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Preliminary design of a centrifugal turbine for ORC applications

M. Pini[1], **G. Persico**[1], E. Casati[1,2], V. Dossena[1]

giacomo.persico@polimi.it

[1]Laboratorio di Fluidodinamica delle Macchine

Dipartimento di Energia

POLITECNICO DI MILANO

[2]Process & Energy Department

TU Delft

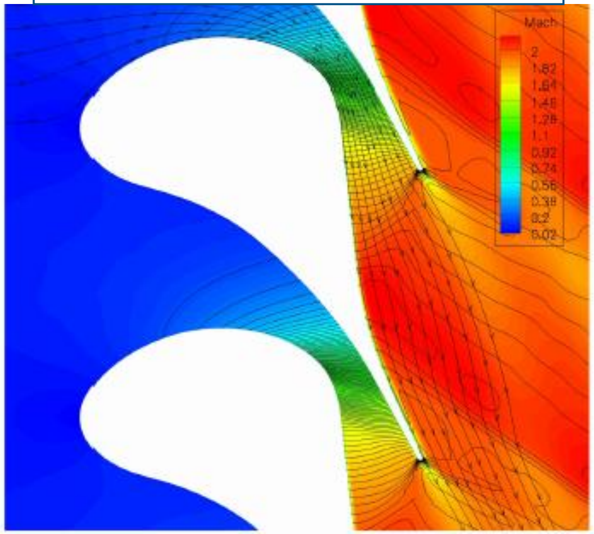
Technische Universiteit Delft

Influence of thermodynamics on ORC turbines

Unconventional thermodynamic expansion → unconventional turbines:

- High expansion ratios → **large passage area ratio**
- Small enthalpy drops → **low number of stages / low peripheral speed**
- Fluid (and flow..) complexity → **low speed of sound, transonic/supersonic turbines**

FluidProp & zFlow



Scientific and technical issues

- Supersonic efflux of dense gases still under study
  → influence on turbine **efficiency** and **design criteria**
- State-of-art modelling tools necessary for reliable predictions
  → advanced **real-gas** models coupled with **CFD**

Implementation of advanced modelling techniques into present-day design tools for effective turbine design

Outline

- Survey on turbine architecture
- Design methodology
- Design exercise of a multi-stage centrifugal turbine
- Conclusions and future works

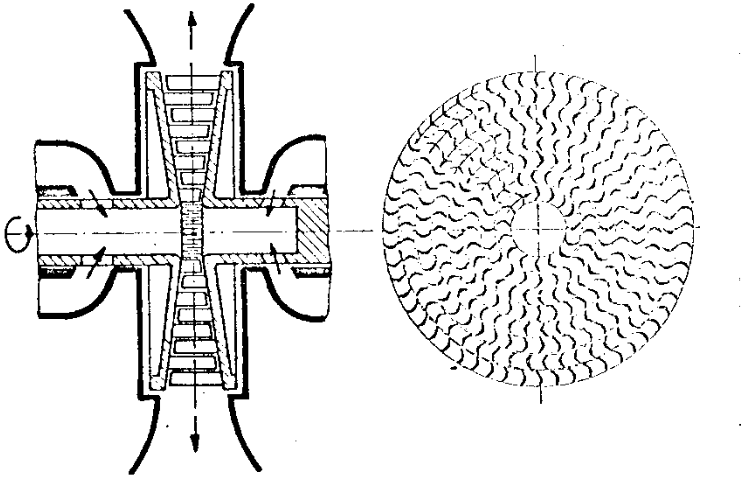
Several architectures are possible

Axial turbines

Centripetal turbines (radial inflow)

Centrifugal turbines (radial outflow):

- ✓ Counter-rotating (Ljiunstrom)
- ✓ **Fixed-rotating centrifugal turbine**
  already suggested by Macchi, VKI LS 1977
  and Gaia et al., 1978



The centrifugal fixed-rotating turbine

Advantages

- passage area naturally increases along the expansion process
  → fits with the huge increase of fluid specific volume
- more compact compared to equivalent axial turbines
  → higher number of stages → lower M, higher efficiency and flexibility

