STANDARDISED FRICTION DAMPER BOLT ASSEMBLIES TIME-RELATED RELAXATION AND INSTALLED TENSION VARIABILITY

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ABSTRACT

The sliding hinge joint is a type of low-damage seismic resistant connection equipped with a bolted friction damper at the bottom beam flange. To accurately control its flexural resistance, it is critical to govern the bolts' preload which depends on complex issues related to the installation procedure, and the short- and long-term phenomena. Despite the influence of these factors on the initial and life-time behaviour of bolts, currently, little information exists. Nevertheless, a statistical characterisation of the variability of the preloading force (initial and during the life-time) would be needed, in order to develop reliable design guidelines for these connections.
Within this framework, this paper examines experimentally, the variability of the preloading force of European bolt assemblies applied in friction dampers, through continuously monitoring the preloading at installation over a period of time. This was done to analyse the accuracy of the standardised installation procedures and the rate of loss of the initial tension over time. The tests have evidenced a higher accuracy of the torque method, highlighting some criticisms of the combined method which, conversely, proved to be inaccurate as currently codified. The short- and mid-term tests have shown that the estimated loss after 50 years, in case of assemblies with normal washers or with European standardised disc springs is, on average, equal to 10% and 27%, respectively. Additionally, in all the cases, the greatest part of the total loss (≅70%) occurred in just 30 days, highlighting that time-dependent phenomena are mainly concentrated in the first days after tightening.

Keywords: Friction damper, tightening procedures, loss of preloading, bolted connections, disc springs

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1. INTRODUCTION

Eurocodes provide to design structures to guarantee minimum performance levels under a set of design load combinations [1,2]. Typically, design procedures are based on checks for Serviceability Limit States (SLS) (related to the most frequent load combinations occurring during the life-time of the structure) and Ultimate Limit States (ULS) where, the structure, in case of the occurrence of rare load combinations (such as those related to seismic events with high return period), can be designed to dissipate energy in selected zones.

In case of steel Moment Resisting Frames (MRFs), according to the design procedures suggested by EC8 [3], the energy dissipation can be concentrated both in beams ends or in joints. In the former case, the design is aimed at promoting the formation of plastic zones in beams through the adoption of full-strength joints and over-strength columns while [4-7], conversely, in the latter case, the joints are designed to transfer to the columns only a part of the bending moment through the adoption of partial-strength joints [3,8,9]. Independently from the adoption of one or another design strategy, the main drawback of the traditional approaches is the need for the development of structural damage which, even though on one hand is used to preserve structural economy and human life, on the other hand represents also the main source of direct and indirect losses occurring in case of rare seismic events. Over the past decades, several strategies have been proposed in order to solve this issue. Among these, supplemental energy dissipation systems have been suggested, attempting to reduce structural damage, and proposing to concentrate energy dissipation in replaceable devices [10]. Nevertheless, even though supplementary energy dissipation design strategies can concentrate the greatest part of damage in specific fuses, the structural damage cannot be completely avoided because, to activate the seismic devices, adequate sway displacements, typically leading to the plastic engagement of the structure, are still needed.

In order to overcome these issues, recently, alternative low-damage strategies, based on the inclusion of friction dampers in the connections of MRFs, have been proposed. One of the first connections applying these principles was initially developed by Grigorian et al. in 1993 [11], providing pioneering studies on slotted bolted connections with friction interfaces made of mill scale steel or mill scale steel and brass. Following these initial works, many other studies have been carried out, especially in New Zealand, developing a further generation of friction connections, the so-called Sliding Hinge Joints (SHJ) which also help to keep any additional costs of the innovative solution to a minimum. These are characterised by extremely simple details based on the inclusion of Asymmetric Friction Connections (AFC) at the beam bottom flange, with shims made of mild steel, aluminium, brass or – in the most recent versions – abrasion-resistant steel (e.g. [12-18]). Similar solutions have also been recently patented in Japan [19], extending a previous concept of a friction-bolted slotted anti-seismic damper [20]. More recently, also in the European framework, similar solutions have been proposed suggesting new layouts, in which the friction damper is realised with steel angles and an additional haunch, welded or bolted to the beam bottom flange [21,22]. This layout, although probably not as simple as the SHJ, provides the possibility to realise the whole damper as a separate element in the shop (the haunch is represented in Fig.1, which can be produced and bolted on site to the beam), allowing, in this way, a better control on the materials' quality (e.g. higher control of the surface conditions, continuous factory controls on the production, control on the employed bolts' quality), and on the application of rigorous bolts installation procedures complying with the relevant European standards. The typical beam-to-column joint, recently proposed in Europe for application in semi-continuous steel Moment Resisting Frames (MRFs), represents an alternative to the classical bolted connections, consisting in a modification of the detail of a Double Split Tee joint (DST) where, in place of the bottom Tee, a bolted friction connection is realised with a slotted haunch slipping on friction shims pre-stressed with high strength bolts (Fig.1). All these elements, practically, realise at the bottom beam flange a Symmetrical Friction Connection (SFC). Adopting such a type of detail, under seismic actions, the beam is forced to rotate around the pin located at the base of the upper T-stub web and the energy dissipation is provided by the alternate slippage of the lower beam flange on friction shims (Fig.1). This detail is beneficial in isolating floor slab contribution and preventing frame elongation.

