Design and testing of a high flow coefficient mixed flow impeller

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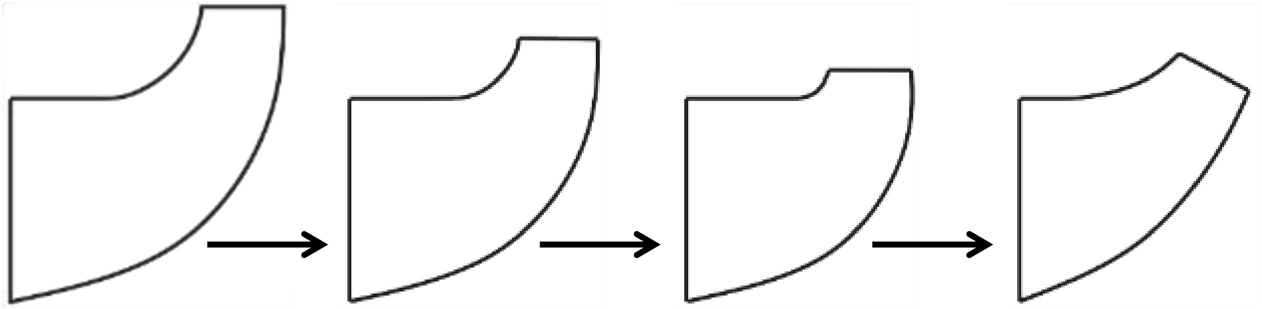
Objectives

- In many turbocharger applications, there is a strong requirement for compact designs to reduce the size and cost of the installation
- This can be achieved by reducing the impeller wheel diameter
- Extreme cases lead to the application of highly loaded mixed flow impellers
- The achievable performance level of such stages have not been fully explored
- The objective was to assess the achievable performance levels in this relatively uncharted territory of the design space



Outline of this talk

- Conceptual design of the stage
- Preliminary design considerations
- Mechanical integrity
- Final impeller design and test data
- Conclusions

Stage pressure rise capacity

- Flow coefficient

$$\phi = \frac{\dot{V}_0}{U_2 D_2^2}$$

- Isentropic pressure rise coefficient:

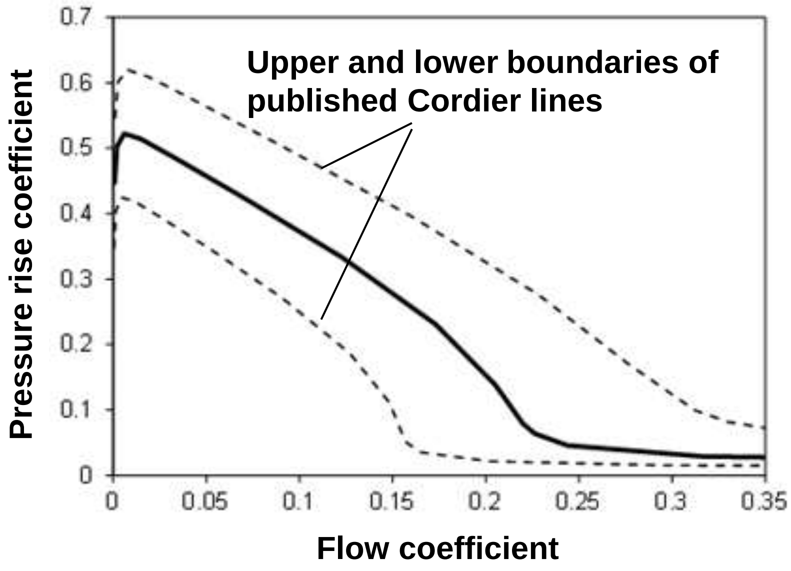
$$\psi = \frac{\Delta H_s}{U_2^2} = \frac{C_p T_{01}\left(\pi^{\frac{\gamma-1}{\gamma}} - 1\right)}{U_2^2}$$

- For the same mass flow, pressure ratio and rotational speed, ϕ and ψ increase as the diameter is reduced

- There is a continuous reduction in the pressure rise capacity of the impeller as the flow capacity is increased and the stage becomes more axial

Modified Cordier diagram

Casey, Zwyssig and Robinson (2010)



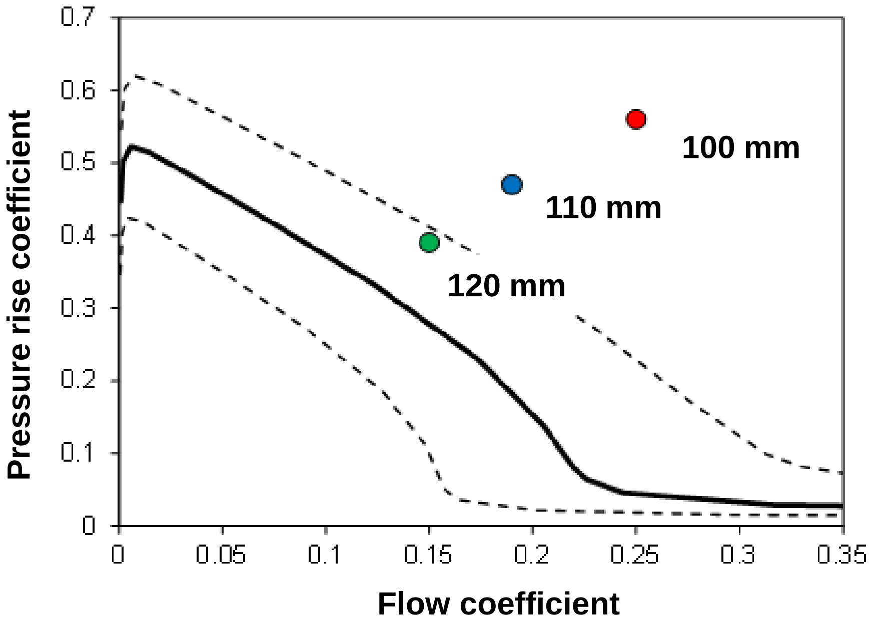
Selection of the design space

- Design targets for the current impeller:
  - Volume flow =1.02 m3/s
  - Pressure ratio = 2.65
  - Rotational speed = 77525 rpm

