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Dynamic & Sequential Monitoring:
Preventative, Predictive and Planned Maintenance

Vibration Analysis Report

Example Company Limited, Example Town, UK: 7th April 2007.

Report Content(s): Furnace Exhaust Fan Vibration Tests.

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	Date of Survey(s): 7th April 2007 **Our Reference Number:** HA07-03	

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Part
1

Report Section

Introduction; Machine Listings; Site Details and Relevant Data

Introduction

Preface

Example Company Limited, situated in Example Town, UK, requested an in-depth *vibration survey* to be conducted on their Furnace Exhaust Fan, following a spate of rotor unbalance problems associated with the installation over the past twelve months of operation.

It was noted that the installation was either '*out of balance*' or approaching a state of unbalance each time the fan was restarted following a shutdown period. Despite numerous attempts to re-balance the fan to normal amplitudes after a shutdown period, the fan approached a state of unbalance over a relatively short period of time. On each occasion, the fan impeller proved to be uncontaminated and did not require cleaning, indicating there was a more serious underlying problem associated with the installation.

Hence, it was the purpose of this survey to highlight the cause of the imbalance and develop a program of work to reduce or eliminate the re-occurrence of unbalance affecting the installation.

Machine Listings

Each stage of the manufacturing process is carried out by various machinery located within purpose built buildings and outdoor enclosures which have been monitored during the course of the on-site *vibration survey*. The following sections at Example Company Limited have been included in this survey:-

1. **Furnace Exhaust Fan:** The Furnace Exhaust Fan located on the upper section of the furnace inside the factory contained the following: (i) one single *horizontally* mounted *belt driven* **Motor**, of 2977 RPM, 75 kW capacity, containing two identical SKF 6316 C4 bearings on the *non drive end* and *drive end* positions of the motor. The fan shaft contained two integral support shaft bearings, with a slight increase of speed of 3050 RPM.

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Data Collection

The machines were monitored using a *Diagnostic Instruments Di460 dual channel* FFT analyser which is a combined portable data collector and sophisticated signal analyser, capable of detecting the following fault types:

- bearing damage
- lubrication problems
- imbalance
- positional misalignment
- angular misalignment
- loading anomalies
- looseness
- gearbox component wear/damage
- conveyor belt tracking
- resonance and structural related problems
- foundation movement
- coupling damage
- excessive stress points
- inadequate support
- application problems
- hysteresis whirl
- faulty belt drives

The analysis conducted on site highlights any of the above fault types and on detection will be documented in this report, stating the exact defect, cause and severity.

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Spectrum Analysis: Description and Purpose

Before presenting this report in further detail, it may be necessary to explain some of the terminology used when referring to the recorded spectra:

Velocity Spectrum

- *Velocity* is movement expressed in terms of time and in our case, measured in "**mm/sec.**". The **ISO10816-3** *vibration severity standard* categorises certain types of machinery (in accordance with capacity; rotational speed and operating circumstances) into certain *zones* such as: "Excellent"; "Acceptable"; "Unsatisfactory" and "Unacceptable".

Acceleration Spectrum

- *Acceleration* is *the rate of change of velocity* (in other words, movement per second per second) and is measured in "**G**". Acceleration is a *high frequency* spectral signal relating to typical faults such as: bearing fatigue; gear wear; complex or multiple component damage; excessive loading anomalies and aerodynamic related problems on rotors or impellers.

Enveloped Spectrum

- *Enveloping* is the relatively new technique, which *filters* and *processes* the normal *high frequency* characteristics naturally produced by the **combined** components of a bearing. In the past, conventional techniques only permitted the analysis of the *overall* signal, which could actually indicate an array of anomalies other than a bearing defect. This method became an increasingly unreliable and misleading technique, especially when levels began to rise - for example, it was impossible to distinguish between a *genuine* bearing defect and a typical loading related problem.

When utilising the *enveloping* technique, the following processes take place:

- The **complex** signal is *demodulated* by use of a *band pass filter* centred on the *resonant carrier waveform*.

- The entire process is *enveloped* (*half wave rectified*).

- The **individual** components of a bearing such as faulty inner/outer races; cages and rollers/balls are now detectable at specific frequencies.

The technique is particularly useful on slow running machinery (e.g., below 10 RPM) where conventional methods fail to produce any relevant signals to analyse.

The investigation can now be classified into the following areas:

1. Site Details.
2. Vibration Measurements on the machines.

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Vibration Survey on Furnace Exhaust Fan

Initial Inspection of the Supporting Structure and Operating System

The Furnace Exhaust Fan installation appeared to be adequately placed upon suitable foundations and supporting structures. However, in order to investigate the possibility of any transmission of vibration and excitation of resonance from *machine* through to *structure*, it was necessary to conduct an additional 'bump' test on various parts of the supporting structure and ducting (see Test 1 below).

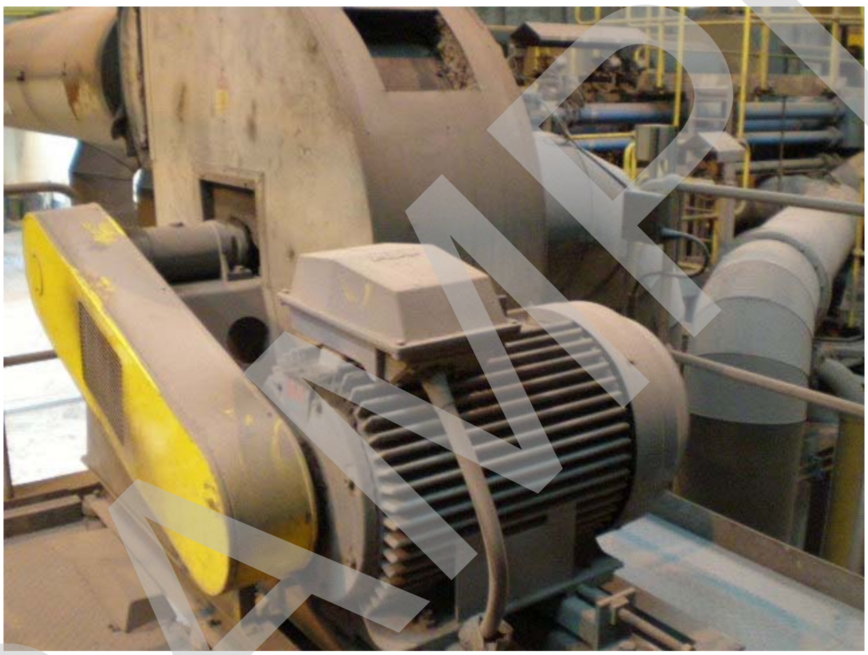

Figure 1: Photograph showing the layout of the Furnace Exhaust Fan located at Example Company.