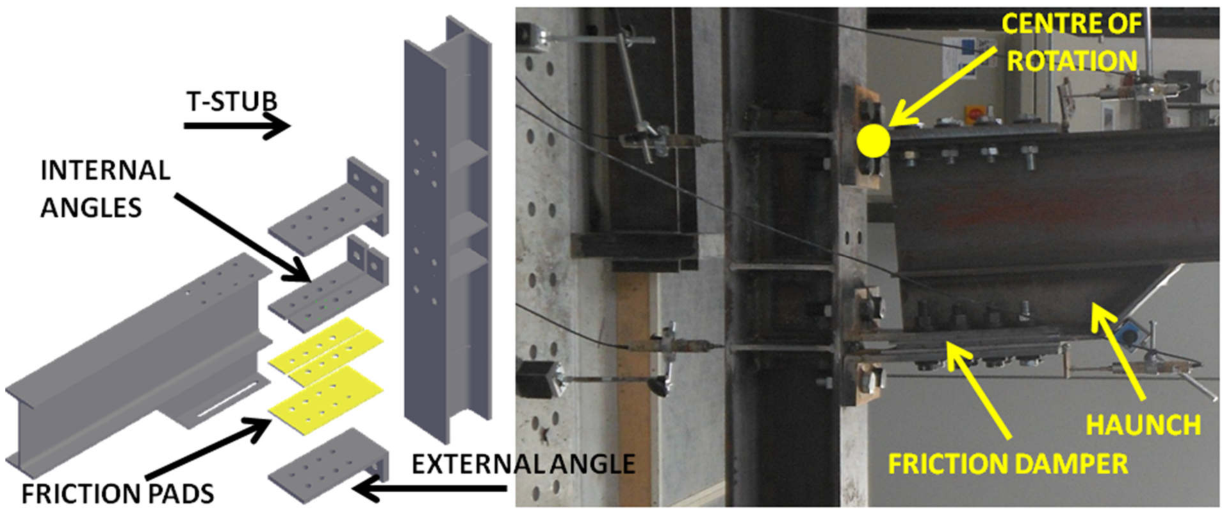


Fig. 1 **–** Typical layout of the connection studied in [21, 22]

This connection, similarly to the SHJ, is intended to behave as rigid under SLS, and to allow beam-to-column inelastic rotation at the ULS. Additionally, through the application of proper hierarchy criteria, both at the global and local level, it can be easily designed to be the only source of energy dissipation of the whole structure. The typical design of these connections should follow these conceptual steps:

1) ***Design of the friction damper*** for the actions deriving from the ULS load combinations. The dampers, in order to avoid the plastic engagement of the beam, should be designed in order to develop a flexural resistance of the connection corresponding at most to the plastic resistance of the beam;
2) ***Design of the non-dissipative parts of the connection***, accounting for the maximum over-strength due to random variability of the friction material. It is worth noting that the component of the over-strength related to the material strain-hardening is negligible, because the friction damper is characterised by a rigid-plastic response. Additionally, in terms of stiffness, being the slip resistance of the friction damper uncoupled from the stiffness of the connection, the joints elements can be designed to achieve a full rigidity, with a clear advantage with respect to the classical semi-continuous design for the serviceability limit state checks [3,8];
3) ***Design of columns*** by means of the adoption of the classical standardised procedures (e.g. EC8) or even by means of more advanced design procedures, such as the Theory of Plastic Mechanism Control (TPMC) [23]. Accurate procedures, such as the TPMC, are able to assure the development of a failure mechanism of global type and the reduction of structural damage occurring in steel members, to zero.