D_2	ϕ	ψ
120 mm	0.146	0.392
110 mm	0.189	0.467
100 mm	0.251	0.564

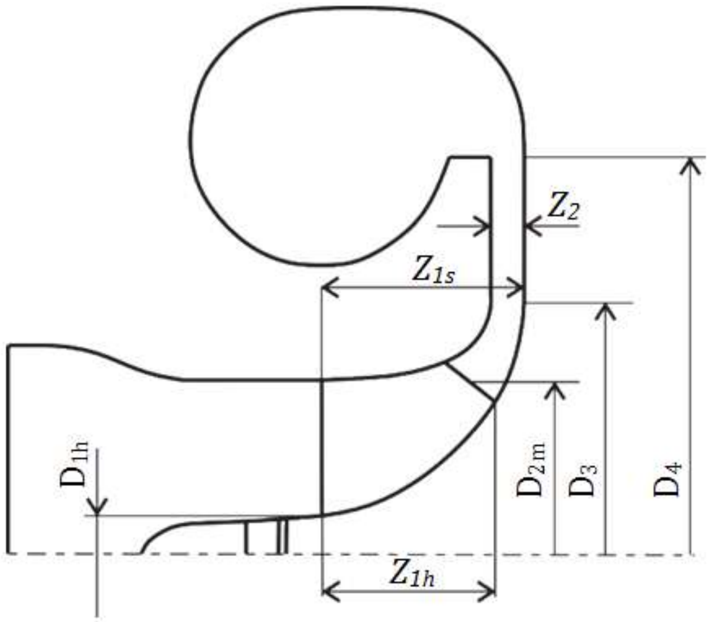
Selected for the current design

Modified Cordier diagram
Casey, Zwyssig and Robinson (2010)



Geometrical constraints

- Several geometrical constraints were imposed on the design:
  - Impeller mean outlet diameter
  - Impeller axial length at the hub and shroud
  - Dimensions of the parallel wall diffuser



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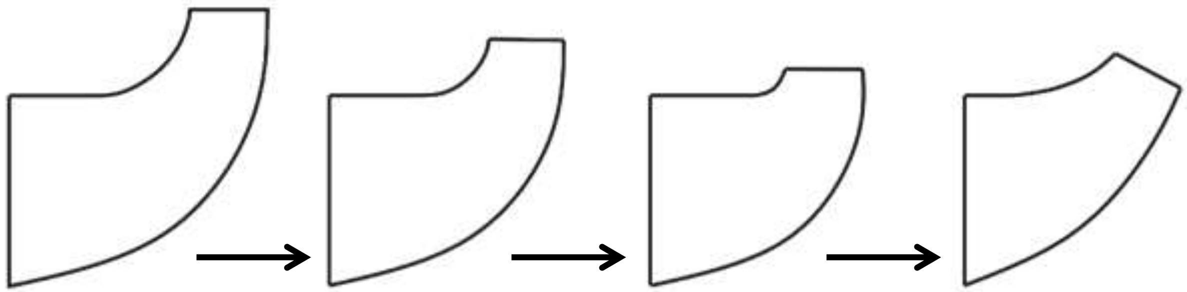
Design cases

- Four different impellers, designed for the same duty and rotational speed are compared
- The following parameters were controlled to produce the target pressure ratio:
  - Impeller tip diameter
  - Impeller backsweep angle
  - Mean absolute flow angle at impeller exit
    - fixed at 55° to match with vaneless diffuser
- The stage polytropic efficiency was assumed to be 80% for all cases



