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

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Presenter/Author Biography

Frank Visser is Principal Engineer at Flowserve, Aftermarket Services & Solutions, in Etten-Leur, The Netherlands. He obtained a B.S. degree (Mechanical Engineering, 1985) from Technical College Alkmaar, The Netherlands, and a M.S. degree (Mechanical Engineering, 1991) and Ph.D. degree (Technical Sciences, 1996) from the University of Twente, The Netherlands. He has received the ASME 2017 Sankaraiyer Gopalakrishnan-Flowserve Pump Technology Award, is a member of the Royal Netherlands Society of Engineers (KIVI), a member of the Industrial Advisory Board for the J.M.Burgerscentrum (JMBC), National Research School for Fluid Mechanics in the Netherlands, a member of ASME, and former Associate Editor for ASME Journal of Fluids Engineering (two terms).

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

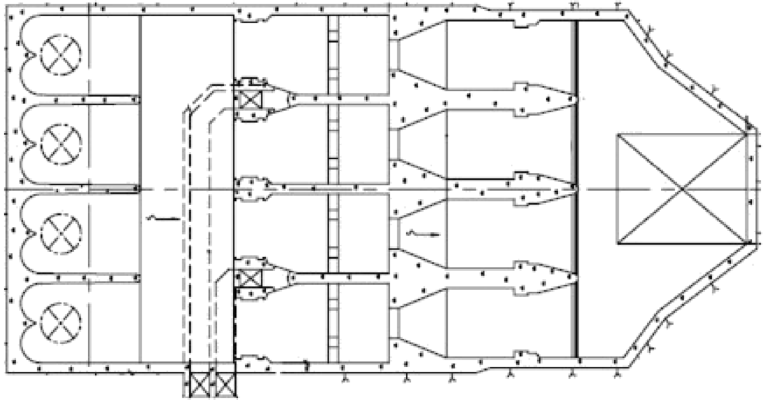
- Background
- In-situ capacity measurement
- Impeller-diffuser CFD study
- Scaled model testing
- Start-up transient
- As built performance

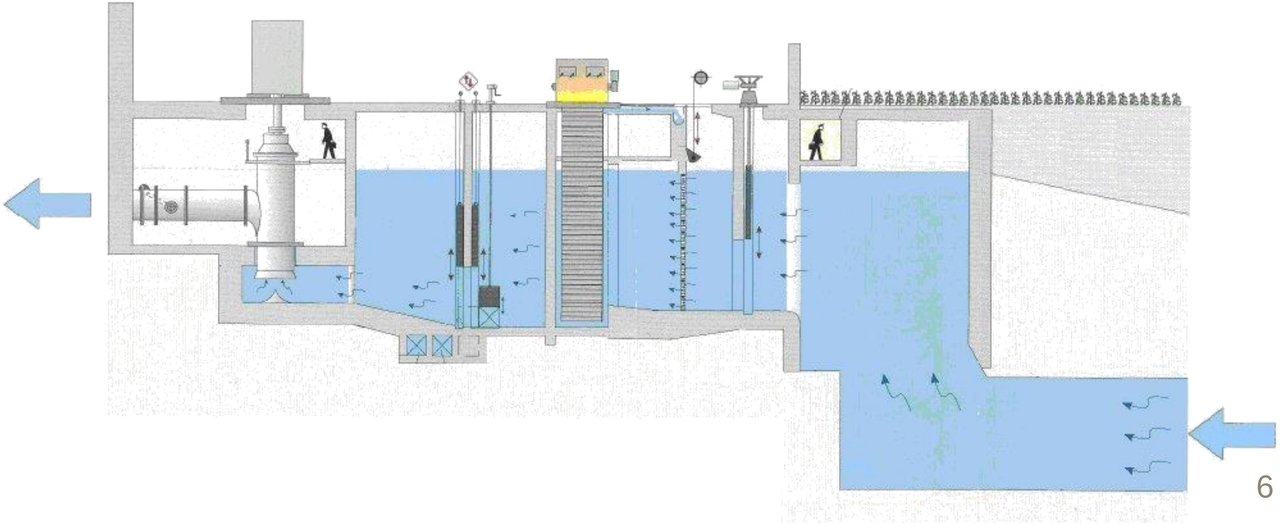
Background

- 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

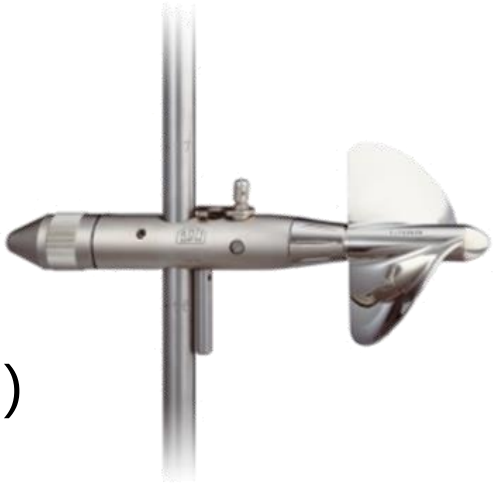
Per unit four (25 percent) CWP's running in parallel feeding the condenser with seawater





In-Situ Cooling Water Capacity Measurement

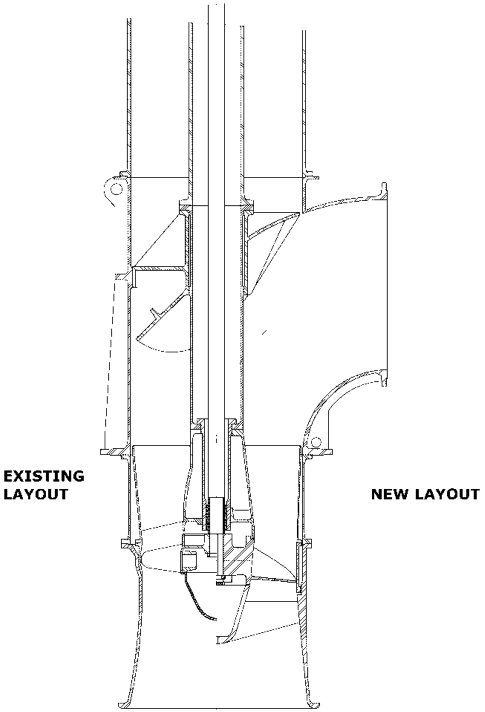
- Required to establish pre-upgrade baseline situation
  - ➢ Total CWPs flow rate measurement
  - ➢ Individual CWP’s head measurement
- Flow rate measurement with OTT-mills
  - ➢ 6 mills on a horizontal scanning bar
  - ➢ 4 throughflow areas scanned (curtain wall)
  - ➢ 6x14 scanning window (14 elevations)



OTT-mill

In-Situ Cooling Water Capacity Measurement

In-Situ Cooling Water Capacity Measurement



	Existing				Up-rate			
Speed	325		r/min		325		r/min	
Capacity	7.8	m^3/s	275	cfs	> 8.81	m^3/s	> 311	cfs
Head	5.7	m	18.7	ft	> 7.3*	m	> 24.0	ft
$N_{s,D}$	12700 4.64		USCU (-)		11200 4.1		USCU (-)	
D_{nom}	56”				58” – 59”			

