Vibration Field Problem Resolved with Analytical Diagnostic Approach and Innovative Impeller Design

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FLOWSERVE

Bruno Schiavello is Director for Fluid Dynamics at Flowserve, Advanced Technology, Pumps Department, in Bethlehem, Pennsylvania.
He started in 1975 with the R&D Department of Worthington Nord (Italy), joined in 1982 the Central R&D of Worthington, USA .

- Co-winner of the H. Worthington European Technical Award in 1979.
- Author of several papers and lecturer at various Seminars for pumps (suction recirculation, cavitation, two-phase flow) .
- Member of ASME (1984), and former Associate Editor for ASME JFE (1996-2002).
- Recipient of the ASME 2006 Fluid Machinery Design Award, 2006
- Co-Lead Organizer of ASME Pumping Machinery Symposium (2005 – 2015)
- Member of Pump Advisory Committee for the International Pump Users Symposium 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 Fluid Dynamics, Rhode St. Genese, Belgium.

Giancarlo Cicatelli is currently holding the position of Engineering and Technology 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.

His principal technical fields of interest are focused on the Fluid Dynamic of Turbomachines, for the design, development and testing of centrifugal pumps and pumping systems. He is author of several scientific publications in the field of Fluid Dynamics applied to turbines and pumps.

He has been member of the Middle East Turbomachinery Symposium Advisory committee and Europump Technical committee member. He is also member of the Organizing committee of the ASME Pumping Machinery Symposium (2009-2015)

ABSTRACT

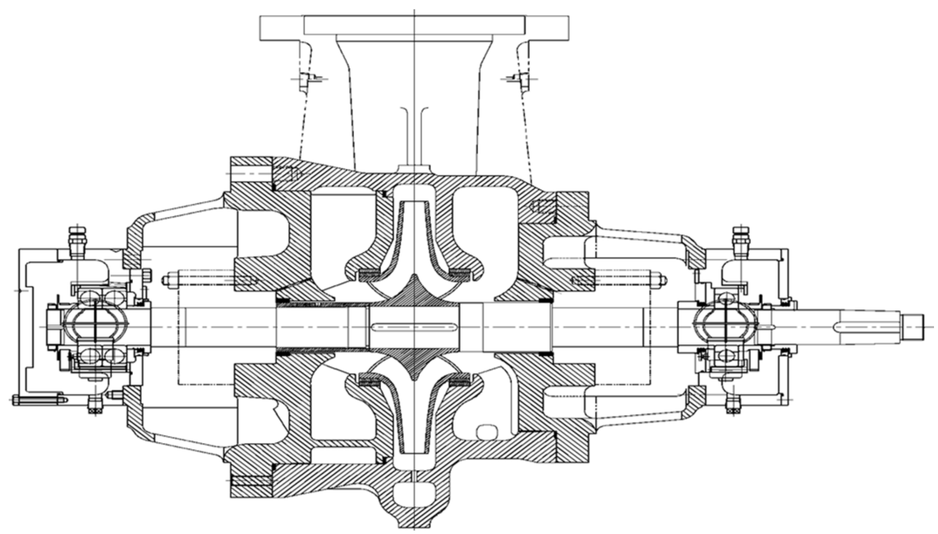
Several pumps of same design exhibited field vibrations above API limits, dominated by
Vane Passage Frequency (VPF).
Root cause analysis included both experimental and theoretical paths.
Experimental investigation , with shop vibration tests and modal analysis, showed
natural frequency of bearing housings at VPF.
The theoretical approach , based on mainly hydraulic analysis, pointed out to discharge recirculation
as primary cause of hydraulic excitation for high VPF vibrations.
The solution was identified with the design of an innovative impeller geometry (5+9 vanes).
Shop tests confirmed both overall performance and drastic reduction of vibrations below API limits .
Pumps with the innovative impeller were installed in the field confirming reduction of vibrations.

Lesson learned: high number of vanes at impeller outlet is a key feature for controlling hydraulic
excitation forces, changing both VPF and amplitude.

Case story

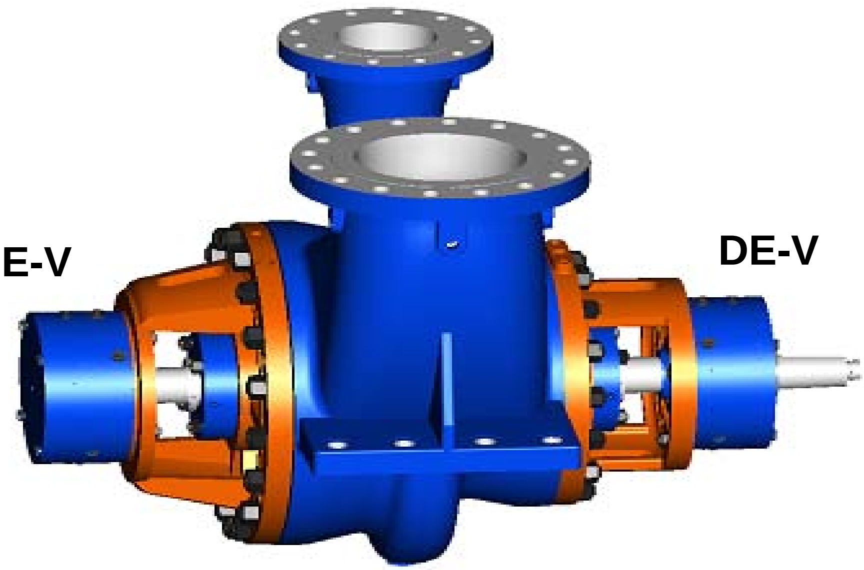
Several pump units of same design installed with different duties, exhibited high vibrations above API limits, from rated point down to minimum continuous flow with process fluids (SG=0.56 to 0.75)

The pump type/size is an API 6" discharge with top-top configuration,
double suction impeller,
double volute,
antifriction bearings configuration,
360° center-mounted (BB2)

