TPS

51ST TURBOMACHINERY & 38TH PUMP SYMPOSIA

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Fast and Ultimate Vibration Field Solution: From Problem Detection to Field Performance Validation

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FLOWSERVE

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Authors' biography

Giancarlo Cicatelli is currently holding the position of Product Engineering Manager for Flowserve in the Engineered Pump Division. He has started his industrial career serving as product design specialist, R&D team leader and manager of the Customer Service for the O&G business. He has his office in Italy.

Previously he was Research assistant at the von Karman Institute in Belgium and at the University of Cambridge (UK), focusing his interests in the field of Fluid Dynamics of Turbomachinery, in the area of gas turbine research.

He holds a PhD in Applied Sciences from the University of Brussels and has a master in Nuclear Plant Construction management from Polytechnic of Milan. He obtained his BS degree in aeronautical engineering from the University of Naples, Italy.

He is author of several scientific publications in the field of Fluid Dynamics applied to turbines and pumps.

He is member of the Advisory committee for the Middle East Turbomachinery Symposium and member of the Europump Technical committee.

Authors' biography

Bruno Schiavello is Consultant, Pump Fluid Dynamics, since May 2020. He retired as Research Fellow, Hydraulics, at Flowserve, Pumps Department, in Bethlehem, Pennsylvania, USA and previously served as Director for Fluid Dynamics with Ingersoll Dresser Pump Company, Phillipsburg, New Jersey, since 1993. He started in 1975 with the R&D Department of Worthington Nord (Italy), joined in 1982 the Central R&D of Worthington Pumps, USA, then Dresser Pump Division.

Mr. Schiavello was co-winner of the H. Worthington European Technical Award in 1979. He has written several papers and lectured at seminars in the area of pump suction recirculation, cavitation, and two-phase flows. He is a life member of ASME, and former Associate Editor for ASME Journal of Fluids Engineering (two terms). He has received the ASME 2006 Fluid Machinery Design Award, the ASME 2016 Henry R. Worthington Medal, and the ASME Medal and Certificate as Eminent Fluids Engineer at the Celebration of the 90th Anniversary of the Fluids Engineering Division, Washington DC, 2016. Also, he has been Co-Lead Organizer of the ASME International Symposium on Pumping Machinery in 2005, 2009, 2011, 2015, 2017 and 2019. He has served on the International Pump Users Symposium Advisory Committee since 1984.

Mr. Schiavello received a B.S. degree (Mechanical Engineering, 1974) from the University of Rome, Italy, and a M.S. degree (Fluid Dynamics, 1975) from Von Karman Institute for FluidDynamics, Rhode St. Genese, Belgium.

Summary

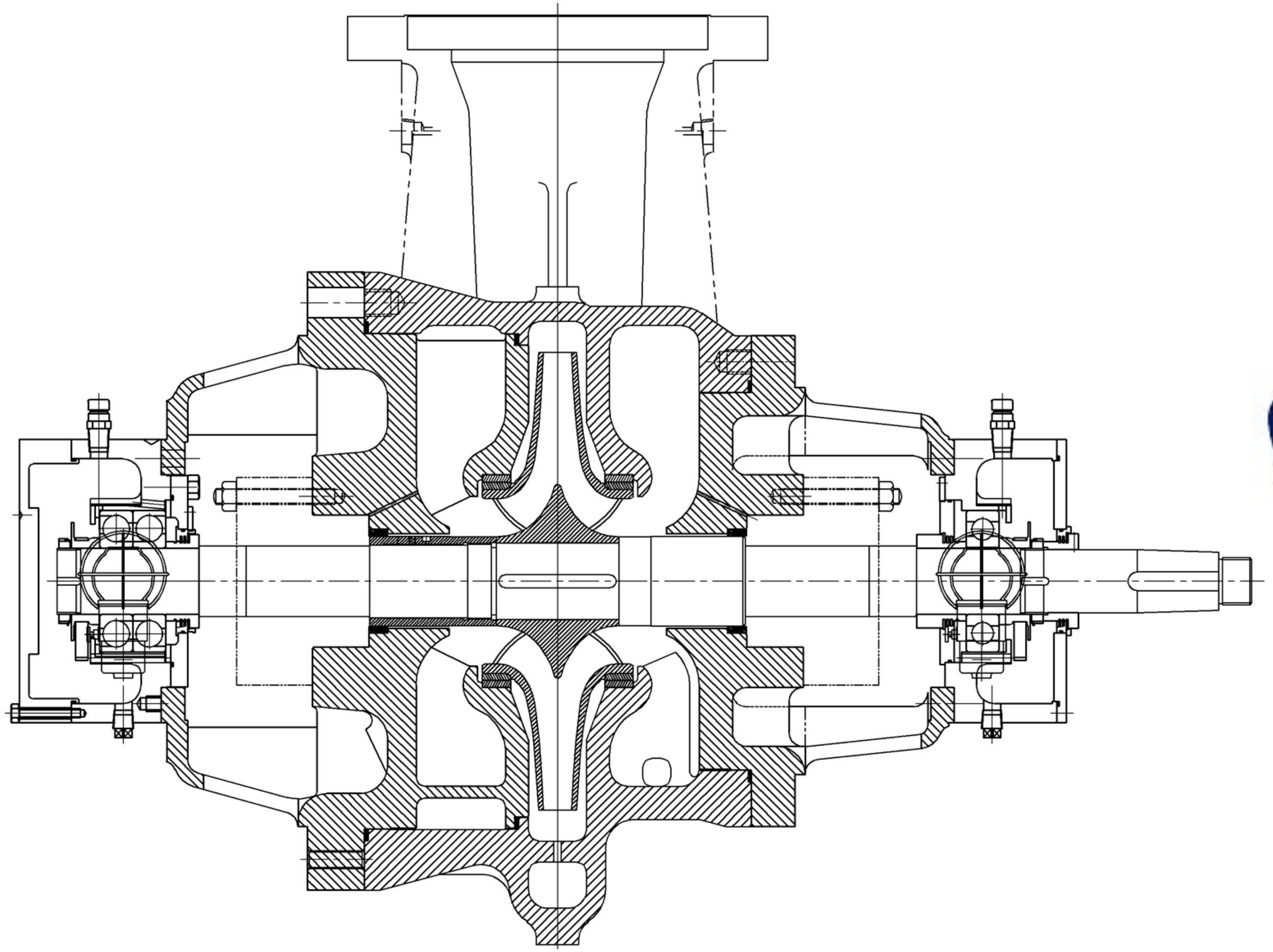
- The case
- Initial analysis
- Root cause analysis
- Solution implementation
- Results
- Conclusions

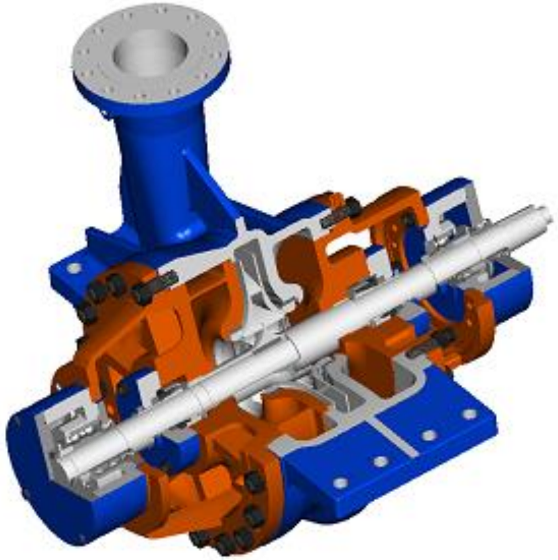
Description of the problem

During commissioning, customer reported unacceptable vibration levels on pumps tested with water.

The pump

The pump type/size is an API 12" discharge with top-top configuration, double suction impeller, double volute, anti-friction bearings configuration, 360°center mounted (BB2)



