Pumping Machinery

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What is Turbomachinery?

Using working fluids to **Boost** output,

either **increase** or **decrease** pressure

by using **Machinery**

High Pressure Fuel Turbopump
Turbomachine Classification

- Turbines. Pumps and Compressors
- Incompressible. Compressible
- Axial-flow, Mixed-flow, Radial-flow geometry
- Single stage. Multi-stage
- Turbo-pump. Turbo-compressor. Torque-converter
- Impulse. Reaction
From Customer Requirements to Final Product

- Specification
- Preliminary Design, Conceptual design, ...
- Component Design
- Component Test, Analysis
- Acceptance Test
- .....
Design Trade-offs

• Performance
• Weight
• Cost
• Life
• Reliability
• Structural Strength
• Maintainability
• Envelope

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Design Process

- In-house design database - scale
- Detail design
  - (2D, Quasi 3D, CFD <-> stress analysis)
- Test Data Evaluation
Turbomachine

\[ Ps = \dot{m} \left[ \left( h + \frac{v^2}{2} + gz \right)_{\text{out}} - \left( h + \frac{v^2}{2} + gz \right)_{\text{in}} \right] \]
Turbines

• Impart Kinetic Energy to rotor as Mechanical Energy of rotation

• Impulse
  – high Pressure, low Flow

• Reaction
  – low Pressure, high Flow
Pump Classification

from the Pump Handbook
Centrifugal Pump

- rotor, stator
  - accelerate flow by imparting kinetic energy
  - decelerate (diffuse) in stator
  - results in increase in fluid pressure
Elements of a Centrifugal Flow Pump

- From Huzel, D. K. and Huang, D. H.
Rotor

- Inducer
- Impeller
- Bearings
- Shaft
Inducer

- Axial flow
- Increase total pressure
- permits non cavitating operation in impeller
- used as boost pump, permits main pump to operate at higher speeds
- e.g. LPOTP is only inducer
Inducer
Stator

- Casing
- Diffuser vanes
- Volute
- Seals
Vane Island Diffuser
(shown without shroud)
Impeller Profiles

Radial Flow          Mixed Flow          Axial Flow

$N_s$

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Pump Configurations

- From Huzel, D. K. and Huang, D. H.
Velocity Triangle

- From Huzel, D. K. and Huang, D. H.

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Velocity Triangle

- From Huzel, D. K. and Huang, D. H.
Impeller Loss Components

• Skin Friction
• Blade Loading
• Incidence
• Wake Mixing
• Impeller-shroud Clearance Leakage
• Disk Friction
• Recirculation
Flow Variables

\[ P_T = P + \frac{1}{2} \rho v^2 \]

\[ P_T = P + \frac{1}{2} \rho (v_\theta^2 + v_m^2) \]

\[ h_T = h + \frac{1}{2} \rho v^2 + gz \]
Dimensionless Quantities

- Head coefficient
  \[ \Psi = \frac{gH}{\Omega^2 R^2} \]

- Flow Coefficient
  \[ \Phi = \frac{Q}{A\Omega R} \]
Head rise

\[ H = \frac{u}{g} (v_{\theta 2} - v_{\theta 1}) \]
Isentropic Enthalpy Rise

$\Delta H = 144 \cdot \frac{\Delta p}{\rho}$

$\Delta P \text{ (psi)}$

$\Delta H \text{ (ft)}$

$\rho \text{ (lb/ft}^3\text{)}$
Affinity Laws

- \( Q \sim \Omega D^3 \)
- \( H \sim \Omega^2 D^2 \)
- \( T \sim \rho \Omega^2 D^5 \)
- \( P \sim \rho \Omega^3 D^5 \)
Engine System Resistance and Pump Characteristics

- From Huzel, D. K. and Huang, D. H.

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Specific speed

- Consistent units
  - $\Omega$ (rad/s)
  - $Q$ (m$^3$/s)
  - $H$ (m)

- US

$$Ns = \frac{\text{RPM} \times (\text{GPM})^{\frac{1}{2}}}{(\text{ft.})^{\frac{3}{4}}}$$

$$\Omega_s = \frac{Ns}{2734.6}$$
Impeller Profiles

From BWIP pump pocket book

Radial Flow         Mixed Flow         Axial Flow

$N_s$

Radial-Vane Area  Francis-Vane Area  Mixed-Flow Area  Axial-Flow Area

Impeller Shrouds  Impeller Shrouds  Impeller Hub

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Effect of $N_s$ on H-Q curve

- From Cameron Hydraulic Data
Profiles and Efficiencies Based on Specific Speed
Issues

• H-Q instability
• Stall
• Cavitation induced dynamic pressure
• Radial loads
• Discharge and suction recirculation
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
Recirculation

Secondary flows in a centrifugal pump from Brennen (1994)

