stage losses

$\eta_l =$ Volumetric efficiency considering leakage losses of shaft

$\eta_m =$ Mechanical efficiency considering bearing friction, ...

Source: Bohl, Strömungsmaschinen 1, Seite 23

Dimensionless Stage Parameters

Flow Coefficient is a dimensionless volume flow:

$$\varphi = \frac{\dot{V}_2}{u_2 . A_2}$$

$$\varphi = \frac{\dot{m}\, v_2}{u_2 . A_2} = \frac{w_2}{u_2} \sin \beta_2$$

Load Coefficient is a dimensionless enthalpy drop:

$$\psi = \frac{\Delta h_s}{u_2^2 / 2}$$

In USA: $\psi = \frac{\Delta h_s}{u_2^2}$

$$\psi = \frac{2}{\eta_{su} u_2^2} (u_1 c_{u1} - u_2 c_{u2})$$ $c_1 = c_2$

$u_1 = u_2 = u$ $$\psi = \frac{1}{\eta_{su}} \frac{2 \Delta c_u}{u}$$

Source: Jericha LN TU Graz

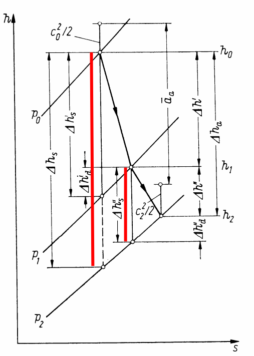
Degree of Reaction

Definition:

r = Isentropic enthalpy drop in rotor / Isentropic stage enthalpy drop

$$r = \frac{\Delta h_s''}{\Delta h_s}$$

$$r = \frac{\left(\frac{w_2}{u_2}\right)^2 \frac{1}{\eta''} - \left(\frac{w_1}{u_2}\right)^2 + \left(\frac{u_1}{u_2}\right)^2 - 1}{\psi}$$



Kinematic degree of reaction:

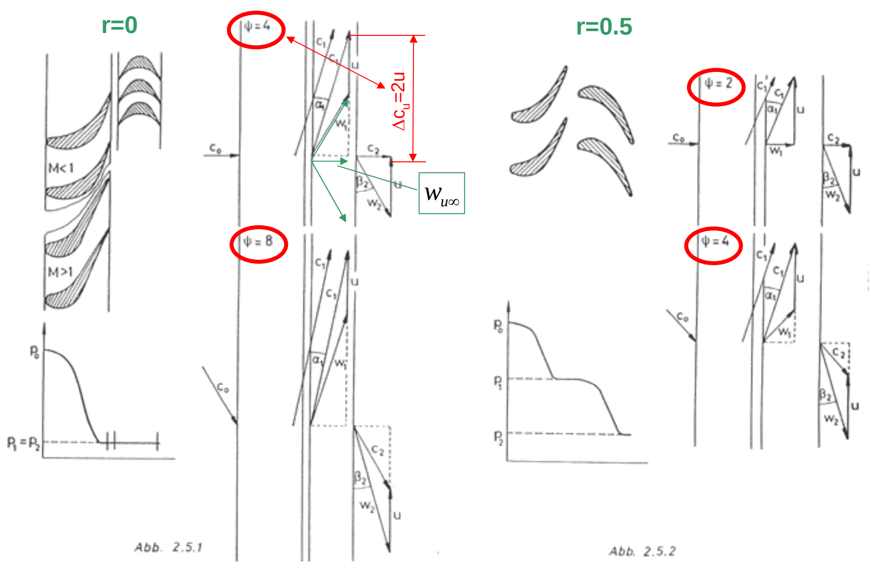
$c_{ax1} = c_{ax2}$, $u_1 = u_2$

$\eta' = \eta'' = 1$

$$r_{kin} = -\left(\frac{\frac{w_{u1}}{u} + \frac{w_{u2}}{u}}{2}\right) = 1 - \frac{\frac{c_{u1}}{u} + \frac{c_{u2}}{u}}{2} = -\frac{w_{u\infty}}{u}$$

Source: Jericha LN TU Graz

Dimensionless Parameters and the Velocity Triangle



$\eta_{su} = 1$

Source: Jericha LN TU Graz

Degree of Reaction = 0

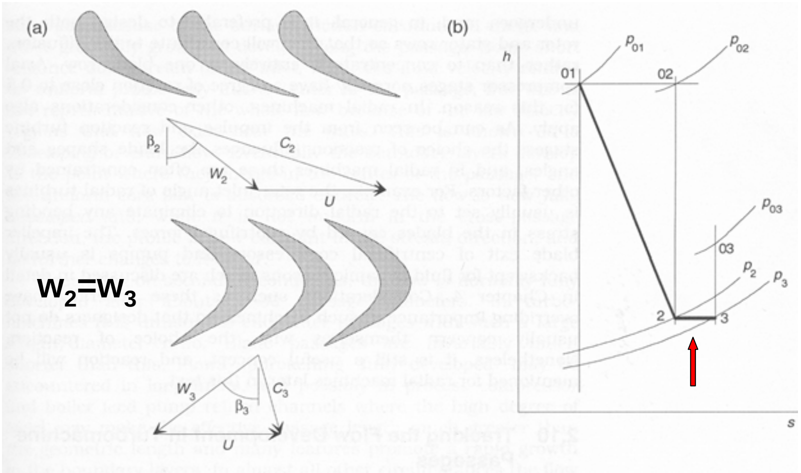


Figure 2.20. Impulse turbine (a) Schematic diagram (b) Mollier diagram.

Source: Intro. Turb. Fig 2.20

Degree of Reaction = 0.5

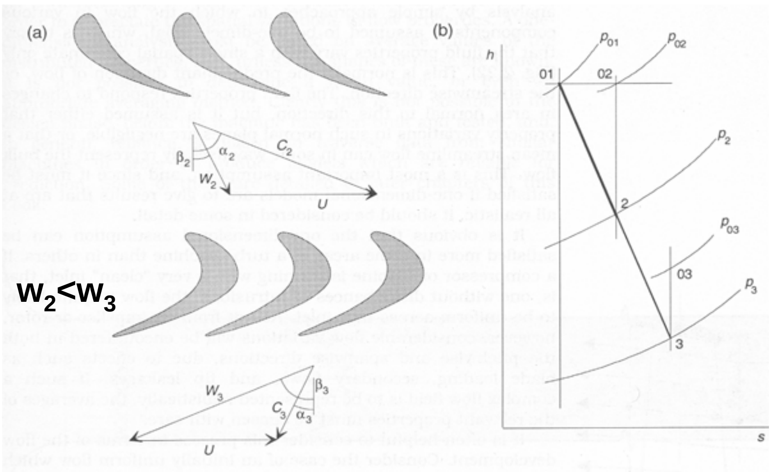


Figure 2.21. Reaction turbine (a) Schematic diagram (b) Mollier diagram.

