Deep Throttle Turbopump Technology Design Concepts

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JANNAF
April 8-12, 2002
Destin FL

Outline

- Objective
- WFR issues
- Concept Evaluation for WFR
- Design-Analysis Methodology Review
- Detailed Diffuser Design

Deep Throttling Turbopump Design Objectives

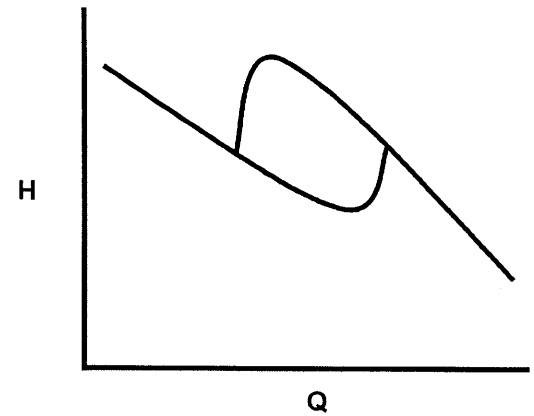
- **Increase the throttling range of turbopumps from 30% to 120% of the design value, while maintaining high performance levels**

Wide Flow Range Issues

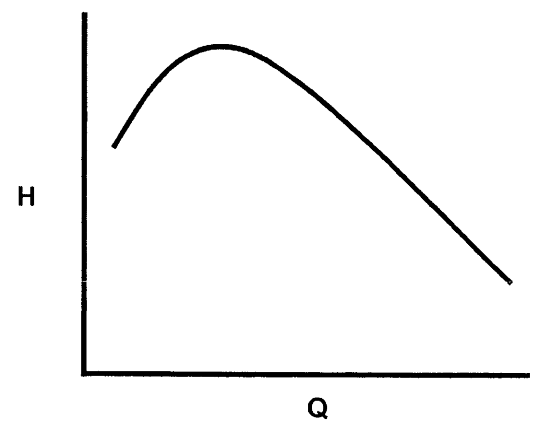
- H-Q characteristic
  - Instability
- Suction Performance
  - Inducer limiting
- Mechanical
  - Dynamic loads from suction and discharge recirculation
- Performance

H-Q Characteristic

- Stall
  - Flow discontinuity within pump that causes a discontinuity in pump characteristic at constant speed



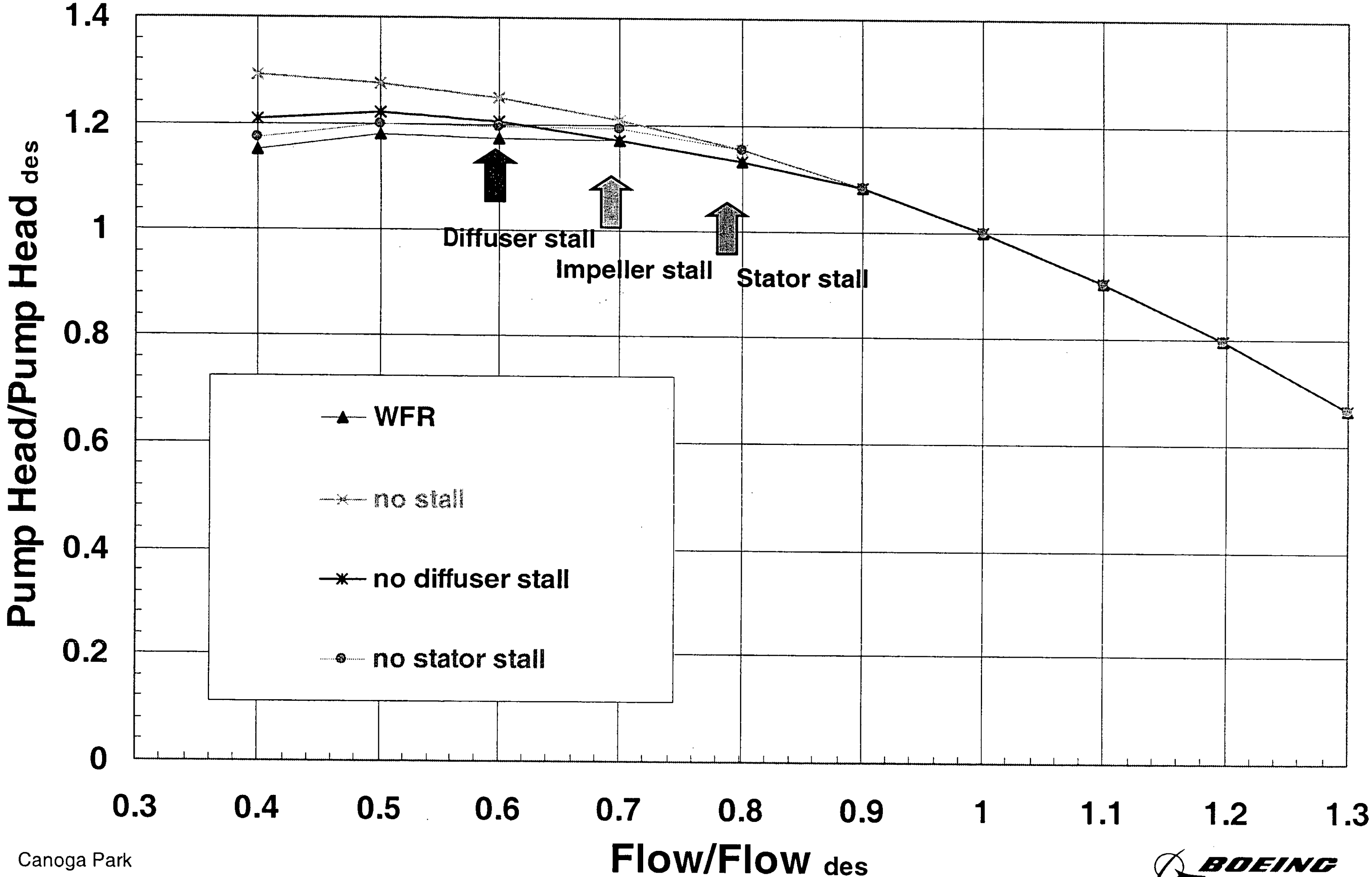
- Surge
  - Instability when the system characteristic reacts with a positive sloping pump characteristic



