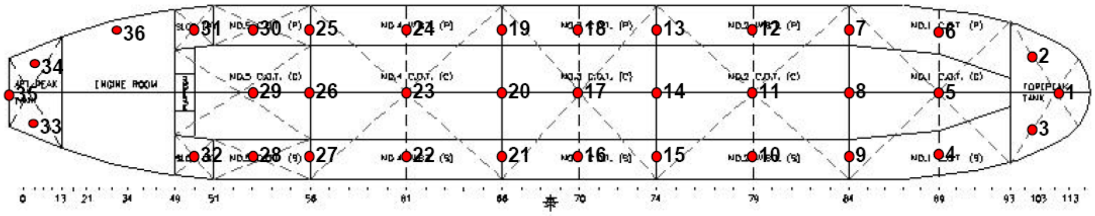
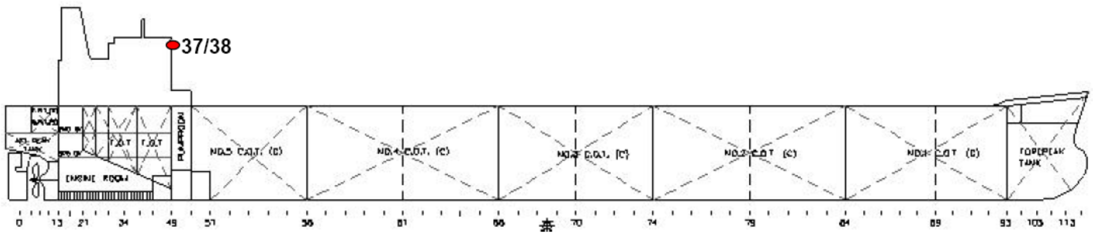
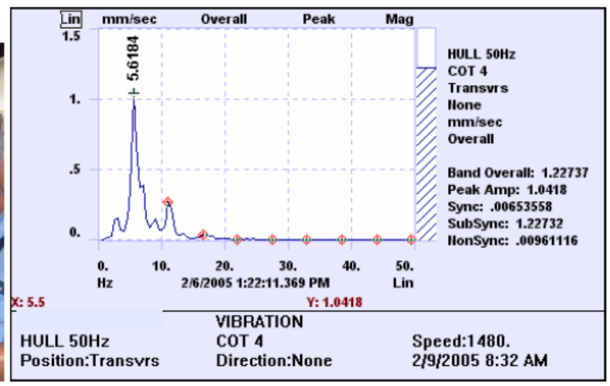
PROFESSIONAL EXPERIENCES

Hull Vibration Analysis (Case Study)





The survey was performed due to the cracks that was repaired at the forward bulkhead of No.4 Water Ballast Tank.



Spectrum 6: Transverse vibration at Side shell Stiffener

Fatigue damage accumulation is associated to the total history of the cyclic wave bending moment rather than just the extreme value of the moment per se; thus the random nature of this history needs to be considered.

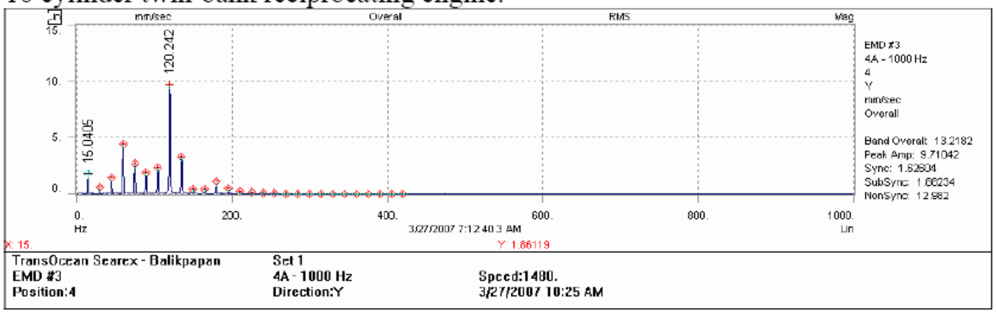
The buckling failure of bottom or deck structures is more complex. Bottom and deck structures are generally grillages stiffened longitudinally, but still presenting transverse structures, named frames, so that different buckling modes can occur: failure of plates between stiffeners, inter-frame flexural buckling of the stiffeners, inter-frame tripping of the stiffeners, inter-frame buckling of the panel and overall grillage failure, usually between bulkheads, involving deflection of both longitudinal and transverse stiffeners.$^{\Phi}$

Vibration forces from the propeller, with combinations of the natural frequency encouraged resonance to occur, and thus contributed to a slow but further cyclic deterioration of the structures. Despite of the vibration level being acceptable as compared against the Hull Vibration Criteria by Llyods Register, the undesirable effect of resonance will be detrimental over a long period, should the ship decides to operate its propulsion at near resonance condition.

Vibration Isolation (Case Study)

4.1 EMD #3 – Vibration Analysis (continue)

Equipment condition was deemed satisfactory. The maximum vibration level of the equipment was 13.2mm/sec rms which occurred at the Generator Non-Drive End in the axial direction, see Spectrum 8. The engine was operating at 900rpm (15Hz), and vibration was dominated by the 8x equipment running harmonic at 120Hz, typical of a 16 cylinder twin-bank reciprocating engine.



