Aerothermal Engineering

C204 Turbomachinery

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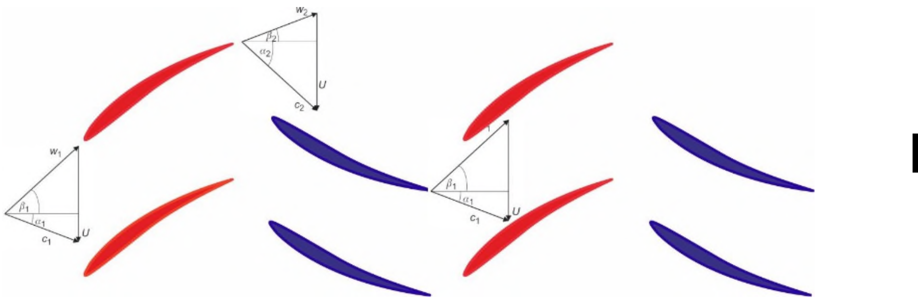
Lecture 3

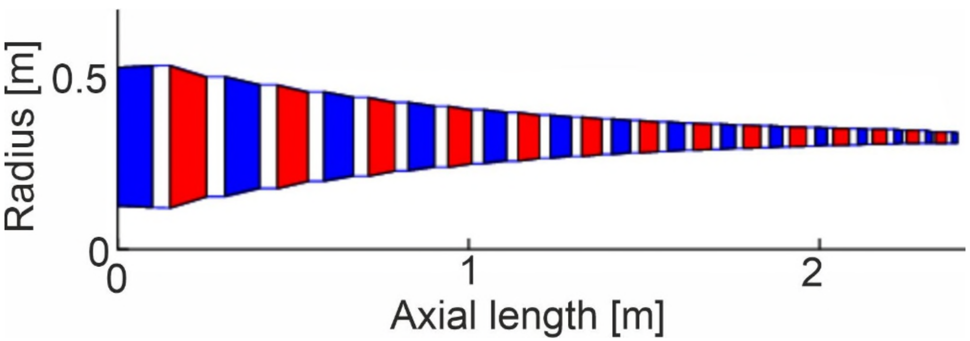
Three-Dimensional Flows in a Turbomachinery Blade Rows

What have we covered so far?

- Mean-line analysis ($\phi, \psi, R, \alpha_1 \rightarrow$ to fix the velocity triangles)
- Compressibility effects (to fix volume flow rate and area)
- Two-dimensional design (M number effects, correlations for losses, flow deviation, blockage)

- Three-dimensional nature of the flow
- Three-dimensional flows
- Three-dimensional design



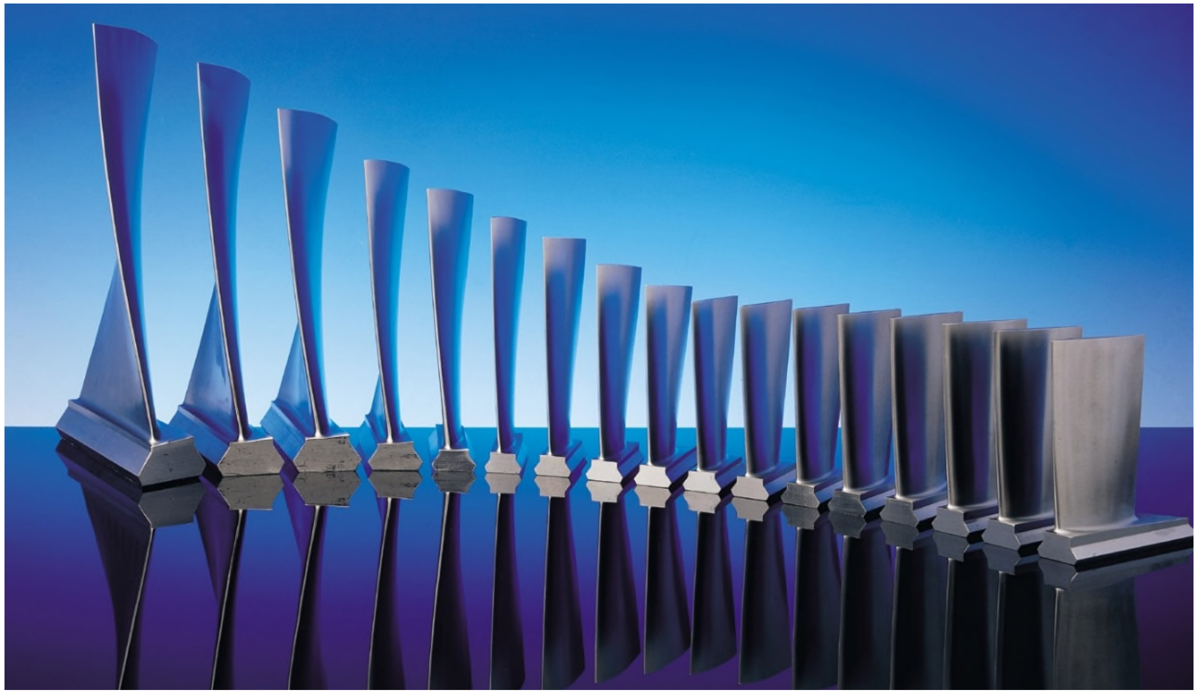


- The velocity triangles were considered at one particular radius (usually mean radius)

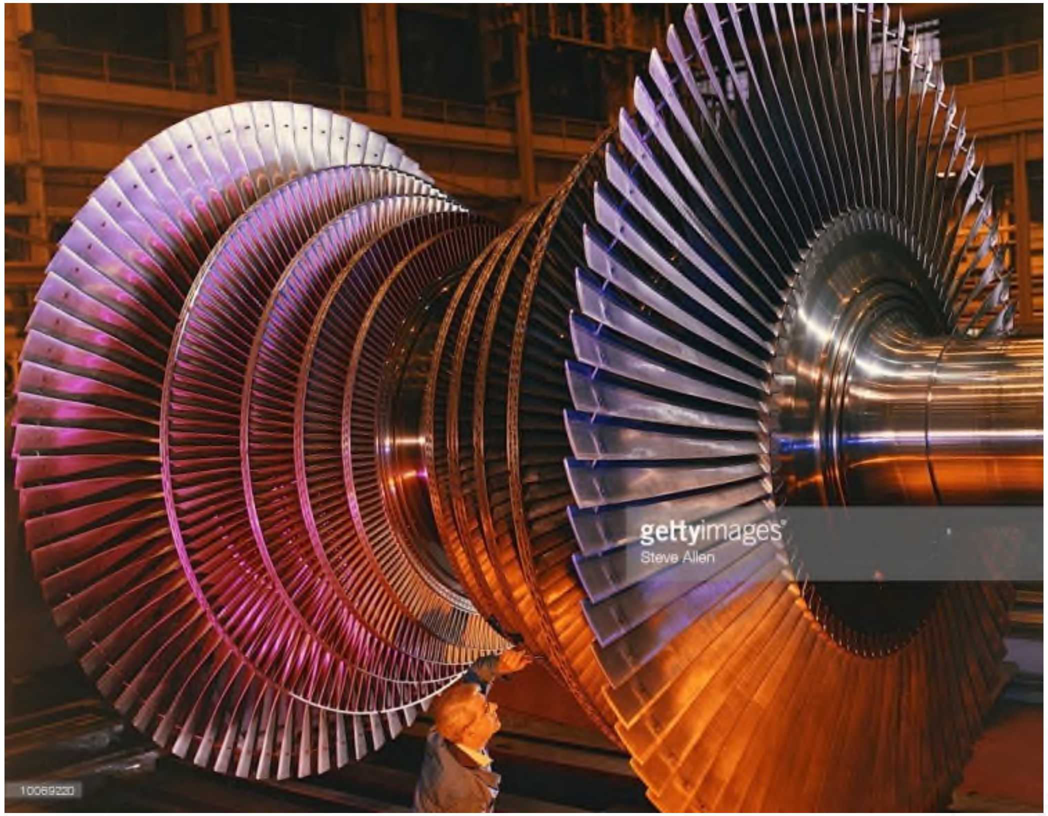
- Actually, 3D blades usually have a significant variation in velocity triangles from hub to tip (especially for high-aspect ratio blades)
- The velocity triangles distribution along the span defines the blade geometry