Separation and Stall

Jet and wake observed in each impeller passage. The eddying wake is seen on the suction side of the channel from Fischer and Thoma, 1932

Stall Characteristics

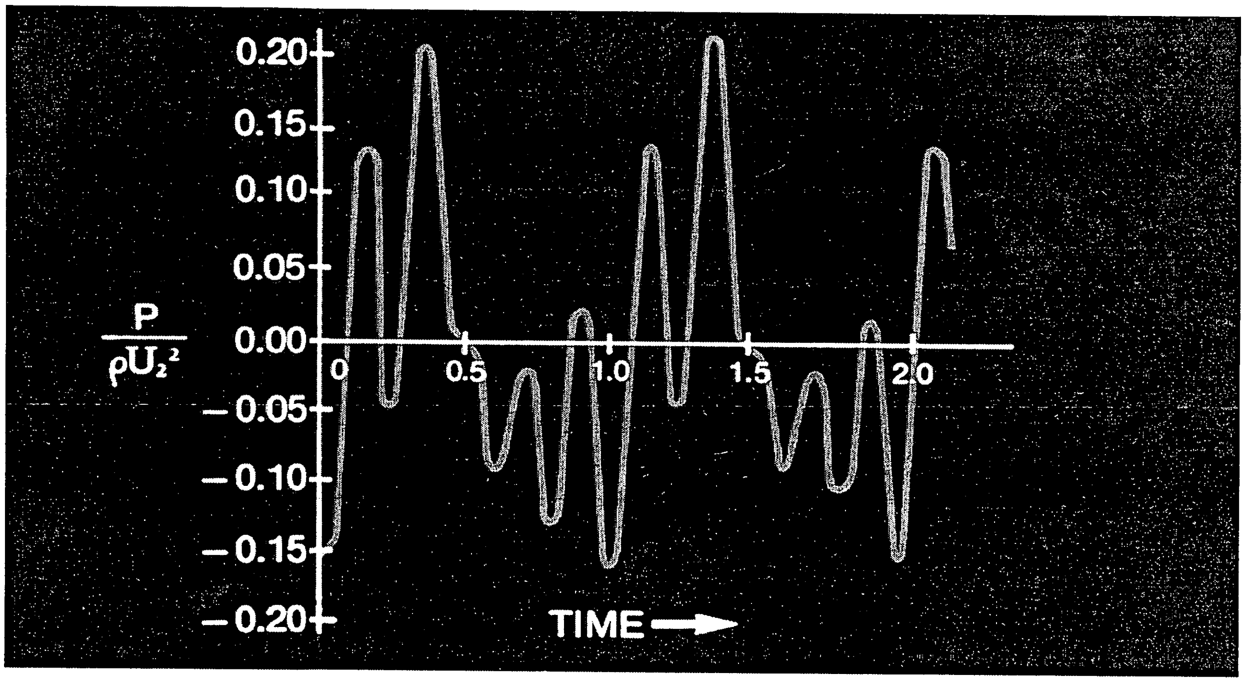


Energy Level

- **Pumps running with recirculation experience pressure pulsations**
  - Recirculation from diffuser and impeller is unsteady
  - Pressure pulsations probably scale with overall pressure rise

- **Diffuser must accommodate these attributes**
  - Gap B (clearance between impeller blades and diffuser blades)
  - Gap A (impeller shrouds and diffuser shrouds)

Pressure Fluctuations at Impeller Exit

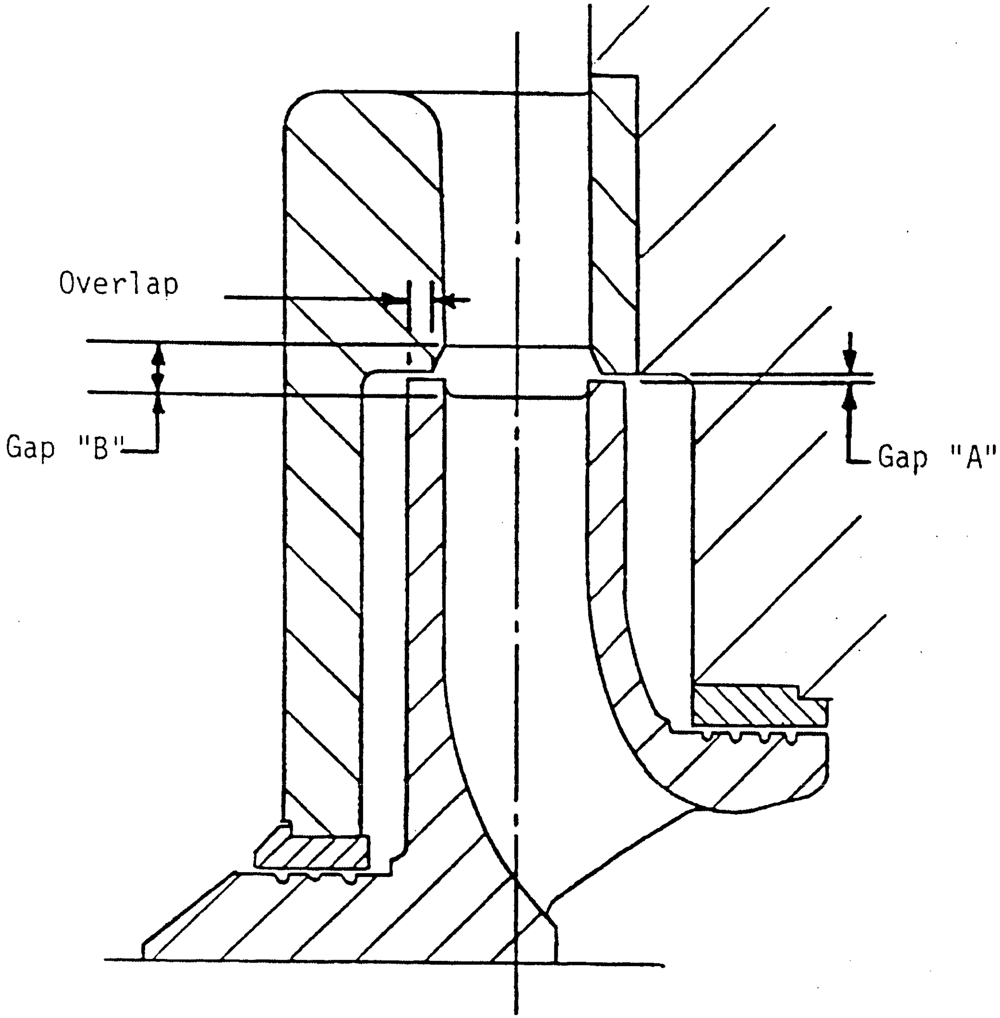


Flow rate = 70% of B.E.P. - Value

Gap "B" = 3% of impeller radius

(From ASME Article by T. Iino)