Spectrum 8: Vibration at the generator non-drive end bearing in the axial direction.

EMD #3 was fitted with a resilient mount instead of spring isolator as installed for EMDs #1 & #2. The isolation efficiency between from Point 16 to Point 17 was a poor 13%, with an overall vibration reduction from 8.2mm/sec to 7.1mm/sec, see Spectra 9 & 10 respectively. Vibration isolation between the EMD to Deck was insufficient isolated. However, the vibration condition experienced on the engine deck was acceptable & will pose no damage to the existing structure, see Table 2A- Criteria for Vibration of Structures

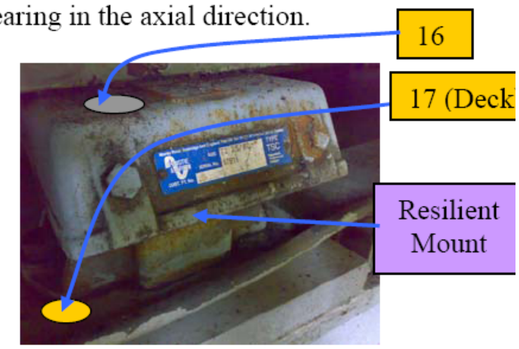
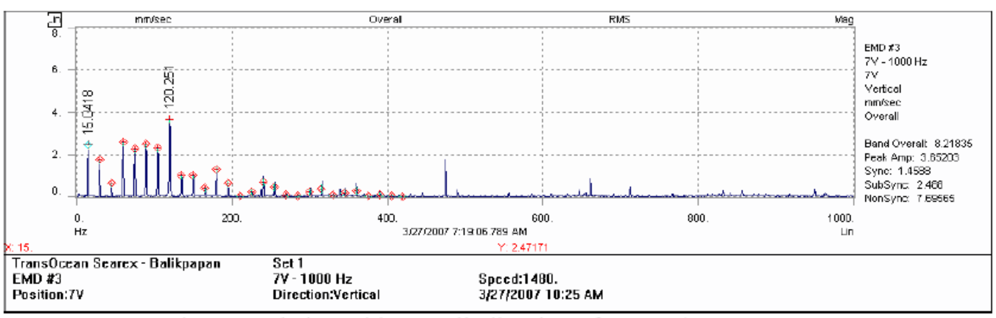
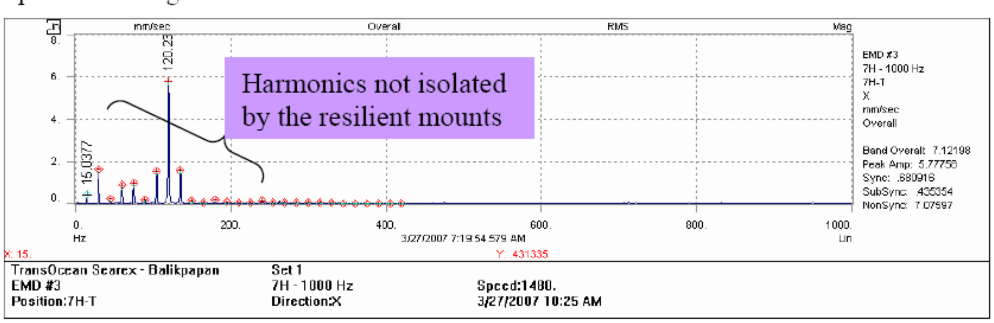


Diagram 2: Resilient Mount

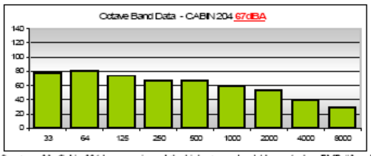


Spectrum 9: Engine Foundation with overall vibration of 8.2mm/sec rms.

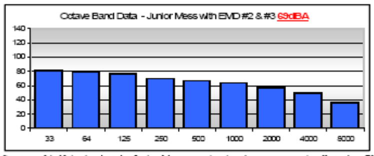


Spectrum 10: Vibration reduction at Engine Deck with overall vibration of 7.1mm/sec rms.

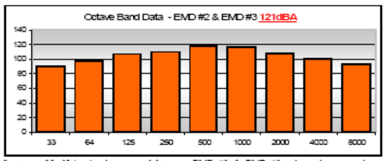
Noise Abatement on Drill Rig (Case Study)



Spectrum 23: Cabin 204 has experienced the highest nose level 'dosage' when EMD #2 and #3 in operation.



Spectrum 24: Noise level at the Junior Mess, experiencing the same acoustic effect when EMD #2 7 #3 were in operation.



Spectrum 25: Noise level measured between EMD #2 & EMD #3, where the super-charge emitted the highest noise 'band'. However, this high frequency has been attenuated by the steel structure of SEAREX. Noise intrusion appeared to be of the lower- frequency band.

