Cavitation in Centrifugal Pumps and Prediction Thereof

Frank C. Visser

Flowserve Pump Division
Etten-Leur, The Netherlands

Tutorial

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Outline

• Part 1: What is cavitation and what does it mean for pumping machinery?
• Part 2: Prediction of cavitation in centrifugal pumps
  – Scaling laws
  – Thermodynamic effect (temperature depression)
  – Effect of dissolved or entrained gases
  – Calculating incipient cavitation ($NPSH$) from CFD
  – Cavity length prediction
Part 1 – What is cavitation

Cavitation is defined as the process of formation and disappearance of the vapour phase of a liquid when it is subjected to reduced and subsequently increased pressures.

The formation of cavities is a process analogous to boiling in a liquid, although it is the result of pressure reduction rather than heat addition.

Cavitation is a thermodynamic change of state with mass transfer from liquid to vapor phase and visa versa (bubble formation & collapse).
Part 1 – What is cavitation (cont.)

Sheet cavity on pump impeller vane leading edge (suction side)

Speed = 2990 RPM
NPSHA = 70 m (230 ft)
Flow rate = 1820 m³/h (8015 gpm)

Vane marker stripes at intervals of 10 mm (0.4 in)

Cavity length = 25-40 mm (1.0 – 1.5 in)

(from Visser et al, 1998)
Part 1 – What is cavitation (cont.)

Cavitation causes or may cause:
- Performance loss (head drop)
- Material damage (cavitation erosion)
- Vibrations
- Noise
- Vapor lock (if suction pressure drops below break-off value)

(Visser et al, 1998)

General Advice: TRY TO AVOID CAVITATION (under normal operation)

Unfortunately, economic or operational considerations often necessitate operation with some cavitation, and then it is particularly important to understand the (negative) effects of cavitation.

→ Design optimization to minimize cavitation
Part 1 – What is cavitation (cont.)

Typical cavitation damages

Centrifugal pump impeller cavitation pitting erosion @ inlet  
(from Dijkers et al, 2000)

Francis turbine runner cavitation damage @ discharge  
(from Brennen, 1994)
Part 1 – What is cavitation (cont.)

Cavitation behavior is typically expressed in terms of cavitation parameters.

- Cavitation number:
  \[ \sigma = \frac{p_1 - p_V}{\frac{1}{2} \rho U^2} \] (Centrifugal Pumps: \( U = U_{eye} = \Omega R_{1T} \))

- Net Positive Suction Head:
  \[ NPSH = \frac{p_{01} - p_V}{\rho g} \]

- Thoma cavitation number:
  \[ \sigma_{TH} = \frac{NPSH}{H} \]
Part 1 – What is cavitation (cont.)

In general, cavitation performance is related to some “critical” value:

\[ NPSHA \text{ (=available)} > NPSHc \text{ or } NPSHR \text{ (=critical or required)} \]

Typical “critical” characteristics identified for centrifugal pumps:

- Incipient cavitation \((NPSHi)\)
- Developed cavitation causing 3% head drop \((NPSH3\% )\)
- Developed cavitation causing complete head breakdown \((\rightarrow \text{vapor lock})\).

Choice of \(NPSHR\) is rather arbitrary, but usually \(NPSHR=NPSH3\%\)
Alternative choices:

- \(NPSHR=NPSH1\%\) or \(NPSHR=NPSH5\%\)
- \(NPSHR=NPSHi\) (cavitation free operation)
Part 1 – What is cavitation (cont.)

Cavitation Phenomena

Q = Constant
N = Constant

Cavitation Inception

Cavitation Break-Off

Three-Percent Head Drop

Total Pump Head

NPSH
Cavitation Visualization Test Pump
0% Head Drop

NPSH (m)

Head (m)

Begin visual cavitation

1% head drop

3% head drop

0% head drop

NPSH (m)

0 10 20 30 40 50 60 70 80 90 100

0 1 2 3 4 5 6 7 8 9 0 1 0 0

4.05

4.00

3.95

3.90

3.85

3.80

3.75

3.70

3.80

3.90

4.00

4.05

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1% Head drop

3% head drop

1% head drop

0% head drop

Begin visual cavitation

**Graph:**

- **Y-axis:** Head (m)
- **X-axis:** NPSH (m)
- Key points:
  - 1% head drop
  - 3% head drop
  - 0% head drop
  - Begin visual cavitation

**Legend:**

- **FLOWSERVE Pump Division**
3% Head drop

Begin visual cavitation

Head (m)

NPSH (m)
Part 1 – What is cavitation (cont.)

- $\text{NPSH}_a$
- $\text{NSPH}_{3\%}$
- $Q_{SR}$
- $Q_{BGP}$
- Cavitation Free

Graph showing the relationship between NPSH (m) and Capacity $Q/Q_{design}$ (%).
Part 1 – What is cavitation (cont.)

Typically (in practice):

- $NPSHA > NPSH3\%$
- $NPSHi > NPSHA$ (especially for low capacity)

→ Pumps run okay, **BUT** with some developed cavitation.

