TPS

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Investigating & Improving the drooping curve of a two-stage feed pump

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Presenter Biography

- Ng Tzuu Bin is the specialist engineer for Aftermarket Services & Solution (AMSS) Engineering team based in Flowserve Singapore. He has 10 years of work experience in the pump and seal aftermarket business. He obtained his PhD from the University of Tasmania Australia, specializing in the unsteady Francis turbine operation for hydroelectric plant.

The Abstract

A two-stage feed pump exhibited a drooping head-flow characteristic during its shop test. Impeller reworks were done to improve the drooping curve. CFD study was performed to examine the pump flow behaviour and a more stringent test procedure was implemented. The key lesson learnt from this case is not to overly push the efficiency of the pump at a single best efficiency point, but to have a more balanced design between achieving good pump efficiency and attaining a stable curve.

The Outline

- The Problem
- The Description of the Pump & the System
- The Rework & the Corresponding Shop Test Results
- The CFD Analysis
- The Refined Test for Shutoff Measurement
- Conclusion

The Problem

- Two-stage between-bearing pumps of the upgraded material were supplied to replace the existing machines.

- The pumps utilized a higher no. of impeller blades to obtain higher efficiency & head at its design point, but exhibited a drooping Q-H characteristic during shop test.

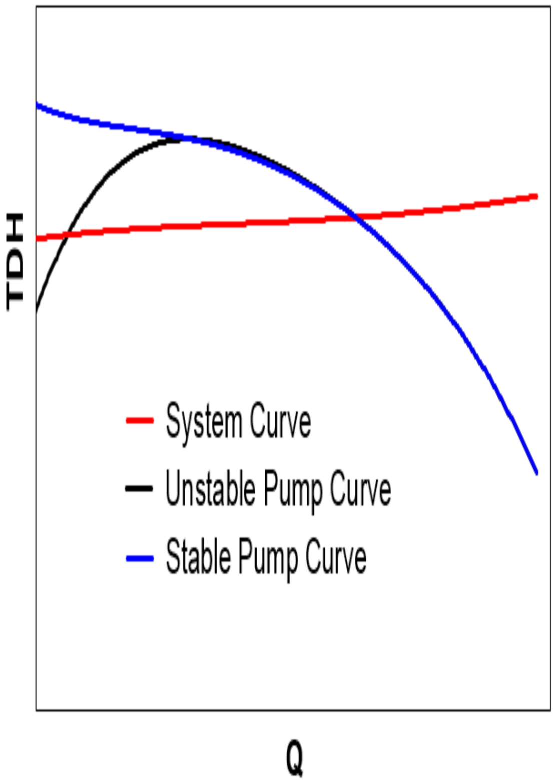
- Only eight serial numbers were found for this pump size. No drooping curve was previously reported. Deviation in cast geometry could be the probable cause.

The Definition of Head Droop

- The head droop (ΔH_d) occurs when the pump TDH does not rise continually when moving from BEP to shutoff.

- This could generate the static instability for a pumping system with high static/ pressure head & negligible friction head.

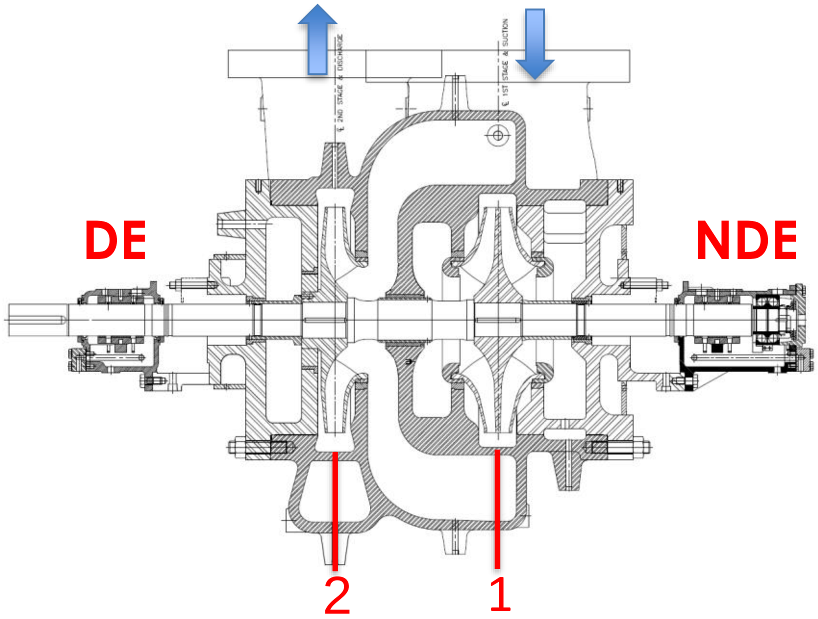
- API 610: "pump with continuous HRTSO is <u>preferred</u> for all applications & is <u>required</u> for parallel operation."



The Pumping System

- The pumps would NOT operate in parallel and at low flow.
- Flow meter was installed at discharge line for monitoring.
- The system curve is made up of the moderately low static/ pressure head and the non-negligible friction head where drooping pump curve will NOT present problem.
- Customer requested OEM to investigate and improve the amount of head droop on the test curve.

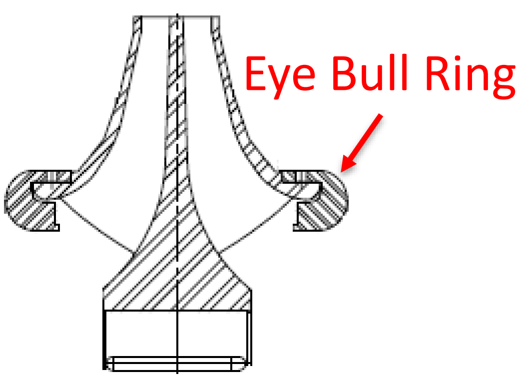
The Pump Information



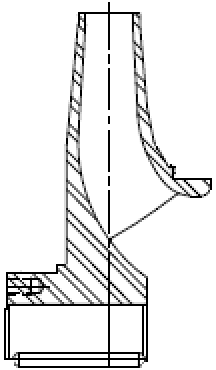
OPERATING CONDITIONS	
Stages	2
Rated Capacity (m^3/hr)	948.6
Suction pressure (kg/cm^2G)	9
Rated Differential Head (m)	256.7
NPSH available (m)	7.1m
LIQUID	
Type of Liquid	Hydrocarbon
Pumping Temperature (°C)	305
Vapor Pressure at Rated Temperature (barA)	0.02
Specific Gravity	0.757
Viscosity at Rated Temperature (cP)	0.833
Corrosive / erosive agent	High TAN Crude
PERFORMANCE	
Rotation (Viewed From Coupling End)	CCW
RPM	1491
Rated power (kW)	695.2 kw
Efficiency (%)	75.8%
NPSHR at rated capacity (m)	4.7 m

