TAEK Ankara

Pump Training for Inspectors

**Agenda for a 3-day training course**

**Day 1**
- Introduction / Basics / Preparation
- Location: TAEK-Office
  - **Morning:** Basics of centrifugal pumps
  - **Afternoon:** Briefing for pump acceptance test according to ISO 9906

**Day 2**
- Practical pump acceptance test
- Location: LayneBowler Pump Company Inc. in Ankara
  - Test of a pump on LayneBowler’s test rig
  - Inspection report

**Day 3**
- Debriefing / Discussion / Questions / Conclusions
- Location: TAEK-Office
TAEK – Pump Training for Inspektors
Basics of centrifugal pumps (CP) / morning

Ambition and motivation

✓ You learn what types of pumps exist
✓ You learn the main differences between centrifugal pumps (CP) and displacement pumps (DP)
✓ You learn how to classify centrifugal pumps
✓ You learn how to do basic hydraulic calculations
✓ You learn the four most important characteristic pump curves
✓ You learn how to evaluate the suction capabilities of pumps
✓ You learn the general characteristic curve of pipe systems
✓ You learn the two most important types of flow regulations
Centrifugal pumps (CP)

Generally:
- A wide range of centrifugal pumps exist
  - Radial pumps
  - Mixed flow pumps
  - Axial pumps

- Name (axial, radial, mixed flow) refers to the direction of outflow:
Some examples for centrifugal pumps

Single stage single flow radial pump (radial means: Outflow in radial direction)

Characteristics:
- axial inflow
- radial outflow
- single flow (one impeller)
- volute casing (cast steel)
- antifriction bearing
- standard pump

Application: Waterpump

Single stage, double flow radial pump

Characteristics:
- single stage (one impeller)
- double flow (two suction sides)
- radial inflow and outflow
- spiral casing (cast steel)

Advantages:
- compakt design
- increased inflow in comparison to single flow pumps
- low suction pressure possible (low NPSH_R)
- nearly balanced axial impeller forces
- inline assembly (no pipe redirection necessary)

Application: Water supply systems (pumps)
Multistage radial pump / modular concept

Characteristics:
- axial inflow
- radial outflow
- 3-stage single flow
- modular pump housing with tie rods
- each impeller with diffuser (guide arrangement/diffuser)

Advantages:
- kompakt and flexible design for different pressure requirements
- depending on no. of stages, only the tie rods and shaft have to be adjusted
- Quick warm-up after start
- low warpage
- good as „stand-by-pumpe“

Disadvantage:
- complex maintenance procedure
- simple exchange of rotor impossible
- possibly leakage problems when heating up/cooling down
Feed water pump / barrel type

+ solid pump housing for high pressure
+ quick maintenance / easy rotor exchange
+ no leakages of pump housing / no static seals
+ only two shaft seals (mechanical seals)
+ ideal for continuous operation

- high thermal warpage
- unsuitable as „stand-by-pump“
- warming-up / preheating before operation necessary, otherwise warpage
- expensive
Canned motor pumps (without shaft seal – no shaft leakages)

Characteristics:
- only static seals (o-rings / gaskets)
- pump is „technically tight“ (no leakages at stuffing box or mechanical seal)
- cooling flow in the motor required (losses of efficiency)
- stator is dry, rotor is wet
- complete balanced rotor (regarding axial impeller forces) due to internal balancing device
- elektr. losses of efficiency due to ≈ 0,5 mm gap (lower total efficiency rate)
- suitable for „smaller“ pumps (clearly below 1 MW)
- very sensitive to any kind of particles in the fluid => only for pure and clean liquids!
Vertical single/multi stage axial tube pumps

**Version 1**
- multistage (5 stages) (relatively high pressure)
- radial impellers
- closed pump housing at the bottom
- connections for suction and discharge pipes on the same vertical level
- low demand of space
- good suction capability

**Version 2**
- single stage
- one axial (or mixed flow) impeller
- relatively low pressure and high flow rates
- no suction line / directly submerged into the water

**Diagram:**
- **Version 1:** 5 radial impellers, discharge side connected to discharge pipe, suction side in open water
- **Version 2:** 1 axial impeller
Displacement pumps (DP)

Generally:
- A wide range of displacement pumps exist
- Rotating Displacement Pumps (RDP) and Oscillating Displacement Pumps (ODP)

