LARGE COOLING WATER PUMP UPGRADE
FOR INCREASED CAPACITY AND
REDUCED IMPELLER CAVITATION

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Short Abstract

This case study discusses the hydraulic upgrade of two sets of four (25 percent) large vertical cooling water pumps. To increase the electrical output of subject power station, the cooling water pumps of block 1 and 2 had to be upgraded in capacity minimally thirteen percent, thereby also attempting to minimize the occurrence of impeller cavitation.

The upgrade consisted of replacing the existing impeller and diffuser plus casing, and installing new electric motors. To corroborate anticipated hydraulic performance the replacement impeller and diffuser were model tested on a 1:4 scale. The model test further served to determine final sizing dimension of the upgrade.
Contents

• Background
• In-situ capacity measurement
• Impeller-diffuser CFD study
• Scaled model testing
• Start-up transient
• As built performance
Background

• Up-rate of Nuclear Power Station (unit 1 & 2)
• Increase of electrical output power required an increase of cooling water capacity (> 13%)
• Existing cooling water pumps (CWPs) are suffering from cavitation attack
• CWP retrofit design objective:
  ➢ Cooling water capacity increase of 13%+
  ➢ Minimize impeller cavitation
• CWP E-motor replacement ($P_{CWP \uparrow}$)
  ➢ System start-up transient analysis
Background

Per unit four (25 percent) CWP’s running in parallel feeding the condenser with seawater
In-Situ Cooling Water Capacity Measurement

- Required to establish pre-upgrade baseline situation
  - Total CWPs flow rate measurement
  - Individual CWP’s head measurement
- Flow rate measurement with OTT-mills
  - 6 mills on a horizontal scanning bar
  - 4 throughflow areas scanned (curtain wall)
  - 6x14 scanning window (14 elevations)
In-Situ Cooling Water Capacity Measurement
In-Situ Cooling Water Capacity Measurement

<table>
<thead>
<tr>
<th></th>
<th>Existing</th>
<th>Up-rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>325 r/min</td>
<td>325 r/min</td>
</tr>
<tr>
<td>Capacity</td>
<td>7.8 m³/s 275 cfs</td>
<td>&gt; 8.81 m³/s &gt; 311 cfs</td>
</tr>
<tr>
<td>Head</td>
<td>5.7 m 18.7 ft</td>
<td>&gt; 7.3* m &gt; 24.0 ft</td>
</tr>
<tr>
<td>(N_{s,D})</td>
<td>12700 USCU (-)</td>
<td>11200 USCU (-)</td>
</tr>
<tr>
<td>(D_{nom})</td>
<td>56”</td>
<td>58” – 59”</td>
</tr>
</tbody>
</table>

* Per quadratic scaling of benchmark system head: \(> 1.13^2 \times 5.7 \text{ m}\)
• **Geometries studied**
  - Existing impeller / diffuser combination*
  - Retrofit impeller / diffuser combination*

• **Objective**
  - Determination of best cavitation point (BCP) → NPSHi
  - Evaluate cavitation development
  - Head comparison

* Both the existing and retrofit design have 4 impeller blades and 7 diffuser blades
Methodology

- **Head calculation**
  
  \[ H = \frac{p_{total, outlet} - p_{total, inlet}}{\rho g} \]

- **Incipient cavitation NPSH**
  
  \[ NPSH_i = \frac{p_{total, inlet} - p_{min}}{\rho g} \]

Mass flow averaged value:

\[ \sum m_j p_{total,j} \]

\[ \sum m_j \]
Impeller-Diffuser CFD Study

Existing Design

Full 360° model

Impeller
- 4 blades

Diffuser
- 7 blades
Impeller-Diffuser CFD Study

New Design

Full 360° model

Impeller
- 4 blades

Diffuser
- 7 blades
Impeller-Diffuser CFD Study

Observations:

1) Improved incipient cavitation NPSH ➔ less cavitation

2) Final size less than 59” (1500 mm)
Impeller-Diffuser CFD Study

New impeller; $p < p_V$ @ design duty
Scaled Model Testing

- **Objective**
  - Verify hydraulic performance
- **Scaled model testing**
  - 1:4 model scale
  - 4:1 speed ratio (325 ⇔ 1300 r/min)
  - Existing impeller-diffuser (350 mm)
  - Up-rate impeller-diffuser (360 mm)
  - Existing parts replicated on scale from 3D scan
  - Up-rate parts modeled directly in 3D CAD
Scaled Model Testing