- BB2: 2-stage/ horizontal/ between bearing/ radially split.
- Pump material upgraded from CS to 316SS for corrosive crude.

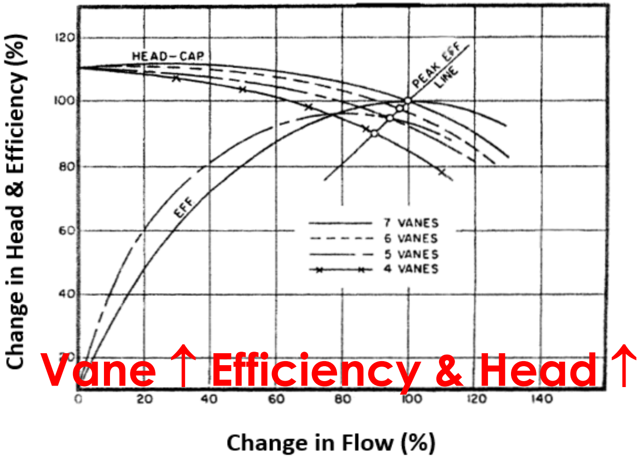
The Original Pump Impeller Design



Stage 1 Impeller



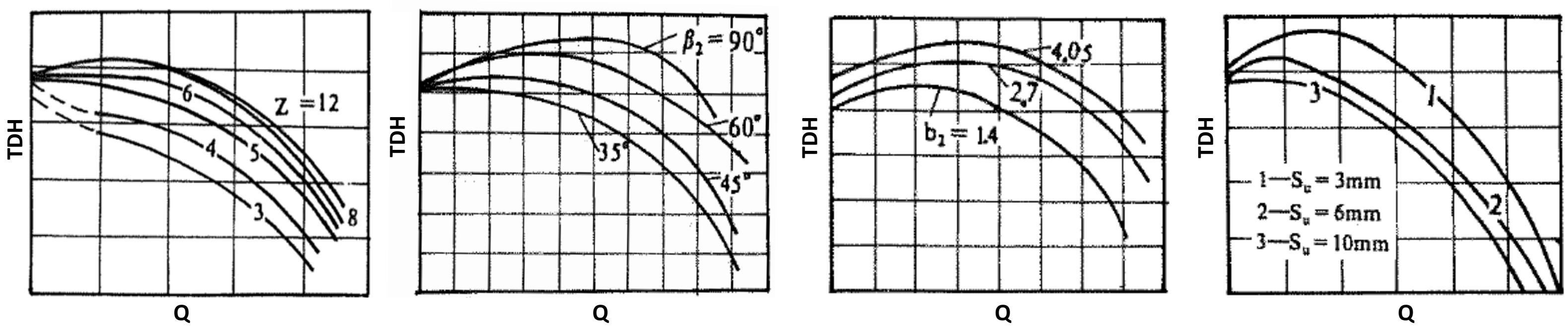
Stage 2 Impeller



Source: Lobanoff & Ross

- Stage 1 impeller: double suction, 5 vanes, eye bull ring.
- Stage 2 impeller: single suction, 8 vanes.
- Design specific speed, Ns ~ 1350.

The Pump Impeller Effect on Droop

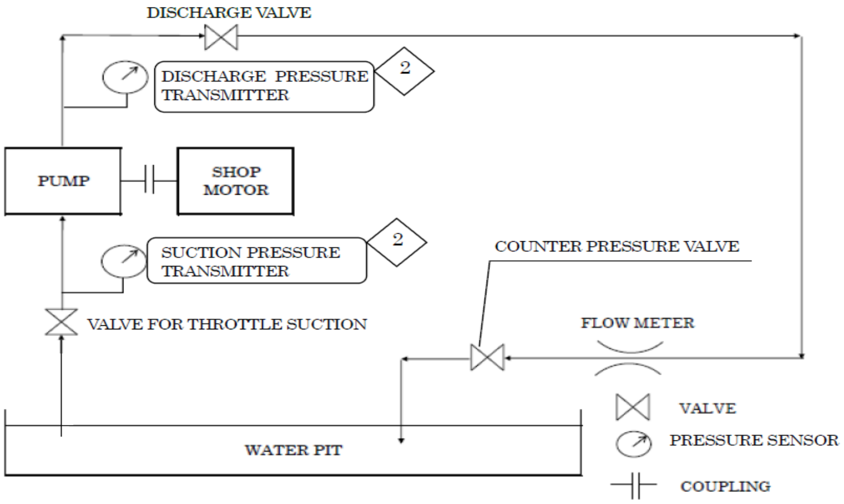


Source: Yuan et al.

- Many design parameters affecting the droop: vane no. (Z), exit angle (β_2), exit width (b_2), exit vane thickness (Su), diameter (D_2), meridional shape, inlet vane position, etc.

- These design parameters are very difficult & expensive to modify once the impeller has been casted. Minor rework is more appealing here.

The Initial Test Setup



- Test carried out in accordance with API 610 11th Ed.
- Pressures, flow, speed & motor power were measured.
- Pressure gauges were installed 2D away from pump nozzles.