Test 1 - Resonance 'bump' tests on supporting structure and ducting

During this particular test, the Furnace Exhaust Fan located at Example Company was inoperative and an applied force was exerted with a mallet on each section of the supporting structure and outlet ducting. The applied force 'excites' the natural resonant frequencies of the supporting structure and it is possible to collect a signature which will identify the dominant natural frequency of the system at that particular location. If this *natural* frequency coincides with any of the *driving* frequencies of the machine, a resultant amplitude can be so great as to induce complete premature failure of the installation.

It is quite rare even nowadays that manufacturers conduct a complete resonance test of the entire system as a whole. Quite often, only the machine parts are considered for these tests (i.e., belt resonance, pulleys, couplings and drive shafts), rather than the *combined* elements containing the structure, ducting, support bed and foundation etc. During the initial test with no loading occurring on the installation, the following sections of the installation at each mounting bolt location were tested:-

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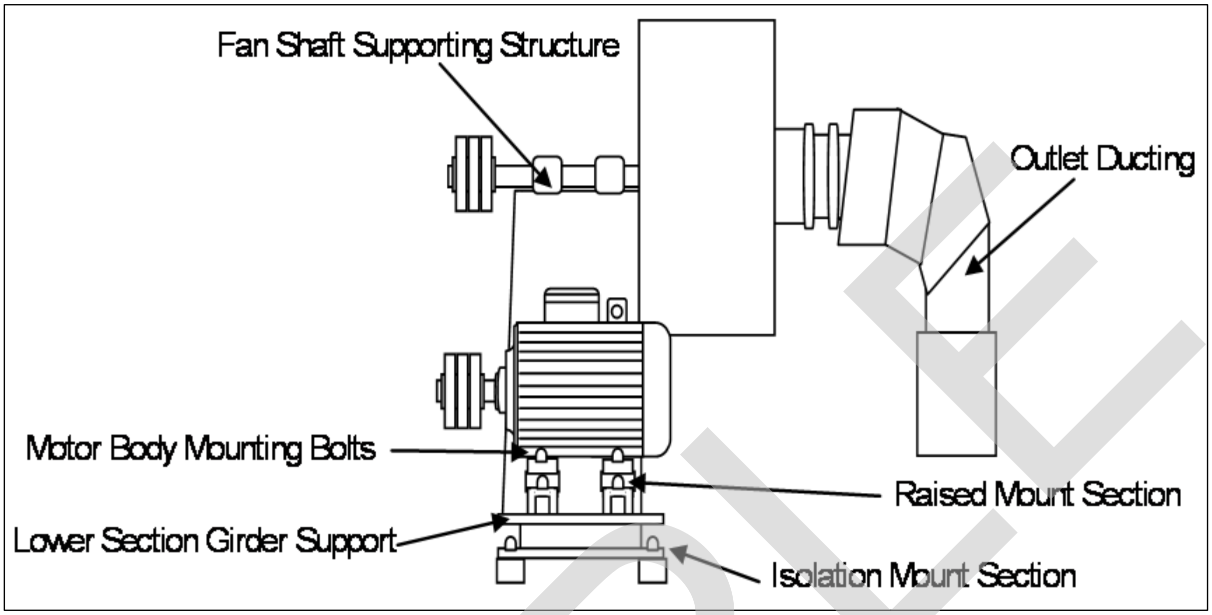

Figure 2: Diagram identifying the different support levels tested for possible resonance related problems.

The resonance tests conducted on the supporting structure and ducting revealed two locations of concern:

(i) The '***raised mount section***' of the motor on the *drive side* position (nearest the pulley) identified a signal approaching one of the *belt passing* frequencies at 11.30 and 15.30 Hz frequencies.

(ii) The **outlet ducting** revealed a natural frequency of 14.10 Hz frequency, also close to one of the natural *belt passing* frequencies of the installation.

It was also noted that the *inlet ducting* on the opposite side of the impeller casing showed multiple harmonics, anyone of which may cause resonance related problems with the installation.

In conclusion, it is thought that the outlet ducting would not interfere with the overall vibration levels experienced on the machine, as the ducting was separated by pieces of material, specifically fitted to avoid any transmission of resonance from one part of the structure to another.

More importantly, the raised mounted section on the *drive side* of the motor may cause resonance related problems leading to imbalance on the installation. The simplest solution would be to ensure all welds are intact at this position and re-weld or build-up additional welds where necessary. It is also advised that the mounting section here is the 'weaker' part of the supporting structure, and it is essential all mounting bolts should be tightened to the correct torque (over-tightening can cause a condition known as 'sprung foot' where the frame becomes distorted, particularly effecting amplitudes at 1x RPM).

Test 2 - Free Run

During this particular test, the Furnace Exhaust Fan located at Example Company was in full operation and readings were taken at each bearing location of the machine; this particular test is analogous to the *condition monitoring* tests performed every three months at Example Company, with the exception that additional *high frequency* and *non-synchronous time waveform* readings were collected, as well as the inclusion of additional monitoring positions.

The results of this test revealed the installation was operating above the '**Unacceptable Zone**' as specified by the Vibration Severity Standard: **ISO10816-3**. The *vertical imbalance* detected on the *fan non drive end* bearing (i.e., the bearing nearest the impeller) attained a *velocity* amplitude of 11.0 mm/sec. at 52.188 Hz frequency. Additional readings also highlighted the possibility of an *eccentric pulley* located on the *motor drive end* bearing, where *belt passing* frequencies attained a *velocity* amplitude of 7.70 mm/sec. at 52.50 Hz frequency within the *horizontal* plane (as well as fluctuating amplitudes on the same position in the *vertical* plane).