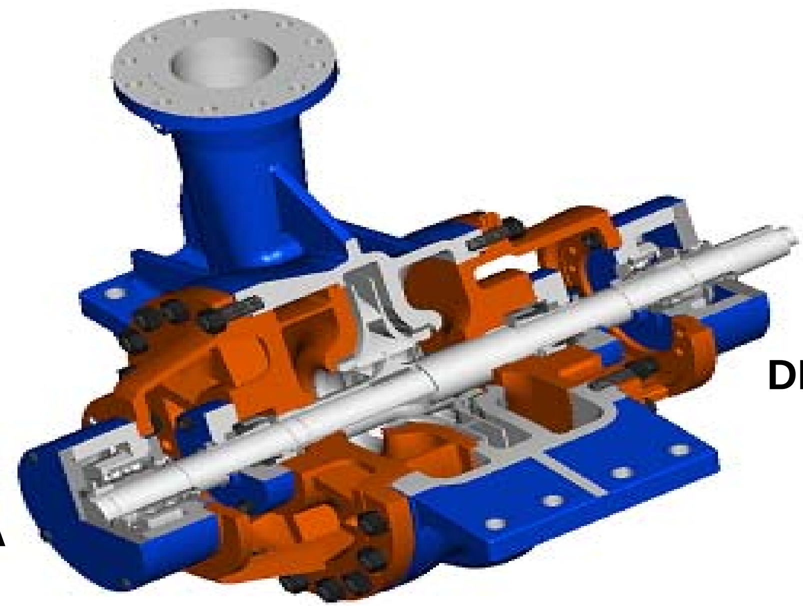


Cross-Section

Pump 3D Model

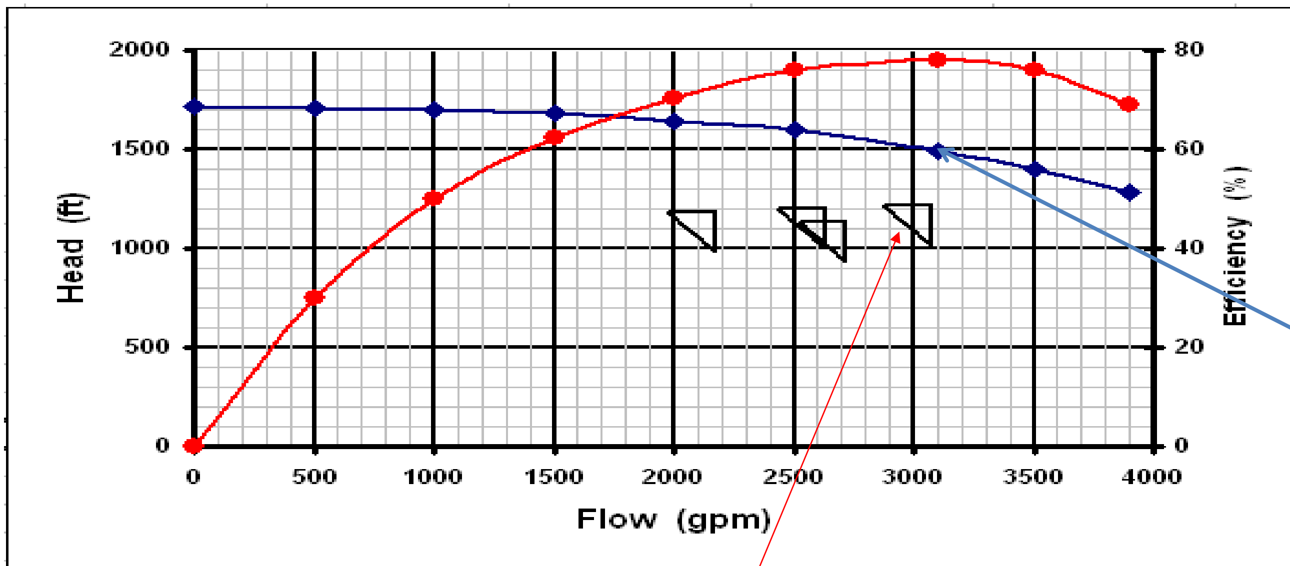


Pump Internal Cut



Bearing Housing – Vibrations Pickups

- **DE-H** **Drive End – Horizontal**
- **DE-V** **Drive End – Vertical**
- **NDE-H** **Non Drive End – Horizontal**
- **NDE-V** **Non Drive End – Vertical**
- **NDE-H** **Non Drive End – Axial**



Pump characteristics

Design conditions

N [rpm]	3565
Capcity Q [gpm]	3100
Head H [ft]	1490
NPSHr [ft]	30
Ns [US]	828
Nss [US]	10950
D2 [in]	18.5
D3 [in]	20.55

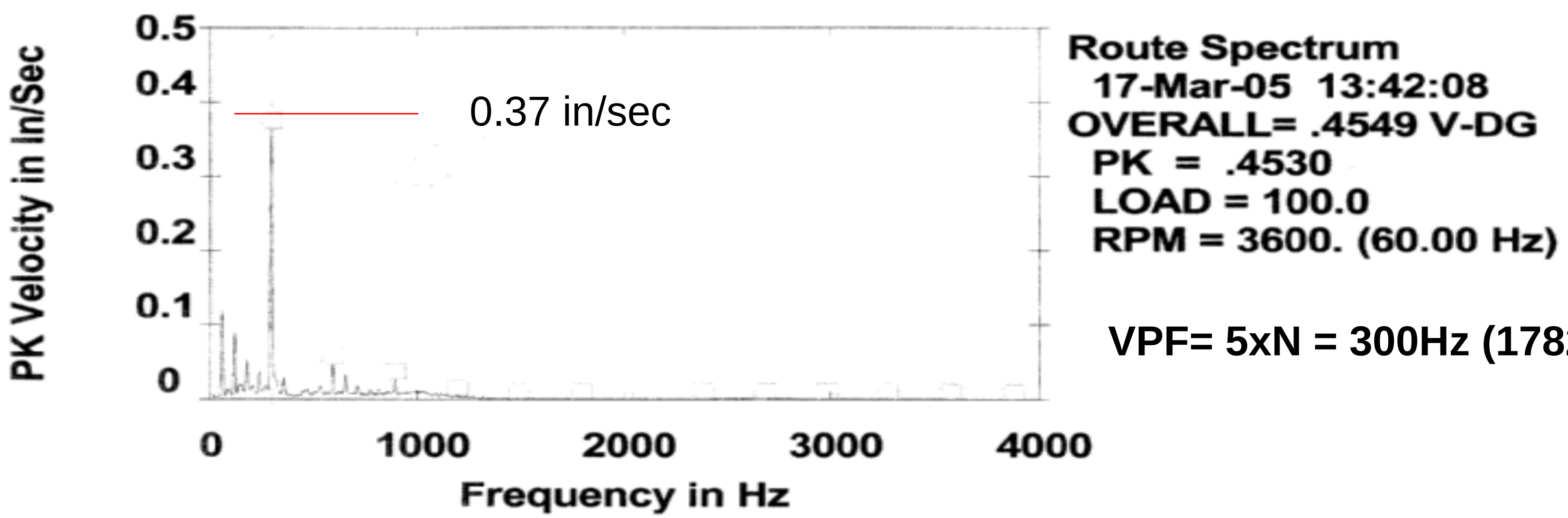
Rated points

Case N.	1	2	3	4
Q (gpm)	2720	3070	2176	2634
H (ft)	1136	1218	1181	1200
D2 (in)	16.73	17.72	16.93	17.32
SG	0.569	0.560	0.724	0.735
MCSF (gpm)	671	1000	917	942
NPSHr (ft)	24.3	25.0	19.2	23.4
NPSHa(ft)	39	35	30.6	52.4
N. of units	2	2	2	2

Vibration spectra at site - DEV

Qoper=2420gpm (78%Qrated)
(Case 2, SG=0.56, N=3565 rpm)

API acceptance criteria = 0.12 in/sec overall (RMS)



VPF= 5xN = 300Hz (17825cpm)

Key observations: CASE 2

ASPECTS

KEY POINT

1) Operating capacity:

$$Q_{rated} / Q_{BEPduty} = 1.03$$
$$Q_{oper} / Q_{rated} = 0.79$$
$$Q_{oper} / Q_{BEPduty} = 0.81$$
$$Q_{oper} / Q_{BEPdesign} = 0.78$$
(at max impeller diameter)

→ **Operating Capacity far below Design Capacity**

2) Margin of NPSHA/NPSHR

NPSHA/NPSHR=1.5

→ **Sufficient NPSH margin**

Key observations: CASE 2

ASPECTS

KEY POINT

3) Frequency activity:
   a) Most up to 1000Hz
   b) Dominant VPF (300Hz)
   c) Absence above 1000Hz up 4000Hz

High suspect of suction recirculation
High suspect of discharge recirculation
Absence or marginal cavitation

4) VPF like spike

Possible structural response of bearing housing

Insights from Background Literature

The above vibration aspects are in principle associated to:

1) suction recirculation

Pressure pulsations with a broadband spectrum

2) discharge recirculation

A sharp increase of the pressure pulsations at VPF
The intensity is increasing with further reduction of flow capacity

3)strong impeller–volute interaction

Pressure pulsations inducing vibration at VPF and increasing with flow reduction

4) structural response of bearing housing at 3565 rpm (mechanical aspect)

Structural vibration modes of the bearing housing are an essential factor in determining the level of vibration in presence of an internal exciting mechanism

Root Cause Investigation

Experimental investigation

Shop tests
(with an "as built" pump)

Modal analysis

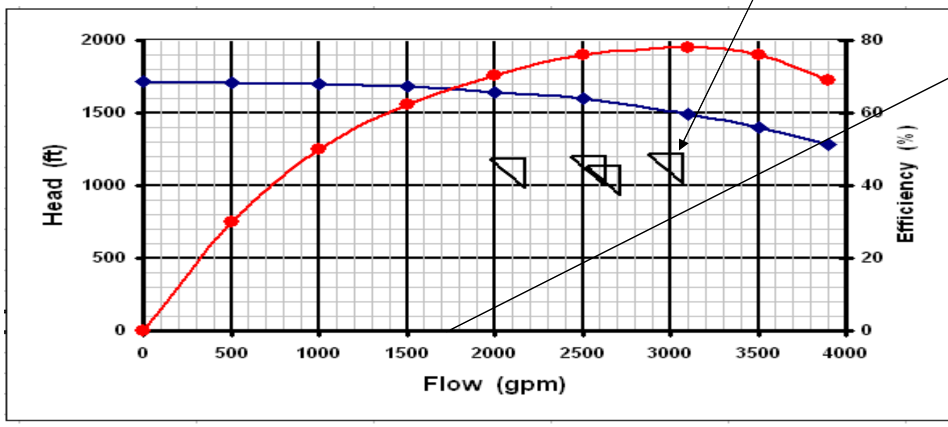
Analytical investigation

Background data

Hydraulic analysis

Shop tests

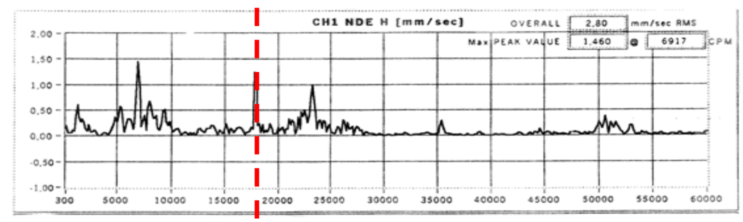
Case 2



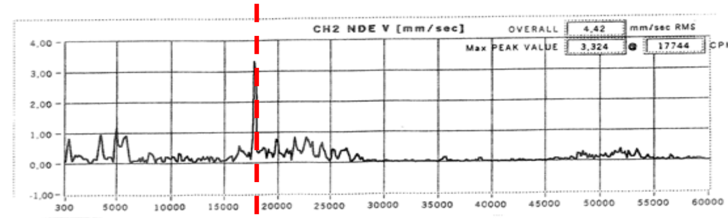
Shop test vibration spectra at 1750 GPM 57%of Qbep-design (PK Velocity In/Sec)

