



**HAL**  
open science

## Damage due to start-stop cycles of turbine runners under high-cycle fatigue

Olivian Savin, Julien Baroth, C. Badina, Sylvie Charbonnier, Christophe  
Bérenguer

► **To cite this version:**

Olivian Savin, Julien Baroth, C. Badina, Sylvie Charbonnier, Christophe Bérenguer. Damage due to start-stop cycles of turbine runners under high-cycle fatigue. *International Journal of Fatigue*, 2021, 153 (December), pp.106458. 10.1016/j.ijfatigue.2021.106458 . hal-03463291

**HAL Id: hal-03463291**

**<https://hal.science/hal-03463291>**

Submitted on 6 Jan 2022

**HAL** is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.



Distributed under a Creative Commons Attribution - NonCommercial - NoDerivatives 4.0  
International License

# Damage due to start-stop cycles of turbine runners under high-cycle fatigue

O. Savin<sup>1,2,3</sup>, J. Baroth<sup>2</sup>, C. Badina<sup>1</sup>, S. Charbonnier<sup>3</sup>, C. Bérenguer<sup>3</sup>

<sup>1</sup> General Technical Division, Electricité de France, F-3800 Saint Martin le Vinoux, France

<sup>2</sup> Univ. Grenoble Alpes, CNRS, Grenoble INP\*\*, 3SR, F-38000 Grenoble, France

<sup>3</sup> Univ. Grenoble Alpes, CNRS, Grenoble INP\*\*, Gipsa-Lab, F-38000 Grenoble, France

Corresponding author: [julien.baroth@univ-grenoble-alpes.fr](mailto:julien.baroth@univ-grenoble-alpes.fr)

## Abstract:

This study presents a strategy to estimate the extra cost due to start-up and shut-down of hydro turbines under high-cycle fatigue. The principle is to compare estimations of cumulated damages from three elementary sequences representing steady and transient regimes (start and stop cycles) of turbine runners, from in-situ strain measurements. Rainflow cycle-counting is used to extract stress cycles from these three sequences. Fatigue design curves are defined, taking into account design codes, laboratory tests and a probabilistic approach. The analysis of these curves, coupled with the Palmgren-Miner rule, allows estimating damage ratio, i.e. extra costs of start and stop events, through an equivalent number of normal operating hours. This approach is applied to Francis and Pelton runners. Different results are discussed in terms of damage distributions and extra costs of start and stops.

**Keywords:** start and stop, turbine, high cycle fatigue, Wöhler curve, turbine, rainflow cycle-counting, Palmgren-Miner, damage, Francis runner, Pelton runner

27 **Table 1**

28 Nomenclature.

Sigle	Description	Sigle	Description
EdF	Électricité de France	SS	Start and stop
SRH	EdF Hydraulic reparation service	<i>S-N curve</i>	Stress vs Number of load cycles
GE	General Electric	FFT	Fast Fourier Transform
ASME	American Society of Mechanical Engineers	$N_{noh}$	Number of normal operating Hours
RCC-M	Design and Construction Rules for Mechanical Components	FEM	Finite Element Model

29 **Table 2**

30 Main notations.

Symbol	Parameter	Unit	Symbol	Parameter	Unit
$YS, UTS$	Yield and ultimate tensile strengths of steel	Pa	$E$	Young's modulus	Pa
$\sigma$	Stress	Pa	$\varepsilon$	Strain	$\mu\text{m}/\text{m}$
$\sigma_a$	Applied stress amplitude	Pa	$\sigma_m$	Mean stress	Pa
$\sigma_{nom}$	Nominal stress amplitude	Pa	$\sigma_{min}, \sigma_{max}$	Minimal, maximal stress	Pa
$FEL$	Fatigue endurance limit	Pa	$D$	Damage	-
$Nr$	Number of load cycles to failure (Fatigue life)	-	$d_{st}, d_{sp}, d_{sy}$	Elementary damages	-
$N$	Number of load cycles	-	$\Delta t_{st}, \Delta t_{sp}, \Delta t_{sy}$	Durations	-
$\sigma_D(N)$	Endurance limit for N cycles	Pa	$t$	Time	s
$K_t$	(Fatigue) stress concentration factor	-	$\mu, s$	Mean, standard deviation	-
$R$	Stress ratio $\sigma_{min} / \sigma_{max}$	-	$CV$	Coefficient of variation CV=s/ $\mu$	-

## 31 **1 Introduction**

32 During the last decades, hydro-power plants have been working with an increasing  
 33 number of start and stop cycles to balance the electric network. Power fluctuations are generated  
 34 to integrate new types of energy in the electrical grid, but start and stop cycles events are  
 35 particularly damaging (Gagnon et al. 2010). As a consequence, estimating the cost incurred by  
 36 start and stop cycles because of increased damage becomes crucial to develop short and long-  
 37 term operating strategies.

38 Start and stop cycles events induce degradation of various mechanical equipments of the  
 39 plants, such as the turbine, penstock and main shut-off valve. Hydroelectric turbines runners  
 40 are one of the most expensive components to be potentially repaired (Savin et al. 2020). It is  
 41 also known that fatigue and cavitation are the two main degradation mechanisms of these  
 42 runners (Dojčinović et al. 2017, Gagnon et al. 2018), among other failure modes (Liu et al.  
 43 2016). Fatigue is often considered as the most difficult to manage, particularly in case of fatigue

44 cracking (Arsić et al. 2013, Dorji & Ghomashchi, 2014, Liu et al., 2016, Luna-Ramirez et al.,  
45 2016).

46 Our aim in this work is to propose a method to quantify the additional cost due to the  
47 increased damage on the runners by fatigue due to the starts and stops. This aim in view, two  
48 approaches are used for the fatigue assessment (Balaji Rao et al. 2019). On the one hand, some  
49 models are based on crack growth and failure criteria (Susmel, 2008, 2009); crack development  
50 can occur on the buckets or blades of the turbine runners (Kokko et al. 2014, Welte 2008),  
51 Doujak 2016), Hassanipour (2015), Gagnon et al. 2010). On the other hand, some models are  
52 based on the relation between the magnitude of the alternating stress history vs the number of  
53 cycles to failure (S-N curves) in combination with Palmgren-Miner's damage accumulation  
54 rule (Marcouiller & Thibaut, 2015). Aiming to quantify the extra cost of start and stops of in-  
55 service turbines, this last kind of models is retained.

56 The proposed approach is based on the analysis of strain measurements near the critical  
57 areas where maximum stresses are concentrated in the stainless-steel runner. The analysis of  
58 these measurements is based on the selection of steady-state and transient sequences and on the  
59 use of rainflow cycle-counting (Matsuishi & Endo, 1969, Amzallag et al. 1994). From the  
60 resulting cycle histograms, damage is estimated using a *S-N* curve (Wöhler curve). Such a  
61 model linking the stress history and the number of cycles to failure is an engineering tool used  
62 for a long time (Wöhler, 1860), but still various works are developed to properly design it for  
63 fatigue life prediction (Susmel et al. 2014, Saga et al. 2020). Particularly, it is crucial to account  
64 for the stochastic nature of fatigue damage process (Schijve, 2005). This aim in view,  
65 probabilistic *S-N* curves, denoted *P-S-N*, allow accounting for large dispersion of fatigue test  
66 data (Gagnon et al. 2013).

67 In this work, we will discuss both the use of design curves, modified using both *P-S-N*  
68 and a regulatory curve, as defined using RCC-M (AFCEN, 2008) or ASME (Chopra & Shack,  
69 2003, ASME, 2013). Then, Palmgren-Miner's law is used to estimate cumulative damage  
70 (Marcouiller & Thibaut, 2015), for each transient sequence (start-up and shut-down). Finally,  
71 we determine the equivalent number of normal operating hours  $N_{noh}$  for each start or stop cycle.  
72 Such a number can be estimated from expert judgment. Nilsson & Sjelvgren (1997) suggested  
73 that one start and stop is equivalent to 15 hours of normal operation. In the same paper, the  
74 value of 50 hours for a start and stop is also reported from Mjøl̄snes, 1999, found in Bakken,  
75 2002 (see also Bakken & Bjørkvoll 2002, Bureau of Reclamation 2014). Anyhow, these values  
76 are not properly justified, and to the best of our knowledge, no scientific publication can be  
77 found to present in detail how to estimate  $N_{noh}$  from in-situ strain measurements, except in brief

78 conference communications in French (Jacquemoud, 1994) and in French confidential reports  
79 (Ballester 1992, EdF & GE, 2018). It is why we propose in the following to present and develop  
80 the approach, specifying all the underlying assumptions.

81 An instrumented Francis runner has been chosen as an example. Such equipment has been  
82 designed to start on Monday and to stop on Friday, but nowadays, the start and stop cycles  
83 frequency has been drastically increased. It is then crucial to better quantify the effect of these  
84 transient sequences.

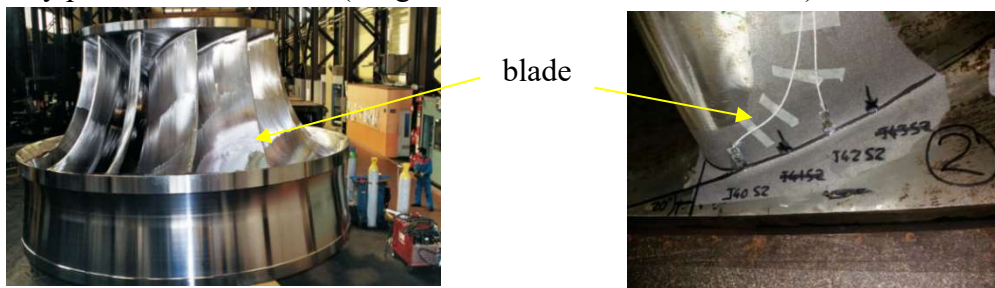
85 The first section of the paper highlights consequences of start and stops on the lifetime of  
86 turbine runners. It also briefly reminds usual practice to account for fatigue phenomenon. Then  
87 a practical fatigue assessment method is presented, estimating damage ratios due to runners  
88 start and stops. Then, the method is applied to Francis runners. Finally, conclusions are drawn,  
89 highlighting limits and perspectives of the approach.