Main disadvantage:

- centrifugal force potential acts against work extraction
  → profile aerodynamics crucial for energy exchange effectiveness

$$L_{eu} = \frac{v_1^2 - v_2^2}{2} + \underbrace{\frac{u_1^2 - u_2^2}{2}}_{\boxed{\textbf{Negative term } (u_2 > u_1)}} - \frac{w_1^2 - w_2^2}{2}$$

Negative term ($u_2 > u_1$) ➡ **Reduction of work**

Outline

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- **Design methodology**
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Overview on design strategy

Difficult task due to the thermodynamic complexity, and transonic flow

Specific turbine design, depending on fluid and thermodynamic conditions

Design techniques:

- ✓ **1D** mean-line method, still used, suited for **initial design** and optimization
- ✓ CFD simulations, three dimensional, steady or unsteady, for blade definition
- ✓ Necessity of a **bridge between 1D and 3D** approach: **throughflow** method
  → **axisymmetric** assumption, including losses, flow deflection and blade blockage: define the 'mean-surface' of the flow across the whole machine
  → effects of shocks, post-expansion, annulus boundary layers and flaring are included

1D meanline (preliminary design) → **Throughflow (flow surface)** → **Fully 3D simulation (blade refinement)**

The 1D mean-line code

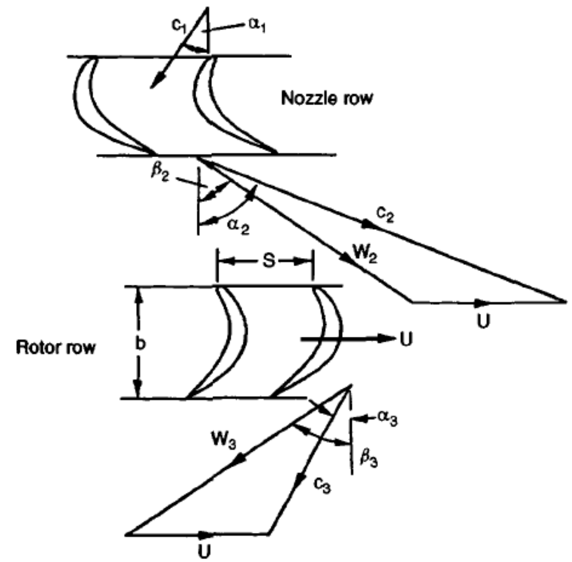
- Based on **mass** conservation, **energy** balance, estimate of **polytropic index**
- Criteria to divide the expansion ratio across the stages and blade rows (χ)
- Geometrical assumptions: inlet h/D, discharge blade angles
- Blade span along the machine computed through mass conservation (choking)
- Work exchange computed from velocity triangles
- Entropy production predicted by **loss correlation** (es. Soderberg, Traupel, Craig&Cox)

Code main features

✓ Applicable to the design of axial and radial turbines

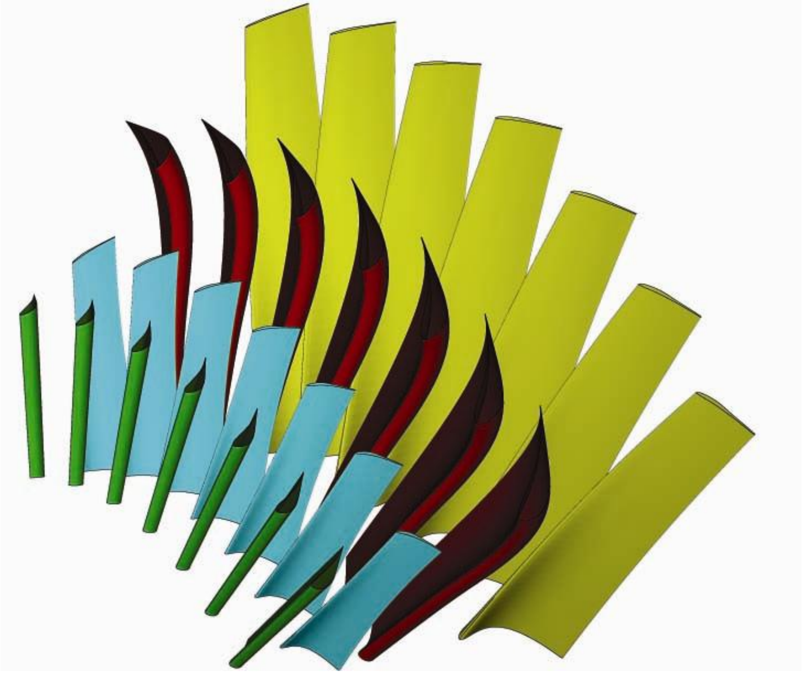
✓ Coupled with FluidProp (Prof. Colonna, TU Delft) for thermodynamic properties calculation

