VIBRATION STUDIES AT THE ESRF

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Abstract

At the ESRF, significant efforts have been made to identify vibration sources, assess their effect and reduce their impact on the site in order to improve the stability of the electron beam. By using various focusing elements, X-ray beam can be focused to sub-micron size. Therefore the X-ray stability is becoming a key issue. This paper presents the vibration studies carried out at the ESRF, including vibration instrumentation, vibration sources identification, vibration damping for machine girder and beamline mirror support, vibration correlation study between X-ray beam and beamline optics, active vibration isolation systems.

Keywords: vibration, beam stability, vibration damping, vibration instrumentation, active vibration isolation

1 Introduction

The e-beam and X-ray beam stability is one of the most important requirements for 3rd generation synchrotron light sources. Significant efforts have been made during the construction and the operation of the ESRF machine to minimise internal and external vibration sources as well as the vibration responses of the girders and quadrupole magnets. Ground vibration has been permanently monitored with a seismic recording network implemented at the ESRF as early as the construction phase [1]. A great amount of vibration measurements have been made to identify vibration sources. Counter-measures were taken to suppress them or reduce their impact. These efforts resulted in an emittance growth of a few percent [2]. Since the start of operation of the ESRF, the vertical emittance has been reduced from a design target value of 0.62 to 0.025 nm.rad, the horizontal emittance has also been reduced from 6.2 to 4 nm.rad.– a routine operation value [3]. With the reduction of the beam size, beam stability has become a more and more important parameter. Damping devices, called ‘damping links’, have been developed to attenuate the resonant vibration of the quadrupoles and girders and installed in the ESRF storage ring. The fundamental resonant vibrations of the magnet girder assemblies have been effectively attenuated by a factor of 5.8 [4,5]. Thus, the electron beam stability has significantly improved. In parallel, the requirement for X-ray beam stability has become more and more demanding with the progress in focusing techniques. Today, hard X-ray beam can be focused to about 100 nm X 100 nm using K-B mirrors for instance [6,7]. The X-ray beam stability related to the sample and optical elements should be better than the focused beam size. Less than 100 nm in stability is required in some experiments. This is challenging for mechanical engineers in view of the typical vibration level on the ESRF ground of about 1 micrometer peak-to peak in the frequency range 1-100 Hz.
This paper reviews stability related issues at the ESRF: ground vibration, vibration source identification, damping device for machine girder, vibration study on beamlines including vibration correlation study between beamline components and X-ray beam, modal testing of mirrors and vessels, damping link for mirror and vessel, liquid nitrogen and nitrogen gas induced vibration on the monochromators.

2 Vibration instrumentation

Vibration measurement instruments essentially consist of three parts: sensors, data acquisition system (DAS) and data processing systems.

2.1 Sensors

Frequently used vibration sensors are geophones (which measure vibration velocity), accelerometers. There are active or passive sensors. Active sensors need electric power for the operation of the included electronic equipment. The key parameters of the sensors are frequency bandwidth, transduction constant, resolution, dynamic range, geometrical size and weight….Table 1 summarizes various sensors available at the ESRF. Most of the measurements are performed in the frequency range 1 to 100Hz as detailed in the ground vibration section of this paper. For that purpose the L4-C geophone from Mark Products is often chosen. The velocity signal can be integrated once in the time domain to give the displacement.

Table 1. Different sensors at the ESRF with their respective utilisation.

Type (measurement)	Model	Bandwidth Hz	Weight kg	Utilisation
Active Geophone (velocity)	Guralp 3ESP	0.033 - 50	10	Measurement of ground vibrations, effects of low frequency phenomenon such as quasi static floor deformation.
Passive Geophone (velocity)	Mark Products L4-C	1 - 100	1	Measurement of ground vibration, operating response of structures
Passive Geophone (velocity)	Mark Products L22	2 - 200	2	Measurement of ground vibration, operating response of structures
Accelerometer (acceleration)	Various models	0 - 5000	0.03 – 0.15	Modal analysis of structures, operating responses on small components
Laser vibrometer (displacement and velocity)	Polytec OFV-512 dual fibre	0 - 20000	N/A	Non-contact measurement of operating responses of small and difficult to access components.

2.2 Data Acquisition Systems (DAS)

The function of a DAS includes possibility of amplification and filtering of signal from sensors, analogue to digital conversion (ADC) of this signal, data storage, and eventually signal visualisation, treatment and data processing. Key parameters of a DAS are dynamic range (presently 16 bit or better), sampling frequency, LSB (Least Significant Bit which measures the “sensibility” of the digitalisation), number of channels, time synchronisation, data storage capability, eventual data processing functionality…. At the ESRF, three systems are used.

- a 4 channel HP35670 spectrum analyser, 16bit (90dB) dynamic, maximum sampling frequency 51.2 kHz, internal DSP for FFT analysis.
- Two 4 channel SIGLAB systems, 16bit (90dB) dynamic, maximum sampling frequency 50 kHz, internal DSP for FFT analysis. These units can be coupled to give 8 simultaneous inputs.
- Five 3 channel RefTek 130-01/3 systems, 24bit (120dB) dynamic, maximum sampling frequency 1000 Hz. These units can be synchronised in a network (more on this in the next section of this paper)

All these DAS’s are easily transportable. The RefTek units can be used as standalone devices for long measurement campaigns (several days) thanks to their internal hard drive.

2.3 Seismic survey system

A seismic survey system based on five RefTek 130-01/3 DAS's has been implemented at the ESRF to continuously measure the ground vibration and the electron beam motion in the frequency range of 1-100 Hz. Fifteen sensors are used in this survey system, which include L4-C geophones mounted around the storage ring tunnel and on two quadrupole magnets of the storage ring, and a number of Beam Position Monitors (BPM). By using a GPS receiver, each DAS has a precise time reference for synchronisation of the data acquisition. All units are connected to a server via ethernet. The server includes a database, an archive containing recorded data corresponding to about one week of continuous recording. Data processing and storage are also performed on this server computer (Fig.1). Vibration noise varying with time, and vibration events such as earthquakes, impact of trucks on irregularities on the roads near the ESRF site are recorded. Measurement data can be monitored from any office at the ESRF using client software. Spectra versus time for any one sensor can be visualised, as well as peak-to-peak displacement versus time for all sensors.