* Per quadratic scaling of benchmark system head: $> 1.13^2 \times 5.7$ m

Impeller-Diffuser CFD Study

- **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**

Impeller-Diffuser CFD Study

Methodology

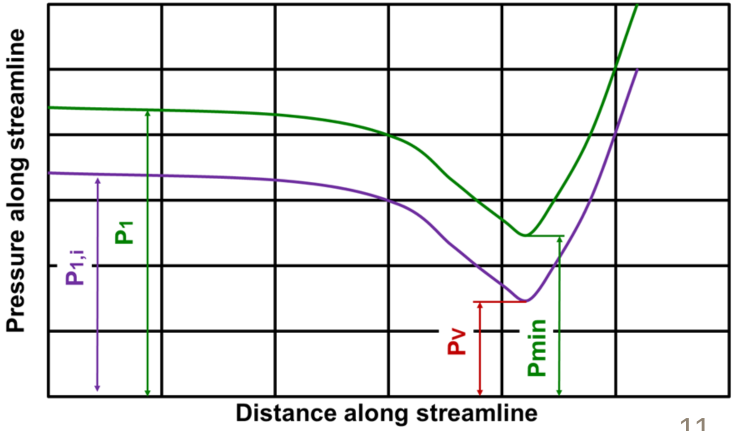
- **Head calculation**

$$H = \frac{p_{total,outlet} - p_{total,inlet}}{\rho g}$$

Mass flow averaged value: $\dfrac{\sum \dot{m}_j p_{total,j}}{\sum \dot{m}_j}$

- **Incipient cavitation NPSH**

$$NPSH_i = \frac{p_{total,inlet} - p_{min}}{\rho g}$$



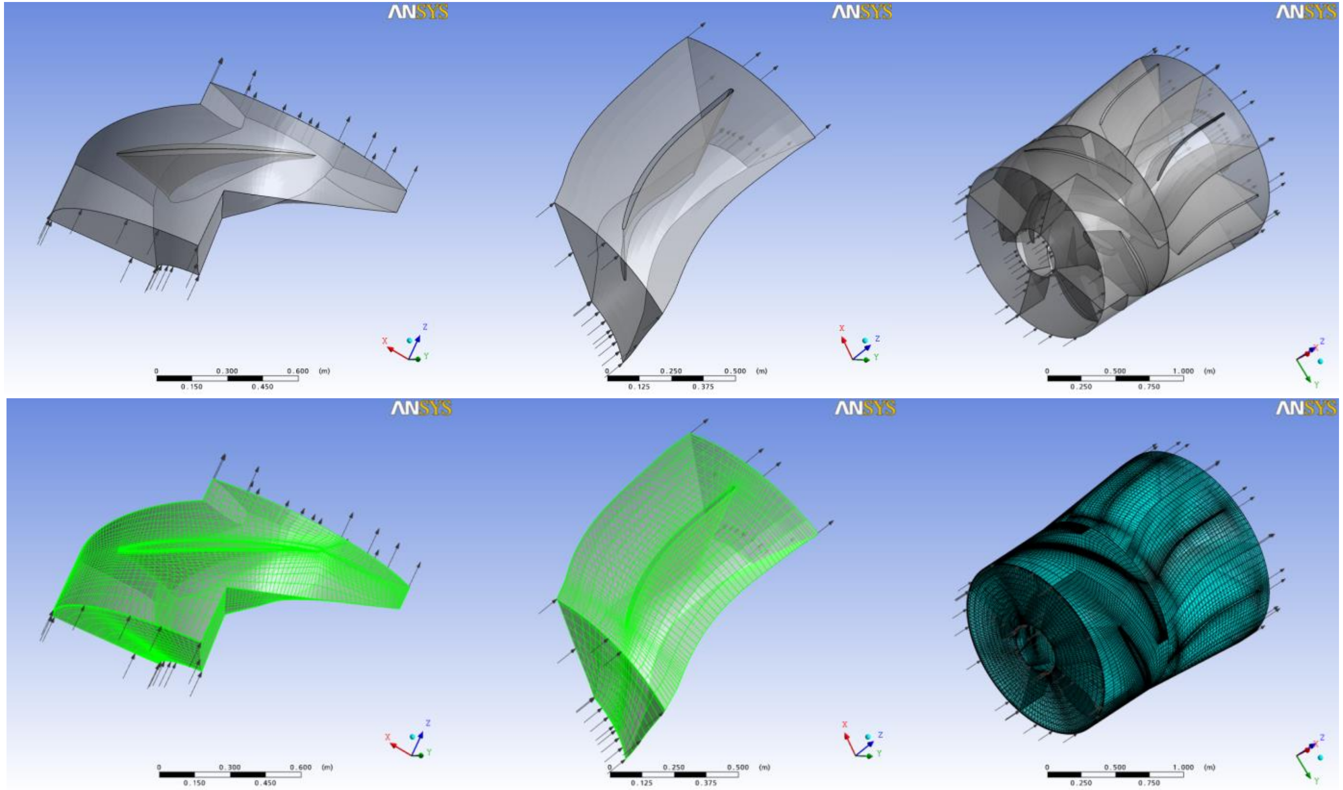