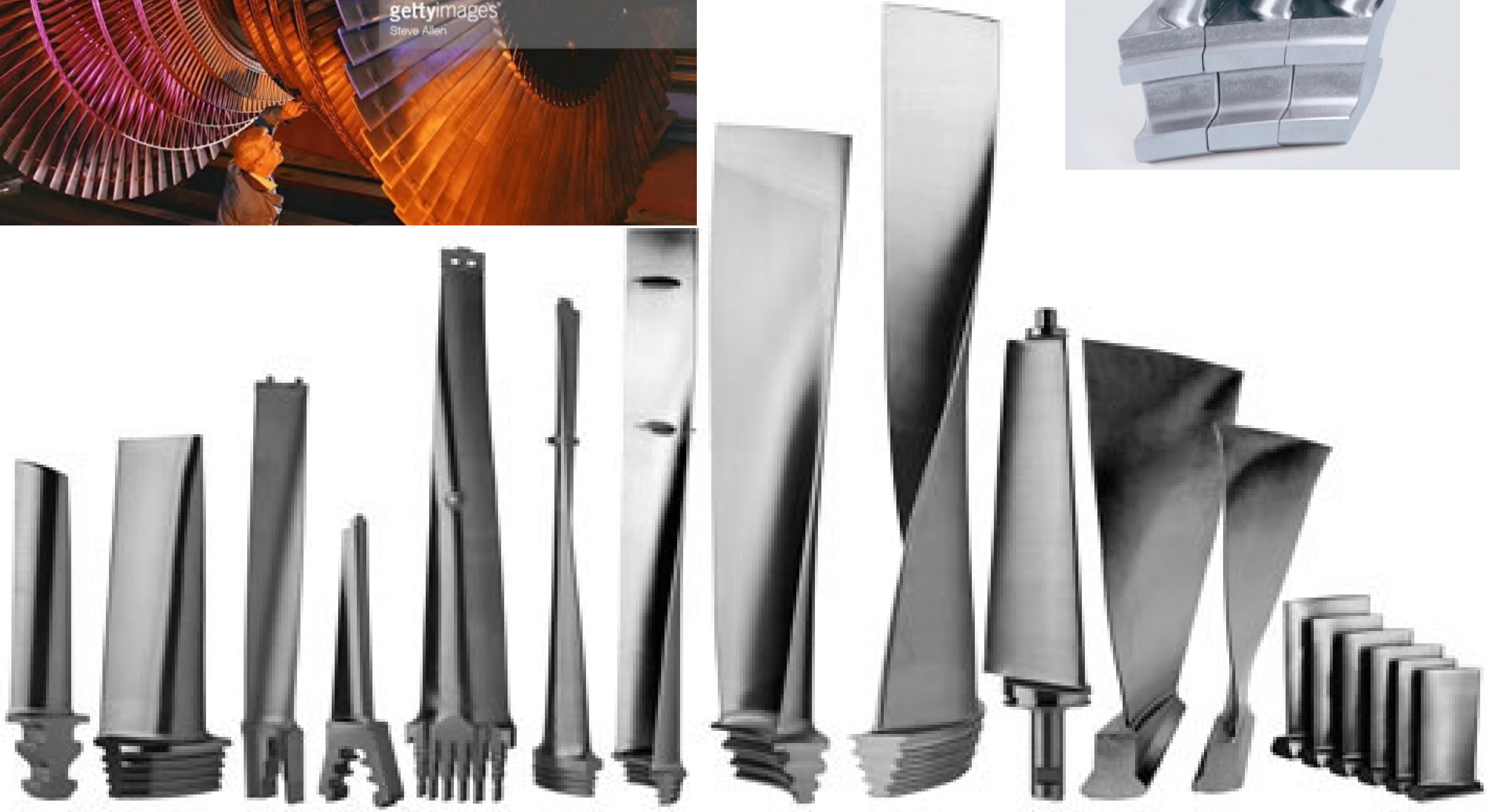
Turbofan Blades

Compressor Blades

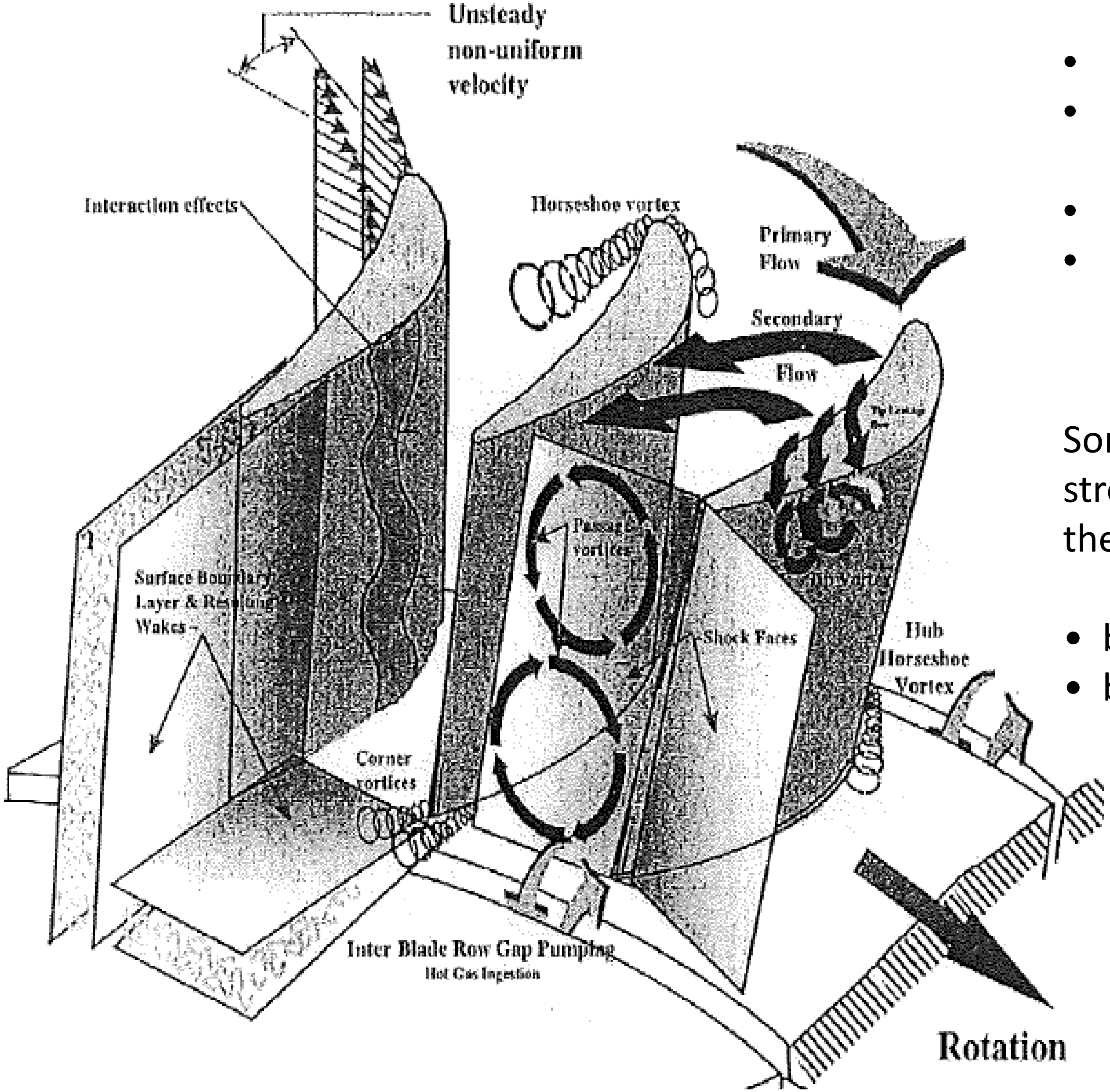


Steam Turbine Blades





Causes of Three-Dimensional Flows in a Turbomachinery Blade Rows



- Unsteadiness, due to relative motion of blade rows
- Secondary flows
- Vortical flow structures, spatial shear boundary layers, viscous effects
- Leakage flows
- Shock waves and their interactions with vortical flow structures.

Some geometrical features can also cause stream surface twist, three-dimensionality of the flow. The most important are:

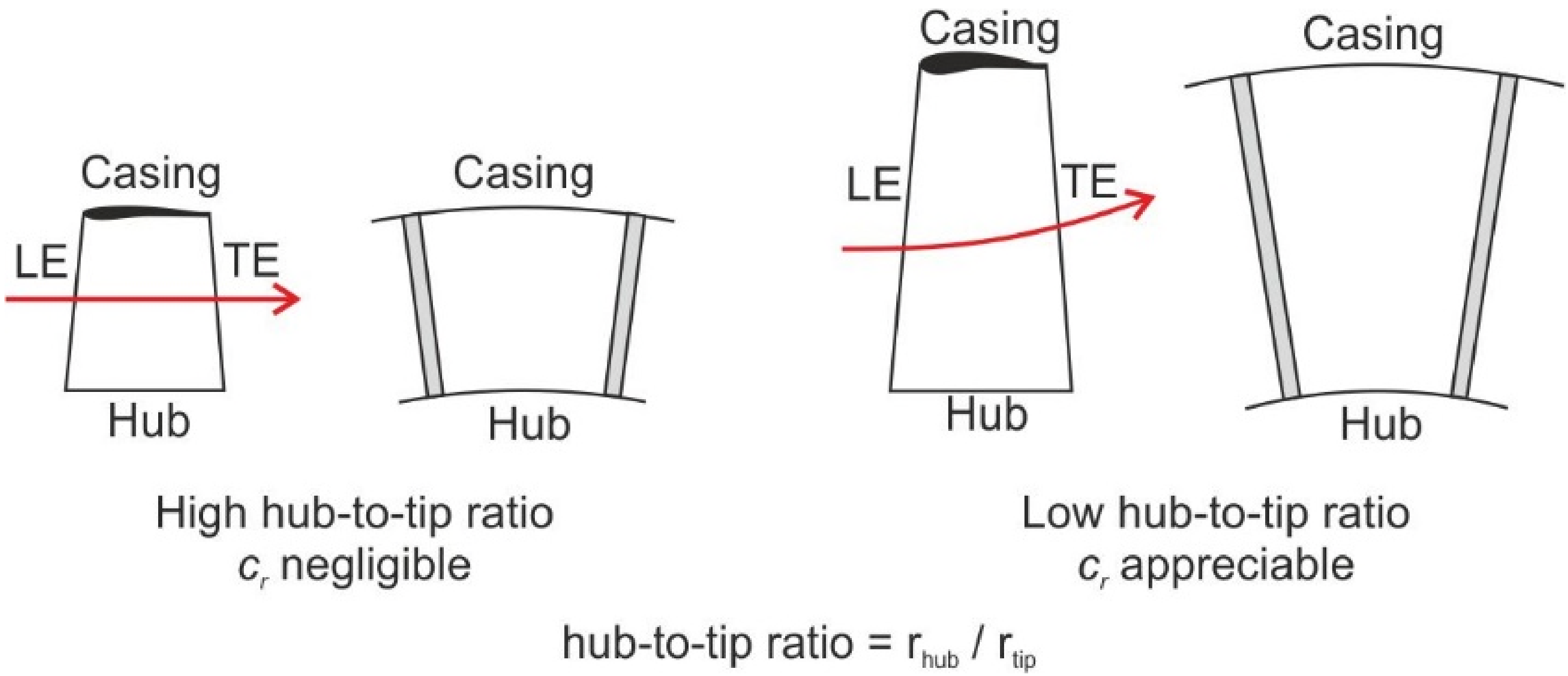
- blade sweep, and
- blade lean.

These geometrical features are sometimes utilised in turbomachinery design to control the flow.

3D Flows in Turbomachinery

So far the fluid motion through blade rows of axial turbomachines was assumed to be two-dimensional – the radial velocity component was neglected.

