The Mechanical Design of High Precision Positioning Instruments, used for X-ray Microscopy at the ESRF

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Abstract

For the last six years the ESRF has been building microscopy beamlines on which various instruments such as high precision sample scanners, micro-beamstops, zone-plate holders, Kirkpatrick-Baez optics, have been developed. This design area covers vibrational stability, high level of reproducibility, micro-stiffness, compactness, no lever arm. Samples are exposed to X-ray beams with spot sizes of between 0.1 µm and 0.5 µm, which requires less than 50 nm vibrations peak to peak and a step size in "zap-scan" mode of less than 100 nm. According to the required accuracy, improvement of stiffness is a permanent concern. Pico-motors against micro-jacks afford alternative solutions which implies a rather high degree of evolution from the instruments.
This presentation will review the instrument design for ID 21, ID22, Kirkpatrick-Baez optics and other existing solutions when micro-focusing is required for some other ESRF beamlines.

Keywords: Micro-positioning, precision, steppers

1. Introduction

At the start of the ESRF, the following strategy was adopted for motorizations:

- stepper motorizations were considered capable of satisfying all positioning requirements;
- open loop control was well adapted to accommodate staff turnover (many ESRF scientists are on time limited contracts): no cultural efforts needed to be developed requiring handling of feedback and naming PID transfer functions (proportional - integral - differential);
- steppers can drive mechanisms with no encoder control. This concept induces a significant cost reduction as commercially purchased mechanical encoders can easily cost ten times that of the motor;
- a logical consequence of this philosophy was the creation of specific ESRF control and power cards, fulfilling basic functions such as series commutation of the phases, half steps commutations (alternating one and two phases), integratation of limit switches and home position (reference point).

2. Compactness Requirements for Microscopy Mechanisms

Generally speaking, experimental set-ups in which sample positioning or scanning is required are designed using a combination of translation and rotation commercial stages. Although, this is rather a relatively easy way of building an instrument, some major drawbacks may occur when large size samples are mixed with a tiny scan area. In fact, stacked stages lead to a rather large moving volume. So, even if opto-mechanical suppliers propose compact 3D combined set-ups in the range of some mm strokes, samples around 20 mm in diameter lead to large scale combination. Figure 1 illustrates a current set-up in air on the ID18 beamline.

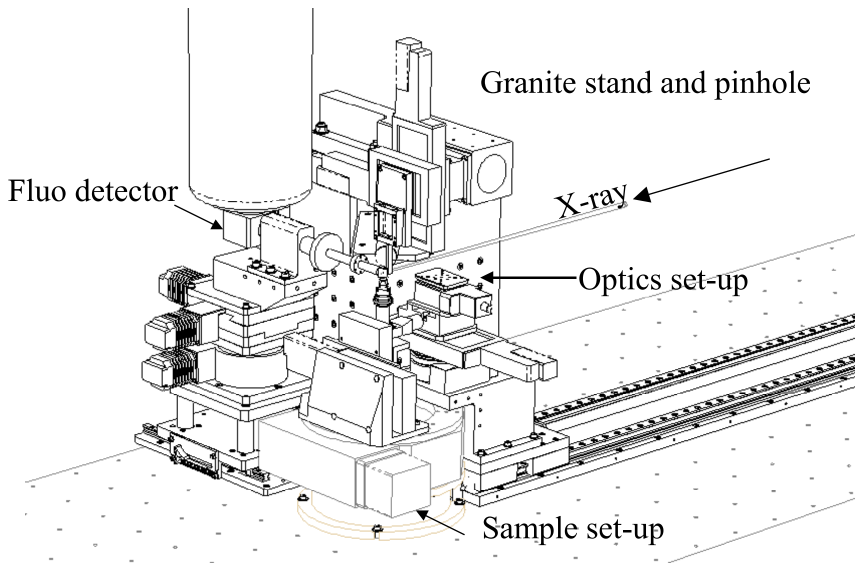


Figure 1: Fluorescence microscopy set-up at ID18 F beamline.

Instrument errors result from many sources and some suggestions for improvement are proposed here:

- Lever arms: if we start with a 20 mm diameter sample, a typical holder should be within a 5 cm size. A 1 µm error space is then given laterally by the angular error of the slides in terms of pitch, roll, yaw $\leq \frac{1\,\mu m}{5\,cm} = 20\ \mu rd$.