Gap A and Gap B



- A: **Gap between shrouds is important. Low flow: reverse flow out from diffuser.**
  - **Small gap A keep from getting into impeller zones. Maintain good HQ.**
- B: **Gap between blades is important.**
  - **Large gap B avoids axial thrust instabilities. Typically Makay ~6%**

WFR Impeller Characteristics

- Require continuously rising head as flowrate is reduced
- If H-Q is positive or zero, the pump will surge (Paul Rothe: "First Order Pump Surge")
- Therefore BEP Head Coefficient must be less than at shut-off
- For centrifugal pumps Ψshut-off ~ 0.6 (Stepanoff) However may be higher for low Ns pump

Diffuser Characteristics

- Impeller produces static pressure rise due to centrifugal effects and diffusion as well as velocity head
- Diffuser must slow down this fluid efficiently i.e. produce static pressure rise that is as close to velocity head decrease
- Flow incidence sensitive
- Leading edge stall phenomenon believed to be cause of loss of diffuser performance
- Rapid head falloff at low flow
- Vaned Diffuser Flattens H-Q curve characteristics

Commercial Diffuser Pumps

- Commercial pumps, such as boiler feed pumps, are required to operate over wide flow ranges including up to shut-off.
- How do they do this?
  - Demand less from diffuser
    - →design lower head coefficient
      - larger diameter, (more U for less $V\theta$)
      - less swirl to be diffused by diffuser
  - Fewer low-angle diffuser blades
  - Wrap around a lot
- This experience will be heeded in the design process

Concepts Evaluated

- Shifted Design Point (to lower ϕ)
- Volute Only
- Volute and Radial Vaneless diffuser
- Vane dedicated to WFR
- Slotted or Tandem Vanes
- Variable Inlet Guide Vane
- Active Control
  - Mechanical complication
  - Increase weight
  - Reliability

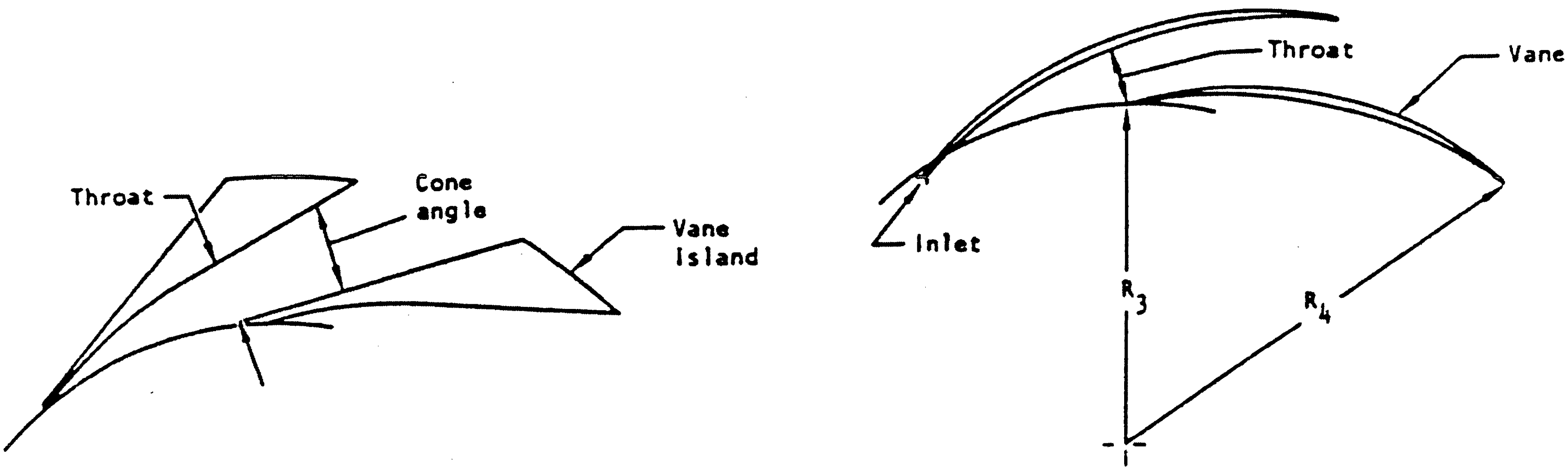
Why Do Vaneless-Volutes Have Wide Flow Range?

- **Strong secondary flows that cleans low momentum fluid off the outer and side walls**
- **Velocity entering throat is lower than when left impeller -> less diffusion to be done**

- **Benefits of vaned diffusers**
  - **Enables load capability**
    - **→Stress consideration**
  - **Needed to get better η on design point**

Vanes

- **Vane-Island, symmetric about center streamline**
- **Constant thickness thin vane type**