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Diagram 7: Halliburton Barge

The increased in noise levels experienced by either the port & starboard spaces were due to the flanking noise from the open machinery deck. The condition appeared to be worsen, i.e. an increased in the overall noise level, particularly in the PMS office and Radio Room when the 'Halliburton' barges berthed along-side Searex IV.

The noise path basically emanated outwards and bounced off the Halliburton silos before reaching the accommodation spaces above, see Diagram 7, represented with red arrows.

The survey has shown that the noise level at the supercharger has been the highest, with a maximum of 121dBA when measured from 1m at a standing height of 1.5m.A feasible solution to reduce the radiating noise from the EMD would be to construct a partial enclosure at the Superchargers.

Diagram 8: Noise path from EMDs.

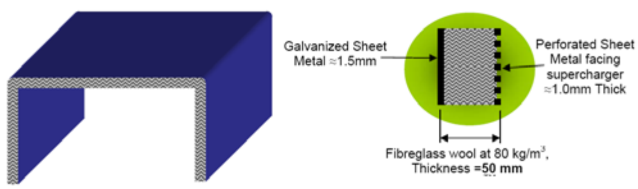


Diagram 9: Proposed Partial Noise Enclosure

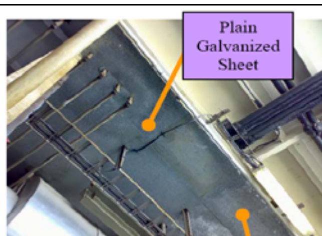


Diagram 5: Galvanized sheet above EMD #3

Physical observation found that the galvanized sheet filled with mineral fiber; once probably assumed to act as thermal insulation as well as an acoustic barrier were found to have little effect on arresting the noise issue. The main reason for such ineffectiveness was the absence of steel perforated sheet which should be in placed instead of a plain galvanized sheet. The plain sheet would act a thermal barrier, but will reflect noise within the machinery space causing acoustic reverberations (incidental waves).

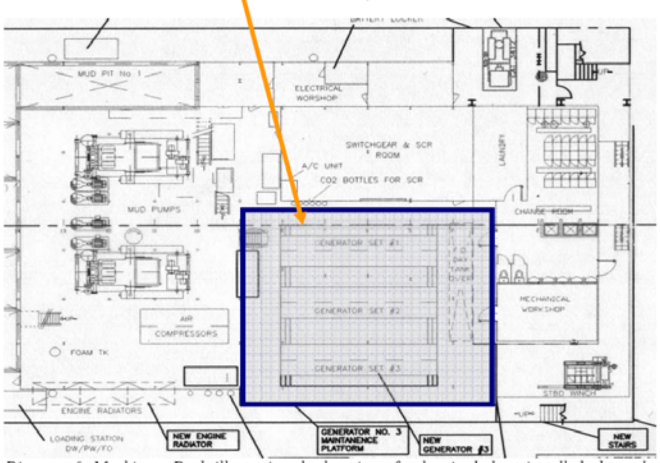


Diagram 6: Machinery Deck illustrating the location of galvanized sheet installed above the EMD Generators.

Pump Foundation Vibration Analysis (Case Study)

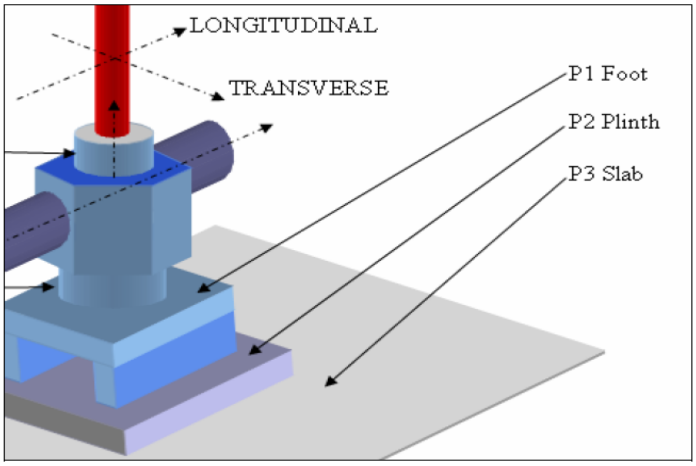
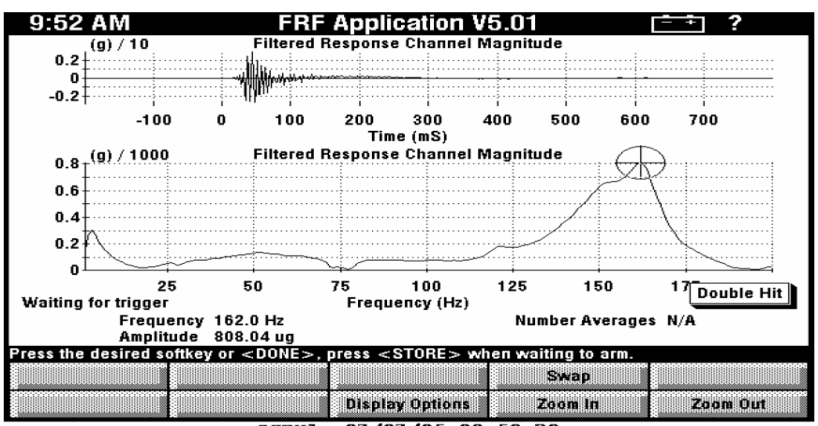