## 90 **2 Problem statement**

91 This section is first focused on turbine runners, undergoing fatigue due to start and stop cycles.  
92 Then  $S-N$  curves classically used for fatigue assessment are introduced. Finally, a discussion  
93 about the extra cost due to start-up and shut down operations is proposed.

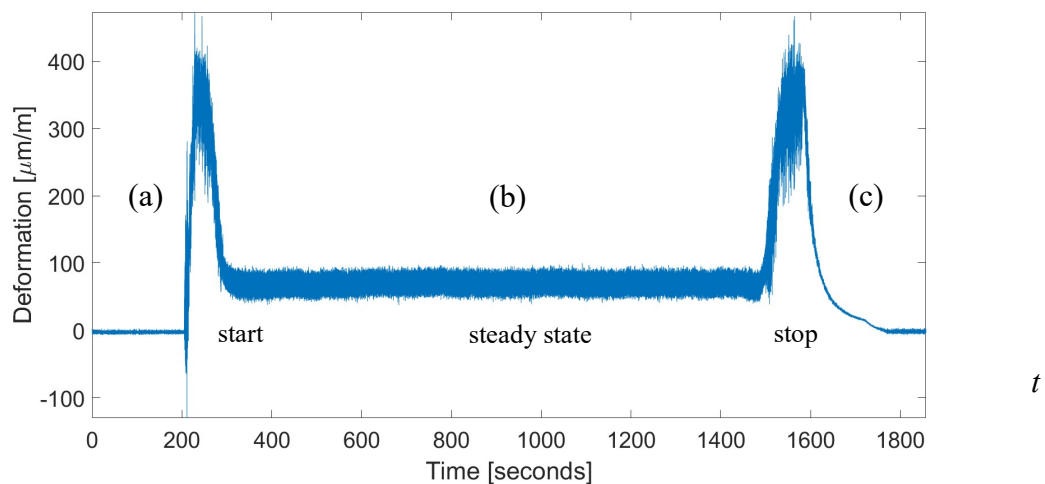
### 94 **2.1 Turbine runners under fatigue**

95 Common Francis turbines are composed of a runner band, runner crown and fixed blades  
96 (Fig. 1). These constructions are casted using stainless steels in 13-17% Cr martensitic  
97 (Buxbaum & Ostermann, 1983), (Sonsino & Dieterich, 1990). Particularly, we will consider  
98 13CR-4Ni (or DIN GX5CrNi 13-4), denoted “13-4 stainless steel” in the following. Runners  
99 undergo various several transient phenomena, as pointed out by Huth (2005): unsteady loading  
100 conditions can be induced by out of phase synchronization during start-up, modification of  
101 operating point, earth fault, (emergency) shut-downs (Nicolet et al. 2003). Particularly, start  
102 and stop cycles of Francis turbines runners are associated to transverse vibration, caused by  
103 high-frequency pressure fluctuations (irregular fluid flow, see Huth, 2005).

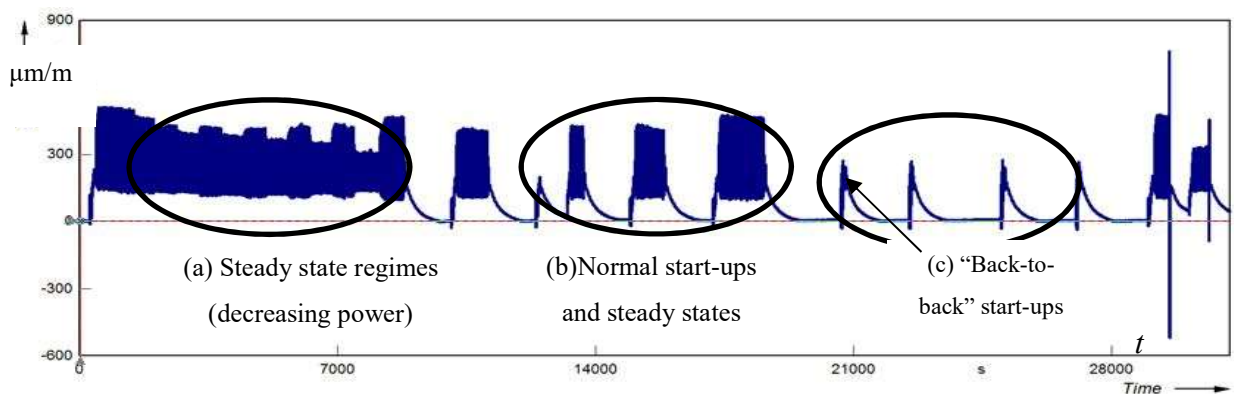


109 **Fig. 1.** Francis turbines: a) example of a runner @EdF, b) welded joint of a blade (EdF & GE, 2018).

110 High-cycle fatigue (HCF) occurs, with associated deformation in every cycle remaining elastic,  
 111 *i.e.* the maximum stress in the runner, denoted  $\sigma_{max}$ , remaining under the steel yield strength  
 112  $YS$  (Jabbado, 2006). Highest stresses usually appear at blade welds with the band and crown  
 113 where each blade is fixed (Fig. 1). Figures 2 and 3 provide examples of strain histories in the  
 114 cases of two different turbines (Francis Fig. 2 and Pelton Fig. 3). Figure 2 shows the turbine  
 115 start (a) and stop (c) phases and the steady operating period (b) in the case of a Francis runner.  
 116 Figure 3 also show examples of unsteady steady sequences, in the case of a Pelton runner. In  
 117 such a figure, different strain evolutions can be observed for different power levels (Fig. 3a)  
 118 and also for start and stop sequences, coupled with steady state sequences (Fig. 3b) and only  
 119 used to help a pump to start (“back-to-back” start-ups, Fig. 3c).  
 120 These evolutions show very different unsteady behaviors. Figure 2 shows that the strain  
 121 amplitude in steady state is less significant compared to a few seconds of start-up or shutdown,  
 122 whereas Therefore Figure 2 seems to highlight that the cost of one hour in steady state is  
 123 insignificant compared to a few seconds start-up or shutdown.



134 **Fig. 2.** Evolution of strain measurements for 3 loading sequences (Francis runner) ©EdF.



143 **Fig. 3.** Evolution of strain measurements for various loading sequences (Pelton runner) ©EdF.

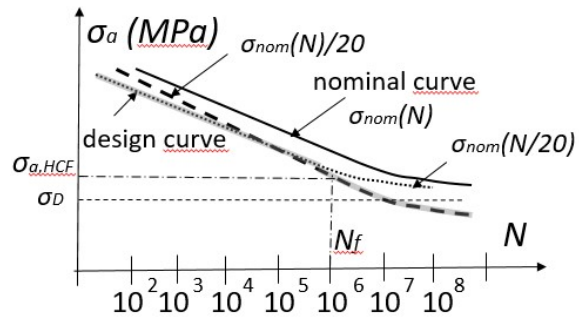
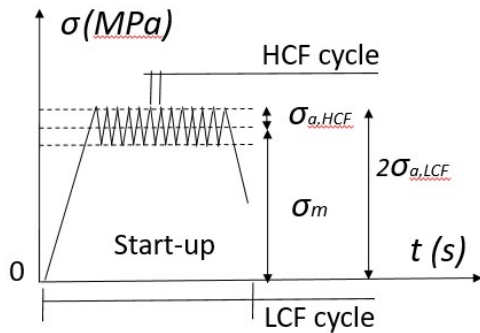
144 On the contrary, Figure 3 displays steady states (with different power ranges) with  
 145 comparable strain amplitudes compared with start-ups and shutdowns (Fig 3a).

146 In the last two figures, the strain gauges were positioned near the critical points  
 147 (“hotspots”), where the stress values (von Mises) are the highest. In the case of a Francis runner  
 148 (figure 2), they were located at the junction between blade and crown and at the junction  
 149 between blade and band; the first was analyzed because of higher recorded values. In the case  
 150 of a Pelton runner (figure 3), gauges were located at the bucket root.

151 The stress history can be deduced from such strain measurements from the gauge the  
 152 closest to the zone with the maximum stress  $\sigma_{max}$  in the runner. Denoting  $\sigma_{nom}$  the nominal  
 153 stress deduced from the gauge, one usually define the stress concentration factor  $K_t$  as the ratio  
 154 between  $\sigma_{max}$  and  $\sigma_{nom}$ .

$$155 \quad K_t = \frac{\sigma_{max}}{\sigma_{nom}} \dots(1)$$

156 In Fig. 4, an example of a stress signal during a simplified start-up sequence is shown,  
 157 highlighting stress HCF cycles. Meanwhile, the whole start-up operation can be considered as  
 158 a low-cycle fatigue (LCF) cycle, as it occurs less than  $10^6$  times in the lifetime of the equipment.  
 159 In this figure, stress amplitudes of HCF and LCF cycles are respectively denoted  
 160  $\sigma_{a,HCF}$  and  $\sigma_{a,LCF}$ . The mean stress is denoted  $\sigma_m$ .



161  
 162 **Fig. 4.** Stress history highlighting a start-up sequence  
 163 (runner scale).