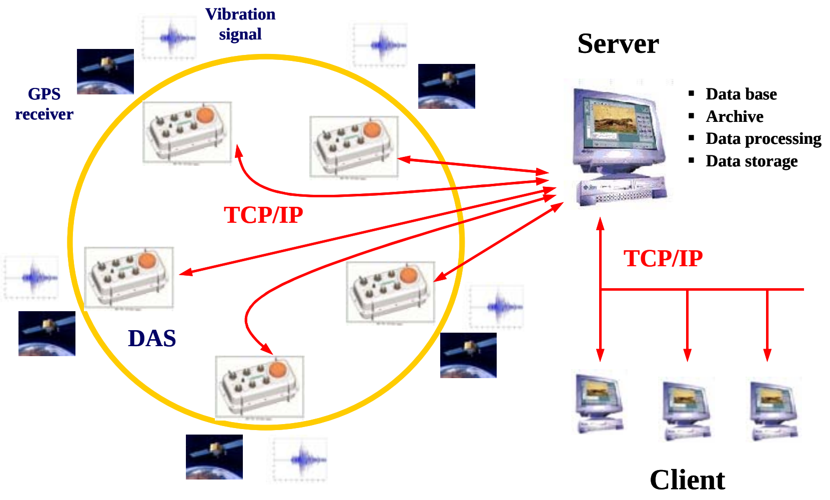


Figure 1: Reftek seismic survey network

3 Ground vibration and vibration sources

3.1 Ground vibration

The ground vibration is usually divided into three frequency ranges: low (f<1 Hz), intermediate (1-100 Hz) and high (f>100 Hz) frequency ranges. In the lower frequency range (f<1 Hz), vibrations are essentially due to ocean waves and micro earthquakes. They are characterised by two peaks centred at 0.14 Hz and 0.07 Hz. The peak around 0.07 Hz is considered to be due to the action of ocean waves on coasts. The peak around 0.14 Hz is due to the pressure from standing ocean waves, which may be formed by waves travelling in opposite directions in the source region of a storm or near the coast. This mechanism generates seismic waves with a frequency twice that of ocean waves [8]. With a typical wave propagation speed of 700 m/s in a sand-gravel soil, the wavelengths are about 5 km and 10 km respectively. The good spatial coherence of this micro-seismic noise within a couple of hundreds of meters results in a negligible differential vibration at any two points in a modern synchrotron radiation facility (diameter<500m) [9]. The intermediate frequency range (1-100 Hz) is the most interesting at the ESRF because the fundamental resonant frequencies of the

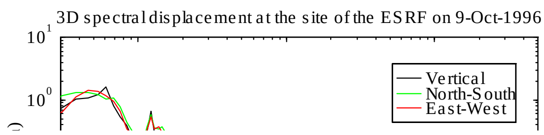


Figure 2: 3D Spectral Displacement of the ground vibration at the ESRF site.

mechanical components usually lie in this frequency range. Therefore, the accelerator

machine is sensitive in this frequency range. Vibration sources are road and train traffic, the operation of heavy machines, the river water flow, the wind, and so on. In the higher frequency range (f>100 Hz), the ground vibration level is much smaller than in the intermediate frequency range because this vibration, mainly generated by small electromechanical devices, is not powerful enough to induce significant ground vibrations. At the ESRF, the seismic survey system described earlier, permanently monitors ground vibration in the intermediate frequency range. This is occasionally extended to the low frequency range. Fig.2 shows a typical spectral displacement in vertical, North-South, East-West directions for the frequency range 0.033-50 Hz. The vibration spectra in the three perpendicular directions are quite similar. Only some differences around the 3Hz peak and between 0.2 and 0.4 Hz can be observed. The peak at 3Hz is a characteristic and frequently observed feature of the ESRF site located in Grenoble. According to advanced geophysical studies, there is a fundamental resonant frequency of 0.3Hz in the central part of the Grenoble basin, and another resonant frequency at 3Hz assigned to a thin surface layer of the ground [10, 11]. The peaks around 3 Hz and between 0.2-0.4 Hz are due to the resonance of the ground in the valley.

The ESRF is located near the junction of two rivers (Isère and Drac) and between two motorways. Road traffic and industrial activities generate considerable vibration noise, fluctuating with time. Fig.3 shows the peak-to-peak displacements versus time at four points around the storage ring for a period of one month. Each value of peak-to-peak displacements was calculated in the bandwidth 1-100Hz over a window of 8.192 seconds. The vibration level clearly varies with time. Generally, the vibration level increases during the day and decreases at night roughly in a sine shape with a 24 hours period. The weekend is quieter than the five working days. There is also a relatively quieter spell between 12:00-13:30 corresponding to the lunch break. It can be remarked that the vibration at cell 28 is significantly higher than at other places. In the next section, we will see that this higher level was due to the road traffic on the nearby road, Avenue des Martyrs, nearest to cell 28. The road was repaired in December 1996, and subsequently, the vibration level at cell 28 is now comparable with other locations on the ESRF site.

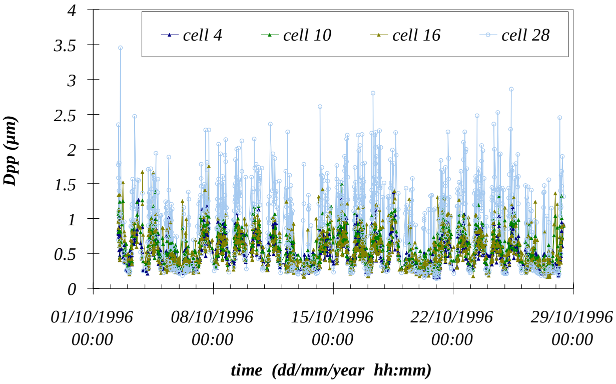


Figure 3: Peak-to-peak displacement of the ground vibration on the ESRF site during one month.

To keep the stability within the tolerance criteria, the ground vibration level at the ESRF should not increase. The vibration sources, both internal and external to the site, have been intensively studied during the construction and the operation of the ESRF. Vibration sources were first identified. Countermeasures were then designed, tested and implemented.