Figure 2: Measurement location and orientation

Operating Speed: 1480 rpm (24.7 Hz)						
ID	**Measurement Location**	**Pump #3**		**Pump #2**		**Remarks**
		Overall µm/s peak	**1X** µm/s peak	Overall µm/s peak	**1X** µm/s peak	
1	Foot P1 1V	1984	1618	179	26	Transmissibility is the real ratio of the output power spectrum to the input power spectrum. Transverse orientation has been selected for this analysis due to its dominance throughout the machine train.
2	Foot P1 1T (Spectrum 5)	2498	2122	188	27	
3	Foot P1 1L	2110	1868	108	9	
4	Plinth P2 2V	2143	1808	211	30	
5	Plinth P2 2T (Spectrum 6)	2470	1961	275	23	
6	Plinth P2 2L	1364	117	231	5	
7	Slab P3 3V	62	25	36	9	
8	Slab P3 3T (Spectrum 7)	83	70	182	5	
9	Slab P3 3L	47	15	54	3	
10	Overall Transmissibility % from P1 – P2 (T)	-1.1	-7.6	46.3	-14.8	
11	Overall Transmissibility % from P2 – P3 (T)	-96.6	-96.4	-33.8	-78.3	
12	Overall Transmissibility % from P1 – P3 (T)	**-96.7**	-96.7	**-3.2**	-81.5	High negativity at Pump #3 represents low transmissibility

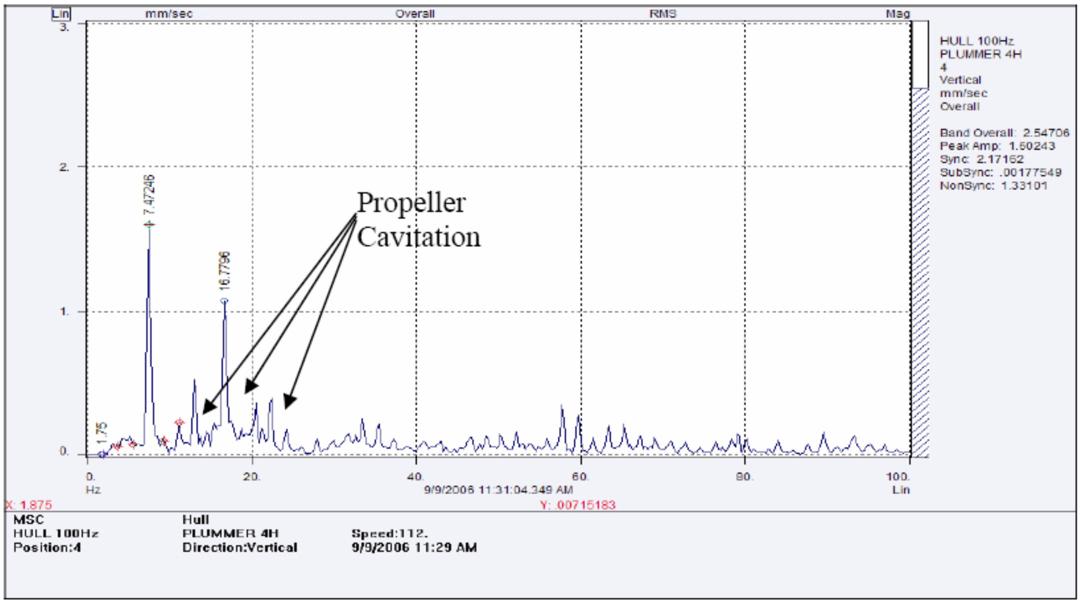
(-)ve denotes less transmissibility and vice versa.

Table 3: Summary of the pump's foundation vibration transmissibility level.



Spectrum 8: Natural Frequency of Pump #3 Vertical – 162Hz

Marine Propeller Cavitation (Case Study)



Spectrum 2: The vibration signature exhibits high blade pass harmonic, typical symptom of excessive pulsation due to poor propeller design and sizing. Poorly designed propeller leads to unsteady flow from the hull through the propeller blades. The unsteady inflow to the propeller blade, while passing through a non-uniform ship wake, causes dynamical changes in the blade pressure distribution. A decrease in pressure to a level below vapour pressure causes the water to boil locally on the propeller blade, e.g. intermittent cavitation occurs.