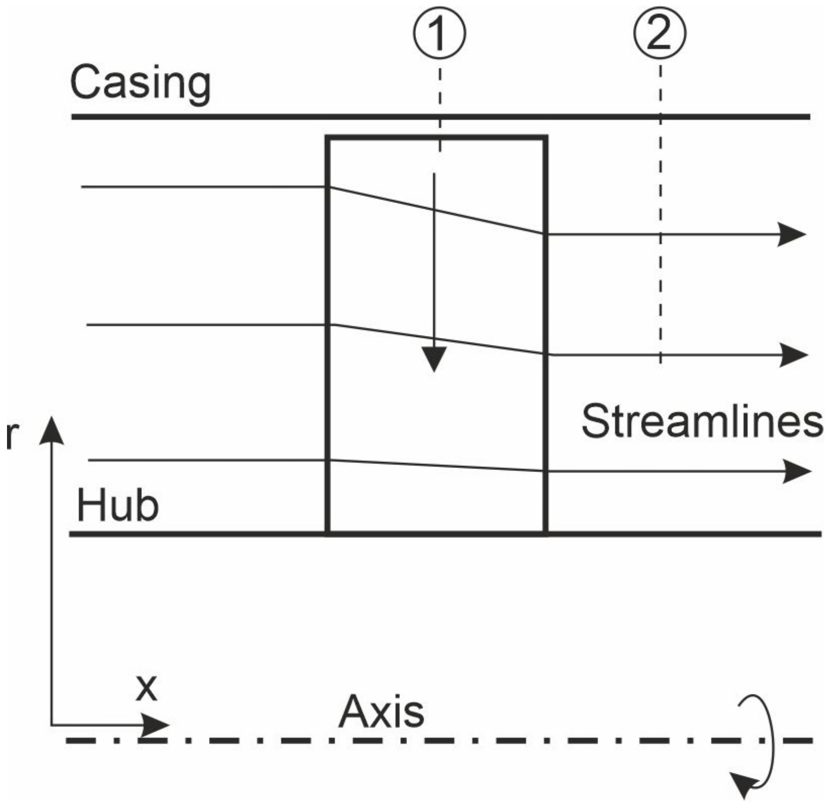
This assumption is reasonable for high hub-to-tip ratio. For low hub-to-tip ratio blades the radial velocity becomes significant to affect the outlet velocity and flow angle distribution.

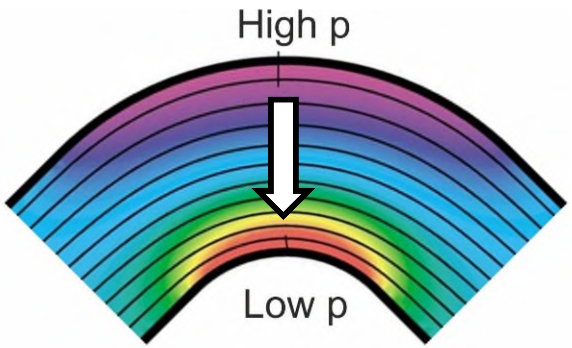


3D Flows in Turbomachinery

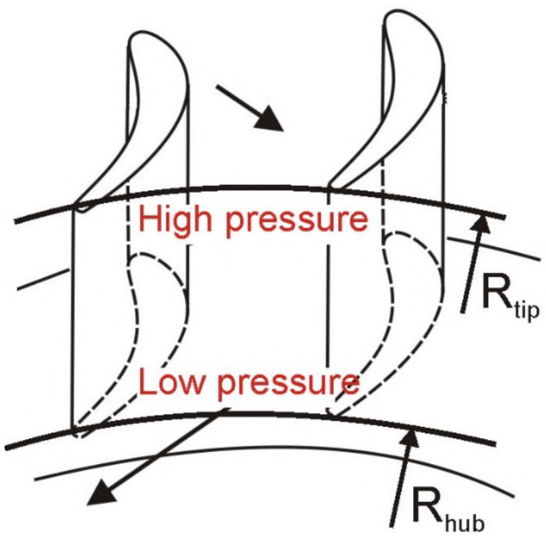
The temporary imbalance between the strong centrifugal forces exerted on the fluid and radial pressures restoring equilibrium is responsible for these radial flows.

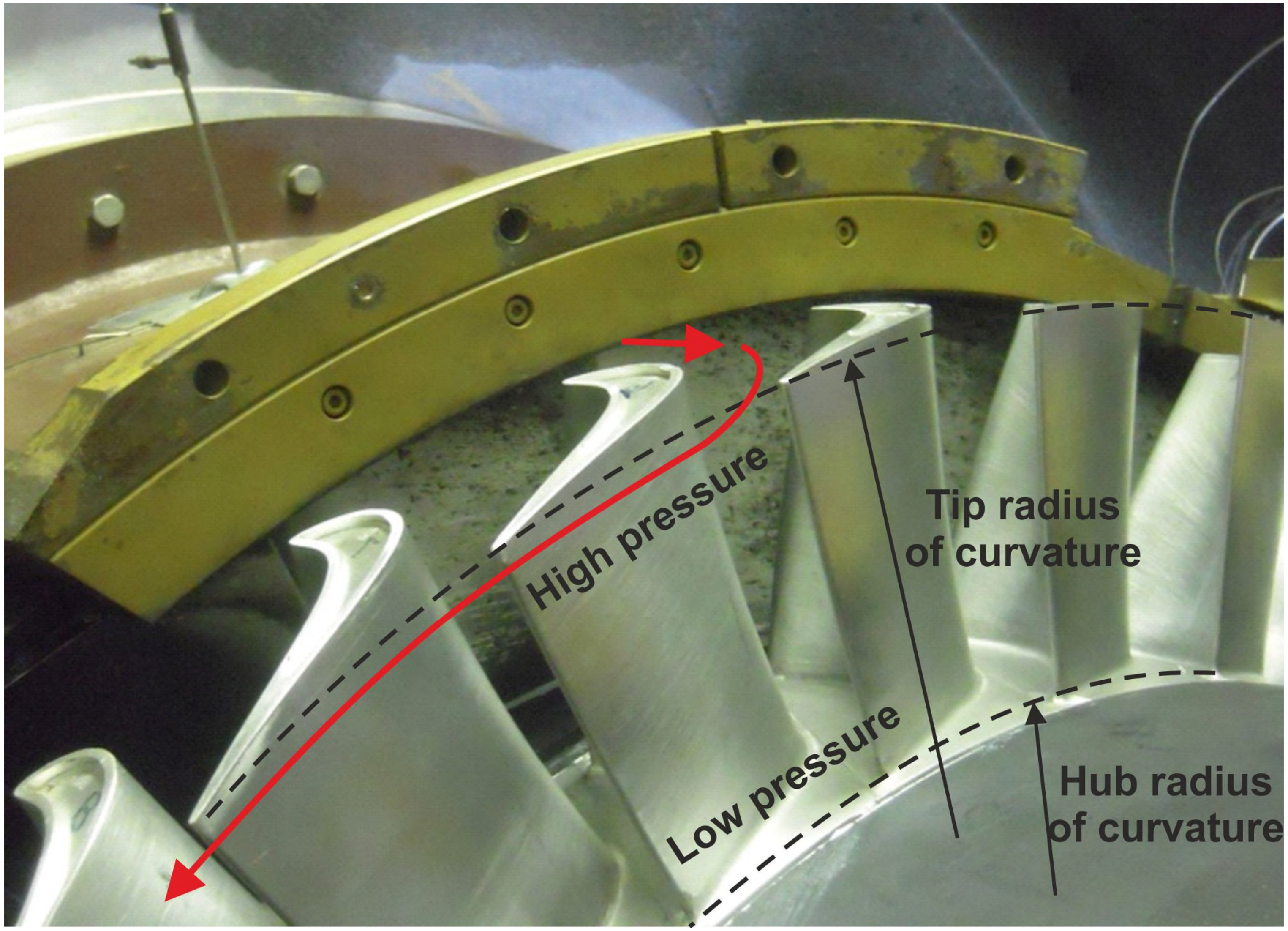
1) To an observer traveling with a fluid particle, within the blade passage, radial motion will continue until sufficient fluid is transported radially to change the pressure distribution to that necessary for equilibrium.
2) The flow in an annular passage in which there is no radial component of velocity, whose streamlines lie in circular, cylindrical surfaces and which is axisymmetric, is commonly known as *radial equilibrium flow*.



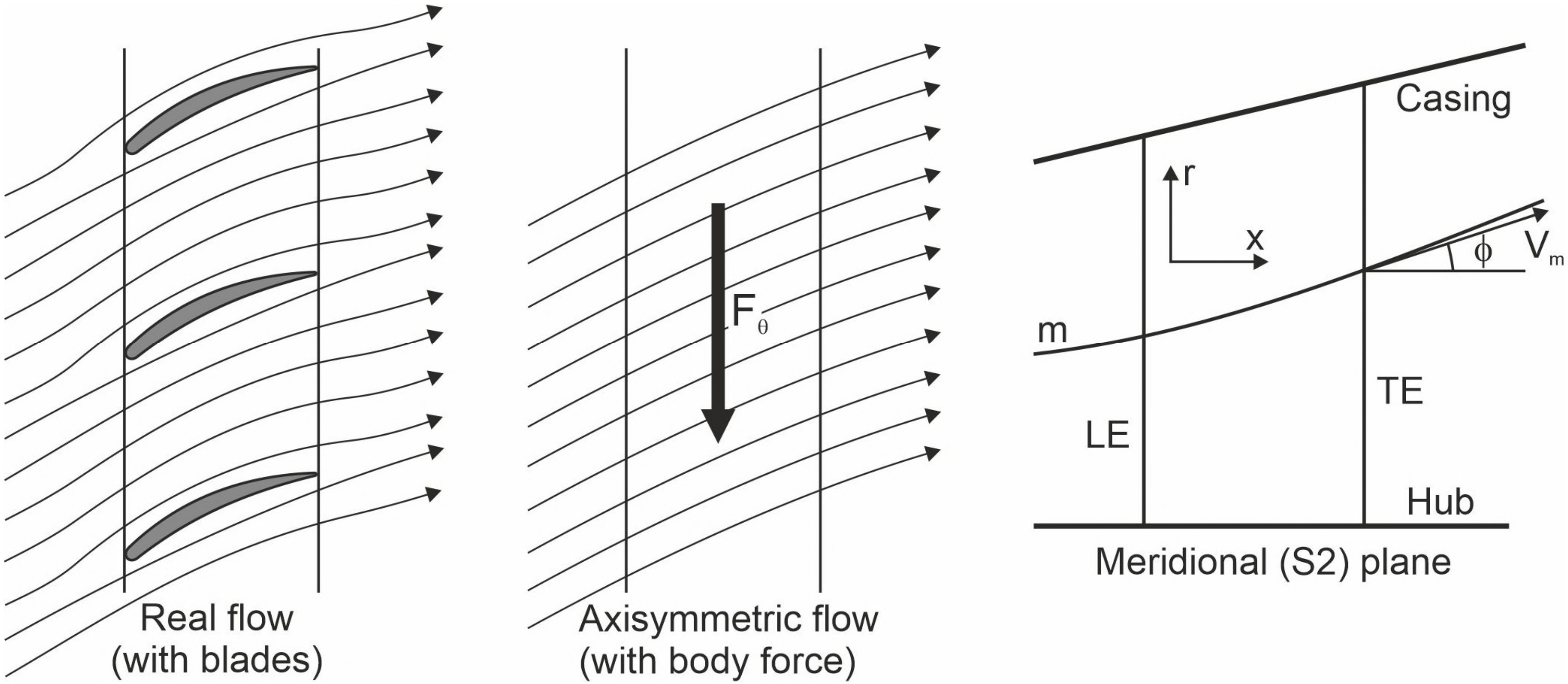


$$-\frac{1}{\rho}\left(\frac{\partial p}{\partial r}\right) = \frac{c^2}{R_c}$$





Meridional Streamline Curvature Equation (Radial Equilibrium) Physical interpretation