Test Loop & Test Set-up

- Q, H, \( \eta \) performance testing
- Cavitation visualization

Flow visualization window with impeller mounted
Scaled Model Testing

Scaling Performance

\[ Q = Q_M f^3 \left( \frac{N}{N_M} \right); \quad H = H_M f^2 \left( \frac{N}{N_M} \right)^2 \]

\[ \eta = \eta_M + \Delta \eta; \quad f = \frac{D}{D_M} \quad \text{(scale or model factor)} \]

\[ \Delta \eta = 0.6 (1 - \eta_M) \left[ 1 - \left( \frac{1}{f^2} \frac{N_M}{N} \right)^{0.2} \right] \quad \text{(IEC-497)} \]
Scaled Model Testing

Full Size Performance from Scaled Model Test of Impeller & Diffuser

- Upgrade @ 59" & 330 r/min
- Existing @ 330 r/min
- Existing Duty

Capacity, [m³/h]

Head, [m]

15000 20000 25000 30000 35000 40000 45000

0 4 8 12 16
Scaled Model Testing

Cavitation – Visual inception (1 mm cavitation bubble)

NPSH [m]

NPSHA
Scaled Model Testing

Cavitation at duty capacity & rated NPSHA

- Minor development of cavitation at blade leading edge
Scaled Model Testing

Cavitation – 10 mm cavity bubble

NPSH [m]

NPSHA

% Duty Capacity

60% 70% 80% 90% 100% 110% 120%
Scaled Model Testing

Cavitation @ 90% duty capacity

CFD Experiment
Start-Up Transient

- **Objective**
  - Check motor capability (torque) to start the pumps
  - Determine start-up time(s)
- **Entire cooling water system is modeled**
Start-Up Transient

• Start-up requirement E-motor
  ➢ 80% Voltage
  ➢ -15% Torque (tolerance per IEC 60034-1)
• Start-up scenario
  ➢ P1 thru P4 are started at 60 sec intervals
• Initially selected motor showed problem when starting 4th pump
  ➢ P4 could not be accelerated to full speed due to insufficient motor torque
  ➢ P4 ended up running against closed (check) valve at intermediate speed
Start-Up Transient

Motor speed-torque curve

Pump curve

Rated Condition
Start-Up Transient

Speed Torque Pump 1 (-15% Tol)

Speed Torque Pump 2 (-15% Tol)

Speed Torque Pump 3 (-15% Tol)

Speed Torque Pump 4 (-15% Tol)
Start-Up Transient

- Required fluid torque P4 is larger than motor torque

- Motor torque
  - $U_n$ 100%; $T$ 100%
  - $U_n$ 80%; $T$ -15%

- Fluid torque @ zero flow
Every next pump needs to develop more head at zero flow to come online.
Start-Up Transient

- Issue resolved by changing the design of the new motors to give better speed-torque characteristic.
- Final simulation shows that the condenser is at full cooling water flow after 200 sec. (with 60 sec start-up intervals.)

Cooling water flow through condenser at start-up
As Built Performance

- Final design built @ 58\(\frac{1}{16}\)” (1475 mm)
- Taking into account:
  - Performance pick-up due to up-scaling
  - Intake & discharge losses not being accounted for in impeller/diffuser CFD study and scaled model testing
  - System resistance line was lowered due to installing power pack (actuator) on check values \(\Rightarrow\) less steep characteristic
  - Contractually required capacity increase of 113%, with +3% tolerance.
As Built Performance

- In-situ field performance check
  - Four pumps running
  - Three pumps running
- Pumps are over-performing (Q-H)
  - Pump Q-H above predicted curve
  - System resistance curve lowered more than expected (power pack)
  - Higher condenser cooling capacity (+)
  - Higher driver power (Δ), but motors are not overloaded

At the end: Everybody Happy!
Thank you for your attention

Questions?