N=3565rpm SG=1.0

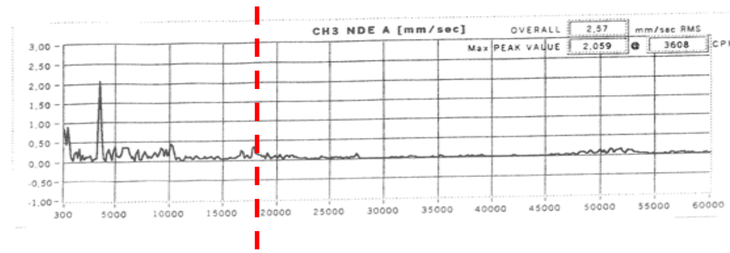
17,825 cpm = VPF (5X)



NDE-H

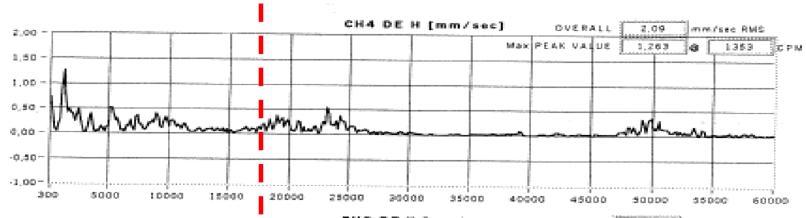


NDE-V

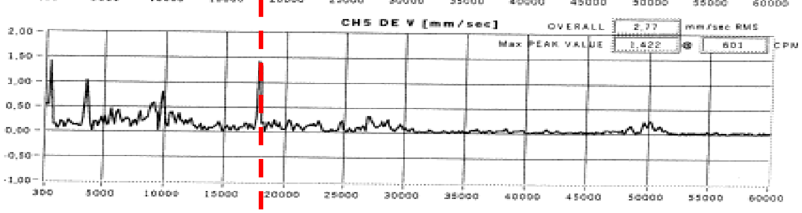


NDE-A

17,825 cpm = VPF (5X)



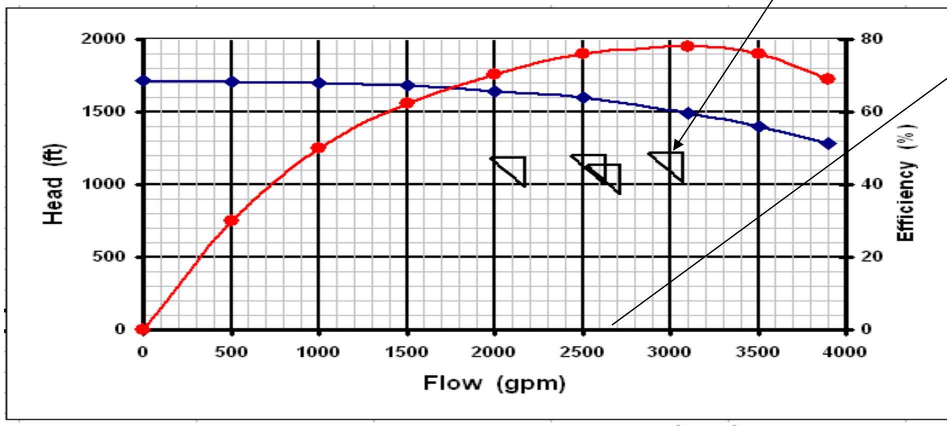
DE-H



DE-V

Shop tests

Case 2



Shop test vibration spectra at 2690 GPM 87% of Qbep-design (PK Velocity In/Sec)

N=3565rpm SG=1.0

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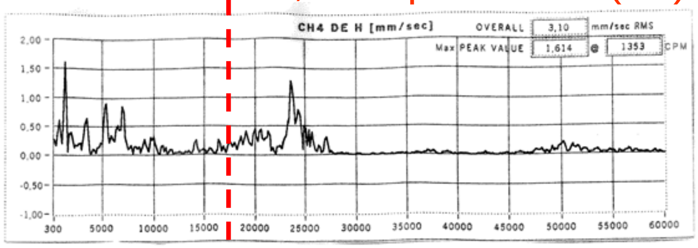
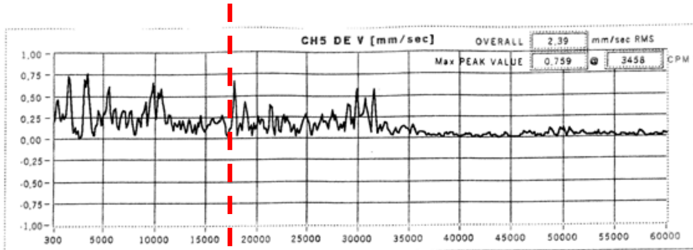
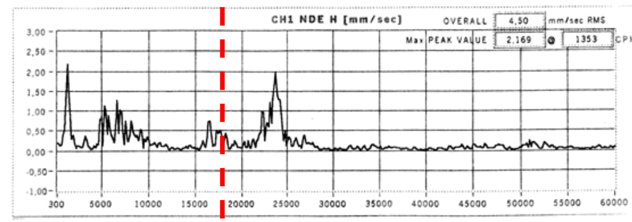


DE-H

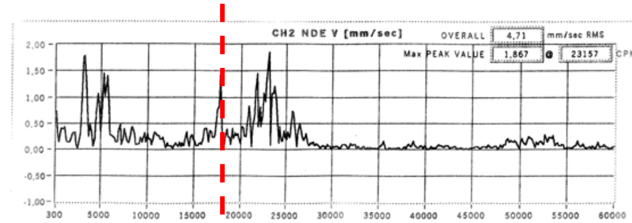


DE-V

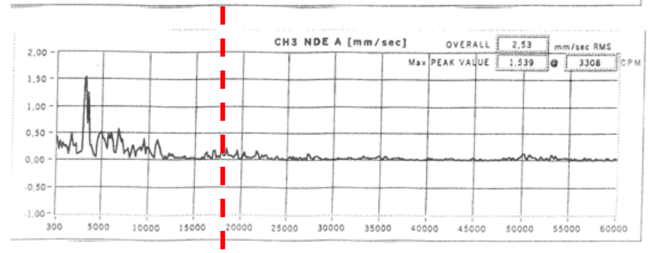
17,825 cpm = VPF (5X)