It was noted that the *fan drive end* bearing became very hot in a relatively short period of time during this particular test; however, additional *enveloping* analysis revealed no untoward bearing related problems associated with the installation. Further readings identified a comparatively high level of *blade passing frequency* vibration in the *vertical* plane on the *fan non drive end* bearing, attaining a *velocity* amplitude of 6.0 mm/sec. at 313.438 Hz frequency. Although *blade passing* frequencies are naturally inherent in *fans*, *pumps* and *compressors* etc., a higher-than-average amplitude may indicate possible diffuser ring wear (uneven impeller rotation); obstructions in ducting and/or eccentric fan rotor.

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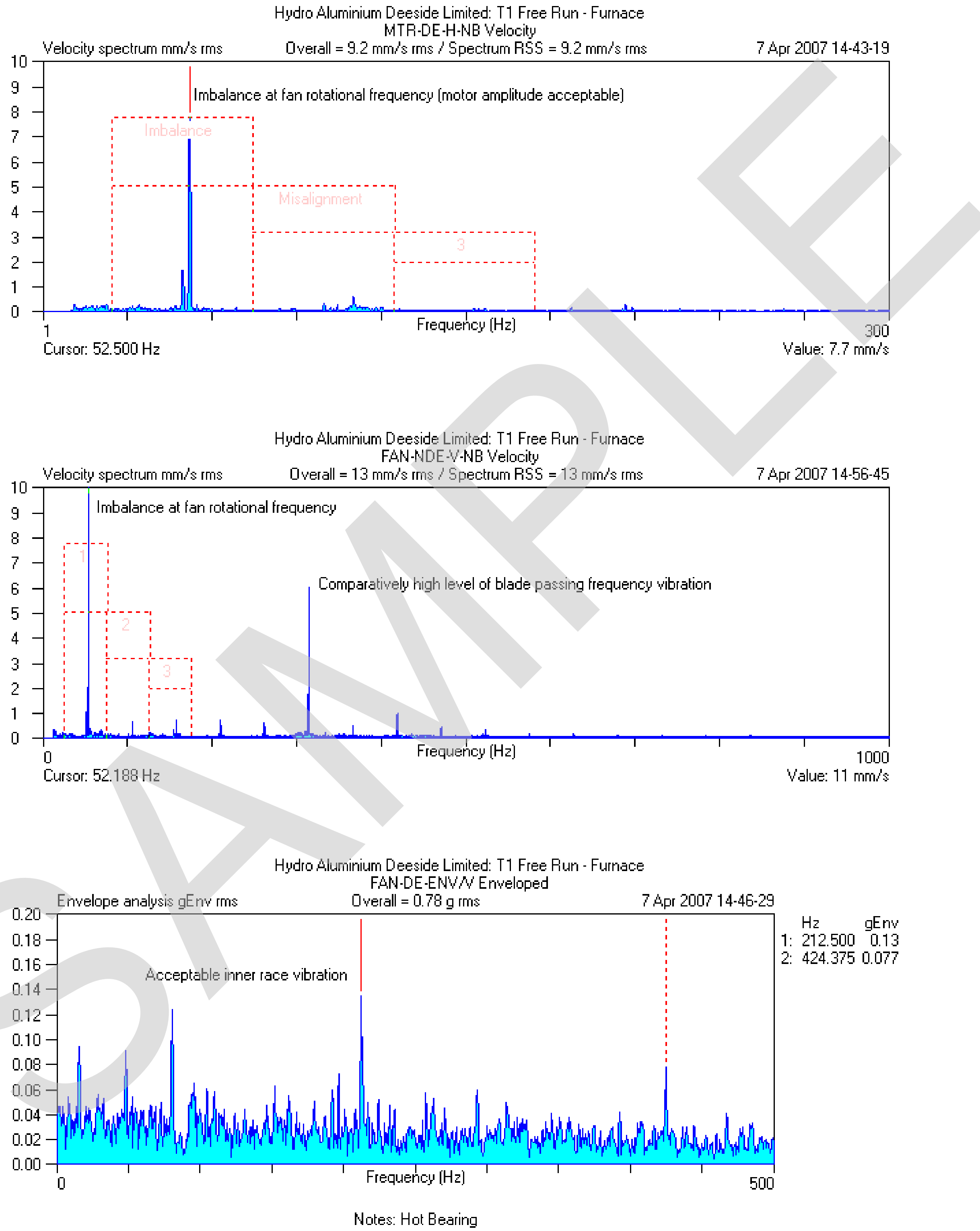

Figure 3: Non-synchronous FFT vibration spectra obtained at various monitoring positions of the Furnace Exhaust Fan.

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Test 3 - Phase and Dual (Cross Channel) Analysis

During this particular test, the Furnace Exhaust Fan located at Example Company was in full operation and readings were taken at each bearing location of the machine in the same manner as Test 3, but with the addition of a tachometer reference and additional accelerometer input (dual cross-channel input). The purpose of this test was to identify the *phase* relationship between the position of the two sensors, which can accurately pinpoint the *type* of mechanical problem inherent on an installation.

It was confirmed in Test 2 that a *vertical* imbalance was present on the fan shaft. There was also the possibility of an *eccentric pulley* located on the drive end position of the motor. The results of Phase and Dual Cross Channel Analysis tests are documented below in Table 1:

Test 3 - Phase and Dual Cross Channel Analysis			
Sensor Positions	Phase	Max. Amplitude	Remarks
Motor Verticals	184°	4.60 mm/sec.	Indicative of 'couple imbalance'
Motor Horizontals	92°	1.40 mm/sec.	None
Motor Pulley (V & H)	318°	2.10 mm/sec.	None
Fan Pulley (V & H)	99°	1.70 mm/sec.	None
Fan Verticals	302°	1.40 mm/sec.	None
Fan Horizontals	64°	5.30 mm/sec.	Indicative of 'force imbalance'

Figure 4: Table 1 identifying the phase relationship between locations of two sensors.

