CFD Simulation and Scale Model Testing of a New Dredge Pump Design

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Scope of project

New dredge pump design:

- *for cutter suction and hopper dredge service.*
- *1100 mm discharge, 1250 mm suction, and 2540 mm impeller.*
- *534 mm sphere clearance.*

CFD simulations.

- *High fidelity, transient, full machine model, including impeller side gaps.*
- *Allows for accurate prediction of performance and internally generated hydraulic loads.*

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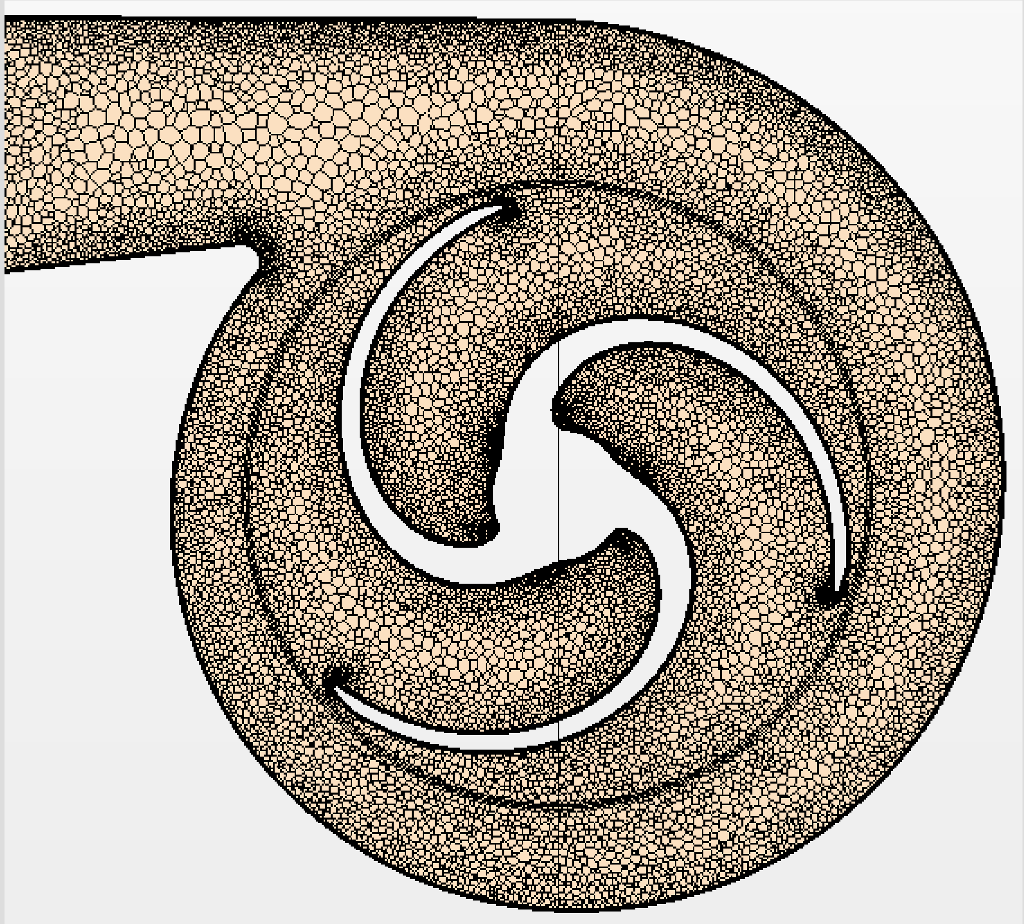
Scope of project

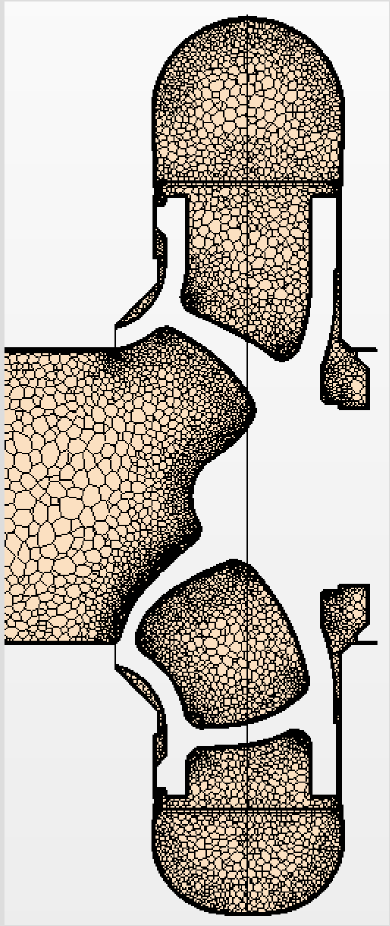
Laboratory model testing

- *0.25 and 0.57 scale models tests.*
- *Rapid prototype impeller castings.*
- *NPSHR scaling validation.*

CFD methodology to improve NPSHR performance.

- *Achieved without sacrificing efficiency or head.*
- *Validated by a third, model pump test.*





Computational Mesh

Polyhedral mesh elements.

Prism layers at the wall to control non-dimensional wall distance (y+).

- *average thickness of wall prism layers = 10 µm.*

More than 24 million cells.

K-omega SST turbulence model.

Flow Rate (m^3/s)	7.365	6.798	6.232	5.665	5.099	4.532	2.833
%BEPQ	116	107	98	89	80	71	45
Inlet Velocity (m/s)	5.94	5.48	5.02	4.57	4.11	3.65	2.28
Number of rotations required for convergence	16.5	20.6	12.6	13.1	16.7	19.6	61.6
Average solver elapsed time per time step (sec)	51.3	49.2	49.4	50.9	50.8	50.5	50
Total solution run time (hours)	84.6	101.3	62.2	66.7	84.8	99.1	308.2

Table 1. Summary of flow rates, required number of rotations, and solver elapsed time.

Summary of analyses

Simulations were performed at 230 rpm.