3.2 External sources

Although earthquakes are common, only a handful per year reaches strong enough magnitude and proximity to be significant for the ESRF. Fig.4 shows an example of such an event. Earthquake response on the ESRF site could be up to 100 times the usual vibration noise level. Most of the other external vibration sources near the ESRF have been studied. They are usually linked to human activities nearby: speed bump at the exit of the

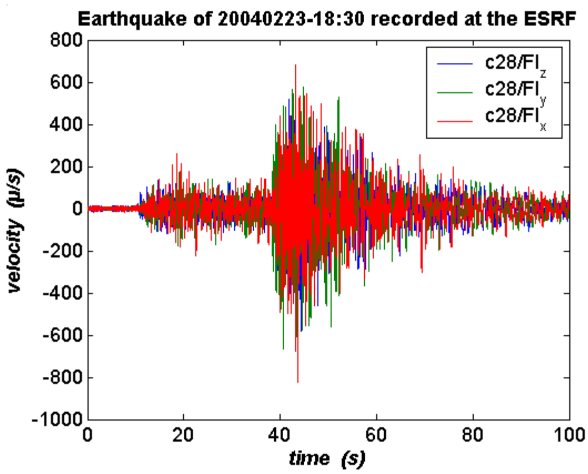


Figure 4: Earthquake magnitude 5.1, ENE of Besançon, 23/02/2004 at 17:31 UTC. Vertical velocity on the floor at the ESRF

motorway, bridges across the two rivers, speed bumps on site, traffic (due to trains, buses, trucks, trolley-bus,…), heavy machines (compressor, pumps, electric-heat co-generator,…).

From long periods of measurement observations previous December 1996 (as shown in Fig.3), it was found that more than 95% of strong vibration events (peak-to-peak displacement >1.5 µm) on site are induced by the passage of heavy vehicles over road irregularities. For example, sewer covers on the Avenue des Martyrs, a local road. Various vibration measurements have been carried out to identify these vibration sources. By placing 10 geophones along this road in the section near the ESRF site, vibration measurements were carried out simultaneously with the triggering of the geophones at its normal position at cell 28. The temporal displacement of a typical event is shown in Fig.5. The sensor No7 is the geophone at its usual position in cell 28 around storage ring and used for triggering measurements. The recorded data showed that the perturbation started near the sensor No4 and propagated to the other sensors with a decrease in amplitude. This clearly suggested that the vibration sources came from near sensor No4. Further investigations revealed the clear correlation of these vibrations with the passage of trucks or buses over a sewer cover on the road near sensor No4. Once this identification was established, a technical proposal for road repairs was submitted to the relevant authority. After road repair and re-arrangement of lanes, the vibration near cell 28 has been reduced to a level comparable to other place on the ESRF site.

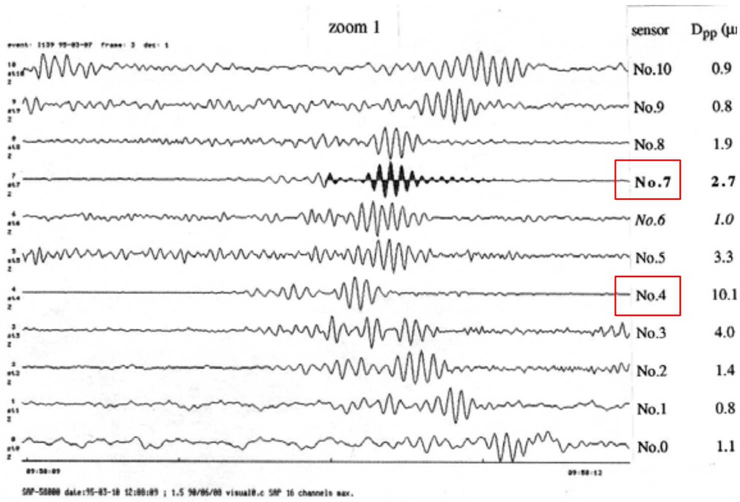


Figure 5: Temporal displacement of a typical event when a truck or bus passes over a sewer cover on the avenue des Martyrs. All sensors were placed along that road in the section near the ESRF site except sensor No7.

Vibration impacts of speed bumps near the ESRF site were also investigated. Some of them inducing significant vibrations at the ESRF site were modified or suppressed.

3.3 Internal Sources

Vibrations generated by various accelerator components have been intensively studied. Some internal sources are the High Quality Power Supply (HQPS), power supply and transformer, the cooling water flow of magnets and thermal absorbers, operation of front-end shutters, the air conditioning units, and the overhead-crane.

With the surrounding Alps, Grenoble has frequent thunderstorms during the summer months. Fluctuations or micro-interruption of the mains caused by the thunderstorms can lead to electron beam loss. An HQPS system [12] was installed at the ESRF to smooth these micro-interruptions. The HQPS consists of ten units of 1MW each with rotating equipment: an alternator, an accumulator and a diesel engine. The vibrations induced by the functioning of these powerful alternator-diesel engines have been carefully controlled. Isolation systems for the engines and for the buildings have been implemented to reduce the vibration. Measurement results show that the vibration amplitude at 30 m from the HQPS building is less than 1.3 µm peak-to-peak, compared to the 1.0 µm when all units are off. The vibration influence on the accelerator machine located at least 100 m far from the HQPS building is negligible.

Rubber dampers were used at the fixations of the cooling water tubes to limit the vibration induced by the cooling water flow. Cooling water flow in magnets and thermal absorbers

generates vibrations above 20 Hz. Fortunately, this is far from the first natural frequencies (7-13 Hz) of the magnet girder assembly. Although the vibration amplitude of magnets above 20 Hz is much higher than the ground vibration, it was still about 10 times lower than the peak at the first natural frequency (Fig.6) before the installation of damping links.
To ensure temperature stability, a great number of air conditioning units have been installed in the storage ring tunnel and in the experimental hall. The air conditioning units in the experimental hall and in the central building generate a very sharp peak at 16.4 Hz. The contribution of this peak in the wide band (1-100 Hz) displacement is negligible.
The vibrations induced by a 6 tons overhead-crane in motion are significant in the frequency range 30-80 Hz for both the storage ring and experimental hall slab. The amplitude in this frequency range is at maximum 10 times higher than the ambient noise. In addition, the quasi-static deformation of the floor under the weight of the crane might affect some experiments on beamlines. Consequently, the use of the overhead-cranes is simply not allowed during machine operation and beamline experiments.