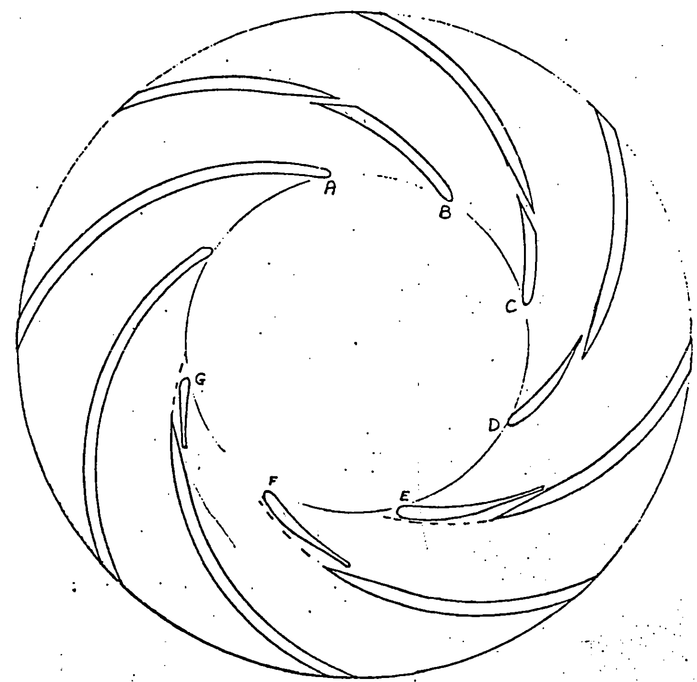


(a) Vane Island diffuser

(b) Vaned diffuser

Slotted or Tandem Vanes

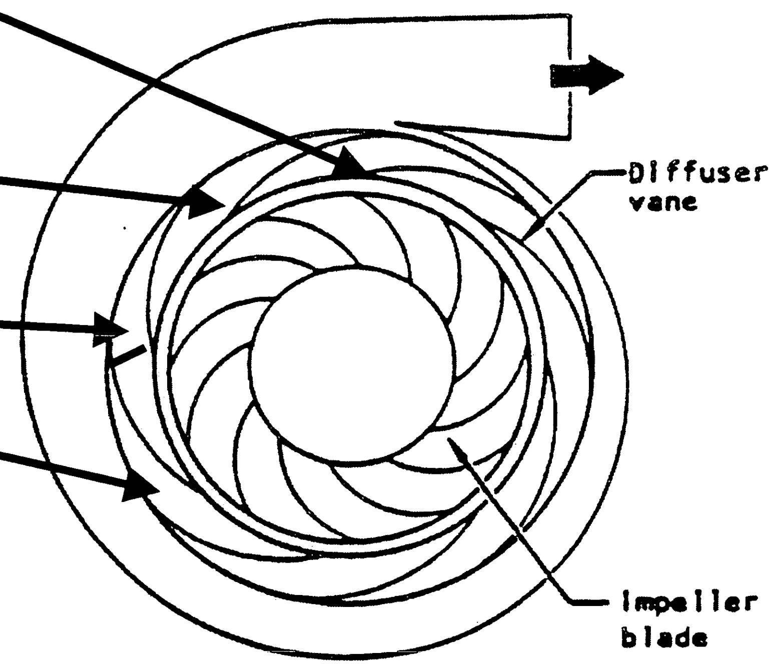
- **Energize suction surface boundary layer with pressure surface flow to prevent separation and cavitation**
  - **Difficult to fabricate**
  - **Requires testing to optimize**
    - **→2D boundary layer code does not account for inward flow on side walls**
    - **→CFD does not handle boundary layers accurately (wall models)**
    - **→Secondary flows make the boundary layer thinner**
    - **→Hard to judge location of slot through analysis only**



Slot geometry configuration optimization from Gostelow and Watson, 1972.

Key Diffuser Design Features Drive Performance and Flow Range

- Impeller - Diffuser Gap
- Number of Blades
- Leading edge
- Leading edge to throat
- Throat
- Diffusion passage
- Exit to volute



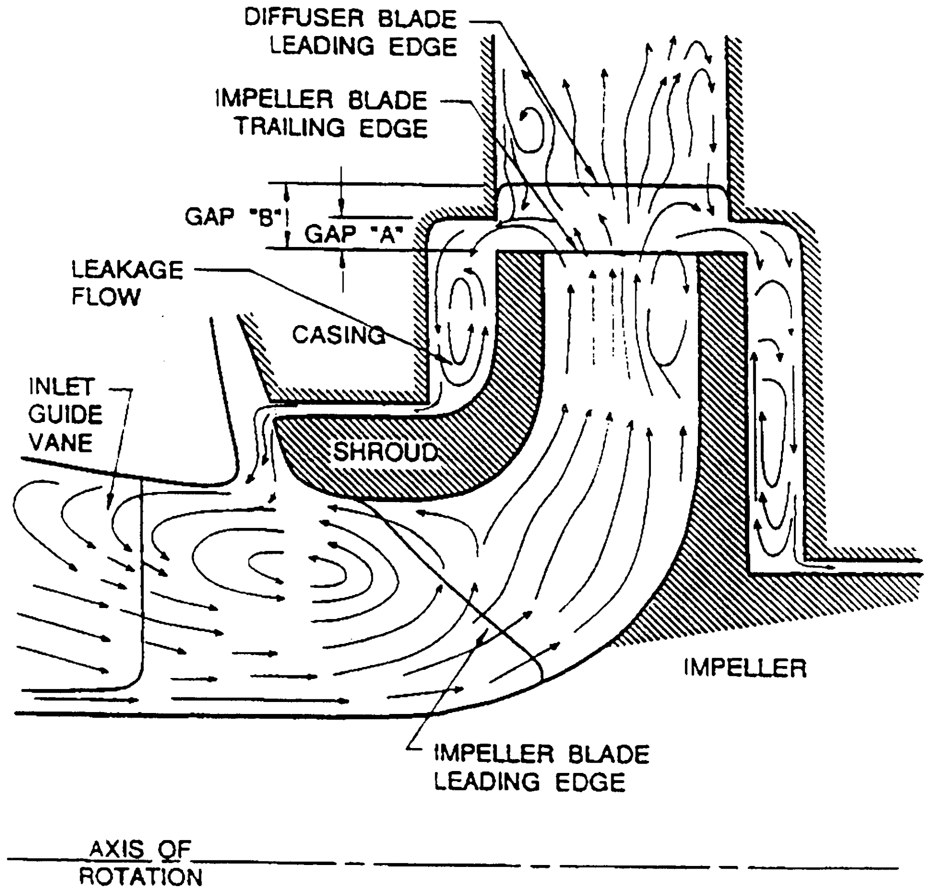
Gap A and Gap B Empirical Selection (based on Makay field work)