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  - Impeller backsw
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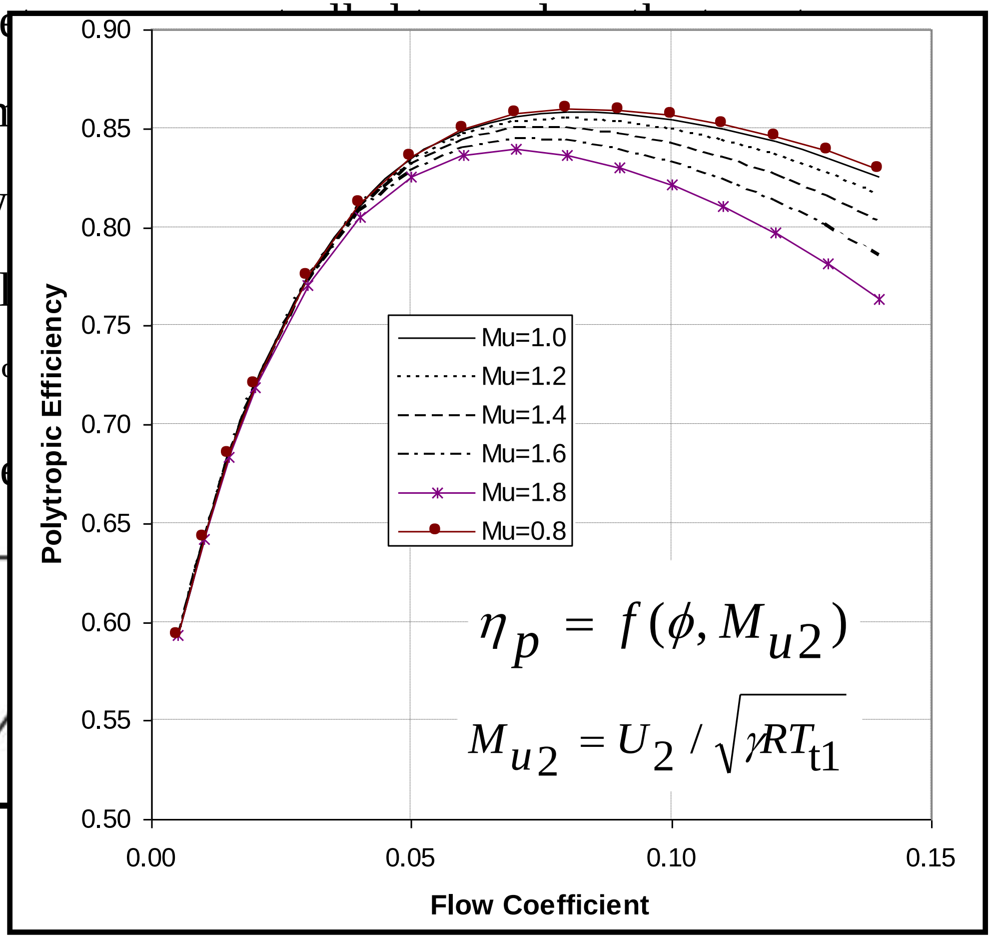
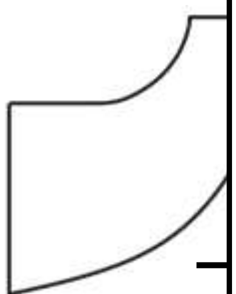
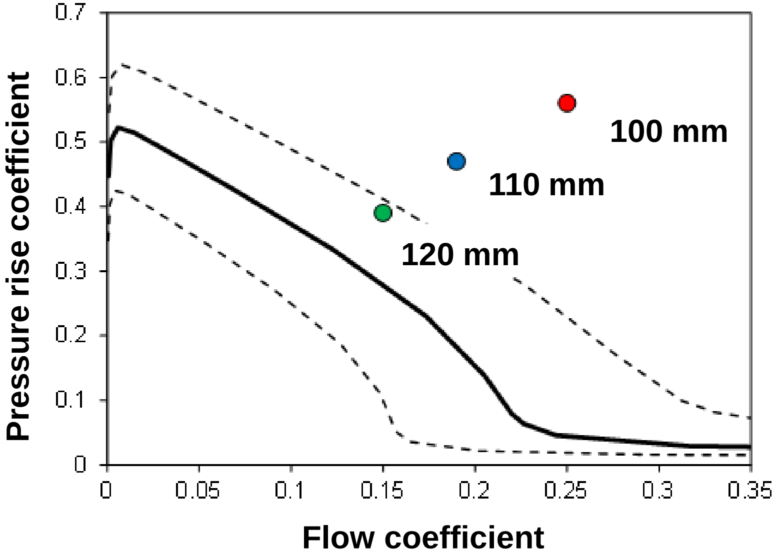
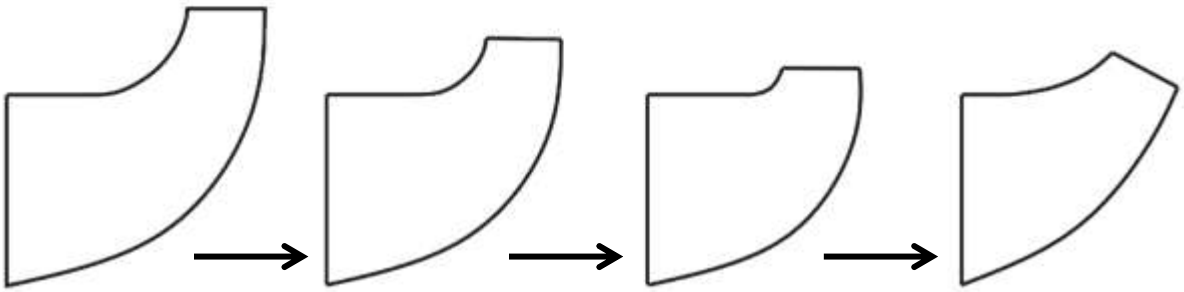




Designs with different impeller diameters

- The difficulty of the design increases as the design space moves to the top-right corner of the diagram with
  - High specific flow
  - High specific pressure rise





$D_{2m} = 120mm$	$D_{2m} = 110mm$	$D_{2m} = 100mm$	$D_{2m} = 100mm$
$\phi = 0.15$	$\phi = 0.19$	$\phi = 0.25$	$\phi = 0.25$
$\psi = 0.39$	$\psi = 0.47$	$\psi = 0.56$	$\psi = 0.56$