Examples for rotating displacement pumps (RDP):
- Gear type pump (single and double flow), internal / external gear pump
- Screw pump (1, 2 or 3 screws)
- Eccentric screw pump

Examples for oscillating displacement pumps (ODP):
- Piston pumps
- Diaphragm pumps

In NPP, most of the pumps are centrifugal pumps!
Example for displacement pump (ODP: 3-piston pump)

Characteristics:
- 3-piston-design to reduce pulsation (3x120°)
- driven by crank shaft
- for high pressure applications
- clearly defined flow rates independent from pressure
- pulsation damper are generally necessary for suction and/or discharge side
- pressure relief valve always necessary
- complex design

Application: For dosage / metering such as boric acid in NPPs
**Centrifugal pumps**

n = konstant

Remark for centrifugal pumps:
- there is always one **Best Efficiency Point (BEP)**
- centrifugal pumps are so-called „hydrodynamic pumps“

**Displacement pumps**

\[ n_1 < n_2 < n_3 < n_4 \quad \text{oder} \quad V_{H1} < V_{H2} < V_{H3} < V_{H4} \]

Variation of flow rates by adjustment of
- speed or/and
- stroke

**For both types of pumps applicable:**
Discharge pressure generated by the pump always depends on the flow resistance of the discharge piping system!

Remark for RDP / ODP:
- there is **no** BEP
- P-Q curves with slight inclination due to internal leakages => internal increasing backflow at high pressure
- RDP/ODP are so-called „hydrostatic pumps“
Definition of operating ranges for centrifugal pumps

- **Zero flow head** (Q = 0)
- **Operation in part load**
- **Operation in over load**

- **BEP** (Best Efficiency Point)
- **Volumetric flow rate Q**
- **Head H ("pressure")**

**Graph:**

1. **System curve**
2. **Pump curve (example)**
3. **BEP (Best Efficiency Point)**
What defines the operating point of a (centrifugal) pump?

It is always the point of intersection between the pump curve and the pipe system curve!
Characteristics of CP and DP - differences between CP and DP (ODP and RDP)

**Centrifugal pumps:**
- transport of low viscosity fluids (approx. up to $v \approx 1,000 \text{ mm}^2/\text{s}$)
- significantly falling efficiency rates with increasing viscosity
- relatively high flow rates at relatively low pressure
- transport of very high flow rates
- flow rates can be adjusted / regulated easily
- CP are relatively cheap pumps in comparison to RDP/ODP
- relatively compact design
- no pressure relief valve necessary

**Displacement pumps** (ODP or RDP):
- transport of high viscosity fluids (no limits as long as it is a liquid)
- flow rates nearly independent from pressure at discharge side
- transport of very small flow rates (e.g. espresso machine)
- perfect for dosage / metering applications
- flow rates easy to regulate by motor speed and/or stroke adjustment (ODP)
- ideal for very high pressure applications
- high efficiency rates
- expensive design
- pressure relief valve always necessary
- ODP: flow rate with undesired pulsation => damping device often necessary
Classification of centrifugal pumps by a numeric definition

Centrifugal pumps are divided in 3 groups:

- radial pumps
- mixed flow pumps
- axial pumps

These 3 expressions refer to the direction of impeller outflow

The distinctive feature is the so-called "Specific Speed \( n_q \)"

\( n_q \) is a characteristic numeric value with an extensive relevance for CP!

Definition

\[
n_q = n \cdot \sqrt[3]{\frac{Q}{H^4}}
\]

\( n_q \) = specific speed \([1/min]\)
\( n \) = real operational speed \([1/min]\)
\( Q \) = flow rate \([m^3/s]\)
\( H \) = head \([m]\)

(if applicable “pressure / (density x gravity acceleration)"

Remark:

- applied values for \( Q \) and \( H \) have to refer to BEP (at \( \eta_{\text{max}} \))!
- ignore the discrepancy for the units
- for multistage pumps take the head of one stage
- for double suction pumps take the half flow rate
Specific speed – numeric values to be regarded generously

radial pump impeller  \[ n_q \approx 10 \ldots 50 \text{ min}^{-1} \]

mixed flow pump impeller  \[ n_q \approx 50 \ldots 150 \text{ min}^{-1} \]

axial pump impeller  \[ n_q \approx 150 \ldots 500 \text{ min}^{-1} \]
Example application for $n_g$:
Read and understand diagrams for the complete range of centrifugal pumps