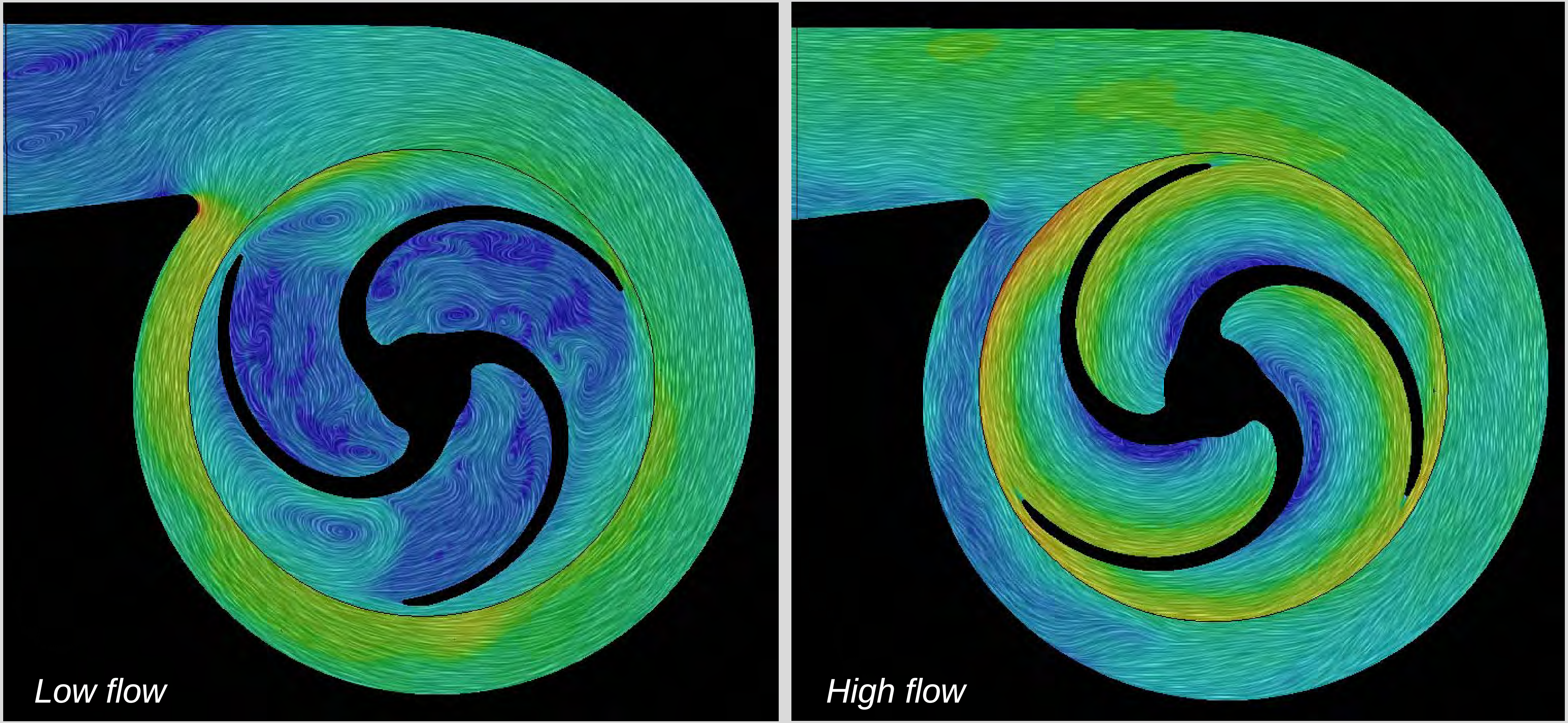
Flow range from 45% to 130% of the *BEPQ*.

Transient solution time-step:

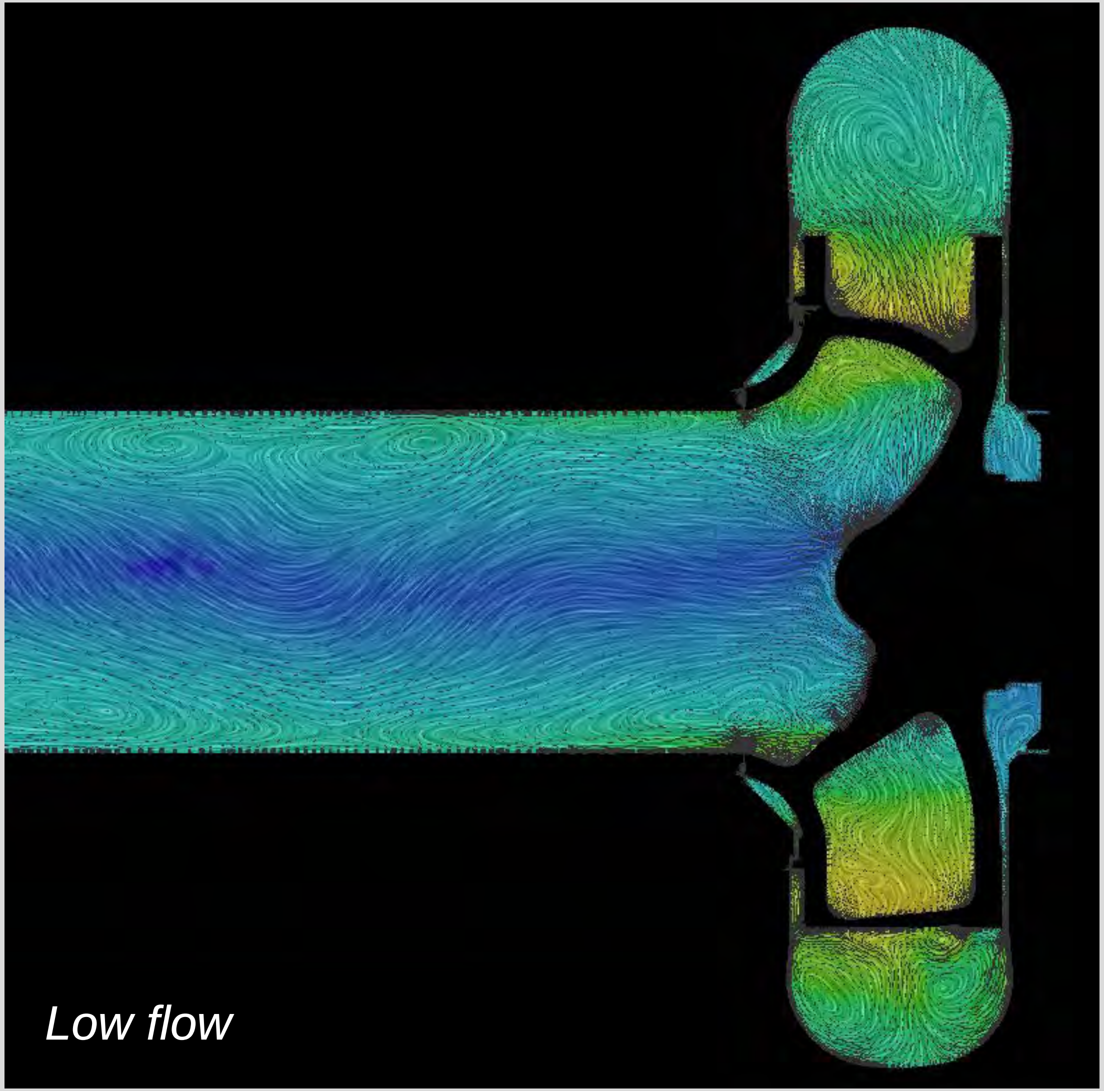
- *Initial: 0.05º (3.6 E-5 s)*
- *Final: 1º (0.72 E-3 s)*