Within this framework, it is clear that the capacity design of all the structural parts depends on the definition of the sliding resistance of the damper. In order to govern the bending strength of the joint and to control the resistance hierarchy of the whole structure, it is critical to characterise two main parameters: the friction coefficient of the interface, and the bolts forces at installation and during the life-time of the connection. Clearly, the friction coefficients – static (to be used in serviceability limit state design) or dynamic (to be used in ultimate limit state design) – of the friction devices depend on the materials employed to realise the dissipative interface and, specifically, they depend on the tribological properties of the shims (micro and macro hardness, shear resistance, roughness, superficial finishing, etc.) used in the damper. This topic has already been subject of studies by several research groups worldwide [24-27] and it is currently in phase of study by the authors, who are providing a statistical characterisation of the friction coefficient values for several possible materials, defining upper and lower bound regression models able to provide the friction coefficient as a variable dependent on the cumulative travel [28,29]. These models can be used to define to design values of the friction coefficient under different load combinations (Serviceability (SLS) and Ultimate Limit State (ULS) design). A possibility for the friction sliding design could be to adopt the lower bound value of the dynamic CoF (Coefficient of Friction) to design the damper at ULS, the upper bound value of the static CoF to design the non-dissipative parts of the structure and the lower bound of the static CoF to design the joint under SLS load conditions. Alternatively, the nominal capacity of the damper could be designed adopting the average CoF using lower bound and upper bound values of the static CoF to determine strength reduction and overstrength factors.

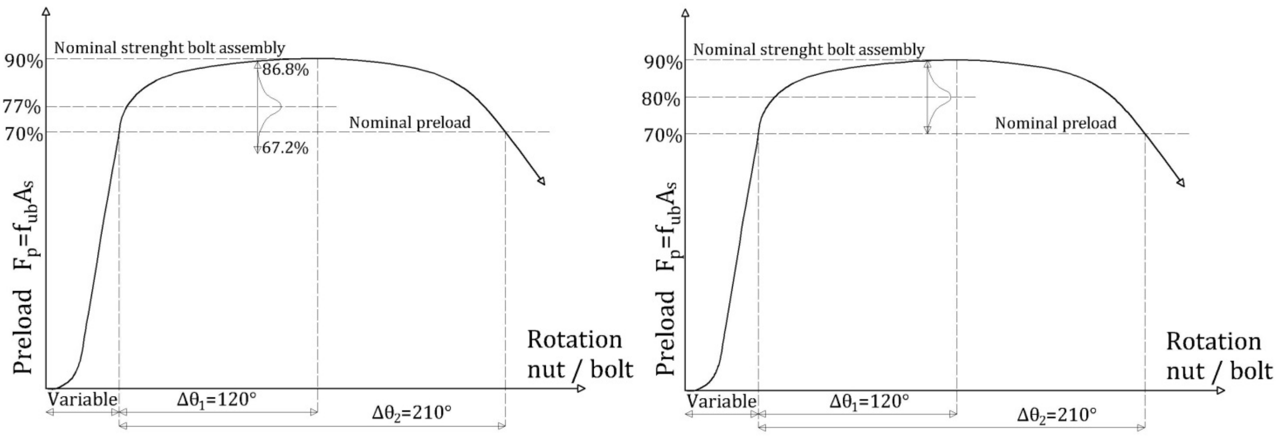
Conversely, the bolt preloading force, after installation and during the life-time of the connection, aside from the procedure applied to tighten, depends on many complex phenomena involving: i) embedment relaxation, ii) bolt creep, iii) vibrations, iv) elastic interactions, and v) differential thermal expansions [30-36]. These effects are usually influenced by phenomena not easy to predict by means of a deterministic modelling, such as the velocity of application of the pre-load with the torque wrench, the tightening procedure, the crushing of micro-spots of the steel plates under the bolt head or nut and the creep of the coating materials and, therefore, to be quantified, need to rely on experimental investigation. Dealing with the tightening procedures, EN 1090-2 [37] currently recommends four different methods for the installation of High Strength bolts (both of HR or HV type) in friction bearing connections (torque method, combined method, HRC, DTI). These methods are calibrated and checked only for traditional friction connection. Therefore, to extend their validity also to AFC or SFC dampers, they need a preliminary experimental study. In fact, while the classical friction connections are realised, all with steel plates uncoated or coated with paintings or zinc, in case of friction joints such as those herein described, the bolts are used to pre-stress particular types of friction shims with possible differences in terms of initial or long-term response.