- operation in over load BEP
- operation in part load
- zero flow head ($Q = 0$)
Basic formulas and calculations for CP

What does a pump do? Three things:

- increase pressure between suction and discharge side (in many cases the decisive part)
- increase flow velocity between suction side and discharge side (if diameter of discharge pipe is smaller than the diameter of the suction pipe)
- lifts fluid from suction side to discharge side (if nozzles have a vertical height difference)

⇒ the sum of these parts is the „head“, unit is [m]
⇒ the head can be measured only indirectly by measuring the three parts and adding them together
⇒ a direct measurement of the head is impossible
Calculation of head

\[
H = \frac{p_D - p_S + \frac{v^2_D - v^2_S}{2}}{\rho \times g} + (z_D - z_S)
\]

- \( p_D \): pressure at discharge nozzle \([\text{Pa}]\)
- \( p_S \): pressure at suction nozzle \([\text{Pa}]\)
- \( v_D \): flow velocity at discharge nozzle \([\text{m/s}]\)
- \( v_S \): flow velocity at suction nozzle \([\text{m/s}]\)
- \( \rho \): density of fluid \([\text{kg/m}^3]\)
- \( g \): acceleration of gravity 9.81 \([\text{m/s}^2]\)
- \( z_D \): height of discharge pressure gauge referring to ground level \([\text{m}]\)
- \( z_S \): height of suction pressure gauge referring to ground level \([\text{m}]\)
Basic recommendations for hydraulic calculations

- First of all: Convert all relevant values (Q, p, ρ …) to **basic units (SI-units)!**
  That is: m, m², m³, s, s², m/s, m/s², m³/s, kg, and thereof derived units like Pa and N
- **Don‘t** calculate directly with units like m³/min, m³/h, l/s, bar, mm, cm, min and h
- After having a result in correct SI-unit, it can be converted to desired units like m³/min, m³/h, l/s, bar, mm, cm, km, min and h
- In case of any doubt‘s check the units

This is very important for a Hydraulic Performance Acceptance Tests at pump manufacturers!
Calculation of driving power / pump efficiency

\[ P = \rho \times g \times H \times Q \]

\[ \eta = \frac{\rho \times g \times H \times Q}{P} \]

- \( P \) = driving power \([W]\)
- \( \rho \) = density of fluid \([kg/m^3]\)
- \( g \) = acceleration of gravity \([m/s^2]\)
- \( H \) = head \([m]\)
- \( Q \) = flow rate / capacity \([m^3/s]\)
- \( \eta \) = pump efficiency rate \([1]\)
Calculation / determination of suction capability – NPSHR and NPSHA

Definition of NPSH:

NPSHR = Net Positive Suction Head (Required)
NPSHR = Suction pressure (divided by ρxg), that is necessary for a satisfactory pump operation; unit is [m]

NPSHA = Net Positive Suction Head (Available)
NPSHA = Suction pressure (divided by ρxg), that is available at the impeller inlet; unit is [m]

Only the usable pressure can be considered

Usable pressure means = Absolut pressure minus the fluid’s vapour pressure
NPSH_{R/A} = (stat. + dyn. pressure + barometric pressure – vapour pressure) / (density x gravity)

NPSH = \frac{(p_{abs} - p_D)}{(\rho x g)} [m]
NPSH = \left( \frac{p_{SN,abs} - p_V}{\rho \times g} \right) + \frac{v^2_S}{2 \times g} = \frac{p_b + p_{SN} - p_V}{\rho \times g} + \frac{v^2_S}{2 \times g} \quad [m]

\begin{align*}
p_{SN,abs} &= \text{absolut static pressure at suction nozzle} \\
p_{SN} &= \text{relative static overpressure at suction nozzle} \\
p_V &= \text{absolute vapour pressure of fluid depending on temperature} \\
p_S &= \text{static overpressure at suction nozzle (vacuum pressure with negative sign)} \\
p_b &= \text{barometric air pressure} \\
v_S &= \text{flow velocity at suction nozzle} \\
\rho &= \text{density of fluid} \\
g &= \text{acceleration of gravity}
\end{align*}

**Question:**
How to determine the NPSH\textsubscript{A}-value at the pump’s suction nozzle?

**Answer:**
NPSH\textsubscript{A} can be calculated “easily“; see formula above

…precondition: A suction pressure sensor has to be available!
And if not? \Rightarrow See next page!
How to calculate $NPSH_A$ when there is a suction chamber above/below the pump and there is no suction pressure sensor in front of the pump?