**Fig. 5.** *S-N* curve: nominal and design curves  
 (laboratory specimen scale).

## 164 2.2 Fatigue assessment using regulatory or P-S-N Curves

165 Fatigue assessment of a runner usually consists in using a *S-N* curve, which is based on strain-  
 166 controlled fatigue tests of small specimens at room temperature in air or in corrosive  
 167 environment (see e.g. Buxbaum & Ostermann, 1983) or (Sonsino & Dieterich, 1990). Figure 5  
 168 displays a nominal *S-N* curve and the corresponding fatigue endurance limit (FEL) under which  
 169 the stress amplitude  $\sigma_a$  will not lead to a failure, whatever the number  $N$  of cycles (see Lei et

170 al. 2021). Figure 5 also displays a design curve. For example, Figure 5 shows the regulatory  
171 number of cycles to failure  $N_f$ , related to the HCF stress amplitude  $\sigma_{a,HCF}$ .

172 The current nuclear design codes such as RCC-M (AFCEN, 2008) or ASME codes (Section III  
173 fatigue design curves, see Chopra & Shack 2003) define nominal curves as best-fit median  
174 curves to the experimental test data accounting for modifications (in particular but not only):

175 - to account for the effects of mean stress using the modified Goodman relationship

176

$$177 \sigma_{nom} = \sigma_a \left(1 - \frac{\sigma_m}{UTS}\right) \dots (2)$$

178

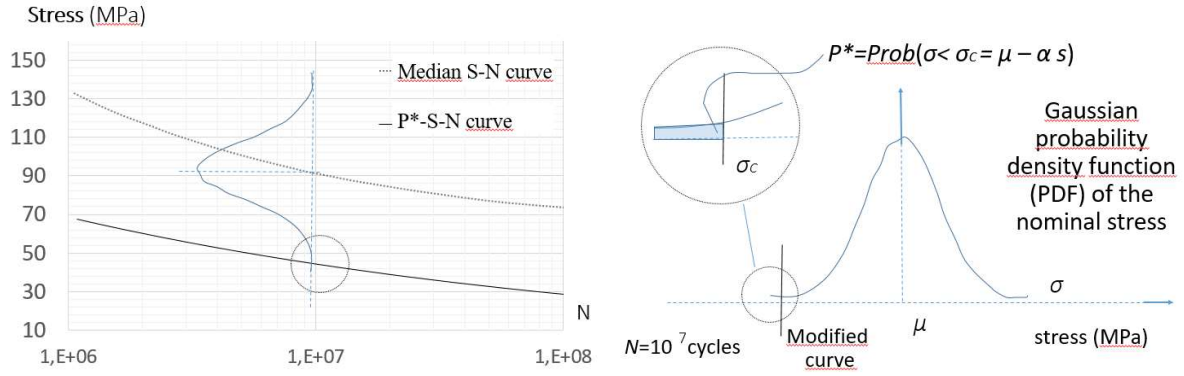
179 with  $UTS$  the ultimate strength of the runner material,  $\sigma_m$  the mean stress,  $\sigma_{nom}$  the nominal  
180 stress, and  $\sigma_a$  the applied stress amplitude (Pa).

181 - and then lowered by a factor  $\gamma_S$  of 2 on stress and  $\gamma_N$  of 20 on cycles to obtain the fatigue  
182 design curves.

$$183 \sigma_d(N) = \text{Min}[\sigma_{nom}(N)/2; \sigma_{nom}\left(\frac{N}{20}\right)] \dots (3)$$

184 These passage factors ( $\gamma_S$ ,  $\gamma_N$ ) are supposed to cover conservatively two different effects: the  
185 variability of the lifetime under fatigue between laboratory tests and a structure within its real  
186 environment on the one hand, and between laboratory specimens on the other hand  
187 (Grandemange & Faidy, 2000), (Sudret, 2011). Chopra & Shack (2003) underlined that these  
188 factors are not safety margins but rather adjustment factors. They are not intended to address  
189 all the environmental effects on fatigue life. For example, fatigue data obtained in the U.S. and  
190 Japan demonstrate that light water reactor environments can have potentially significant effects  
191 on the fatigue resistance of steels, such that the current ASME Code design curve can become  
192 nonconservative for certain materials. Moreover, RCC-M or ASME codes seem rather to be  
193 validated for fatigue lives under  $N = 10^5$  and  $N = 10^6$  respectively. These last two limits of  
194 regulatory curves encourage to use probabilistic  $S-N$  curves (Balaji Rao et al. 2013, Fouchereau,  
195 2014, Seddik, 2017). Indeed,  $P-S-N$  curves can correspond to a given probability to get a lower  
196 stress amplitude. Figure 6 depicts as an example a  $P-S-N$  curve based on the common  
197 assumption of a Gaussian probability law modelling the nominal stress.





198  
199 **Fig. 6.** Example of a design curve defined as a probabilistic  $S-N$  curve.

200

201 The position of the design curve therefore depends on the knowledge of a standard deviation  $s$   
202 and a target probability  $P^*$ , defined in Eq. 5 (cf. Fig. 6 and Table 3).

203

204 **Table 3**

205 Target probability values  $P^*$  depending on  $\alpha$ , such that  $P^* = Prob(\sigma < \sigma_c = \mu - \alpha s)$ .

Probability $P^*$	0.05	$10^{-2}$	$10^{-3}$	$10^{-4}$	$10^{-5}$
$\alpha$	1.645	2.33	3.1	3.72	4.26

206 Passage factors ( $\gamma_s, \gamma_N$ ) become finally particularly relevant if a corresponding  $P-S-N$  curve can  
207 be defined. For instance, the use of a characteristic value can be considered, as it is defined in  
208 design codes such as Eurocode (CEN, 2002). Particularly, a 5% characteristic value  $\sigma_k$ ,  
209 associated to a partial safety coefficient  $\gamma$ , could be defined such as

210 
$$\sigma_k = \sigma_m - 1.645 s \quad \text{and} \quad P(\sigma < \sigma_k) = 0.05 \quad \dots(4)$$

211 
$$P^* = P(\sigma < \sigma_c = \sigma_k / \gamma) \quad \dots(5)$$

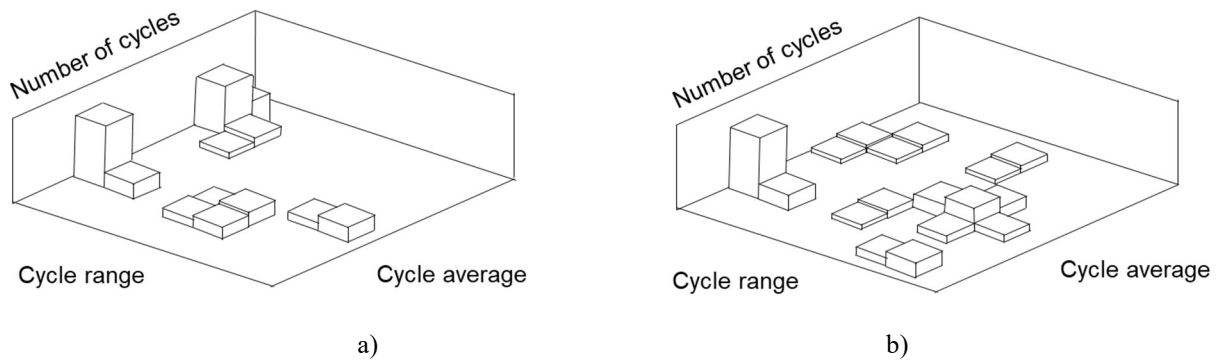
212 **2.3 The extra cost due to the increase of start and stop cycles**

213 In the simplified case presented Fig. 4, there is only one HCF stress amplitude  $\sigma_{a,HCF}$ , for  $n_{HCF}$   
214 cycles, and one LCF cycle with stress amplitude  $\sigma_{a,LCF}$ . Then the damage caused by these  
215 cycles can be written:

$$d = \frac{1}{N_f^{LCF}} + \frac{n_{HCF}}{N_f^{HCF}} \quad \dots(6)$$

218 More realistically, an on-site stress history leads to a complex series of stress amplitudes  
219 (see Fig. 2 and 3). For such complex loading history, histograms of stress amplitudes can be  
220 extracted using a cycle-counting method such as rainflow method, Markov chain or ARMA  
221 model methods (Ling et al. 2011). Figure 7 displays simplified distributions of cycles obtained  
222 for the two damaging sequences of a Pelton turbine (Ballester 1992): full-power steady state

223 (Fig 7a, corresponding to Fig. 3b setting), and “back-to-back” start-up sequence (Fig. 7b,  
 224 corresponding to Fig. 3c setting).



225  
 226  
 227 **Fig. 7.** Simplified distributions of cycles extracted after a cycle-counting procedure: case (a) full-power  
 228 steady state, and case (b) start-up sequence.  
 229

230 Damage corresponding to these cycles distributions is estimated using Palmgren-Miner’s  
 231 damage accumulation rule, summing elementary damages, more generally denoted  $n_j$ , such that

$$d = \sum_{j=1}^r \frac{n_j}{N_{f,j}} \quad \dots(7)$$

234 The well-known cumulative damage method (EN 1993-1-9, 2005) means that a total  
 235 damage is reached for  $d=1$ . However, such a rule does not help to estimate the lifetime of  
 236 runners, given that this lifetime can be impacted by phenomena that cannot be measured using  
 237 damage estimation.

238 In this paper, damage estimation principle only aims at comparing steady and unsteady  
 239 sequences. The idea is to use cycle-counting to different sequences, then to extract damages  
 240 and finally to compare them. Damage ratios could then help quantifying how start of stop  
 241 sequences are more damaging (or not) compared to a supposed steady state.