Impeller rotations as needed to achieve a stable cyclic flow:

- *50-60 at the lowest flow rates*
- *10-20 near BEPQ*



Effect of %BEPQ on streamlines and velocities



Effect of %BEPQ on streamlines and velocities

Figure 11. Static pressure contour plots, on center-planes at 100% BEPQ

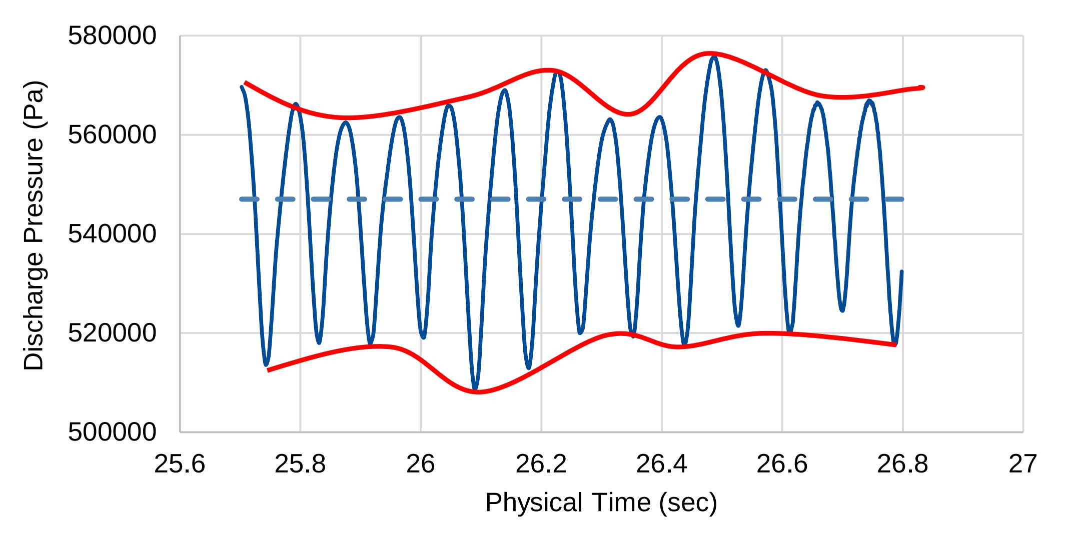


Figure 6. Discharge pressure vs. time @ 45 % BEPQ, dashed line represents mean value.

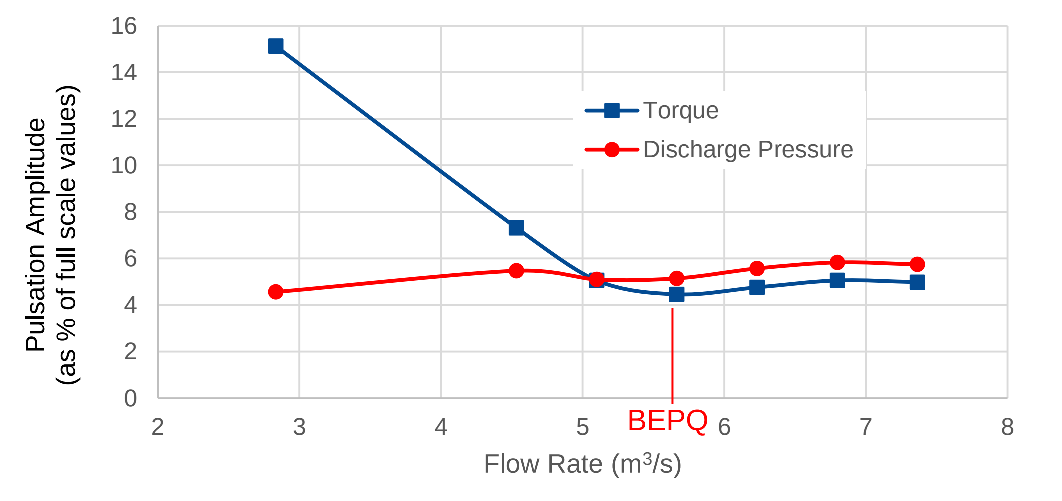


Figure 7. Relative torque and discharge pressure pulsation amplitudes vs. flow rate.

Pulsations

Pressure pulsations are slightly non-symmetric in shape: rounded at the maxima and sharper at the minima.

Vane pass frequency dominates, but secondary recirculation frequencies are also visible at low flowrates.

Magnitude of pressure pulses relatively constant with respect to %BEPQ.

However, torque pulses increase significantly at lower flowrates.