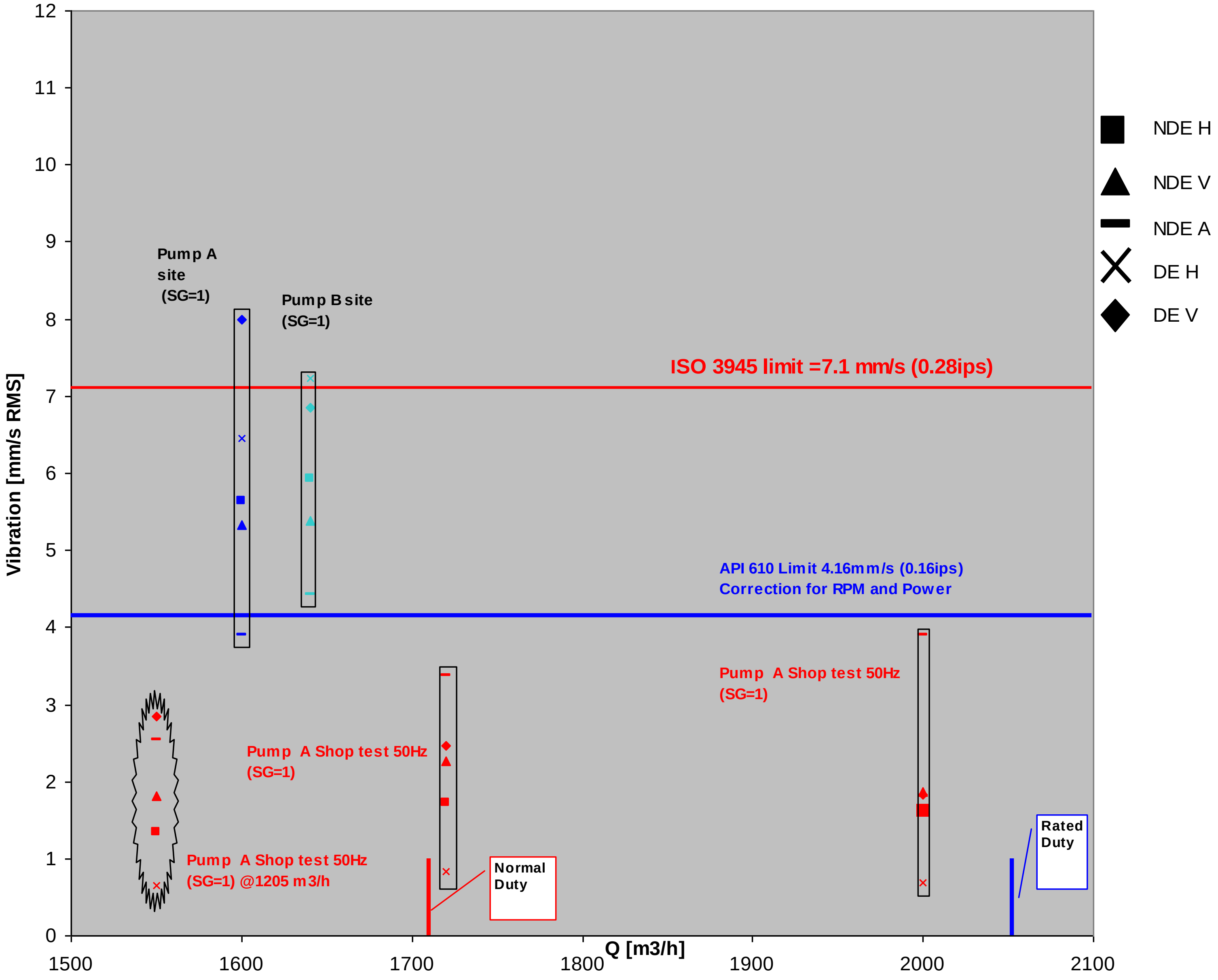


First steps of the investigation

On site inspection to verify mechanical integrity of pumps

Campaign of vibration measurement on all installed pumps

Initial analysis



51ST TURBOMACHINERY & 38TH PUMP SYMPOSIA

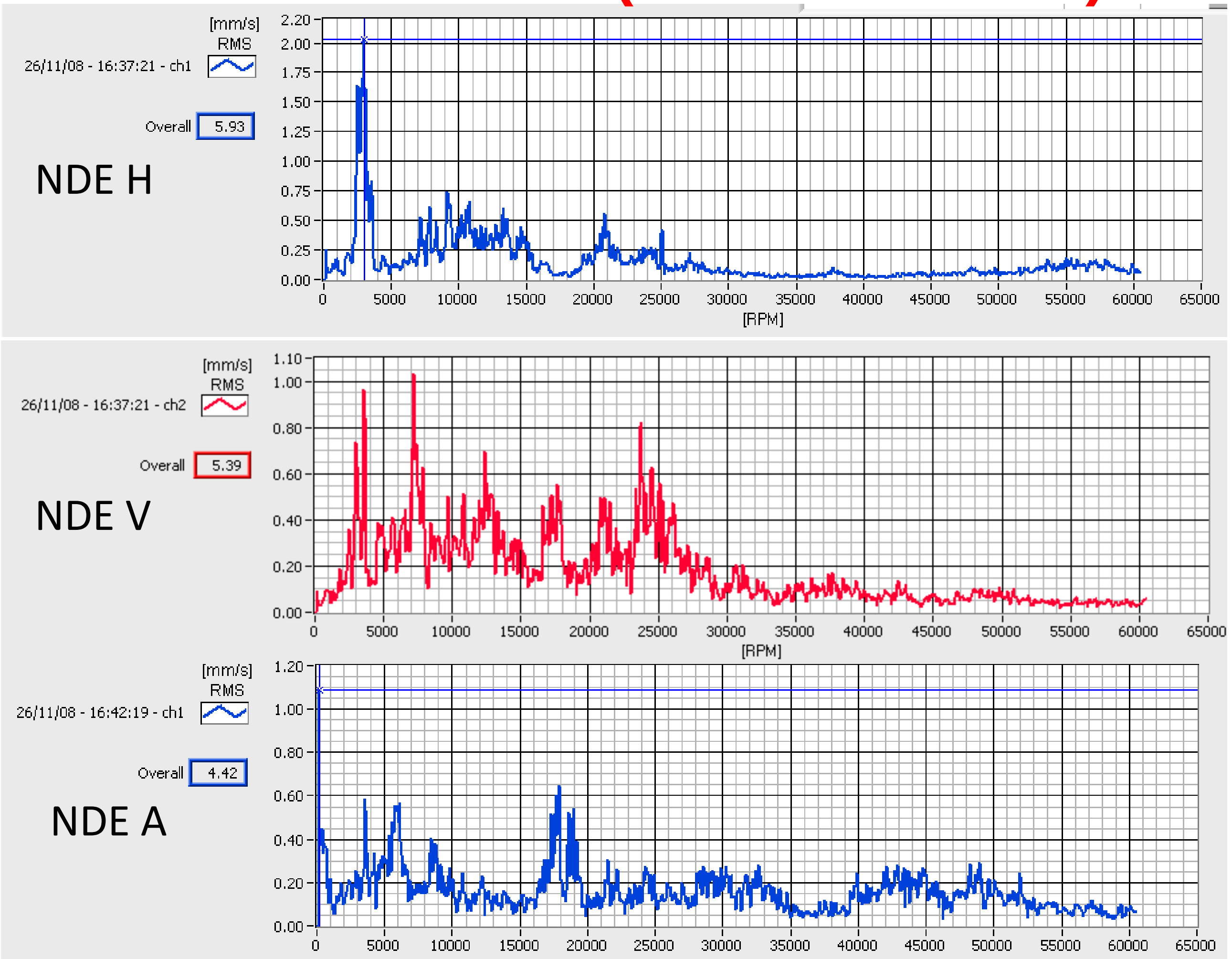
Initial analysis

Spectra analysis and main outcomes

- Confirmation of the measure taken by customer
- Similar behaviour on the two pumps
- Frequency spectra with broadband showing peaks distributed for many frequencies up to 500 -700 Hz (low - medium range).
- Filtered vibrations at key characteristic frequencies have (1x, VPF) have amplitude around 1.0 – 1.5 mm/s (0.04 – 0.06 ips). But overall value is around 5-6 mm/s(0.2-0.24 ips), due to the high number of peaks
- Spectra instability, with high variations in different moments
- Phase not stable
- The higher vibration values were detected on pump casing, and not on the bearing

Initial analysis – Vibration spectra

Field data (November 2008)



NDE vibration spectra

N=3580 rpm
1x=3580 rpm
VPF= 7x=25060 rpm

Q=1700 m^3/hr (7490 gpm)
Close to Normal duty
SG=1
T=35°C
NPSHA/NPSHR=2.14

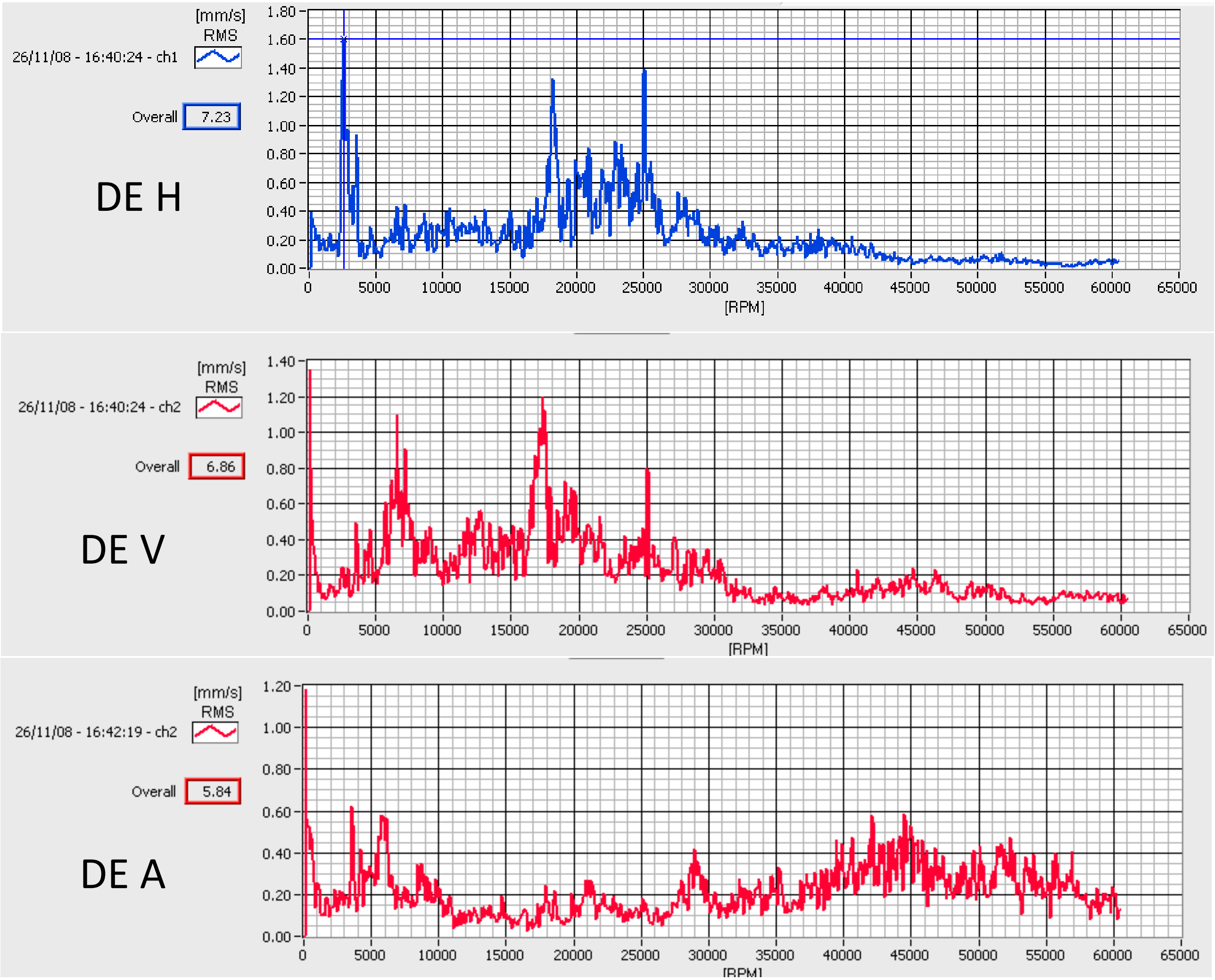
Remarks:

a) Low amplitude at VPF (< 1 mm/s = 0.04 ips)

b) High activity mainly across a range up 500 Hz (30000 rpm)

Initial analysis – Vibration spectra

Field data (November 2008)