**General misconception:**

$NPSHA > NPSHR \Rightarrow$ No Cavitation

(This will only hold if $NPSHR = NPSHi$.)
Part 2 – Cavitation prediction

- Scaling laws
- Thermodynamic effect
- Effect of dissolved or entrained gases
- Calculating incipient cavitation ($NPSHi$) from CFD
- Cavity length prediction
Part 2 – Cavitation prediction (cont.)

Predicting NPSH at speeds other than reference or test speed (scaling laws)

$NPSH_i: \quad NPSH_i \propto N^2 \Rightarrow NPSH_i = NPSH_{i,REF} \left( \frac{N}{N_{REF}} \right)^2$

$(\sigma_{TH} = \frac{NPSH}{H} = \text{constant})$

$NPSH_{3\%}: \quad NPSH_{3\%,i} = f \cdot NPSH_{3\%,REF} \left( \frac{N}{N_{REF}} \right)^2$

$N < N_{REF}, f \geq 1; \quad N > N_{REF}, f \leq 1$

“Postulate”: Amount of developed cavitation depends on residence time $\rightarrow f$ depends on size of the pump and ratio $N/N_{REF}$
Part 2 – Cavitation prediction (cont.)

Alternative approach to account for deviation from affinity law:

\[
NPSH_{3\%} = NPSH_{3\%,\text{REF}} \left( \frac{N}{N_{\text{REF}}} \right)^\alpha
\]

\[1 \leq \alpha \leq 2\]

Choice of \(\alpha\) is rather arbitrary and relies heavily on empiricism

Conservative choice:

\(N < N_{\text{REF}}, \ \alpha = 1\)

\(N > N_{\text{REF}}, \ \alpha = 2\)
Part 2 – Cavitation prediction (cont.)

Thermodynamic effect (temperature depression)

Cavitation performance depends on:
- Temperature of liquid
- Type of liquid

→ \(NPSHR\) reduction
  (E.g. Stepanoff method, or Hydraulic Institute correction chart)

(from Brennen, 1994)
Part 2 – Cavitation prediction (cont.)

Predicting thermodynamic effect

\[ NPSH_{3\%} = NPSH_{3\%,REF} - \Delta NPSH \]

Equilibrium theory:

\[ \Delta NPSH = B \frac{\rho_L}{\rho_V} \frac{h_{fg}^2}{v_{fg} g C_p T} \quad ; \quad B = \frac{V_V}{V_L} \]

Stepanoff (1965, 1978):

\[ B = B_1 \Delta NPSH \]

\[ B_1 = \left( \frac{\rho_L}{\rho_V} \right)^2 \frac{g C_p T}{h_{fg}^2} \quad ; \quad [m^{-1}] \text{ or } [ft^{-1}] \]

\[ \Delta NPSH = \frac{29}{H_V} B_1^{-\frac{2}{3}} \quad ; \quad [m^{-1}] \text{ or } \Delta NPSH = \frac{64}{H_V} B_1^{-\frac{2}{3}} \quad ; \quad [ft^{-1}] \]

Non-equilibrium theory → bubble dynamic (CFD) calculations, involving time-dependent two-phase flow calculations
Part 2 – Cavitation prediction (cont.)

Influence of dissolved and/or entrained gases:
→ “conceptual effective or artificial” vapor pressure:

\[
P_E = P_V + \Delta \\
P_E = yP_0
\]

(Chen, 1993)

Key characteristic:
Performance (breakdown) comes from gas evolution and gas expansion, rather than classical vapor formation.

Dissolved and/or entrained gases result in reduction of (effective) field \( NPSHA \):

\[
NPSHA^* = \frac{(P_{01} - P_E)}{\rho g}
\]

“Hidden danger”: \( NPSHA > NPSHR \) but \( NPSHA^* < NPSHR \)
Part 2 – Cavitation prediction (cont.)

Predicting incipient cavitation \( (NPSHi) \) from CFD

Typical approach:

1. Create 3D geometry model/grid of impeller passage
2. Solve flow field with CFD code (non-cavitating)
3. Calculate incipient \( NPSH \) from CFD pressure field (next slide)
Part 2 – Cavitation prediction (cont.)

Streamline through point of minimum pressure

\[
NPSH_i = \frac{p_{01,i} - p_V}{\rho g}
\]

\[
p_{01,i} = p_{1,i} + \frac{1}{2} \rho U^2
\]

\[
p_{1,i} = p_1 - (p_{\text{min}} - p_V)
\]

\[
NPSH_i = \frac{p_{01} - p_{\text{min}}}{\rho g}
\]

So: \(NPSH_i\) follows from \(p_{\text{min}}\) and \(p_{01}\) of calculated pressure field, and does not require \(p_V\) to be known!
Running simulations for several flow rates produces \( NPSH_i \) curve:

![Graph showing incipient NPSH vs. flow rate (Q/Qrated (%)).(From Visser, 2001)]
Part 2 – Cavitation prediction (cont.)