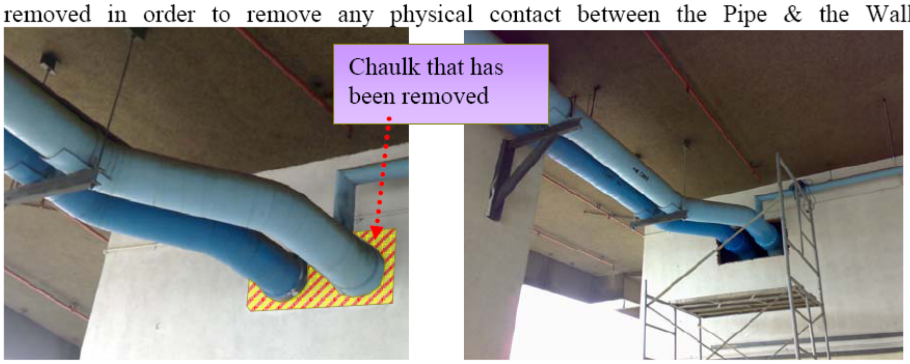
Examining Photos 1 & 2 clearly exhibit evidence of white-water as a result of probable cavitation from the propeller. This picture was taken at 116rpm propeller speed.


Photo 1: Turbulent propeller wake


Photo 2: Propeller cavitation.

Pipe Vibration (Case Study)

Feasible rectification efforts have been carried at the pipe penetration in order to improve the vibration condition of the office at Point 1 & 2. The existing penetrating chaulk was removed in order to remove any physical contact between the Pipe & the Wall.

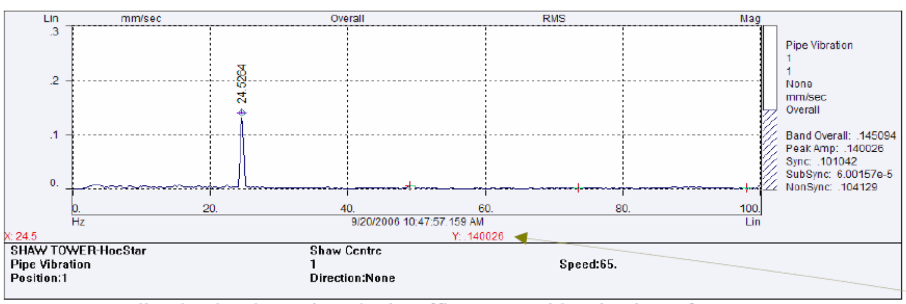


Rubber snubber was also installed between the Pipe Hanger & Stud to reduce the vibration transmission from the 'gurgling' pipe to the above ceiling.

Measurement Location		Vibration levels at indicated frequencies μm/sec rms @ Hz								
		Vertical			X - Horizontal			Y - Horizontal		
Point 1 Before Repair	Office Space	**140**	**@**	**24.5**	10	@	3.5	7	@	3.5
Point 1 After Repair	Office Space	**93**	**@**	**24.5**	21	@	24.5	6	@	24.5
Point 2 Before Repair	Office Space	54	@	24.5	10	@	3.3	7	@	2.8
Point 2 After Repair	Office Space	84	@	24.5	12	@	24.5	8	@	24.5

Table 2: Summary of Maximum Floor Vibration Results in μm/sec rms after Repair

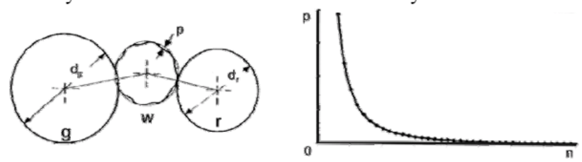
The maximum vibration level prior to the repair has reduced from 140 μm/sec to 93 μm/sec rms in the Vertical Direction, see Spectra 6 & 7 respectively.



Spectrum 6: Vibration level at Point 1 in the office space with a dominant frequency at 24.5Hz, which corresponded to the chilled water pump running speed. The magnitude was 140 μm/sec.

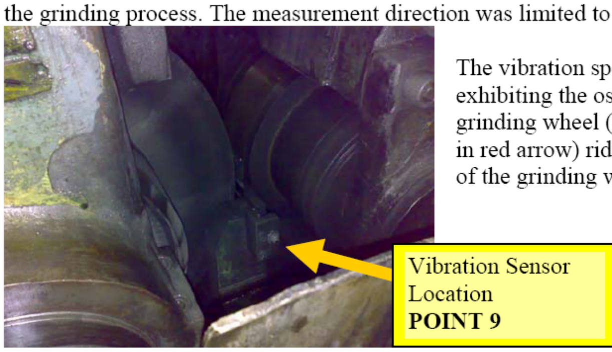
Investigation of Production Defect (Case Study)

Centreless (centerless) grinding can perform excellent roundness of the work piece. However, caused by the simultaneous suspending and machining of the work piece surface it is possible that process typical roundness errors are generated. Proper adjustment of the grinding machine and the grinding slot geometry is essential. When a high spot comes in contact with the regulating wheel, then on the other side of the work piece a low point will be ground. However this low point must not be exactly in the opposite side of the work piece. The grinding machine has to be set up in a way that a polygon form is ground with so many corners that it is almost round finally.