A: 1% R_2

shroud overlap ~ 4 x gap A

$D_3/D_2=1.1$

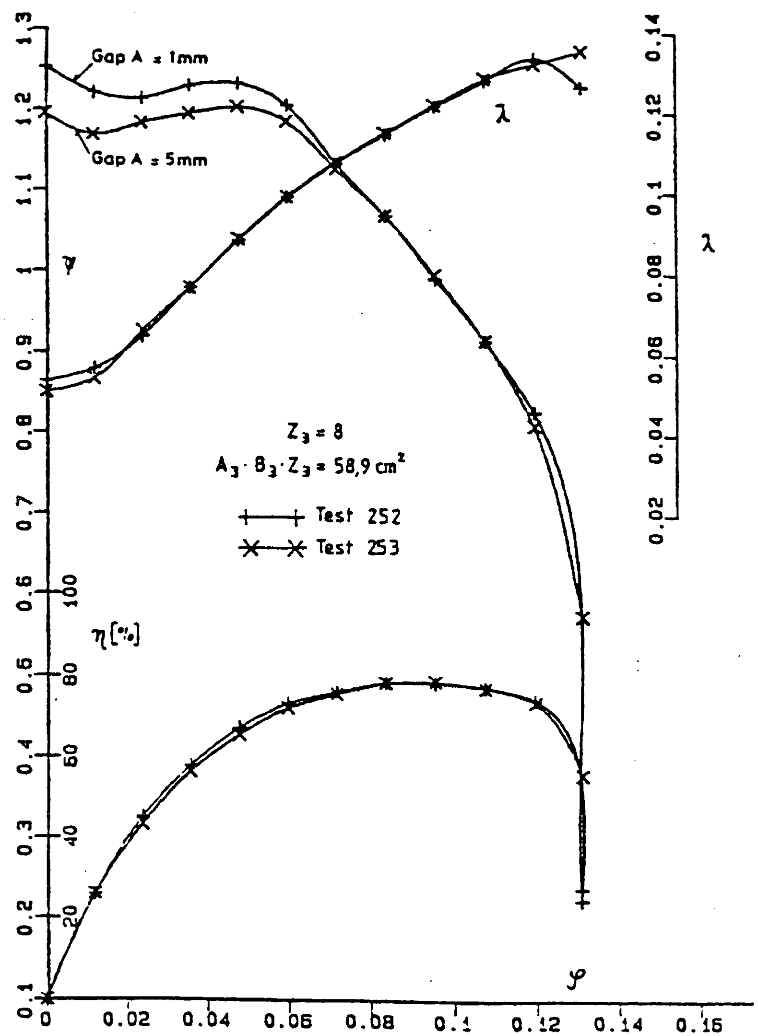
B: 10% R_2



Secondary flows in a centrifugal pump from Brennen (1994)

A: Gap Between Shrouds

- Low flow: reverse flow.
- Small gap A. Keep flow out diffuser from getting into impeller zones.
  - Gap A study by Guelich et al.
- Maintain good H-Q.



B: Impeller/Diffuser Clearance

- High energy pumps => a lot of interaction (jets, wakes)
- Need large to avoid axial thrust instabilities
  - Typically ~ 6% (Makay)
- < 4% dangerous in high energy pumps operating off-design.
- Slurry pumps up to 40%!!!
  - Do not want too big or get vaneless diffuser instability.
  - Seen in compressors with high exit angles => stall cells)

Number of Blades

- Avoid stator rotor interaction per Bolleter's paper
- 5 or 7 ok for 12 bladed impeller
  - Less blades longer diffuser (heavier pump)
  - Less blades shallower angle for WFR
  - (rarely exceed 7 in traditional commercial pumps)

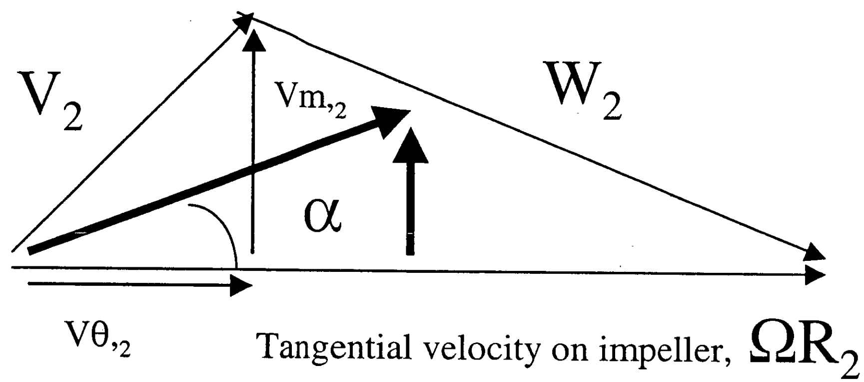
Leading Edge Characteristics

- Shape like airfoil: wedge
  - Blade thickness development
- Match diffuser flow approach angle to diffuser blade camber line
  - Diffuser blade leading edge set by the impeller discharge flow
    - →6 degrees.
- Suction side is critical
- Pressure side determined from suction side
- Elliptic nose