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Recirculation

Sudden increase in pressure pulsation from Fraser (1981)

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Stator Effect on Head Characteristics

(steepens H-Q curve)

- reduce impeller inlet Cu at low flow
- increase impeller inlet Cu at high flow
- provide stability over wide operating range
- increase stator and impeller incidence angle at off design
- reduces inception of stall with negative incidence
Vaned Diffuser Effect on Head Characteristics

(flattens H-Q curve)

• convert kinetic energy of fluid leaving the impeller into static pressure rise
• flow incidence sensitive
• leading edge stall phenomenon believed to be cause of loss of diffuser performance
• rapid head falloff at low flow
Stall Characteristics

Flow/Flow_{des}

Pump Head/Pump Head_{des}

- WFR
- no stall
- no diffuser stall
- no stator stall

Diffuser stall
Impeller stall
Stator stall

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Two-Dimensional Diffuser Map


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Slot Optimization

Slot geometry configuration optimization from Gostelow and Watson, 1972.

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Blade Loading

From Guinzburg et al. (1997)

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CFD as a Tool

• Before using a particular CFD code in a rotating machinery component design process, it is important to bracket the accuracy of the code results for that particular type of component.
Interpretation of CFD

- Another important issue is how accurately the component *inlet flow boundary conditions* have to be known (pre-computation) to get results that are consistent with test data.
CFD Process

• Validate a computational fluid dynamic code for integration into the impeller design process.

• The validation process consists of computing the impeller flow for a range of inlet conditions.
Accuracy of the CFD Results

- number of nodes used to discretize the flow domain
- accuracy of the numerical discretization scheme
- type of turbulence model used.
CFD Capabilities

- Transient Analysis
- Two Phase Flow
- Heat Transfer
- Temperature Dependent Properties
- Moving Mesh
- Non-inertial Reference Frames
- Selection of Turbulence Models
- Wall Function Models
Diffusion

The diffusion factor $D$, can be adapted for pumps from Lieblein (1965) as follows:

$$D = \left(1 - \frac{W_2}{W_1}\right) + \frac{V_{\theta_2} - \frac{r_1}{r_2} V_{\theta_1}}{2 \sigma W_1}$$

Duncombe (1964) explicitly examined the diffusion on both the suction ($s$) and pressure ($p$) sides of the blade and expressed the result as follows:

$$D = \left(1 - \frac{W_2}{W_{s,\text{max}}}\right) + \left(1 - \frac{W_{p,\text{min}}}{W_1}\right)$$
Cavitation

Typical cavity configuration within an impeller. Flowrate is half that of BEP; so, the cavity is broken up by recirculating flow. From Sloteman et al (1995).
Cavitation

- Thoma number, cavitation number

\[ \sigma = \frac{p - p_v}{\frac{1}{2} \rho v^2} \]
Suction Specific speed

- Consistent units: \( \Omega_{ss} = \frac{\Omega Q^{\frac{1}{2}}}{(g\text{NPSH})^{\frac{3}{4}}} \)
  - \( \Omega \) (rad/s)
  - \( Q \) (m\(^3\)/s)
  - \( \text{NPSH} \) (m)

- US: \( \text{Nss} = \frac{\text{RPM.(GPM)}^{\frac{1}{2}}}{(\text{ft.})^{\frac{3}{4}}} \)  \( \Omega_{ss} = \frac{\text{Nss}}{2734.6} \)
Pump Suction Performance

- From Huzel, D. K. and Huang, D. H.

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Bucket Curve

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Experimental Inducer Cavitation

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Leading Edge Cavitation Damage

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Typical Pump performance curve, showing NPSH required a) to maintain hydraulic performance or pump head (NPSH$_R$), b) to limit cavitation damage and therefore maintain pump life (NPSH$_d$), c) to prevent bubble formation entirely (NPSH$_i$) from Cooper and Antunes (1983)
Pump Losses

- mechanical
- hydraulic
- disk friction
- leakage
Radial Load

Comparison of radial forces

*Figure 12. Radial Thrust Vs Flow for Single Volute, Double Volute, and Diffuser Designs (Courtesy of Stork Pumps).*
Radial Load Profiles for Volutes of Different Specific Speed

Fig. 8 Comparison of the effect of three casing designs on radial force for $N_s = 55$

Fig. 9 Comparison of the effect of three casing designs on radial force for $N_s = 100$

Fig. 10 Comparison of the effect of three casing designs on radial force for $N_s = 165$

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Axial Calculation
Axial Load Balancing Schemes

• Seal Leakage Return Path
• Pump out ribs or vanes
• Balance Drum
• Balance Disk
Pump Balance Piston

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Rotordynamics

Relationship between the forces in the pump frame and the rotordynamic forces from Brennen (1994)

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Impact Testing
References

- BWIP Pump Pocket Book.

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