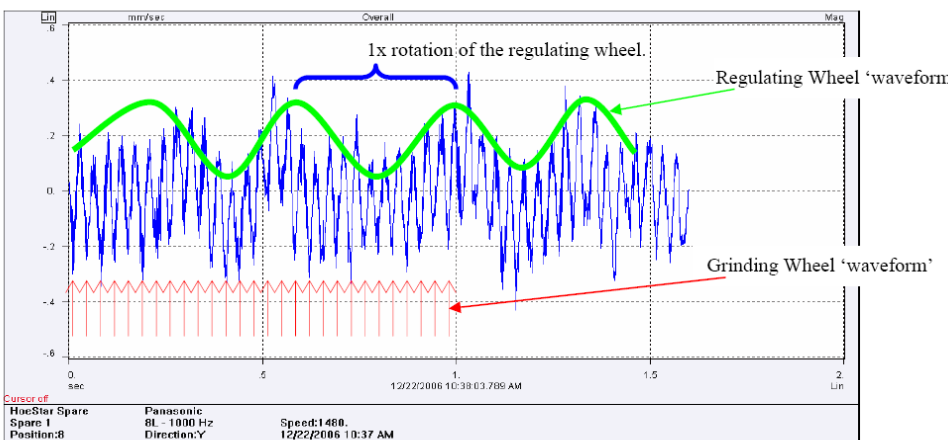


g: grinding wheel - r: regulating wheel - w: work piece - d_g: diameter grinding wheel - d_r: diameter regulating wheel - p: penetration depth - n: polygon order

Vibration sensor was also placed at the work-piece blade support to confirm rigidity during the grinding process. The measurement direction was limited to 'Axial' orientation only.



The vibration spectrum is shown below, exhibiting the oscillation frequency of the grinding wheel (small waveform indicated in red arrow) riding on the oscillation wave of the grinding wheel (big green waveform).



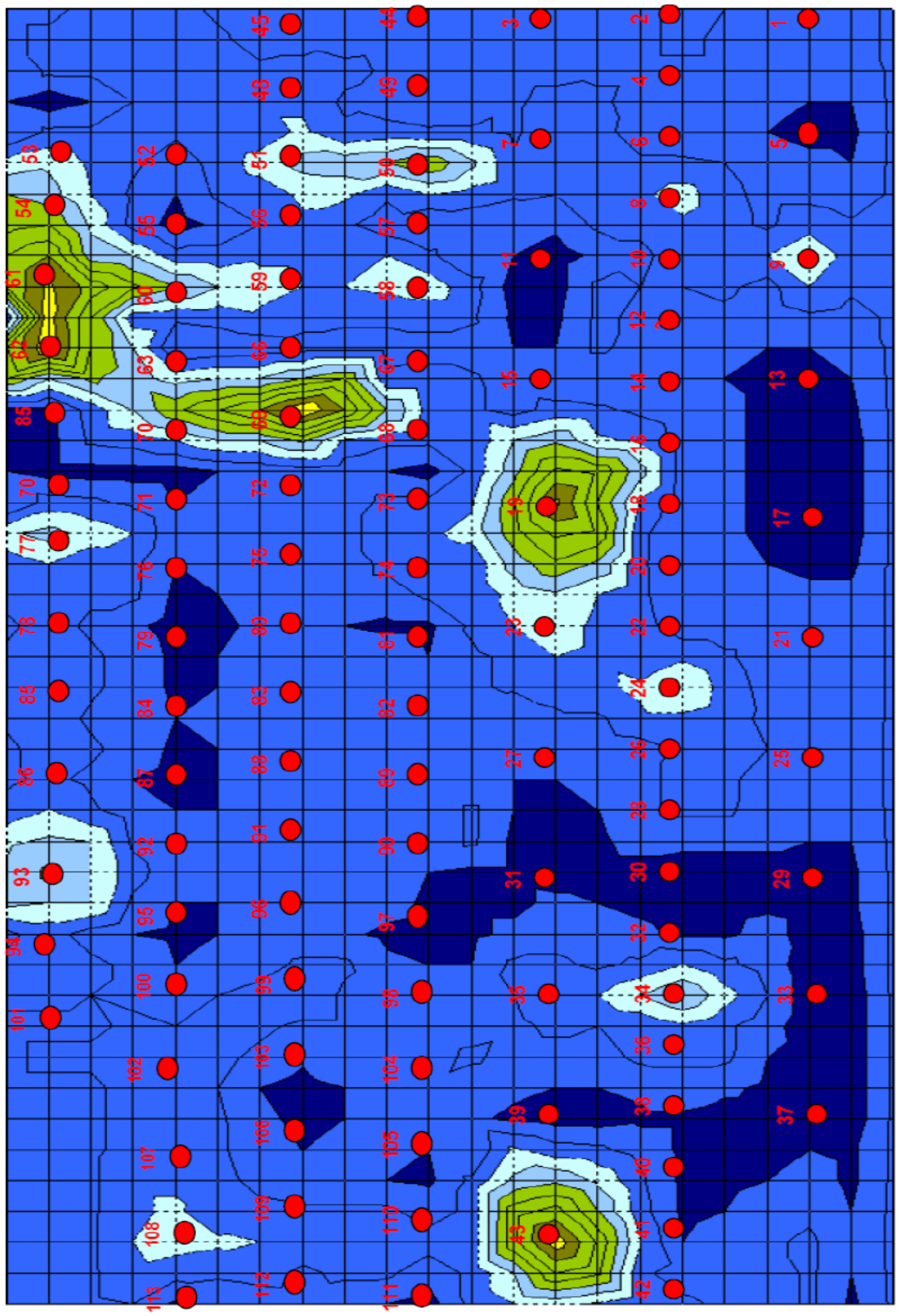
Waveform 1: Vibration signature obtained from Point 9, i.e. work-piece blade support measure on 22nd December 2006 during grinding process.