Source: Intro. Turb. Fig 2.21

Turbines of different degree of reaction

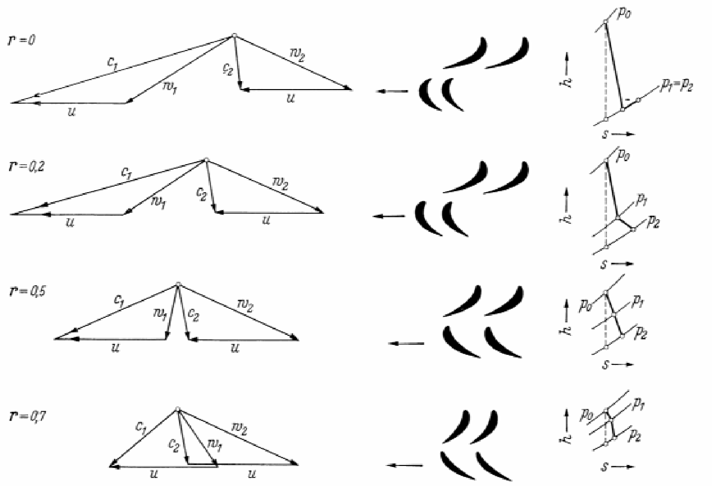


Abb. 4.1.3 Turbinenschaufelungen verschiedener Reaktionsgrade.

Source: Traupel 1 STM 140

Compressor Cascade

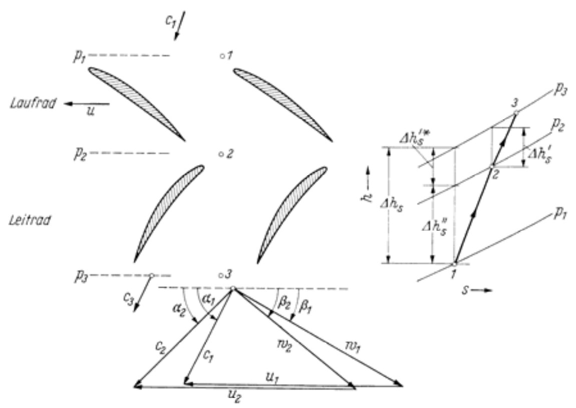


Abb. 4.2.2 Arbeitsweise einer Axialverdichterstufe.

Source: Traupel 1 STM 140

Compressors of different degree of reaction

$$r = \frac{\Delta h_s''}{\Delta h_s' + \Delta h_s''} \approx \frac{\Delta h_s''}{\Delta h_s} \qquad 4.2(1)$$

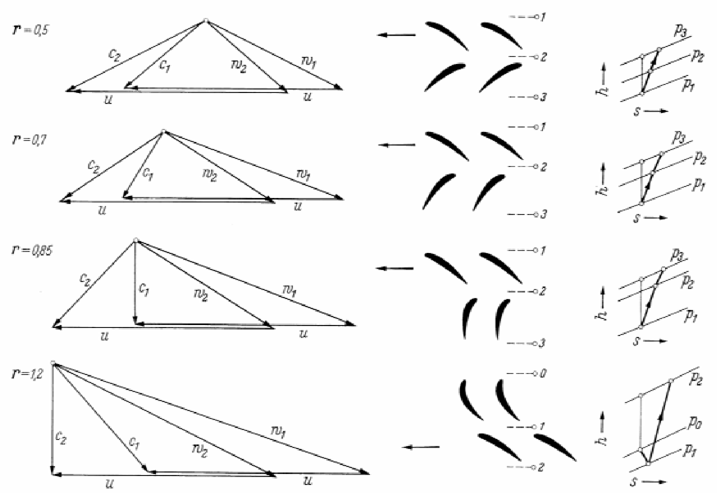


Abb. 4.2.3 Axialverdichterschaufelungen verschiedenen Reaktionsgrades.

Source: Traupel 1 STM 140

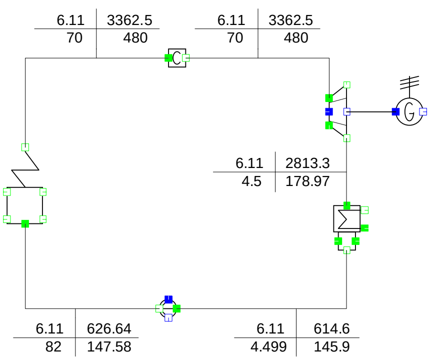
From Thermodynamic Layout to a Turbine Cascade

What is needed to design a stage?

- **Stage enthalpy drop**
- **Rotational speed**
- **Mean diameter**
- **Degree of reaction**
- **Stator and Rotor efficiency**

Power	3.13	MW
Efficiency	16.83	%
Turbine Dh_is	669.68	kJ/kg

mass[kg/s]	h[kJ/kg]
p[bar]	t[°C]



Layout of a Stage Group

$$u = \Omega R = D\pi n / 60$$

$$\dot{V} = \frac{u}{\cos\alpha} \sin\alpha \, D\pi \frac{D}{D/l}$$ **(axial outflow)** → $\varphi = \tan\alpha$

Generally: $\varphi = f.\tan\alpha$ **with** $f = \frac{\Psi}{4} + (1 - r)$

Limit for short blades: $\alpha = 14^\circ$, $D/l = 10$

Limit for long blades: $\alpha = 35^\circ$, $D/l = 3$

$$D = \sqrt[3]{\frac{60\,\dot{V}\,(D/l)}{n\,\pi^2\,f\,\tan\alpha}}$$

$$D_a = D\left(1 + \frac{1}{D/l}\right) \qquad D_i = D\left(1 - \frac{1}{D/l}\right)$$