→Tolerate large variation of incidence angle

Incidence Losses Off-design

- **Flow velocity coming off impeller is lower at BEP and higher α**



Q = Flow

A = diffuser area

V or C = absolute velocity

W = relative velocity

2 = impeller exit

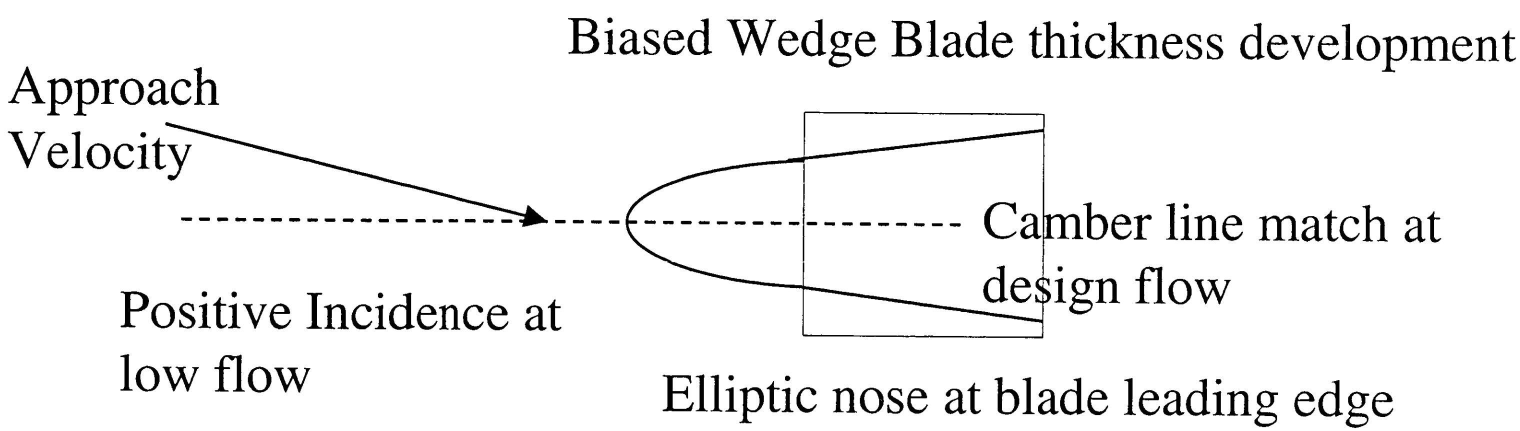
3 = diffuser inlet

α = angle of absolute velocity vector from tangential direction

  - **at BEP diffuses from V_2 to $V_3 = Q_{bep}/A$**
  - **low flow, larger V_2 to smaller V_3**

Leading Edge Model

- Biased Wedge [Cooper et al.] applied to a diffuser blade
- Tolerate wide variation of incidence angle
- Concept first applied to impeller of a high energy pump at Flowserve (pump OEM)



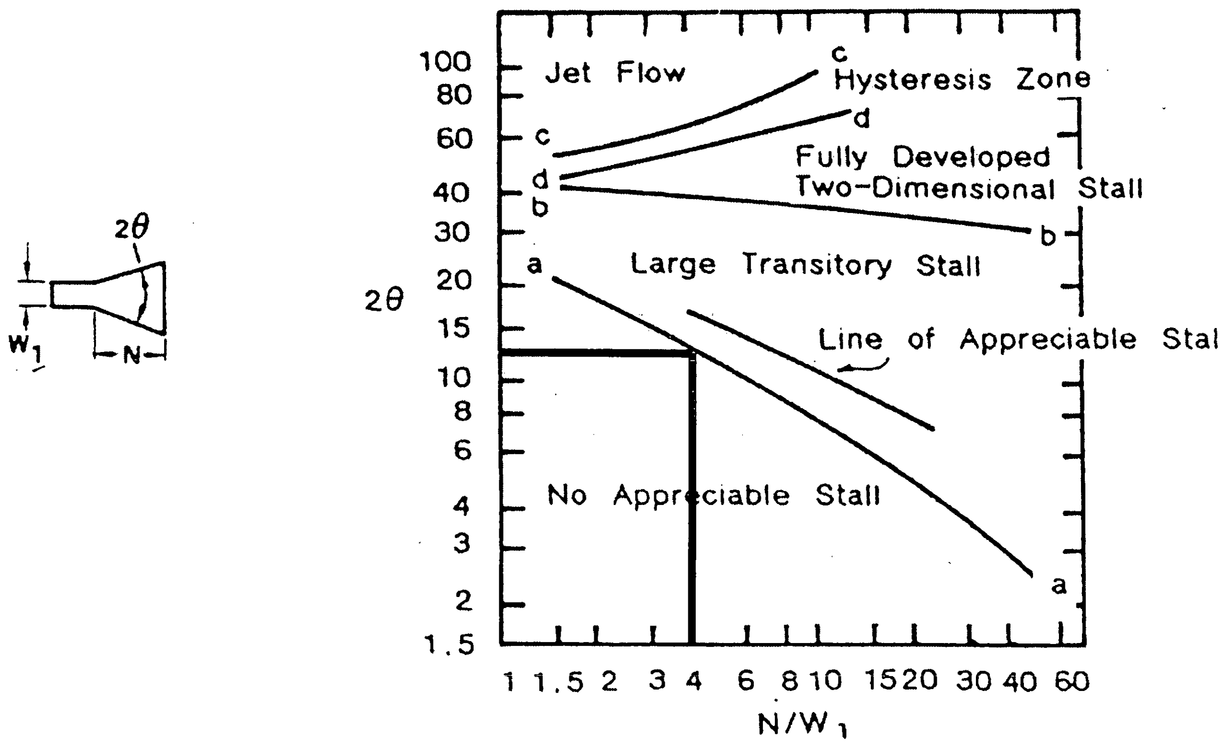
Throat Model

- **Correct area driven by BEP**
  - L2F measurements available at 100% flow
- **Diffuser width, b_3 set larger than impeller width, b_2**
  - $b_3 = 1.2\ b_2$
  - Pump OEM practice (Pump Handbook)
- **Throat area (Kovats)**
  - $b_3 . w_1$
  - $r_T V_T \sim 0.8\ r_2 V_{\tau 2}$
    - $\rightarrow w_1$
  - Blockage (conservative estimate of 9%)
  - Aspect ratio = 1.2
    - →(conservative estimate: losses 10% of recovery losses; Cp=0.71 instead of 0.73)
    - →trade with greater blade number and therefore greater blade angle

Diffusion Passage Model

- **Vaned portion performs rest of diffusion**
- **Pump industry**
  - **Typical combination: $A_{ex}/A_T = 1.8$, and $N/w_1 = 4$**
  - **per Kline charts for stable diffuser flow**

Two-Dimensional Diffuser Map



Flow Regimes in Straight Wall, Two-Dimensional Diffusers from Moore and Kline, 1955.

Why Design with Few Long Blades with Large Wrap?