✓ Directly connected with the throughflow solver

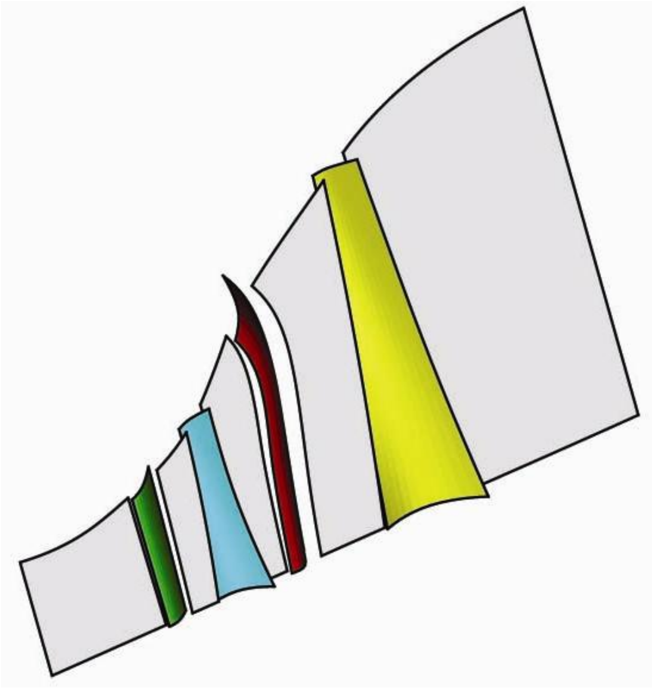


Throughflow concept

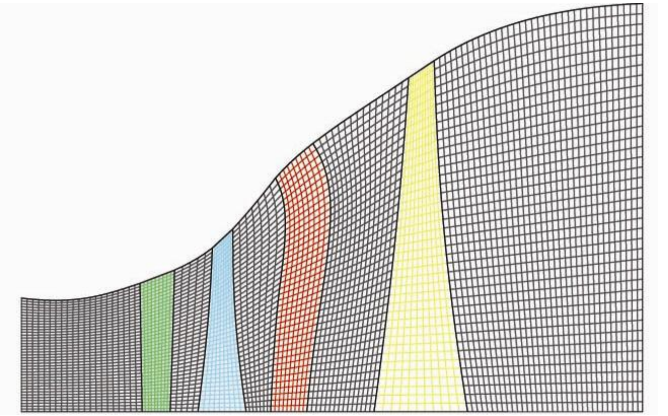
Multistage LP axial turbine: fully 3D unsteady: CFD affordable but with very high computational cost (no optimization)