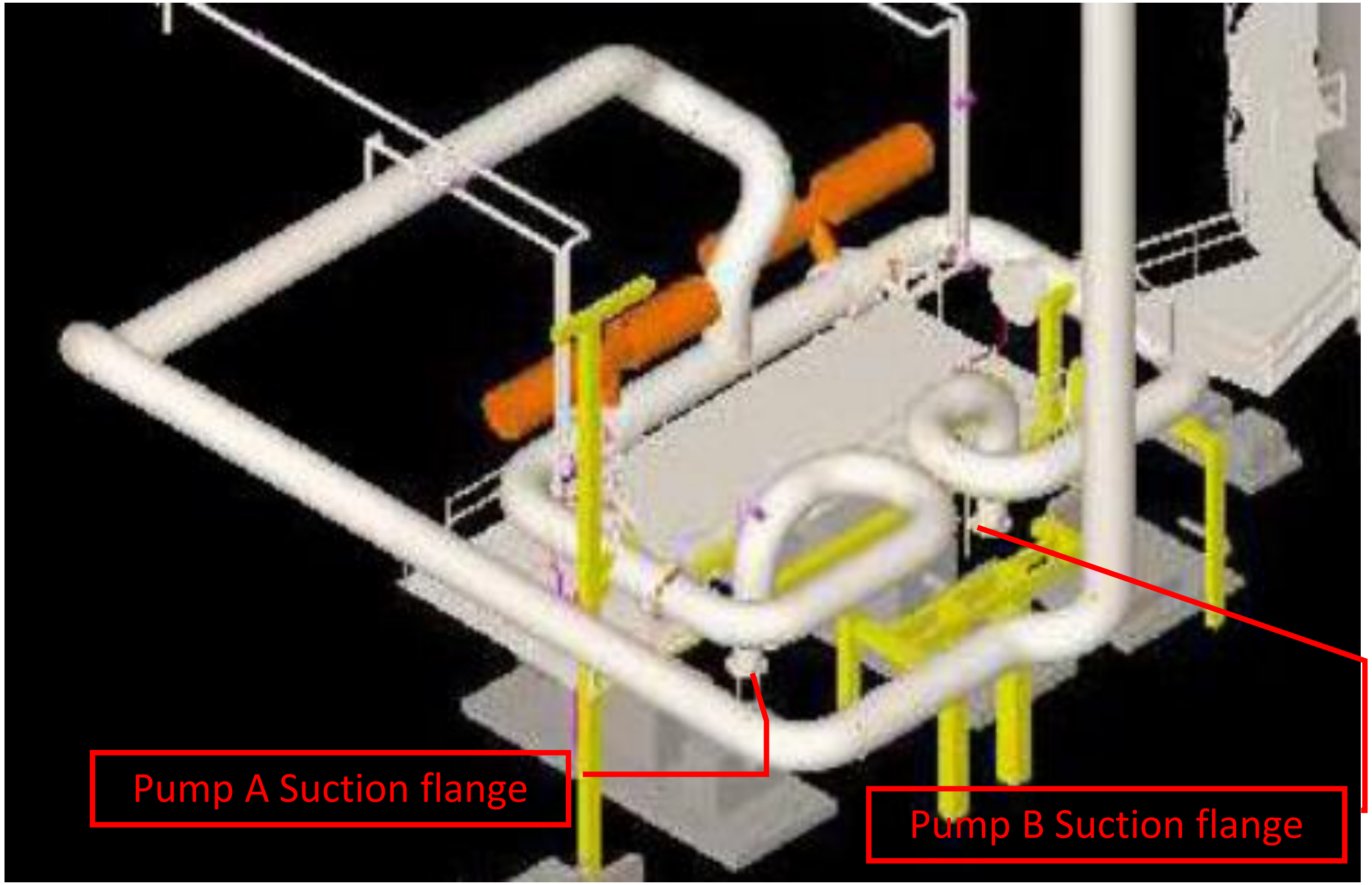
DE vibration spectra

Remarks :

a) Max amplitude at VPF : 1.4 mm/s = 0.06 ips)

b) High activity distributed and dominant across a range up 500 Hz (30000 rpm)

Initial analysis – Suction piping



Root Cause Analysis

Following the results and data collected in the first site campaign, a thorough Root Cause Analysis was conducted by pump designer

Potential Root Cause Analysis 1)

POSSIBLE CAUSE	Why yes	Why not	Result
Mechanical behaviour of the pump	High level of vibration is due to the mechanic of the pump (misalignment, unbalance,etc)	1. The spectra don't show evidence of the mechanical problem 2. Dismantling of pump A didn't highlight any issue	EXCLUDED
Major internal looseness Broken parts	Extreme bearing wear, internal looseness or broken parts can justify a low noise level like background in the spectra	1. Bearing when inspected, didn't show any major damage 2. Dismantling of pump A didn't highlight any major looseness	EXCLUDED
Resonance	Resonance can justify a unstable phase	Resonance is centered on defined frequencies, and these frequencies are always the same. It's not compatible with the spectra variations measured	EXCLUDED

Potential Root Cause Analysis 2)

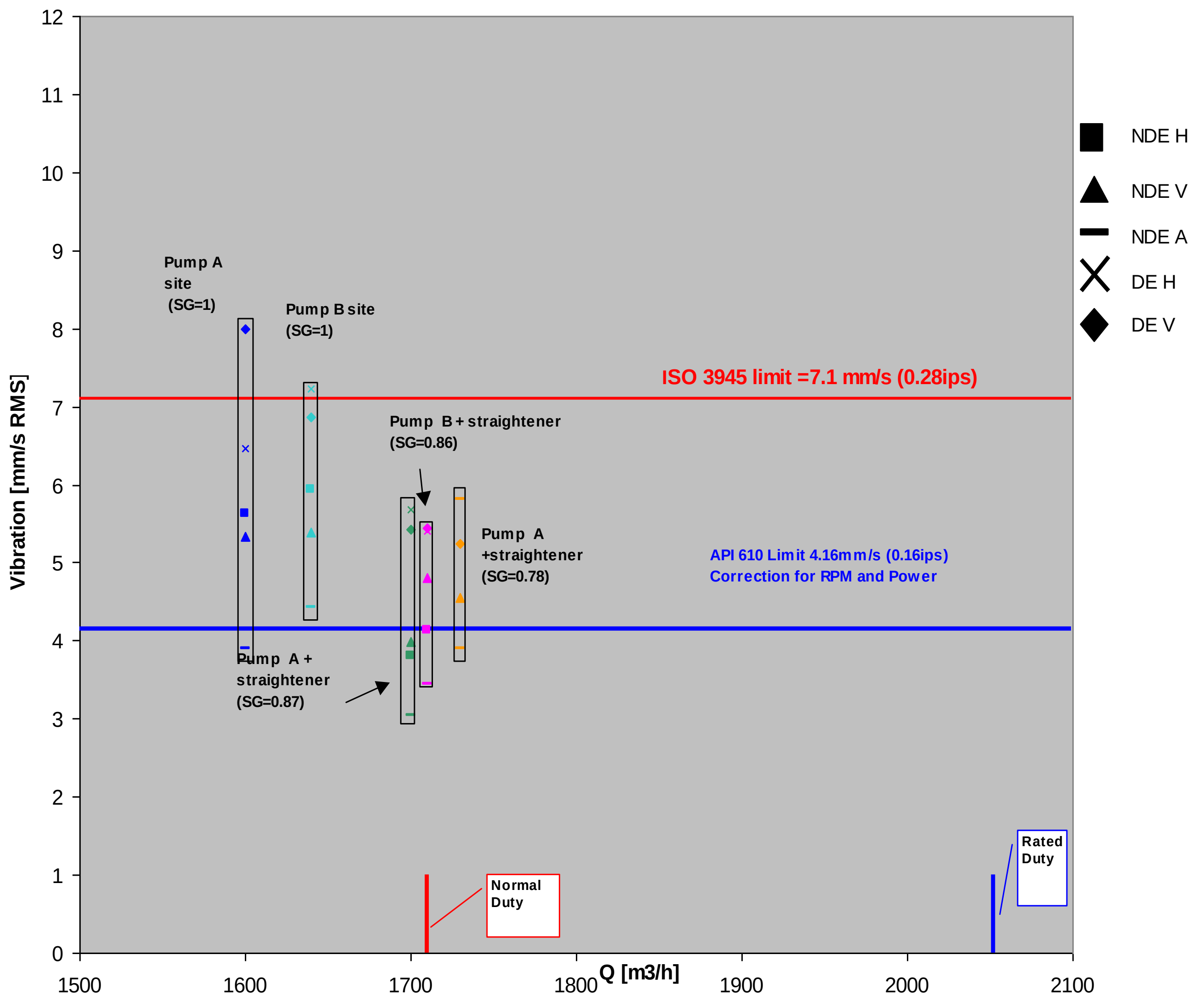
POSSIBLE CAUSE	Why yes	Why not	Result
Fluid dynamics of the piping	Unsteady and random spectra with a broadband distribution of many peaks of low frequencies are indicative of intense turbulence. Piping was not fully compliant with HI recommendations	Piping designed according to customer best practice	PROBABLE CAUSE
Fluid dynamics of the pump	Unsteady and random spectra with a broadband distribution of many peaks of low frequencies are indicative of intense turbulence. Pump operation at capacity below BEP is potential source of high turbulence	Same type of pump running well in other applications	PROBABLE CAUSE

Implementation of 1st phase:Suction piping

- The customer modified the piping layout as to have it compliant to Hydraulic Institute recommendation.
- The results of the modification showed visible reduction in vibration level, though not within the required acceptance limits.

Implementation of 1st phase

Suction piping with flow straightener (February-March 2009)



The results of the modification showed visible reduction in vibration level, though not within the required acceptance limits.

Root Cause Analysis – 2nd phase
Pump hydraulic design

- The solution has been focused on the pump hydraulic, as the remaining cause pointed out in the Root Cause Analysis
- The hydraulic design of the pump was studied with respect to the vibration analysis

Implementation of 2nd phase

General considerations

1) The peculiarity of broadband frequency spectra with presence of many peaks up to 500 – 700 Hz could be associated with turbulent flow induced by flow separation inside the impeller either at inlet (suction recirculation) and/or at outlet (discharge recirculation).