The paper has two main objectives. The first one is to assess the accuracy of some of the European standardised tightening procedures on sub-assemblies of symmetrical friction dampers. The second one is to assess the possible influence of time-related effects on the initial bolt tension, with specific tests carried out for periods of time, extending up to one month. This duration has been sufficient in almost all cases to determine the tangent of the displacement-log time curve allowing, therefore, to extrapolate the data according to a procedure similar to that suggested by EN 1090-2 for creep tests. In addition, the efficiency of the only European standardised type of Belleville spring washer (which complies to DIN 6796 [38]) in maintaining the preloading of the bolt assemblies constant during the life-time, is assessed. In order to analyse all these aspects, an experimental analysis regarding 84 tightening tests on specimens of friction dampers is presented. All the specimens have been tested measuring with annular load cells the force applied to the bolts, in order to evaluate the initial achievement of the target preload adopting one of the standardised tightening procedures. Additionally, in 26 cases, the loss of preload occurring over a period of time has also been monitored. The additional loss of preloading, induced by the alternate slippage of the connection occurring in case of rare seismic events, has been neglected in this work. This is being evaluated in another separate research regarding the behaviour of sub-assemblies of friction dampers subjected to low velocity and high velocity cyclic loads [29]. The obtained results are hereinafter critically discussed, proposing, at the end, a statistical characterisation and a regression study of the loss of preloading in order to provide a simple tool for design.

2. TIGHTENING PROCEDURES ACCORDING TO EN 1090-2

EN 14399 [39], EN 898-1 [40] and EN 15048 [41] identify the different types of bolts to be used in steel structures as well as their geometrical and mechanical features. European bolt classes are divided into two macro-categories depending on the type of assembly: pre-loadable assemblies of HV, HR, and HRC type, and standard assemblies (SB) for normal connections. Dealing with the first category, for friction bearing connections, EN 1090-2 [37] introduces four different methods to tighten bolts: torque, combined, HRC, and DTI.