Axisymmetric flow assumption



2D problem: very fast
hours → minutes



Intrinsically suitable for optimization

CFD-based throughflow model

Axisymmetric inviscid model with volume forces: deflection (L), losses (D)

$$\frac{\partial \boldsymbol{u}}{\partial t} + \frac{\partial \boldsymbol{f}^x}{\partial x} + \frac{1}{r}\frac{\partial (r\boldsymbol{f}^r)}{\partial r} + \frac{1}{r}\boldsymbol{s}^r + \frac{1}{b}\frac{\partial b}{\partial x}\boldsymbol{f}^x + \frac{1}{b}\frac{\partial b}{\partial x}\boldsymbol{f}^r + \boldsymbol{s}^b = \boldsymbol{s}^v$$

$$\boldsymbol{u} = \begin{bmatrix} \rho \\ \rho e_0 \\ \rho v_x \\ \rho v_r \\ \rho v_\theta \end{bmatrix}, \quad \boldsymbol{f}^x = \begin{bmatrix} \rho v_x \\ \rho h_0 v_x \\ \rho v_x^2 + P \\ \rho v_r v_x \\ \rho v_\theta v_x \end{bmatrix}, \quad \boldsymbol{f}^r = \begin{bmatrix} \rho v_r \\ \rho h_0 v_r \\ \rho v_x v_r \\ \rho v_r^2 + P \\ \rho v_\theta v_r \end{bmatrix} \qquad b(x,r) = \frac{N[\theta_p(x,r) - \theta_s(x,r)]}{2\pi}$$

$$\boldsymbol{s}^r = \begin{bmatrix} 0 \\ 0 \\ 0 \\ -(\rho v_\theta^2 + P) \\ \rho v_r v_\theta \end{bmatrix}, \quad \boldsymbol{s}^b = \begin{bmatrix} 0 \\ 0 \\ -P\partial b/\partial x \\ -P\partial b/\partial r \\ 0 \end{bmatrix}, \quad \boldsymbol{s}^v = \begin{bmatrix} 0 \\ \rho(\mathbf{L}+\mathbf{D})\cdot \omega r \mathbf{e}_\theta \\ \rho(\mathbf{L}+\mathbf{D})\cdot \mathbf{e}_x \\ \rho(\mathbf{L}+\mathbf{D})\cdot \mathbf{e}_r \\ \rho(\mathbf{L}+\mathbf{D})\cdot \mathbf{e}_\theta \end{bmatrix}$$

$$\mathbf{L} = L\mathbf{n}_g \longrightarrow L = L(\boldsymbol{u}) = -K\mathbf{w}\cdot\mathbf{n}_g = -K\left(\mathbf{v} - \omega r \mathbf{e}_\theta\right)\cdot\mathbf{n}_g$$

$$\mathbf{D} = -D\mathbf{t} = -D\frac{\mathbf{w}}{||\mathbf{w}||} \longrightarrow D = T\nabla s \cdot \frac{\mathbf{w}}{||\mathbf{w}||} = T\nabla s \cdot \mathbf{t}$$

Throughflow package

Numerical scheme

- Implemented as an extension of the *zFlow* code (Prof. Rebay, Univ. of Brescia)
- Based on hybrid FE/FV formulation, uses fully implicit time-marching methods
  → very efficient and accurate computational model (especially for 2D problems)
- Coupled with FluidProp (Prof. Colonna, TU Delft) for thermodynamic properties
  → accurate treatment to handle arbitrary EoS (thus to deal with organic fluids)

Blade modeler & grid generator

- Throughflow solver integrated by a blade modeler with mesh generator
- Blade mean line and blockage factor managed through a generalized blade modeler
- Structured multi-block mesh generator for axial, radial and mixed-flow machines