Impeller-Diffuser CFD Study

Existing Design

Full 360^0 model

Impeller
- **4 blades**

Diffuser
- **7 blades**



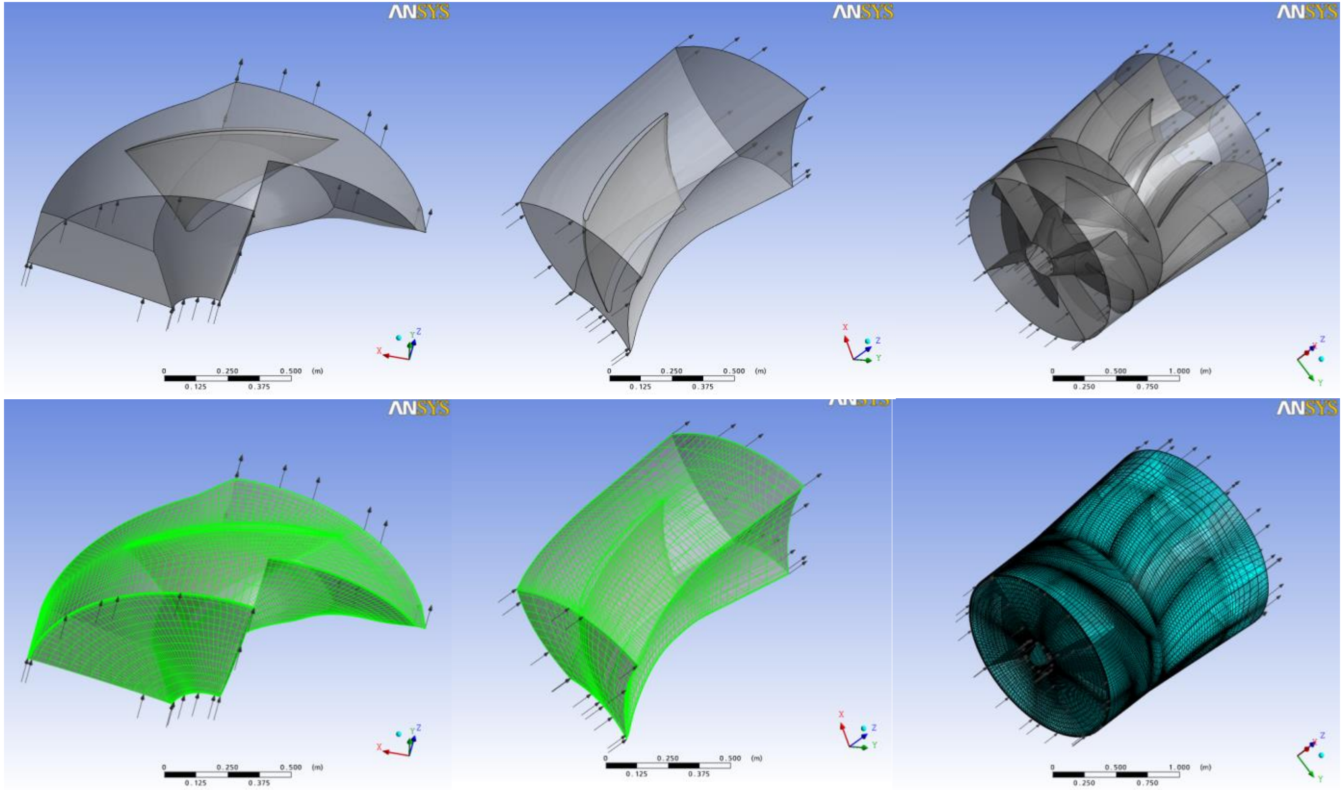
Impeller-Diffuser CFD Study

New Design

Full 360^0 model

Impeller
- **4 blades**

Diffuser
- **7 blades**

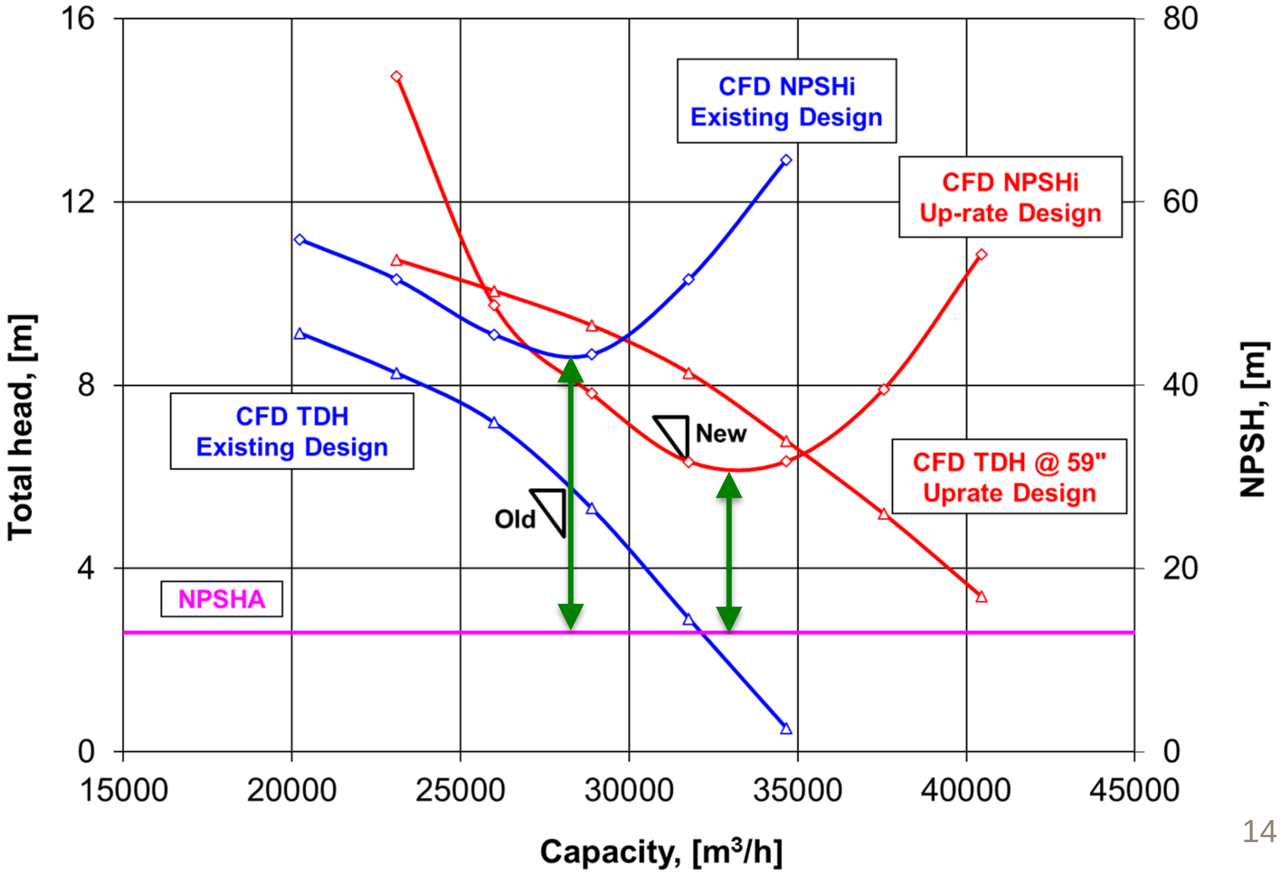


Impeller-Diffuser CFD Study

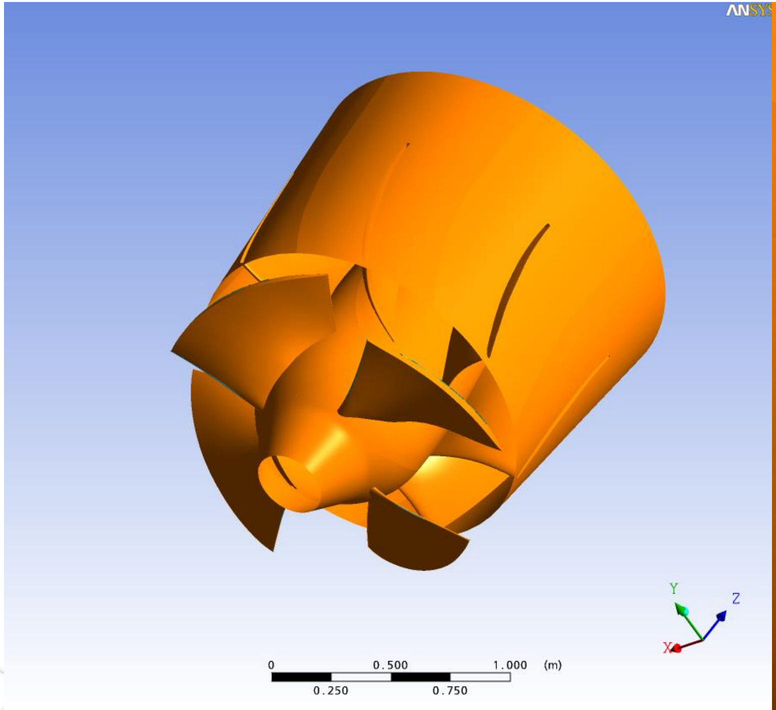
Observations:

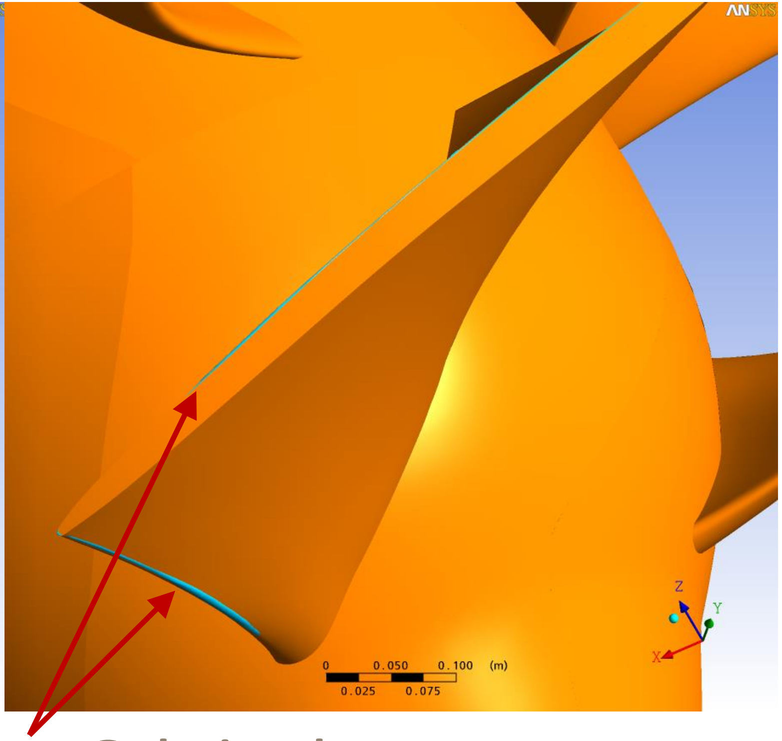
1) Improved incipient cavitation NPSH
➔ less cavitation

2) Final size less than 59” (1500 mm)