242 The next section will present such an approach.

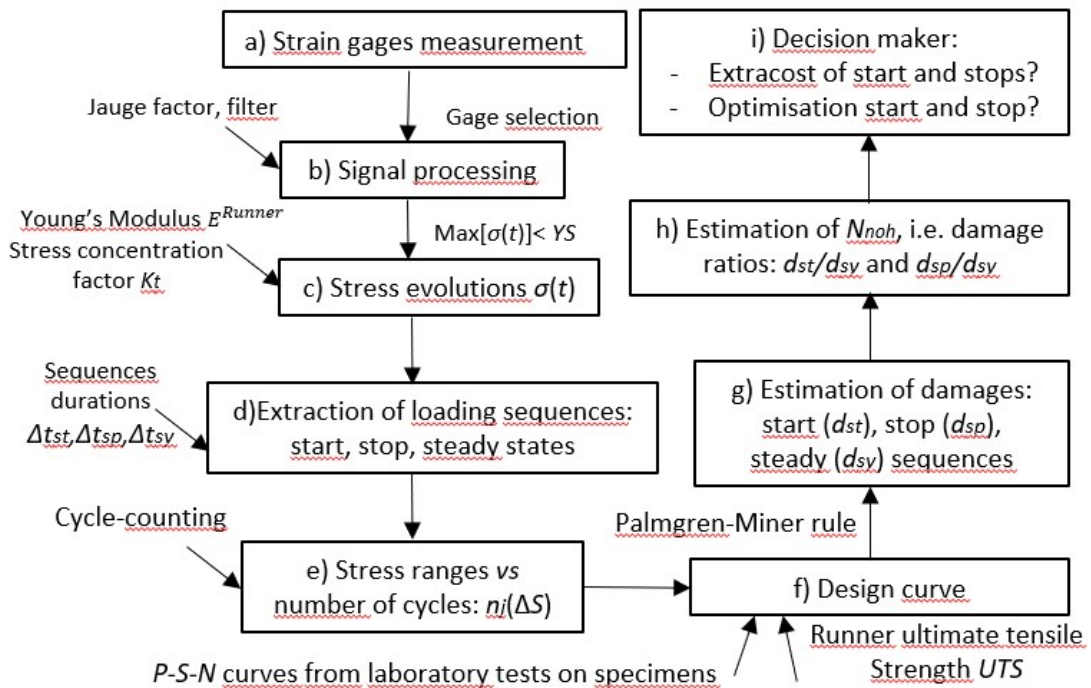
### 243 **3 Estimation of damage ratios due to runners start and stop**

244 This section presents how to estimate equivalent number of operating hours for start and stop  
 245 events. Principle of this estimation is resumed in Fig. 8 and is explained. Then, a probabilistic  
 246  $S-N$  curve is determined, based on fatigue tests data.

#### 247 **3.1 Presentation of the approach**

248 The proposed method consists first in the instrumentation of the runners with strain gages  
 249 (Fig. 8a) (Lofflad et al. 2014). The question whether extensometers (unidimensional gages

250 instead of rosette gages) are sufficient has to be discussed. To do this, a finite element (FE)  
 251 model can be used to estimate the stress concentration factor  $K_t$  (Bryla et al. 2013) and to  
 252 confirm that uniaxial strain are representative of the real strain state (Morissette et al. 2016).  
 253 These strains are derived after gages signal processing, accounting for a gauge factor and after  
 254 filtering, accounting for signal noise (Fig. 8b). Finally, stresses are deduced by means of  
 255 Hooke's law and Young's modulus of the runner  $E^{runner}$  and by checking that the maximum  
 256 stress is below the yield strength  $YS$  (Fig. 8c). Loading sequences are selected for start and stop  
 257 and steady states, with durations  $\Delta t_{st}$ ,  $\Delta t_{sp}$ ,  $\Delta t_{sy}$  (Fig. 8d). Then cycle counting allows obtaining  
 258 stress amplitudes  $\sigma_a$  vs number of loading cycles  $n_j$ , for each sequence (Fig. 8e). A design  
 259 curve is defined using  $P-S-N$  curves based on laboratory tests on specimens (Fig. 8f). As a  
 260 consequence, Palmgren-Miner's damage accumulation rule leads to elementary damages  $d_{st}$ ,  
 261  $d_{sp}$ ,  $d_{sy}$  (Fig. 8g) and damage ratios  $d_{st}/d_{sy}$  and  $d_{sp}/d_{sy}$  (Fig. 8h) are deduced. Figure 8 summarizes  
 262 the main steps of the approach.



263

264

**Fig. 8.** Flowchart of the method estimating the numbers of normal operating hours  $N_{noh}$ .

265

From these damage ratios are derived: equivalent numbers  $N_{noh}$  of normal operating hours

266 
$$N_{noh}^i = \frac{\Delta t_{sy} d_i}{\Delta t_i d_{sy}} \dots \quad (8)$$

267

where  $(N_{noh}^i, \Delta t_i, d_i)$  denotes  $(N_{noh}^{st}, \Delta t_{st}, d_{st})$  and  $(N_{noh}^{sp}, \Delta t_{sp}, d_{sp})$  for start and stop transient sequences respectively.

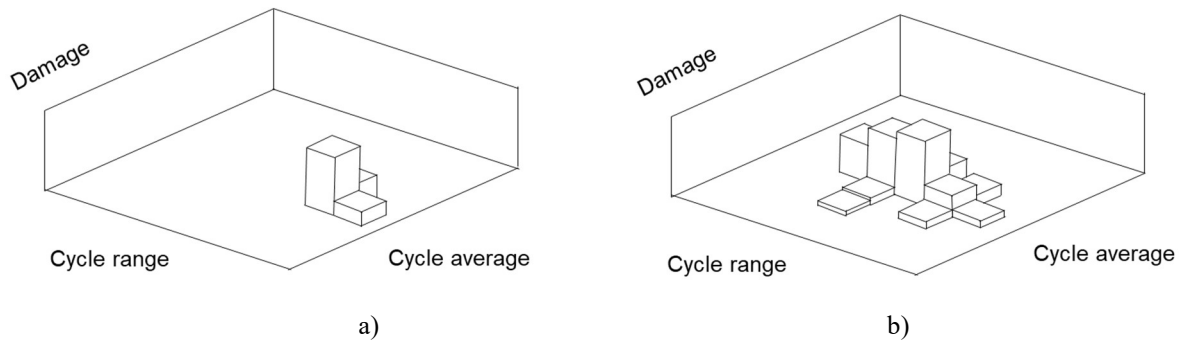
268

269

Elementary damages  $d_{st}$ ,  $d_{sp}$ ,  $d_{sy}$ , denoted as  $d_j$  in the following, are the ratios between the numbers of loading cycles  $n_j$  and  $N_{fj}$ . The number of cycles  $n_j$  is given by a cycle-counting

270

271 method (Fig. 8e). For example, Figure 9 displays distributions of elementary damage  $d$ , deduced  
 272 from histograms (Fig. 7) and using numbers of cycles to failure  $N_{fj}$  estimated from a  $P^*=1/100$   
 273  $S-N$  curve (Ballester 1992).



274  
 275  
 276 **Fig. 9.** Simplified distributions of damage: case (a) full-power steady state, compared to start-up sequence  
 277 case (b), with more cycles and cycle ranges, leading to more damage.

278 In the case of full-power steady state (Fig. 7bis a), most damaging cycles are those  
 279 corresponding to the stresses after water impact: on the blades in the case of Francis runners,  
 280 on the buckets in the case of Pelton runners. As an example, in the case of back-to-back start of  
 281 a pump (Fig. 7bis b), Ballester (1992) estimated that the accumulation of damage of the Pelton  
 282 runner leads to  $N_{noh} = 3$  times the total damage found in case a).

283 The next section presents how to define the cycles number at failure  $N_{fj}$  using a design curve,  
 284 based both using probabilistic and regulatory  $S-N$  curves, as proposed.

### 285 3.2 Probabilistic $S-N$ curves

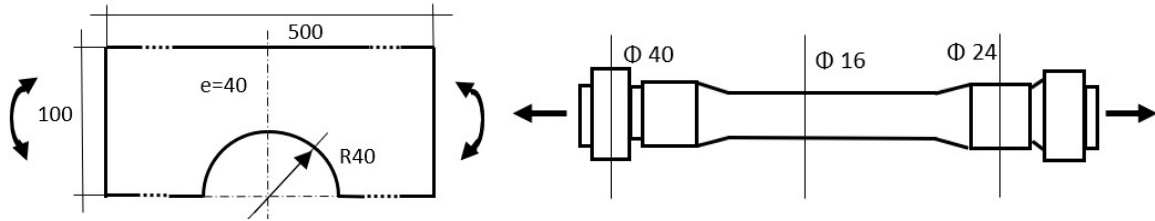
286 This section is divided into three parts: i) the presentation of fatigue test series, ii) the  
 287 modified Goodman relationship, taking into account the effect of mean stress, iii) the estimation  
 288 of  $P-S-N$  curves in accordance with the design rules.

#### 289 Fatigue test series

290 Series of tests were carried out on dozens of specimens, under axial and also combined axial  
 291 and lateral cyclic loading, on 2 types of specimens and several steel grades (Buxbaum &  
 292 Ostermann, 1983). These tests have been carried out in the same conditions as industrial  
 293 runners. They are extracted from large-sized casted blocks. The size of the specimens is  
 294 consistent with the dimensions of the runners. Only 13.4 steel with yield strength  $YS = 628$  MPa  
 295 and ultimate strength  $UTS = 804$  MPa is used here. Figure 10 shows the two types of specimens,  
 296 which correspond to:

- 297 - axial cyclic tests: the stress is assumed to be homogeneous in cross-section, which leads to a
- 298 stress concentration factor  $K_t$  of 1;

299 - combined axial and lateral cyclic tests composed of notched prismatic specimens, with  $K_t$   
 300 equal theoretically to 1.36.



301  
 302 **Fig. 10.** Specimen under flexural bending (left) and tension (right) (Buxbaum & Ostermann, 1983), cf. Table 4  
 303  
 304

305 Table 4 gathers configurations depending on test conditions in air or in corrosive environment  
 306 and on the zero mean stress levels ( $R=-1$ ), equal to 204 MPa, or  $\sigma_{nom}/2$  ( $R=0$ ). For each  
 307 combined axial and lateral cyclic test, stresses are deduced from measurements at the edge or  
 308 center of the specimens (series 4b and 4c). A theoretical stress concentration factor  $K_t$  is given.  
 309 In the following, we particularly study tests series 3 and 4, because they have been carried out  
 310 in corrosive environment similar to the one of industrial runners.