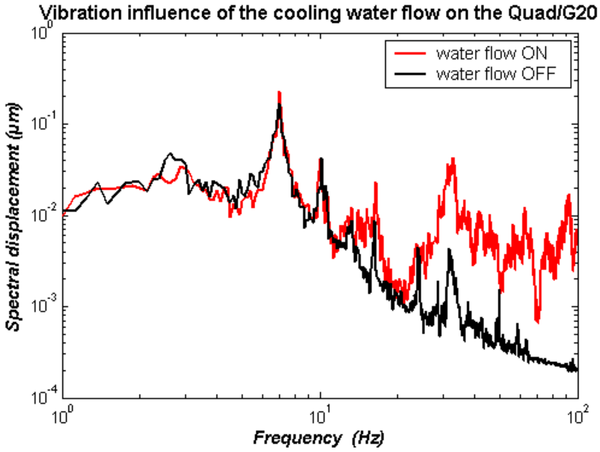


Figure 6: Spectral displacements of a Quadrupole magnet with and without cooling water flow before the installation of damping links

The opening and closing of the front-end shutter by fast pneumatic valves had induced very strong vibrations on the magnets and on the girder. The peak-to-peak displacement of a quadrupole could be increased from 3.2 µm (noise) to 74 µm in the lateral direction, and from 0.9 µm (noise) to 21 µm in the vertical direction during the fast closing (50 msec). It was identified that the fast closing affects the e-beam stability, especially in the case of single bunch mode. The slow closing (200 msec) generates a vibration amplitude on the quadrupole six times lower than the fast closing. This mode was implemented. Analysis of the vibration response of the quadrupole G20 during the fast closing of the front-end shutter showed that the magnet-girder assembly was excited in the lateral direction at 7 Hz, which was the first natural frequency. The excitation lasted about 5 seconds. The modal damping ratio has been roughly estimated at 1.6%, which is very low. After the installation of the damping links and modification of the opening and closing mechanism of the front-end shutter, the vibrations induced by the action of these shutters were significantly reduced. It should be noted that the opening and closing action is occasional, the vibration impacts are limited both in duration and in the vicinity of the shutter.

4 Damping devices for machine girders

The fundamental resonant vibration mode of the ESRF quadrupole magnet girder assemblies was a horizontal rocking motion at about 7 Hz [13, 14]. This was the origin of the dominant motion of the e-beam, and of the X-ray beam instability. A damping device, the so-called ‘damping link’, has been developed to attenuate the vibrations of the magnet girder assemblies.
The damping link design consists of adding a ViscoElastic link between the girder and the floor. It consists of three parts (Fig.7): (1) a sandwich structure with aluminium plates and ViscoElastic Material (VEM) (Al + VEM + Al), (2) a girder mounting fixture (GMF) linking the sandwich structure to the girder, (3) a floor mounting fixture (FMF) linking the sandwich structure to the floor. The idea was to use the sandwich structure with VEM to absorb the dynamic strain energy of the MGA related to the rocking motion. The damping links were installed on the two extremities of the girder and floor (as shown in Fig.7) in parallel with the existing jacks. Therefore the required lateral stiffness was maintained. This installation allowed attenuation of both lateral rocking motion (1st mode) and horizontal rotation around the vertical axis at the centre of the girder at about 13.6 Hz (3rd mode) [13, 14]. The mounting fixtures (GMF, FMF) should both accommodate the environment in the tunnel and be stiff enough to transmit maximal dynamic strain energy of the MGA to the VEM layer which then dissipates this energy. The VEM

sandwich was optimised to attenuate the 1st resonant vibration with an operation condition tolerating up to 2 mm shear displacement in the vertical direction. This 2mm displacement corresponds to the maximum possible accumulated stroke required by alignment for two years. Test results show that the MGA with the damping links could be adjusted 2mm in vertical and lateral directions, and that the damping performance is not degraded by this amount of adjustment. The damping links are fully compatible with the alignment operation [15].

A complete installation of damping links in the ESRF storage ring was performed after the March 2001 shutdown. The fundamental resonant vibrations of the magnet girder assemblies have been effectively attenuated by a factor of 5.8 [5]. The RMS displacement of the electron beam in the horizontal direction has been reduced from 10 to 2.7µm in the bandwidth of 4-12 Hz. Power Spectral Density (PSD) of the horizontal displacement of the electron beam is shown in Fig.8. Before the installation of the damping links, there was a huge peak at 6.8 Hz in the horizontal displacement PSD. When the storage ring was totally equipped with damping links, the peak at 6.8 Hz in the PSD was dramatically attenuated by a factor of 49.

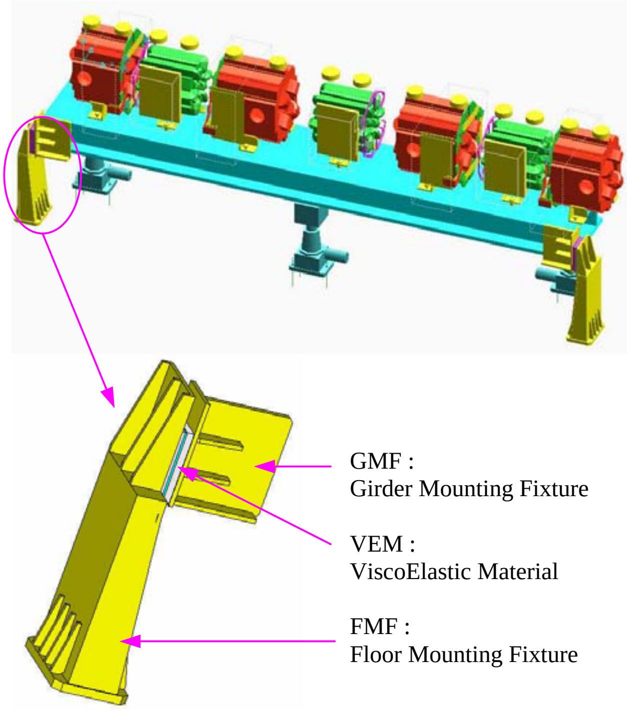


Figure 7: Damping link and installation on a G20 magnet girder assembly in the ESRF storage ring

A wide peak around 30 Hz was also observed on the PSD. The damping links have no effect on that peak. This is because this 30Hz peak is due to the lateral rocking motion of the quadrupole QF2 (or QF7) relative to the girder. The resonant motions of the quadrupoles QF2 and QF7 at 30 Hz are excited by the water flow in the cooling circuits. As the girder itself does not move at this particular mode, the damping links have neither beneficial effect for the vibration of the quadrupoles, and therefore nor for the motion of the electron beam around 30 Hz. Some countermeasures to reduce the vibrations of quadrupoles QF2 and QF7 have been studied by finite element simulation, and could prove very effective.