NDE-H



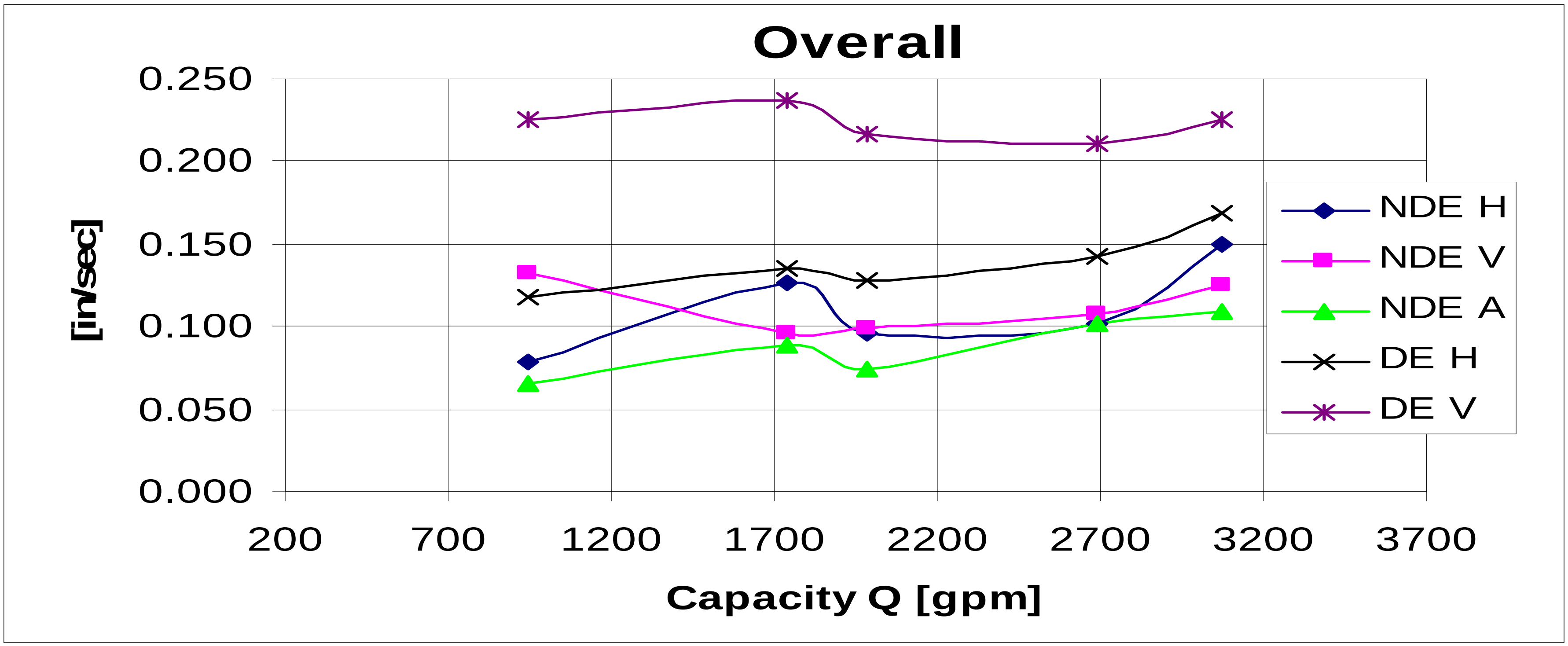
NDE-V



NDE-A

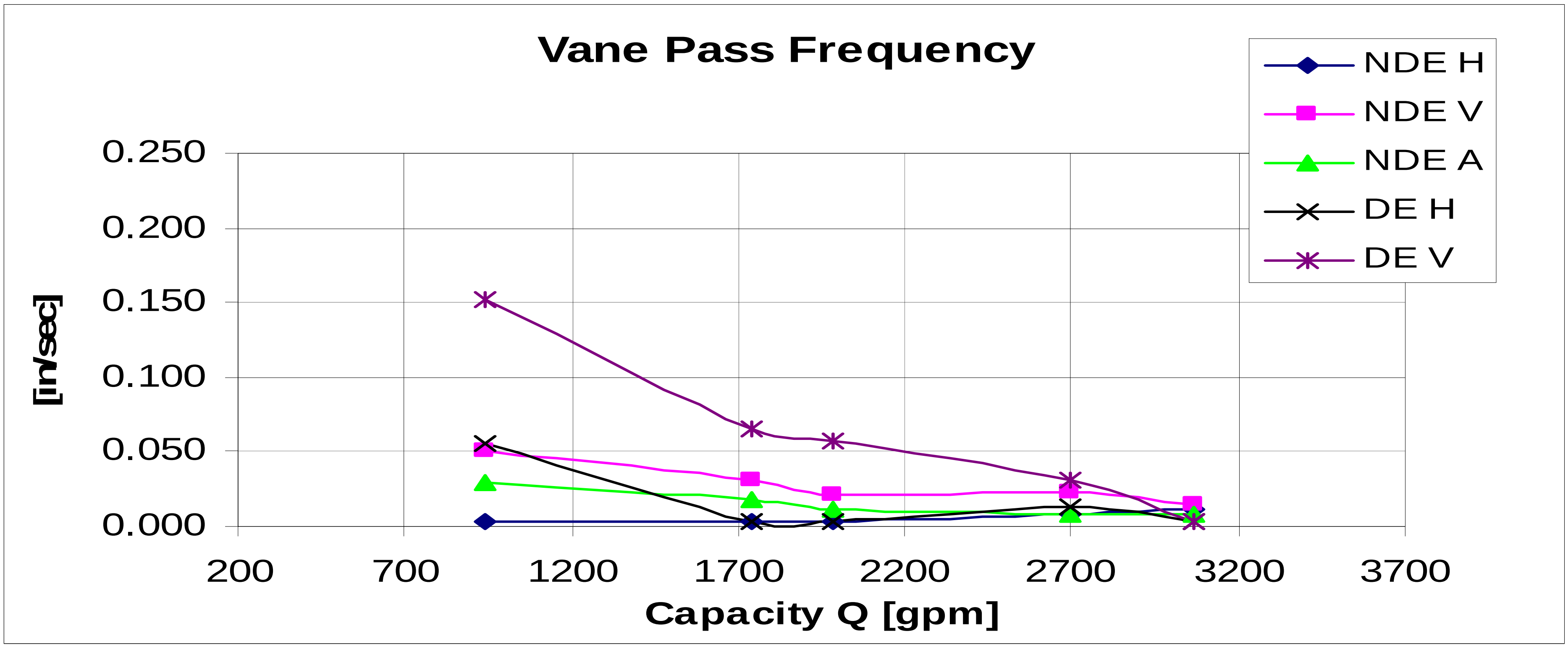
Original impeller

Shop test vibration spectra
Velocity PK In/Sec
N=3565rpm SG=1.0



Original impeller

Shop test vibration spectra
Velocity PK In/Sec
N=3565rpm SG=1.0



Modal analysis(Hammer test)

ORIGINAL BEARINGS HOUSINGS

IMPACT	Natural frequency [Hz]	
HORIZONTAL	DE	NDE
DE	290	-
	423	-
NDE	-	320
	-	477
VERTICAL	DE	NDE
DE	250	-
	468	-
NDE	-	291
	-	490

MODIFIED BEARINGS HOUSINGS –
ADDED MASS : 65 LB –OB & 55 LB –IB

IMPACT	Natural frequency [Hz]	
HORIZONTAL	DE	NDE
DE	262-300	-
	361-391	-
NDE	-	285
	-	405
VERTICAL	DE	NDE
DE	234	-
	420	-
NDE	-	286
	-	488

Pump speed = 3560cpm = 60Hz

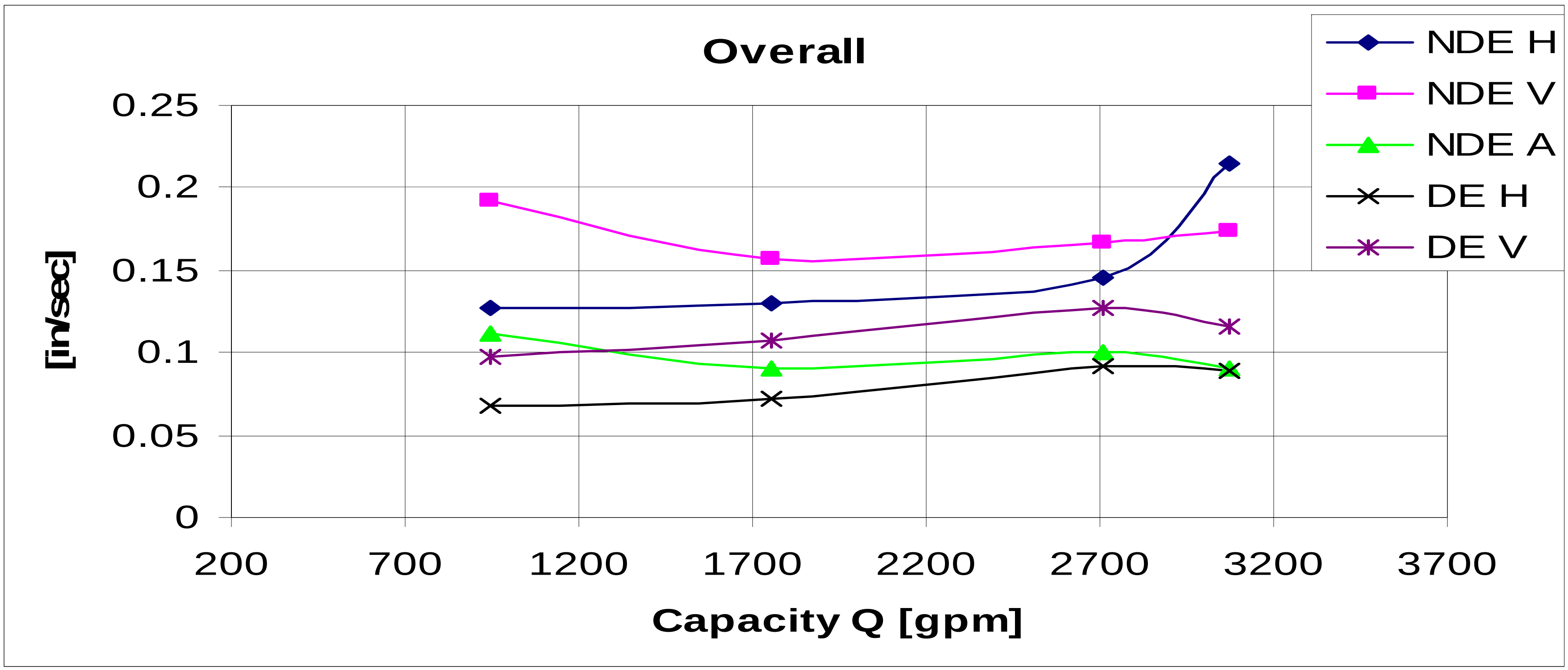
VPF = 17,825cpm = 297Hz

Modified pump (Mass added)