Meridional blade view

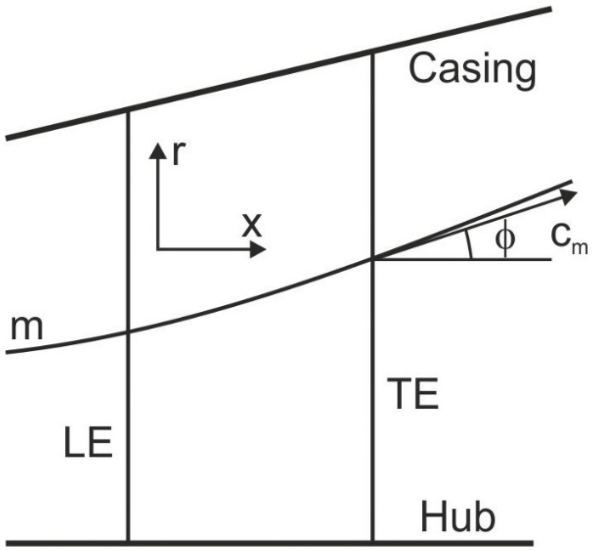
$$c_m = \sqrt{c_x^2 + c_r^2}$$

<u>Main assumption:</u>

Flow is axisymmetric – implying no circumferential variation of velocity or fluid properties $\left(\frac{\partial}{\partial\theta} = 0\right)$

Not a real picture. The effect of blade-to-blade pressure differences is represented by a body force, F, equal to the required change in tangential momentum to turn the flow

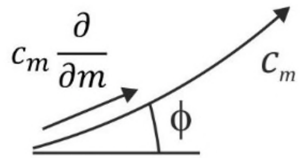
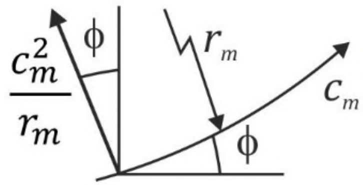
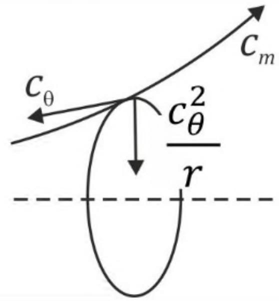
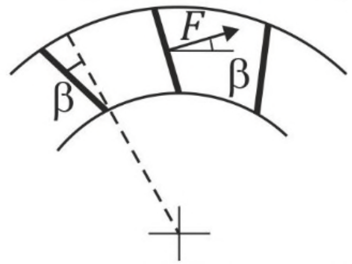
Meridional Streamline Curvature Equation (Radial Equilibrium)
Physical interpretation



Meridional (S2) plane

Radial momentum equation:

$$\boxed{c_m sin\phi \frac{\partial c_m}{\partial m} + \frac{c_m^2}{r_m} cos\phi - \frac{c_\theta^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + \frac{F_r}{\rho}}$$

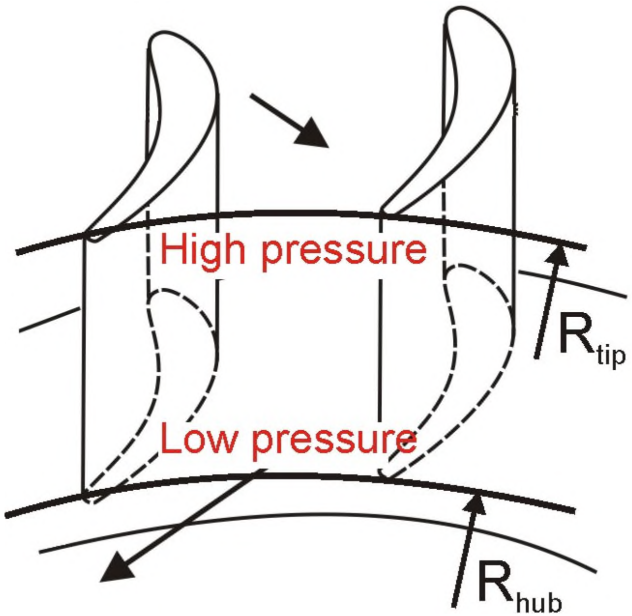
$c_m sin\phi \frac{\partial c_m}{\partial m}$	*Radial component of acceleration along the streamline*	
$\frac{c_m^2}{r_m} cos\phi$	*Radial component of centripetal acceleration due to meridional curvature*	
$-\frac{c_\theta^2}{r}$	*Radially inward centripetal acceleration due to swirl*	
$-\frac{1}{\rho}\frac{\partial p}{\partial r}$	*Radial pressure gradient (provides majority of force)*	
$\frac{F_r}{\rho}$	*Blade force F acts normal to blade and may have radial component $F_r = F sin\beta$ (β = blade lean)*	

Reduction to Simple Radial Equilibrium Equation

$$\boxed{c_m sin\phi \frac{\partial c_m}{\partial m} + \frac{c_m^2}{r_m} cos\phi - \frac{c_\theta^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + \frac{F_r}{\rho}}$$

$$c_m sin\phi \frac{\partial c_m}{\partial m} \begin{cases} = 0 & if\ \phi = 0\ \ (cylindrical\ surface) \\ small & if\ \partial c_m/\partial m\ \ is\ small \end{cases}$$

$$\frac{c_m^2}{r_m} cos\phi \begin{cases} = 0\ \ if\ parallel\ streamlines\ (r_m \to \infty) \\ small\ if\ r_m \gg r\ (compared\ to\ c_\theta^2/r) \end{cases}$$



$$Radially\ stacked\ blades: Fr = 0$$

$$\cancel{c_m sin\phi \frac{\partial c_m}{\partial m}} + \cancel{\frac{c_m^2}{r_m}} cos\phi - \frac{c_\theta^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + \cancel{\frac{F_r}{\rho}}$$

$$\boxed{-\frac{c_\theta^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r}}$$

Alternative Derivation of Simple Radial Equilibrium Equation

Small element of fluid of mass dm. The element is in radial equilibrium – the pressure forces balance the centrifugal forces.

$$(p + dp)(r + dr)d\theta - prd\theta - \left(p + \frac{1}{2}dp\right)drd\theta = dmc_\theta^2/r$$

$$dm = \rho rd\theta dr$$

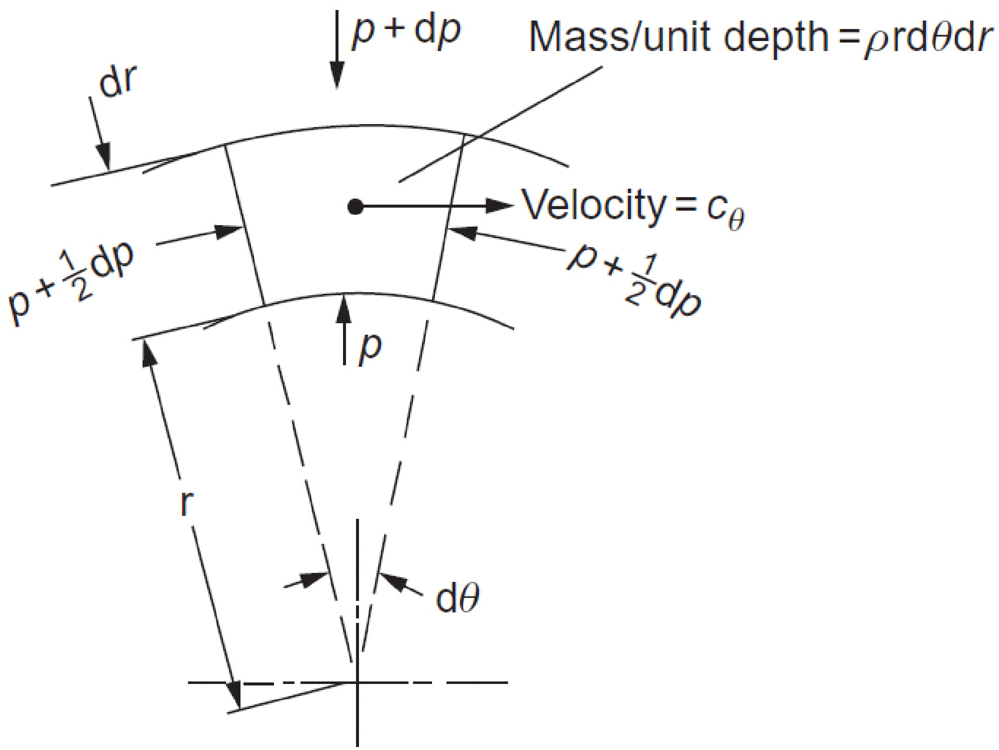
Ignore the second order terms: $$\boxed{\frac{1}{\rho}\frac{dp}{dr} = \frac{c_\theta^2}{r}} \qquad (1)$$

If the swirl velocity c_θ and density are known functions of the radius, the radial pressure variation along the blade length can be determined as:

$$p_{tip} - p_{root} = \int_{root}^{tip} \rho\, c_\theta^2 \frac{dr}{r} \qquad (2)$$

For incompressible fluid:

$$p_{tip} - p_{root} = \rho \int_{root}^{tip} c_\theta^2 \frac{dr}{r}$$



The stagnation enthalpy (assuming $c_r = 0$) $$h_0 = h + \frac{1}{2}(c_x^2 + c_\theta^2) \quad (3)$$

$$\frac{dh_0}{dr} = \frac{dh}{dr} + c_x \frac{dc_x}{dr} + c_\theta \frac{dc_\theta}{dr} \quad (4)$$

$$\frac{1}{\rho}\frac{dp}{dr} = \frac{c_\theta^2}{r}$$

Using: $Tds = dh - \frac{dp}{\rho}$ $$T\frac{ds}{dr} = \frac{dh}{dr} - \frac{1}{\rho}\frac{dp}{dr} \quad (5)$$

The radial equilibrium may be obtained in form (by eliminating dp/dr and dh/dr):

$$\boxed{\frac{dh_0}{dr} - T\frac{ds}{dr} = c_x \frac{dc_x}{dr} + \frac{c_\theta}{r}\frac{d}{dr}(rc_\theta)} \quad (6a)$$

If the stagnation enthalpy, h_0 and entropy, s, remain constant along the radius: $dh_0/dr = ds/dr = 0$

$$\boxed{c_x \frac{dc_x}{dr} + \frac{c_\theta}{r}\frac{d}{dr}(rc_\theta) = 0} \quad (6b)$$

This will hold for the flow between the rows of an adiabatic, reversible (ideal) turbomachine in which rotor rows either deliver or receive equal work at all radii. If the flow is incompressible, instead of Eq. (3) use $p_0 = p + (1/2)\rho(c_x^2 + c_\theta^2)$:

$$\frac{1}{\rho}\frac{dp_0}{dr} = \frac{1}{\rho}\frac{dp}{dr} + c_x \frac{dc_x}{dr} + c_\theta \frac{dc_\theta}{dr} \quad (7)$$

Combining with $\frac{1}{\rho}\frac{dp}{dr} = \frac{c_\theta^2}{r}$

$$\frac{1}{\rho}\frac{dp_0}{dr} = c_x \frac{dc_x}{dr} + \frac{c_\theta}{r}\frac{d}{dr}(rc_\theta) \quad (8)$$

Equation (8) reduces to Eq. (6b) in a turbomachine in which equal work is delivered at all radii and the total pressure losses across a row are uniform with radius.