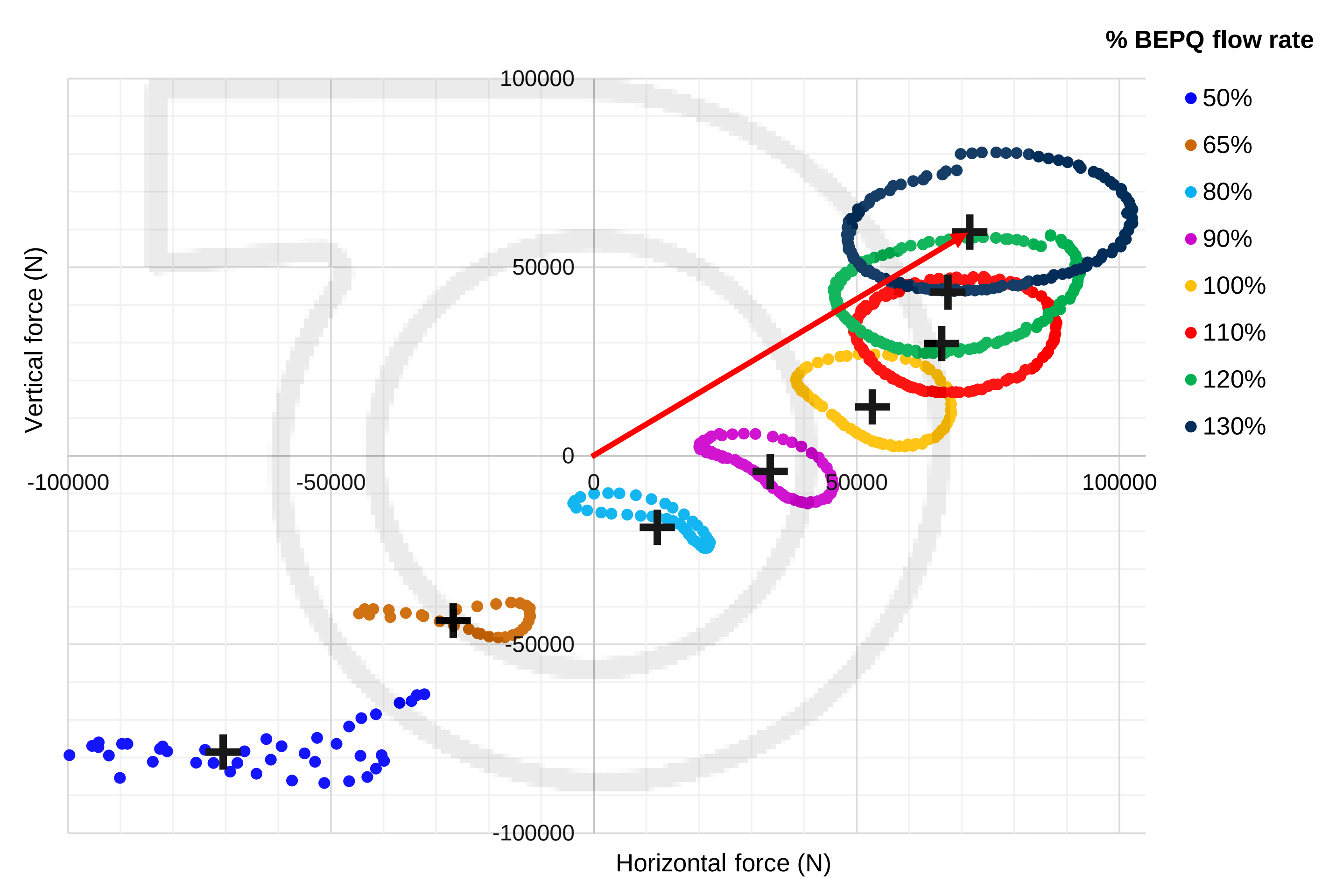


Figure 8. Vertical vs. horizontal radial load on impeller, as a function of %BEPQ. Point of view is from the pump drive side with casing at top horizontal, left facing discharge, (+) symbols indicate mean values.

Hydraulic Radial Load

Hydraulic radial load on the impeller calculated by integrating pressure and shear forces over all surfaces.

Transient results plotted for one vane pass.

Magnitude and direction of average force at each flowrate given by vector from origin to centroid "+".

Magnitude and direction vary greatly with flowrate.

Minimum near 80% BEPQ.

"Vapor Volumes"

Some indication of *NPSHR* performance can be determined by evaluating the volume of low pressure liquid in the flow stream.

By minimizing these low pressure volumes, improved *NPSHR* performance can be achieved.