$$NPSH_{A(Q)} = \pm H_{geo} + \frac{P_{air} - P_v - H_{v(Q)}}{\rho \times g}$$

with:

- $H_{geo}$ = geodetic vertical difference between water level of suction chamber and impeller inlet (NPSH-reference level)
- $H_{geo}$ is positive if the pump is below the water level of the suction chamber
- $H_{geo}$ is negative if the pump is above the water level of the suction chamber
- $P_{air}$ = absolute gas (air) pressure above the water level
- $P_v$ = vapour pressure of water depending on temperature
- $\rho$ = density of water (fluid)
- $g$ = acceleration of gravity
- $H_{v(Q)}$ = pressure drop ($\Delta P_v / (\rho \times g)$) in the suction pipe between water level and suction nozzle, unit [m]
Question:
How to determine the NPSH\textsubscript{R}-value of a pump?

Answer:
• a pump designer /manufacturer can estimate the value based on experience and impeller specific calculations (difficult)
• generally, it is measured on a pump test rig
• varying parameter on the test rig is the suction line pressure
• a criterion for definition has to be defined!

Possible criterions for NPSH\textsubscript{R} are:
• drop of head for 3% \Rightarrow \text{NPSH}_{3\%} (normal case!)
• incipient cavitation \Rightarrow \text{NPSH}_{ic} (special case)
• magnitude of vapour bubbles (e.g. 5 mm); only for special applications
• noise emission (only for small and cheap pumps)
• magnitude of vibrations (only for small and cheap pumps)
• for DP only: Drop of capacity (flow rate) of 2%, 3% or 5%
Example of measurement curve

One of these curves defines one measurement point in the Q-NPSH$_{R3\%}$-curve (see below). In order to get a complete Q-NPSH$_{R3\%}$-curve (see curve below), several curves (see curve above) have to be measured. The procedure is time-consuming and therefore expensive.

For this curve the flow rate is constant; a curve for each flow rate has to be measured!
ignore these 4 curves!
Question:
Why is the most common $\text{NPSH}_R$-criterion the 3% drop of head (for CP)?

Answer:
It's technically only a rough orientation and not very prudent, but it's relatively easy to measure!
The four most important characteristic pump curves

Characteristic curves for centrifugal pumps:
- Head $H$ [m]
- Driving power $P$ [kW]
- Suction capability $NPSH_R$ [m]
- Wirkungsgrad $\eta$ [1]

Each parameter depends on the flow rate $Q$ [m³/s]
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**Head – flow rate (Q-H curve)**

- **Remark:**
  Flat curves in the relevant operating range are not welcome, as flow regulation can get difficult (possibly acute-angled intersection point between pump curve and system curve)

**Important for all curves:**

- Declaration of
  - speed
  - outer diameter of impeller $D_2$, for axial pumps angle of blade
  - type of fluid (temperature, viscosity; generally cold water)
Driving power – flow rate (Q-P curve)

Shape of curve depends on the specific speed $n_q$ (type of impeller):

Case a: Radial impeller $n_q \approx 25 \text{ min}^{-1}$
Case b: Mixed flow impeller $n_q \approx 90 \text{ min}^{-1}$
Case c: Axial impeller $n_q \approx 180 \text{ min}^{-1}$

Power demand depending on outer impeller diameter $D_2$

Power demand increases / decreases significantly with variations of outer impeller diameter $D_2$!
**Efficiency – flow rate (Q-η curve)**

- For all centrifugal pumps (radial/mixed flow/axial or 10 min$^{-1} < n_q < 300$ min$^{-1}$) the general shape of the Q-η curve is nearly identical.

- For radial pumps the area of good efficiency is relatively wide. Pump operation in moderate part load or over load is no problem. For axial pumps operation in part load / over load is more restricted.

- Big axial flow pumps should be operated only close to (or in) the BEP, as part load or over load generates big losses and undesired effects.

![Graphs showing pump efficiency](image)

radial pump ($n_q \approx 25$ min$^{-1}$)  mixed flow pump ($n_q \approx 70$ min$^{-1}$)  axial flow pump ($n_q \approx 180$ min$^{-1}$)
Suction capability – flow rate (Q-NPSH$_R$ curve) for 3% drop of Head H

- In the over load area significant increase of NPSH$_R$ for all types of pumps
- In the part load area the pump type is decisive

![Diagram showing suction capability and flow rate](image-url)
Complete curve diagrams with variable impeller diameter

The following illustration of curves is often used by manufacturers:

Remark:
For better overview the efficiency rate is often integrated in the Q-H diagram.