- **Long diffuser blades enables secondary flow to develop**
  - migration of low-momentum fluid from pressure side along sidewalls (hub and shroud) onto suction side farther downstream
    - →Time for low momentum fluid to add gradually to thickening wake
- **Accommodates incidence variation onto diffuser vanes**
  - effective at allowing stall to develop gradually over a wide flow range

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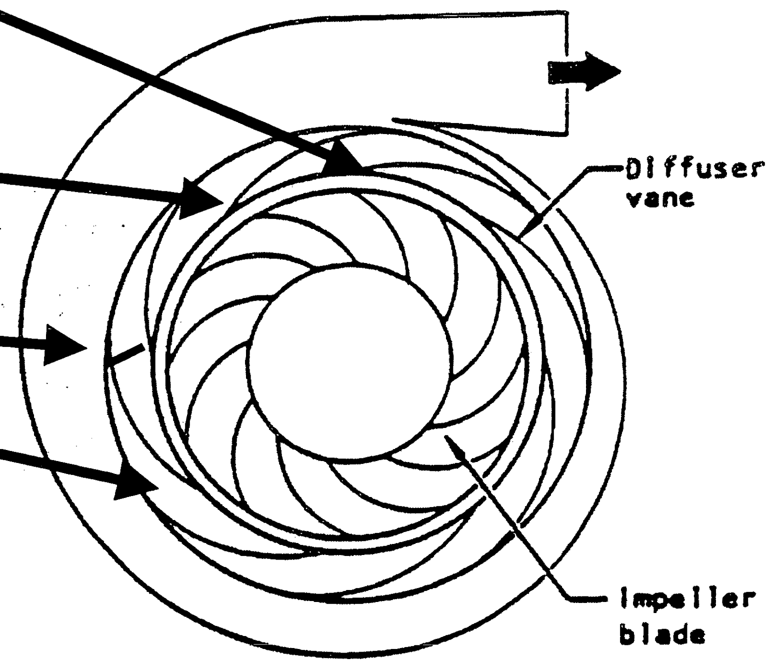
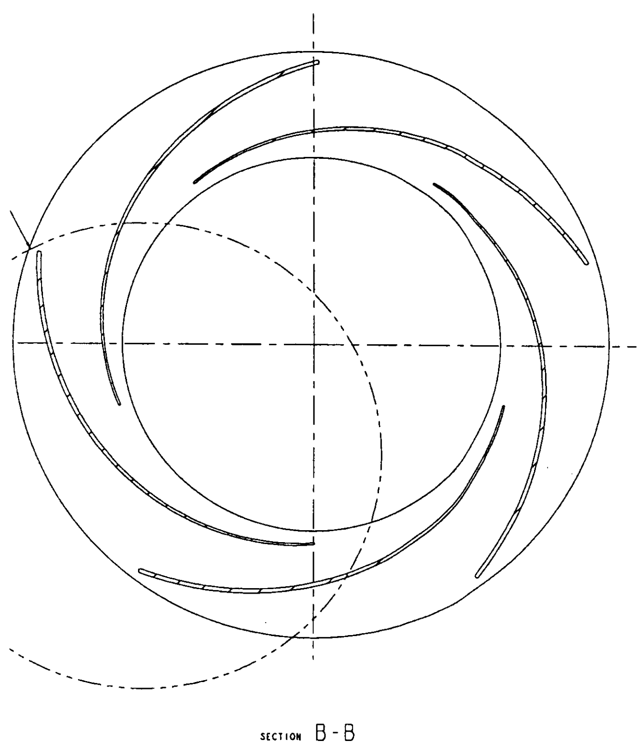
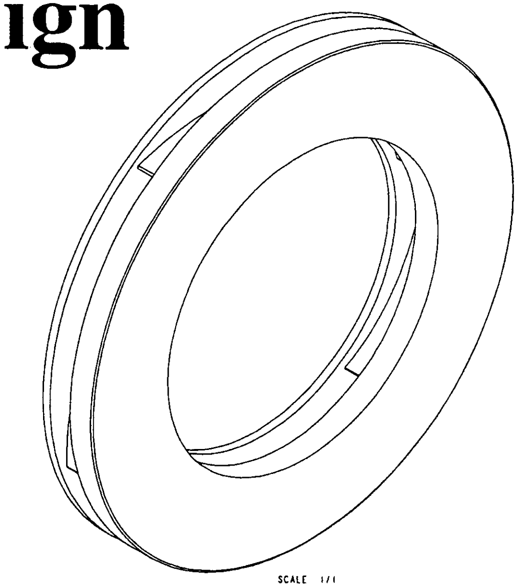


Detailed Design





CFD Methodology

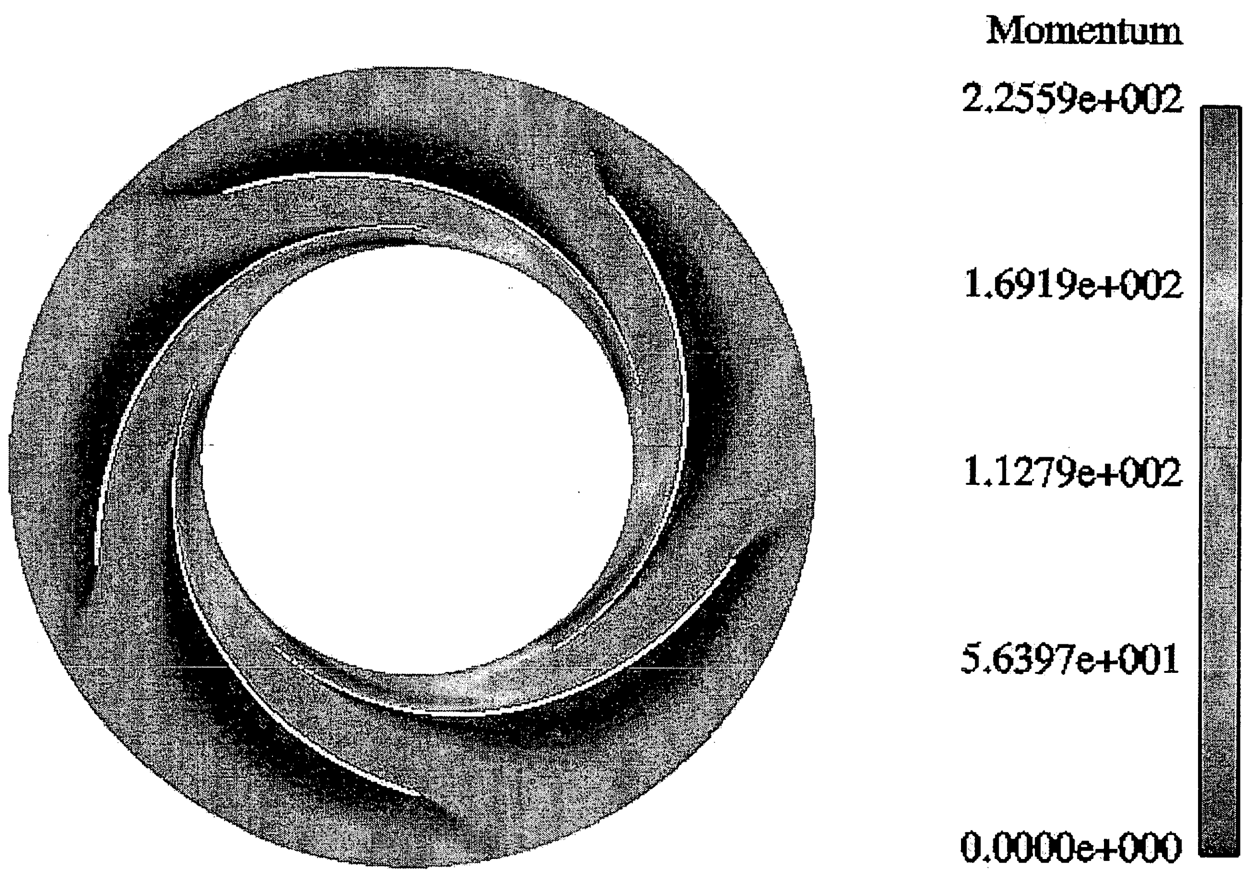
- **Objectives**
  - **Simulate WFR Constant Vane Thickness Diffuser Design**
    - →Note: CFD tends to give an optimistic view of stall performance