D … mean diameter
V … volume flow
$D_{i,a}$ … inner/outer diameter
n … rotational speed [rpm]
α … nozzle angle
u … circumferential velocity
l … blade length

Number of Stages

Isentropic stage enthalpy drop: $h_{st} = \psi \frac{u^2}{2}$

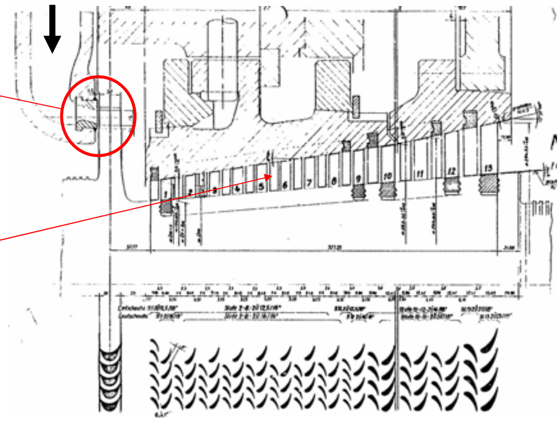
Stage number z: $z \cdot \frac{h_{St\,1} + h_{St\,z}}{2} = \Delta h_{Teilturbine}$

By varying the parameters r, ψ, α, D/l and n for the first and last stage of a stage group feasible dimensions and stage number can be found!

Layout of reaction stages

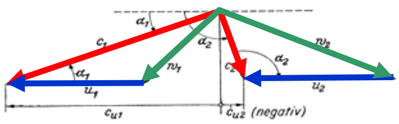
Control stage: 20 % of turbine enthalpy drop

13 reaction stages:
Δh_is = 536 kJ/kg
V_1 = 0.426 m^3/s
V_2 = 2.75 m^3/s



⇨ **Excel**

Assumption of stator and rotor efficiency then allows the layout of the velocity triangles!

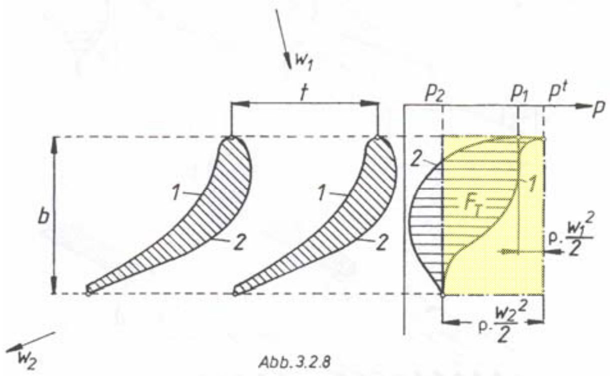


Source: Jericha LN TU Graz

Blade Number

Zweifel method:

Zweifel investigated the ratio of the tangential force F_T to an idealised tangential force F_{Tid} and found a common value of ψ_T=0.8



$$F_{T_{id}} = \frac{\rho}{2}\cdot w_2^2 \cdot b \qquad F_T = \rho \cdot w_{ax} \cdot (w_{2u} - w_{1u}) \cdot t \qquad (5)$$

$$\psi_T = \frac{F_T}{F_{T_{id}}} = 2 \cdot \frac{w_{ax}\cdot(w_{2u} - w_{1u})}{w_2^2} \cdot \frac{t}{b}$$

Source: Jericha LN TU Graz

Blade Number

Zweifel method:

With ψ_T=0.8 an optimum ratio between spacing t and blade width b can be found!

$$\psi_T = 0{,}8 \qquad \left(\frac{t}{b}\right)_{opt} = \frac{0{,}8}{2} \cdot \frac{w_2^2}{w_{ax} \cdot \Delta w_u}$$

$$\left(\frac{t}{b}\right)_{opt} = \frac{0{,}8}{2} \cdot \frac{\sin\beta_1}{\sin\beta_2 \cdot \sin(\beta_1 - \beta_2)} = \frac{0{,}8}{2} \cdot \frac{1}{\sin^2\beta_2(\cot\beta_2 - \cot\beta_1)}$$

For compressible flows:

$$\left(\frac{t}{b}\right)_{opt} = 0{,}8 \frac{\Delta p}{\rho_2 w_{ax_2} \cdot \Delta w_u}$$

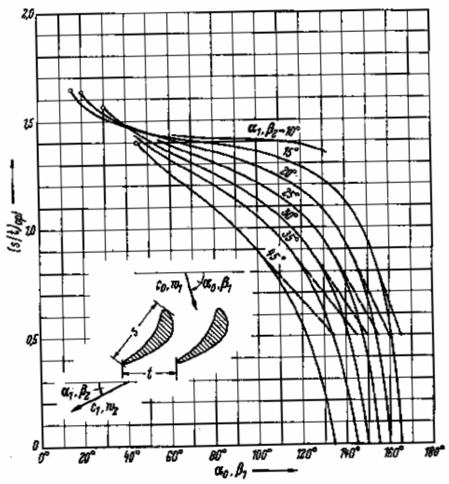
with $\Delta p = p_{1t} - p_2$

The blade width b can be chosen or is given by strength considerations!

Source: Jericha LN TU Graz

Blade Number

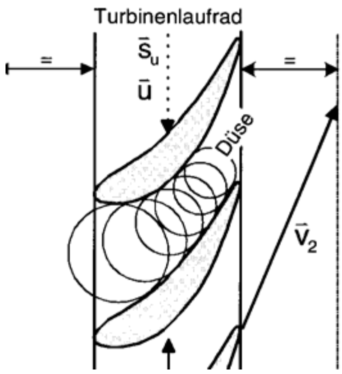
Alternatively following diagram by Traupel can be used to find the optimum spacing depending on the blade turning:



Source: Traupel I, 405

Next Step: Blade Geometry

- Based on the velocity triangle the blade geometry is designed
- The flow channel should allow a continuous acceleration (or deceleration)
- The narrowest area (throat) is usually at the blade trailing edge!



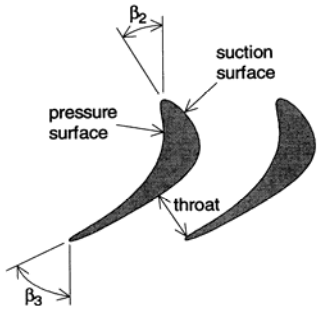


Figure 6.5. Turbine stage: principal objectives.