311 **Table 4**

312 Synthesis of test series (Buxbaum & Ostermann, 1983), cf. Fig. 10.

Series	Environment	Type of mechanical	(Ratios of) stress	Theoretical stress
d'essais	corrosive	loading	min., moy. max.	concentration factor
0	air	Alternate tension	$R = -1, \sigma_m = \sigma_{nom}/2$	1
1	air	Alternate tension	$R = 0, \sigma_{min}=0$ MPa	1
3b (3c)	corrosive	Alternate bending	$R = 0, \sigma_{min}=0$ MPa	1.36 (1)
4b (4c)	corrosive	Alternate bending	$\sigma_m=204$ MPa	1.36 (1)

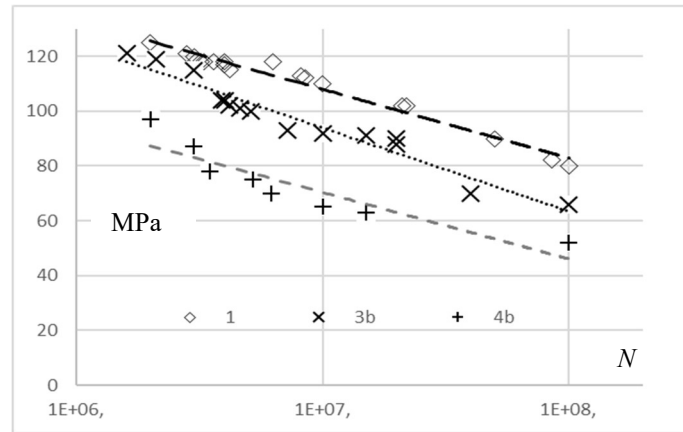
313 Table 5 presents some statistics on selected test series. The dispersion of the measurements is  
 314 estimated by the coefficient of variation of the stress amplitude  $\sigma_a$  for  $N=10^7$  cycles. Figure 11  
 315 also shows the dispersion of the measurements. This coefficient of variation fluctuates from  
 316 about 10 to 20% depending on the series, with an increasing trend for higher mean stress.  
 317

318 **Table 5**

319 Stress amplitudes  $\sigma_a$  in cases of series 0, 1, 3, 4 (Buxbaum & Ostermann, 1983).

Tests	Tests	$N$ variation	Stress $\sigma_a$	Mean stress (MPa)	Coefficient
series	number	domain $\Delta$	$N=10^7$ (MPa)	(Standard deviation)	of variation
0	15	$4 \cdot 10^5 - 2,3 \cdot 10^7$	182	32 (24)	0.21
1	18	$2 \cdot 10^6 - 10^8$	110	13 (9)	0.14
3b	15	$1.6 \cdot 10^6 - 10^8$	92	9 (8)	0.14
3c	8	$2.5 \cdot 10^6 - 10^8$	74	5 (5)	0.11
4b	9	$6 \cdot 10^5 - 10^8$	65	6 (4)	0.09
4c	14	$2 \cdot 10^6 - 10^8$	52	6 (5)	0.16

320



321  
322  
323  
324

**Fig. 11.** Examples of measurements series (1, 3b et 4b) (Buxbaum & Ostermann, 1983), see. Table 3. Series correspond to tensile stresses (Fig. 10, right) and series (3b, 4b) to bending stresses (Fig. 10, left).

325 **Accounting for the mean stress and stress concentration factor**

326 Table 6 presents these modifications for  $N = 10^7$  and  $10^8$  load cycles, in the case of series 3 and  
327 4, with measurements at the edge and in the center of the specimens. Two types of modification  
328 can then be estimated:

- 329 - taking into account the mean stress: modified Goodman relationship, from 4 to 25%;
- 330 - the stress concentration factor  $K_t$ , estimated here between 1.23 and 1.4 (1.36 theoretically).

331 **Table 6**  
332 Modified stress amplitudes (MPa) depending on the mean stress, for test series 3, 4 de (Buxbaum & Ostermann,  
333 1983), 13.4 steel, ultimate limit strength of 804 MPa, from measurements at the edge and center of specimens

Specimen edge	Specimen center (close to the notch)			$K_t$ tests				
<i>Series 3b (edge) and 3c (center) : case <math>R=0</math>, i.e. <math>\sigma_{min}=0MPa</math> and <math>\sigma_m = \sigma_a/2</math></i>								
$\sigma_a$	Correct.(%)	$\sigma_{nom}$	$\sigma_a$	Correct.(%)	$\sigma_{nom}$			
$N = 10^7$	72	4.5	75	$N = 10^7$	59	3.7	61.2	1.23
$N = 10^8$	90	5.6	95	$N = 10^8$	74	4.6	77.6	1.23
<i>Series 4b (edge) and 4c (center): case <math>\sigma_m = 204 MPa</math></i>								
$\sigma_a$	Correct.(%)	$\sigma_{nom}$	$\sigma_a$	Correct.(%)	$\sigma_{nom}$			
$N = 10^7$	52	25.4	70	$N = 10^7$	37	25.4	49.6	1.41
$N = 10^8$	70	25.4	94	$N = 10^8$	54	25.4	72.4	1.30

334 The stress concentration factor of 1.36 is not exactly found, partly because of the uncertainties  
335 in the stress estimate and partly because it seems unlikely that the maximum stress was  
336 accurately measured at the center of the specimens. For safety reasons,  $K_t = 1.36$  will be  
337 considered in the following.

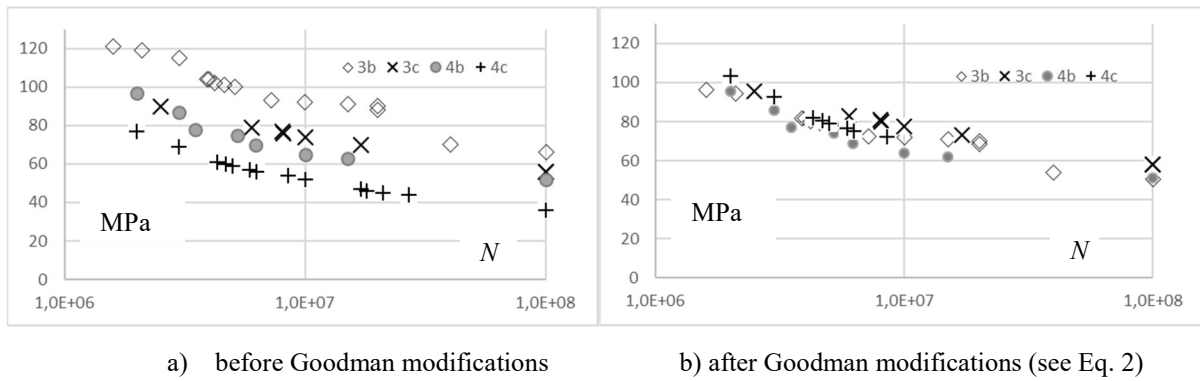


Fig. 12. Mean  $S-N$  curves (13.4 steel), series 3b, 3c, 4b, 4c

### Looking for nominal and design $S-N$ curves

In order to carry out a fatigue design, it is necessary to define a nominal curve, which can in principle be deduced from each of the 4 above-mentioned curves. Figures 12a and 12b show the mean Wöhler curves of series 3 and 4, derived from measurements at the edge and center of the specimens, before and after Goodman modifications. If we do not use the mean curves, but all the stress amplitudes, we obtain an average dispersion of 13%, which is the average coefficient of variation that can be deduced from Table 3. Defining a nominal curve could then consist of a regression of one curve among others or of the mean points of Fig. 12b. Equation 9 provides the equation of the regression for all series 3 and 4.

$$\sigma_{\text{nom}} = -10.66 \ln(N) + 245.19 \dots (9)$$

This 50%  $S-N$  curve is plotted on Fig. 13., with different corrected design curves. Indeed, one considers at least three options:

- a) a  $P-S-N$  curve; Figure 13a proposes 0.1% and 0.01%  $S-N$  curves, deduced from the 50%  $S-N$  curve (Eq. 8), and following Table 1 and Eqs. 4-5;
- b) a regulatory curve deduced from the 50%  $S-N$  curve (Eq. 8), applying the design rule (Eq. 3); Figure 13a shows that this curve is very close to the 0.01%  $S-N$  curve;
- c) regulatory curves deduced from the design code (ASME, 2013).

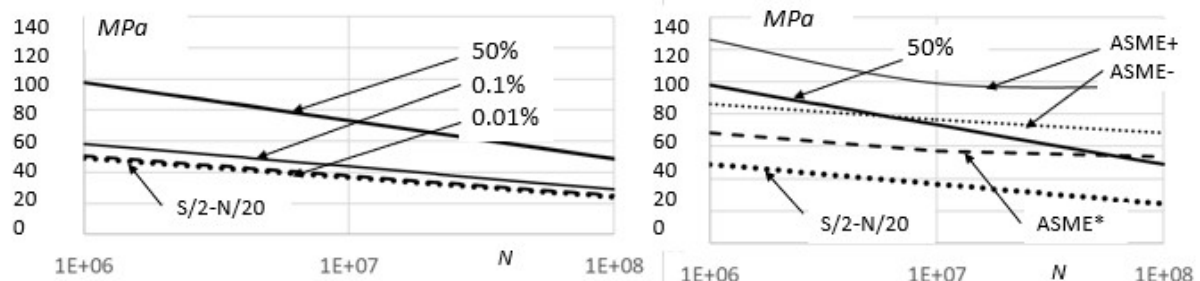


Fig. 13. Corrected design  $S-N$  curves deduced from the mean experimental series (a), from ASME code (b).