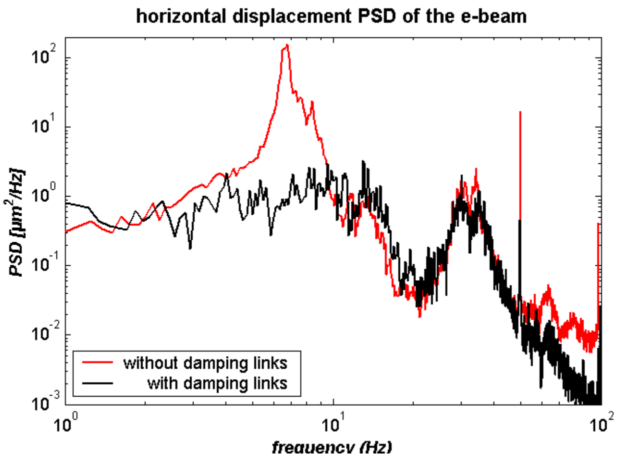
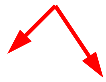


Figure 8: Horizontal displacement PSD of the electron beam before and after the installation of the damping links in the storage ring.

5 Vibration of beamline components

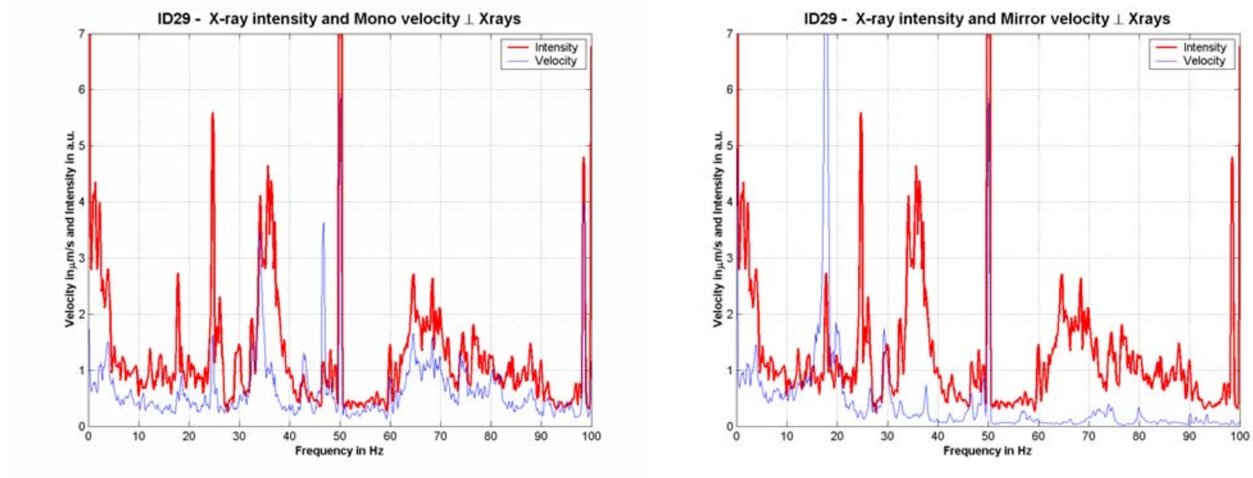
In parallel to the significant improvement of the electron beam stability, the X-ray beam stability becomes more and more demanding with the huge progress in X-ray optics, particularly in focusing techniques. The applications with small sample and small size of X-ray beams increase constantly. Micro-focusing beam and micro-tomography tend to nano-focusing beam and nano-tomography. The stability of the X-ray beam and beamline components (essentially optic components and sample environment) is crucial for the success of the experiments.



The stability of the supports of optical components such as mirrors and monochromators is essential for most ESRF beamlines. A thorough investigation of the vibrations can lead to the dynamic characterisation of these elements. For instance, information such as structural resonances and local vibration sources can be extracted. In addition, the correlation of the X-ray beam intensity with the vibration data usually reveals important clues. This is then used as the basis for mechanical modifications or specific operating conditions, e.g.: for pumps. To illustrate various types of vibrations studies carried out on ESRF beamlines, we present some vibration investigations for ID29 beamline [16]. This is a macromolecular crystallography beamline where multiple anomalous diffraction (MAD) experiments take place. Most of the following discussion will focus on the mirror structure (Fig.9). This is composed of a vacuum vessel mounted on a steel frame clamped to the concrete floor. The mirror is mechanically de-coupled from the vessel, its vertical position being adjusted via three motorised jacks.

5.1 Vibration characterization and correlation with X-ray intensity

The 1st step consists of an operating response measurement (ORM) where geophones are placed on strategic positions on the structure. In this case, the top of the vacuum vessel was selected and the floor, acting as a reference. This allows the extraction of vibration amplitudes, peak frequencies and amplification with respect to floor (Fig.10). For example, a horizontal vibration mode of the mirror structure is clearly visible at 18Hz. This relatively low frequency indicates a lack of rigidity of the mechanical assembly in this direction. Since the mirror is actually not directly linked to the vessel, there could be other more localised modes at different frequencies. To investigate the influences of the mechanical vibration of the mirror, X-ray beam intensity fluctuation was measured with a photodiode exposed to the beam at the sample position.
This measurement was simultaneously carried out with the vibration on the structures.