Shop test vibration results

Velocity PK In/Sec

N=3565rpm SG=1.0

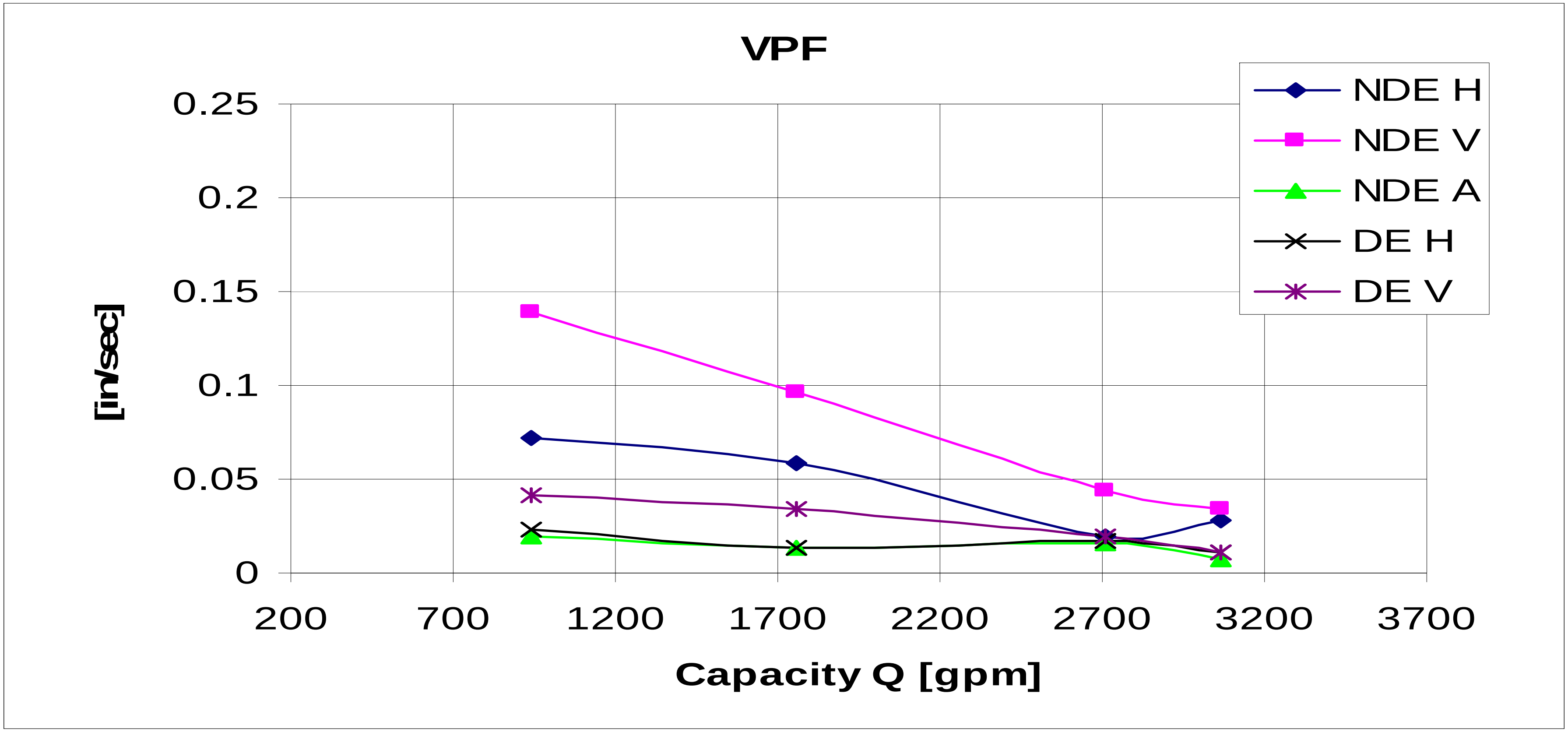


Modified pump (Mass added)

Shop test vibration results

Velocity PK In/Sec

N=3565rpm SG=1.0



Key observations

Critical vibrational aspects:

a) High vane pass frequency

b) Increasing with flow reduction: Flow dependent

c) Bearing housing responding to vane pass

d) Added mass not effective

Key point: Vibration source is internal hydraulic excitation at vane pass frequency (Forced vibrations)

Root Cause Investigation
Hydraulic analysis

Analytical design approach

1) Off design evaluation

2) Impeller - Volute matching capacity

3) Onset determination of suction and
   discharge recirculation

4) Cutwater- impeller diameter ratio (Gap B)

Root Cause Investigation
Hydraulic analysis

Results

	CASE 1	CASE 2	CASE 3	CASE 4
Qrated /Qbep-max	0.90	1.02	0.72	0.87
Qrated / Qbep-duty	1.00	1.06	0.8	0.87
Qrated / Qrs (*)	1.10	1.24	0.87	1.06
Qrated / Qrd (*)	0.96	1.03	0.77	0.90
D3 /D2duty	1.22	1.16	1.21	1.18

Qrs= suction recirculation onset capacity (Fraser 1981)
Qrd= discharge recirculation onset capacity (" ")

Key conclusions:
Discharge recirculation is the hydraulic triggering mechanism of vibrations
Discharge recirculation related with the impeller design
Volute interaction negligible (Large B-Gap)

New impeller design strategy

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

A) Delay the discharge recirculation onset capacity below lowest rated flow for all cases (Rank A)
B) Reduce the discharge recirculation strength (energy level of the exciting hydraulic forces or dynamic radial loads (Rank A)
C) Change the impeller vane number to reduce vane loading, pressure pulsations and move away from possible natural frequency of bearing housing (increased number of vanes) (Rank A)
D) Reduce the suction recirculation onset capacity and intensity (Rank A)
E) Keep or increase volute cutwater-impeller clearance: i.e. keep or reduce impeller duty diameter (Rank A)
F) Keep rated flow, head and NPSHR (Rank A)
G) Maintain efficiency at rated flow close to contractual value within maximum negative deviation of 1 point (Rank A/B)
H) Revise and marginally increase MCSF for each Case thus redefining the lower end of the operating range within API vibrations limit (Rank B

Impeller design features

An existing patented design (double row of vanes) has been cosidered. The key features are:

A) Inlet geometry: number and shape of vanes:

Optimized to meet suction conditions design target (Low number of blades)