368  
369  
370  
371  
372  
373  
374  
375  
376  
377  
378  
379

Last option, c) curves denoted ASME+ and ASME-, respectively, correspond to carbon and low-alloy steels (Young's modulus  $E$  of 206.8 GPa) and austenitic stainless steels ( $E = 195.1$  GPa) (ASME, 2013). Contrary to  $P$ - $S$ - $N$  curves deduced from the presented experimental series, ASME  $S$ - $N$  curves do not account for any corrosive environment, justifying arbitrarily corrected  $S$ - $N$  curve by dividing nominal stress values by a safety factor, e.g. 1.5 (see Brand et al., 1999). Therefore Figure 13b shows the revised  $S$ - $N$  curve, denoted ASME\*, accounting for this coefficient and also a 200 GPa Young's modulus. It is worth noting that if this revised curve is conservative for  $N < 10^7$ , it becomes far less conservative for larger number of cycles, which corroborates the analysis found in Chopra & Shack (2003), in the case of a usual fatigue assessment.

#### 4 Application to Francis and Pelton runners

Method presented in the section 2 is applied here to Francis and Pelton runners. Considered runners are listed in Table 7. They are all made using 13.4 steel, with Young's modulus of 200 GPa. Signal filtering and cycle counting are applied in four cases, before estimating damage ratios, i.e. equivalent number of normal operating hours. For all studies, signals were acquired at a 2400 Hz frequency with an anti-aliasing filter.

**Table 7**  
List of runners (EdF fleet of hydroelectric power plants).

	<b>Runner type</b>	<b>Effective power</b>
Runner 1	Francis (turbine)	28 MW
Runner 2	Francis (turbine)	2 MW
Runner 3	Pelton (turbine)	> 80MW
Runner 4	Pelton (Ballester, 1992)	152 MW

##### 4.1 Case of a 28 MW Francis runner (runner 1)

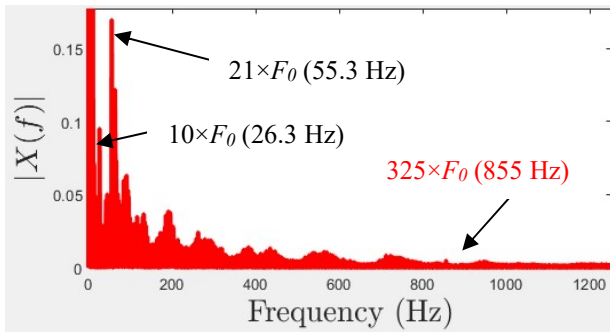
This runner has been studied estimating a maximum stress of 60 MPa (EdF & GE, 2018), for an estimated maximum deformation of 300  $\mu\text{m/m}$ . However, the maximum deformation found (J1S8 gauge) is 132.5  $\mu\text{m/m}$ , which leads us to consider a stress concentration factor  $K_t$  of 2.27 with an accuracy of about 3% ( $K_t$  estimated between 2.16 and 2.34, EdF & GE, 2018). For the power of 28 MW,  $K_t = 2.16$ , is the coefficient we propose to consider in the following.

394  
395

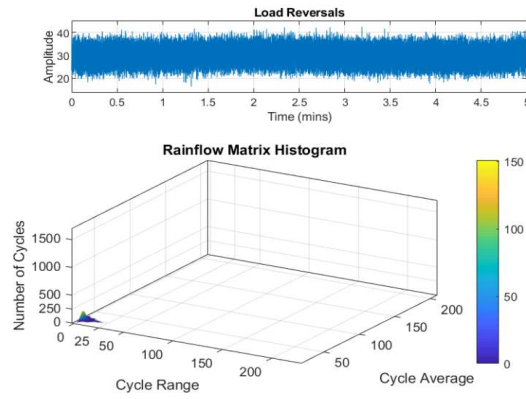


396 **Signal filtering and cycle-counting**

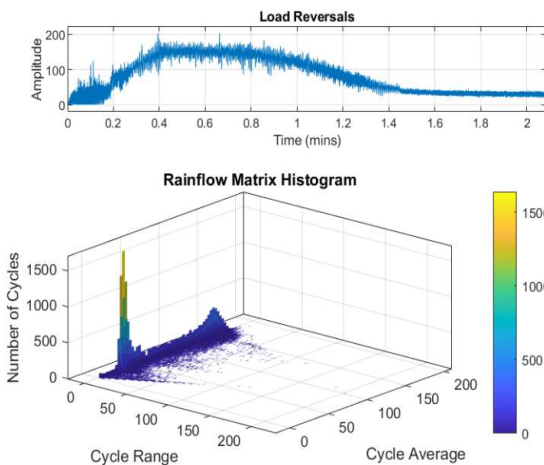
397 A spectral analysis is performed to quantify the impact of all degradation phenomena occurring  
 398 at the runner, and determine the filter to be applied to separate the real effects from the signal  
 399 noise. Fast Fourier Transform was performed during 5 minutes-long stabilized operating signal  
 400 (Fig. 14). The eigen frequency  $F_0 = n_0/60 = 2.63$  Hz directly depends on the rotational frequency  
 401 of the runner, where  $n_0 = 158$  rpm is the rated synchronous speed of the runner.



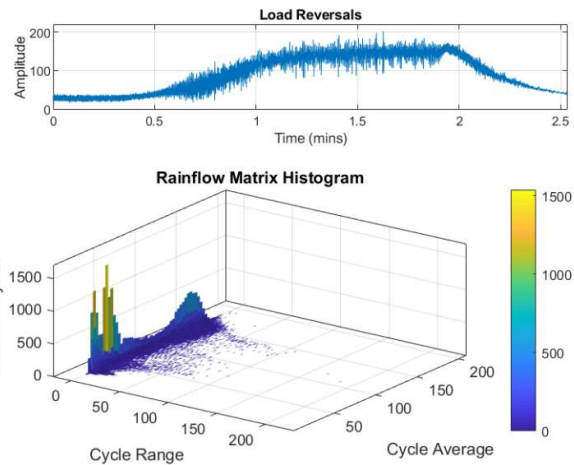
402  
 403 **Fig. 14.** Spectral analysis of the signal using a fast Fourier  
 404 transform (FFT): evolution FT modulus vs frequency.



**Fig. 15.** Representation of the rainflow counting  
 matrix using loading history of a steady regime.



a)



b)

415 **Fig. 16.** Representation of the rainflow counting matrix using a load history data in case of 28 MW startup (a)  
 416 and stop sequences (b) (Runner 1).

417  
 418 The analysis shows harmonics at  $2 \times F_0$ ,  $3 \times F_0$  up to  $21 \times F_0$ , with a natural frequency  $F_0$  equal to  
 419 2.63 Hz. The harmonics captured in the spectrum could probably correspond to hydraulic  
 420 instabilities e.g. to the  $Z_r = 13$  blades ( $Z_r F_0 = 34.2$  Hz), to the  $Z_s = 24$  wicket gates ( $Z_s F_0 =$   
 421 63.2 Hz)). The value of 855 Hz is an example that corresponds in fact to  $25 Z_r F_0$ , harmonic of  
 422 the interaction between the stationary pressure field (on the distributor side) and a rotating  
 423 pressure field - (on the runner side)). At 55.3 Hz is detected the rotor-stator interaction and at  
 424 26.3 Hz interblades vortices.

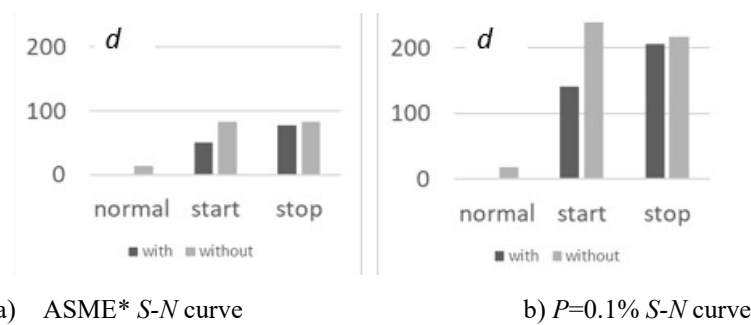
425 The 1 kHz filter is eventually applied in the following, given that ISO 20816-5 (2018)  
 426 also recommends ~~not~~ filtering not below 1 kHz.

427 Rainflow counting algorithm (ASTM, 2011) is used to translate time-dependant stress  
 428 loading into stress cycles amplitudes. For illustration, Figure 15 displays the steady state regime  
 429 during  $\Delta t_{sy} = 5$  minutes, for a power stabilized at 28 MW, and a representation of the rainflow  
 430 counting matrix, with histograms depending on amplitude (cycle range), mean stress (cycle  
 431 average) and number of cycles.

432 Figures 16 represent the rainflow counting matrix using a load history data in the same  
 433 case of a 28 MW start-up (a) and shutdown sequences (b). Their durations are respectively  
 434 equal to  $\Delta t_{st} = 2$  min and  $\Delta t_{sp} = 2.5$  min. On the histogram, fairly high numbers of cycles are  
 435 reported compared to steady state sequence, where average cycles clearly less than 50 MPa.

### 436 Estimations of damage and normal operating hours

437 From this last observation, it seems irrelevant to account for an endurance limit. This can be  
 438 checked displaying damage estimations for steady, start and stop sequences, with and without  
 439 30 MPa fatigue endurance limit (FEL). This comparison is proposed in Fig. 17, accounting for  
 440 arbitrarily  $FEL = 30$  MPa, using ASME\* (a) and  $P=10^{-4}$  (b)  $S-N$  curves respectively. This figure  
 441 highlights that the steady regime corresponds to stress amplitude mostly lower than the FEL.  
 442 As a consequence, normal operating hours estimated in Tab. 6 increase drastically.