The Pump TDH Calculation

$$\text{TDH} = (h_d - h_s) + \left(\frac{V_d^2}{2g} - \frac{V_s^2}{2g}\right) + (Z_d - Z_s)$$

Where $h_{s,d} = P_{s,d}/\rho g$ = suction/ discharge pressure head

$V_{s,d}^2/2g = Q^2/2gA_{s,d}^2$ = suction/ discharge velocity head

$Z_d - Z_s$ = suction & discharge gauge height differential

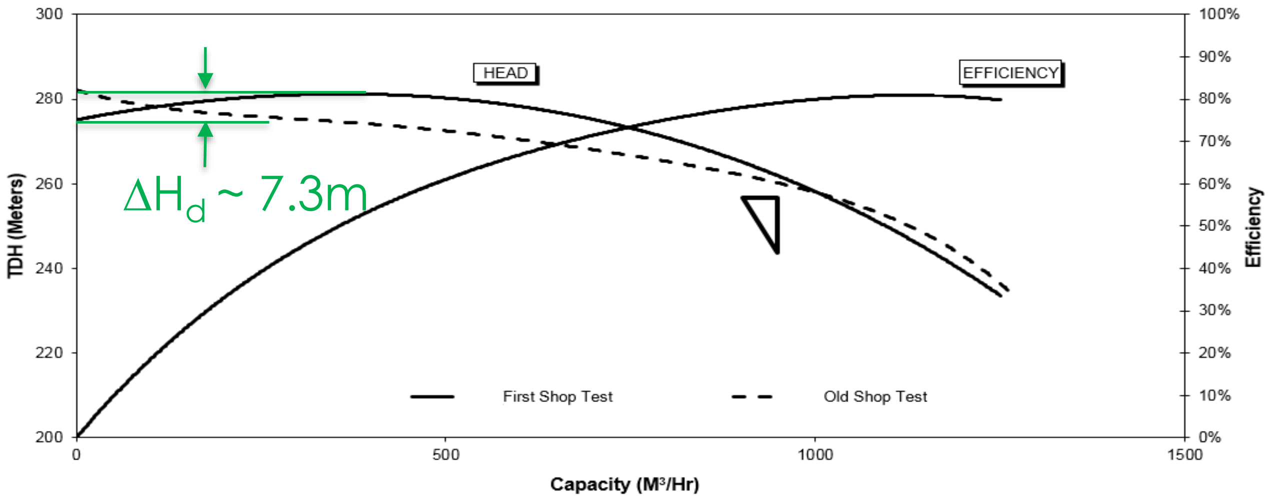
$P_{s,d}$ = measured suction/ discharge pressure

Q = measured flow

$A_{s,d}$ = suction/ discharge pipe cross sectional area

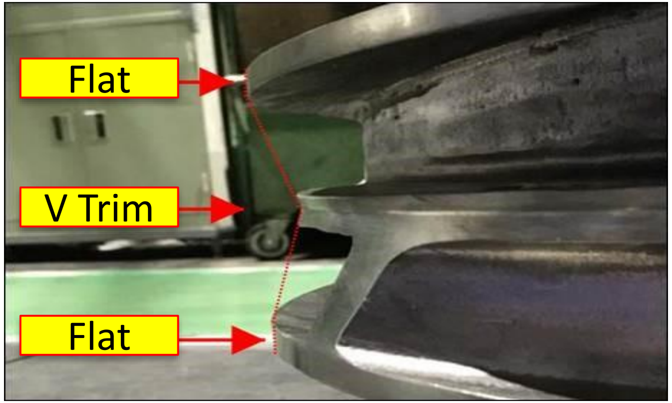
- Water density ρ was assumed constant.
- Flow velocity V was assumed uniform over pipe cross section.

The First Shop Test Result

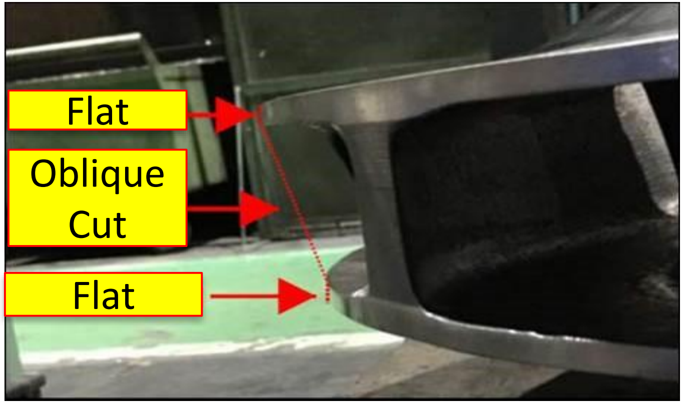


- TDH peaked at ~35% BEP flow in the first shop test. 7.3m head droop was observed here.
- Head droop was not reported in the test done 22 years ago.