The ***torque method*** is, basically, a force control (torque control) procedure, and it is divided into two steps. During the first step, 75% of the torque reference value (i.e. $T_r=0.7A_b f_{ub} kd$, where A_b is the bolt's net area, f_{ub} is the ultimate stress of the bolt's material, d is the bolt's nominal diameter and k is a constant depending on the bolt's assembly) is applied to all the bolts of the connections. During the second step, the torque is increased to the 110% of the torque reference value in all the bolts of the connection. This increase of the 10% of the tightening with respect to the nominal value is implicitly aimed at achieving a target mean preload equal to $(1+1.65V_k)x0.7A_b f_{ub}$, being V_k the coefficient of variation, assumed by the code as equal to 0.06 (see clause 8.5.3 of [37]). Therefore, the increment of the 10% is meant to consider the random variation of the bolts tightening in order to guarantee contemporarily that the 5% fractile of the pre-load (lower bound value) is higher than the nominal preload ($0.7A_bf_{ub}$), and that the 95% fractile (upper bound value) is lower than the bolt's nominal tensile strength ($0.9A_bf_{ub}$) (Fig.2a). The first condition is to assure that the actual preload in the assembly is higher than the design value, the second one, to guarantee that the applied tightening torque does not exceed the bolt yield strength. Nevertheless, even though this is the aim of the code, it is easy to verify that, as already demonstrated by Berenbak [28], the torque method, as currently codified, is not able to assure the minimum 95% reliability required by EC0 [1]. In fact, the adoption of a coefficient of variation for the bolt assemblies equal to 0.06 [28] does not lead to the needed reliability due to the influence of other random effects that should be considered in design. In fact, aside from the random variation of the k-factor, equal to 0.06, there are other random parameters that influence the estimate of k, namely the accuracy of the bolt measuring device used in the k-test (± 0.02) and its repeatability (± 0.01), the accuracy of the bolt torque measurer used in the k-test (± 0.01) and its repeatability and the accuracy of the torque wrench used to apply the preload which, as required by the EN 1090-2, for the torque method, must be at most equal to the ±4%. All these factors, if combined with the maximum coefficient of variation allowed for bolt assemblies (0.06), would lead to a combined value of the coefficient of variation equal to 0.077. Due to this, it could be verified with simple calculations that when a target mean value of $1.1x0.7A_bf_{ub}$ is assumed, the reliability with respect to the upper bound value is guaranteed (98.6%), but it does not guarantee the reliability for the nominal preload which is only equal to the 88.2% (Fig.2a). It is worth noting that, in order to respect the minimum reliability of the 95% for both bound pre-load values, the torque procedure could be easily corrected, imposing a target mean value equal to $0.8A_bf_{ub}$ [42] (Fig.2b). Additionally, it must be underlined that the torque method, relying on a pre-qualification of the bolts based on the so-called k-test (the test used to determine the correlation factor between torque and pretension), needs to be applied only on bolts which are exactly in the same conditions of the tested bolts. Therefore, they must have no rust or dust in the threads, and they must be in the same lubrication state. This requirement can be practically considered satisfied only if the installed bolts are delivered and taken from closed boxes and if, before installation, it is possible to verify that the nut is able to turn freely through performing a free-nut turn check.



a) b)

Fig. 2 **–** Torque method. a) Current procedure with $V_k=0.06$ and mean value equal to $0.77f_{ub}A_b$; b) Improved procedure with mean value equal to $0.80f_{ub}A_b$.

Conversely, the ***combined method*** is substantially a displacement-control (turn-of-the-nut control) procedure, divided into two phases: a first one equal to the first step of the torque method, and a second consisting in a controlled part-turn rotation of the nut determined as a function of the thickness of the assembly. This procedure, as opposed to the previous one, is able to provide a 100% reliability with respect to the minimum tightening force required, because the displacement control phase is meant to assure that the preloading force achieves the plastic branch of the pre-load/elongation curve. Nevertheless, as shown hereinafter, this procedure does not allow a clear control of the upper bound value of the preloading force and, in fact, can lead in many cases to the development of pre-loads higher than the bolt nominal tensile strength (Fig.3).

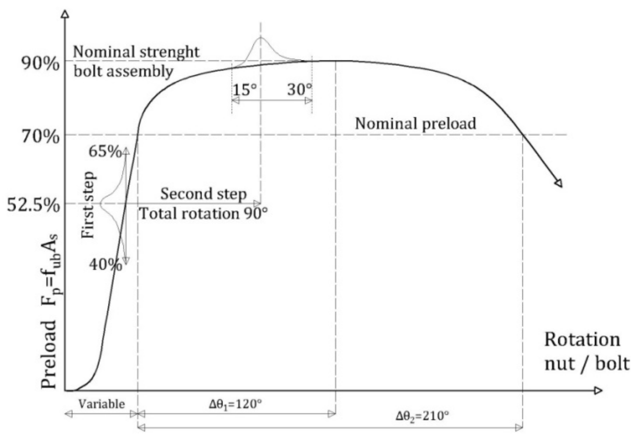


Fig. 3 **–** Combined method, distribution preload according to EN 1090-2

The ***HRC system*** is very different from the other methods because it requires a particular assembly and the employment of a special wrench equipped with two co-axial sockets that react against each other to tighten the bolt. In this case, it is not necessary to calibrate the wrench because the maximum torque is determined by the strength of the splined end. In any case, the accuracy of this method also depends on the quality of the threads which must be checked prior to installation. Finally, the ***Direct Tension Indicator*** (DTI) method comprises the use of special washers with compressible nibs on one face that deform to indicate when the maximum preload has been reached. The tightening process is divided into two steps: during the first step, the bolt assembly is tightened until the nibs just begin to deform, while during the second step the bolt assembly is tightened until reaching the full compression of the protrusion.