  This value suggests the used of high purity slides supplied by press tool companies.

- Physical resolution of carriages: the 1 µm physical resolution does not depend uniquely on the resolution of the motor. Other constraints such as stick-slip and pre-loading have a significant influence. A review of which parameter plays a key role in this process is made in this paper.

3. ESRF Design Concept

3.1 Carriage Guiding Strategy - A High Transverse Stiffness

In the light of our experience, we initially based the architecture of the carriages on ball guide systems marketed by various companies such as Mahr, Steinel, Feinprüf. These components, illustrated in Figure 2 are carefully mounted on a special casing which affords numerous advantages:

- Minimization of the number of parts (some microscope vessels must be pumped in 20 mn at 10^{-6} mbars)
- Ball pre-loading by construction with no special mechanism (only choice of diameter).
- Very high stiffness and straightness.
- Easy coating process (WS_2).

No re-circulation ball process, using the concept described in Ref. [1]

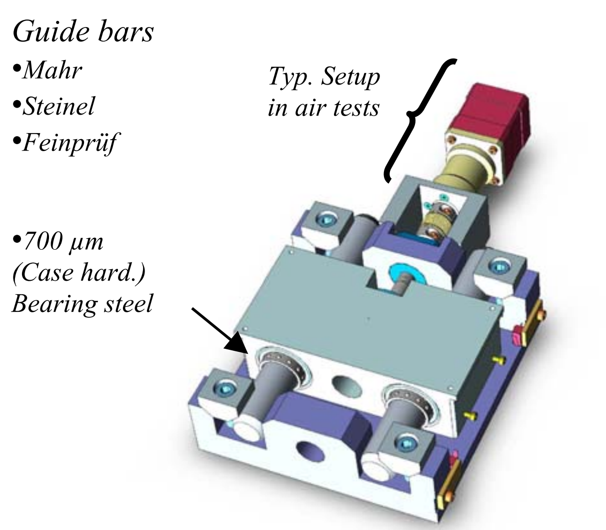


Figure 2: Design for higher stiffness: Technical proposal for as-new custom stages.

Another advantage was found in designing with the following statement: the pre-loading of the balls comes from two sources. One is the radial compression caused by the choice of diameter, typically the ball selection is made at the assembly end where they are classified by diameter steps of 1 µm. The other comes from the distance between ball guides. Figure 3 shows a sketch of the mounting principle which provides the adequate tunability.

In this example, tunability is the pursuit of a correct symmetrical loading. A good balance must be found between the screw clamping and the body bracket B (Fig. 3). Contact points P_1 and P_2 give a stiffness of 100 N/µm with a small deformation and then a low resolution. The flexion of bracket B can provide a means of improving this precious aspect with a smooth compression of the central shoulder.

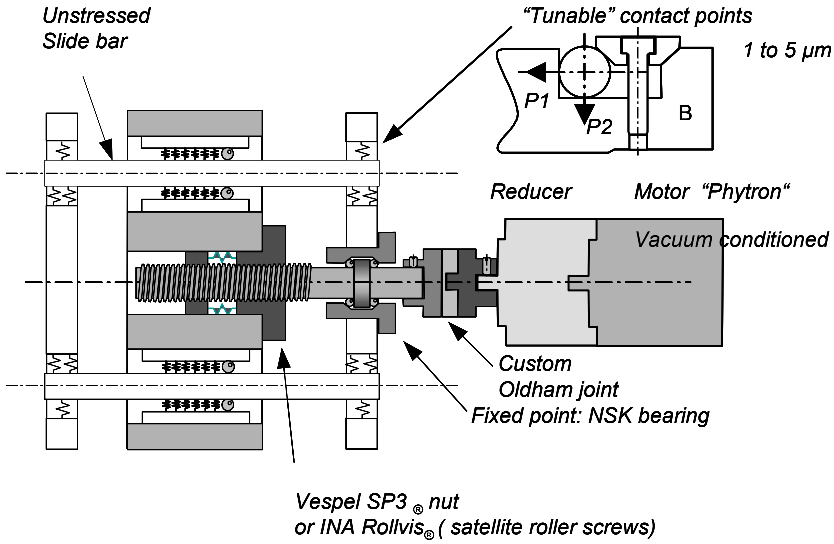


Figure 3: Design for higher stiffness: breakdown of technical issues.