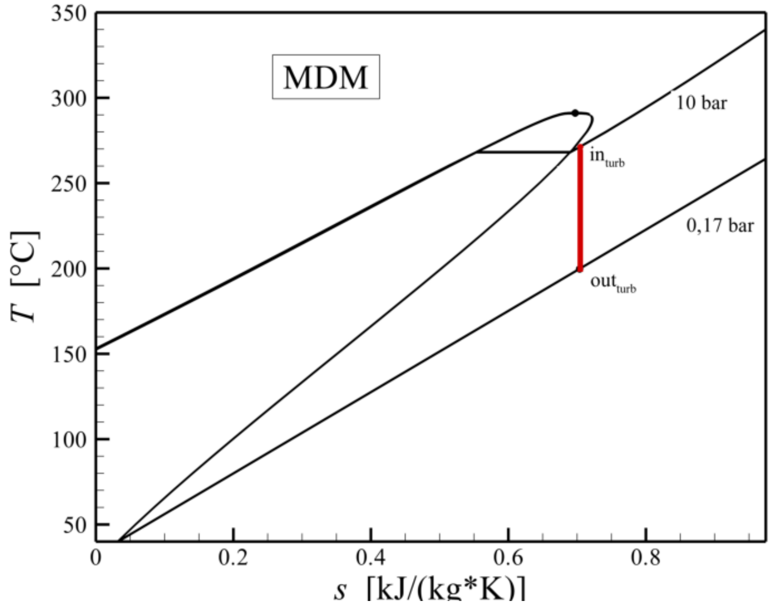
Outline

1. The radial-outflow turbine
2. Turbine design method
3. **Design exercise of a multi-stage centrifugal turbine**
4. Conclusions and future works

Design exercise: 1D mean-line design

Turbine case: 'mainstream ORC'

1. Inlet thermodynamic conditions:
   - ✓ Total inlet T: **274 °C**
   - ✓ Total inlet P: **10 bar**
2. Working fluid: **siloxane MDM**
3. Expansion ratio: **60**
4. Target power: **1÷2 MW**



Design constraints:

1. $M_{MAX} = 1.1$
2. $\chi \approx 0.5$
3. $\psi_{MAX} = 30°$

Outcome of parametric study:

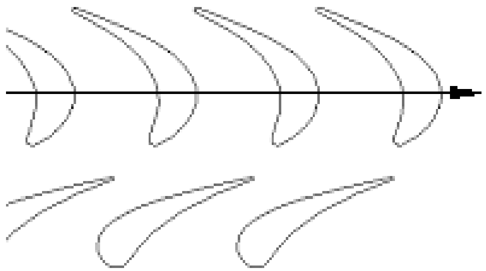
1. $N_{ST} = 6$
2. $R_{EXT} = 0.7$ m
3. n = 3000 rpm
4. P = 1.3 MW
5. $\eta_{TS} = 0.894$

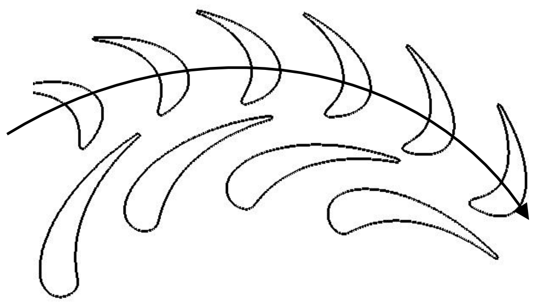
Stage	$P_{T,out}$ (bar)	P_{out} (bar)	T_{out} (°C)	s (kJ/kgK)	W (%)
1	5.62	5.07	261.5	0.718	11.6
2	2.72	2.57	253.5	0.720	17.3
3	1.34	1.30	246.7	0.722	18.6
4	0.70	0.66	240.2	0.723	18
5	0.37	0.34	234	0.724	17.8
6	0.20	0.17	227.7	0.725	17.6