2) Vibration amplitude at VPF is in general a symptom more related with discharge recirculation which appears unlikely (low VPF level in all spectra).

Suction recirculation looks as the most probable mechanism of high turbulence and vibration source. Therefore, the focus has to be directed to:

a) Pump operation: if and how much below BEP and /or
b) Impeller design: if suitable for the application (primarily inlet geometry)

Implementation of 2nd phase

Keywords

***Recirculation*:**

For a trimmed impeller, the onset of suction recirculation may be closer to the normal point, even if this looks at first glance reasonable and complying with the API criteria.

***Incidence angle at blade tip*:**

An incidence angle far away from the shock-less condition may lead to flow separation with flow unsteadiness inducing vibrations.

For pumps with high energy level at inlet - peripheral velocity at the impeller eye diameter above 35 m/s (115 ft/s) - the overall level of vibrations can be high even above acceptable limits for the bearing housings.

Implementation of 2nd phase

Hydraulic analysis

Design point:

N = 3580 rpm

Q = 2900 m^3/h (12775 gpm)

H = 418 m (1373 ft)

D_2 = 490 mm (19.3 inch) max dia

NSPHR= 28 m (92 ft)

Ns_{des}= 1794

Nss_{des}= 9630 (reasonable)

Z = 7 vanes, staggered

Dcw/D2 = 1.08 (B-Gap)

D_{eye}= 280 mm (11 inch)

U_{eye}= 52.5 m/s (172.4 ft/s) (moderate)

Qsl = 3190 m3/h (14053 gpm)

Qsl/Qdes = 1.1 (sl = shockless)

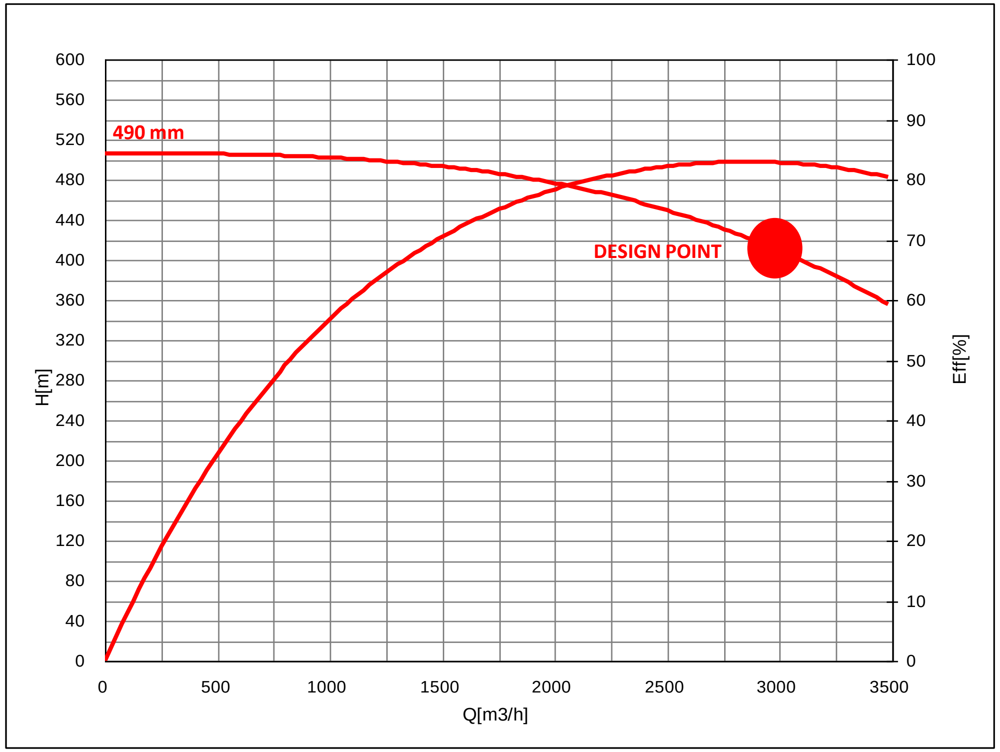
Qsr = 2090 m3/h (9207 gpm)

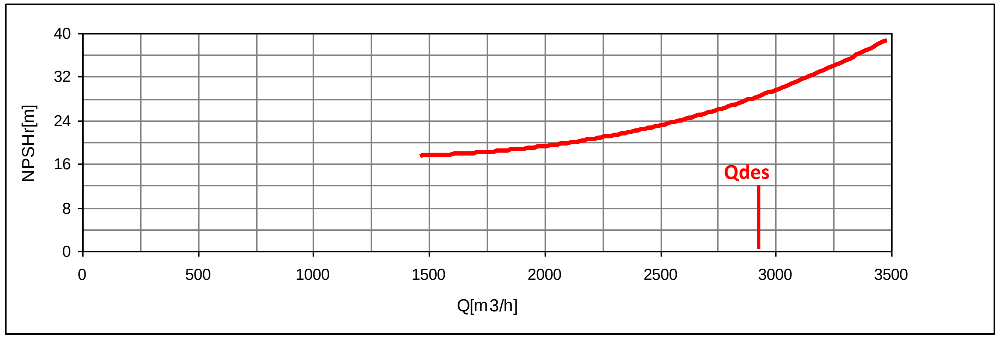
(sr = suction recirculation)

Qsr/Qdes = 0.72

Qrs/Qsl = 0.65

Test Curves



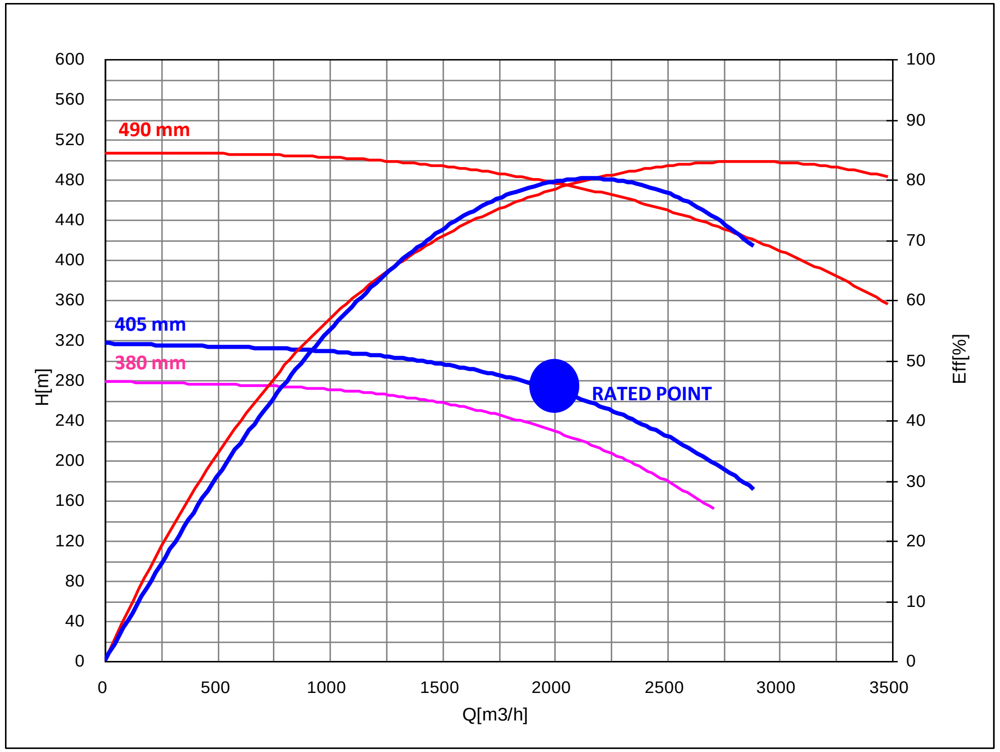


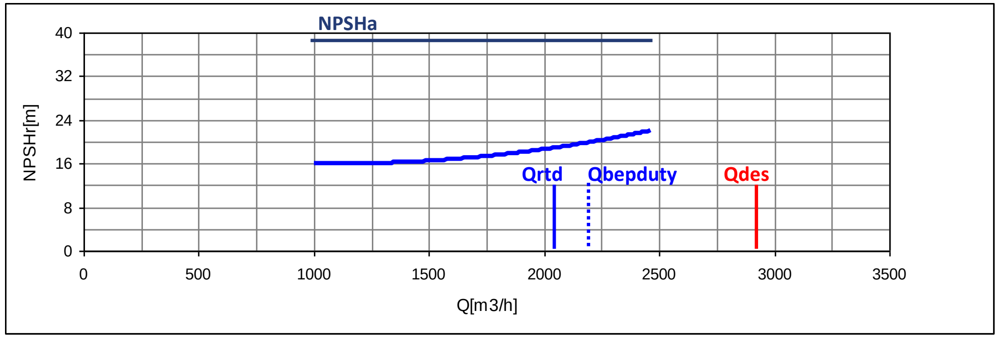
Implementation of 2nd phase
Impeller trimming

Rated point:

N = 3580 rpm
Q = 2052 m^3/h (9040 gpm)
H = 259 m (850.6 ft)
$D2_{duty}$ = 405 mm (15.9 inch)
NPSHR = 19.4 m (63.7 ft)
NPSHA = 39.6 m (130 ft)
NPSHA / NPSHR = 2.04
D2duty/ D2des = 0.83
$Q_{bepduty}$= 2150 m^3/h (9471gpm)
$Q_{rated}/Q_{bepduty}$= 0.95 (Looks good !)
Q_{rated}/Q_{design}= 0.74 (Too low)
Q_{rated}/Q_{sl} = 0.64 (Too low)
Q_{rated}/Q_{sr} = 0.98 (Possibility of suction recirculation start)
D_{cw}/D_{2duty}= 1.30 (B-Gap : very large i.e . low vibrations at VPF)

Test Curves





Implementation of 2nd phase

Impeller trimming

Normal point (specified):

N = 3580 rpm

Q = 1710 m^3/h (7533 gpm)

H = 283 m (929.4 ft)

D_{2duty}= 405 mm (15.9 inch)

NPSHR = 18.5 m (60.7 ft)

NPSHA = 39.6 m (130 ft)

NPSHA / NPSHR = 2.14

Q_{normal}/Q_{rated}= 0.83

$Q_{normal}/Q_{bepduty}$= 0.8 (OK for API 610)