The equation 6a: $\boxed{\frac{dh_0}{dr} - T\frac{ds}{dr} = c_x\frac{dc_x}{dr} + \frac{c_\theta}{r}\frac{d}{dr}(rc_\theta)}$ can be used in:

Design (indirect) mode The swirl velocities, rc_θ, are specified through the machine (i.e. fixed the work distribution) and equation is used to predict the required blade angles.

Analysis (direct) mode We hold geometry (relative flow angles) fixed and use the equation to analyse velocity (pressure) distribution through the machine.

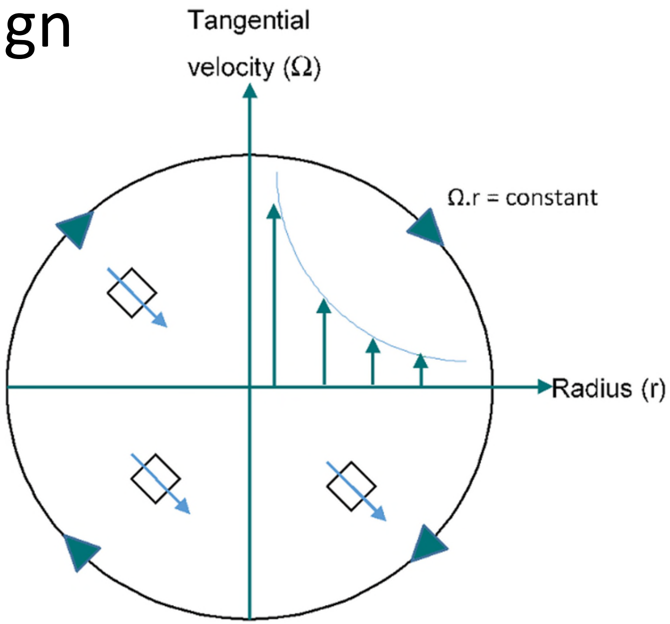
Vortex Distribution

The variation of tangential velocity component, c_θ, along the blade span, known as ***the vortex distribution***, determines the axial velocity and the pressure distribution once the loss and h_0 distributions are known.

The choice of the vortex distribution is essential for determining turbomachinery design.

Different vortex designs (distribution of c_θ) can be considered. It is theoretically possible to prescribe the rotor exit flow to give uniform work along blade height .

The Design Mode – Vortex Design



Free-Vortex Flow

A flow where the product of radius and tangential velocity remains constant (i.e., $rc_\theta = K$, a constant). The term *vortex free* is appropriate as the vorticity (to be precise we mean axial vorticity component) is then zero.

Consider an element of an ideal inviscid fluid rotating about some fixed axis. The circulation Γ is defined as the line integral of velocity around a curve enclosing an area A, or $\Gamma = \oint c\,ds$. The *vorticity* at a point is defined as the limiting value of circulation δΓ divided by area δA, as δA becomes vanishingly small.

Thus, vorticity, $\omega = d\Gamma/dA$.

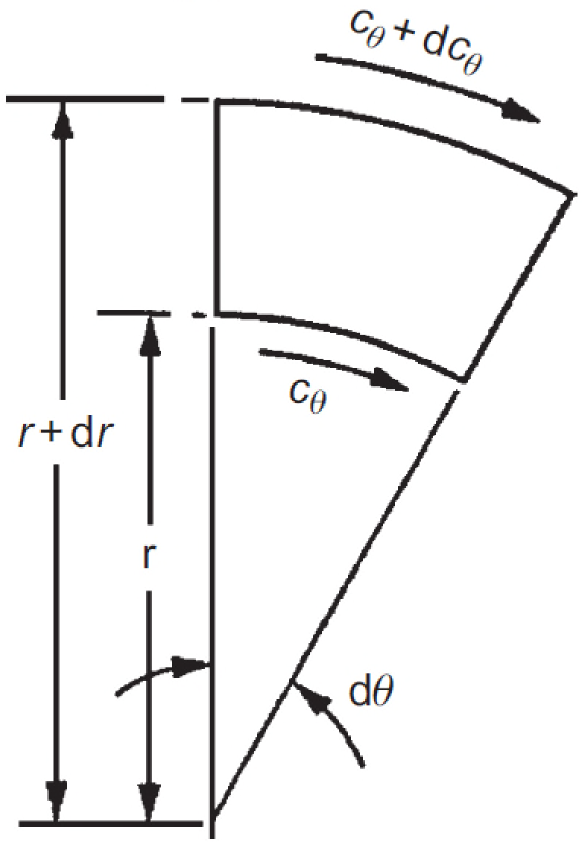
For the element in the figure, $c_r = 0$

$$d\Gamma = (c_\theta + dc_\theta)(r + dr)d\theta - c_\theta r d\theta = \left(\frac{dc_\theta}{dr} + \frac{c_\theta}{r}\right) r\,d\theta\,dr$$

(ignoring, small, second order terms)

$$d\Gamma = (c_\theta + dc_\theta)(r + dr)d\theta - c_\theta r d\theta = \left(\frac{dc_\theta}{dr} + \frac{c_\theta}{r}\right) r\,d\theta\,dr$$

$$\frac{1}{r}\frac{d}{dr}(rc_\theta) = \frac{1}{r}\left(r\frac{dc_\theta}{dr} + c_\theta\frac{dr}{dr}\right) = \frac{1}{r}\left(r\frac{dc_\theta}{dr} + c_\theta\right)$$



$\omega = \frac{d\Gamma}{dA} = \frac{1}{r}\frac{d(rc_\theta)}{dr}$ If the vorticity is zero, $d(rc_\theta)/dr$ is zero, and therefore rc_θ is constant with radius

Putting $rc_\theta = const$ in Eq. (6b), then $dc_x/dr = 0$ and so $c_x = const$.

This can be applied to the incompressible flow through a free-vortex compressor or turbine stage, enabling the radial variation in flow angles, reaction, and work to be found.

Compressor Stage

Consider the case of a compressor stage in which $rc_{\theta 1} = K_1$ before the rotor and $rc_{\theta 2} = K_2$ after the rotor, where K_1 and K_2 are constants. The work done by the rotor on unit mass of fluid is

$$\Delta W = U(c_{\theta 2} - c_{\theta 1}) = \Omega r\left(\frac{K_2}{r} - \frac{K_1}{r}\right) = constant$$

Thus, the work done is equal at all radii.

The relative flow angles (Compressor stage velocity triangles, Lecture 1) upstream and downstream of rotor:

$$tan\beta_1 = \frac{U}{c_x} - tan\alpha_1 = \frac{\Omega r - K_1/r}{c_x} \qquad (tan\alpha_1 = \frac{c_{\theta 1}}{c_x})$$

$$tan\beta_2 = \frac{U}{c_x} - tan\alpha_2 = \frac{\Omega r - K_2/r}{c_x}$$

In which $c_{x1} = c_{x2} = c_x$ for incompressible flow.

Stage reaction in an axial compressor: $R = \frac{\Delta h_{rotor}}{\Delta h_{stage}}$

For a repeating stage where $\alpha_1 = \alpha_3$ with c_x constant across the stage, the reaction can be shown to be:

$$R = \frac{c_x}{2U}(tan\beta_1 + tan\beta_2)\ \ check\ B19\ lecture\ notes\ for\ this$$

$$R = \frac{c_x}{2U}\left(\frac{\Omega r - K_1/r}{c_x} + \frac{\Omega r - K_2/r}{c_x}\right) = \frac{1}{2\Omega r}\left(2\Omega r\ - \frac{K_1 + K_2}{r}\right) = 1 - \frac{K_1 + K_2}{2\Omega r^2} = 1 - \frac{k}{r^2}$$