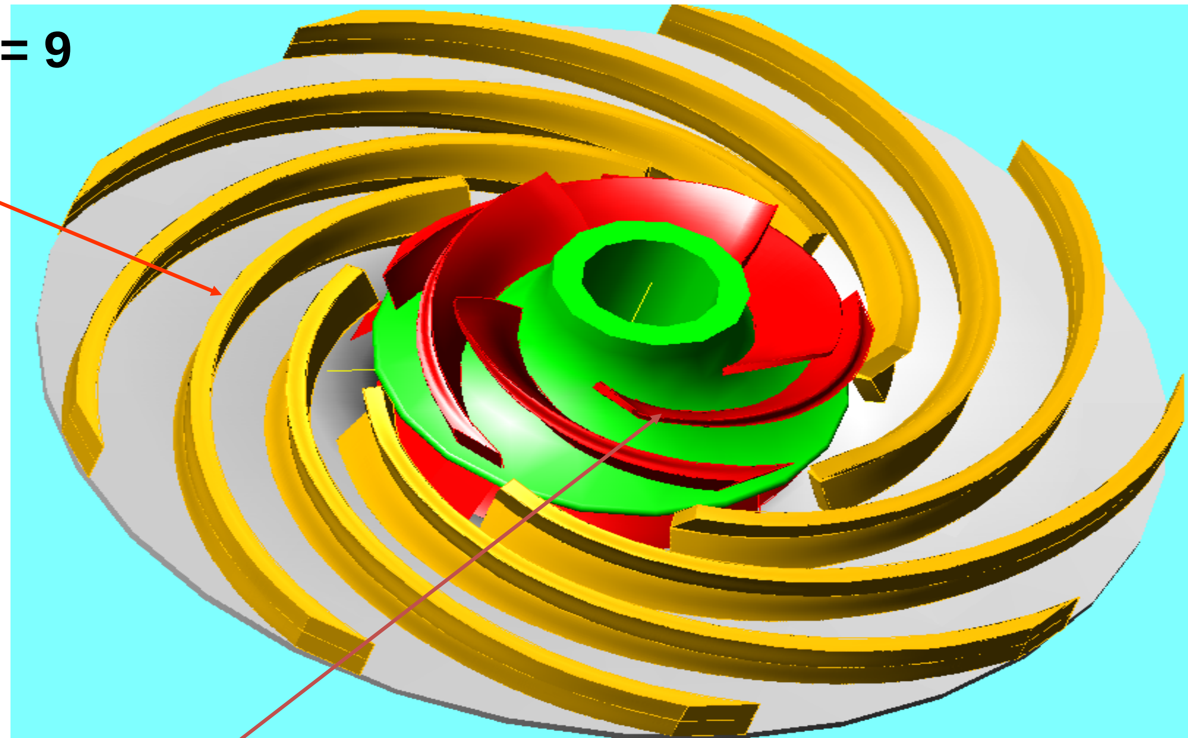
B) Outlet geometry: number and shape of vanes:

Optimized for reduction of hydraulic excitation strength
(high number of vanes and appropriate shape)
Compatible efficiency design target

The above features have been optimized for the specific requirements following the design priorities (Rank A/B)

New impeller design

Discharge impeller vanes= 9



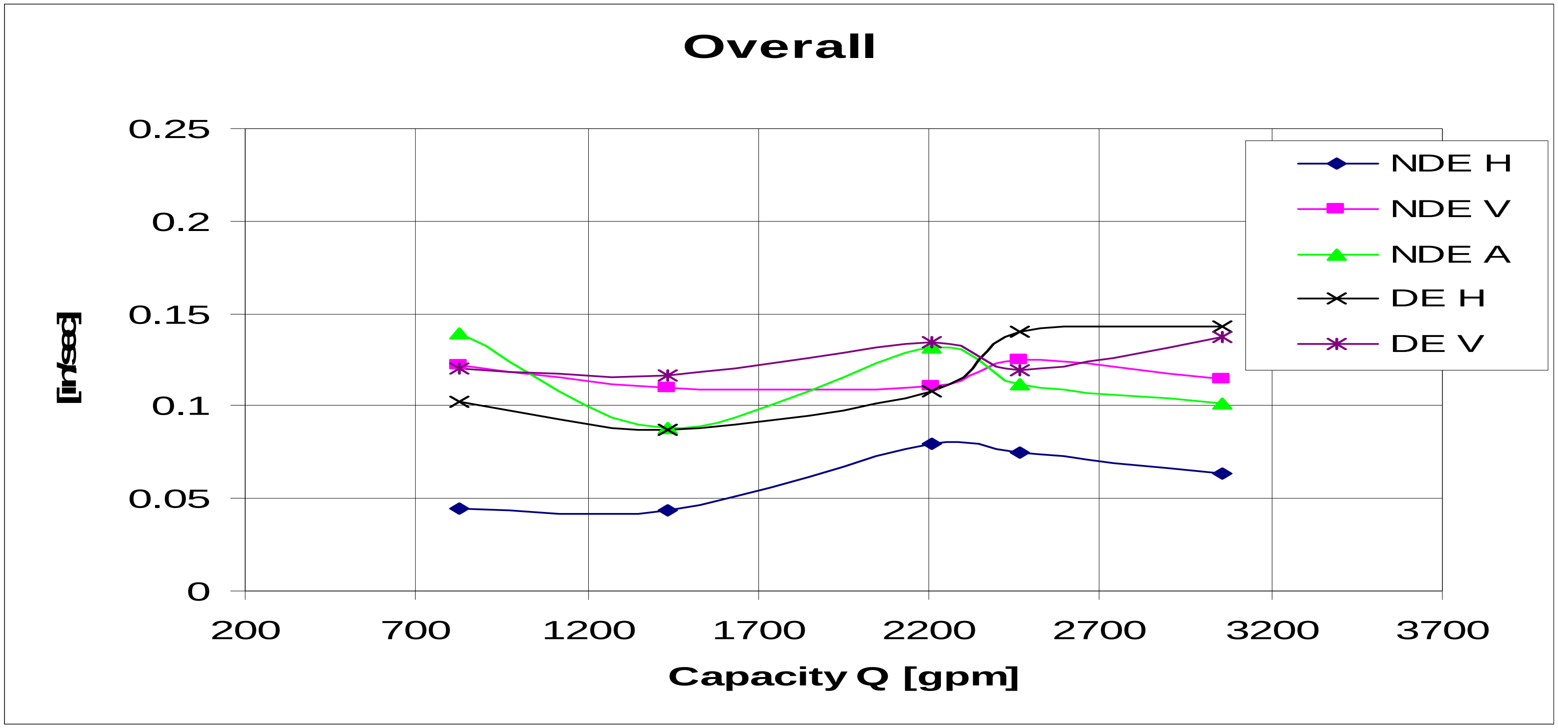
Suction impeller vanes= 5

New Impeller Design

Shop test vibration results

Velocity PK In/Sec

N=3565rpm SG=1.0

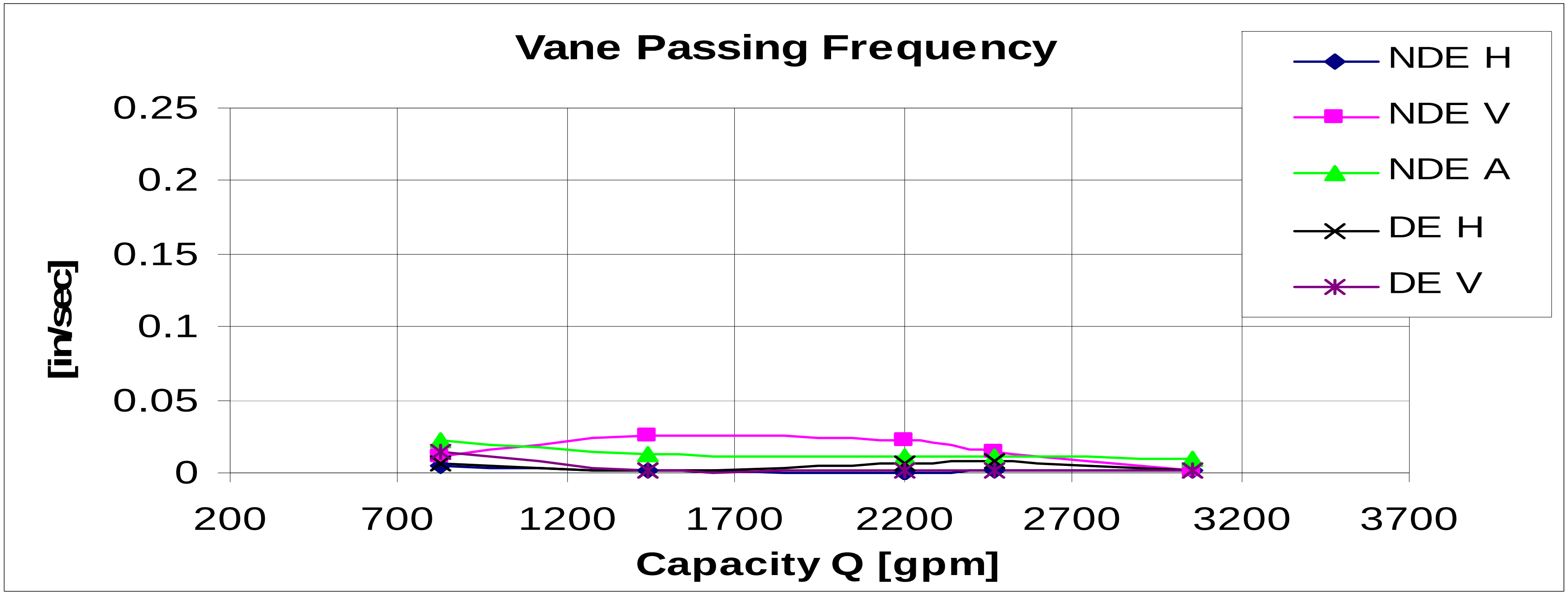


Key remark :