The Rework 1: Impeller Oblique Cut & V-Trim



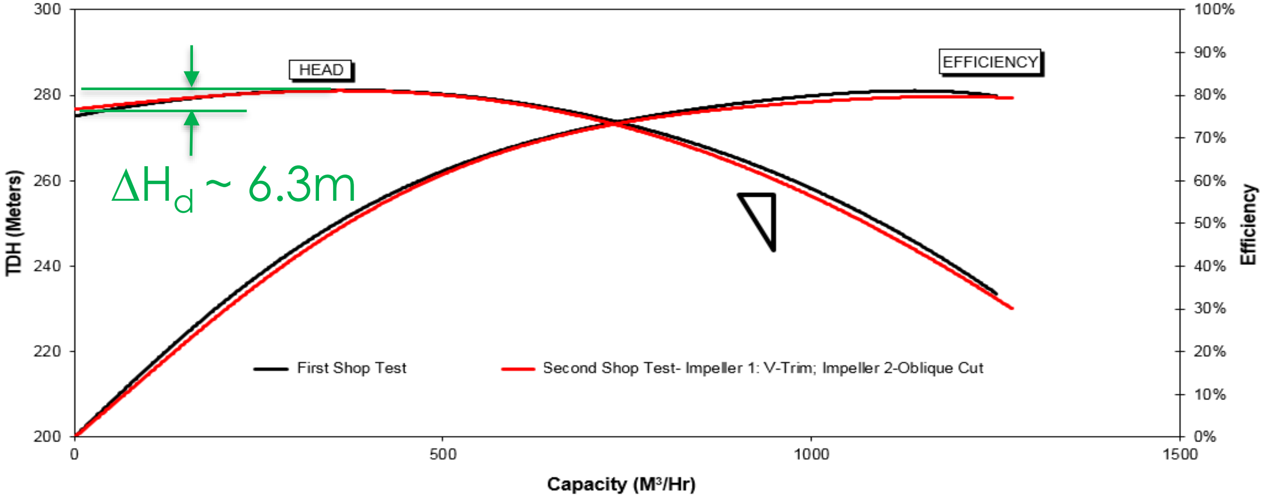
Stage 1 Impeller



Stage 2 Impeller

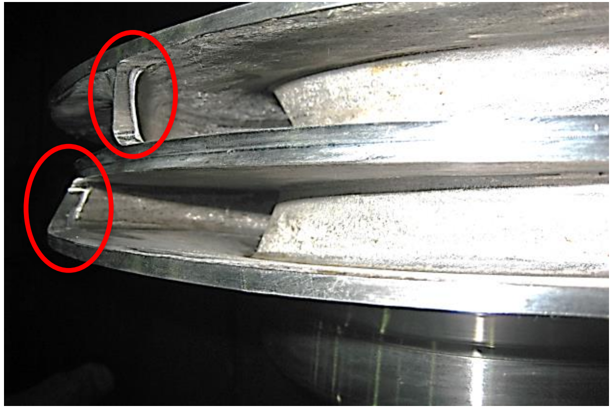
- Stage 1 Impeller (DS): V-trim angle 20°.
- Stage 2 Impeller (SS): Oblique cut angle 13°.
- This will reduce the secondary flow loss caused by streamline differences between impeller shroud and hub.

The Second Shop Test Result – Rework 1



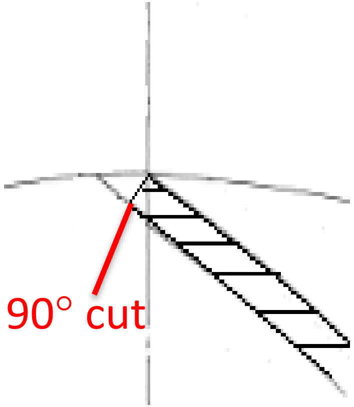
- Minor improvement observed. 6.3m of head droop remained in the test.
- Rated efficiency dropped ~1.1% (Compared to first test).

The Rework 2: Impeller Vane Squaring



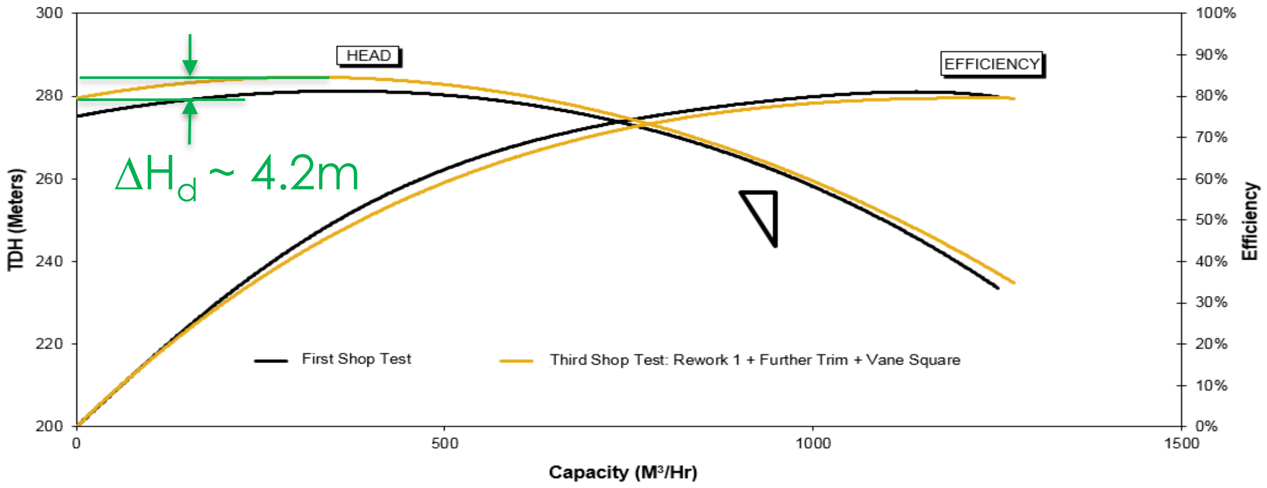
Stage 1 Impeller

Stage 2 Impeller



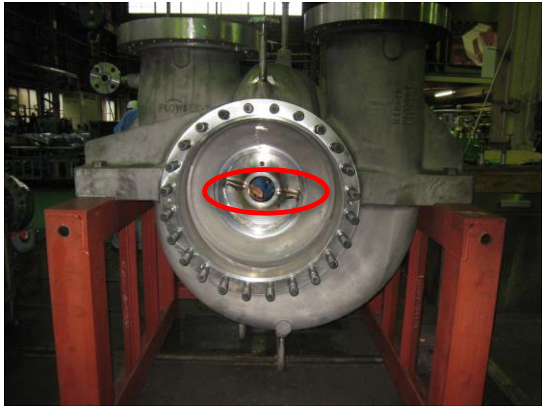
- Both Impellers: Cut the exit edge of vanes radially.
- This could reduce the wake shedding area due to traversing flow at the exit. Further trimming is needed, as the rework will rise the TDH at all capacities.

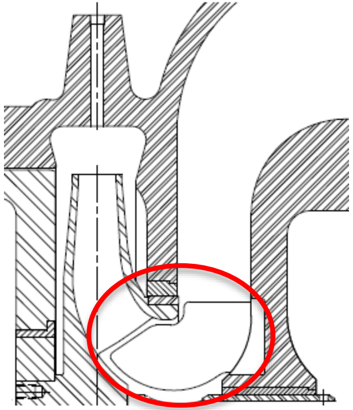
The Third Shop Test Result – Rework 2



- Substantial improvement obtained, but 4.2m of head droop remained in the test.
- Rated efficiency dropped ~1.3% (Compared to first test).