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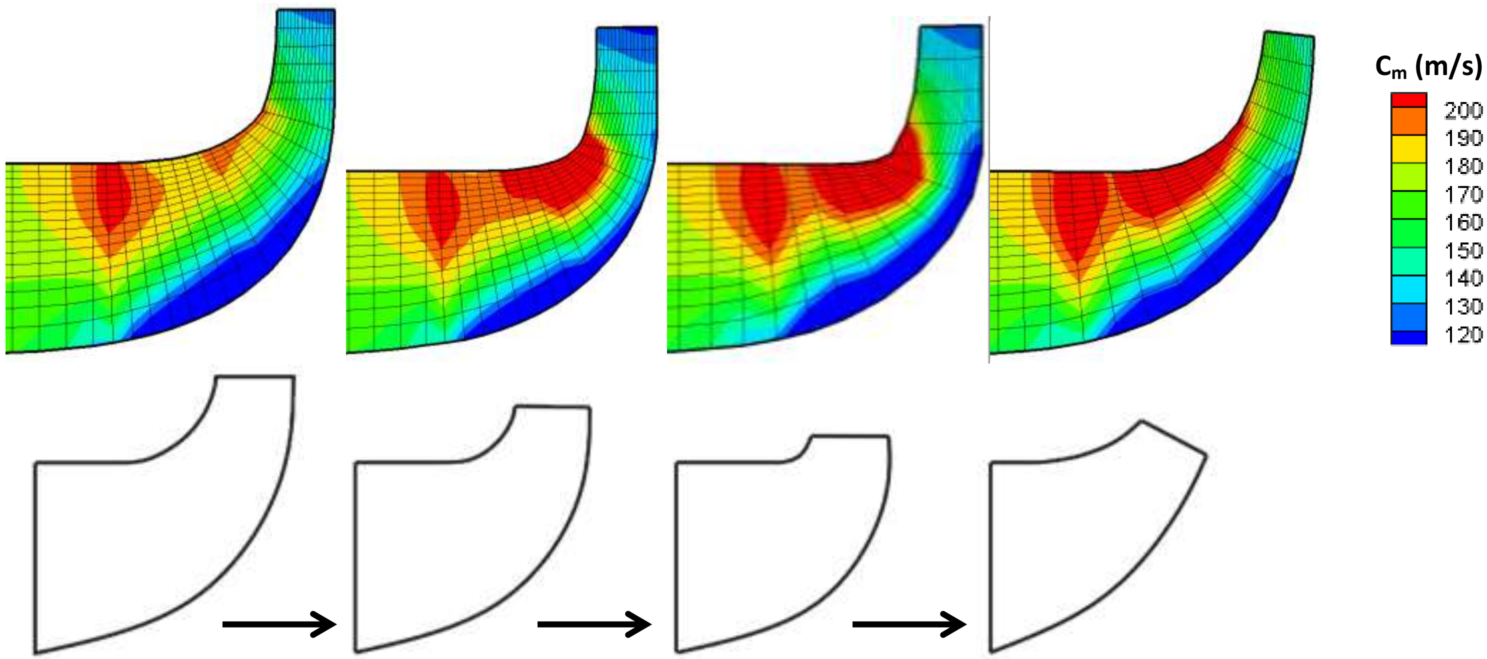
Shroud curvature

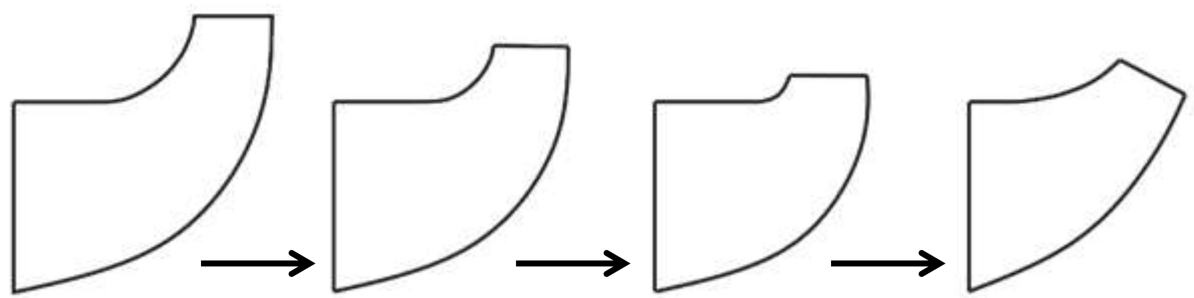
- Through-flow calculations of the radial designs show acceleration of the flow on the shroud contour due to increased curvature with a risk of flow separation at sharp turn
- Mixed flow design relieves this shroud curvature effect



$D_{2m} = 120mm$	$D_{2m} = 110mm$	$D_{2m} = 100mm$	$D_{2m} = 100mm$
$D_{2t}/D_{1t} = 1.28$	$D_{2t}/D_{1t} = 1.17$	$D_{2t}/D_{1t} = 1.07$	$D_{2t}/D_{1t} = 1.16$

Impeller back sweep

- In a typical radial compressor about 50% of the pressure rise is due to the centrifugal effects
- Lower backsweep is required to compensate for the reduction in centrifugal effects at high flow coefficient