The results of Test 3 reveal there is no evidence to support the theory of an eccentric or offset pulley, where a phase difference of 0° or 180° is expected between relative *vertical* and *horizontal* positions. However, it was noted that the phase relationship on the motor revealed a typical 'couple imbalance' where balancing would require the placement of weights in at least two planes to correct (i.e., a phase difference of approximately 180°). The phase relationship identified on the fan bearings identified a typical 'force imbalance', where only one balance weight would be required to correct vibration to acceptable limits.

Hence, it is concluded that there are two types of imbalance affecting the installation; a 'couple imbalance' on the motor and 'force imbalance' on the impeller. The 'couple imbalance' identified on the motor may be attributed to the supporting frame 'flexing' and becoming distorted (it was noted also that the amplitude fluctuated greatly in the vertical plane on the motor).

The only other comment to make at this stage is the unknown effects of *thermal stratification* on the impeller, whereby excessively high temperatures can in some cases cause the impeller blades to bend outward under the influence of centrifugal force, resulting in an uneven mass distribution and force imbalance.

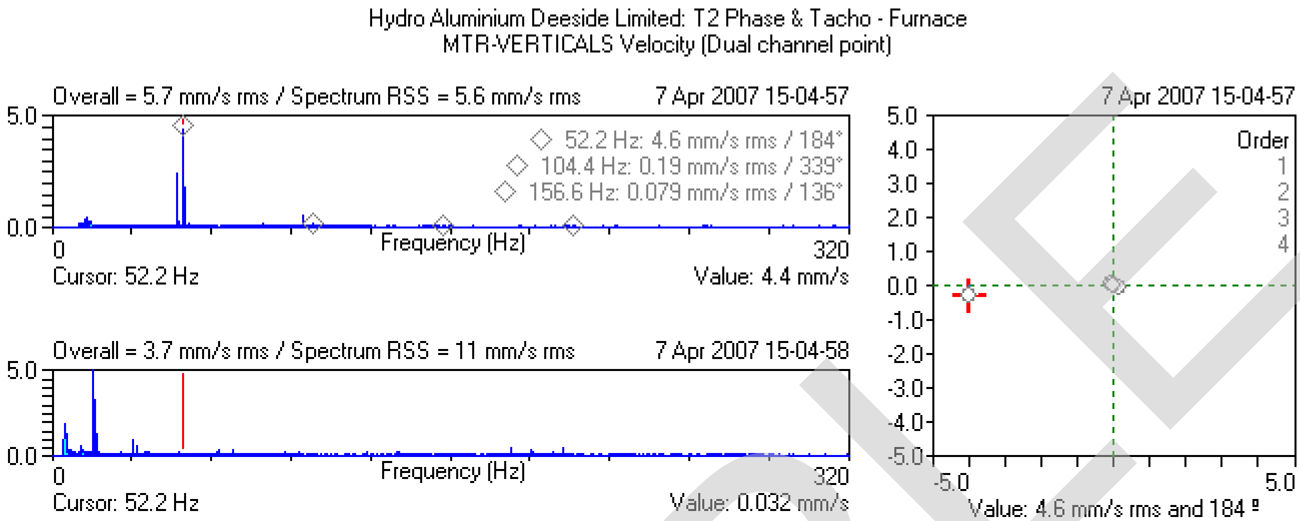

Figure 5: Dual spectra taken from the vertical positions of the motor on the *non drive end* and *drive end* positions, identifying the combined phase relationship between the locations of the two sensors.

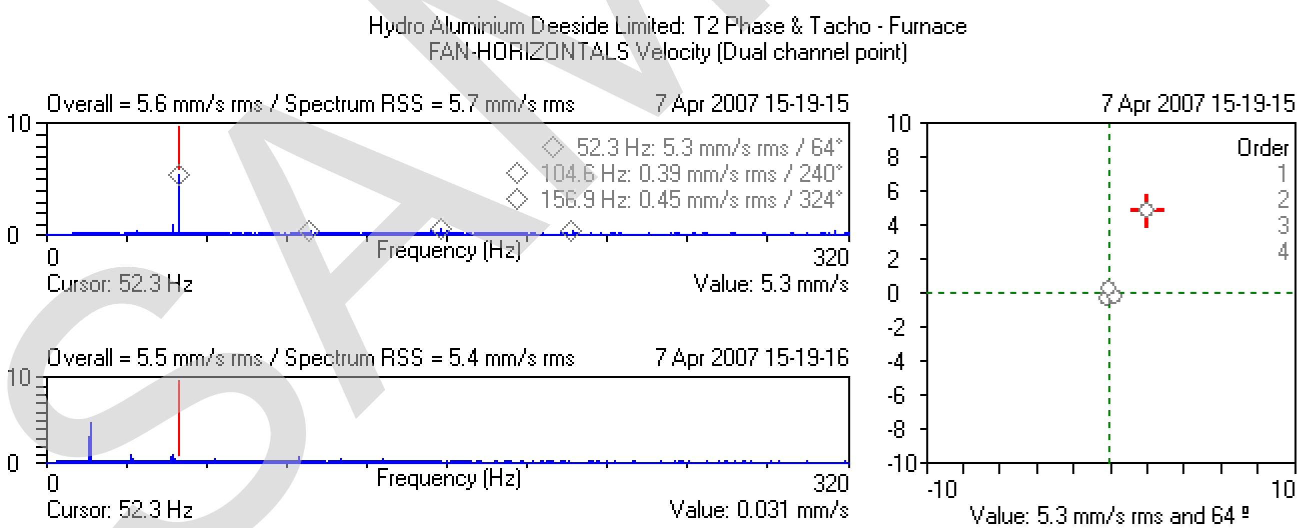

Figure 6: Dual spectra taken from the horizontal positions of the fan bearings on the *non drive end* and *drive end* positions, identifying the combined phase relationship between the locations of the two sensors.