443  
 444 a) ASME\*  $S-N$  curve  
 445 b)  $P=0.1\%$   $S-N$  curve  
 445 **Fig. 17.** Representation of damage estimations for steady, start and stop sequences, with and without 30 MPa  
 446 endurance limit, accounting for ASME\* (a) and  $P=10^{-4}$  (b)  $S-N$  curves respectively (runner 1).  
 447

448 Table 8 gathers estimations of normal operating hours ( $N_{noh}$ , defined in Eq. 8) obtained for  
 449 runner 1, in cases of start and stop events. Three design curves presented in section 3 (Fig. 13)  
 450 have been used to estimate damage ratios, considered as  $N_{noh}$  estimations for these events (Fig.  
 451 16). For example, the first column provides the damage ratio (extracost) of a start-up, using the  
 452 ASME\* curve (Fig. 13b), without accounting for a fatigue endurance limit (FEL). It means that  
 453 stress amplitudes below the FEL have been ignored to estimate elementary damages.

454

455 **Table 8**

456 Number of normal operating hours ( $N_{noh}$ ), start-up of the 28 MW Francis (runner 1)

<b>Sequence</b>	<i>ASME*</i>	<i>ASME*</i>	<i>P=0.1%</i>	<i>P=0.1%</i>	<i>P=0.01%</i>
<i>(with/without FEL)</i>	<i>without</i>	<i>with</i>	<i>without</i>	<i>with</i>	<i>without</i>
Start-up	6	51	13	141	658
Stop	6	78	12	205	423

457 Results from Table 8 seems conservative to account for the FEL. Anyhow, since most stress  
 458 amplitudes of steady regime sequence being below the FEL, the operator better should consider  
 459 them. Table 8 also shows that estimations of extra cost of start and stops highly depend on the  
 460 choice of the  $P$ - $S$ - $N$  curve:  $N_{noh}$  increases as  $P$  decreases. For example,  $N_{noh}=141$ h means that a  
 461 start-up corresponds to 141h of normal operating hours if we consider a  $P$ - $S$ - $N$  design curve  
 462 with  $P=0.1\%$  and if we account for the FEL in the estimation of the damage ratio  $N_{noh}$ .  
 463 Moreover, runners' operator should bear in mind that  $P$ - $S$ - $N$  curves are estimated from a limited  
 464 number of tests, between  $N=10^6$  and  $10^8$  cycles, whereas damage estimation needs to use  $S$ - $N$   
 465 curves for higher  $N$  values. This last observation would then tend to recommend a regulatory  
 466 curve such as ASME ones, defined to  $N=10^6$  cycles. The operator has finally to choose a safety  
 467 factor, arbitrarily equal to 1.5 in this work, but which should probably be taken larger.

468

469 **4.2 Other runners and discussion**

470

471 This section gathers results obtained using other runners (Tab. 5).

472

473 **Runner 2 (2 MW Francis turbine)**

474 This study was conducted using histories observed working on a runner similar to  
 475 runner 1. We consider the same material and also the same stress concentration factor. For  
 476 technical reasons, we were only able to focus on a start-up of 2 MW. Figure 18 represents  
 477 histograms of a start-up (a) and the steady state regime (b). In this case, a startup of less than  
 478 2 minutes corresponds to around 35 minutes of steady regime, using the  $P=1\%$   $S$ - $N$  curve.

479

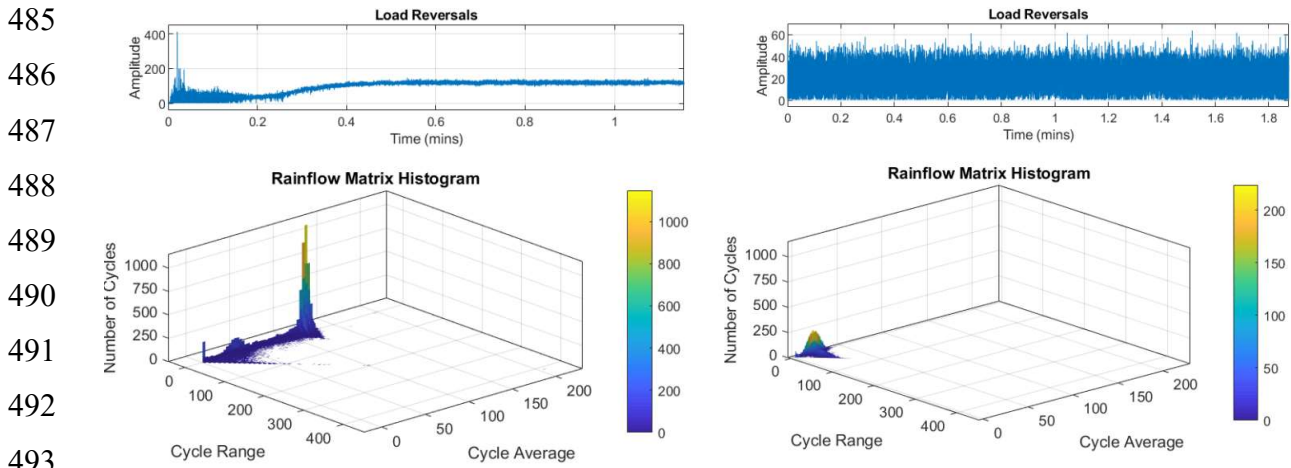
480

481

482

483

484



494 **Fig. 18.** Representation of the rainflow counting matrix for a Francis turbine in case of 2 MW startup (a) and  
 495 steady regime (b) (runner 2)  
 496

497 **Pelton runners 3 and 4**

498 Considered Pelton runners 3 and 4 are used to start a reversible machine in pump mode.  
 499 Runner 3, with power greater than 80 MW, can be used to start a turbine-pump less powerful.  
 500 In this case, there is clearly no extra cost due to start or stop.  
 501 Runner 4 has the same power as the turbine-pump to be started (Ballester 1992). In this case,  
 502 the start-up of this runner is equivalent to 3 normal operating hours.

503 **5 Conclusion**

504 The paper presents a strategy to estimate the extra cost of start and stop of hydro turbines, under  
 505 high-cycle fatigue. The method compares estimations of cumulated damages from three  
 506 elementary sequences representing steady and transient regimes (start and stop cycles) of  
 507 turbine runners. These estimations are based on regulatory or probabilistic *S-N* curves,  
 508 determined using fatigue tests data. The approach has been applied to Francis runners, using  
 509 also a past result on a Pelton runner, published only in French (Ballester 1992).

510 The proposed approach is interesting to detect and quantify extra cost of start and stop in terms  
 511 of normal operating hours. However the operator has to bear in mind that these estimations  
 512 depend on the reference *S-N* curve used to estimate elementary damages. Laboratory tests or  
 513 regulatory curves are useful, but one has to use them with caution. For example, one of the  
 514 available ASME curves was designed for austenitic steel used in nuclear industry, but not used  
 515 for runners of hydraulic plants.

516 Once an extra cost has been detected, the recommendation from Gagnon et al. (2010) can then  
517 be followed if possible, i.e. optimize this start-up scheme with regard to fatigue damage, in  
518 order to extend Francis runner life expectancy.

519 As perspectives, this approach could be applied to other ageing components such as penstock,  
520 or main shut-off valve. Ling et al. 2011 experienced to implement Markov chain or ARMA  
521 model method instead of rainflow cycle-counting method, that could also be improved (Anthes,  
522 1997, Yung-Li et al. 2012).

523

## 524 **Acknowledgements**

525 The authors would also like to thank Philippe Bryla, Stéphan Courtin, Nicolas Dutruel, Thibaut  
526 Autrusson, for their scientific advices.

## 527 **6 References**

- 528 AFCEN (2008): RCC-M - Edition 2007 – Addendum dec. 2008: Design and Construction Rules for  
529 Mechanical Components of PWR Nuclear Islands.
- 530 Amzallag, C., Gerey, J.P, Robert, J.L. and Bahuaud J., « Standardisation of the rainflow counting  
531 method for fatigue analysis », Int. J of Fatigue, Vol 16(4), 287-293, 1994.
- 532 Anthes R. J.. Modified rainflow counting keeping the load sequence. Int. J. of Fatigue, 19(7) :529—535,  
533 1997.
- 534 Arsić, M., Karić, R., Sedmak, A., Burzić, M. and Vistać B., Methodological Approach to Integrity  
535 Assessment and Service Life of Rotating Equipment at Hydropower Plant – Turbine Shaft, Structural  
536 Integrity and Life, Vol.13, No.2, 2013, 117–124.
- 537 ASME, American Society of Mechanical Engineers Boiler and Pressure Vessel Code (ASME Code),  
538 Appendix I to Section III, “Rules for Construction of Nuclear Facility Components”, 2013.
- 539 ASTM E1049-85(2017), "Standard Practices for Cycle Counting in Fatigue Analysis." West  
540 Conshohocken, PA: ASTM International, 2011.
- 541 Ballester, J.L. « Contribution to the study of the ageing of Pelton wheels: comparative damage ratios »  
542 (in French). Internal report EdF DTG MPSH, D.4100/EHM-G-92/08, 1992.
- 543 Bakken, B.H., (2002). Hydro unit startup costs and their impact on the short term scheduling strategies  
544 of swedish power producers. IEEE Power Engineering Society Summer Meeting.
- 545 Bakken B.H., Bjørkvoll T., « Hydropower Unit Start-up Costs », IEEE, 2002.
- 546 Balaji Rao, K., Anoop, M.B., Raghava, G., Prakash, M. and Rajadurai, A., Probabilistic fatigue life  
547 analysis of welded steel plate railway bridge girders using S-N curve approach. 2013, Proceedings  
548 of the Institution of Mechanical Engineers, Part O: Journal of Risk and Reliability 227: 385
- 549 Bureau of Reclamation, Hydrogenerator Start / Stop Cost, US 2014.
- 550 Brand, A., Flavenot J. F.; Gregoire R. and Tournier C., Technological data on fatigue, Senlis, CETIM,  
551 1999, 4<sup>e</sup> ed., 383 p. (in French).
- 552 Bryla, P., Pillou, M. and Bennebach, M., Operation of the Bois plant hydraulic turbine in bypass  
553 operating mode: analysis of the fatigue behavior, Procedia Engrg 66 (2013), 413-450.
- 554 Buxbaum, O. & Ostermann, H. Ausfallsichere Bemessung von Laufrädern für Wasserkraftmaschinen  
555 aus rostfreiem Stahlguss unter Berücksichtigung von Korrosion und Gefügezustand, Fraunhofer  
556 Institute for Structural Durability, Darmstadt, Germany, 1983. (Technical report in German)
- 557 CEN, 2002. Eurocode 0. Bases of Structural Design. Brussels: CEN.
- 558 Chopra, O.K. & Shack, W.J., Argonne National Laboratory, Review of the Margins for ASME Code  
559 Fatigue Design Curve - Effects of Surface Roughness and Material Variability. U.S. Nuclear  
560 Regulatory Commission Office of Nuclear Regulatory Research Washington, DC 20555-0001, 2003.