3.2 Carriage Resolution: A High Longitudinal Stiffness

This section deals with the design concepts of the nut-screw system. The principle is to move the carriage by means of a pre-loaded nut based on high tech polymers such as Poly Imide (Vespel) or Peek (Ketron). This nut is a spring design with the addition of a moveable part using a combination of slots. The couple polymer/steel can be operated in air without coating. However, an interesting friction aspect is the possibility of making a multiplayer hard/soft material (WS_2), the so-called Dicronite [2] coating. Table 1 gives an idea of the different friction coefficients according to the material couple and the type of lubrification.

It is important to note the difference between the static and kinetic friction coefficients. This variation is the key point leading to stick-slip operations. This aspect is quantified in the next chapter of this paper. A performance of 0.1 µm resolution per motor step is easily achieved by dividing the screw pitch p by a reducer r, as follows:

$$\partial x_o = \frac{p}{stp}\frac{1}{r} \quad ,$$

where p = screw pitch, stp = number of steps per turn of the motor, and r = reducer rate.

Table 1: Friction Coefficients

Coefficients of Friction MATERIALS	μ_o Static	μ_k Kinetic	$\Delta\mu$ S-K
MoS2 to MoS2	0.03	0.03	0
Steel to ceramics	0.13	0.12	0.01
Steel to stainless steel	0.27	0.23	0.04
Copper to stainless steel	0.28	0.24	0.04
Copper to copper	038	0.3	0.08
Nickel to chromium	0.41	0.36	0.05
Vespel to steel (*)	0.45	0.35	0.1

(*) : these values are highly influenced by cleaning procedures

It could therefore be suggested that r may be chosen to be as large as possible to provide the resolution, this however is emitting the velocity or acceleration requirement which, for this specific aspect, needs r to be as small as possible. This adverse aspect is dealt with in the next chapter.

4. Detailed Design of Longitudinal "Stiffness"

When the selection of the NSK fixed point of the screw and the design of the spring loaded nut has been made, the essential remaining aspect is the "electrical" rigidity. This corresponds to a fine definition of the motor torque capability. This is done starting from the configuration of the movement.

In view of the motor dimensioning, the basic relation which must be used is:

$T_{max} = T_{dynamic} + T_{static}$ (during the acceleration phases) (1)

$T_{dynamic}$ = Required torque to provide acceleration movement.

T_{static} = Required torque to balance all fixed resistance forces.

Expressed in terms of final load sample parameters, this gives [3,4]:

$$T_{max} = J_o \, r\,\gamma + \frac{1}{r\,\eta_{red}}\left[J\,\gamma + T_{stat}\right], \tag{2}$$

where:

J_o = motor inertia (kg.m²)

r = reducer rate

γ = final stage acceleration (ms^{-2}) ($r\gamma$ is then the motor acceleration).

J = Total moving inertia except motor (kg.m²)

$$T_{stat} = \frac{F}{\eta_{sc}} \frac{p}{2\pi} \tag{3}$$

F = sum of all fixed forces, such as direct pre-loading of nut or all resistances (N)

p = lead screw pitch

η_{red} = reducer efficiency

η_{sc} = screw efficiency

J is the stage inertia "seen" by the motor through the screw, so the linear velocity must be traduced in terms of rotation (this is what is know as "reported terms" to the motor). All this gives:

$$\frac{1}{2}J\omega^2 = \frac{1}{2}Mv^2 \text{ and finally} \tag{4}$$

$$J = M\left[\frac{p}{2\pi}\right]^2 + J_{screw} \quad . \tag{5}$$

Expression 2 gives a behaviour which is represented in Figure 4.