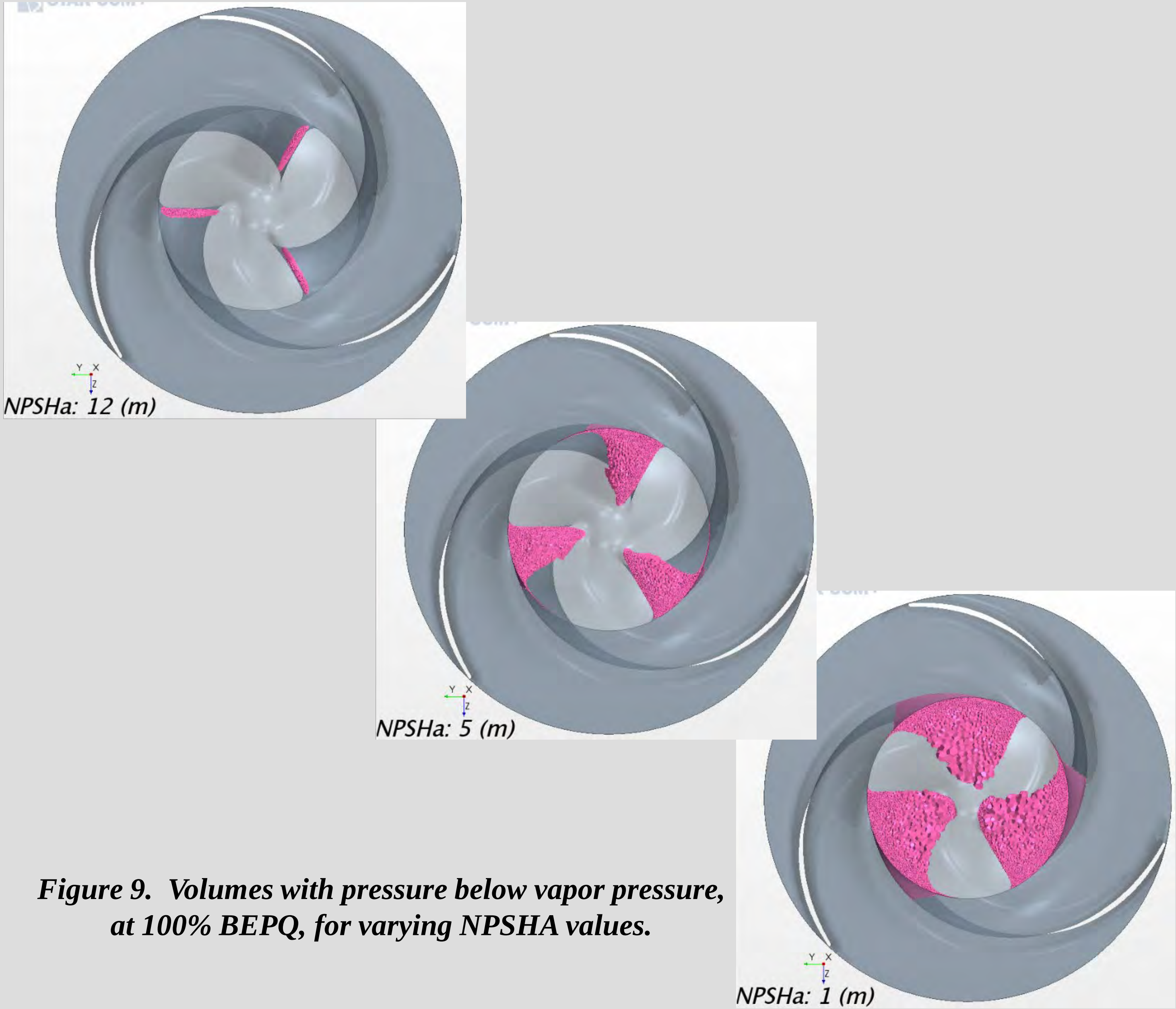


Figure 9. Volumes with pressure below vapor pressure, at 100% BEPQ, for varying NPSHA values.

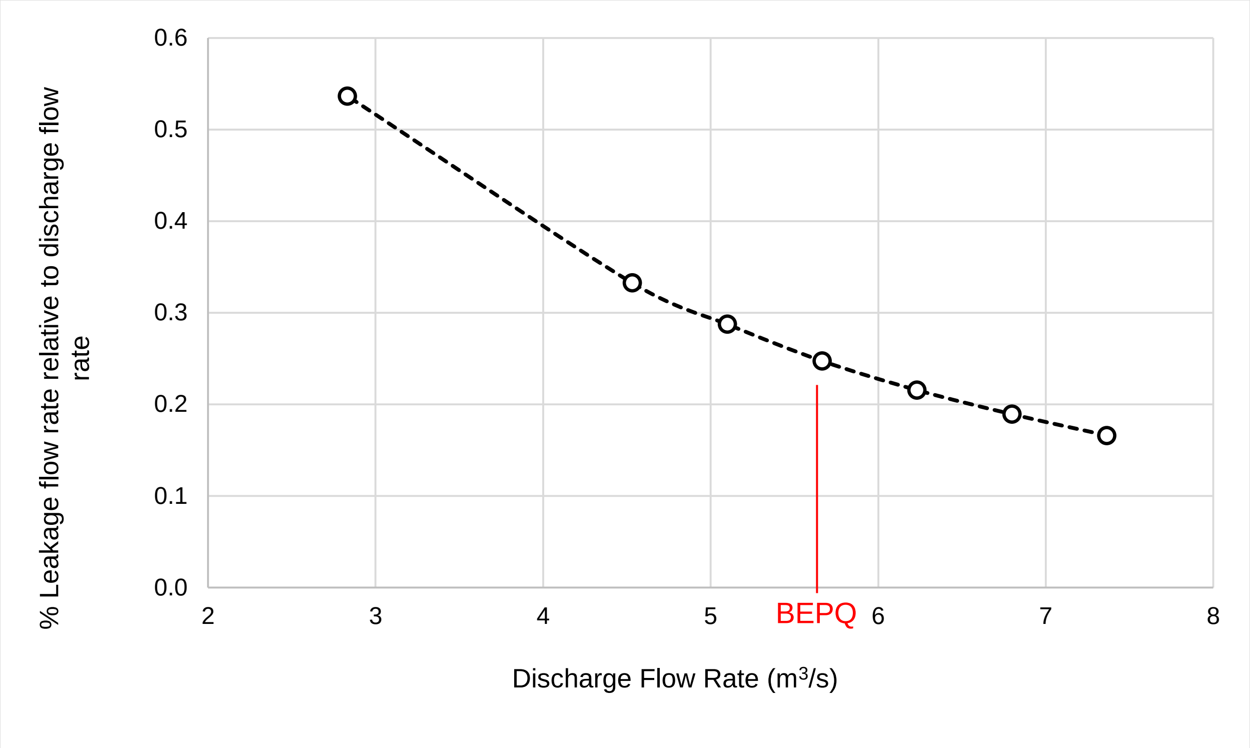


Figure 10. Leakage flow rate as a % of pump discharge flow rate.

Leakage Flowrate

Simulations were performed with and without side gaps.

The results showed that side gaps must be included for accurate prediction of efficiency, radial load, and torque, especially at flows away from 100% *BEPQ*.

Side gap modelling also allows prediction of the side gap leakage flowrates and velocities.

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Model test pumps

Full size pump:
S=1.0, D_2=2540 mm

Model 1:
S=0.25, D_2=635 mm

Model 2:
S=0.57, D_2=1447.8 mm

Model 1:

- *0.25 scale*
- *On GIW LCC 100 mm shaft frame.*

Model 2:

- *0.57 scale*
- *On GIW LSA 260 mm shaft frame.*

For both models:

- *Impeller and side liner custom made in pure scale.*
- *Impeller cast in ductile iron using a "printed" sand mold produced by rapid prototype technology from the 3D CAD model.*

Performance scaling

Head and flow are scaled according to the well-known centrifugal pump “affinity laws”.

Efficiency scaling according to Yoda et. al. (2016) and Anderson (1980).

- *More detailed Yoda treatment used for following evaluations, but less complex Anderson method gave similar results.*

Scaling of head and flow

$$Q_2 = Q_1 \left(\frac{n_2}{n_1}\right)\left(\frac{D_2}{D_1}\right)^3 \qquad H_2 = H_1 \left(\frac{n_2}{n_1}\right)^2 \left(\frac{D_2}{D_1}\right)^2$$

Efficiency scaling according to Yoda et. al.