Where: $k = \frac{K_1 + K_2}{2\Omega}$

$\boxed{k > 0\text{, the stage reaction increases from root to tip.}}$

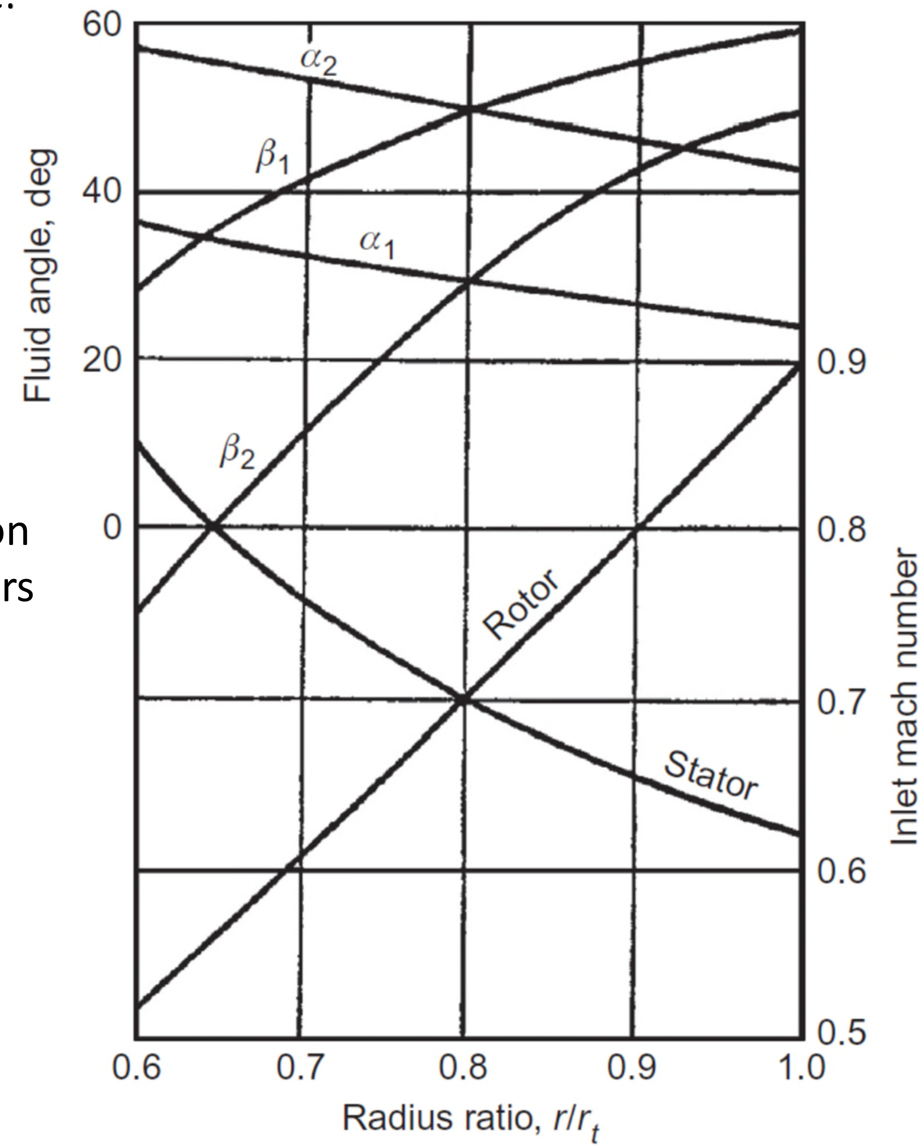
From (1) we observe that as c_θ^2/r is always positive, so static pressure increases from root to tip.

The simplicity of the flow under free-vortex conditions is attractive to the designer and many compressors have been designed to conform to this flow.

Characteristic of this flow are the large fluid deflections near the inner wall and high Mach numbers near the outer wall, both effects being detrimental to efficient performance.

A further serious disadvantage is the large amount of rotor twist from root to tip, which adds to the difficulty of blade manufacture.

The figure illustrates the variation of fluid angles and Mach numbers of a typical compressor stage designed for free vortex flow.



Example

An axial compressor stage is designed to have free-vortex tangential velocity distributions for all radii before and after the rotor blade row. Both the tip diameter (1.0 m) and the hub diameter (0.9 m) are constant for the stage. At the rotor tip, the flow angles are as follows:
absolute inlet angle, $\alpha_1 = 30°$;
relative inlet angle, $\beta_1 = 60°$;
absolute outlet angle, $\alpha_1 = 60°$;
relative outlet angle, $\beta_2 = 30°$.

Determine:
a. the axial velocity;
b. the mass flow rate;
c. the power absorbed by the stage;
d. the flow angles at the hub;
e. the reaction ratio of the stage at the hub;

The rotational speed of the rotor is 6000 rev/min and the gas density is 1.5 kg/m^3, which can be assumed constant for the stage. It can be further assumed that stagnation enthalpy and entropy are constant before and after the rotor row for the purpose of simplifying the calculations.

Solution

a) The rotational speed: $\Omega = 2\pi N/60 = 628.4\ rad/s$

$Blade\ tip\ speed$: $U_t = \Omega r_t = 314.2\ m/s$ $Blade\ hub\ speed$: $U_h = \Omega r_h = 282.5\ m/s$

From the velocity triangles: $U_t = c_x(tan60° + tan30°) = 2.309c_x$

$c_x = 136\ m/s$ (constant at all radii because the flow is a free vortex)

b) The mass flow rate $\dot{m} = \pi\left(r_t^2 - r_h^2\right)\rho c_x = \pi(0.5^2 - 0.45^2)1.5 \cdot 136 = 30.4\ kg/s$

c) The power absorbed by the stage: $\dot{W} = \dot{m}U_t(c_{\theta 2t} - c_{\theta 1t}) = \dot{m}U_t c_x(tan\alpha_{2t} - tan\alpha_{1t})$

$= 30.4 \cdot 314.2 \cdot 136\left(\sqrt{3} - 1/\sqrt{3}\right) = 1.5\ MW$

d) At the inlet to the rotor tip: $c_{\theta 1t} = c_x tan\alpha_1 = 136/\sqrt{3} = 78.6\ m/s$

The absolute flow if a free vortex, $rc_\theta = constant$.
Therefore, $c_{\theta 1h} = c_{\theta 1t}(r_t/r_h) = 78.6(0.5/0.45) = 87.3\ m/s$

At outlet to the rotor tip: $c_{\theta 2t} = c_x tan\alpha_2 = 136\sqrt{3} = 235.6\ m/s$

$c_{\theta 2h} = c_{\theta 2t}(r_t/r_h) = 235.6(0.5/0.45) = 262\ m/s$

The flow angles at the hub: $tan\alpha_1 = c_{\theta 1h}/c_x = 87.3/136 = 0.642 \quad \rightarrow \alpha_1 = 32.75°$

$$tan\beta_1 = U_h/c_x - tan\alpha_1 = 1.436 \rightarrow \beta_1 = 55.15°$$

$$tan\alpha_2 = c_{\theta 2h}/c_x = 262/136 = 1.928 \rightarrow \alpha_2 = 62.6°$$

$$tan\beta_2 = U_h/c_x - tan\alpha_2 = 0.152 \rightarrow \beta_2 = 8.64°$$

e) The reaction at the hub: $R = 1 - \dfrac{k}{r^2}$

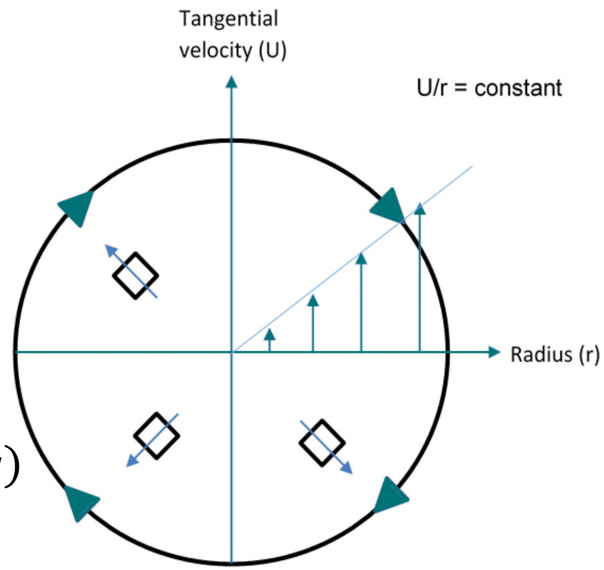
From the symmetry of the velocity triangles at the tip region → $R = 0.5, \quad r = r_t, \quad k = 0.5r_t^2$

$$r_h = 1 - 0.5\left(\frac{0.5}{0.45}\right)^2 = 0.382m$$

Different Vortex Designs

Vortex distribution: variation of c_θ along the blade span

Many different types of vortex design have been proposed to overcome some of the disadvantages of free-vortex design.



Forced Vortex

Sometimes called solid-body rotation because c_θ varies directly with r.
At the entry to the rotor assume $h_{01} = const$ and $c_{\theta 1} = K_1 r$.
It reduces the pressure gradient at the hub (removes problem of free-vortex design)

Variable (Combined) Vortex Design

The tangential velocity distribution is:

$$c_{\theta 1} = ar^n - b/r \quad (before\ rotor)$$
$$c_{\theta 2} = ar^n + b/r \quad (after\ rotor)$$

The work distribution for all values of the index n is constant with radius so the if h_{01} is uniform h_{02} is also uniform with radius.

$$\Delta W = h_{02} - h_{01} = U(c_{\theta 2} - c_{\theta 1}) = 2b\Omega$$

Selecting different values of n gives several of the tangential velocity distributions commonly used in compressor design.
With $n = 0$, or ***zero power blading***, it leads to the so-called exponential type of stage design. With $n = 1$, or ***first power blading***, the stage design is called (incorrectly) constant reaction.

Mixed Vortex Design

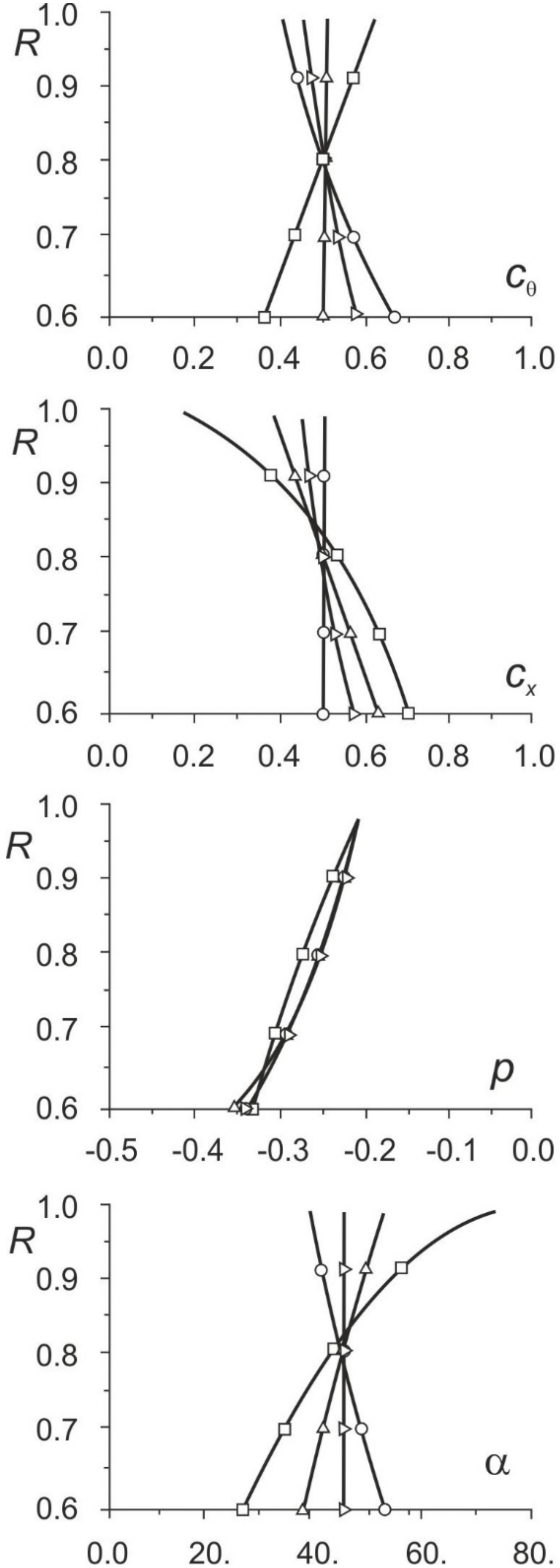
A problem arises with the use of the free-vortex design, (especially when it is applied to low hub-to-tip ratio stages), which is the large radial variation in flow angles, reaction, and tangential velocities. This leads to highly twisted blades which are difficult and expensive to manufacture and can be the cause of large total pressure losses. Various strategies have been recommended to reduce these flow extremes.