Q_{normal}/Q_{bepdes} = 0.59 (Too low)

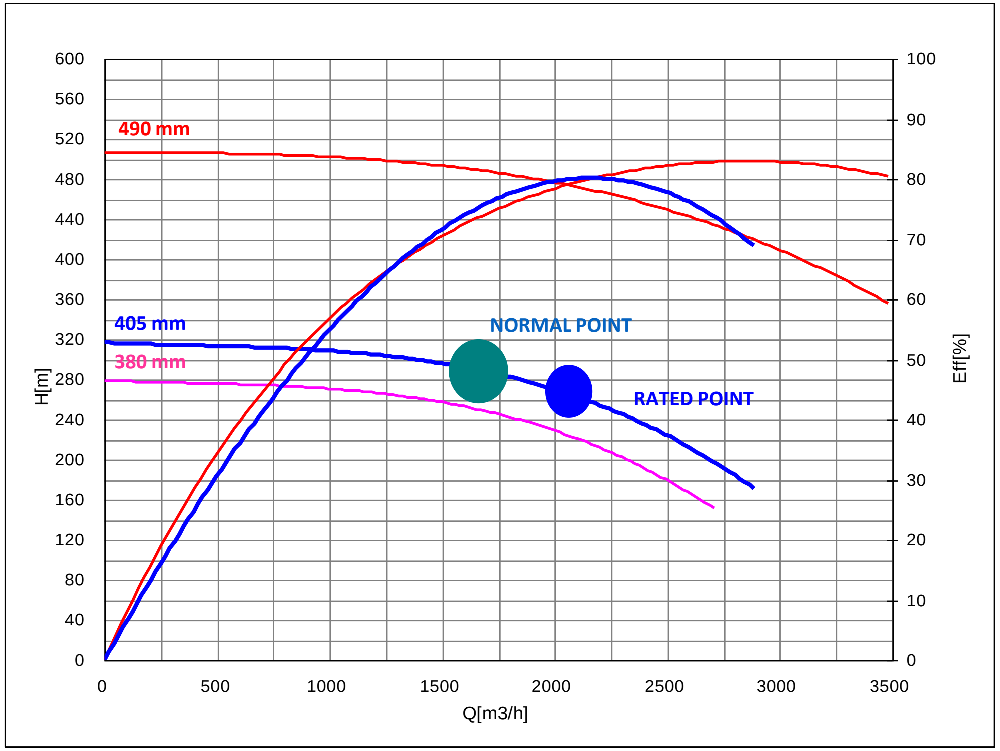
Q_{normal}/Q_{sl} = 0.54 (Too low)

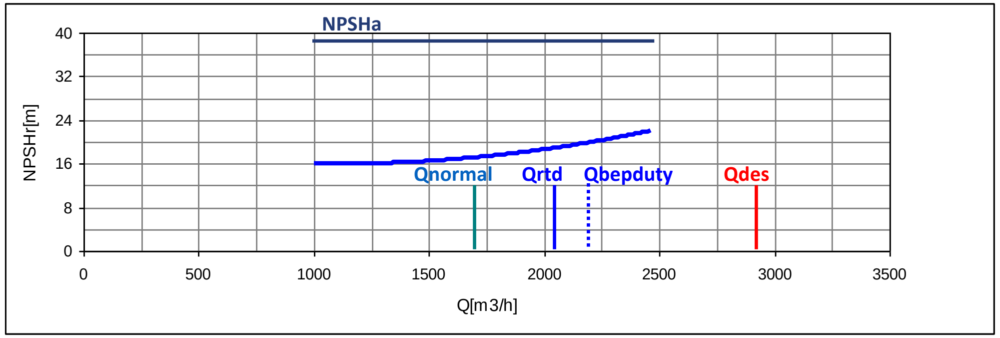
Qn = 59% of Qdes – RED FLAG

Qnormal / Qsr = 0.82

Suction recirculation is root cause of vibrations

Test Curves



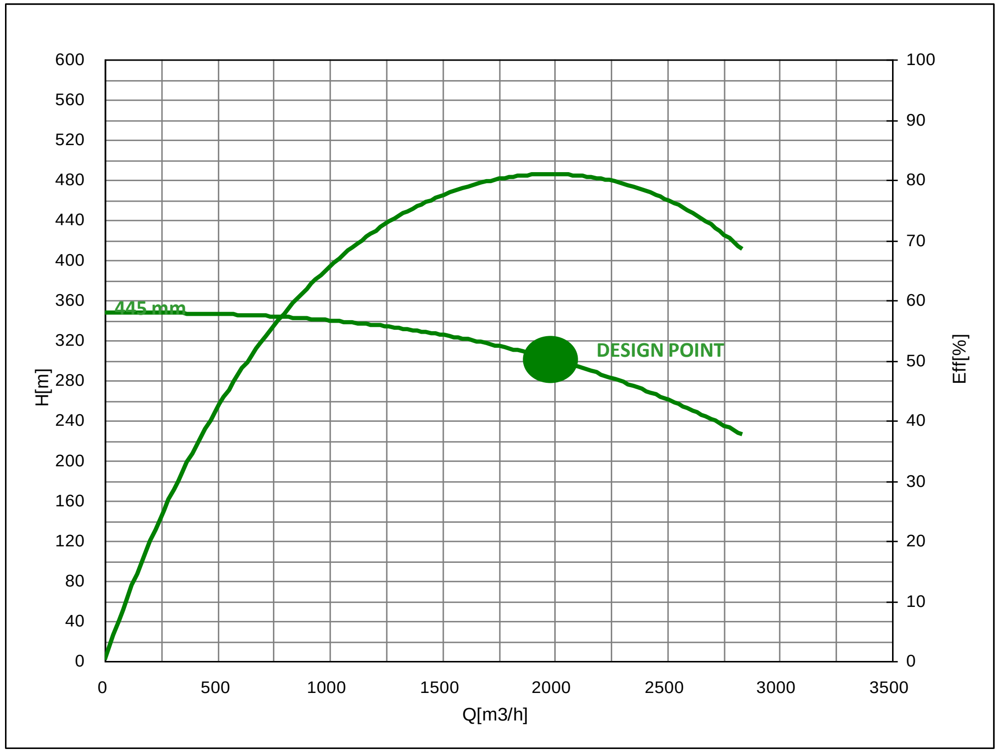


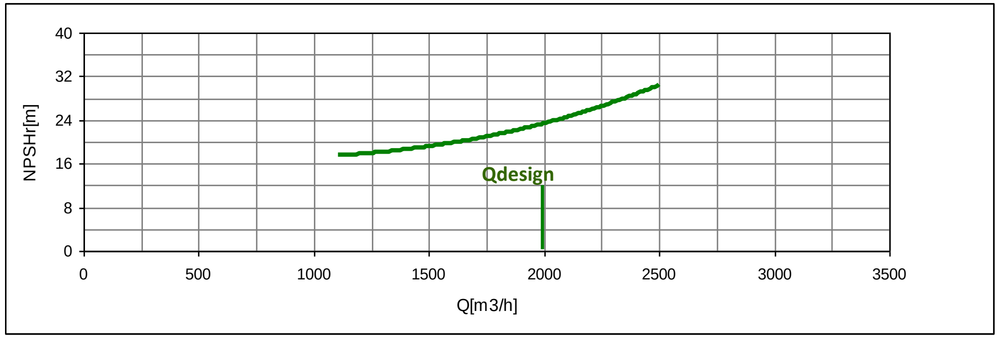
Upgraded impeller design

New Impeller design point:

N = 3580 rpm
Q = 2000 m^3/h (8810 gpm) close to rated
H = 300 m (985 ft)
D_2 = 445 mm (17.5 inch)
NSPHR= 23 m (75.5ft)
Ns_{des}= 1911
Nss_{des}= 9270 (reasonable)
Z=7 vanes, rake - no stagger
Dcw/D2=1.18 (B-Gap, ample)
D_{eye}= 255 mm (10 inch)
U_{eye}= 47.8 m/s (157 ft/s) (reduced)
Qsl = 2120 m3/h (9340 gpm)
Qsl/Qdes = 1.06 (sl = shockless)
Qsr = 1400 m3/h (6167 gpm)
(sr = suction recirculation)
Qsr/Qdes = 0.70
Qsr/Qsl = 0.66
Qsr/Qrated= 0.68 (<<1)

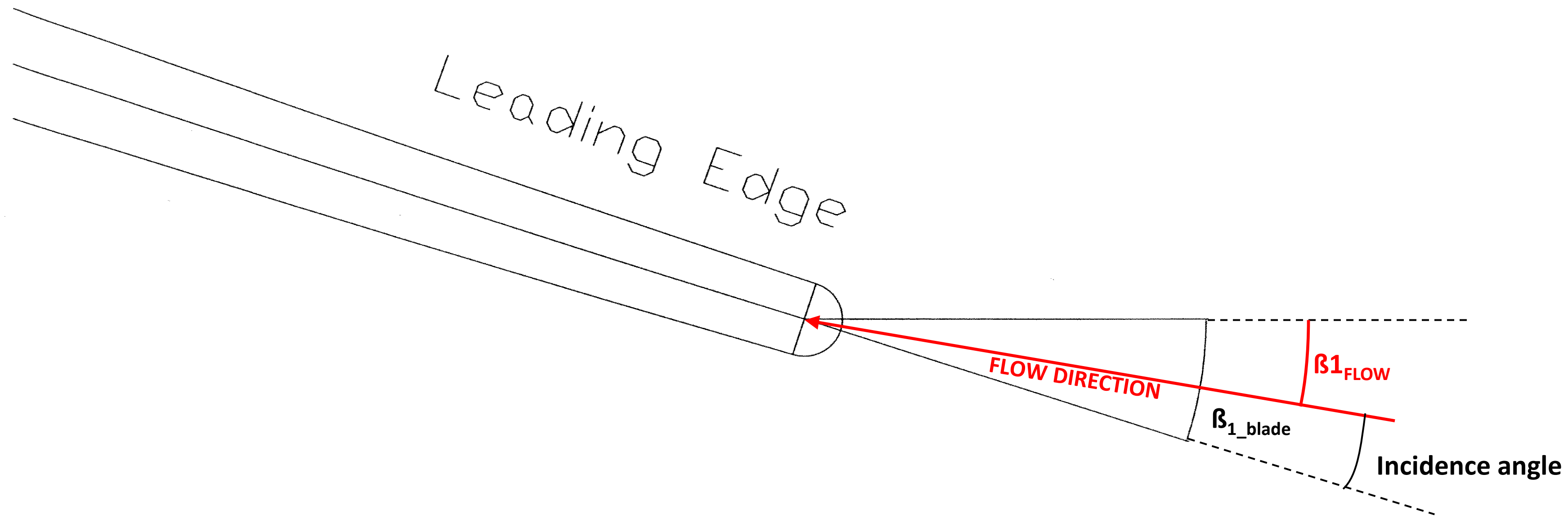
Expected Curves





Implementation of 2nd phase

- Incidence angle = $ß1_{blade}$ – $ß1_{FLOW}$



Implementation of 2nd phase

Hydraulic analysis (March 2009)