$$\eta_2 = F_h \cdot F_m \cdot F_v \cdot F_b \cdot \eta_1$$

Where subscripts h, m, v and b refer to the hydraulic friction (main flow passages), disc friction (side gaps), volumetric leakage, and bearing losses respectively. In the present case:

$$\eta_{S=1.00} = 1.008 \cdot \eta_{S=0.57} = 1.039 \cdot \eta_{S=0.25}$$

Efficiency scaling according to Anderson

In the present case:

$$\eta_{opt} = 0.94 - 0.011378[\ (BEPQ/N)\ ]^{-0.2133} - 0.29[\ \log_{10}(44.26/n_q)\ ]^2$$

$$\eta_{S=1.00} = 1.012 \cdot \eta_{S=0.57} = 1.041 \cdot \eta_{S=0.25}$$

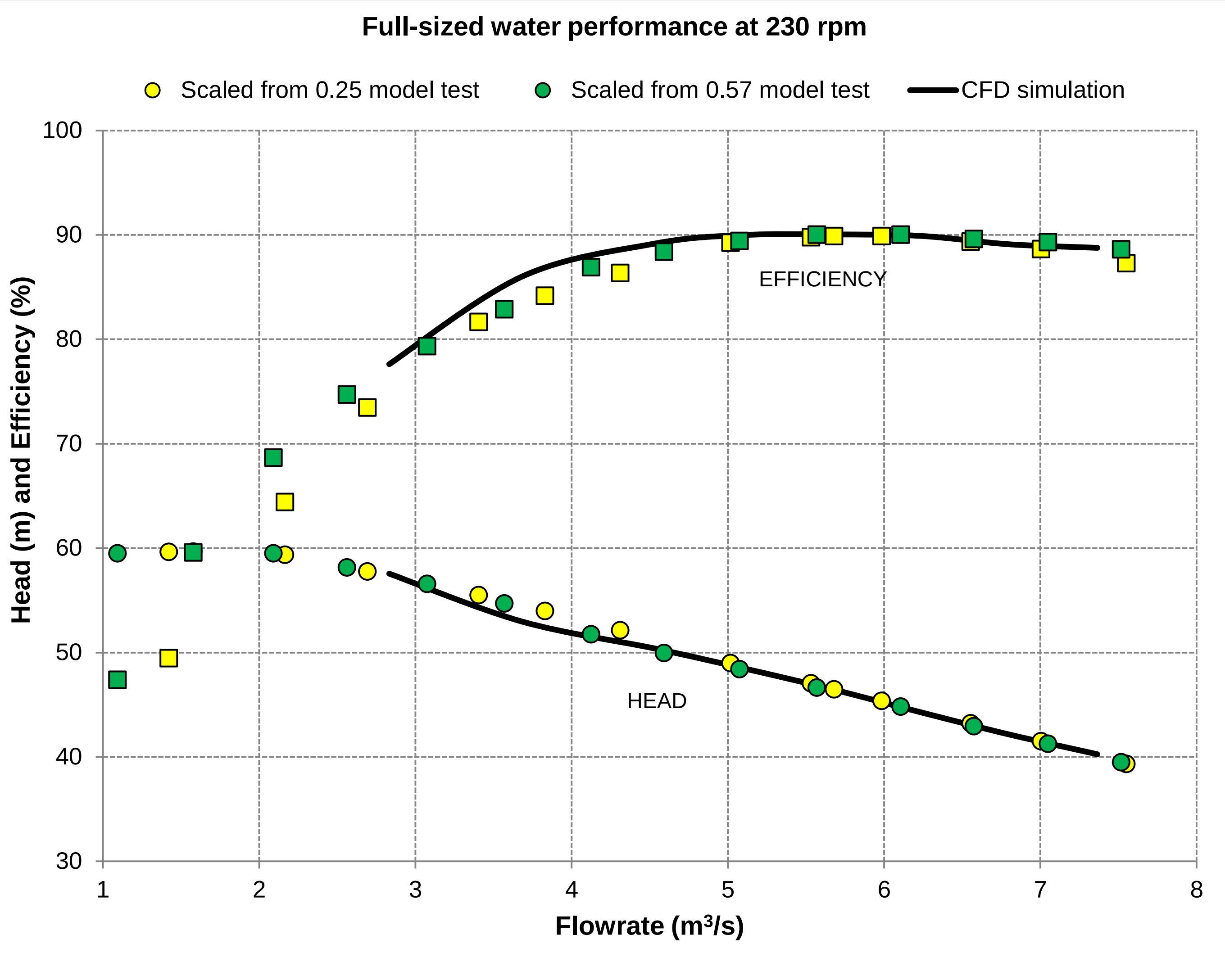

Figure 16. Head and Efficiency scaling for the original design.

Test results

Scaled head and efficiency closely follow CFD predictions.

Very good correlation in head between the two model tests.

Very good correlation in efficiency between the model tests at 100% BEPQ, but deviating at higher and lower flows, with the smaller pump under-predicting efficiency.

A correction factor for this "off-BEPQ" effect is proposed:

$$\eta_2 = [\,1 + abs(Q\text{-}BEPQ)/BEPQ\,]^{Bexp} \cdot F \cdot \eta_1$$

where:

F = *Efficiency scale ratio at BEPQ*

$Bexp = 0.02\ (D_2/D_1 - 1)$

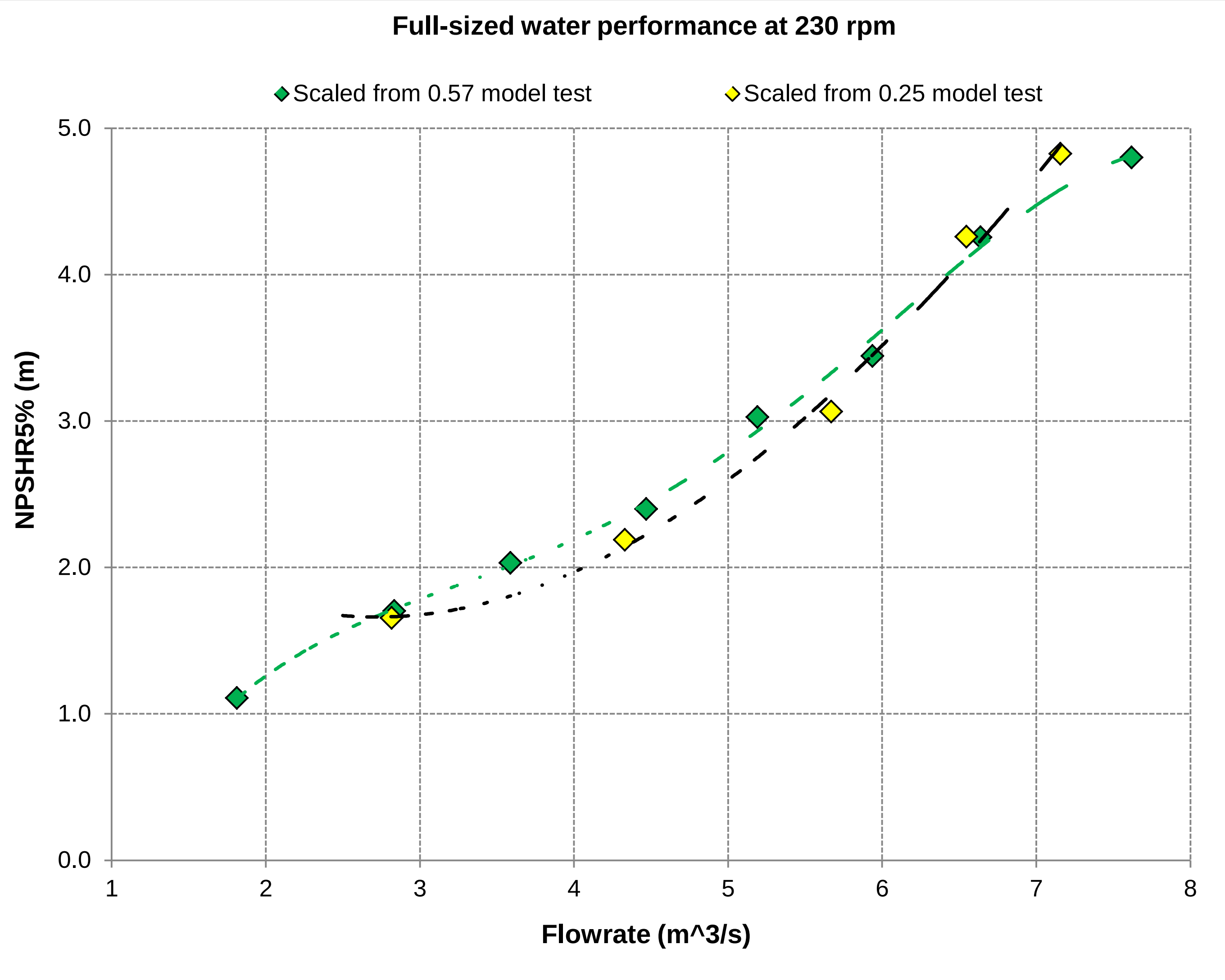

Figure 17 . NPSHR Scaling for the original design.

NPSHR scaling

In theory, the affinity laws for head should also apply to NPSHR.

In practice, the behavior of vapor in the impeller passages may vary with pump size.

- *Large passage impellers may carry a large volume of vapor before head breakdown occurs.*

The correlation in NPSHR between the two model tests did show some variation.

However, the scaled results show good agreement. No correction will be proposed without further study.

Figure 18 . Development of "vapor pressure volumes" in two designs.
Top: original design with box showing result at tested NPSHR5%.
Bottom: new design with box showing same "vapor volume" at lower NPSHA.

NPSHR improvement

It was desired to further improve the NPSHR performance.

However, CFD simulation of cavitation in a centrifugal pump is difficult to accomplish with the desired accuracy.

A method for incremental improvement, correlating NPSHR to the volume of low-pressure fluid was proposed.

A new design was created, manufactured and re-tested in the 0.57 scale model pump.

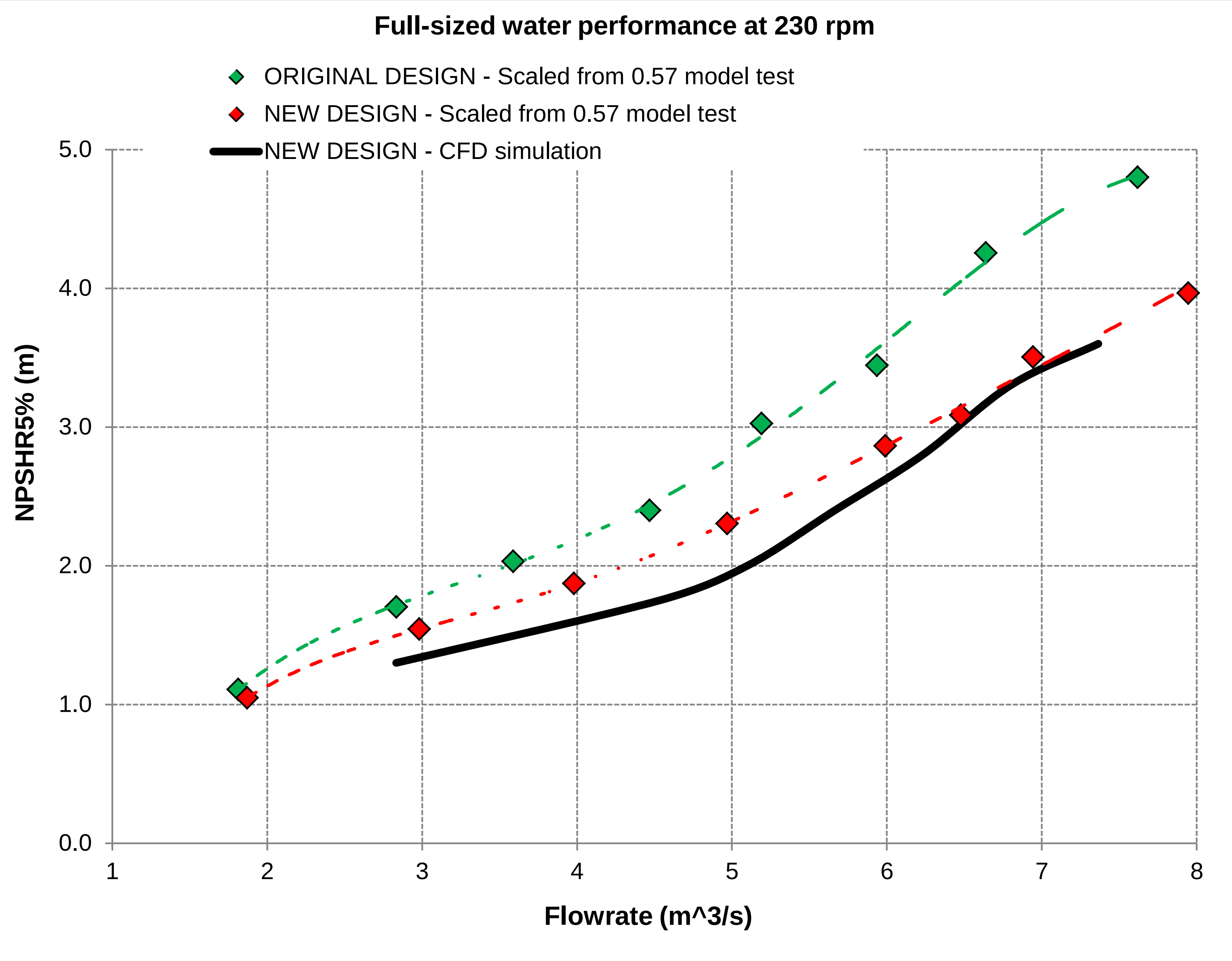

Figure 19 . CFD prediction and tested NPSHR improvement for new design.