$D_{2m} = 120mm$
$\alpha_2 = 55^{\circ}$
$\beta_2 = 54^{\circ}$

$D_{2m} = 110mm$
$\alpha_2 = 55^{\circ}$
$\beta_2 = 40^{\circ}$

$D_{2m} = 100mm$
$\alpha_2 = 55^{\circ}$
$\beta_2 = 18^{\circ}$

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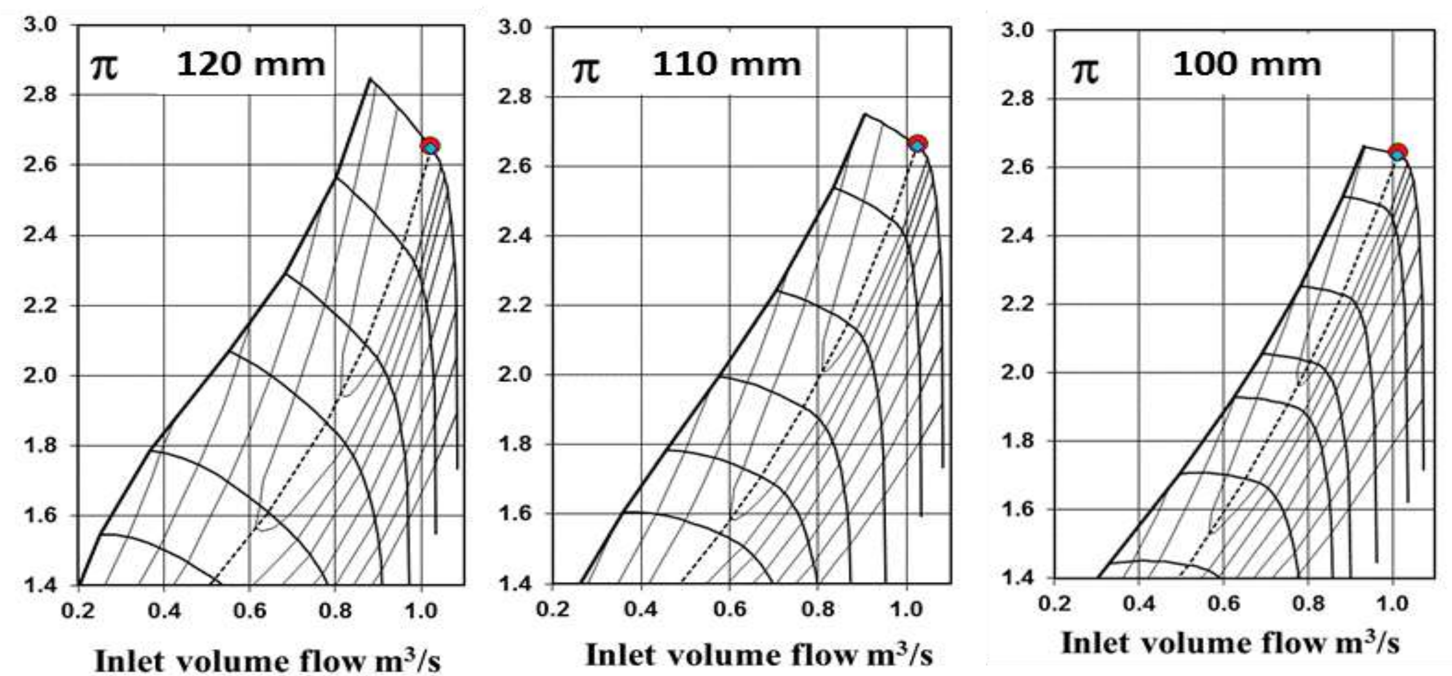
Impeller back sweep

- Casey-Robinson map prediction method shows flatter compressor characteristics and narrower operating range with low backsweep angles



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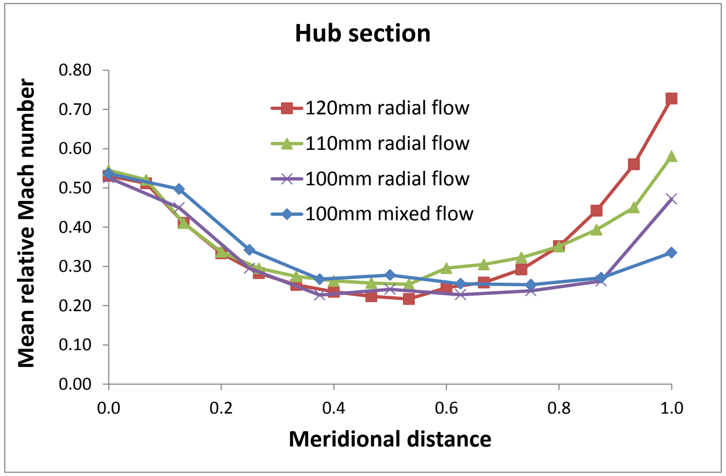
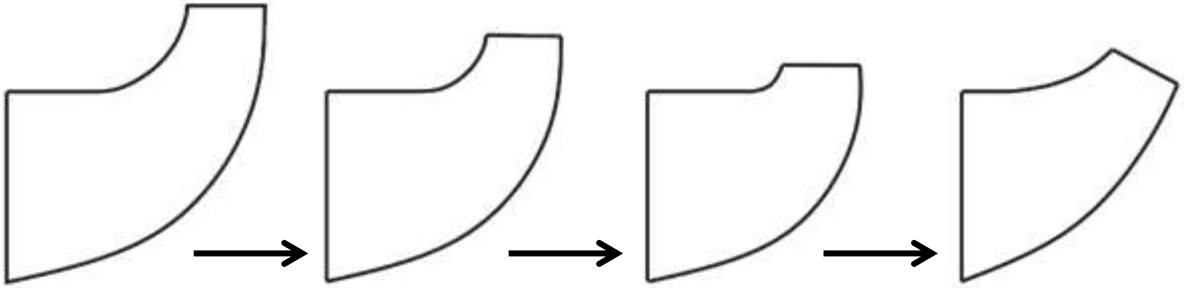
Relative diffusion

- In a typical centrifugal compressor:
  - Flow accelerates on the hub contour
  - Flow decelerates on the shroud contour
  - Deceleration on the shroud limits the total diffusion in the passage
  - Flow in the hub region is less sensitive to blade design