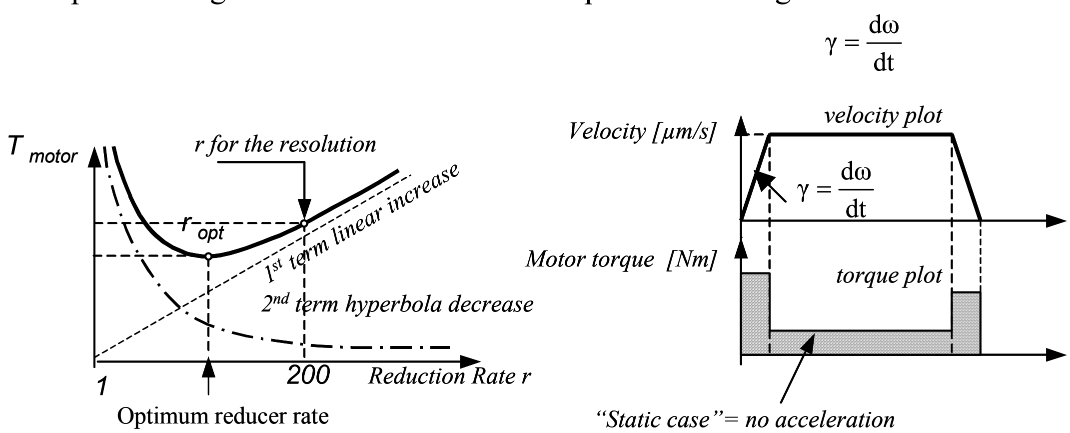


Figure 4: Evolution of motor torque versus r in static and dynamic aspects.

The total expression of the maximum torque with all these expressions becomes:

$$T_{motor} = J_o r \frac{d\omega_{load}}{dt} + \frac{1}{r\,\eta_{red}}\left[\left[M\left(\frac{p}{2\pi}\right)^2 + J_{screw}\right]\frac{d\omega_{load}}{dt} + \frac{F}{\eta_{sc}}\frac{p}{2\pi}\right]. \tag{6}$$

$\gamma = \frac{d\omega}{dr}$ is related to the sample, if we guess the motor acceleration itself, we must bear in mind that $\gamma_{motor} = r\,\gamma_{load.}$ Thus, the dynamic term $J_o r\,\gamma$ means that the higher the reducer, the higher the energy requirement to the motor when γ increases. The second term in brackets is highly influenced by the screw efficiency, which for self-locking reasons, must be very low. In fact, η_{sc} is defined by:

$$\eta_{sc} = \frac{\text{torque with no friction}}{\text{torque with friction}} \quad . \tag{7}$$

If the friction of the nut is $\mu = 0.4$, which we measure using the inclined plane method, then $\eta_{sc} = 0.06$ (which is very low!).

Globally interpreting the expression (6) leads to the possible following assessment illustrated in Figure 5.

$F := 105$	[N]	Required force to move the load [N]
$p := 1$	[mm]	Pitch of the lead screw [mm]
$f_c := 15 \cdot 10^3$	[Hz]	Commutation frequency of the motor [Hz]
$t_{acc} := .01$	[s]	Acceleration time [s]

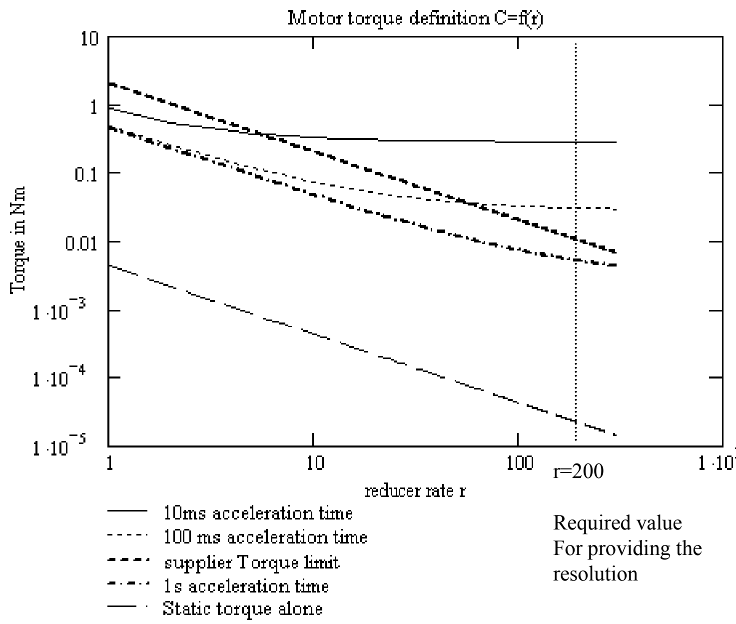


Figure 5: Evolution of motor torque versus r in static and dynamic aspects.

As a final result, beneath the allowable torque value, it was necessary to modify the following parameters:

t_{acc} = 1s (increase by 10^2!)
f_c = can not be changed. It is a scientific parameter (about 1 mm/s).