NPSHR improvement

This method provided a useful tool for evaluating potential designs.

The estimated improvement in NPSHR determined for the final design correlated well with the model test results.

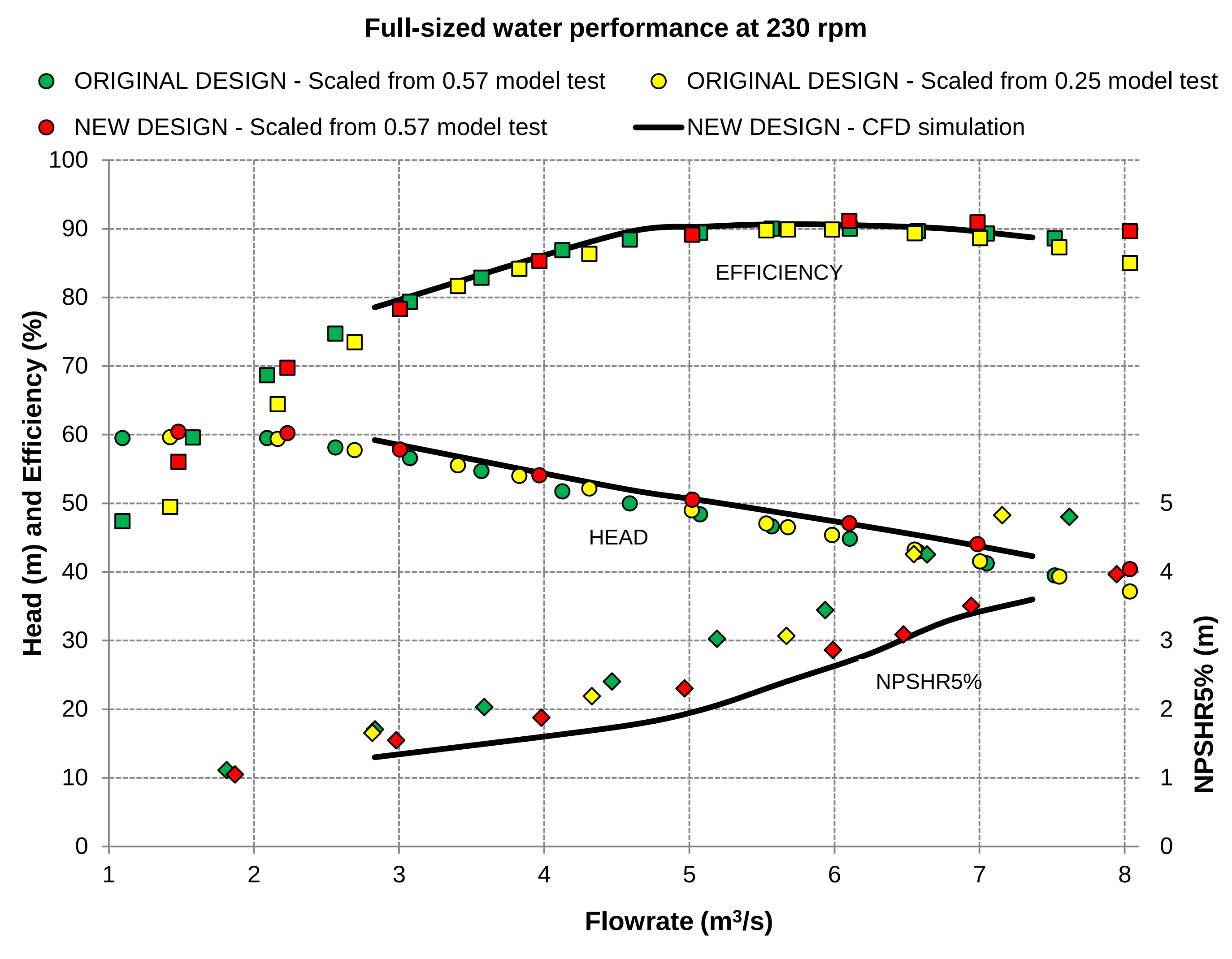

Figure 20 . Comparison of the three model pump tests and final design CFD.

Summary of results

For the final design with improved *NPSHR*, the head and efficiency are increased at higher flowrates.

The measured best efficiency of 90.4% for the new design at 0.57 scale exceeds the optimum efficiency predicted by Anderson.

- The fact that Anderson's optimum efficiency prediction is exceeded in a 3 vane dredge pump impeller with heavy section vanes and large sphere clearance validates the quality of the design and testifies to the improvements in design and analysis capabilities made since the 1980s.

Conclusions

1. Model testing can provide very accurate predictions of centrifugal pump performance (head, efficiency and NPSHR) over large scale ratios for low vane-number, large sphere clearance designs.

2. Scaling accuracy is best near the best efficiency flow rate (*BEPQ*).

3. Efficiency deviation away from the *BEPQ* increases with an increase in the modelling scale ratio, with smaller model pumps (relative to full-size) predicting lower efficiency. A correction factor for this effect is proposed.

4. NPSHR scaling showed a similar trend, but was still accurate enough for reasonable prediction of full-sized performance.

5. Comprehensive “full machine” transient CFD flow analyses including impeller, casing and side gaps can provide accurate predictions of pump head, efficiency and incremental NPSHR improvement over a wide range of flows for large passage, 3 vane dredge pumps.

6. Three vane centrifugal pump designs of large sphere clearance, low NPSHR and efficiencies approaching the empirically predicted optimums are possible, when supported by comprehensive and accurate CFD analyses.

7. NPSHR improvement relative to an existing, tested design can be reasonably predicted using single phase CFD analysis targeted to minimizing the volume of flow below the liquid vapor pressure.