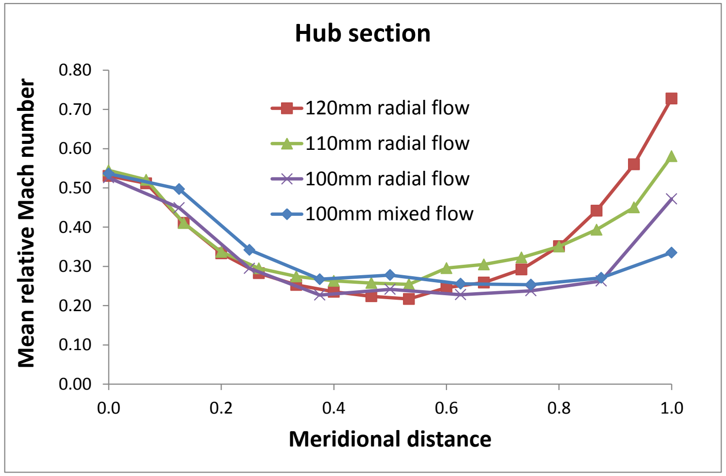


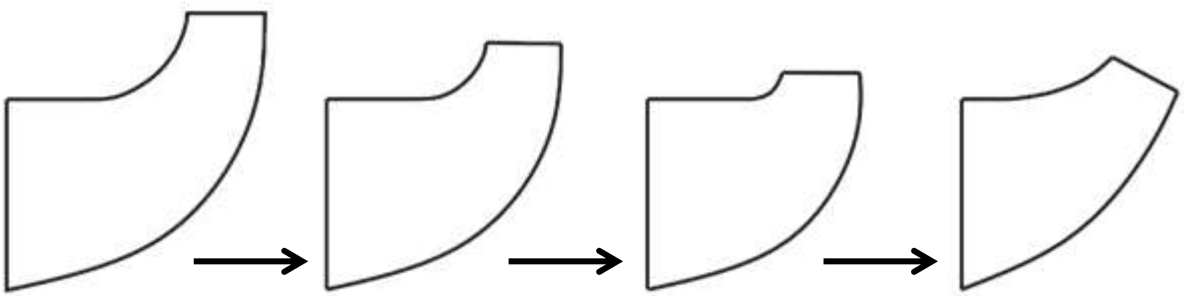


$D_{2m} = 120mm$	$D_{2m} = 110mm$	$D_{2m} = 100mm$	$D_{2m} = 100mm$
$W_{2t}/W_{1t} = 0.69$	$W_{2t}/W_{1t} = 0.59$	$W_{2t}/W_{1t} = 0.52$	$W_{2t}/W_{1t} = 0.61$
$W_{2h}/W_{1h} = 1.22$	$W_{2h}/W_{1h} = 1.05$	$W_{2h}/W_{1h} = 0.93$	$W_{2h}/W_{1h} = 0.80$

Relative diffusion

- As the impeller diameter is reduced:
  - Higher relative diffusion is needed in the passage to further increase the pressure rise
  - Stronger diffusion on the shroud contour
  - Diffusion of the flow on the hub contour





$$\begin{aligned} D_{2m} &= 120mm \\ W_{2t}/W_{1t} &= 0.69 \\ W_{2h}/W_{1h} &= 1.22 \end{aligned}$$

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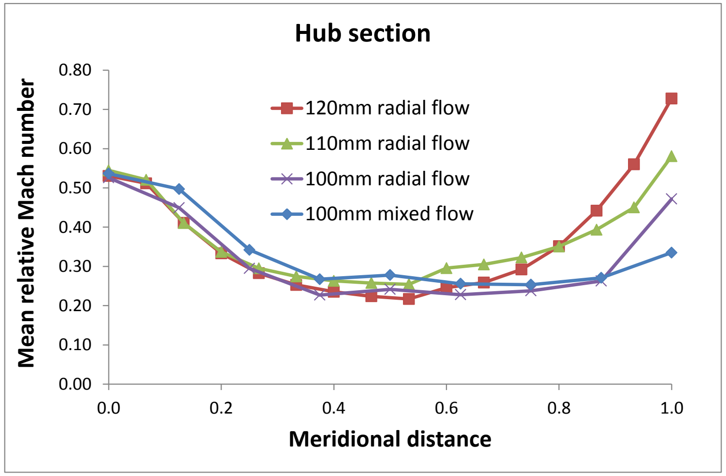
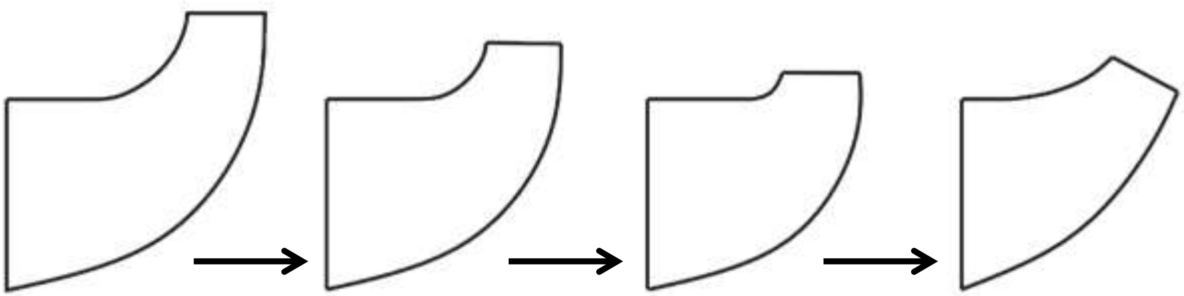
$$\begin{aligned} D_{2m} &= 100mm \\ W_{2t}/W_{1t} &= 0.61 \\ W_{2h}/W_{1h} &= 0.80 \end{aligned}$$