4.1 Design of Longitudinal Resolutions in the Stick-Slip Case

As a result of the friction, the stick-slip occurs when there is a significant difference between the static and kinetic friction coefficient. The stick-slip theory uses a model which leads to a rather simple conclusion. Figure 6 defines the condition and parameters involved.

μ_o = static friction coefficient
μ_d = kinetic friction coefficient
N = Pre-load force (no clearance mechanism)
k = total stiffness of moving mechanism
v_o = (ω_o) motor velocity
t = time
m = moving mass

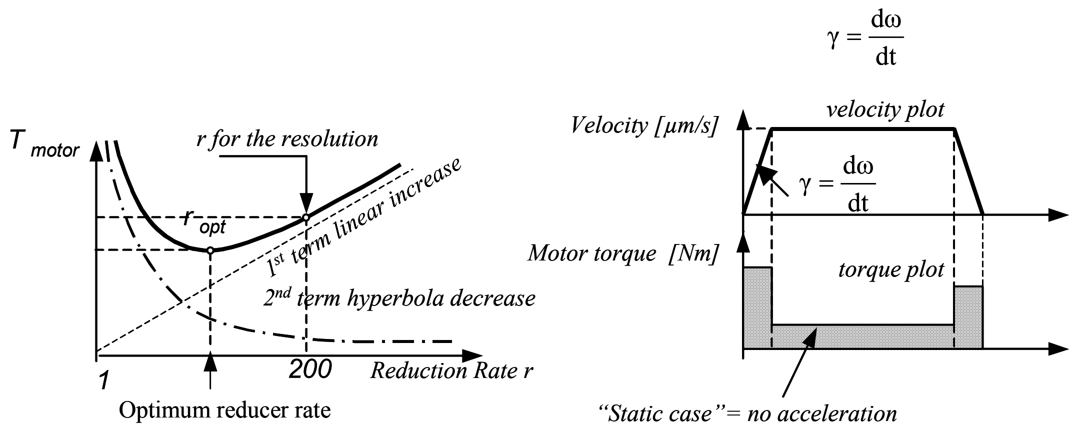


Figure 6: Stick slip is triggered by the friction variation.

This leads to the following equation [5]:

$$m\ddot{x} + kx = (\mu_o - \mu_d)N + kV_o t \quad . \tag{8}$$

Finally the solution is:

$$x = V_o t + \frac{\mu_o - \mu_d}{k} N[1 - \cos\omega_o t] - V_o \sin\omega_o t \quad ,$$

where $\omega_o = \sqrt{\frac{k}{m}}$ is the longitudinal harmonic oscillator pulsation. The minimum term of x is compared to the imposed theoretical address $s = v_o t$. This is defined by:

$$\Delta x\big|_{min} = 2N\left(\frac{\mu_o - \mu_d}{k}\right).$$

This relation is important to understand the physical possible resolution as it suggests the following statements:

for Δx small:

N = should be as small as possible. (Not easy to achieve and even more difficult to quantify), however provided by WS_2 type.

μ_o-μ_d = smallest differential friction between the static and kinetic. No so easy to quantify.

k = Rigidity of the whole mechanism as high as possible. This is almost the only quantifiable parameter which we can adjust.

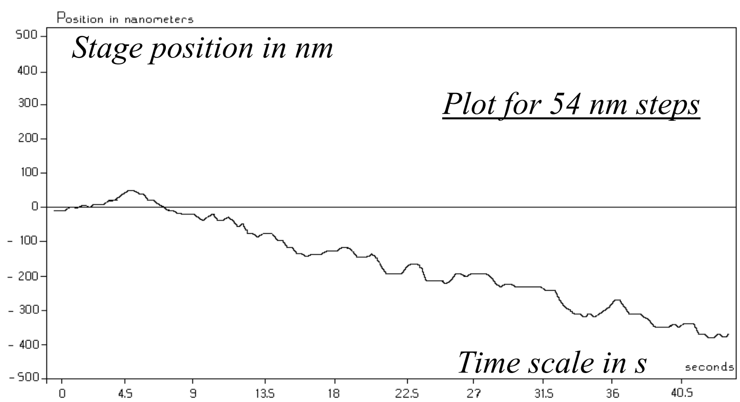
5. Examples of Realizations

The ESRF produced five prototypes, of which four were assigned to the final set-up. Table 2 gives the results of their geometrical quality compared to commercial items.