561 Dojčinović, M., Arsić, M., Bošnjak, S., Murariu, A. and Malešević Z., Cavitation resistance of turbine  
562 runner blades at the hydropower plant 'DJERDAP', *Structural Integrity and Life*, Vol.17(1), 2017,  
563 55–60.

564 Dorji, U. & Ghomashchi, R. « Hydro turbine failure mechanisms: An overview », *Engineering failure*  
565 *analysis* 44 (2014) 136–147, Elsevier, 2014.

566 EN 1993-1-9 (2005): Eurocode 3: Design of steel structures - Part 1-9: Fatigue [Authority: The European  
567 Union Per Regulation 305/2011, Directive 98/34/EC, Directive 2004/18/EC].

568 EdF & GE, « Prototype Air Injection Tests », confidential report, 2018

569 Fouchereau, R., Probabilistic modelling of S-N curves, PhD thesis, University Paris-Sud, 2014. (in  
570 French)

571 Gagnon, M., Tahan, S. A., Bocher, P. and Thibault, D. « Impact of start-up scheme on Francis runner  
572 life expectancy » 25th Iahp Symposium on Hydraulic Machinery and Systems, Iop Conf. Series :  
573 Earth and Environmental Science 12 (2010) 012107

574 Gagnon, M., Tahan, S. A., Bocher, P. and Thibault, D. « A probabilistic model for the onset of High  
575 Cycle Fatigue (HCF) crack propagation: Application to hydroelectric turbine runner » *Int. J. Fatigue*  
576 47 (2013) 300–307.

577 Gagnon, M., Thibault, D. and Blain, M., « On the expected monetary value of hydroelectric turbine  
578 start-up protocol optimisation », 13<sup>th</sup> world congress on Engrg Asset Managt, WCEAM, 2018.

579 Grandemange, J-M, Faigy C., General approach to preventing the risk of fatigue damage to nuclear  
580 boiler equipment: from design to monitoring during operation SFEN-ST2 Day "Fatigue damage to  
581 nuclear installations", Paris, 23 November 2000 (in French)

582 Hassanipour, M, Y. Verreman, Y., Lanteigne, J. and Chen J., (2015) Effect of Periodic Underloads on  
583 Fatigue Crack Growth in Three Steels Used in Hydraulic Turbine Runners, *Int. J. of Fatigue* 85,  
584 DOI: [10.1016/j.ijfatigue.2015.11.027](https://doi.org/10.1016/j.ijfatigue.2015.11.027)

585 Huth, H-J, *Fatigue Design of Hydraulic Turbine Runners*, Doctoral thesis, Fakultet for  
586 ingeniørvitenskap og teknologi, 2005

587 ISO 20816-5:2018 Mechanical vibration — Measurement and evaluation of machine vibration — Part  
588 5: Machine sets in hydraulic power generating and pump-storage plants (2018)

589 Jabbado, M. Polycyclic fatigue of steel structures: life under varying loads. *Mechanics of materials*  
590 Doctoral thesis of the Ecole Polytechnique, 2006 (In French)

591 Jacquemoud, R. "Towards a reliable operating range for a Kaplan turbine, beyond the guaranteed range,  
592 based accounting for material fatigue damage processes", XVII IA HR symposium, Beijing, China  
593 1994. (in French)

594 Kokko, V. (2014). Ageing due to start-stop cycles and frequent regulation in lifetime estimation of hydro  
595 turbines and generators. SHF, Grenoble.

596 Lei, X., Yuan, L., Peng, L., Sun, C., Wei, B. and Wei Y., Fatigue endurance limit and crack front  
597 evolution in metallic glass, *Int. J. Fatigue* 143 (2021) 106004.

598 Ling, Y., Shantz, C., Mahadevan, S. and Sankararaman, S., 2011. Stochastic prediction of fatigue  
599 loading using real-time monitoring data. *International Journal of Fatigue* 33, 868–879.  
600 <https://doi.org/10.1016/j.ijfatigue.2011.01.015>

601 Liu, X., Luo, Y. and Wang Z., A review on fatigue damage mechanism in hydroturbines, *Renewable*  
602 *and Sustainable Energy Reviews* 54(2016), 1–14.

603 Lofflad J., Eissner M. and Graf B. « Strain Gauge measurements of rotating parts with telemetry » 9th  
604 International Conference on Hydraulic Efficiency Measurements, Trondheim, 2014.

605 Luna-Ramírez A., Campos-Amezcuca A., Dorantes-Gómez O., Mazur-Czerwiec Z., Muñoz-Quezada R.,  
606 Failure analysis of runner blades in a Francis hydraulic turbine — Case study, *Engineering Failure*  
607 *Analysis*, Vol. 59, 2016, pp 314-325. <https://doi.org/10.1016/j.engfailanal.2015.10.020>

608 Marcouiller, L. & Thibault, D. « Obtaining stress measurements on runners as a key contribution to  
609 reducing their degradation and improving the reliability of hydroelectric production units » HYDRO  
610 conference, Bordeaux, 2015.

611 Matsuishi, M. & Endo, T. (1969), « Fatigue of metals subjected to varying stress », In *Proceedings of*  
612 *the Kyushu Branch of Japan Soc. of Mech. Engrg*, Fukuoka, Japan (in Japanese), 37-40.

613 Mjøltnes, J. "Frequent Start/Stop. Costs for Hydropower Generators," *Statkraft Engineering*, Report  
614 99/87, Sept. 1999 (in Norwegian).

615 Morissette, J.F., Chamberland-Lauzon J., Nennemann B., Monette C., Giroux A.M., Coutu A. and  
 616 Nicolle J., Stress predictions in a Francis turbine at no-load operating regime, 28th IAHR symposium  
 617 on Hydraulic Machinery and Systems, 2016, doi:10.1088/1755-1315/49/7/072016  
 618 Nicolet, C., Avellan, F., Allenbach, P., Sapin, A., Simond, J.-J., Kvicinsky, S., and Crahan, M.  
 619 Simulation of transient phenomena in Francis turbine power plants: hydroelectric interaction, 1–12.  
 620 13th conference on Waterpower, July 29–31, 2003 in Buffalo, New York, USA.  
 621 Nilsson, O. & Sjelvgren, D. (1997). Hydropower unit startup costs. IEEE Transactions on Power  
 622 Systems.  
 623 Savin, O., Badina, C., Baroth, J., Charbonnier, S. and Bérenguer, C., Start and stop costs for hydro  
 624 power plants: a critical review, Proc. of the 30th Europ. Safety and Reliability Conf. & the 15th Prob.  
 625 Safety Assessment and Management Conf., 2020.  
 626 Schijve, J., Statistical distribution functions and fatigue of structures, Int. J. Fatigue 27 (2005) 1031–  
 627 1039.  
 628 Seddik, R., Ben Sghaier, R., Atig, A. and Fathallah, R., Fatigue reliability prediction of metallic shot  
 629 peened-parts based on Wöhler curve, Journal of Constructional Steel Research 130 (2017) 222–233  
 630 Sonsino, C.M. & Dieterich, K., Fatigue strength of steel types G - X5CrNi 13 4 and G - X5CrNi 17 4  
 631 used for blades of water - power turbines and pumps, Werkstoffe und Korrosion 41, 330-342, 1990.  
 632 Sudret, B. (2011). Probabilistic design of structures submitted to fatigue, Fatigue of materials and  
 633 structures, chapter 5 (C. Bathias and A. Pineau (Eds)), 223\_263. Wiley & Sons.  
 634 Susmel, L. Modified Wöhler Curve Method, theory of critical distances and Eurocode 3: a novel  
 635 engineering procedure to predict the lifetime of steel welded joints subjected to both uniaxial and  
 636 multiaxial fatigue loading. Int J Fatigue 2008; 30: 888–907.  
 637 Susmel, L. The Modified Wöhler Curve Method calibrated by using standard fatigue curves and applied  
 638 in conjunction with the theory of critical distances to estimate fatigue lifetime of aluminium  
 639 weldments. Int. J. Fatigue 2009; 31: 197–212.  
 640 Yung-Li, Lee, Y.-L. and Tjhung, T. A new definition of the rainflow cycle counting method, Metal  
 641 Fatigue Analysis Handbook, 89-114, 2012.  
 642 Welte, T. Deterioration and maintenance models for components in hydropower plants, Doctoral thesis,  
 643 Norwegian University of Science and Technology (NTNU), 2008.  
 644 Wöhler, A., 1860. Versuche zur Ermittlung der auf die Eisenbahnwagenachsen einwirkenden Kräfte  
 645 und die Widerstands higkeit der Wagen-Achsen. Zitschrift für Bauwesen, Vol. 10, 583- 616. (In  
 646 German).  
 647