One of these attempts is the use of the so-called “mixed vortex,” which combines a free vortex with a forced vortex or solid-body rotation. For the flow after a rotor row, this combined flow produces a tangential velocity distribution given by

$$c_{\theta 2} = \frac{a}{r} + br$$

Comparison of Different Vortex Designs

(45° swirl, 0.6 Hub-to-Tip Ratio)



○ *Free-vortex design*
□ *Forced-vortex design*
△ *Constant c_θ design*
▷ *Constant angle design*

Free vortex: *rc_θ = const*

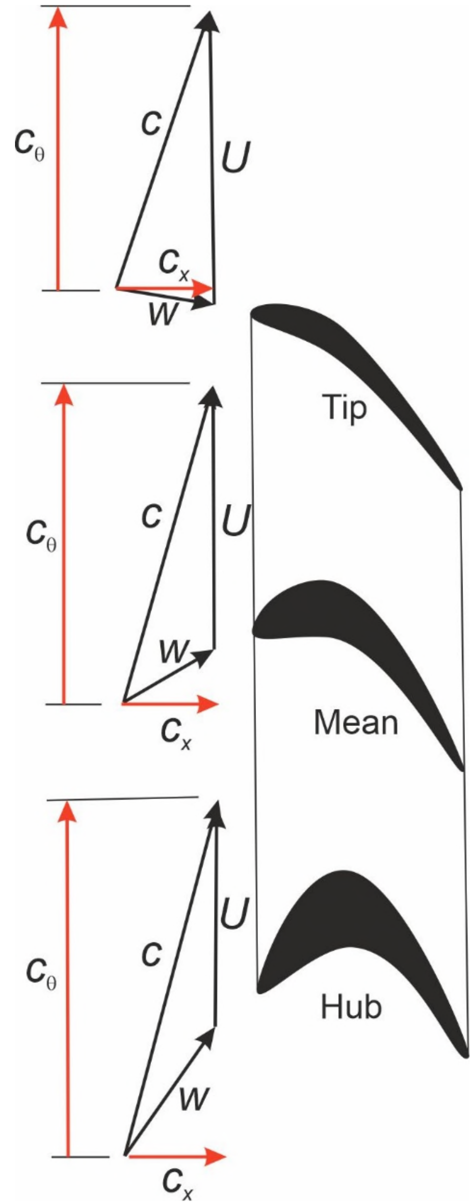
- *Large pressure gradient variation at hub*
- *Constant c_x*
- *Change in α compatible with c_θ*

Forced vortex: *c_θ/r = const*

- *Relief hub pressure gradient*
- *Steep cx distribution forcing too much flow through hub*
- *Large α variation*

Other dwo designs have moderate variation

Example of free-vortex design



The Analysis (Direct) Mode

The flow angle variation is specified in the direct problem and the radial equilibrium equation enables the solution of c_x and c_θ to be found.

The radial equilibrium equation is:

$$c_x \frac{dc_x}{dr} + \frac{c_\theta}{r}\frac{d}{dr}(rc_\theta) = \frac{dh_0}{dr} - T\frac{ds}{dr}$$

Substituting $c_\theta = csin\alpha$ and $c_x = ccos\alpha$:

$$c_x \frac{dc_x}{dr} + \frac{csin\alpha}{r}\frac{d}{dr}(rcsin\alpha) = \frac{dh_0}{dr} - T\frac{ds}{dr}$$

$$ccos\alpha \frac{d}{dr}(ccos\alpha) + \frac{csin\alpha}{r}\frac{d}{dr}(rcsin\alpha) = \frac{dh_0}{dr} - T\frac{ds}{dr}$$

$$\frac{csin\alpha}{r}\left[csin\alpha + rsin\alpha\frac{dc}{dr} + rccos\alpha\frac{d\alpha}{dr}\right] + ccos\alpha\left[\frac{dc}{dr}cos\alpha - csin\alpha\frac{d\alpha}{dr}\right] = \frac{dh_0}{dr} - T\frac{ds}{dr}$$

Multiplying out and simplifying:

$$c\frac{dc}{dr} + \frac{c^2}{r}sin^2\alpha = \frac{dh_0}{dr} - T\frac{ds}{dr}$$

Special cases:

1. If both dh_0/dr and ds/dr are zero, by integrating: $lnc = -\int sin^2\alpha \frac{dr}{r} + constant$

If $c = c_m$ at $r = r_m$, then: $\frac{c}{c_m} = exp\left(-\int sin^2\alpha \frac{dr}{r}\right)$

2. If the flow angle α is made constant, then the above equation simplifies to:

$$\frac{c}{c_m} = \frac{c_x}{c_{xm}} = \frac{c_\theta}{c_{\theta m}} = \left(\frac{r}{r_m}\right)^{-sin^2\alpha}$$

This vortex distribution is often employed in practice as untwisted blades are much simpler to manufacture.

Summary of Simple Radial Equilibrium

The equation is concerned with the way in which swirling flow through turbomachines is matched radially:

i) The radial pressure gradient provides the force required for the swirling flow (centripetal acceleration)

ii) The axial velocity must be consistent with the pressure field, tangential velocity and the stagnation pressure

iii) The total mass flow must be correct

How to use the Simple Radial Equilibrium Equation:

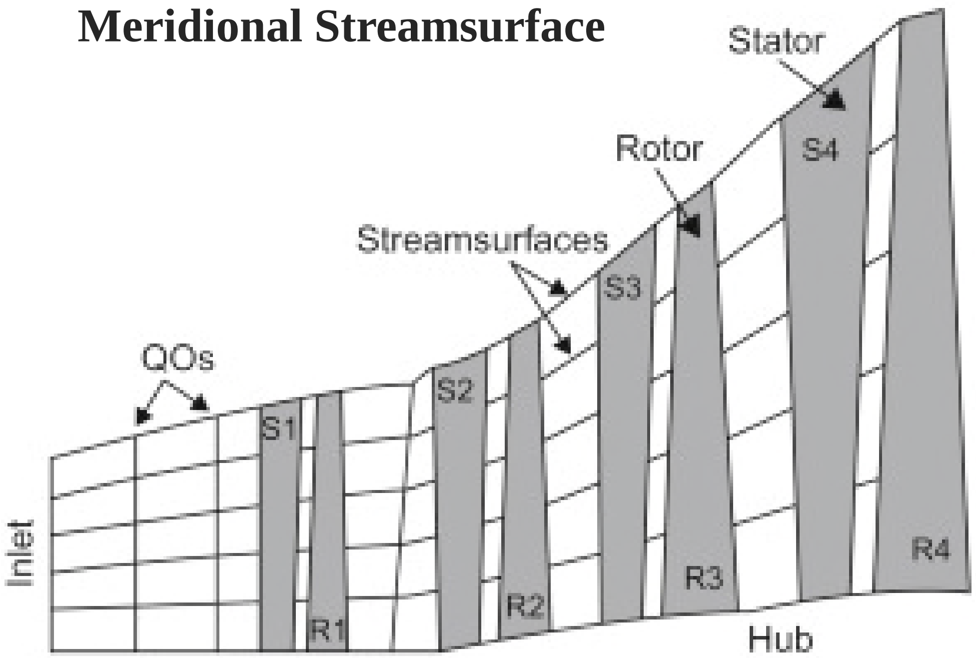
Given the tangential velocity distribution $c_\theta(r)$, known as vortex design, and the stagnation enthalpy and entropy (losses) (commonly both could be assumed to be uniform), then:

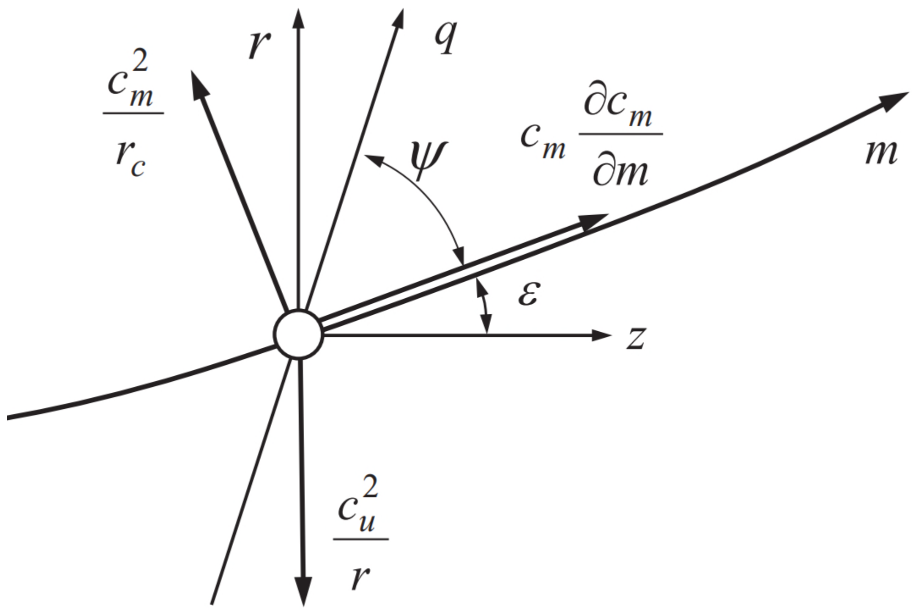
a) Solve the Simple Radial Equilibrium Equation to obtain $c_x(r)$, up to an arbitrary constant c_{xref}^2

b) Find the arbitrary constant from mass conservation.

Streamline Curvature Throughflow Method (SCM)

The flow field is represented as a set of radially distributed meridional stream surfaces (these are surfaces of revolution, along which fluid is assumed to move through the machine) set at equal mass intervals.