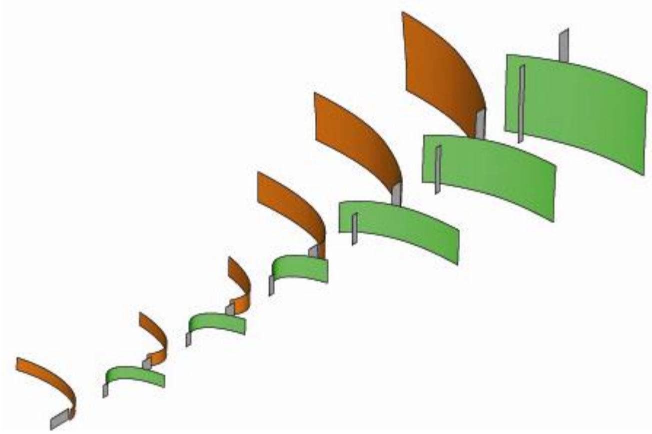
Design exercise: throughflow model

From lumped-parameter mean-line to continuous mean-blade-surface

- Design of meridional channel
  → blade span evolution along the machine from 1D model
  → relevant risk of separation close to the endwalls (gaps act as vaneless diffusers)
  → constant flaring angle in each bladed region, inter-row gaps as small as possible
- Definition of the blade shape: mean line and blade thickness
  → basic 2D profiles developed for axial turbines
  → profile deformation to match in/out design blade angles
  → conformal transformations in polar coordinates to conserve blade angles