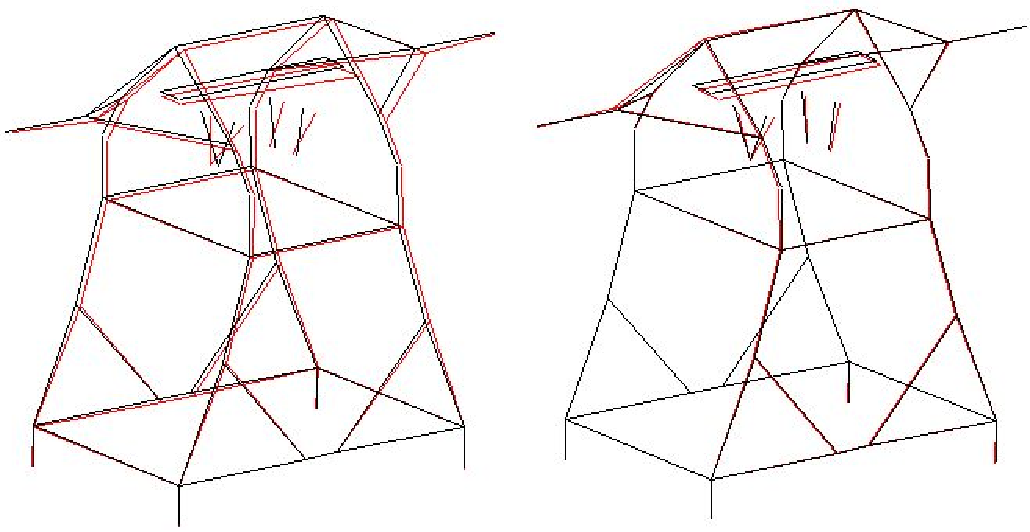
(a) (b)

Figure 12: Spectrum of the X-ray beam intensity and spectral velocity monochromator (a) and mirror (b)

Coherence of the X-ray beam intensity with the structure can be calculated. Fig.11 shows this coherence with monochromator, mirror vessels and experimental table. The high coherence with the monochromator vibration in the frequency range 35-40Hz and 60-80Hz reveals the vibration influences of the monochromator on the X-ray beam stability. One can remark the mirror contribution at 18Hz (Fig.11). Indeed, a direct comparison of the frequency spectra confirms these observations (Fig.12).

5.2 Modal testing of the mirror and vessel

In the light of the above measurements it was decided to concentrate further studies mainly on the mirror structure. As pointed out earlier, the mirror itself is de-coupled from the supporting structure. Therefore, a modal analysis was performed with measuring points on the actual mirror to extract more information about its dynamic behaviour. This analysis was aimed mainly at understanding how the structure deforms at its main resonance frequencies rather than extracting



(a) (b)

Figure 13: 1st (a) and 4th (b) horizontal vibration modes of the mirror structure. The deformed shapes are represented in red

quantitative modal parameters, for finite element modelling for instance. The procedure consists in hitting the structure with an impact hammer thus providing an excitation force over a sufficient frequency range. The response of the structure at carefully selected points is recorded by a 3D accelerometer. Since the response in acceleration is normalised to the input force, the measurements are performed sequentially. Ultimately, a complete grid of points is taken from which the spatial resolution determines the actual modal shape accuracy.

Results once processed and displayed in a user friendly format, showed both rigid body vibration modes in the lower end of the frequency spectra and local modes of the mirror at higher frequencies. For example, for the 1st horizontal resonance at 18Hz, all points on the structure move in phase perpendicularly to the X-ray beam (Fig.13a). Hence this is a rigid body mode underlying the relatively low stiffness to mass ratio of the mechanical assembly and in particular the feet to floor interface. In contrast, the 4th horizontal mode at 29Hz is a local motion of the mirror (Fig.13b). The mirror supports highlighted in red on Fig.13 are the weaker stiffness points.

Since the 18Hz mode is by far the mode with the greatest amplitude, it was decided to improve the stability of the structure by means of improved rigidity.

5.3 Improving the structure behaviour using Damping Link

In order to modify the vibration behaviour of the structure, one can play with the stiffness, the mass and the damping characteristic of the whole structure or selected components. In view of the very good results obtained with the damping links on the machine girders (cf. Section 4 of this paper), a similar solution was chosen. However, due to the limited space available on the floor, the damping links were connected from the concrete wall next to the structure (Fig.9). The improvement in stability is fairly obvious when looking at Fig.14, the 18Hz mode has vanished. A much weaker peak near 24Hz has now replaced it.

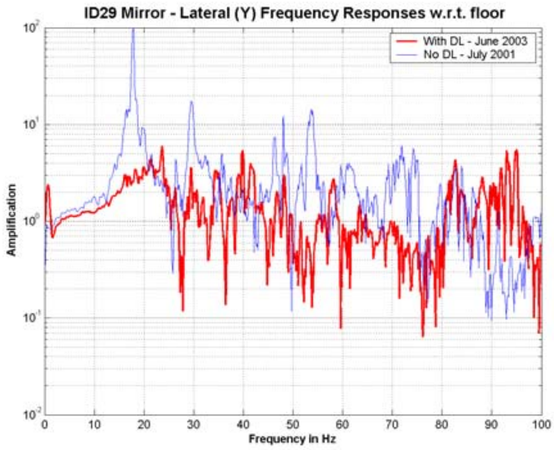


Figure 14: Horizontal frequency response spectra showing the improved stability of the ID29 mirror structure after damping link installation.

5.4 Liquid nitrogen induced vibration on ID22 monochromator crystal

Liquid nitrogen (LN_2) cooling is mostly used in the silicon monochromator for 3rd generation light sources [17-20]. To ensure an efficient cooling, the LN_2 is generally in turbulent flow in the cooling channels. To investigate the vibration induced by the LN_2 flow and optimise the function of the LN_2 pump, vibration of the ID22 monochromator crystal was measured by using a tiny accelerometer at variable operating frequency of the LN_2 pump. Results (Fig.15) show large increase of the vibration level when the operating frequency of the LN_2 pump is higher than 40 Hz. This sudden deterioration of the stability of the crystal is likely due to turbulent flow of the LN_2 since the frequency analysis shows a strong wide band noise rather than sharp spectral contributions linked to the pump frequency. Therefore the LN_2 pump speed was set at a value both compatible with efficient crystal cooling and away from the turbulent flow regime.

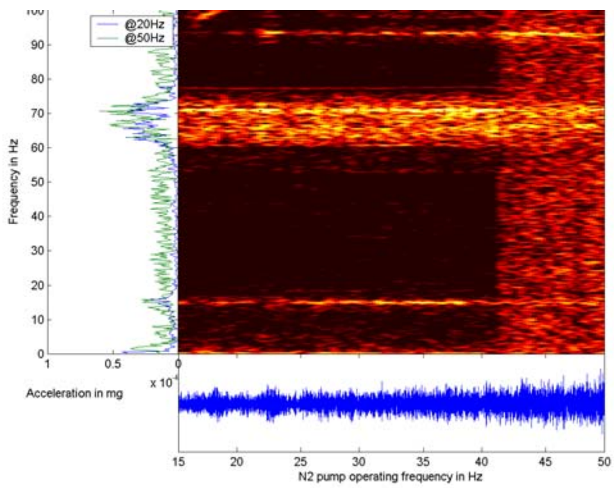


Figure 15: Spectrogram representation of the acceleration level measured on the 1st (upper) monochromator crystal. The cooling LN_2 flow rate was linearly ramped up by increasing the pump rotation spe expressed in Hz here.