- Incidence analysis (*existing impeller*)

Point	Flow [m^3/h]	ß1_blade (tip)	ß1flow	INCIDENCE
DESIGN	2900	17°	15.5°	*1.5°*
RATED	2052	17°	10.2°	*6.8°*
NORMAL	1710	17°	8.4°	*8.6°*

<u>Could lead to suction recirculation (flow separation) with high level of broadband vibration for high energy pumps</u>

New impeller design strategy

Constrains:

1) Upgrade impeller design with new pattern

2) Stringent expected delivery time from Contractor and End User

Impeller to be interchangeable with present pump configuration, i.e. double suction, double volute, existing bearing housing

Upgraded impeller design (April 2009)

Incidence angle (design strategy for *new customized impeller*)

Point	Flow [m³/h]	ß1_blade (tip)	ß1flow(tip)	INCIDENCE
DESIGN	2000	16°	15.2°	*0.8°*
RATED	2052	16°	10.2°	*0.4°*
NORMAL	1710	16°	8.4°	*3.1°*

The incidence is near to the shockless condition for the rated capacity. Also, it is far below the critical value (causing flow separation and suction recirculation) for the normal point

Comparison upgraded vs original impeller

Upgraded impeller ("Customized design")

$N = 3580$ rpm

$D2_{duty}$ = 423 mm (16.7 inch)

$D2_{duty}/D2_{des}$ = 0.95

$D_{cw}/D2_{duty}$ = 1.25

$Q_{bepduty}$= 1900 m3/h (8370 gpm)

Rated point

NPSHR = 23.2 M (76.2 ft)

NPSHA / NPSHR = 1.70

$Q_{rated}/Q_{bepduty}$= 1.08 (Good)

Q_{rated}/Q_{design}= 1.03 (Good)

Q_{rated}/Q_{sl} = 0.97 (Good)

Q_{rated}/Q_{rs} = 1.46 (Well above suction recirculation onset)

Normal Point

NPSHR = 21 m (69 ft)

NPSHA / NPSHR = 1.89

$Q_{normal}/Q_{bepduty}$= 0.9 (Good for efficiency)

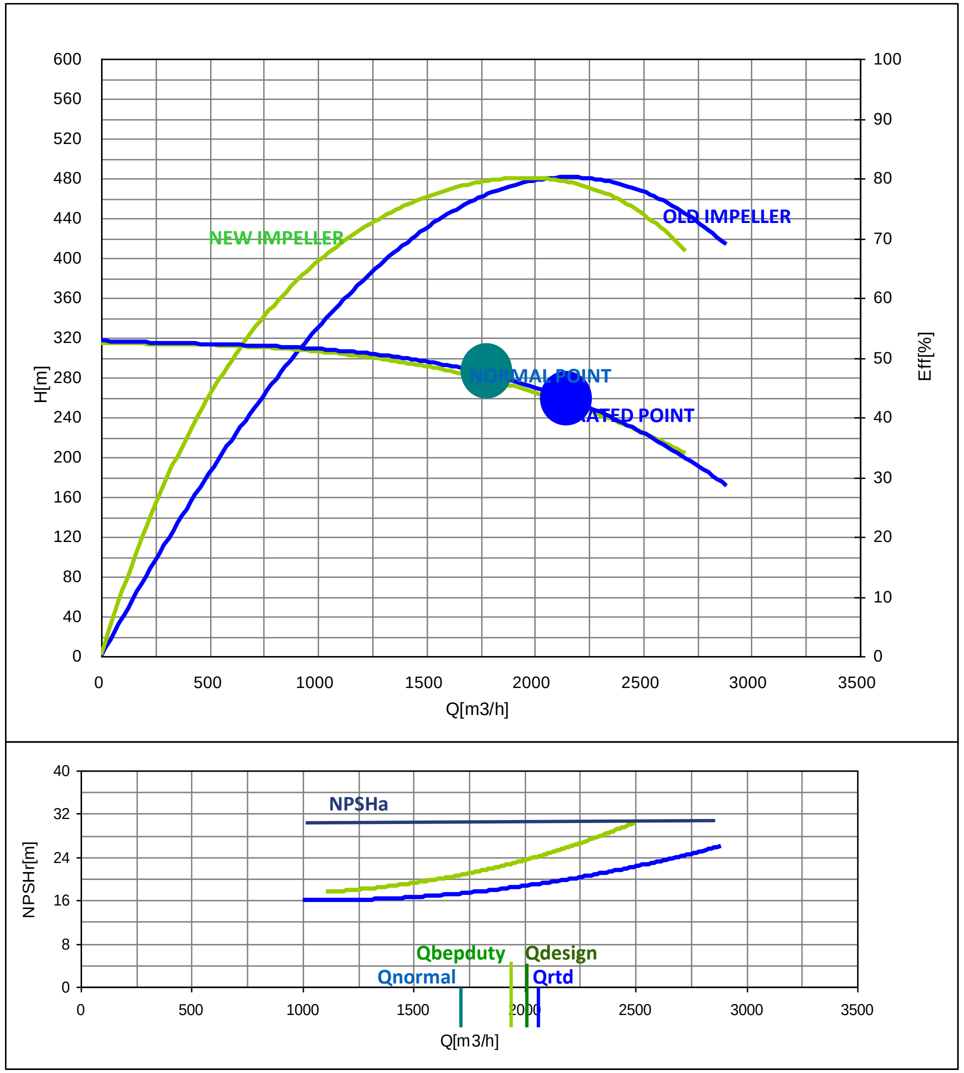
Q_{normal}/Q_{bepdes} = 0.85 (Reasonable)

Q_{normal}/Q_{sl} = 0.81 (Acceptable)