This work seeks to evaluate the possibility to extend the two easiest procedures to symmetric friction dampers, namely the torque, and combined methods. These tightening methods can be applied using only HV or HR bolts. The HV and HR assemblies are very similar, and they can both be used for friction bearing connections. Their main difference is the shape of the threads and the height of the nut, which lead to a different failure mechanism under tension: in the HR system, the failure occurs in the bolt's threaded part, while in the HV system the failure occurs in the threads of the nut. This failure mode of the HV system is not typical in other countries but, nevertheless, represents the most common type of pre-loadable bolt assembly in Europe. Further discussions on this topic are reported in [43, 46].

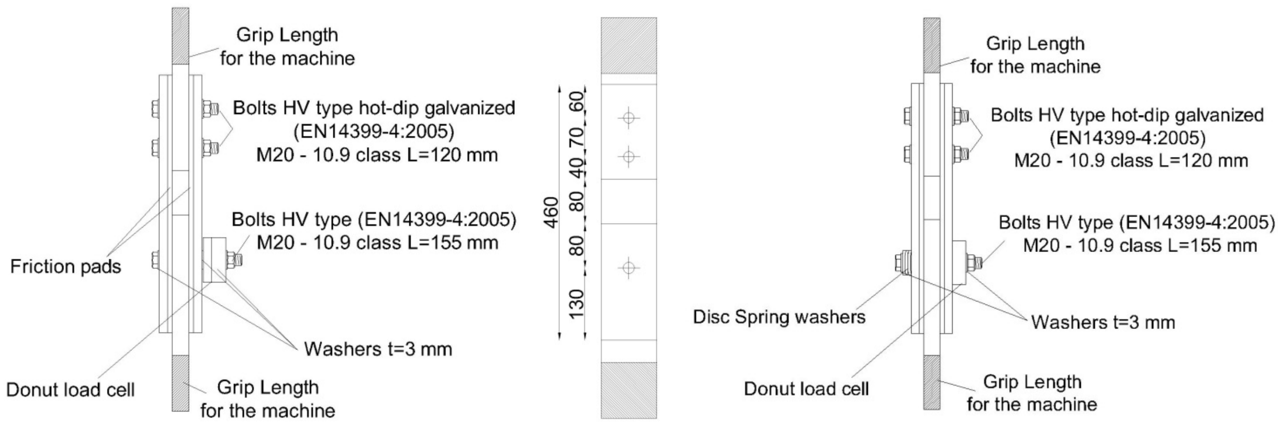
From the point of view of the qualification procedures for bolts belonging to preloaded assemblies, the relevant European standards, EN 14399-3/4 [44,45], and EN 1090-2 [37], divide the bolts into three classes: K0 class (no requirements for the k-factor); K1 class (minimum 5 tests on bolts and $0.10 \le k_i \le 0.16$); and K2 class (minimum 5 tests on bolts and $0.10 \le k_m \le 0.23$ with $V_k \le 0.10$). K2 bolts can be used for both, the combined and the torque method, assuming $k=k_m$ (declared by manufacturer), while K1 bolts can be used only for the combined method. However, in this case, the value of the *k* coefficient is assumed equal to 0.13 for the first tightening step (the mean value between 0.10 and 0.16), if not differently specified. It is worth noting that currently, the minimum requirements for the *k*-classes are poorly defined. In fact, as explained before, the torque method for a consistent application needs a maximum value of the coefficient of variation to be equal to 0.06 as required by EN 1090-2 [37], while EN 14399 [44,45] allows the use of the torque method for bolts having a coefficient of variation up to 0.10. Hence, in order to fulfil both the requirements of EN 1090-2 and EN 14399-3/4, bolts which are contemporarily complying with the requirements for K1 and K2 classes with $0.10 \le k_i \le 0.16$, $0.10 \le k_m \le 0.16$, with a coefficient of variation lower than

0.06 are currently available on the market. In the tests reported subsequently, bolt assemblies of this type have been used and their use is suggested, in general, for the application of the torque method.