5.5 N_2 gas flow induced vibration on ID21 Monochromator

When a vibration measurement is required inside a vacuum vessel on a sensitive optical element, even a small accelerometer might not offer a sensible solution. In that case, a laser vibrometer offers a valid alternative of a non-contact measurement. An in-situ vibration measurement of the

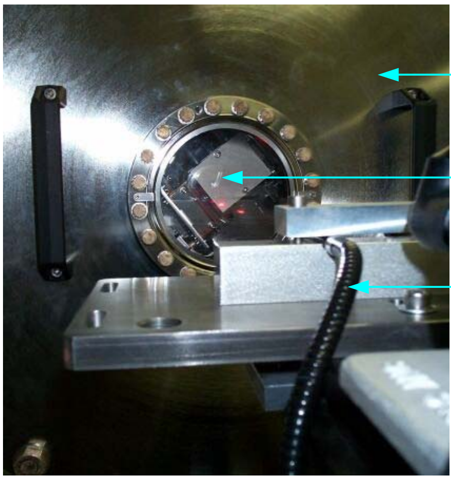


Figure 16: Vibration measurement of the monochromator crystal by laser vibrometer

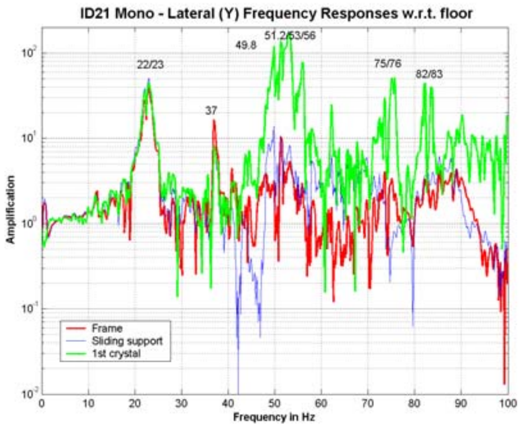
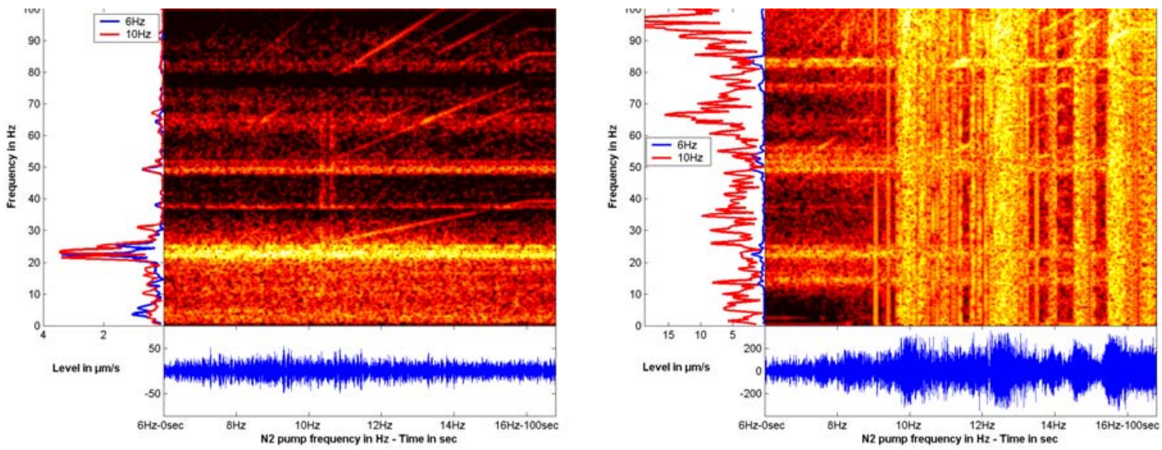


Figure17: Horizontal frequency responses of the monochromator crystal compared to other locations on the structure.

N_2 gas cooled crystal using a Polytec laser vibrometer was performed on the ID21 monochromator in order to investigate the influence of the gas circulation (Fig.16). The dual fibre laser vibrometer was used here with one measuring fibre (the other one being the optical path reference). This measurement was combined with geophone data recorded on other points of the structure, namely the frame and the mechanical structure supporting the crystals. In the lateral direction (⊥ to X-ray beam), the 1st mode was found to be at 22Hz and coupled to a 2nd mode at 23Hz. The modal shape although not actually measured is likely a translation along the measurement direction (likely with rocking component).



(a) (b)

Figure 18: Spectrogram of the velocity measured on the monochromator vessel (a) and on the crystal (b). The horizontal axis represents time (proportional to the cooling pump frequency) while the vertical axis is the frequency of the measured signals. PSD spectra at low and high instability time slices are also shown in red and blue (vertically oriented)

All the components either structural or the monochromator crystals have a similar response. Fig.17 confirms that the frequency response spectra with respect to floor are very similar up to 40Hz at the three different positions. But the laser vibrometer measurement on the crystal shows additional modes or forced vibration components at higher frequencies. These must be the result of local vibration excitation on the crystal itself. The influence of the N_2 gas flow was also investigated. The frequency of the pump was ramped gradually (and as linearly as possible: slope of 0.1Hz/sec) from 6Hz up to 16Hz over a 100 second duration. The results illustrated in Fig.18 shows that this cooling gas flow had a significant effect on the 1st crystal stability. There is a striking contrast between the vibrations measured on the structure and the direct measurement on the optics. Although on the structure the harmonic of the N_2 pump are clearly visible as oblique lines, no major amplitude increase is observed. In contrast, on the 1st crystal, at certain pump operating regimes, the vibration amplitudes vary greatly between region of relative quiet and very noisy spans. This measurement shows clearly the interest of in-situ vibration measurements directly on the optic elements.