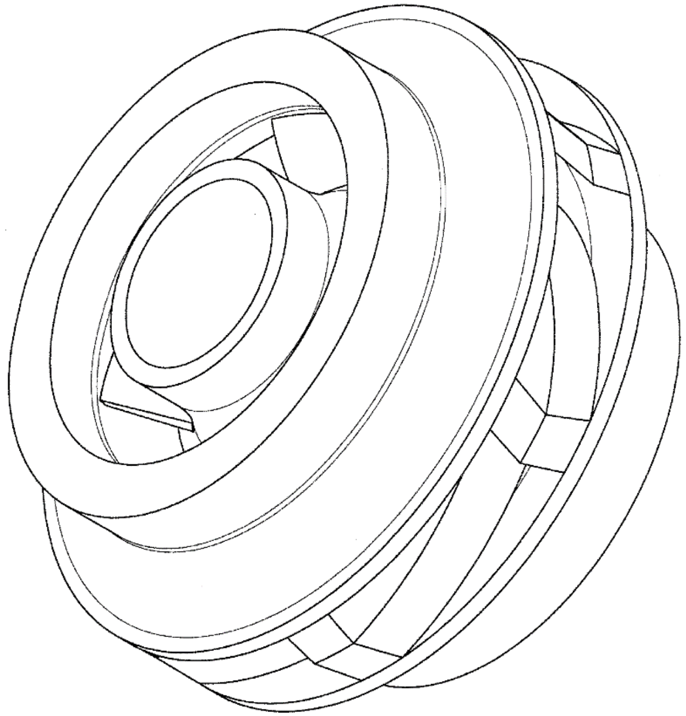
Qnormal / Qrs = 1.22

No suction recirculation

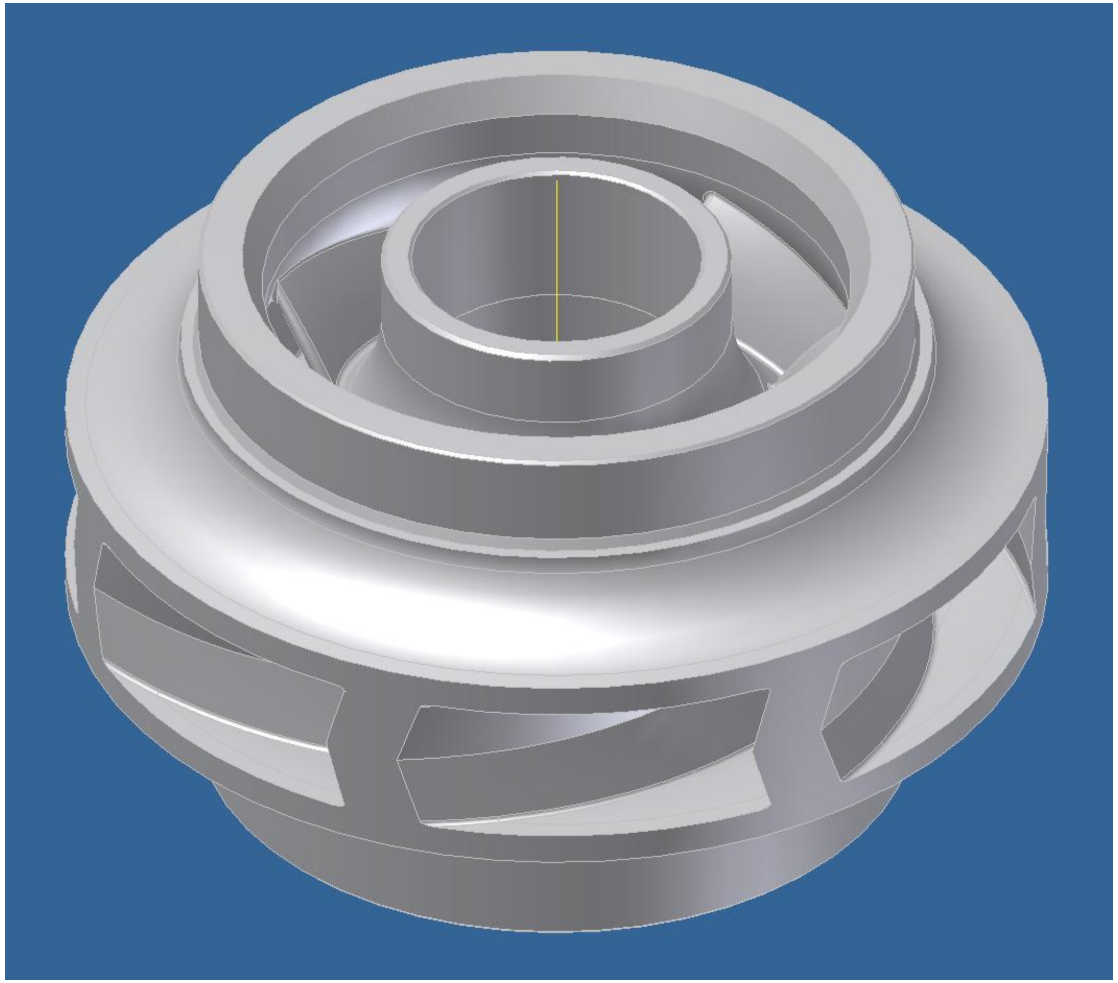
Expected versus Test Curves



Upgraded impeller design



Features: Blade rake – No stagger



3D Virtual solid model

Fast impeller production

The virtual solid model was post processed to obtain
all the pattern components through Rapid Prototyping
for fast production as required by Contractor and End User
to complete the plant commissioning and release to production

3D scanning for accurate casting inspection

Once the casting was obtained a 3D scanning of the impeller allowed the complete geometrical inspection to verify the compliance of the casting to the original design.

This step was needed because:

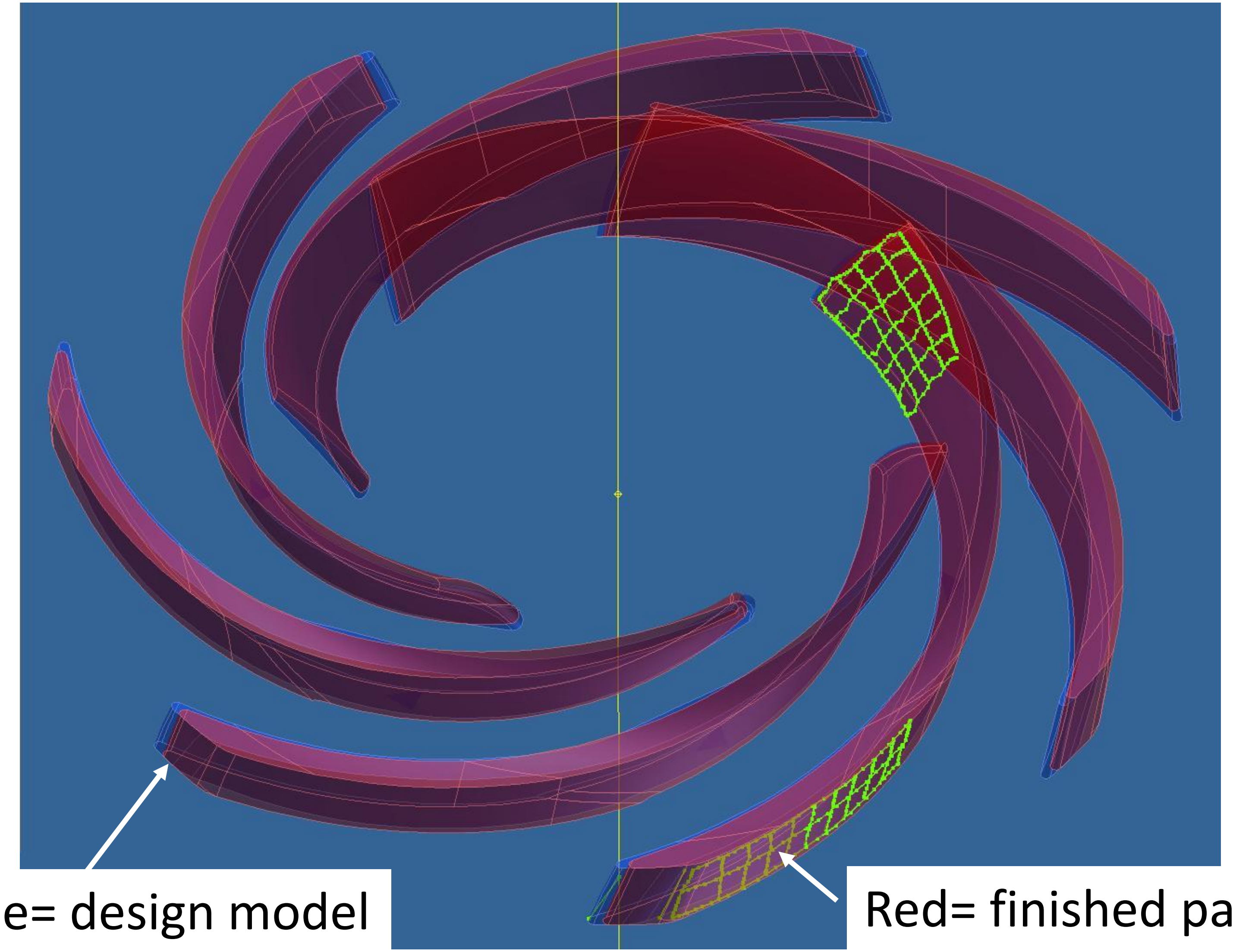
a) Incidence angle is very sensitive parameter.

In relation to suction recirculation onset and cavitation behaviour only tight tolerance for incidence and inlet blade angle is allowed (+/-0.5°)

b) The new impeller could not be tested at the shop.

The rotor had to be directly installed at site for quick plant restart, possibly avoiding any rework i.e. impeller outlet diameter to readjust head for any geometrical deviation (out of tolerance) at blade outlet (angle, span, thickness)

Shrouded Impeller blades are 3D scanned from casting (laser scan +point probe)



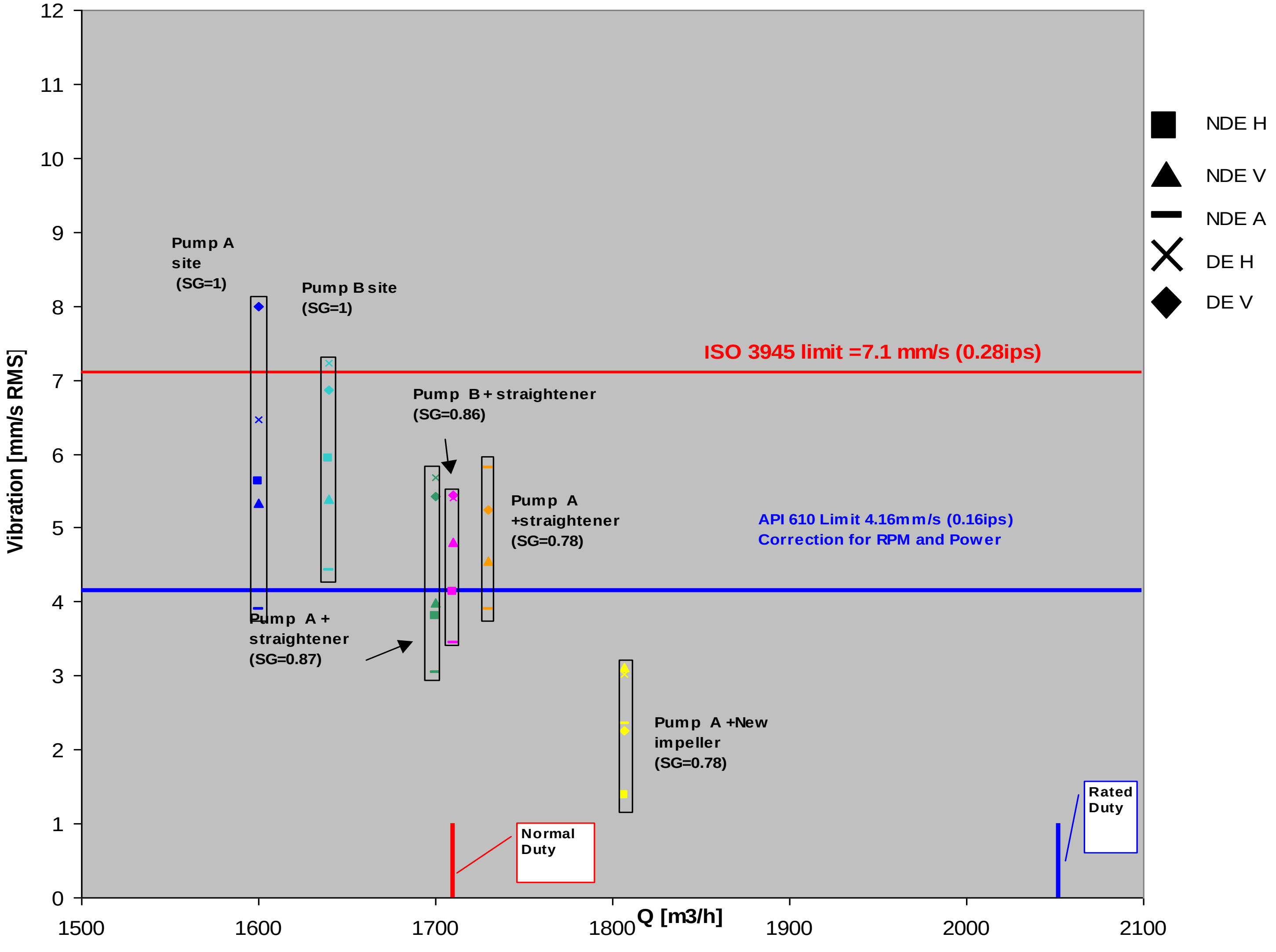
Blue= design model

Red= finished part

Machined impeller as shipped (June 2009)