Test 4 - Coast Up/Down Real-Time Analysis

During this final test, the Furnace Exhaust Fan located at Example Company was powered up/down and real-time data collected in the same monitoring positions as in Test 3, with a tachometer speed reference. The purpose of this test was to highlight the potential of any speed related *shaft critical* vibration and induced resonance vibration at specific frequencies, as well as identifying any random elements, which can often be missed using FFT spectral analysis in isolation.

It was confirmed in Test 2 that a *vertical* imbalance was present on the fan shaft and Test 3 concluded that there were two types of imbalance affecting the installation; a 'couple imbalance' on the motor and 'force imbalance' on the impeller. The results of the Phase and Dual Cross Channel Analysis tests are documented below in Table 2:

Test 4 - Coast Up / Down Analysis			
Monitoring Position	Coast	Max.	Remarks
Motor NDE Vertical	UP	15.70 g peak	Cold start; evidence of severe rotor eccentricity
Motor NDE Vertical	DOWN	1.76 g peak	1x RPM; BPF and RBPF dominates; stable waveform.
Motor NDE Horizontal	UP	1.55 g peak	Some low activity but stable throughout
Motor NDE Horizontal	DOWN	0.919 g peak	Stable waveform
Motor DE Vertical	UP	1.57 g peak	Stable waveform
Motor DE Vertical	DOWN	1.81 g peak	Clear impacts on coast down
Motor DE Horizontal	UP	1.95 g peak	Stable waveform
Motor DE Horizontal	DOWN	1.53 g peak	Stable waveform
Fan DE Vertical	UP	4.50 g peak	Unsettled waveform; note increase in amplitude
Fan DE Vertical	DOWN	2.59 g peak	Clear impacts on coast down
Fan DE Horizontal	UP	3.54 g peak	Impacts observed throughout waveform
Fan DE Horizontal	DOWN	2.13 g peak	Clear impacts on coast down
Fan NDE Vertical	UP	10.90 g peak	Increase in amplitude; unsettled for approximately 6s
Fan NDE Vertical	DOWN	3.05 g peak	Clear impacts on coast down
Fan NDE Horizontal	UP	6.10 g peak	Unsettled waveform and impacts observed
Fan NDE Horizontal	DOWN	3.36 g peak	Evidence of non-linear rotation

Figure 7: Table 2 identifying the phase relationship between locations of two sensors.

<u>KEY TO ABBREVIATIONS USED IN TABLE 2</u>:

RPM: Revolutions Per Minute
BPF: Blade Passing Frequency
RBPF: Rotor Bar Passing Frequency

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The results of Test 4 highlighted an unacceptable *acceleration* amplitude observed on the *motor non drive end* bearing in the *vertical* position, attaining an amplitude of 15.70 g peak during a 'coast-up' analysis. It is thought that the high amplitude witnessed here is the result of operating the installation from a stand-still position to full load (i.e., no soft start) resulting in the rotor taking over 6 seconds to locate its magnetic centre (the results of the waveform equate to rotor bar passing frequency).

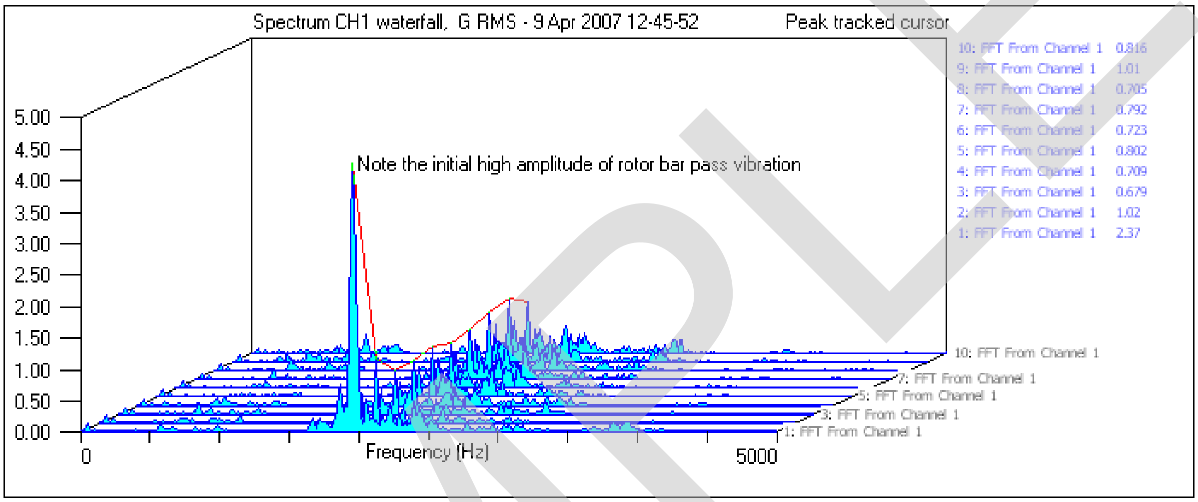

Figure 8a: FFT Waterfall spectrum showing coast up analysis of motor measured on the *non drive end* bearing within the *vertical* plane. The initial levels of rotor bar pass vibration is considerably high in amplitude, but settles to an acceptable amplitude following 6 seconds of operation.