The principle of SCM: Inviscid pitch-wise averaged radial momentum conservation along lines roughly perpendicular to stream surfaces, QOs – quasi-orthogonals

$$c_m \frac{dc_m}{dq} = \frac{dh_0}{dq} - T\frac{ds}{dq} - \frac{1}{2r^2}\frac{d\left(r^2 c_\theta^2\right)}{dq} + sin\psi \frac{c_m^2}{r_c} + cos\psi c_m \frac{\partial c_m}{\partial m} + tan\gamma \frac{c_m}{r}\frac{\partial (rc_\theta)}{\partial m}$$

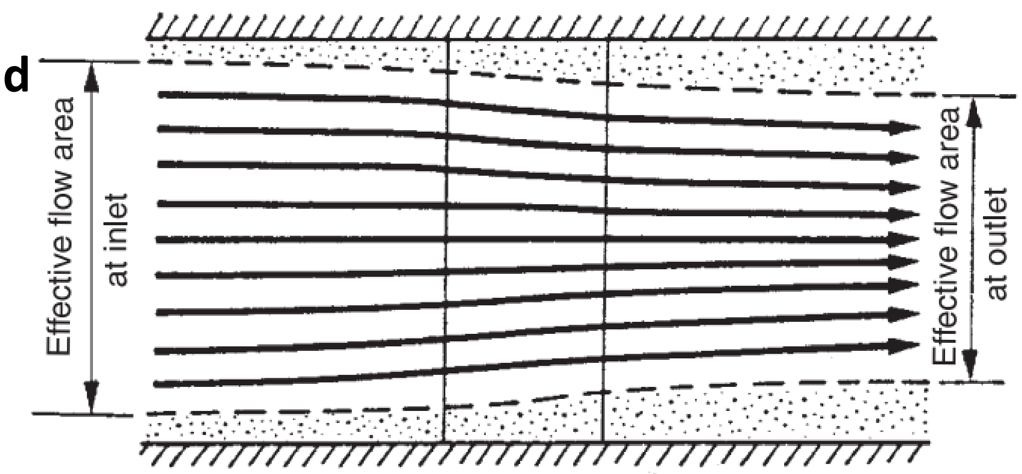
This equation for the variation of c_m along a QO is subject to continuity constraint, i.e. must be solved together with **the continuity equation for the flow across the QO:**

$$\int_{hub}^{casing} \rho c_m sin\psi (2\pi r - Zt_u)(1-B)dq = \dot{m}$$

Streamline Curvature Throughflow Method (SCM)

Empirical Correlations (based on know-how and manufacturing experience) are used for:

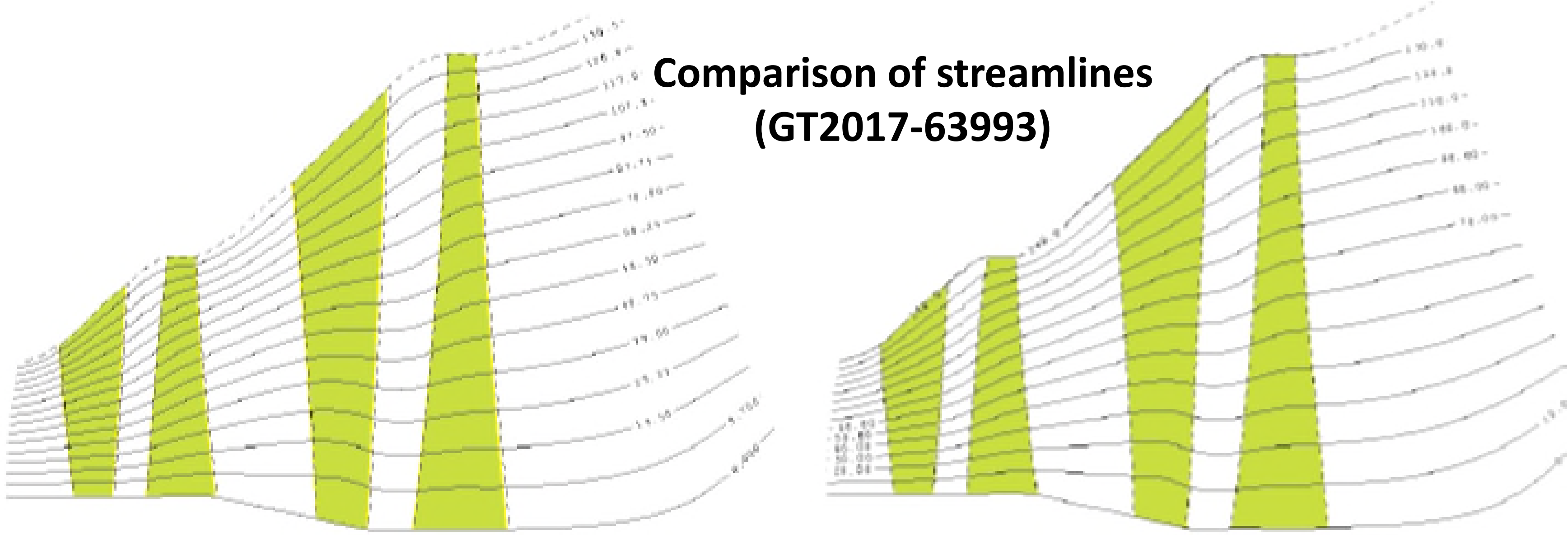
- Blockage
- Deviation and incidence
- Supersonic deviation and unique incidence
- Spanwise mixing
- Radial loss distribution
- Model "3-D effects"



Blockage due to boundary layer effects according to Suder (1998):

$$B = 1 - \frac{A - \int \delta^* dr}{A} \qquad \delta^* - displacement\ thickness$$

Comparison of streamlines (GT2017-63993)



Throughflow

Full 3D calculations (Pitchwise averaged)

Turbomachinery Design Process*

**non-examinable*

1) Specify the overall duty
    Inlet flow conditions
    Overall stagnation pressure ratio
    Mass flow rate
    Mean blade speed/RPM (Mechanical constraints)
2) Choose sufficient details of the flow to determine the velocity triangles
    Stage loading, flow coefficient, stage reaction, amount of swirl, α_1
    Fixing three of these is sufficient to fix velocity triangles for repeating stages
3) Guess the efficiency
    Based on past experience (Smith's chart)
4) Calculate the overall stagnation temperature change
    Using the pressure ratio and efficiency
5) Mean-line one-dimensional calculation
    Gives $c_x, p, T, \rho,$ after each blade row
    Empirical loss coefficients are used for each blare row
    Use to improve efficiency in step 3)
6) Determine the annulus area
    $\dot{m} = \rho A c_x \quad \rightarrow \quad A\ (after\ each\ blade\ row)$
    With defined the mean radius it gives $r_{hub},\ r_{tip}$
7) Throughflow calculation
    Specify the spanwise variation in blade angles
    Choose the type of vortex distribution
    Setup and perform an axisymmetric throughflow calculation

Turbomachinery Design Process (2)

8) Determine a realistic value for efficiency or blade loss coefficient
9) Inspect throughflow results

   Check if there are undesirable features such as:

   - highly non-uniform velocities
   - regions of too low or high reaction

   After necessary modifications return to step 7 and perform another throughflow calc.

10) Design blade section

    The throughflow calculations give us:

    - The inlet and outlet angles to every blade
    - Change of stream surface radius
    - Change of stream tube thickness
    - Blade-to-blade calculation on stream surface
    - Blade section must be designed to:
      - Accept the calculated inlet flow direction
      - Produce required outlet flow direction (for any deviation)

    They must have good surface pressure distribution to minimise losses and avoid danger of B/L separation.

11) Produce the complete blade

    'Stack' the blade sections – their centroids lie on a radial line (Centrifugal loading produces only tensile stresses and no bending stresses)

12) Full Three-Dimensional flow calculation

    Removes the assumptions of 2D flows (difference between 3D and quasi 3D)

13) All blade rows are designed in this way
14) Check the off-design performance

Turbomachinery design system freely available online

Professor John D Denton
Whittle Laboratory, Cambridge

Proceedings of ASME Turbo Expo 2017: Turbomachinery Technical Conference and Exposition
GT2017
June 26-30, 2017, Charlotte, NC, USA

GT2017-63993

MULTALL- AN OPEN SOURCE, CFD BASED, TURBOMACHINERY DESIGN SYSTEM

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Cambridge University Engineering Dept.
Cambridge, UK.

ABSTRACT

Turbomachinery design systems are usually the jealously guarded property of large companies, the author is not aware of any for which the source code is freely available. The present paper is aimed providing a freely available system that can be used by individuals or small companies who do not have access to an in-house system.

The design system is based on the 3D CFD solver Multall, which has been developed over many years. Multall can obtain solutions for individual blade rows or for multi-stage machines, it can also perform quasi-3D blade-to-blade calculations on a prescribed stream surface and axisymmetric throughflow calculations. Multall is combined with a one-dimensional mean-line program, Meangen, which predicts the blading parameters on a mean stream surface and writes an input file for Stagen. Stagen is a blade geometry generation and manipulation program which generates and stacks the blading, combines it into stages, and writes an input file for Multall.

The system can be used to design the main blade path of all types of turbomachines. Although it cannot design complex features such as shroud seals and individual cooling holes these features can be modeled and their effect on overall performance predicted. The system is intended to be as simple and easy to use as possible and the solver is also very fast compared to most CFD codes. A great deal of user experience ensures that the overall performance is reasonably well predicted for a wide variety of machines.

This paper describes the system in outline and gives an example of its use. The source codes are written in FORTRAN77 and are freely available for other users to try.

INTRODUCTION

Turbomachinery design systems differ greatly in their details but the general approach is likely to follow the following guidelines.

1. Specify overall parameters such as mass flow rate, mean diameter, rotational speed, inlet flow conditions, exit pressure.
2. Perform a one-dimensional mean-line calculation to obtain the annulus shape and mid-span blade angles.
3. Perform a 2D axisymmetric throughflow calculation in the inverse (design) mode to obtain the variation of flow angles along the span.
4. Repeat the throughflow calculation in analysis mode to predict the blade losses, machine efficiency and stream surface thickness distributions.
5. Perform quasi-3D (Q3D) blade-to-blade calculations at several spanwise sections on each blade row to design the blade shapes.
6. Perform relatively coarse grid, multistage, 3D, viscous calculations for the main flow path of the whole machine to optimise the blade stacking.
7. Perform more detailed 3D calculations to include the effects of leakage flows, endwall bleeds and cavities and coolant flows. These will give the final prediction of machine overall performance.

Each of these stages is likely to be repeated several times with return to repeat previous steps often being necessary.

The author has long maintained [1] that steps 3, 4 and 5 should be omitted and that, given a flexible geometry