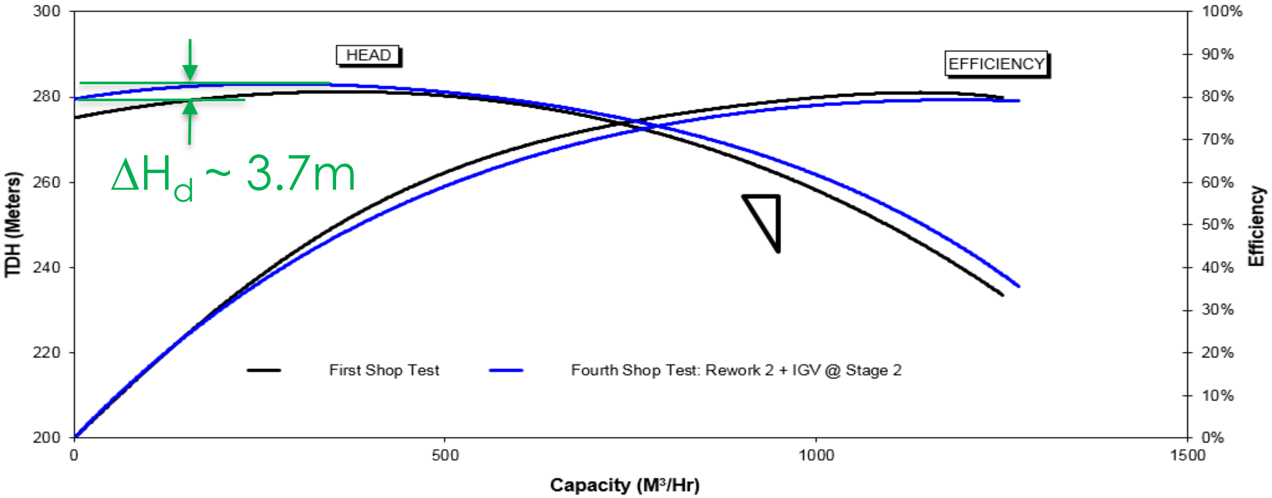
The Rework 3: Inlet Guide Vane at Casing





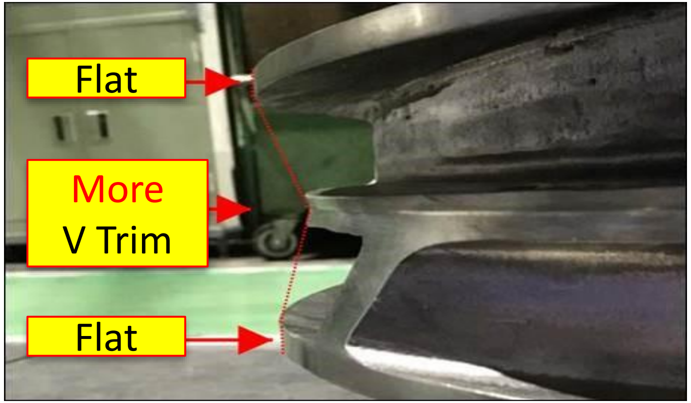
- Install the guide vanes at inlet of stage 2 impeller. The guide vanes extend right into the suction eye.
- This will reduce the hydraulic loss due to inlet pre-rotation.

The Fourth Shop Test Result – Rework 3

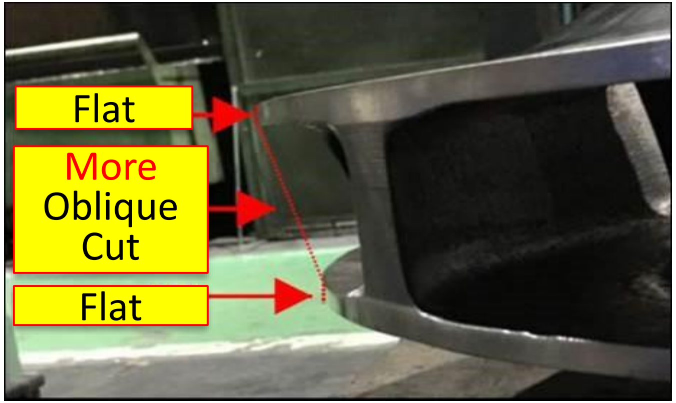


- Minimal improvement at low flow. Effect was more obvious at high flow. A 3.7m of head droop recorded in the test.
- Rated efficiency dropped ~1.5% (Compared to first test).

The Rework 4: More Impeller Oblique Cut & V-Trim



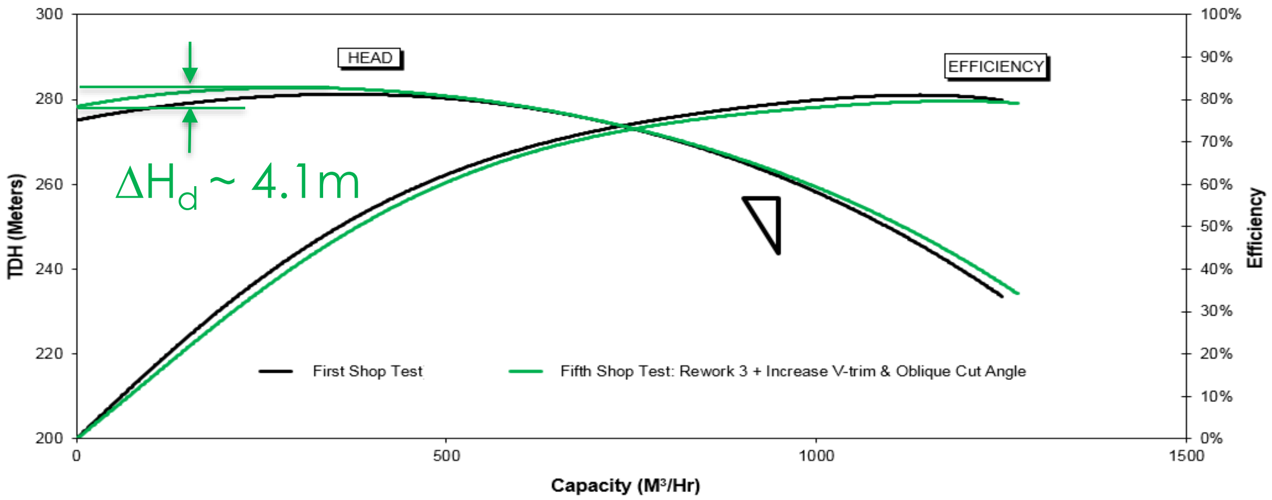
Stage 1 Impeller



Stage 2 Impeller

- Stage 1 Impeller (DS): Increase V-trim angle to 29°.
- Stage 2 Impeller (SS): Increase oblique cut angle to 32°.
- Positive outcome obtained for this method in the past.