Relative diffusion

- Mixed flow design reduces the diffusion on the shroud but increases the diffusion on the hub
    - More balanced diffusion on the hub and shroud contours
- In axial compressors the hub loading becomes the limiting factor





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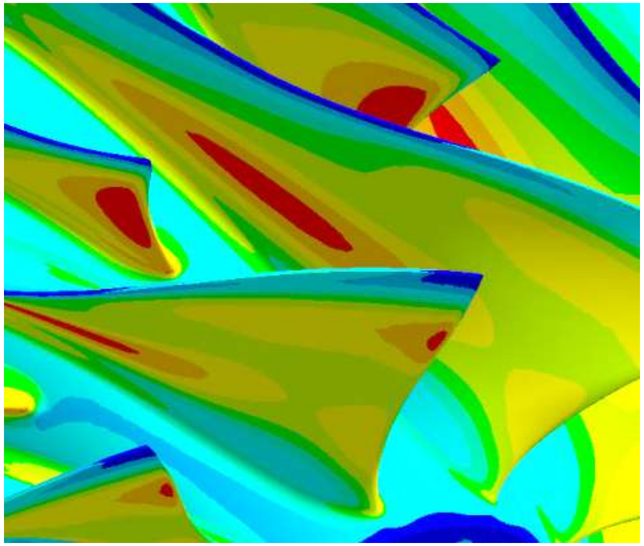
Mechanical integrity

- Mechanical design criteria:
  - Maximum allowable stress should not be exceeded
  - Vane first eigenmode frequency should be above 4EO frequency
  - Burst margin criteria should not be exceeded
- Aerodynamic design requirements:
  - Small camber in the front part of the tip section to control the supersonic flow (lower vane natural frequencies)
  - Aerodynamic optimization along the span to control the shock losses (higher stress in the blade)
  - Less flexibility in aerodynamic design of the hub section due to high speed flow and high relative diffusion near the hub
  - Limitation on the number of the vanes, vane thickness and position of the splitter vanes on the hub

Mechanical integrity

- The compromised design:
  - Thick blades with compromised aerodynamic performance at lower span
  - Backward swep near the hub to reduce the stress
  - Splitter leading edge placed further downstream to avoid strong flow acceleration near the hub



Stress distribution in the impeller

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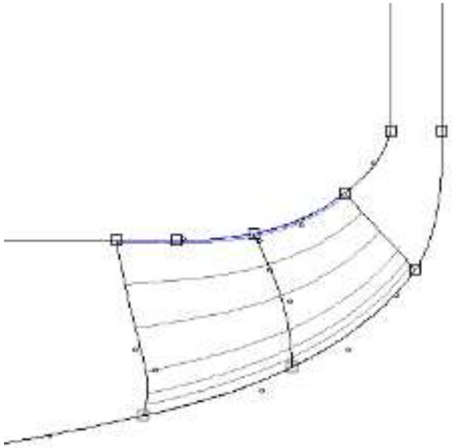
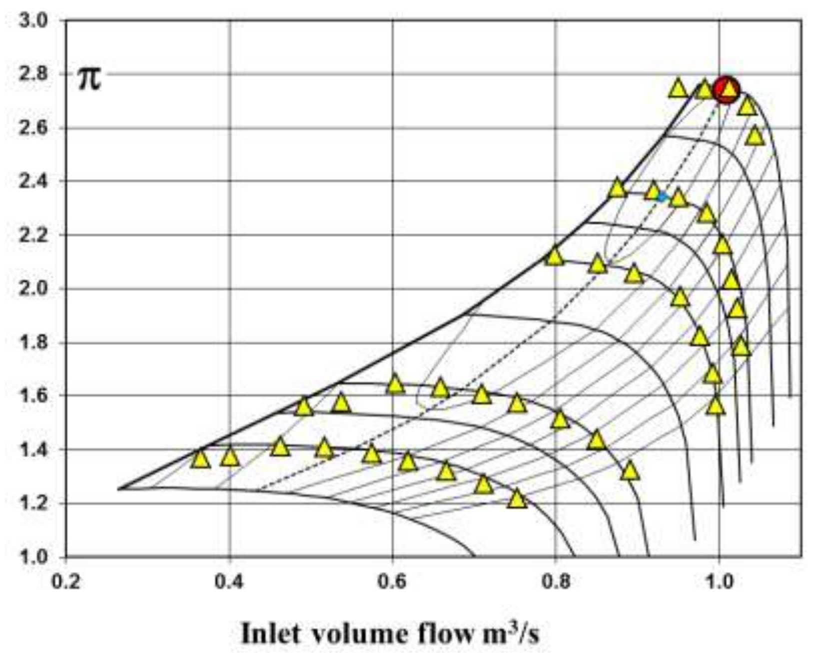
Final impeller design