Impeller-Diffuser CFD Study

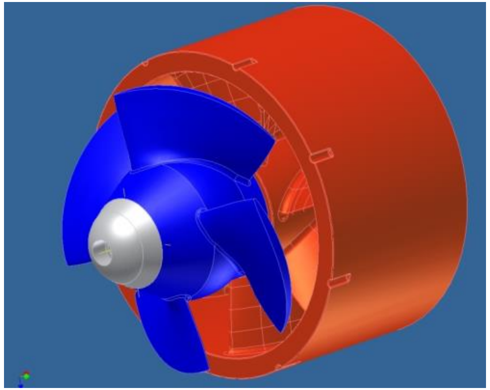


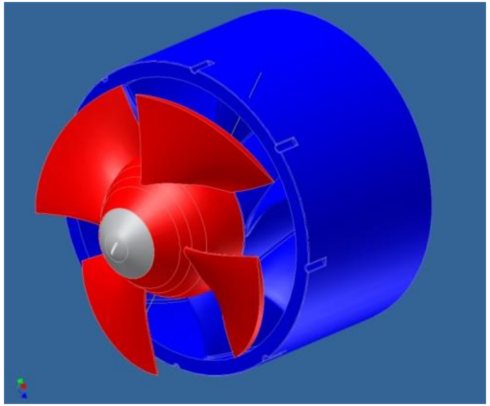


New impeller; $p < p_V$ @ design duty

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

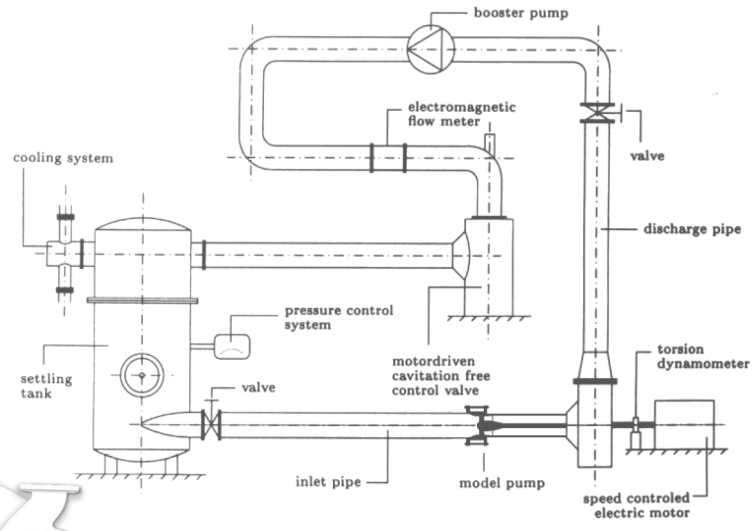


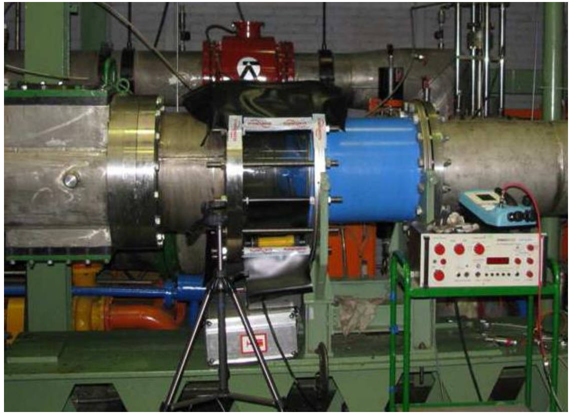


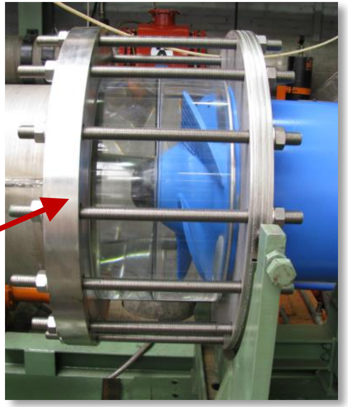
Scaled Model Testing

Test Loop & Test Set-up

- Q, H, η performance testing
- Cavitation visualization







Flow visualization window with impeller mounted

Scaled Model Testing

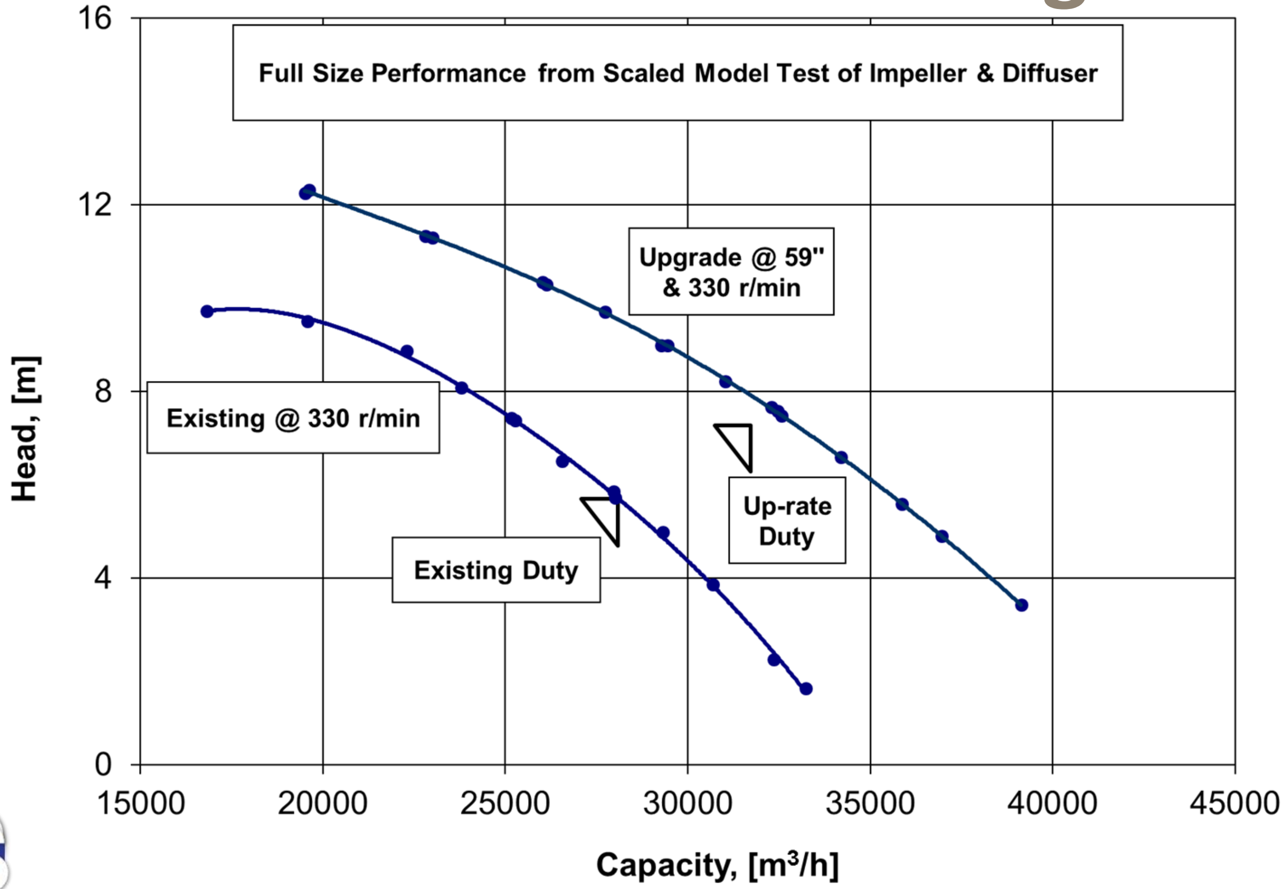
Scaling Performance

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

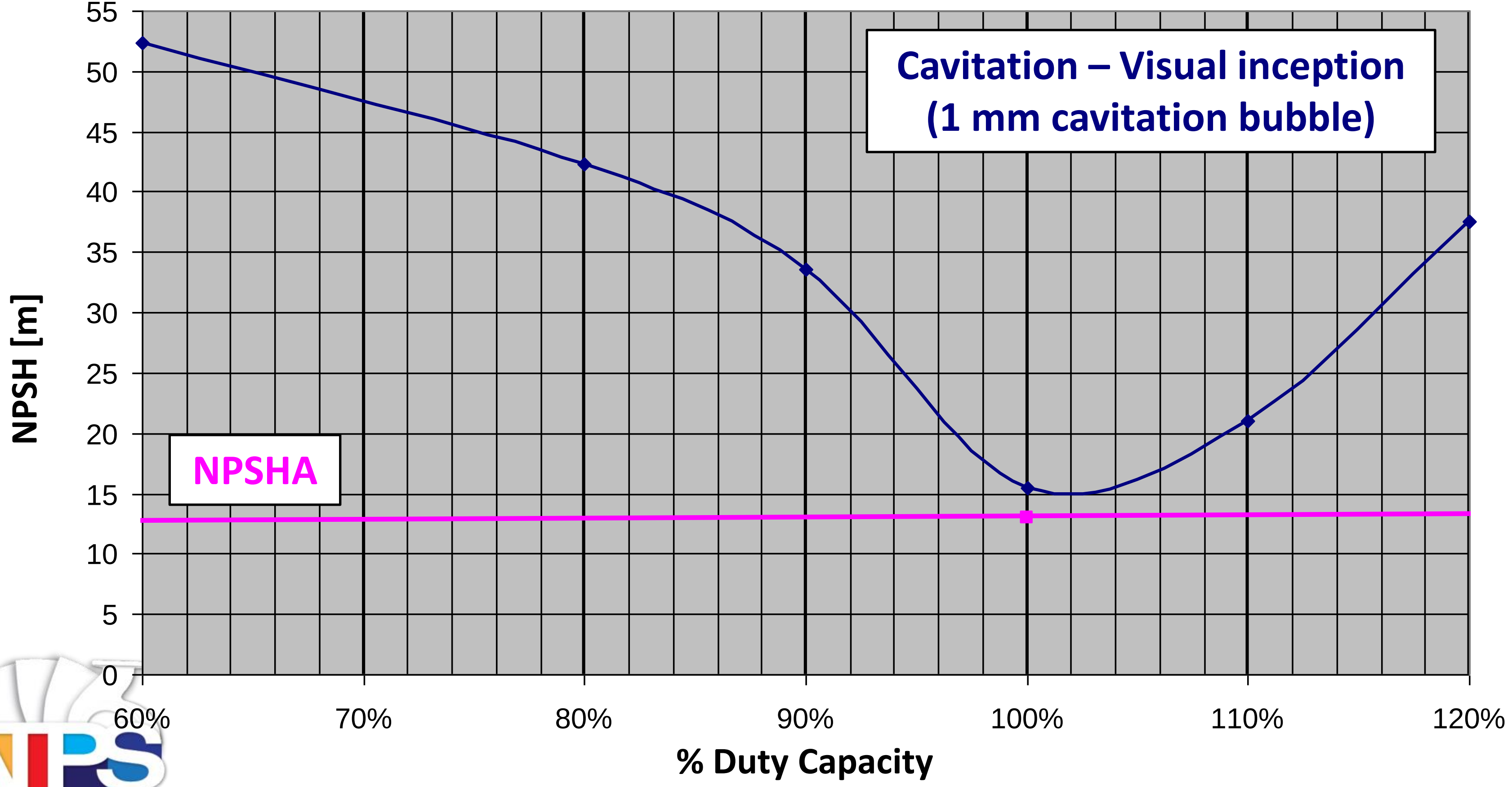
$$\eta = \eta_M + \Delta\eta; \qquad f = \frac{D}{D_M} \text{ (scale or model factor)}$$