The NPSH<sub>R</sub>-curve is (nearly) constant for all outer diameters, as small changes of outer diameter D<sub>2</sub> have no significant influence to the suction side.
Summary: Characteristic curves for pumps with different specific speeds

Attention:
Axial flow pumps shall not be operated in part load \( (Q/Q_{opt} \ll 1) \), because

- Driving power increases too much (torque and el. current are too high)
- Q-H curve is getting unstable (poor flow control mode)
- \( NPSH_R \) increases significantly (danger of cavitation)
- Discharge pressure increases significantly (and is possibly too high for the pump housing and the mechanical seal)

Recommendation for pump start:
- Axial flow pumps: Start with low discharge pressure (and high flow rate / over load) and then change to operating point
- Radial flow pumps: Start with high discharge pressure (low flow rate / part load) and then change to operating point
Characteristic curves of pipe systems

- **Static part** (is sometimes zero)
  - Geodetic altitude difference
  - Pressure difference between suction side and discharge side, if applicable

- **Dynamic part** (pressure drop due to hydraulic friction losses in the pipe, in installation parts, pipe elbows etc.)

\[ H_A = H_{\text{stat}} + H_j \quad \text{bzw.} \quad H_A = H_{\text{geo}} + H_{\Delta p} + H_j ; \text{mathematical:} \quad H_A = H_{\text{stat}} + k \times Q^2 \]

![Graph of characteristic curves of pipe systems](image)

- Typical pipe system curve at turbulent flow with static and dynamic part

\[
H_{\text{stat}} = \frac{p_2 - p_1}{\rho \cdot g} + H_{\text{geo}}
\]

\[
H_j = \sum \lambda \cdot \frac{1}{D} \cdot \frac{V_{\text{vol}}^2}{2g} + \sum \zeta_k \cdot \frac{V_{\text{vol}}^2}{2g}
\]

mit

- \( \lambda \) ... Rohrreibungskoeffizient z.B. aus Moody-Diagramm
- \( \zeta_k \) ... Einbauteil-Druckverlustkoeffizient
- \( l \) ... Länge des betrachteten Rohres
- \( D \) ... Durchmesser des betrachteten Rohres
- \( V_{\text{vol}} \) ... Volumetrisch gemittelte Strömungsgeschwindigkeit (Förderstrom durch Strömungsquerschnitt)
Flow regulation for centrifugal pumps

How can the pump’s flow be regulated?

By shifting the point of intersection between pump curve and system curve:

Examples

- possible operating points
- pump curves (red)
- possible system curves (blue)
Flow regulation for centrifugal pumps

Adjustable and reversible
Modification of pipe system curve
Regulation by flow restriction (throttle) in the discharge pipe

Modification of pump or motor
Flow regulation by variable pump speed

Adjustable, but not reversible
Modification of impeller
Reduction of outer diameter $D_2$
Sharpening of outer edges of impeller vanes
Regulation by flow restriction (throttle) in the discharge pipe

System curves before and after flow restriction

\[ B_1 = \text{operating point before flow restriction} \]

\[ B_2 = \text{operating point after flow restriction} \]

Power saving for a radial pump

\[ P_1 \text{ and } P_2 \]
Flow regulation by variable pump speed

\[ Q_2 = Q_1 \times \left(\frac{n_2}{n_1}\right)^1 \]
\[ H_2 = H_1 \times \left(\frac{n_2}{n_1}\right)^2 \]
\[ P_2 = P_1 \times \left(\frac{n_2}{n_1}\right)^3 \]
Modification of impeller in order to adjust head (increase or reduce)

During hydraulic performance acceptance tests it might be observed and measured that the head is too high/low (out of guarantee). As a result, suitable measures are required to adjust the Q-H curve. There are two possibilities to do this:

Reduce head: Reduce outer impeller $D_2$
Increase head: Underfiling of outer edges of impeller blades (at the „inconvenient“ side!)

Reduction of $D_2$:

Underfiling:
Three different types of outer edges of vanes:
1. Original shape
2. Normal underfiling
3. Max. underfiling

Modified Q-H curves (dashed)