The Fifth Shop Test Result – Rework 4

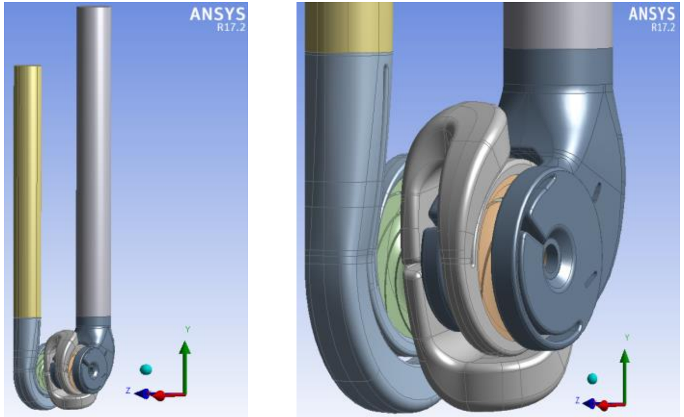


- Head droop increased to 4.1m against the expectation. Flow recirculation overwhelmed the impeller streamline difference.
- Negligible change on rated efficiency.

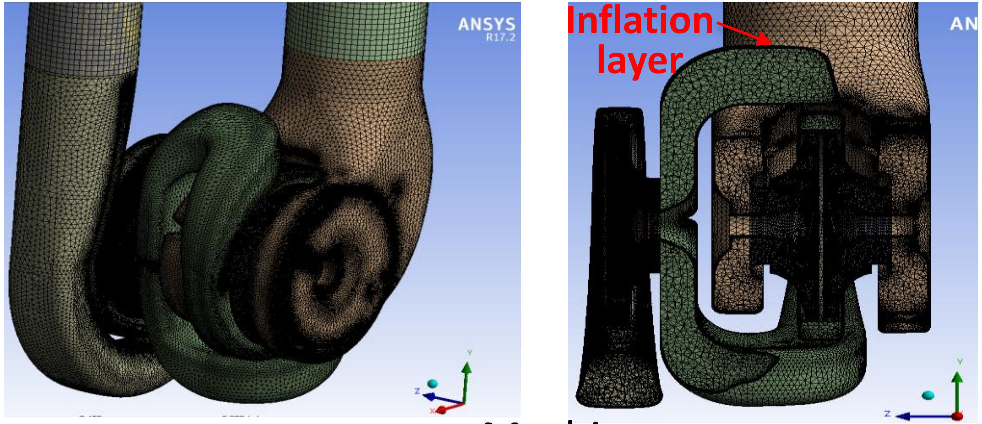
Summary of the Rework

- Impeller V-trim/ oblique cut generated minor recovery on head droop. Higher cut angle could have negative impact.
- Impeller vane squaring showed significant improvement in head droop, but further trim is required for rated TDH.
- Extended inlet guide vanes at stage 2 casing produced very minimal impact on head droop recovery.
- Improvement on the head droop was obtained at the expense of pump efficiency. None of these well known methods could completely remove the head droop here.