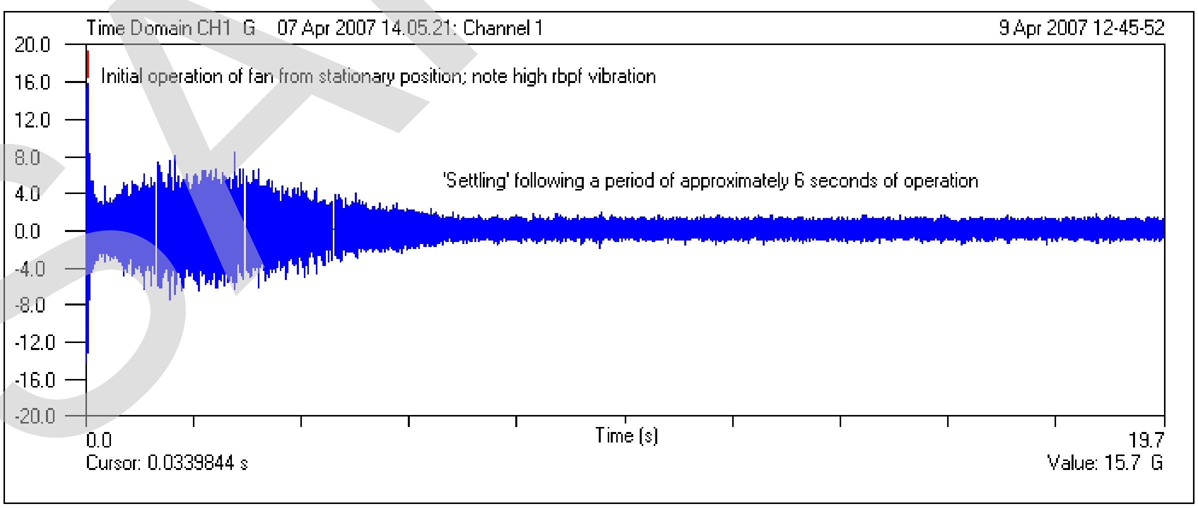

Figure 8b: Time waveform taken over a 19.70 second period collected from the same monitoring position as figure 4a. Note the initial 'unsettling period' (identified as a 'swelling') clearly showing unsteady rotor bar passing vibration.

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In addition to the initial rotor bar passing vibration identified on the *non drive end* monitoring position of the motor, the coast up/down analysis revealed some unsettling on the *fan non drive end* bearing, also over a period of 6 seconds and resulting in high-than-expected amplitudes (10.90 g peak). The aerodynamics of the fan impeller is thought to be attributable to the unsettling period witnessed here and this had been identified as a comparatively high amplitude of *blade passing frequency* vibration.

The opposing *fan drive end* bearing also revealed an increase in levels, albeit at a lower amplitude than the *fan non drive end* bearing. It is also noteworthy that these higher amplitudes have been observed in the *vertical* plane on both fan bearings and the *motor non drive end* bearing.

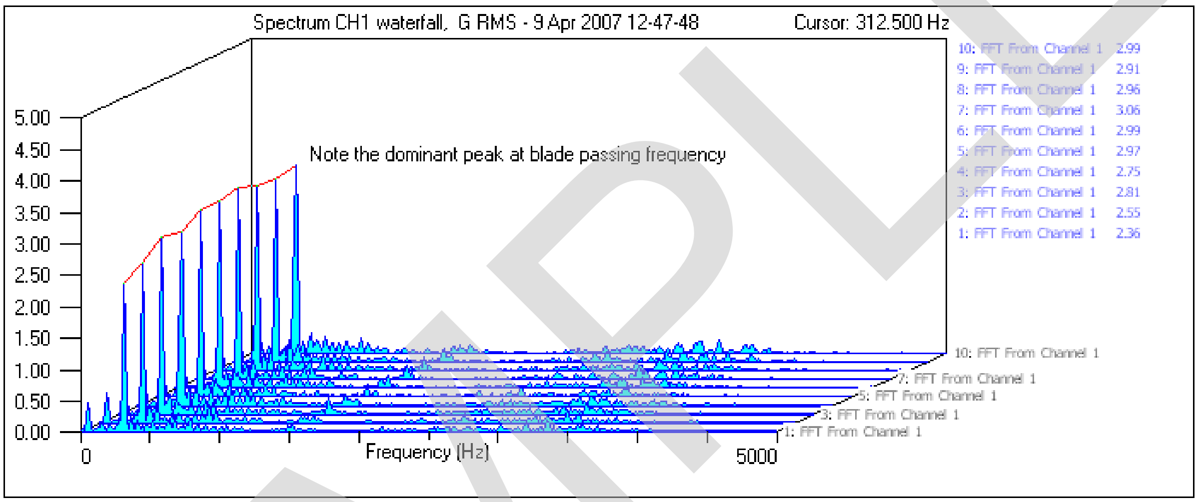

Figure 9a: FFT Waterfall spectrum showing coast up analysis of fan non drive end bearing measured within the *vertical* plane. The initial levels of blade passing vibration are unsettled due to the aerodynamic properties of the installation.

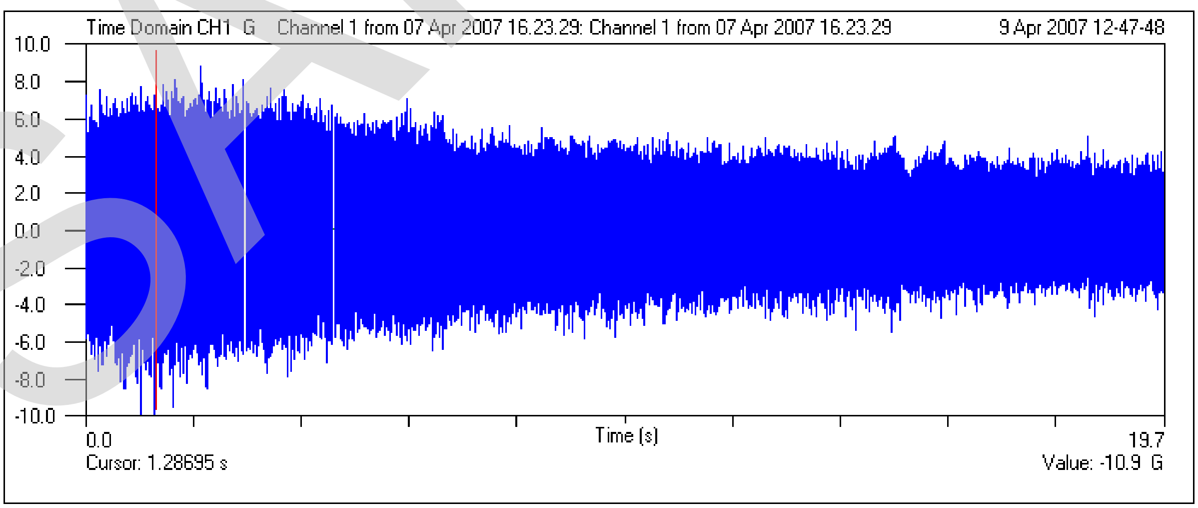

Figure 9b: Time waveform taken over a 19.70 second period collected from the same monitoring position as figure 4a. Note the initial 'unsettling period' (identified as a 'swelling') at the beginning of the waveform.