Source: Bild 6.1 STM 152 Japikse Introduction into Turbomachinery

Examples for Blade Design based on Zweifel method

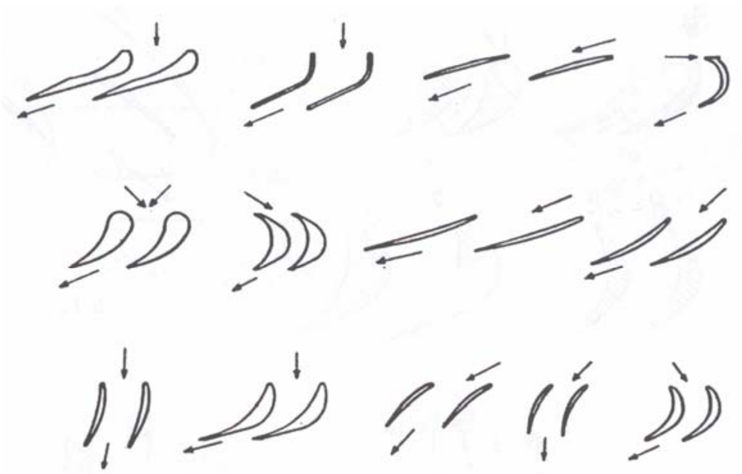
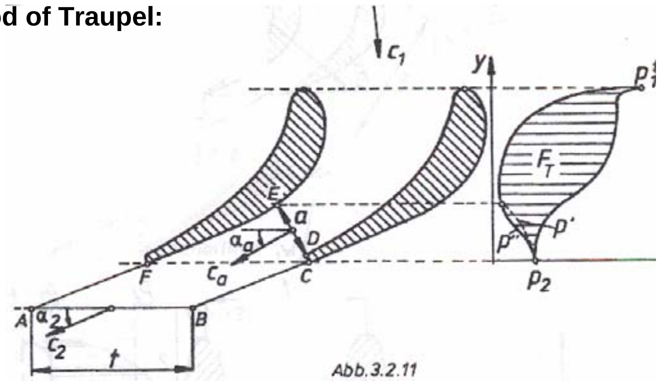


Abb. 3.2.9

Source: Jericha LN TU Graz

Does the flow follow the geometry?

Method of Traupel:



Continuity:
$$t \cdot c_2 \cdot \sin\alpha_2 = a \cdot c_a$$

Impulse in circumferential direction:
$$J_E = \dot{m} \cdot c_a \cdot \cos\alpha_a \qquad J_A = \dot{m} \cdot c_2 \cdot \cos\alpha_2$$
$$c_2 \cdot \cos\alpha_2 = c_a \cdot \cos\alpha_a$$

Source: Jericha LN TU Graz

Method of Traupel

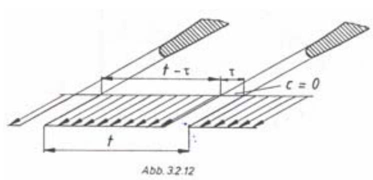
$$\sin\alpha_2 = \frac{a}{t} \cdot \frac{c_a}{c_2} = \frac{a}{t} \cdot \frac{\cos\alpha_2}{\cos\alpha_a}$$

Thus the blade exit flow angle depends on the throat area and the throat flow angle

$$\tan\alpha_2 = \frac{1}{\cos\alpha_a} \cdot \frac{a}{t}$$

Considering the thickness of the blade trailing edge, following formula is obtained:

$$tg\alpha_2 = \frac{1}{\cos\alpha_a} \cdot \frac{a}{t-\tau}$$

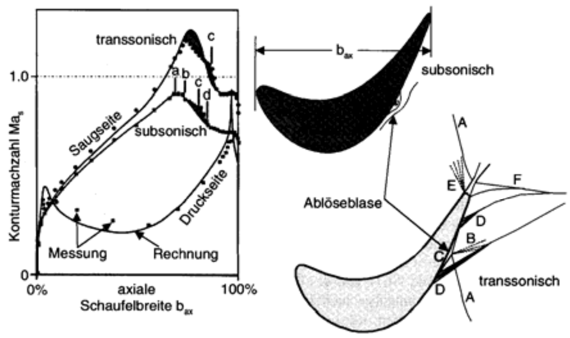


The blade geometry has to be adapted so that α_2 corresponds to the velocity triangle.

Source: Jericha LN TU Graz

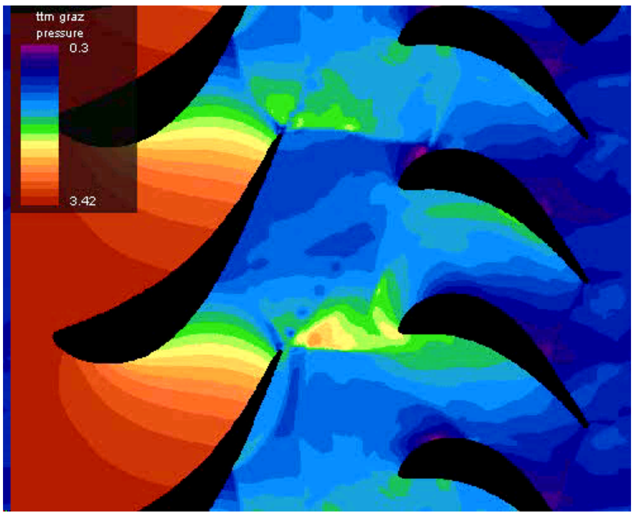
Turbomachinery Flow

Flow in turbomachinery is very complex:
- **three-dimensional**
- **laminar, transitiónal or turbulent**
- **unsteady with stator-rotor interaction**
- **transonic with shock – boundary layer interaction**
- **high pressure gradients lead to vortices**



Mach number distribution along suction and pressure side of a turbine at subsonic and transonic conditions

Instantaneous pressure distribution in a transonic turbine



Three-dimensional Blade Design

Until now the blade geometry was determined only for a representative mid section.