Table 2: Records of Angular Errors of Tables, Measured at the Precision Engineering Laboratory (PEL)

Identification	Stroke	Motor	Linear Positioning Tx	Unidir. Repeat.	Bidir. Repeat.	Angular Error Roll	Pitch	Yaw	Positioning Resolution	
ESRF design:	mm		µm	µm	µm	µrad	µrad	µrad	µm	steps
ID21/Prototype	85	stepping	12	2	6	14	22	21	0,2	3
ID21/Stage 1	90	stepping	20	1	10	49	25	7	0,2	3
ID21/Stage 2	90	stepping	6	1	5	30	17	19	0,2	4
ID21/Stage 3	9	stepping	11	0,15	9	57	4	6	0,1	3
ID21/Stage 4	7,5	stepping	7	1	5	8	11	8	0,1	6
Outside manufacturers:										
Huber 5101.3	300	stepping	17	-	14	32	194	22	-	-
Melles Griot - flexure	10	DC+PZT	23	-	3,7	227	51	32	-	-
Micro-Controle UT 100 PP	100	stepping	15	2,7	5,9	165	115	133	-	-
Micro-Controle UT 100 PP	100	stepping	12	2,8	2,8	26	73	40	-	-
Physik Instrumente	25	DC	12,2	3,9	9	-	46	13	0,1	1
Physik Instrumente	25	DC	9,8	0,3	4,7	18	22	50	0,1	1
Schneeberger	98	stepping	37	1,2	1,5	75	108	203	-	-

As an example of resolution achievement, Figure 7 also shows the search for a minimum step made at the Precision Engineering Laboratory (PEL).



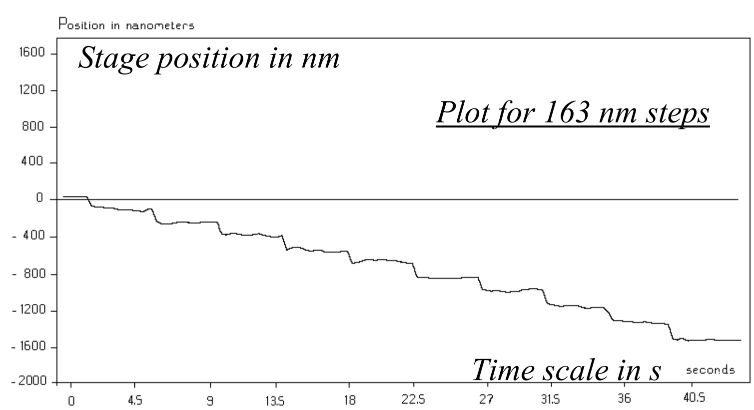


Figure 7: Detection of the minimum step resolution.

A comment concerning the nut pre-loading should be made; the friction process injects some heat into the screw. A simple quantification of this is given by the expression:

$\dot{P} = \mu p v$ with $\dot{P}$ = surface power density w/m²
μ = kinetic friction coefficient
p = contact pressure N/m²
v = contact relative velocity m/s

All this leads to an extension of the screw of Δl = 230 nm for a stroke of 25 mm within 25 seconds, of course decreasing $\dot{P}$ by an order of magnitude is a key of improving dynamic precision. This leads, at the minimum, to lower as far as possible the friction coefficient, which as it is demonstrated throughout this paper, solves most of the precision aspects.

6. Conclusion

The ESRF is bound to make stepper control evolution in phase with scientific requirements.

Attempts to provide high precision are currently being experimented using standard ESRF equipment.

At present, velocity is adapted to resolution although these aspects are antagonistic. The next generation of drivers is being designed with a very high speed stepper driven by a more complex electronic crate which implies an encoder feedback.

This paper has been presented a very precise static and dynamic dimensioning of the motor torque as being a key point, confirmed by torque bench measurements. We also presented designs in which highly over constrained mechanisms enables the required stiffness and then precision to be met.

The friction process acts at every stage of the design and it must be one the major design parameter concerns. Unfortunately this aspect remains one of the most difficult to satisfy. Nonetheless, attempts to achieve a 10 to 4 nm resolution are currently in progress at the ESRF.

7. Acknowledgements

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8. References

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