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

Scaled Model Testing



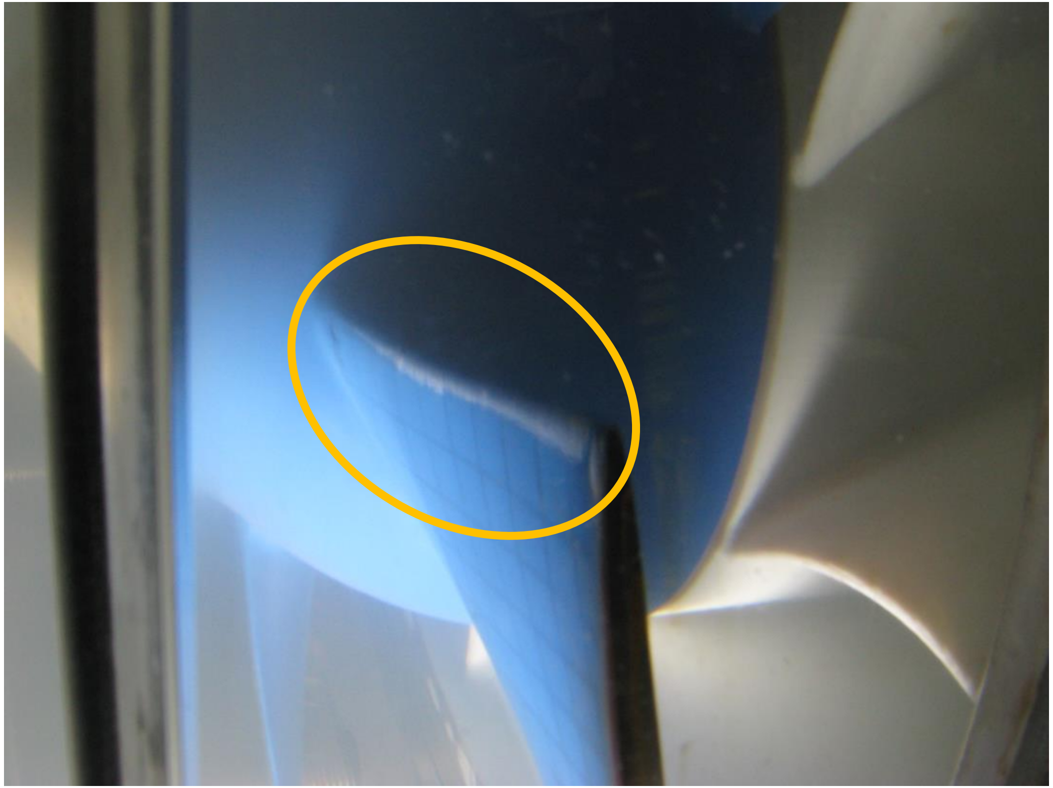
Scaled Model Testing



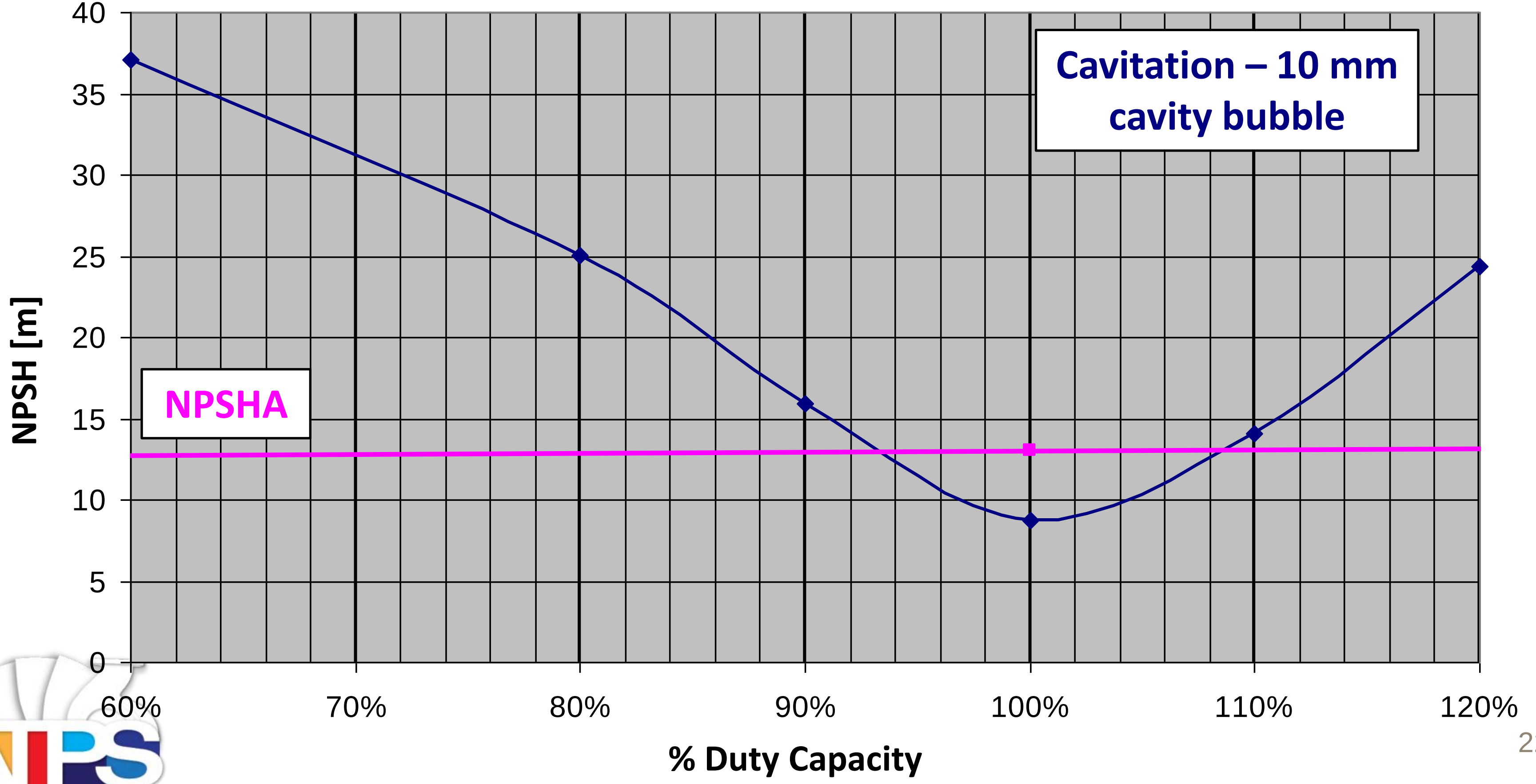
Scaled Model Testing

Cavitation at duty capacity & rated NPSHA

- Minor development of cavitation at blade leading edge

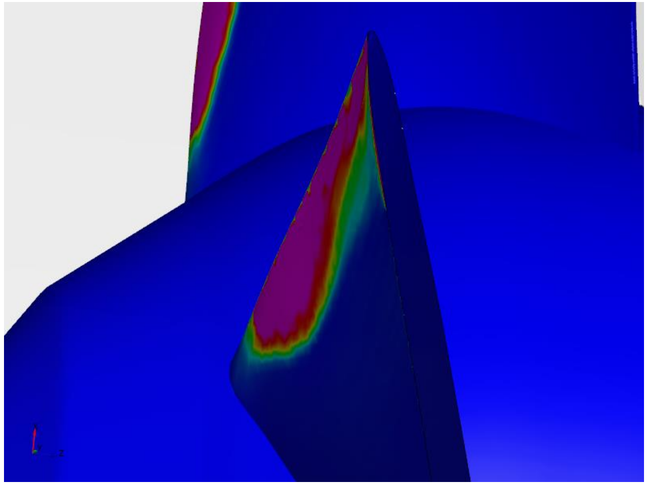


Scaled Model Testing

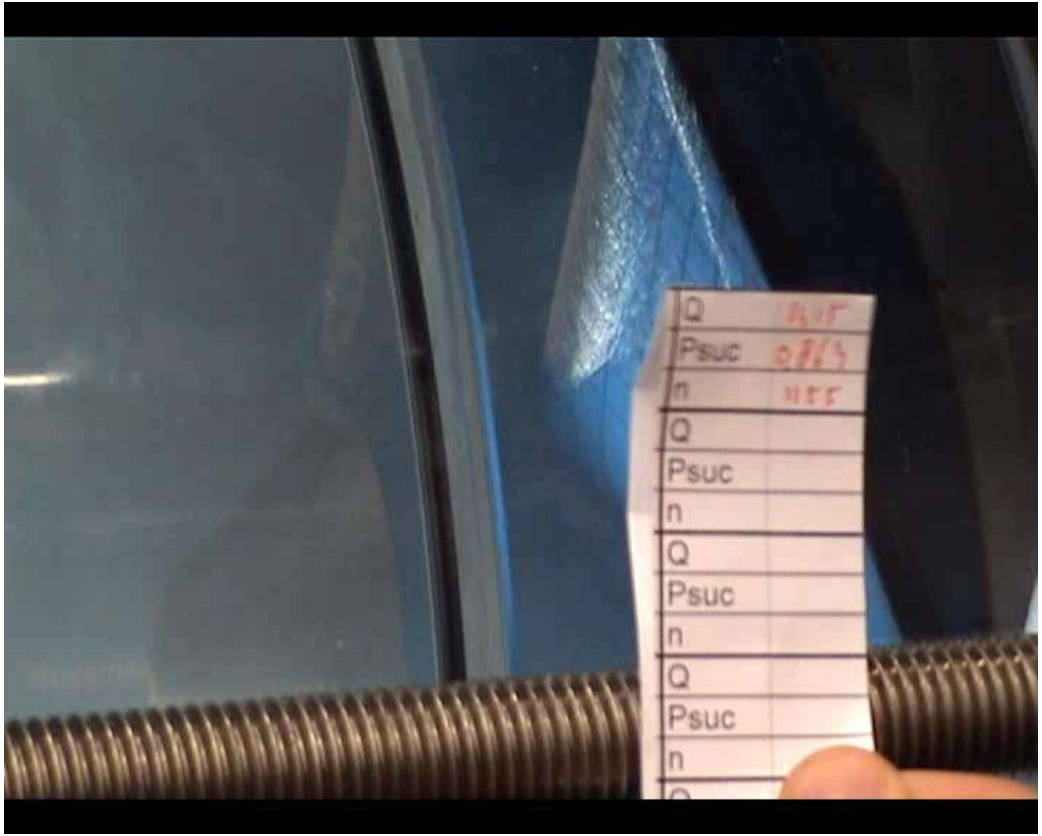


Scaled Model Testing

Cavitation
@ 90% duty capacity