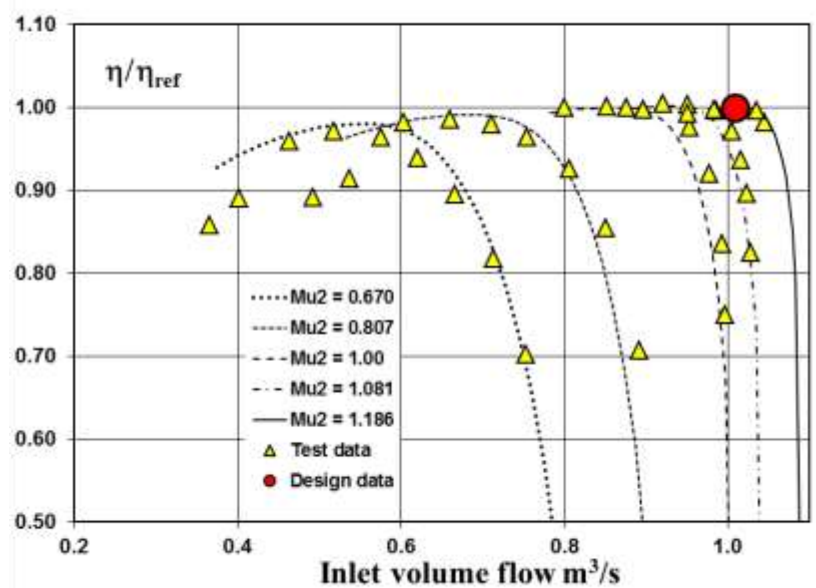
- Final impeller:
  - 100 mm mean outlet diameter
  - 9+9 vanes
  - Backsweep angle of 28°
  - Forward swept at the tip
  - Curved line generators were used (impeller was manufactured by point milling) to allow better control of the high speed flow along the span



Test results

- 1D map predictions agree very well with the test results
- A peak total-to-total isentropic efficiency of above 80% was achieved at a pressure ratio of 2.75
- The isentropic pressure rise was 6% higher than the design target





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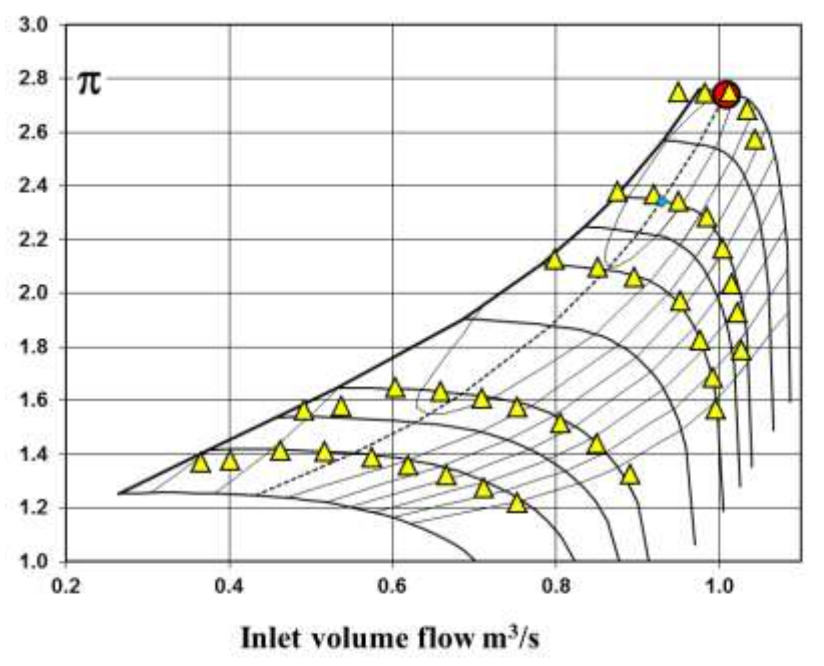
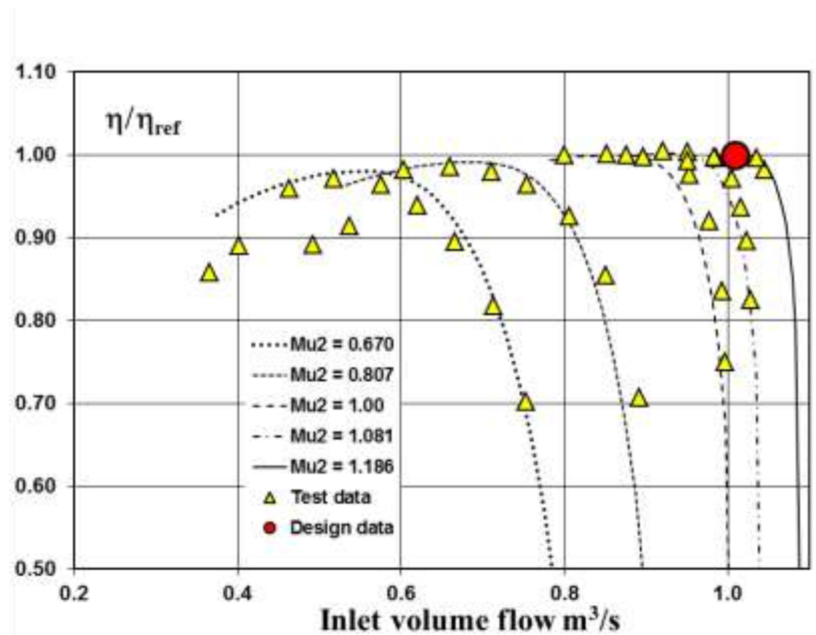
Test results

- Surge margin at design speed =

  (Design flow rate – Surge flow rate) / (Design flow rate) = 6%

- Map Width Enhancement devices may be necessary to widen the operating range





Conclusions

- The conceptual design and testing of a mixed flow compressor stage with an extremely high flow and pressure rise is described.
- The preliminary design methods identified many of the difficulties expected from such an unconventional design, in particular the need for a mixed flow stage, the expected narrow operating range and the difficult mechanical issues.
- The test data showed that an efficiency level of above 80% is achievable in this extreme design space
- The final testing of the stage identified the need for some form of recirculating bleed system to increase the operating range of the stage.