3. TIGHTENING TESTS AND SHORT-TERM RELAXATION TESTS ON SFC SUB-ASSEMBLIES

3.1 Experimental layout

In order to evaluate the accuracy of the EN 1090-2 tightening procedures and the possible short- and mid-term loss of the initial tension of bolts installed in friction dampers, 58 tightening tests on simple bolt assemblies and 26 relaxation tests were performed at the laboratory STRENGTH (STRuctural ENGineering Testing Hall) of the University of Salerno. The typical specimen adopted to evaluate the accuracy of the tightening procedures is a simple modification of the specimen suggested by EN 1090-2 [34] for slip tests adapted to the case of a SFC connection. It is constituted by 8 mm friction shims, realised with sand-blasted steel plates coated with three different coating materials, a couple of 15 mm hot-dip galvanised external steel plates and a couple of plates, one realised with S275JR steel and normal holes (upper part of the specimen, Fig.4) and the other realised with stainless steel and a slotted hole (steel equivalent to the AISI 304 [47], lower part of the specimen, Fig.4). The mild steel plates (made exception for the friction shims) are all hot-dip galvanized, while the internal stainless steel plate is untreated. The specimen is designed to be similar to the typical configuration of an SFC employed in SHJs recently tested and similar to the SFC reported in [21,22] (Fig.1). In fact, the stainless-steel plate simulates the internal plate of the haunch located under the bottom flange of the beam (Fig.1), while the external plates simulate the angles used to fasten the friction pads to the column. In the upper and lower part of the specimen two bolts and one bolt M20 10.9 HV have been used, respectively, to tighten the specimen. The bolts used in the experimental analysis are produced by SBE-Varvit S.p.A (Italy).



a) b)

Fig. 4 **–** Configurations adopted for the tightening tests. a) HV washers; b) HV + Disc Spring washers.

The tests were performed measuring the preload applied, the tightening torque and the rotation of the nut using specific devices. The tightening torque, applied through a hand torque wrench calibrated according to EN 1090-2 [37] in order to reach the accuracy of +/-4%, was also monitored, using a torque sensor FUTEK TAT430 with a maximum capacity equal to 680Nm. Conversely, the preload applied to the bolts was measured through a donut load cell, FUTEK LTH500, with maximum capacity of 222.4 kN located on the side of the bolt's nut. Finally, to measure the rotation of the nut, a digital angle meter, USAG 831A, with a tolerance of ±2% was used.

The tested bolts are 10.9 class HV with size M20x155 mm with the following characteristics certified by the manufacturer: km=0.119 and V_k<0.06, so that they can be classified both as K2 or K1 class according to EN 14399-2 [48] and they can be tightened using the torque method or the combined method, indifferently (EN 1090-2 [37]). Two different configurations have been tested: one with standard flat washers (EN 14399-6 [49]) (Fig.4a) and another one with flat and disc springs washers (DIN 6796 [38]) (Fig.4b). The normal flat washers have been used also in the second case to comply with the value of the k-factor provided by the bolt's manufacturer. The disc spring washers, also called Belleville, are conical washers made of high strength steel, able to compress elastically, provided that they are properly pre-set, until the full squash load [50]. Conversely, the standardised European disc spring washers (complying to DIN 6796), which are not pre-set, provide an elastic behaviour until reaching a threshold value beyond which they show a significant increase of stiffness (Fig.5).