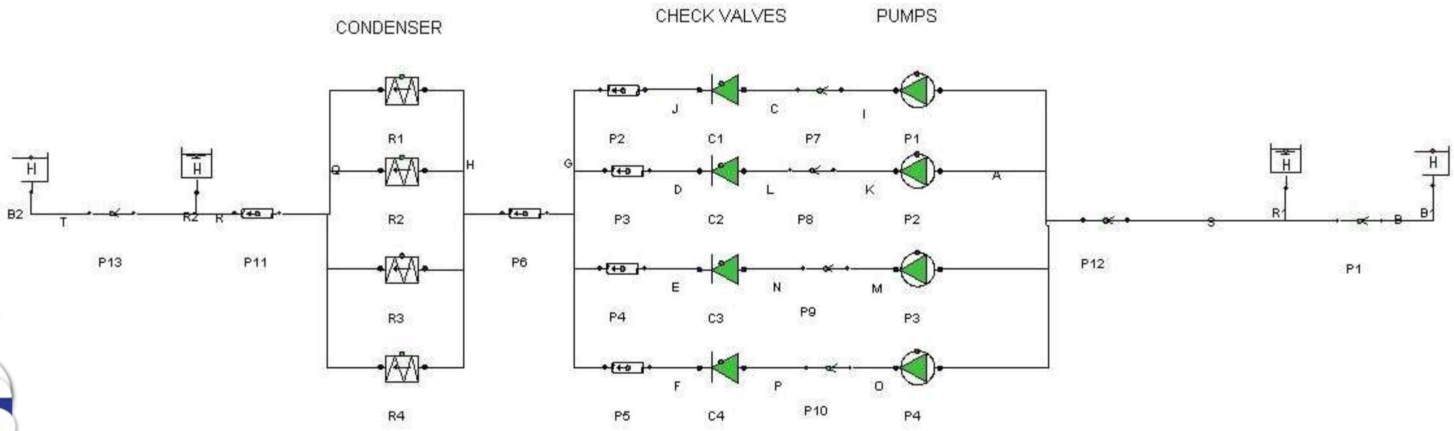
CFD



Experiment

Start-Up Transient

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

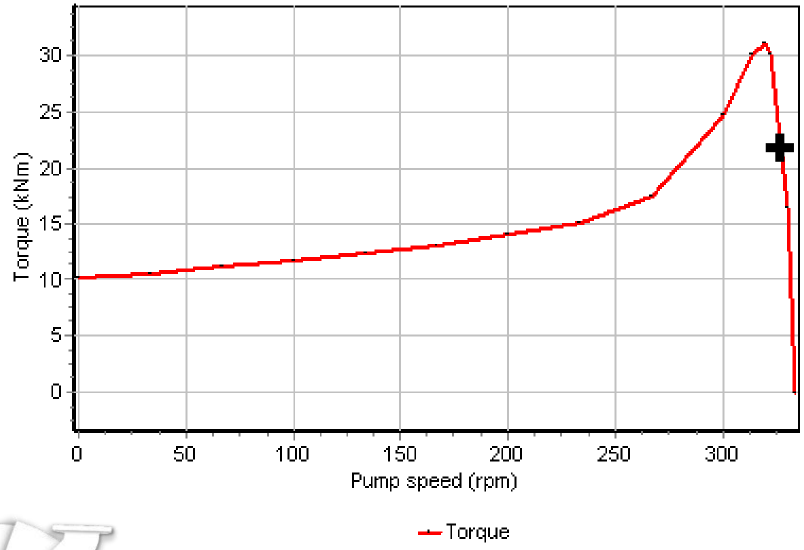


Start-Up Transient

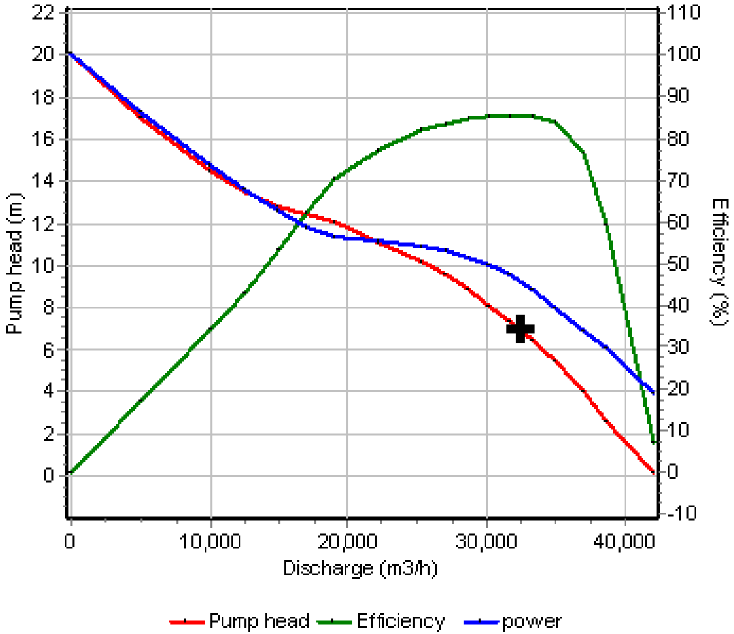
- Start-up requirement E-motor
  - 80% Voltage
  - -15% Torque (tolerance per IEC 60034-1)
- Start-up scenario
  - P1 thru P4 are started at 60 sec intervals
- Initially selected motor showed problem when starting 4th pump