CFD Shows Potential Problems



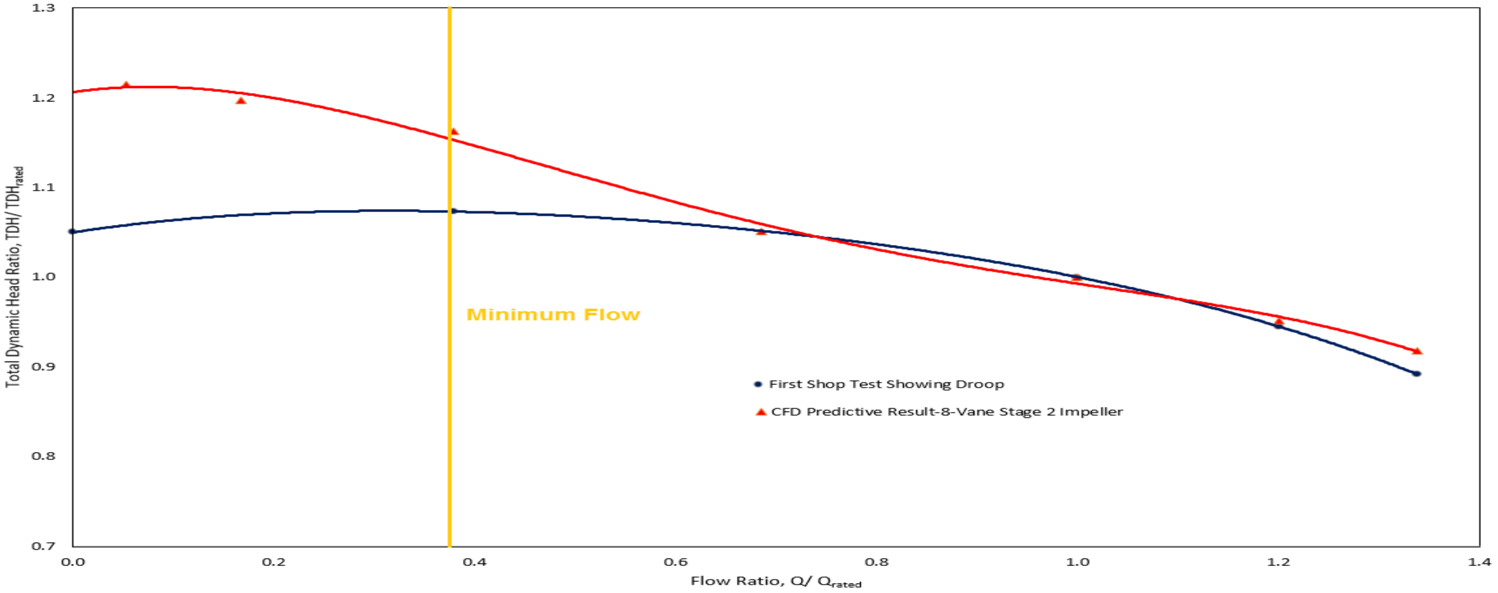
Flow Domain



Meshing

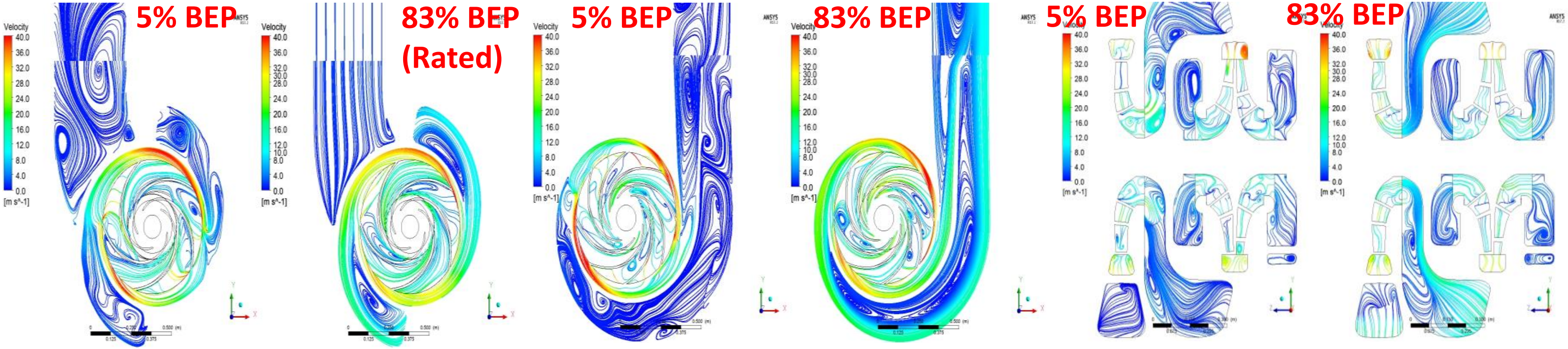
- Inlet/ outlet boundaries extended 10D away.
- 30M unstructured mesh elements with inflation layer.
- Frozen rotor approach & SST turbulence model applied.

CFD Result – Pump TDH prediction



- CFD over-predicted the TDH rise below the minimum flow.
- CFD did not clearly reproduce the head droop at the test.

CFD Result – 2D Streamline Plot



Stage 1: 5-Vane Impeller

Stage 2: 8-Vane Impeller

Crossover

- Flow recirculation & separation in the volute & crossover. Such phenomena become more severe at low flow and will extend further into the suction/ discharge pipe.
- Highly transient flow observed below 35% BEP.

Summary of the CFD Analysis

- CFD provides a good qualitative assessment, although it doesn’t clearly reproduce the head droop effect from test.
- Flow recirculation & separation in the casing cause huge static and total pressure losses, especially at low flow.
- Pump flow is highly transient below 35% BEP. More time is needed for pump to settle down at low flow.
- Impeller rework will not likely resolve the head droop. Further casing modification is very costly.