6 Active vibration isolation systems

Considering the relatively large vibration levels on the ESRF site, it is difficult to fulfil the stability requirement in the sub-micrometer range that some beamlines now expect. To work around this problem, active vibration isolation systems are a possible alternative. The idea is to have a combination of a passive isolation system with a low resonant frequency and an actuator driven by an electronic feedback loop acting mainly on this particular frequency. The passive part filters out the high frequency component of the excitation vibration above its resonance while the active part attenuates the amplitude of this resonance. A commercial active vibration isolation system was investigated at the ESRF on an atomic force microscopy (AFM) measurement bench (Fig.19). The active isolation system is a Halcyonics MOD-1 device with built-in electromagnetic actuators [21]. Results show very good performances in the vertical direction (Fig.20). Horizontally, a low frequency contribution remains significant at about 1Hz (Fig.21). This leads to actual amplification of the floor vibrations. However, for a standalone application like the AFM, this is not a fundamental issue. Indeed, relative motion between the sample and the microscope will remain very small. On the other hand, for an optical component part of a beamline, this is not acceptable since it would lead to large relative motion between various devices.

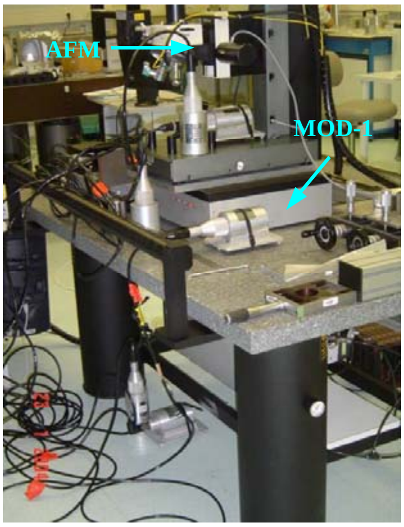
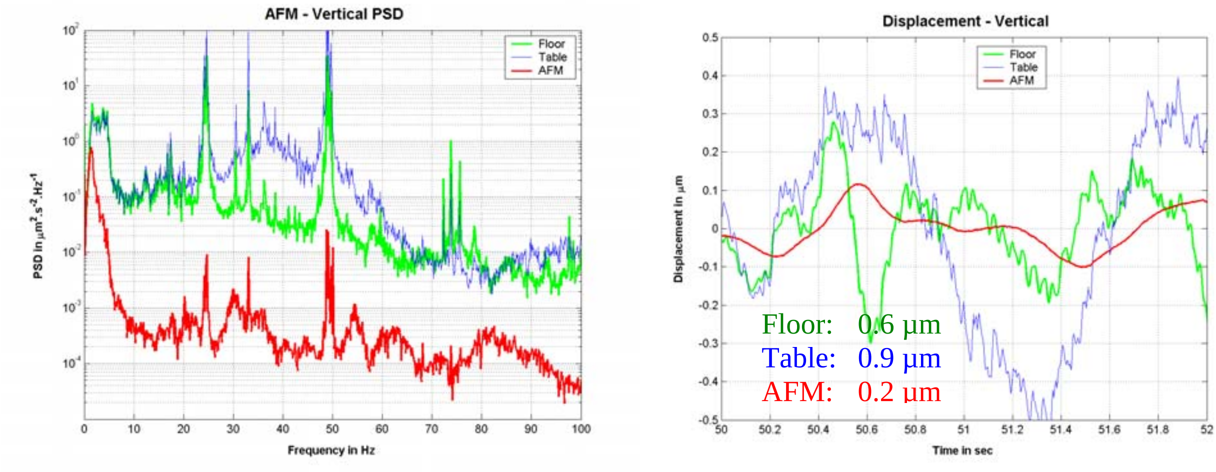
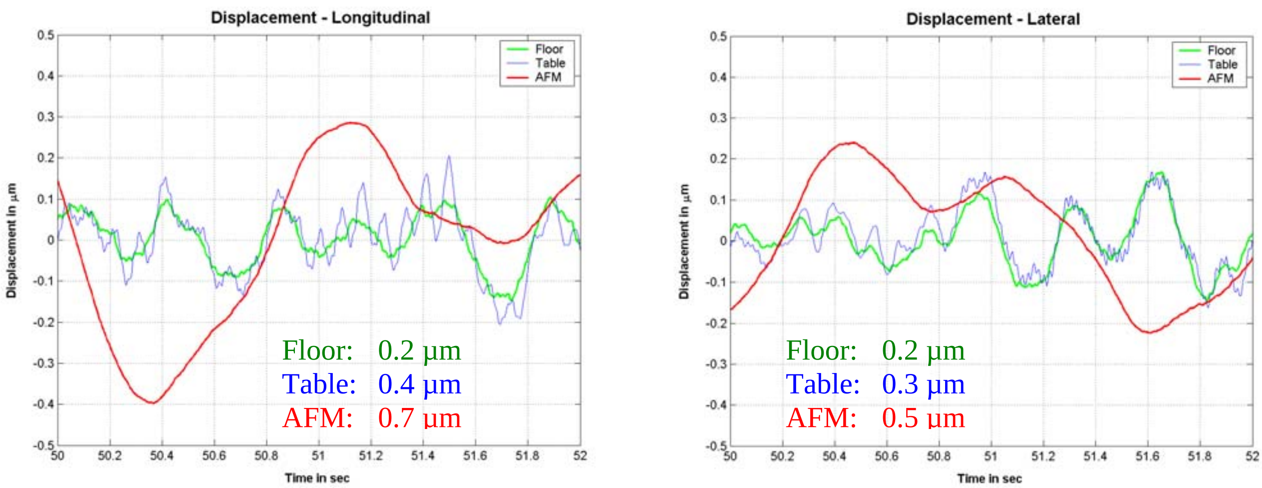


Figure 19: The Halcyonics MOD-1 active isolation system was placed underneath the AFM.



(a) (b)

Figure 20: Power Spectral Densities (a) and temporal displacement (b) in vertical direction on the floor, the supporting table and the AFM with active isolation system.



(a) (b)

Figure 21: Temporal displacement in longitudinal (a) and lateral (b) directions on the floor, the supporting table and the AFM with active isolation system.

7 Concluding remarks

The ultimate goal of the vibration studies performed at the ESRF is to provide stable X-ray beam to the beamlines users. This involves the use of adequate sensors, data acquisition systems and analysis capabilities. The stability of the electron beam, of the beamline monochromators and mirrors is crucial, great progress have been made into the characterisation, the analysis and the development of solution in order to improve the stability. In particular, vibration source identification and vibration damping of structures have seen major successful developments. The area of active vibration control and active vibration isolation is going to be pushed further in order to gain stability levels compatible with sub-micrometer orders of magnitude.

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