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Part
2

Discussion

Advice and Recommendations

1 Recommendations for Furnace Exhaust Fan

1.1 Exhaust Furnace Fan - PRIORITY (couple and force imbalance on motor / fan)

The Exhaust Furnace Fan located at Example Company proved to be operating above the '**Unacceptable Zone**' as specified by the Vibration Severity Standard: **ISO10816-3**. The *force imbalance* occurring on the *fan non drive end* bearing attained a *velocity* amplitude of 11.0 mm/sec. at 52.188 Hz frequency within the *vertical* plane, which can be corrected by placing a single balance weight on the impeller.

In addition to the *force imbalance* occurring on the *fan non drive end* bearing, there is a *couple imbalance* occurring on the *motor drive end* bearing within the *horizontal* plane. There is evidence to suggest that the supporting frame below the drive end of the motor is flexing, causing a 180° phase shift between readings; this can be corrected by reinforcing/replacing the supporting frame and ensuring all welds are intact (please refer to Diagram 2 on page 7 for details).

It was noted that the *fan drive end* bearing became extremely hot in a very short period of time. Although the *enveloped spectra* revealed no untoward bearing related problems associated with the installation, it is thought that this heat is being generated by the combination of the *couple imbalance* on the *motor drive end* position (flexing frame) and *force imbalance* of the impeller. The *fan drive end* bearing is in *stress* within two separate planes and could catastrophically fail without little warning. Replacing the bearing per sé would not necessarily cure the problem of heat transfer until the issue of *imbalance* has been resolved on the motor and fan.

During the coast up/down tests, it was noted that the internal rotor took over 6 seconds to settle down from an initial start up period, resulting in some very high amplitudes of *rotor bar passing* frequency vibration before magnetic centre was established. In addition to the internal rotor unsettling period, the fan impeller also revealed higher-than-expected levels of *blade passing frequency* vibration over the same time period of 6 seconds, indicating the two faults may be interrelated (i.e., 6 seconds is the joint time taken for the aerodynamic related properties of the impeller and rotor bar passing elements to settle).

There is a danger of the rotor rubbing against the motor casing in this instance and the case for a soft-start system is should be seriously considered, due to the amplitudes observed here. Likewise, if the impeller becomes eccentric or unstable during start-up, the blades may touch the outer casing, resulting in damaged edges and uneven air gaps.

Continued Overleaf…

The only other comment to make at this stage is the unknown effects of *thermal stratification* on the impeller, particularly when the fan has been shutdown over a period of time. Impellers that are subjected to excessively high temperatures can in some cases cause the impeller blades to bend outward under the influence of centrifugal force, resulting in an uneven mass distribution and force imbalance. The resultant *warping* of the blades cannot be easily rectified and full replacement of the impeller is usually necessary.

It is therefore paramount that the fan impeller should be allowed to rotate freely in the air-stream during a shutdown period, especially if any outlet valves are opened, as the residual heat contained within the furnace may cause the blades to distort under extreme temperatures when the impeller is stationary.

If there is any aspect of this report you wish to discuss in greater detail, or any further points you would like to raise yourself, then please do not hesitate to contact Anthony Riseley on Tel. 01352 710600 or Email sales@dynaseq.co.uk.

Further Remarks

Basic Concepts of Vibration Analysis

What is Vibration?

Introduction

Vibration is the disturbance from equilibrium, which propagates in time from one place to another and exists in all rotating and reciprocating machinery. An ideal machine would produce no vibration at all because all energy would be channelled into the machine function. A good design will produce low levels of inherent vibration, however, as the machine wears, foundations settle and parts deform, subtle changes in the dynamic properties of the machine begin to occur. Shafts become misaligned, parts begin to wear, rotors become unbalanced and tolerances increase. All of these factors are reflected in an increase in vibration energy, which dissipates throughout the machine, excites resonance and puts considerable strain on bearings. Cause and effect reinforce each other and the machine progresses towards ultimate breakdown.

A machine may contain many complex vibrations, made up of a wide-range of superimposed sinusoidal and random components. This multi-complex signal can be broken down into its constituent frequency components by using **F.F.T.** analysis (Fast Fourier Transform) commonly referred to as a **Spectrum**. The following table highlights the many common faults and their characteristic frequencies in terms of rotation speeds:

Nature of Fault	Frequency of Dominant Vibration (Hz=rpm/60)	Direction	Remarks
Rotating members out of balance	1 x rpm	Radial	A common cause of excess vibration in machinery.
Misalignment & Bent Shafts	Usually 1 x rpm; Often 2 x rpm and sometimes 3 or 4 x rpm	Radial & Axial	A common fault.
Damaged rolling element bearings (ball, roller etc.)	Impact rates for the individual bearing component Also vibrations at high frequencies (2 to 60 KHz) often related to radial resonance in bearings	Radial and Axial	Uneven vibration levels, often with shocks. **Impact Rates f (Hz)** Outer Race defect: f(Hz) = n/2 x f_r (1-BD/PD x Cos ø) Inner Race defect: f(Hz) = n/2 x f_r (1+ BD/PD x Cos ø) Ball defect: f(Hz) = PD/BD x f_r [1-(BD/PD x Cos ø)^2] N = number of balls or rollers f_r = relative rev./s between races
Journal bearings loose in housing	Sub-harmonics of shaft rpm, exactly ½ or 1/3 x rpm	Primarily Radial	Looseness may only develop at operating speed and temperature (e.g. turbo machines).