- **Methodology**
  - **eTango featuring Enigma CFD**
    - →Incompressible, steady-state 3-D flow model
      - Non-cavitating flow model
    - →Standard two-equation turbulence model
  - **APPT Grid Generation (~50,000 Flow Nodes)**
    - →Pro-E IBL to APPT

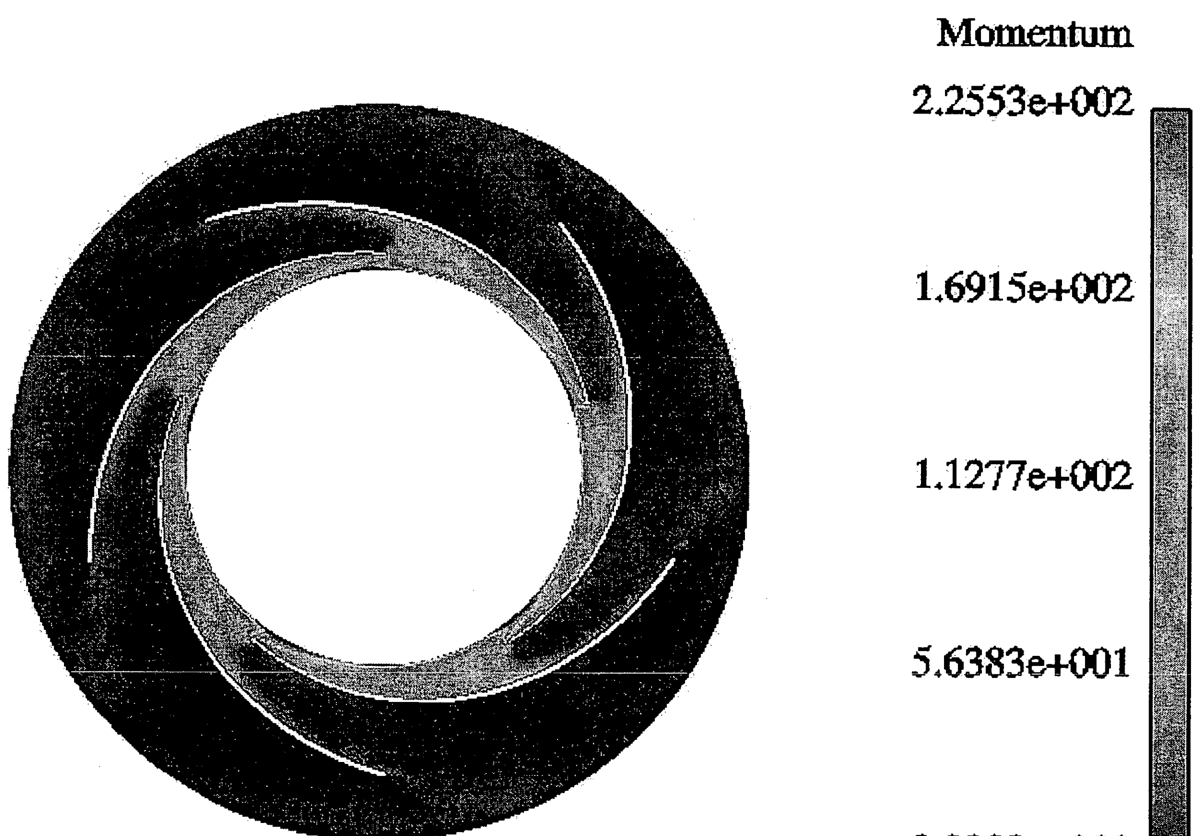
CFD Flow Cases

- Case1 - 5 Blades, Q design (thin vane)
- Case2 - 5 Blades, Q design (final design)
- Case3 -   20% Q design
- Case4 - 5 Blades, Q design (leading edge angle variation)
- Case5 -   20% Q design
- Case6 - 7 Blades, Q design

5 Blades, Q_{design}
Velocity Magnitude



5 Blades, 20% Q_{design}
Velocity Magnitude



CFD Flow Cases Results
static pressure rise

Type	Design (psf)	Off-design (psf)
Consortium Vane Island	29500	14 600
Thin Vane	29 648	
design #1	**29 563**	**21 245**
Δ angle	27 085	
7 blades	26 554	

WFR Diffuser Characteristics

- Impeller/Diffuser gaps
- Fewer low-angle diffuser blades
  - keep incidence low
- Camber line at leading edge
  - Minimize angle of attack
- Trade bulbous nose with blade thickness and therefore increased blade angle (analogy venetian blinds)
  - risk separation on suction side of diffuser
- Throat governs the flow
- Large Wrap