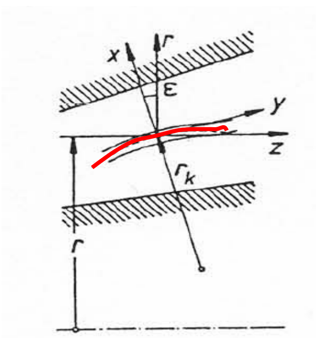
But there are differences along the blade height:

- The circumferential velocities vary along the blade height.
- The radial balance between centrifugal and pressure forces leads to a radial pressure distribution.

This leads to different velocity triangles along the blade height and thus a twist of the 3D blade is necessary (except for very small D/l ratios)

Three-dimensional Force Balance

Stream line in a meridional channel without blades:



Force Balance in x-direction:

$$-\underbrace{\frac{c_u^2}{r}\cos\varepsilon}_{a} - \underbrace{\frac{c_y^2}{r_k}}_{b} = -\underbrace{\frac{1}{\rho}\frac{\partial p}{\partial x}}_{c}$$

Force Balance in y-direction:

$$-\underbrace{\frac{c_u^2}{r}\sin\varepsilon}_{a} + \underbrace{c_y\frac{\partial c_y}{\partial y}}_{d} = -\underbrace{\frac{1}{\rho}\frac{\partial p}{\partial y}}_{c}$$

a ... centrifugal force by circumferential velocity
b ... centrifugal force by streamline curvature
c ... pressure force
d ... convective force

Source: Jericha LN TU Graz

Three-dimensional Force Balance

Some transformations and the assumption of isentropic flow lead to:

$$\frac{\partial h_t}{\partial r} - c_u \cdot \frac{\partial c_u}{\partial r} - c_y \cdot \frac{\partial c_y}{\partial r} = \frac{c_u^2}{r} + \frac{c_y^2}{r_k} \cdot \cos\varepsilon - c_y \cdot \frac{\partial c_y}{\partial r} \cdot \sin\varepsilon$$

Assumption of no curvature and parallel-to-axis flow lead to:

$$r_k = \infty, \quad \varepsilon = 0 \quad \longrightarrow \quad c_y = c_z$$

$$\frac{\partial h_t}{\partial r} - c_u \frac{\partial c_u}{\partial r} - c_y \frac{\partial c_y}{\partial r} = \frac{c_u^2}{r}$$

Assumption of the same enthalpy drop along the radius:

$$\frac{\partial h_t}{\partial r} = 0,$$

$$-c_u \cdot \frac{\partial c_u}{\partial r} - c_z \cdot \frac{\partial c_z}{\partial r} = \frac{c_u^2}{r}$$

Source: Jericha LN TU Graz

Constant Nozzle Angle Design

Postulation of a constant nozzle exit angle:

$$\alpha = const. \qquad \frac{c_z}{c_u} = \tan\alpha$$

$$-c_u \cdot \frac{\partial c_u}{\partial r} - c_z \cdot \frac{\partial c_z}{\partial r} = \frac{c_u^2}{r}$$

$$-\frac{\partial c_u}{\partial r} \cdot (1 + \tan^2\alpha) = \frac{c_u}{r}$$

$$1 + \tan^2\alpha = \frac{1}{\cos^2\alpha}$$

$$\frac{dr}{r} = -\frac{dc_u}{c_u} \frac{1}{\cos^2\alpha}$$

$$\ln r = -\frac{1}{\cos^2\alpha} \cdot \ln c_u + K$$

$$c_u = \left(\frac{K}{r}\right)^{\cos^2\alpha}$$

$$c_u = c_{um} \cdot \left(\frac{r_m}{r}\right)^{\cos^2\alpha}$$

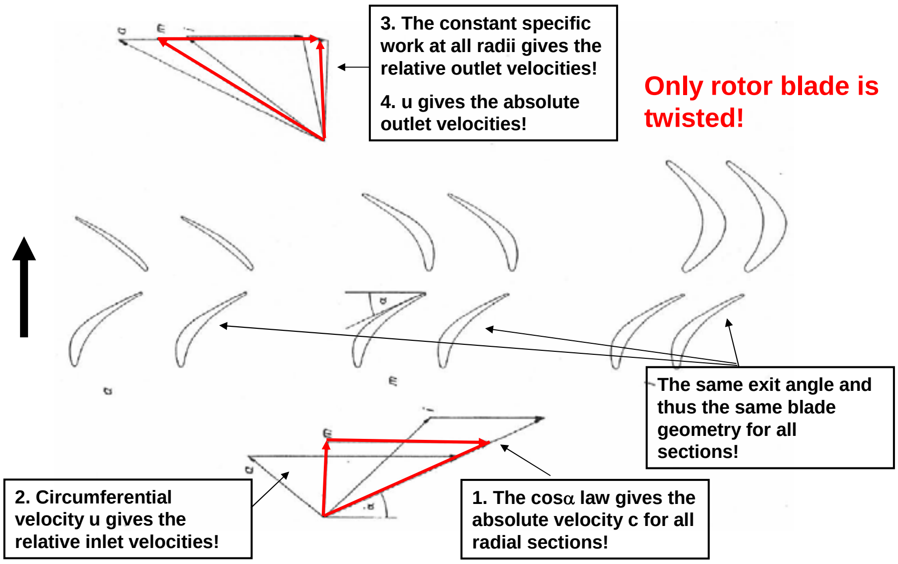
$$c_z = c_{zm} \left(\frac{r_m}{r}\right)^{\cos^2\alpha}$$

Effective in the axial gap between stator and rotor

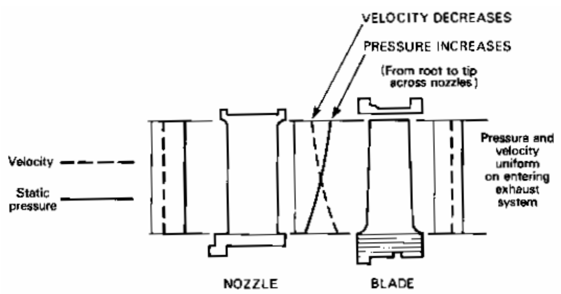
Source: Jericha LN TU Graz

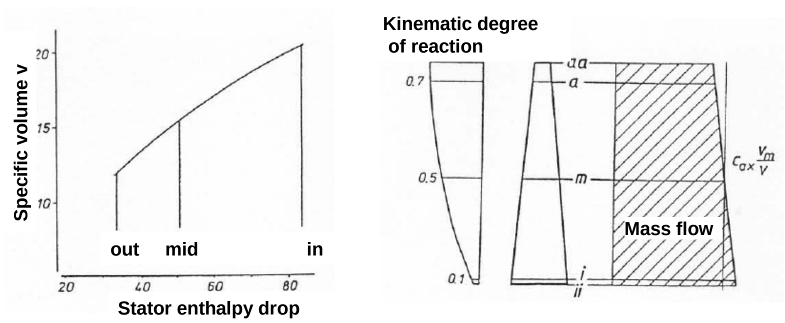
Constant Nozzle Angle Design

Starting point is the velocity triangle at mid section:



Constant Nozzle Angle Design





Source: Jericha LN TU Graz

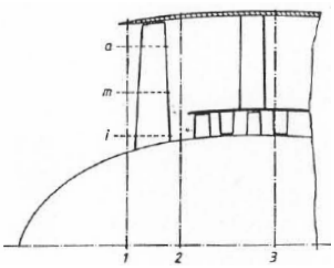
Free-Vortex Design

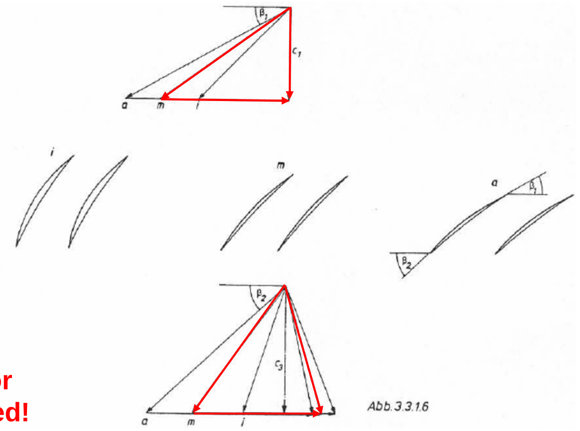
Postulation of a constant axial velocity between stator and rotor:

$$c_z = const.$$

$$-c_u \cdot \frac{\partial c_u}{\partial r} = \frac{c_u^2}{r} \qquad \frac{dc_u}{c_u} = -\frac{dr}{r} \qquad \boxed{c_u \cdot r = const.}$$

3D layout of a fan stage:





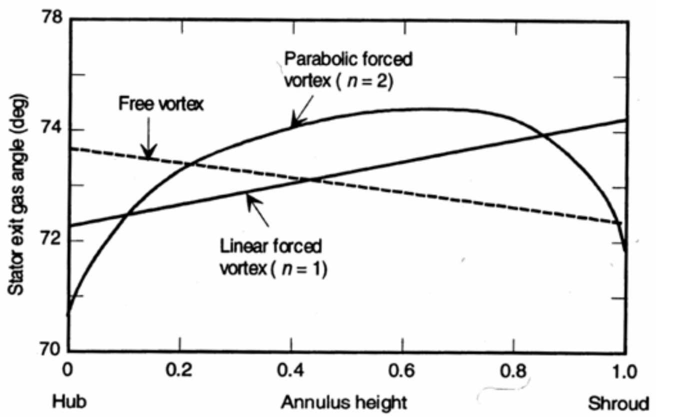
Rotor and stator blade are twisted!

Source: Jericha LN TU Graz

Three-dimensional Blade Design

The linear forced vortex law reduces the turning at the blade hub, the quadratic forced vortex law also reduces the turning at the blade tip.

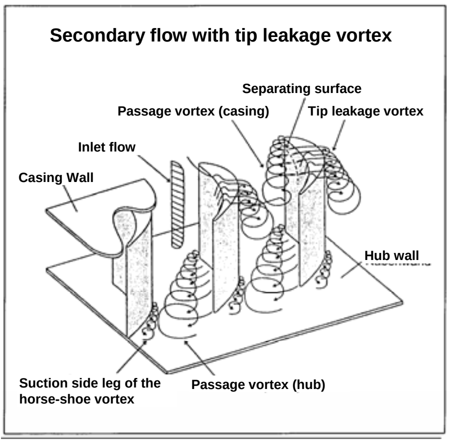
This results in a decrease of secondary losses!

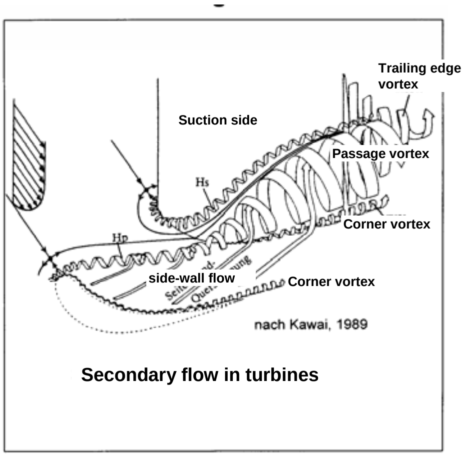


Variation of nozzle exit angle across the annulus for various vortex designs

Source: Fig 6.1 STM 152 Japikse Introduction into Turbomachinery

Secondary Flows





Hp ... Pressure side leg of the horse-shoe vortex

Hs ... Suction side leg of the horse-shoe vortex

Modern 3D Blade geometry

Where do the highest secondary losses occur?