Note: CFD calculated characteristic is for impeller flow!

To project it on pump throughput one needs to account for volumetric efficiency (→ eye wear ring leakage flow):

\[ Q_{\text{impeller}} = Q_{\text{pump}} + Q_{\text{leakage}} \iff Q_{\text{pump}} = Q_{\text{impeller}} - Q_{\text{leakage}} \]

→ Computed curve shifts left by amount \( \Delta Q = Q_{\text{leakage}} \)

\[ Q_{\text{leakage}} = f(\Delta p, D, L, \delta, \nu, \rho) \sim \frac{1}{2} \pi D \delta \bar{u} \; ; \; \bar{u} = \sqrt{\frac{\Delta p \delta}{\lambda L \rho}} \]

\[ \lambda_{\text{laminar}} = 24 / \text{Re} \; ; \; \lambda_{\text{turbulent}} = 0.2373 / \text{Re}^{0.25} \; ; \; \text{Re} = \frac{\bar{u} \delta}{2 \nu} \]

It becomes particularly important to take \( Q_{\text{leakage}} \) into account for low \( N_S \) (specific speed) impellers. For high \( N_S \) the relative influence is less.
Part 2 – Cavitation prediction (cont.)

What if \( NPSHA < NPSHi \) ?

\[ \rightarrow \text{Find region on impeller blade surface where } p < p_V \]

- physically unrealistic, but it gives
- first “indication” of cavitation area, and
- first approximation of cavity bubble length

**Note:** The actual cavity will be bigger

\[ \rightarrow \text{bubble length will be underestimated} \]
Part 2 – Cavitation prediction (cont.)

To visualize $p < p_V$ region from non-cavitating flow simulation:

→ Plot isotimic surface for threshold value $p_V^*$

\[
p_V^* = p_V + (p_1 - p_{1,A})
= p_1 + \frac{1}{2} \rho U^2 - (p_{1,A} + \frac{1}{2} \rho U^2 - p_V)
= p_{01} - \rho g NPSHA
= p_{01} - NPSPA
\]
Example:

Plot of $p < p_V$ region

$NPSHA = 15.5 \text{ m (51 ft)}$
$NPSHi = 28 \text{ m (92 ft)}$
$N = 2980 \text{ RPM}$
$Q = 400 \text{ m}^3/\text{h}$
(1760 USGPM)

Cavitation on blade suction side
Part 2 – Cavitation prediction (cont.)

Putting $L_{CAV} = mL(p<p_v)$, $m=\alpha(3)$, one can get some impression of expected cavitation erosion rate.


$$\dot{E} = C \left( \frac{L_{CAV}}{L_{CAV,10}} \right)^n \left( \tau_A - \phi^2 \right)^3 U_e^6 \rho^3 A \left( 8T_s^2 \right)$$

or

$$\dot{E} \propto L_{CAV}^n \Leftrightarrow \dot{E}_2 = \hat{E}_1 \left( \frac{L_2}{L_1} \right)^n$$  (*)

with  

$n = 2.83$ for blade suction side and  

$n = 2.6$ for blade pressure side.

Equation (*) is especially powerful when comparing designs and evaluate susceptibility to cavitation erosion (in a relative sense).

→ Design optimization studies
Part 2 – Cavitation prediction (cont.)

• Results and theory thus far do not require two-phase flow calculations.

• Still it provides important information of an impeller design regarding cavitation performance.

• Next level of improvement has to come from CFD calculations with cavitation model.

• Calculations with a cavitation model are time consuming and tend to be “CPU-expensive”

• Several cavitation models exist to date, and development of cavitation models is still ongoing
Part 2 – Cavitation prediction (cont.)

CFD Cavitation models

Typically two approaches:

• Equilibrium models
  – Barotropic or pseudo density models; $\rho = \rho(p)$
  – Somewhat “simplistic”, yet
  – Attractive since they can be used in single phase codes

• Bubble dynamic models
  – Rayleigh-Plesset equation
  – Vapor-liquid interaction (time-dependent mass & heat transfer)
  – Closer to reality
  – More complicated and more “CPU-expensive”
  – E.g. Volume of Fluid (VOF) model
Example:

Plot of cavity bubble

Equilibrium model
CFX-TASCflow
(CEV-model)

\(NPSHA = 15.5\) m (51 ft)
\(NPSHi = 28\) m (92 ft)
\(N = 2980\) RPM
\(Q = 400\) m³/h
(1760 USGPM)
\(m \approx 3\)

Cavitation on blade suction side
Application:
With CFD cavitation models one can predict $NPSH_{3\%}$ from CFD calculated head drop curves

(from Visser, 2001; CEV-model prediction)
Concluding Remarks

• Cavitation is a phenomenon which can seriously impact performance and operation of pumps.

• Predicting cavitation performance is an important topic, not only for pumps, but for fluid machinery in general.

• Traditional (scaling) methods are still important and useful.

• CFD methods provide further insight and are becoming more and more common.

• Bubble dynamic (CFD) methods are emerging and hold a promise for the future.
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