Vibration oscillation shown in Waveform 1 reveals the rotational speed of the regulating wheel and its impacting rotational force exerted from the grinding wheel being transferred via the work-piece. Examining each rotation of the regulating wheel within the blue bracket, we have 14 oscillating & impacting frequency of the grinding wheel. Based on linear velocity conversion, the angular displacement of the grinding wheel acting upon the work-piece is being fed by the grinding wheel would be 0.7 μm peak to peak. This displaced value can be considered small, with virtually no impact to product quality. Should the oscillating displacement of the regulating wheel, in this case the 'movable' member become excessive, the feed-force would cause angular impingement on the ground surface, hence affecting the cross-sectional profile of the finished work-piece.

Floor Vibration Profile (Case Study)

Floor Vibration Summary – Concrete Floor

The following vibration profile has been derived from the concrete slab obtained from the clean room space (Points 1 - 43) and future production space (Points 44 – 114).



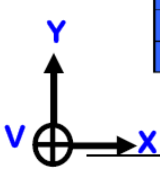
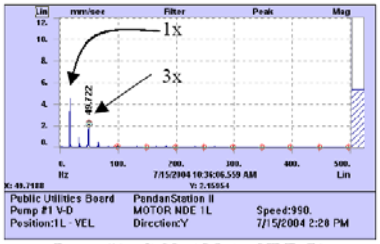


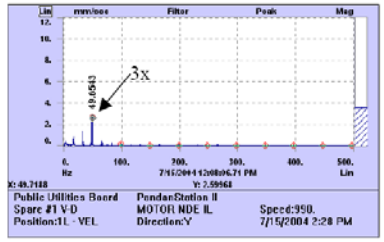
Figure 1A: Vibration Profile at MSA 1.5 at Concrete Floor

Detection of Crack Rotor (Case Study)

Despite of the magnitude of the unbalance component at 1x to have been reduced by 50% when hot, the harmonics at 3x motor running speed (49.7Hz) has remained regardless of the change in the external and internal temperature of the motor, see spectrum below.



Pump #1 : Cold at Motor NDE (L)



Pump #1 : Hot at Motor NDE (L)

A detail examination of the above spectrum in a logarithmic plot reveals the presence of probable sidebands around the 1x, 2x, 3x, 4x ….at motor running speed, see Spectrum 13. The symptom is often associated with a loose or/and crack rotor bar.

A cracked rotor bar will cause localized heating of the rotor, which causes uneven expansion of metal and rotor bowing. This will result in combinations of significant unbalance component at 1x running speed vibration with sidebands related to the slip frequency.

Infrared thermography has been employed to confirm a good stator condition i.e. by observing the heat transfer of probable deficient stator windings to the external motor casing if any.

Figure 1 shows the thermal profile of motor #1 prior to start-up. The temperature has been uniformly maintained with the aid of a space heater.

Figure 2 was taken approximately 2 hours after start-up. Uniform temperature profile is typical of acceptable stator windings.

Vibration analysis does not show any signs of stator problem, which is usually represented by the presence of 2x line frequency at 100Hz. Hence motor stator condition is acceptable.

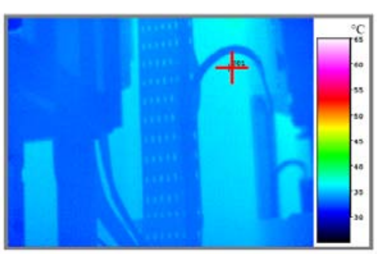


Fig1: 'Cold' Spot Temperature = 35°C

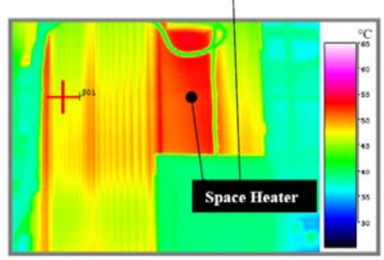


Fig 2: 'Hot' Spot Temperature = 50°C

Finite Element Analysis of 'Resonating' Pump(Case Study)

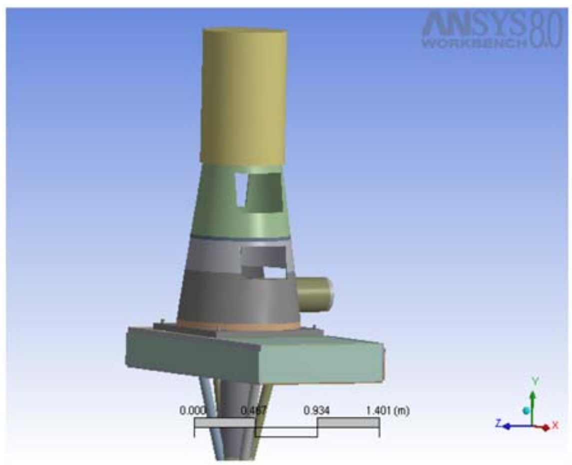
Frequency Results (Post Treatment)