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Oil film whirl or whip in journal bearings	Slightly less than half shaft speed (42% to 48%)	Primarily Radial	Applicable to high speed (e.g. turbo) machines.
Hysteresis Whirl	Shaft critical speed	Primarily Radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by checking tightness of rotor components.
Damaged or worn gears	Tooth meshing frequencies (shaft rpm x number of teeth) and harmonics	Radial and Axial	Sidebands around tooth meshing frequencies indicate modulation (e.g. eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-band analysis and cepstrum.
Mechanical looseness	2 x rpm		Also sub and inter-harmonics, as for loose journal bearings.
Faulty belt drive	1,2,3 & 4 x rpm of belt	Radial	The precise problem can be usually identified visually with the help of a stroboscope.
Unbalanced reciprocating forces and couplings	1 x rpm and/or multiples for higher order imbalance	Primarily Radial	
Increased turbulence	Blade and vane passing frequencies and harmonics	Radial and Axial	Increasing levels indicate increasing turbulence.
Electrically induced vibrations	1 x rpm or 1 or 2 times synchronous frequency	Radial and Axial	Should disappear when turning off the power.

Figure 10: Table 3 identifying common machinery faults and characteristic frequencies in terms of rotational speed.

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Evaluation of Rotating Machine Condition using ISO10816-3 Vibration Severity Standard

The ISO Committee have completely revised the old ISO2372 Vibration Severity Standard for evaluating in-situ performance of rotating machines. The new standard: **ISO10816-3** accommodates the many changes that have taken place in the design and operating frequencies of modern process machinery.

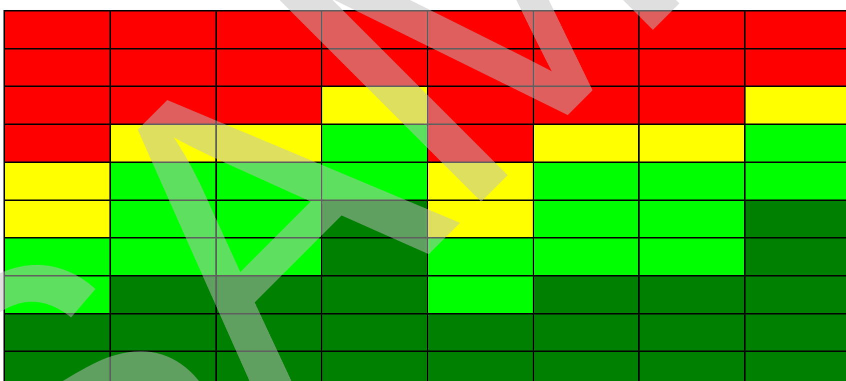
The vibration criteria in this standard applies to the machine sets, with for example steam turbine or electric drives, having a capacity above 15 kW and operating between speeds of 120 RPM and 15000 RPM. Machine sets covered by this standard include: (i) Steam Turbines with a capacity up to 50 MW; (ii) Steam turbine sets with a capacity greater than 50 MW and speeds below 1500 RPM; (iii) Rotary Compressors; (iv) Industrial gas turbines up to 3.0 MW capacity; (v) Pumps of centrifugal, mixed or axial flow type; (vi) Electrical motors of any type and (vii) Blowers or fans, not of lightweight sheet metal construction.

Classification according to Machine Type and Application

Significant differences in the design; type or bearings and support structures requires a separation into different groups. Machines in these groups may have horizontal, vertical or inclined shafts and can be mounted on rigid or flexible supports.

- **Group 1:** Large machines rated above 300 kW; electrical machines with a shaft height H ≥ 315 mm.
- **Group 2:** Medium machines with a rated power above 15 kW up to and including 300 kW: electric machines with a shaft height 160 mm ≤ H ≤ 315 mm.
- **Group 3:** Pumps with multi vane impeller and separate driver, rated above 15 kW capacity.
- **Group 4:** Pumps with multi vane impeller and integrated driver, rated above 15 kW capacity.

ISO10816-3 Vibration Severity Standard



RMS	RMS	Velocity 10 - 1000 Hz r > 600 rpm 2 - 1000 Hz r > 120 rpm
11	0.44	
7.1	0.28	
4.5	0.18	
3.5	0.11	
2.8	0.07	
2.3	0.04	
1.4	0.03	
0.71	0.02	
mm/sec.	inch/s	

rigid	flexible	rigid	flexible	rigid	flexible	rigid	flexible	FOUNDATION
Pumps > 15 kW radial, axial, mixed flow				Medium sized machines 15 kW < P ≤ 300 kW		Large machines 300 kW < P < 50 MW		MACHINE TYPE
Integrated driver		External driver		Motors 160 mm ≤ H < 315 mm		Motors 315 mm ≤ H		
Group 4		**Group 3**		**Group 2**		**Group 1**		GROUP CLASSIFICATION

Figure 11: Table identifying ISO10816-3 Vibration Severity Standard.

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Human exposure to vibration within buildings

Structural vibration in buildings can be detected by the occupants and can affect them in many ways: their quality of life can be reduced as also their working efficiency. Building vibrations as they affect people may be classified as impulsive or continuous as follows:

(a) Impulsive vibration is a rapid build-up to a peak followed by a damped decay which may or may not involve several cycles of vibration. It can also consist of a sudden application of several cycles at approximately the same amplitude, providing that duration is short, i.e. less than 2 seconds.

(b) Continuous vibration is vibration which continues uninterrupted for either a day time period of 16 hours or a night time period of 8 hours, for example 23-00 to 07-00.

When measuring vibration in these circumstances, data is normally acquired on a building structural surface supporting a human body. In some cases, measurements may have to be taken outside the structure, or on some other surface other than the point of entry to the human subject.

The "*evaluation of human exposure to vibration in buildings*" standard: BS 6472 defines the factors which influence human response to repeated and prolonged vibration exposure.

Economic benefits of an effective Vibration Monitoring program

When implemented correctly, vibration monitoring will bring about the following:

- Increase in the average time between overhauls.
- Increased productivity and reduced maintenance costs.
- Virtual elimination of unexpected breakdowns.
- Increased reliability.
- Elimination of secondary damage.
- Elimination of component waste (no replacement or serviceable components).
- Reduction of spares and stock.
- Reduction in business interruption and damage insurance premiums.
- Reduced repair duration.
- Improved product quality.
- Downtime may be scheduled.

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