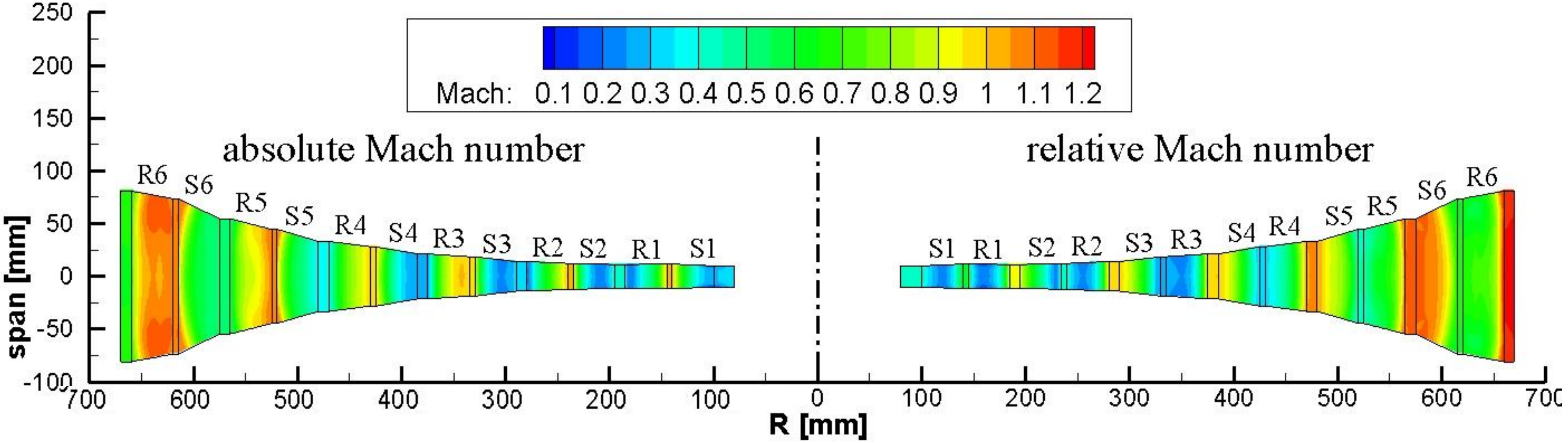




Design exercise: throughflow calculation

Soderberg correlation for core flow + annulus boundary layer loss model

21 (span) x 400 (stream) grid, computational time ~5 min



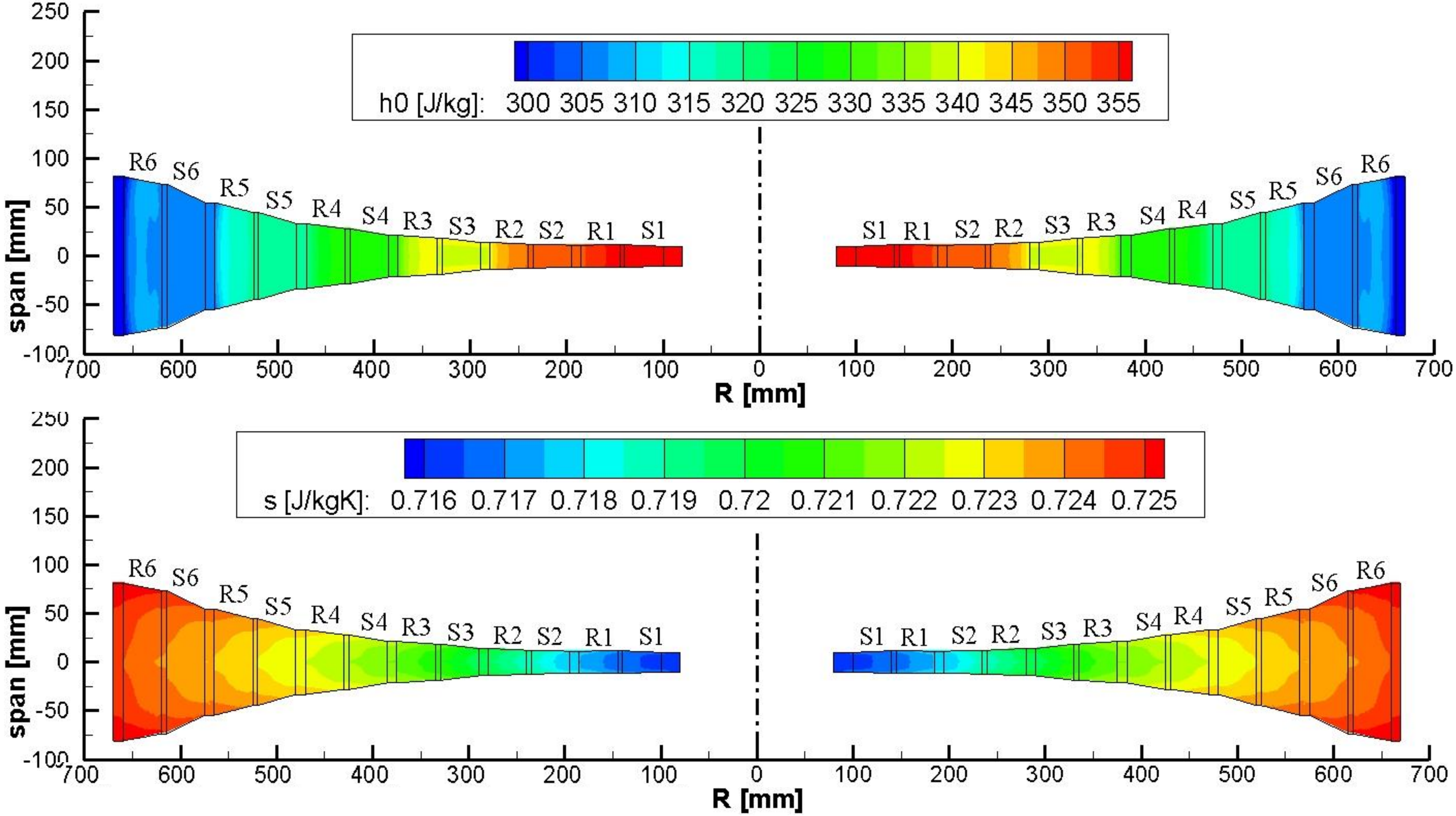
Flow distribution on the whole meridional surface

Maximum abs/rel Mach number target nearly confirmed (1.1)

Significant deviations from 1D model in the last stages

Design exercise: throughflow calculation

Streamwise evolution of thermodynamic quantities



Computed efficiency 88.8% (< 0.5% 1D)

Conclusions and **future works**

- Multistage centrifugal turbine architecture considered
- Design methodology for multistage ORC turbines proposed, based on:
  - mean-line code
  - CFD-based throughflow model
- Design exercise for a 'mainstream' ORC turbine (P ~1.3 MW, Pt = 10bar, Tt=270°C)
  → 6 stages, 3000 rpm, R = 0.7 m, η ~ 89%
- Throughflow model: 1D integral data corrections, large spanwise gradients

- Future work will move mainly in three-directions:
  → blade-to-blade calculations to update loss and slip factor correlations
  → unsteady calculations to investigate blade-row interaction (very small gaps!)
  → coupling with optimization methods to determine the optimal streamsurface

THANK YOU!