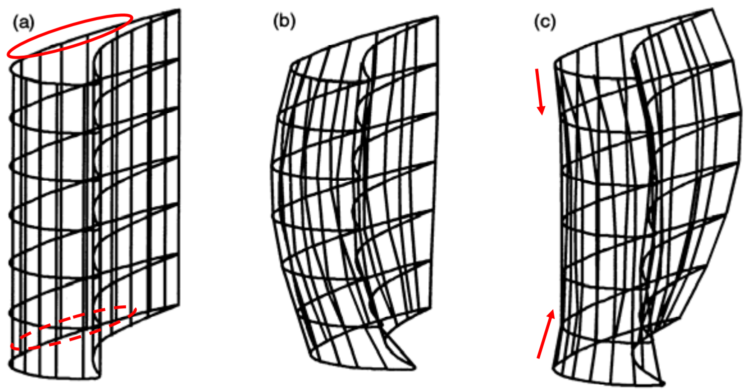
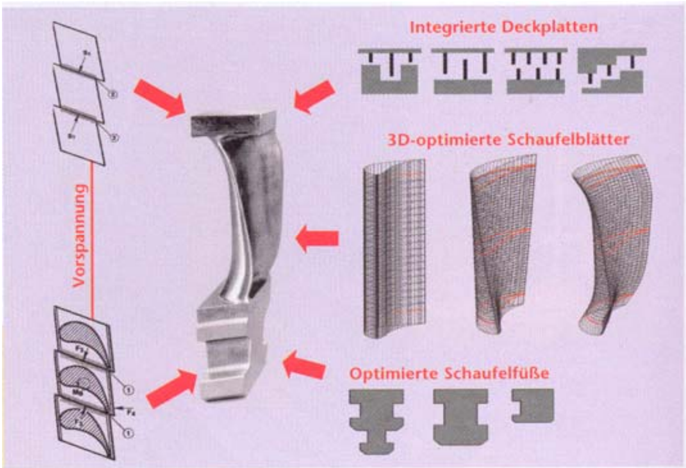


Figure 6.11. Section stacking: (a) radial stack (b) and (c) 3D stacks.

Stacking of two-dimensional sections along radius

Source: Fig. 6.1 STM 152 Japikse Introduction into Turbomachinery

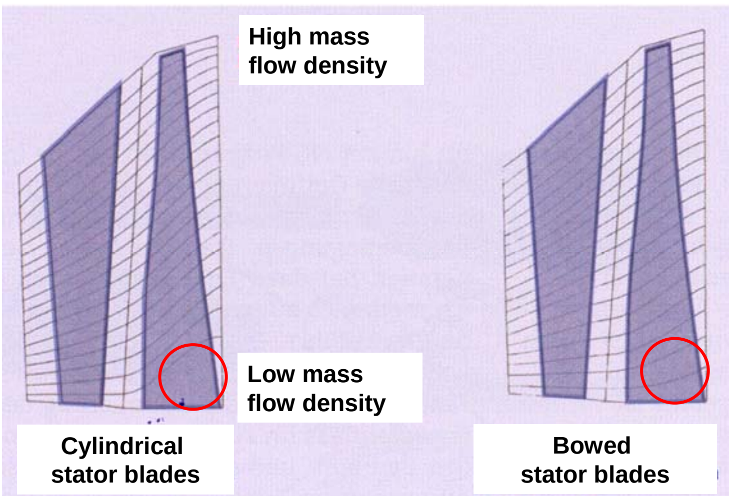
Modern 3D Blade geometry



Source: BWK 3/2005, Seite 58, Bild 3

Modern 3D Blade geometry

Meridian stream lines



Source: BWK 3/2005, Seite 60, Bild 4

GE-90 3D Fan Blade Design

Source: GE

Loss Calculation by Traupel

Stator/rotor efficiency: $\eta', \eta'' = 1 - (\zeta_p + \zeta_{SWR} + \zeta_{Zus})$

- ζ_p: Profile loss
- ζ_{SWR}: Side wall friction
- ζ_{Zus}: Additional losses

$$\zeta_p = \zeta_{p0} \chi_R \chi_M \chi_\delta + \zeta_\delta + \zeta_F$$

- ζ_{p0}: Basic profile loss due to flow turning
- χ_R: Mach number influence
- χ_M: Mach number influence
- χ_δ: Trailing edge influence
- ζ_δ: Trailing edge influence
- ζ_F: Trailing edge influence

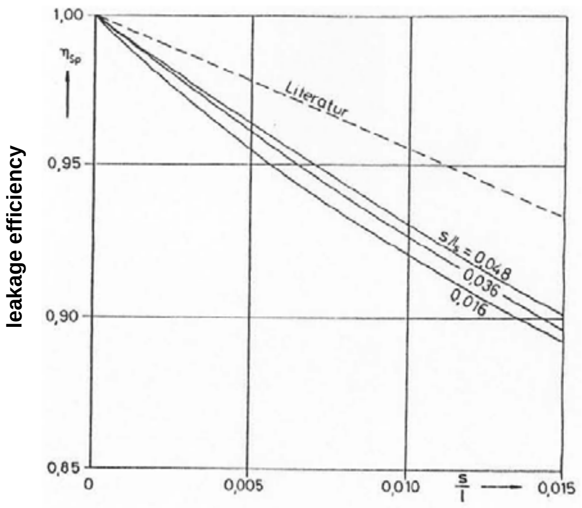
Circumferential efficiency: $\eta_u = f(\eta', \eta'', \alpha_i, \beta_i)$

Inner efficiency: $\eta_i = \eta_u - (\zeta_{Sp}' + \zeta_{Sp}'' + \zeta_R + \zeta_V)$

- ζ_{Sp}': Stator leakage loss
- ζ_{Sp}'': Stator leakage loss
- ζ_R: Wheel friction loss
- ζ_V: Ventilation loss

Source: Jericha LN TU Graz

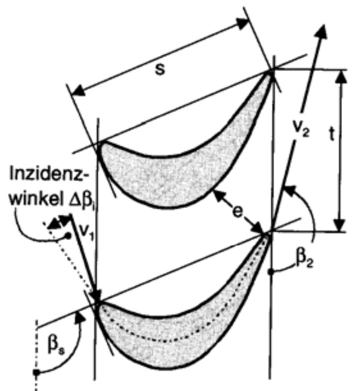
Tip leakage loss by Jericha

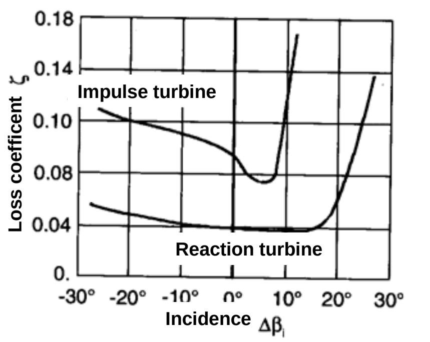


s … leakage height
l … blade height
l_s … blade chord length

Source: Jericha LN TU Graz

Part Load Operation



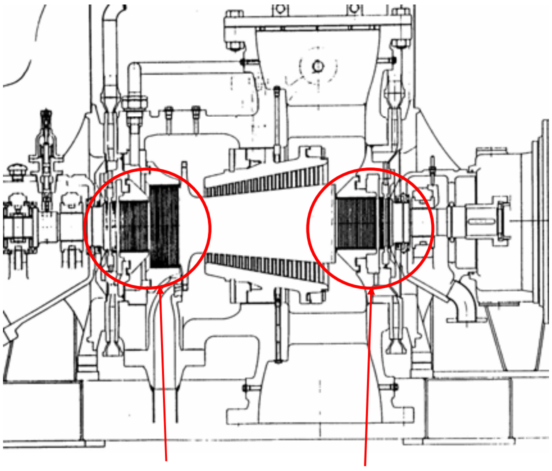


Losses of impulse and reaction turbine dependent on the incidence angle (Dietrichs et al, 1987)