Overall Vibration reduced by 50% and close to API limits of 0.12 IPS even with SG = 1.0

New Impeller Design

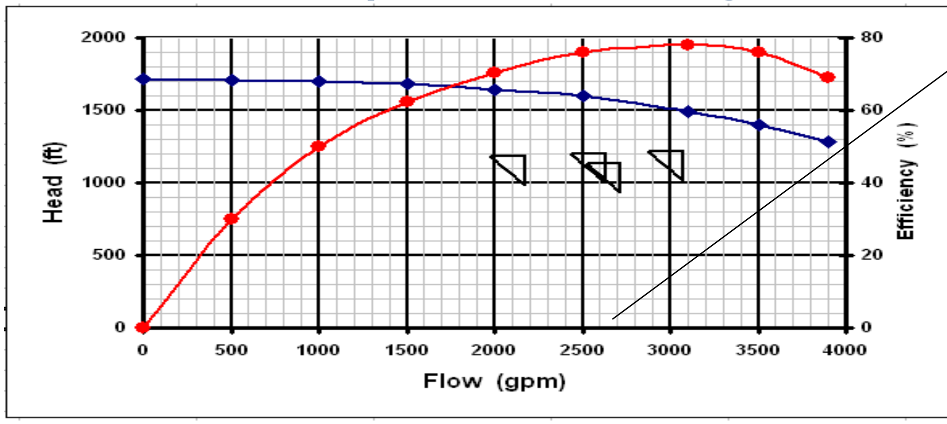
Shop test vibration results
Velocity PK In/Sec
N=3565rpm SG=1.0



Key remark :

All Vibration components at VPF drastically reduced below 0.03 IPS even with SG = 1.0

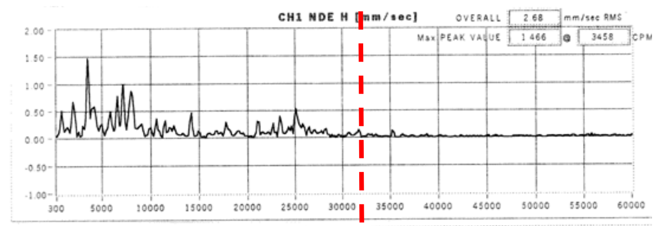
New Impeller Design



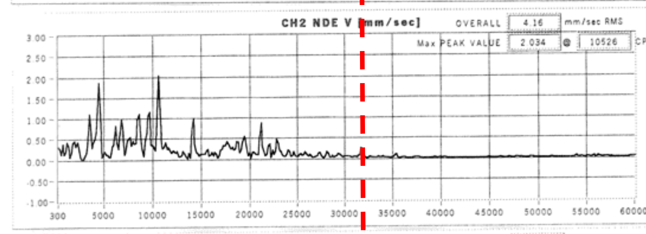
Shop test vibration spectra at 2690 GPM 87% of Qbep-design (PK VelocIty In/Sec)

N=3565rpm SG=1.0

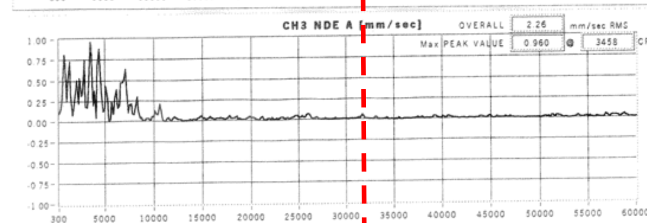
32,085cpm = VPF (9x)



NDE-H

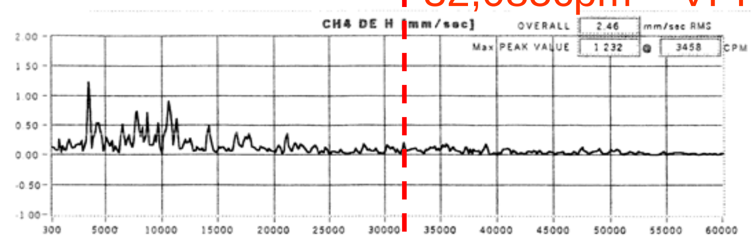


NDE-V

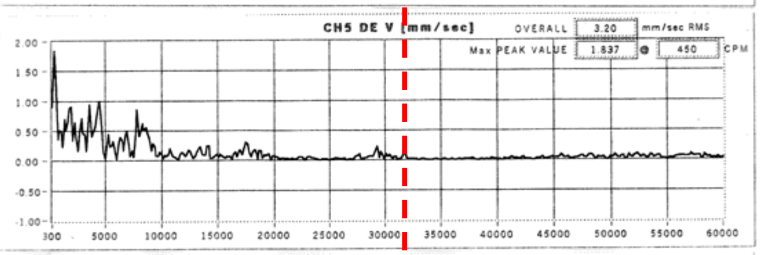


NDE-A

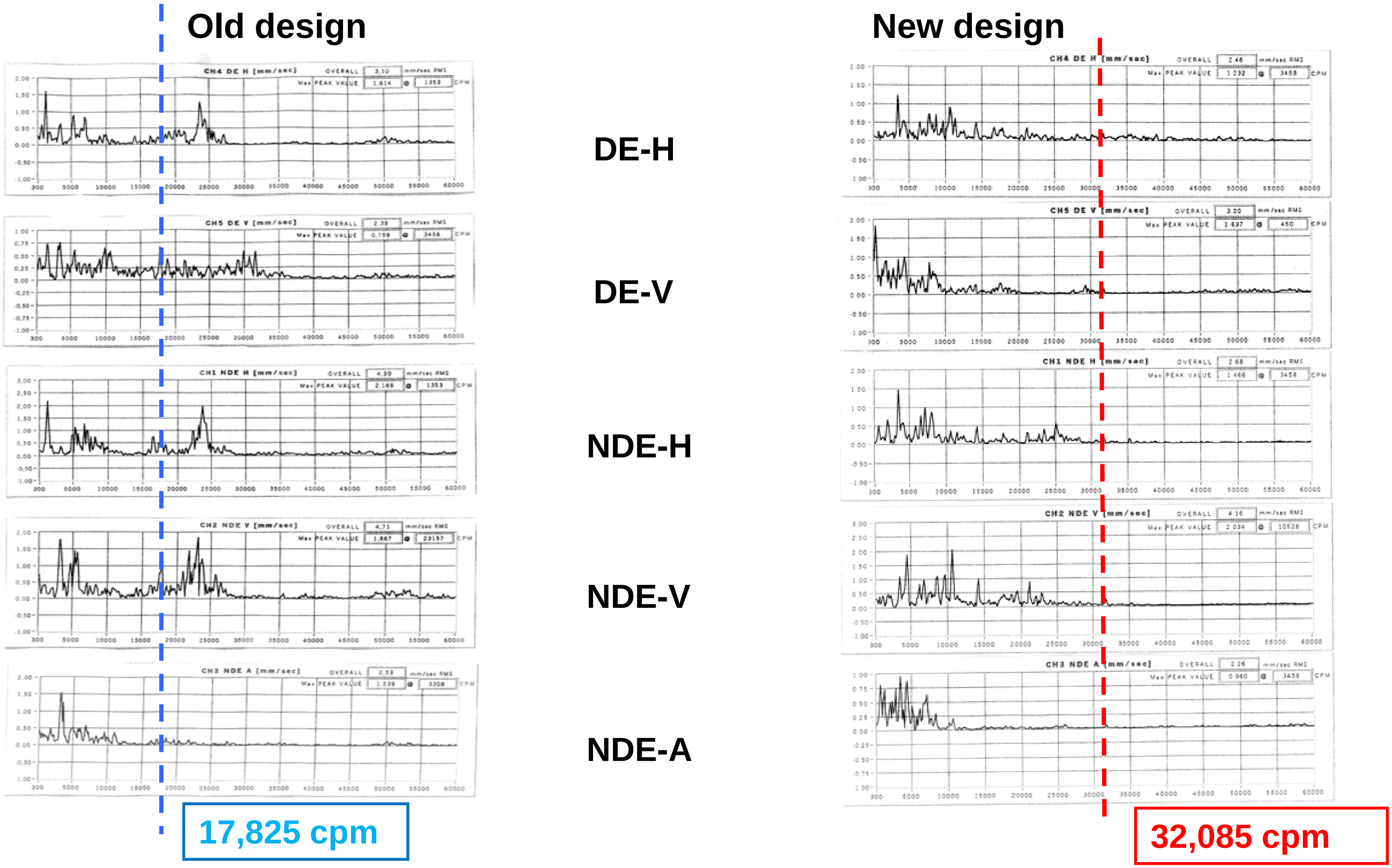
32,085cpm = VPF (9x)