  - P4 could not be accelerated to full speed due to insufficient motor torque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

Start-Up Transient

Motor speed-torque curve



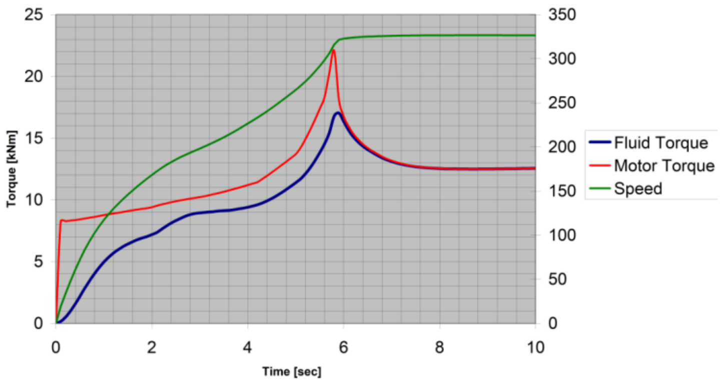
Pump curve



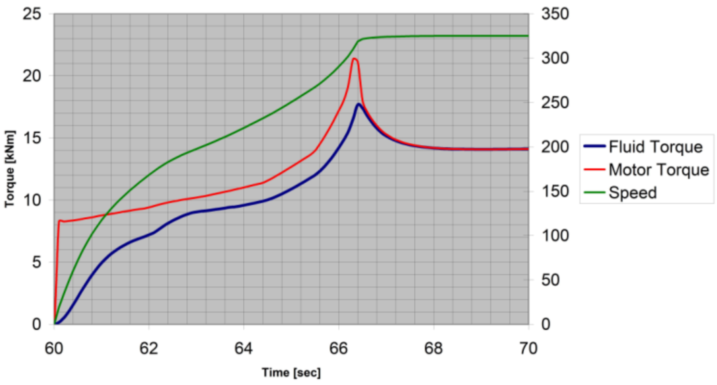
✚ **Rated Condition**

Start-Up Transient

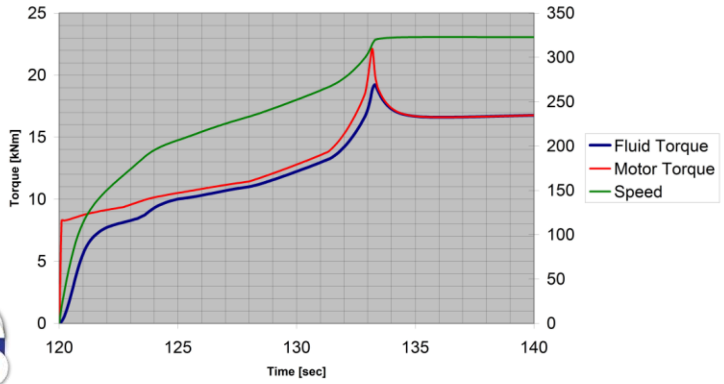
Speed Torque Pump 1 (-15% Tol)



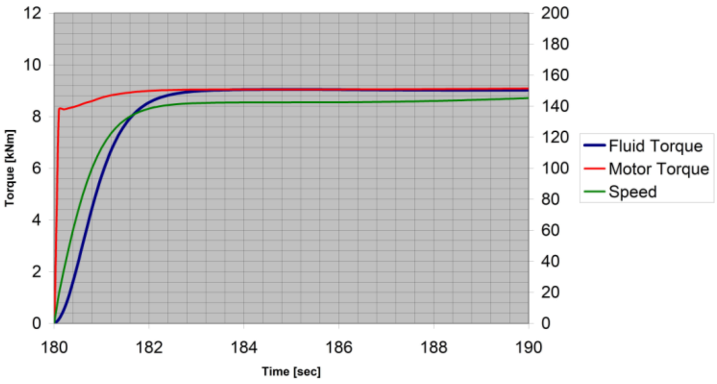
Speed Torque Pump 2 (-15% Tol)



Speed Torque Pump 3 (-15% Tol)