Source: Dietrichs et al., 1987

Labyrinth seals

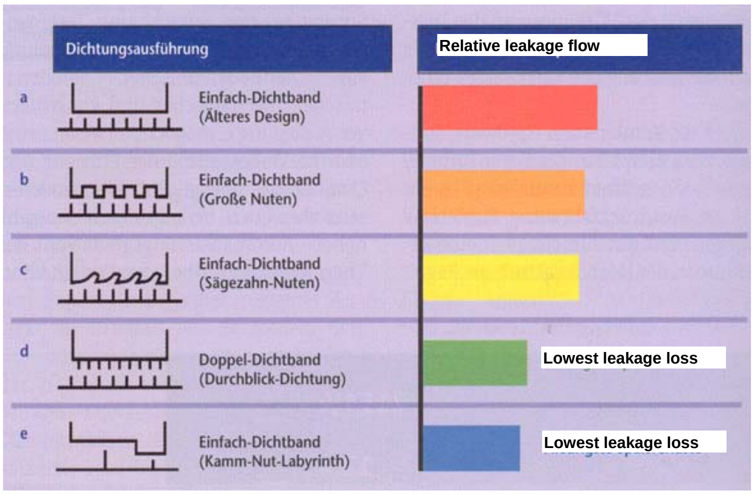
Steam turbine



Labyrinth seals to prevent steam leakage

Source: Jericha LN TU Graz

Different sealing designs



Flow through sealings is a repeated throttle flow

Source: BWK 3/2005, Seite 60, Bild 6

Estimate of sealing mass flow

Estimation of sealing gap height:

$$s_{Sp} = \left(k \frac{D}{1000} + 0{,}2 \right) \quad \text{[mm]}$$

with k = 0.6 … for compressor

k=0.85 … for turbines with a ferritic runner

k=1.3 … for turbines with an austenitic runner

Estimation of sealing mass flow (simplified):

$$\dot{m} = A\,\Phi \sqrt{\frac{p_1}{v_1}}$$

A … gap area

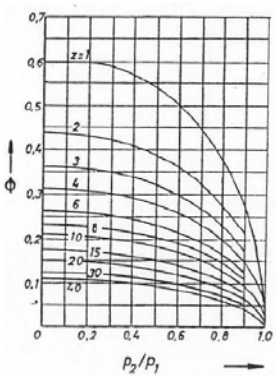
p_1 … pressure before first seal

p_2 … pressure after last seal

v_1 … specific volume before first seal

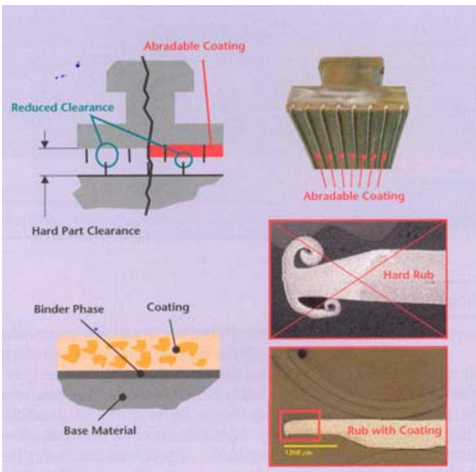
z … number of seal spikes

Φ … flow coefficient, depending on sealing geometry



Source: Jericha LN TU Graz

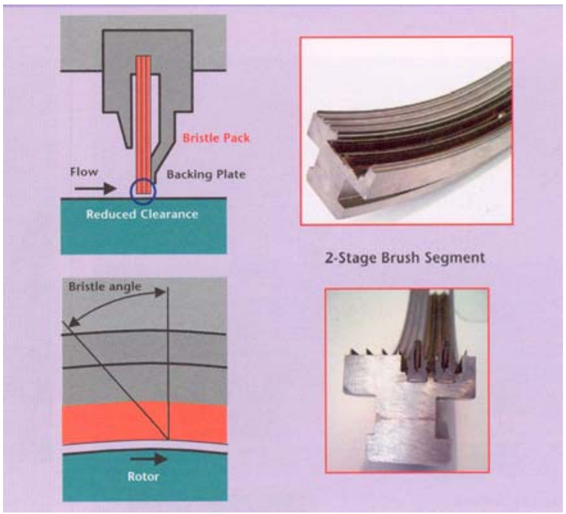
Abradable Coatings



Abradable coatings allow a reduced clearance

Source: BWK 3/2005, Seite 61, Bild 7

Brush Sealings

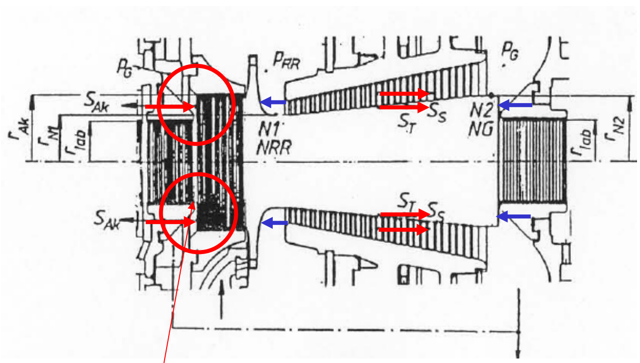


Brush sealings allow a further clearance reduction

Source: BWK 3/2005, Seite 62, Bild 8

Axial Thrust Balancing

Example of a steam turbine with control stage:



Balance piston allows balancing of axial forces

Source: Jericha LN TU Graz