DE-H

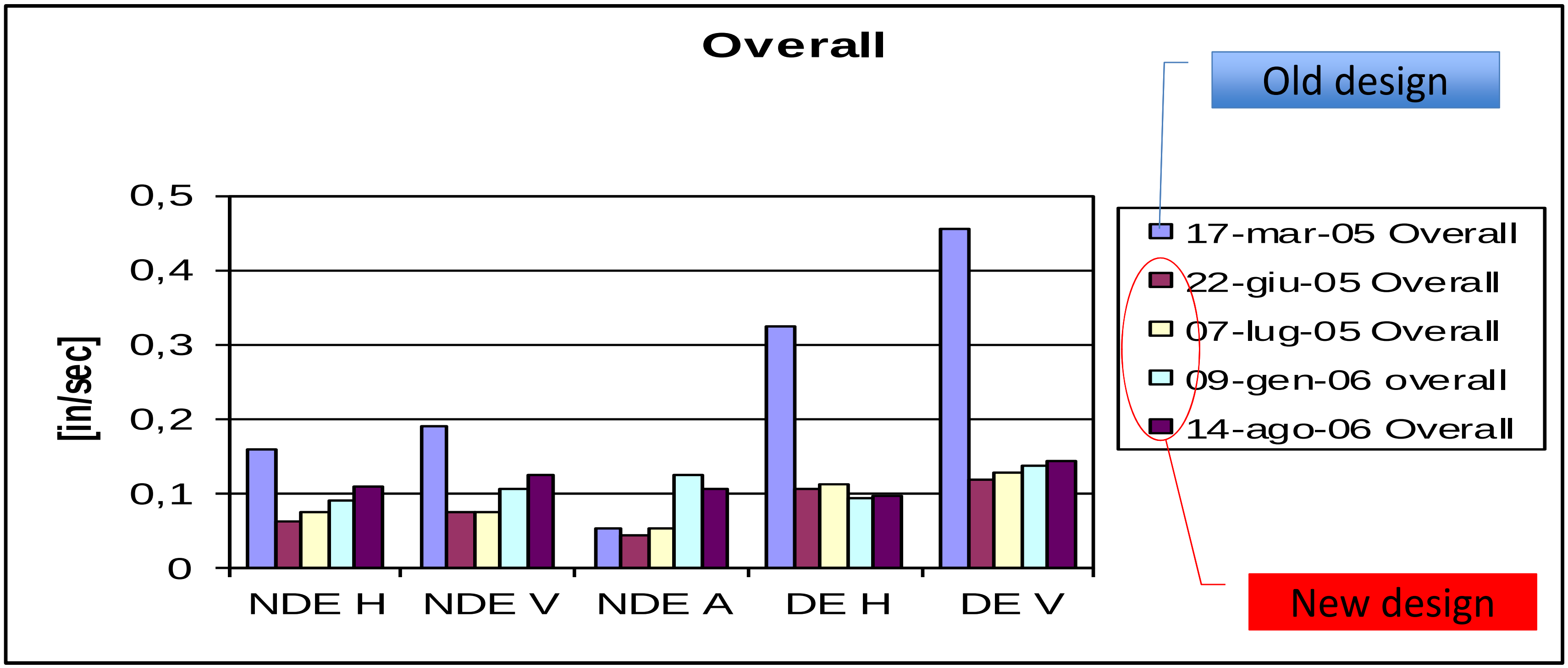


DE-V



Shop test vibration spectra at 2690 GPM – N = 3565 rpm – SG=1.0 (PK Velocity In/Sec)

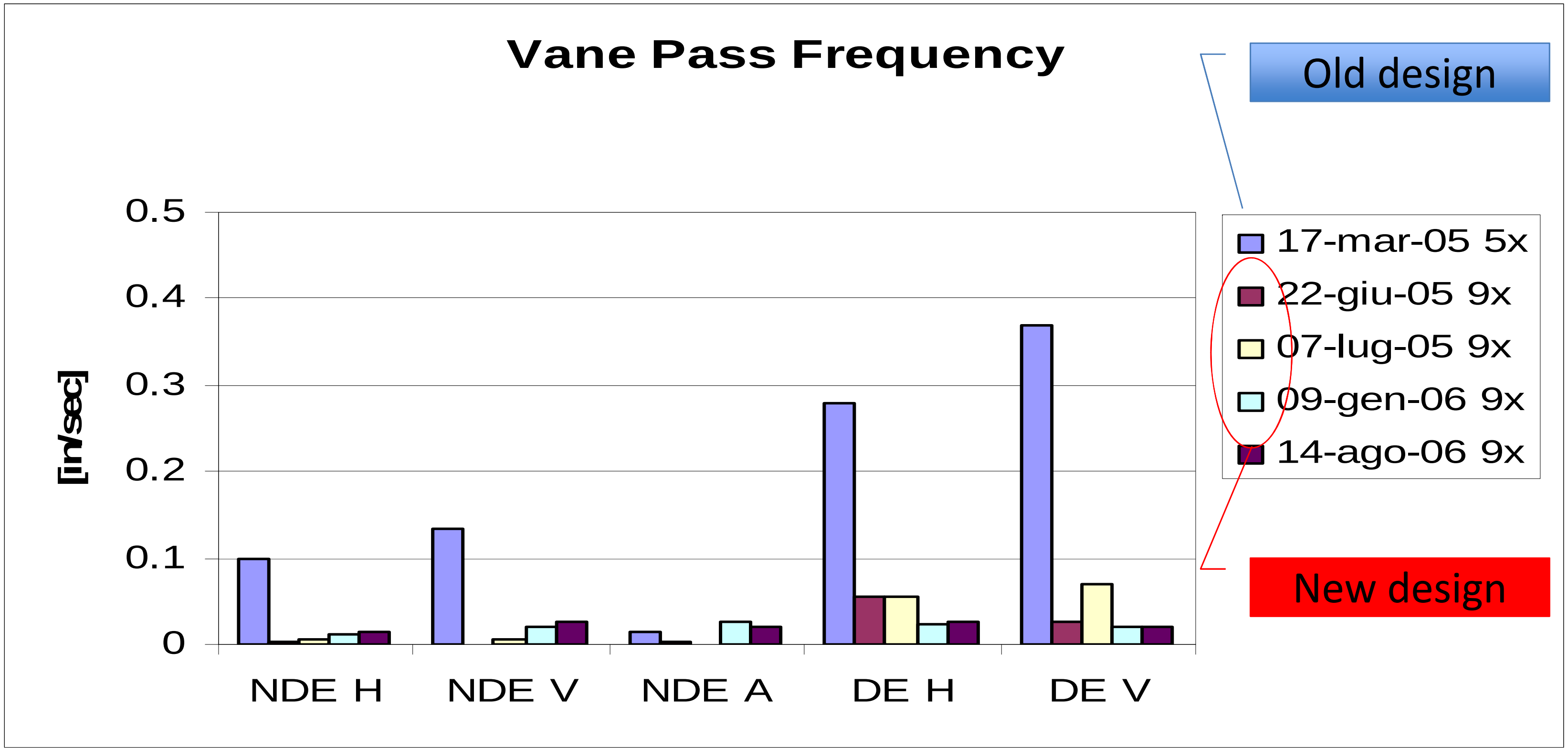
New Impeller - Vibration reading
Site results (Velocity PK In/Sec)
CASE 2, N=3565rpm SG=0.56



Key remark :

Original impeller design : Mar 2005 , New impeller design : Jun 2005 – Aug 2006

New Impeller - Vibration reading
Site results (Velocity PK In/Sec)
CASE 2, N=3565rpm SG=0.56



Key remark :

Original impeller design : Mar 2005 , New impeller design : Jun 2005 - Aug 2006

Conclusions

An analytical diagnostics approach has been applied along with experimental investigation for identifying the vibration root cause. The vibration source was identified as an internal hydraulic excitation at vane pass frequency due to impeller discharge recirculation

An innovative blade impeller design with double row of blades was optimized to meet the specific requirements

The new impeller has been implemented and shop test revealed a drastic reduction of all vibration components below API acceptance level.

Field data show further vibrations reduction with full satisfaction of the customer

Lesson Learned

An innovative blade impeller design with double row of blades was optimized to meet the specific requirements

The new impeller has been implemented and shop test revealed a drastic reduction of all vibration components below API acceptance level.

Field data show further vibrations reduction with full satisfaction of the customer

Thank you for the attention!