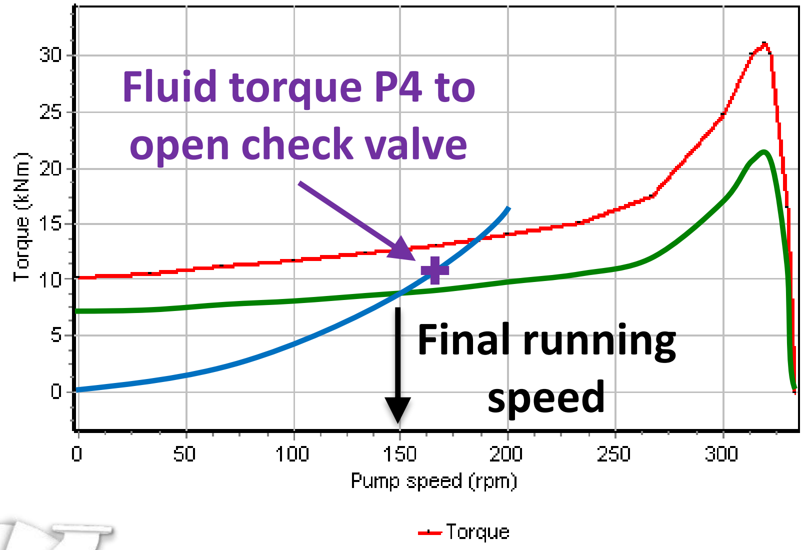


Speed Torque Pump 4 (-15% Tol)



Start-Up Transient

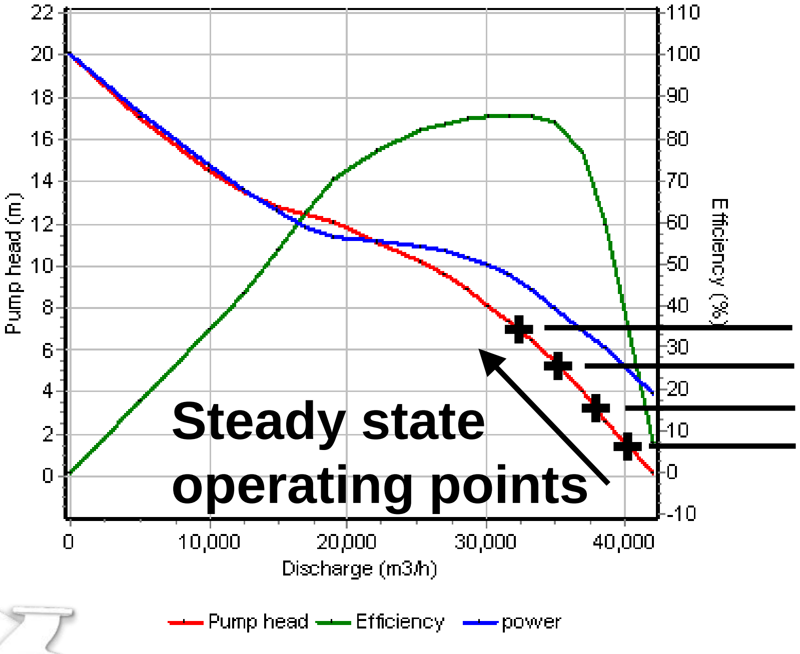
➔Required fluid torque P4 is larger than motor torque



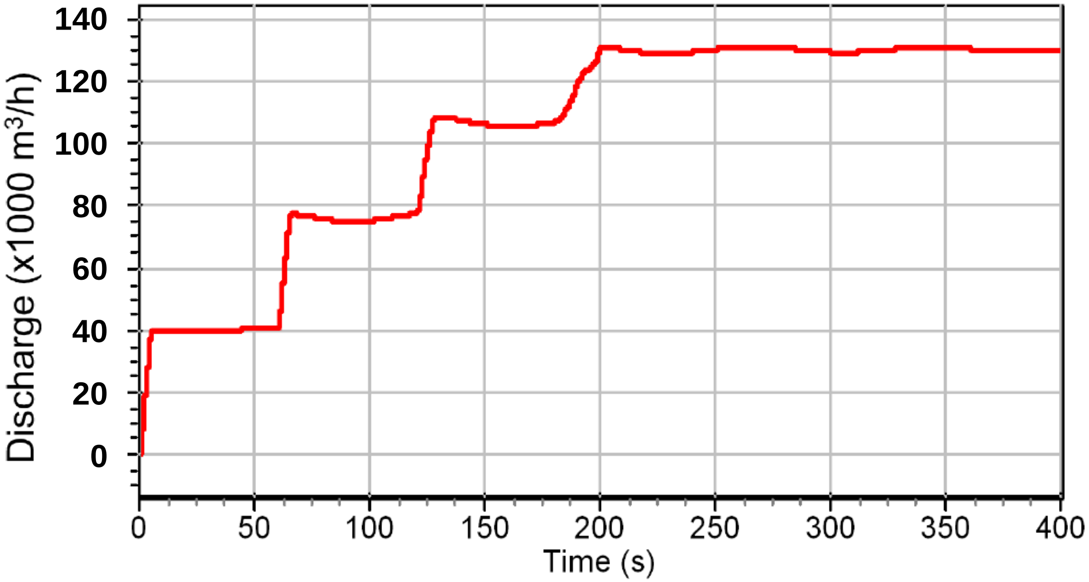
- **Motor torque**
  - ➢ **U_n 100%; T 100%**
- **Motor torque**
  - ➢ **U_n 80%; T -15%**
- **Fluid torque @ zero flow**

Start-Up Transient

- Every next pump needs to develop more head at zero flow to come online



Start-Up Transient

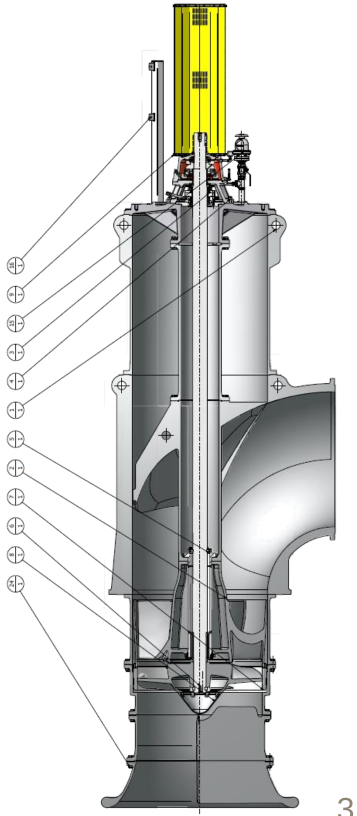


Cooling water flow through condenser at start-up

- 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)

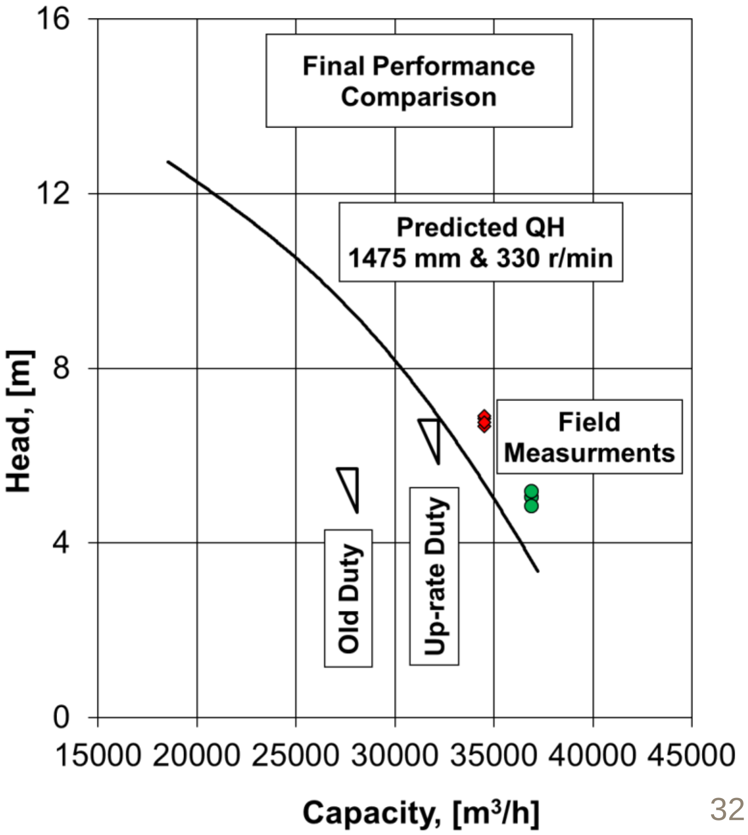
As Built Performance

- Final design built @ 58 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 ➔ 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?