Remark: No shop test

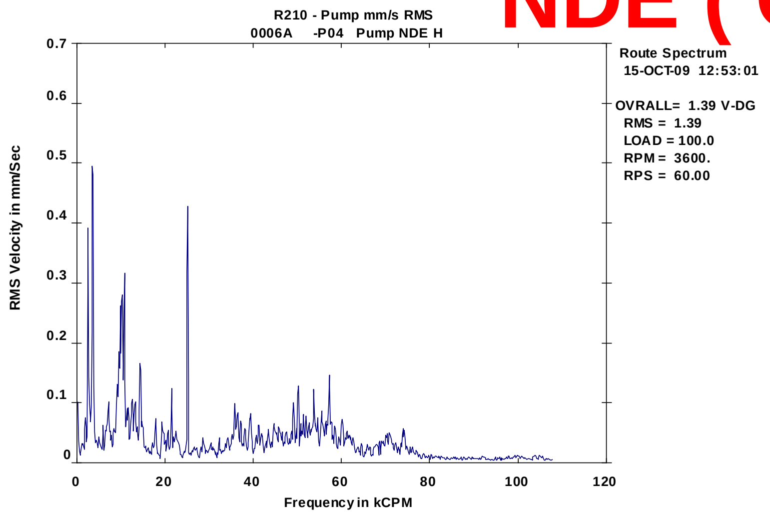
Preliminary field results pump A (July 2009)

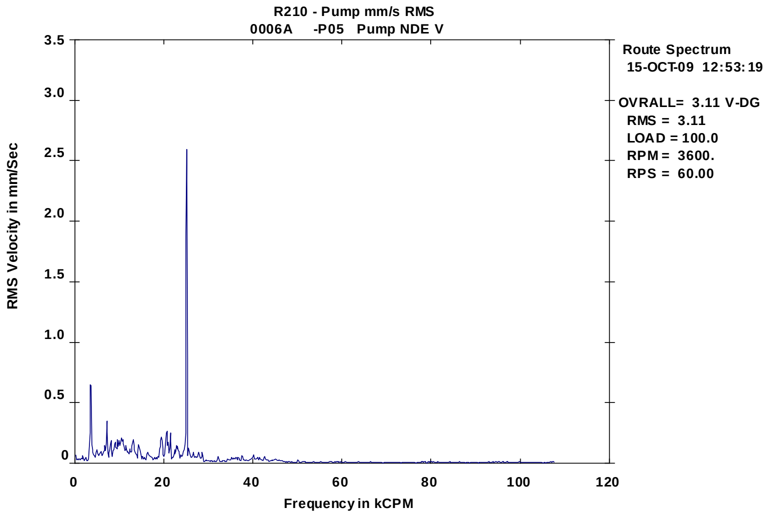


Pump accepted : August 2009

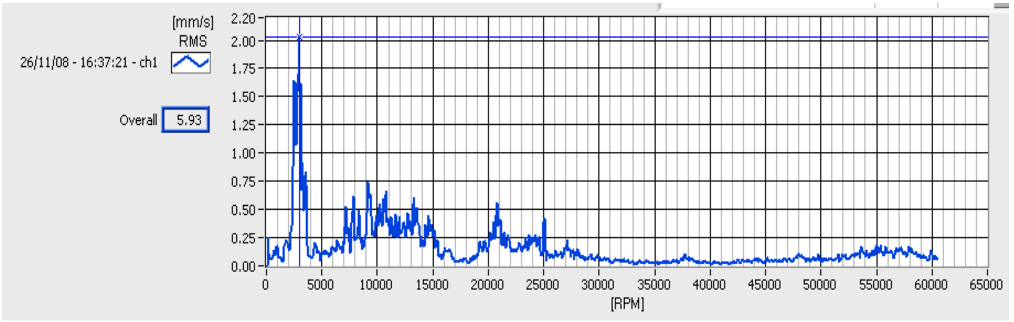
Plant released to full production : September 2009

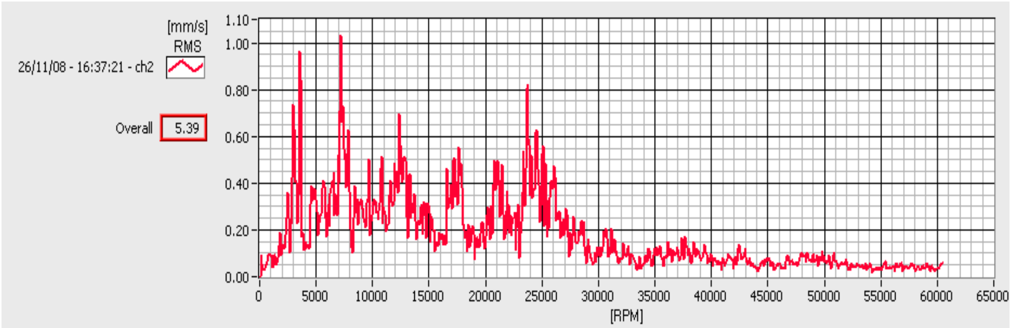
Field data comparison at normal capacity NDE (October 2009)





Final field data with new impeller

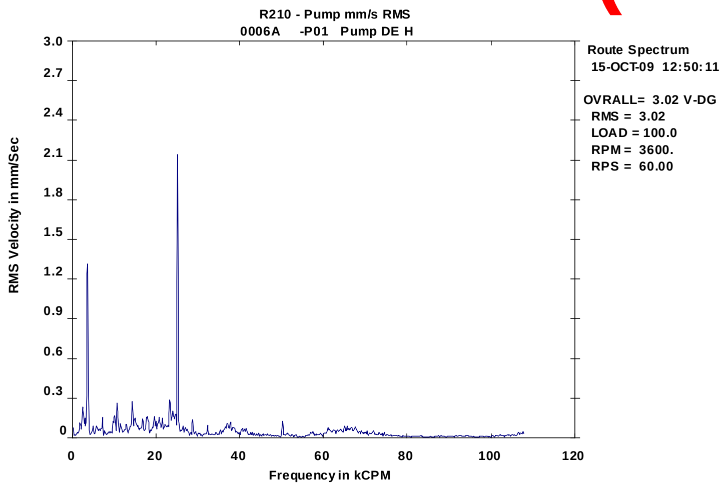


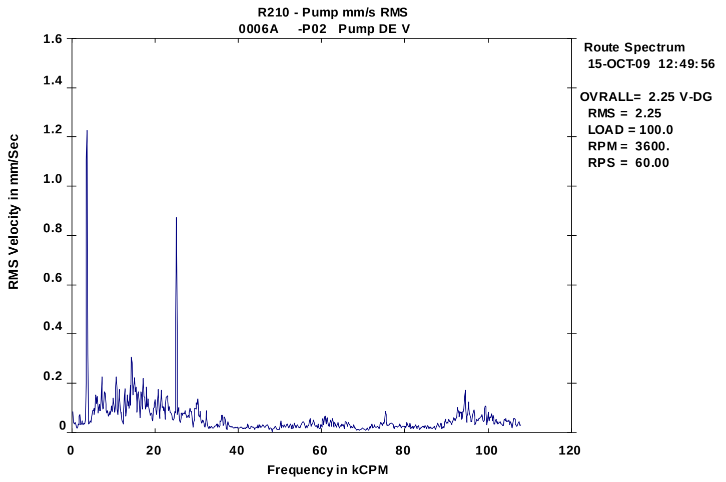


Initial field data with old impeller

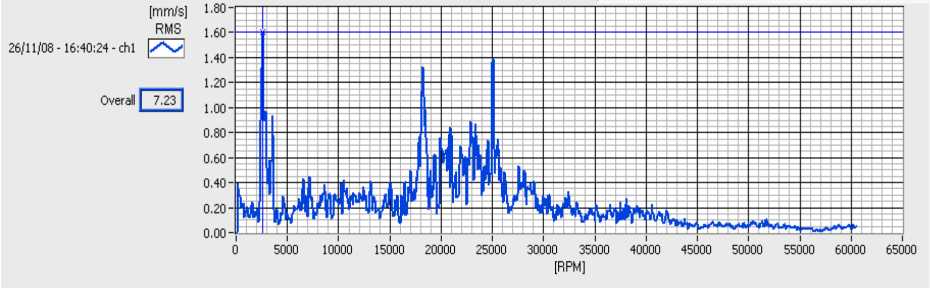
	Old	New
H mm/s(ips)	5.93 (0.23)	1.39 (0.05)
V mm/s(ips)	5.39 (0.21)	3.11 (0.12)

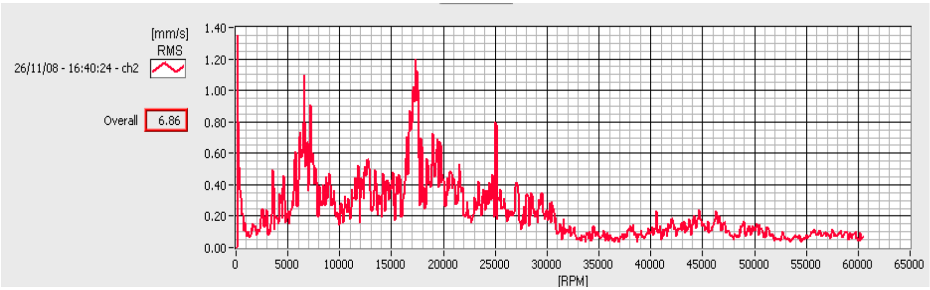
Field data comparison at normal capacity DE (October 2009)





Final field data with new impeller





Initial field data with old impeller

	Old	New
H mm/s(ips)	7.23 (0.28)	3.02 (0.12)
V mm/s(ips)	6.86 (0.27)	2.25 (0.09)

Conclusions

An analytical diagnostics approach has been applied along with experimental investigation for identifying the vibration root cause.
The vibration source was identified as mainly an internal hydraulic excitation due to high vane inlet angle not suitable for the expected operating range

A new impeller was designed with geometry fully optimized for the intended operating range , particularly the inlet geometry (customized design).

The new impellers were manufactured using a Rapid Prototyping process to meet customer impellent needs.

A 3D scanning protocol has been used to verify consistency of casting to the design and allow straight installation at site with minimal risk

The new impellers have been installed in the pumps and field data show a drastic reduction of all vibration components below API acceptance level with full satisfaction of Contractor and End User for ultimate solution of pump vibrations with fast field implementation allowing the start of plant production according schedule.