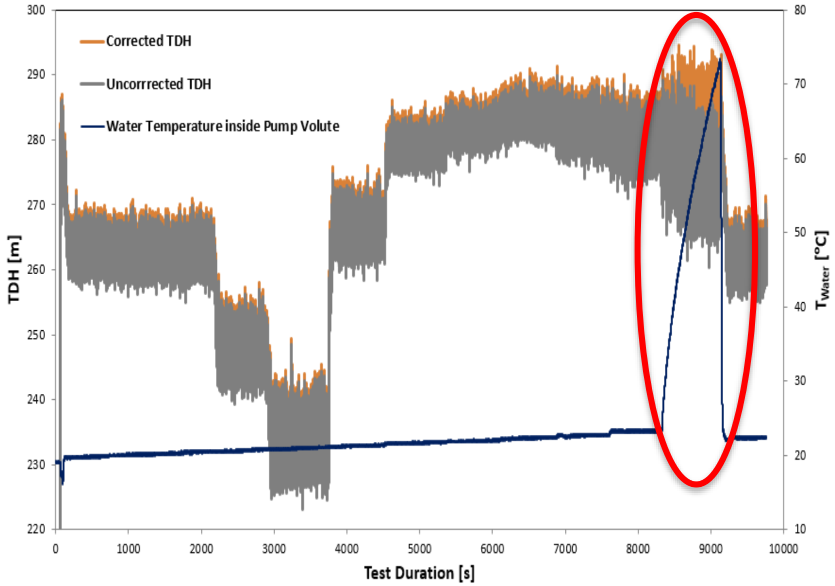
Testing at Pump Shutoff

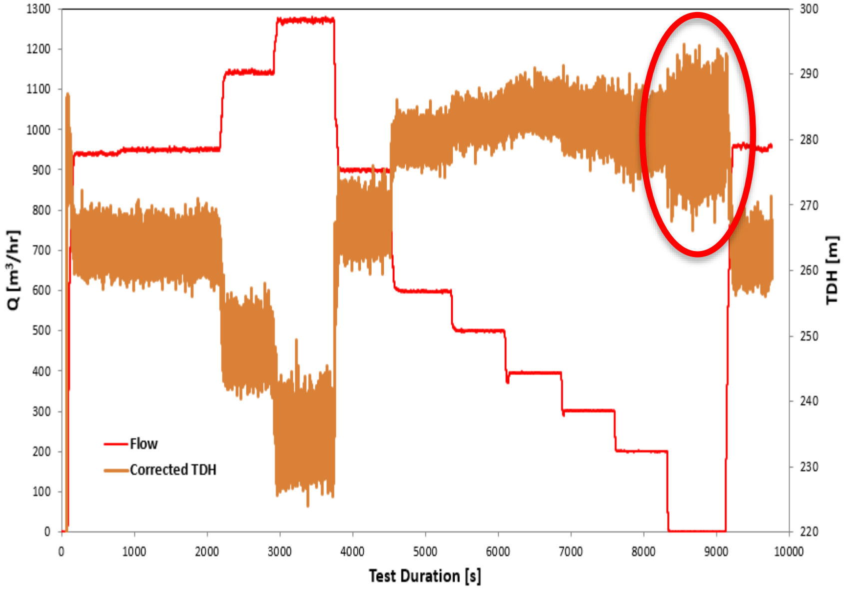
- Two problems:
  - Flow is highly transient as learnt from CFD. More time is required for pump to settle down during shutoff test.
  - Continuous pumping temperature rise at shutoff. Calculated temperature rise at shutoff is **4-9°C/min**. Static pressure head will be affected as the fluid density is temperature dependent.

The Modified Test Setup

- Fast response RTD probe was installed at casing vent to monitor the temperature rise.
- Higher sampling rate was set in the data logger.

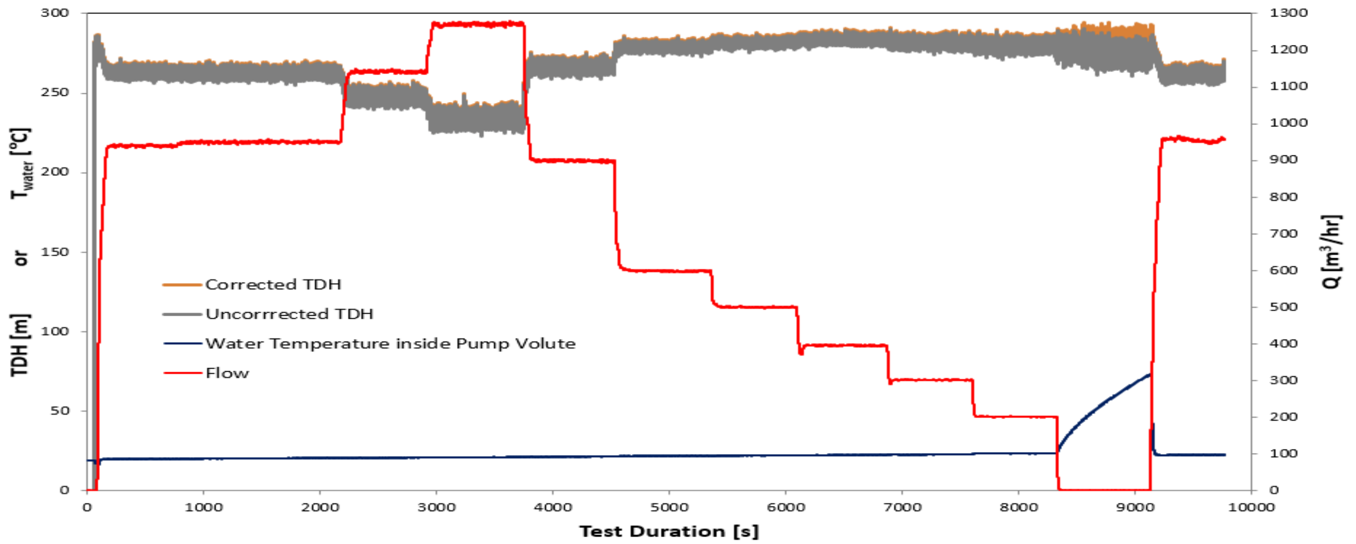
The Pump Shutoff Test Result





- Water temperature rise effect must be corrected at shutoff.
- TDH fluctuated more at the reduced flow.

The Pump Shutoff Test Result



- No hunting curve effect was observed during the low flow test.
- Unsteady dynamic pressure head was not accounted in the TDH calculation.

The Pump Shutoff Test Result

% BEP	Flow Rate, Q [m3/h]		Uncorrected TDH [m]		Corrected TDH to Pump Temp [m]		
	Mean	Standard Devation	Mean	Standard Deviation	Mean	Standard Deviation	
111.3%	1271.0	3.0	233.7	2.8	234.7	2.8	
100.0%	1141.7	2.5	247.3	2.4	248.3	2.4	
83.2%	949.8	2.0	262.4	1.9	263.5	1.9	Rated Point
78.7%	898.9	2.1	266.2	2.1	267.3	2.1	
52.4%	598.4	1.6	279.0	1.5	280.2	1.5	
43.8%	499.8	1.1	281.5	1.5	282.3	1.5	
34.6%	395.4	0.9	283.5	1.5	284.0	1.5	Drooping Point
26.4%	301.2	0.8	281.6	1.9	282.9	1.9	
17.6%	200.9	0.7	279.8	2.3	281.3	2.3	
0.0%	0.5	0.5	275.8	4.1	281.3	3.9	Shutoff Point

- Standard deviation of the measured TDH ↑ as Q ↓.
- The measured head droop was statistically insignificant.
- Customer witnessed the test & accepted the result.

Conclusion – Lessons Learnt

- A balance between achieving good pump efficiency and attaining a stable curve must be considered at early design stage. Fixing head droop at the test stand is expensive (>$100K) and time consuming (4-6 months).
- Minor rework could improve the head droop but at the expense of pump efficiency.
- CFD provides good qualitative assessment on the head droop.
- Test setup must be refined for pump showing <5% HRTSO.

References

- De Neumann, B., " The interaction between geometry and performance of a centrifugal pump", Mechanical Engineering Publications UK, 1991.
- Lobanoff, V.S. & Ross, R.R., "Centrifugal pumps design & application", 2nd ed., Gulf Publishing USA, 1992.
- Yuan et al., "Design method of obtaining stable head-flow curves of centrifugal pumps", Pumping Machinery Symposium, ASME FED Vol. 154, ASME, 1993.