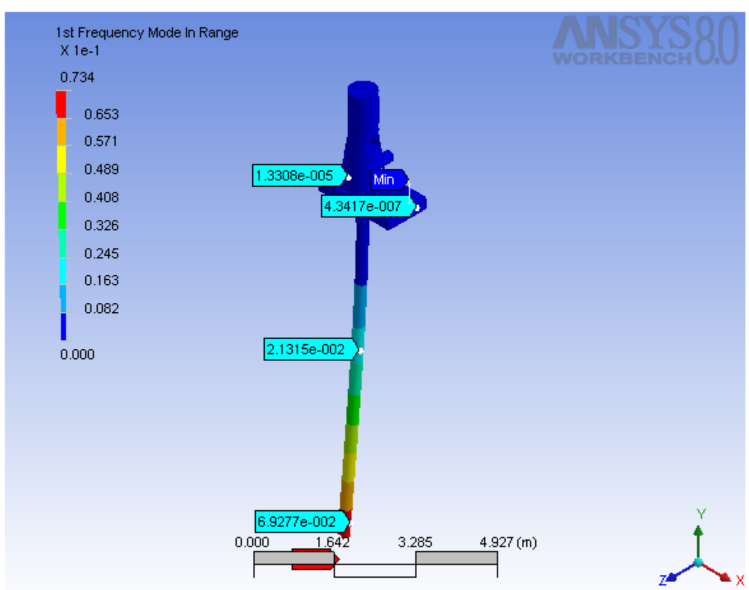
Frequency results apply to all active bodies in *"Model"*.

First 6 Natural Frequencies			
Name	**Mode**	**Frequency**	**Alert Criteria**
"1st Frequency Mode In Range"	1	4.4 Hz	none
"2nd Frequency Mode In Range"	2	4.7 Hz	none
"3rd Frequency Mode In Range"	3	**26.47 Hz**	none
"4th Frequency Mode In Range"	4	**28.18 Hz**	none
"5th Frequency Mode In Range"	5	31.98 Hz	none
"6th Frequency Mode In Range"	6	32.46 Hz	none

Tabel 1: Pre-Treatment frequency results

- Description: *"Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1"*
- Material data file: *"C:\Program Files\ANSYS Inc\v80\AISOL\CommonFiles\Language\en-us\EngineeringData\Materials\Structural_Steel.xml"*

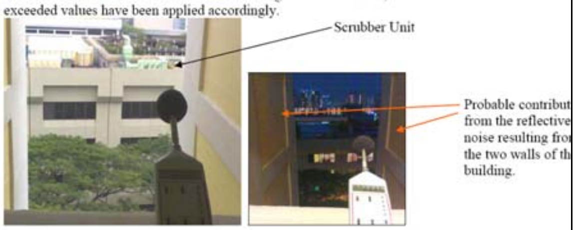




Noise Abatement –Environmental (Case Study)

5.2 Noise Level at Residential, Block 114 Level 13.

The scrubber unit located at the roof-top of HP's facility appeared to be tonally dominant throughout the measurement event. The din of vehicular noise has been the contributor to the 'rise & fall' of the overall noise level during the measurement, thus correction factor on exceeded values have been applied accordingly.



Condition (TWA*)	7 am – 7 pm	7 pm – 11 pm	11 pm – 7 am
Criteria Residential	65 dBA	62 dBA (60+2)	56 dBA (55+1)
Block 114, Level 13	64.3 dBA	62.6dBA	59.0 dBA
Difference from Criteria	- 0.7 dBA	0.6 dBA	3.0 dBA

*TWA is Time Weighted Average over the given period herein

Condition (Leq5**)	7 am – 7 pm	7 pm – 11 pm	11 pm – 7 am
Criteria Residential	70 dBA	65 dBA	60 dBA
Block 114, Ground Floor	65.6 dBA	63.2 dBA	59.6 dBA
Difference from Criteria	- 4.4 dBA	- 1.8 dBA	- 0.4 dBA

Within Criteria | Exceeds Criteria

Taken that the discharge noise of fan to be 85dBA.
Distance of CES fans to louver ≈ 4 metres.
Distance of Louver to ground at boundary ≈ 25metres.

Noise from CES x3 to main Louver
$= 10\log(10^{8.5}+10^{8.5}+10^{8.5})$
$= 89.8$dBA.

Empirical data to subtract 6dB for the Louver from free field reduction.
$84.8 - 6 \Rightarrow 83.8$dB

∴ Noise from the boundary would be $= 89.8 - 20\log(\frac{25}{1})$
$= 55.8$ dBA.

Conclusion

Acoustic Louver would ideally have a size of approximately 9'x 4' with 11-slits at 3" gap. The noise emission to the ground has been conservatively calculated to have a resultant of 55.8dBA.

General Arrangement of Acoustic Plenum & Louver.