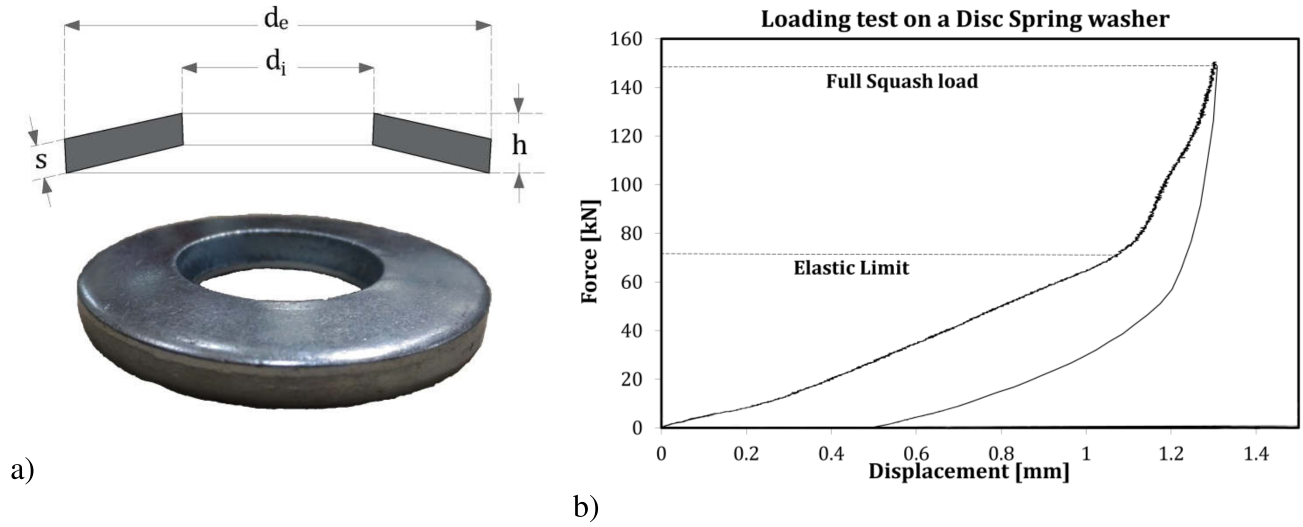


Fig. 5 – Disc Spring washers. a) Geometric features of a Disc Spring washer; b) Experimental behaviour of a disc spring washer.

The aim of the introduction of these washers in SFC is to increase the axial deformability of the bolt assembly in order to limit the loss of preload due to long-term relaxation, vibration, and thermal effects [17, 18, 30-36, 51]. In fact, during the life-time of a friction connection, when the bolts relax or the coatings of the friction shims creep, the disc washers act as springs, pushing the bolt and restoring the preload force initially applied. The Belleville springs can be assembled in different ways to create a system of desired stiffness. Usually, it is possible to stack them one over the other, obtaining an increase of stiffness and resistance proportional to the number of disc springs (parallel configuration); face-to-face, obtaining an increase of the deformability proportional to the number of disc springs (series configuration), or in groups of series and parallels. Preliminarily, in order to determine the load-bearing capacity of the disc springs, a compression test has been performed highlighting that the standardised European disc springs for M20 bolts are able to resist to a force of about 73 kN elastically, value beyond which they exhibit an increase of stiffness until complete flattening (Fig.5b). Considering that the single disc spring can resist to 73kN, three disc springs in parallel have been necessary to withstand the upper bound value of the tightening force which, as explained before, may be at most equal to the bolt's yielding resistance (for M20 bolts, class 10.9, 245x900=220.5 kN). Therefore, in order to understand the influence of the disc springs over the tightening procedure and the short- or mid-term relaxation, as an alternative to the configuration with normal washers, a simple configuration composed of three disc springs arranged in parallel has been considered. Obviously, this configuration is selected only to provide a first comparison of the behaviour of normal flat washers and standardised Belleville washers. Further experimental efforts should be, eventually, devoted to understanding the influence of the configurations or of other disc springs typologies on the short-term or long-term behaviour of the bolt assembly.

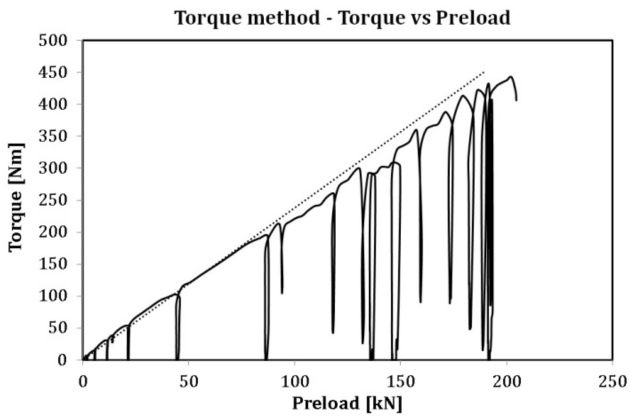
3.2 Tightening tests

The accuracy of the EN1090-2 tightening methods applied to SFC dampers has been verified carrying out 36 tightening tests on bolt assemblies employing normal flat washers and 22 tightening tests on bolt assemblies employing also disc springs. The typology and number of tests performed are summarised in Table 1.

Table 1. Tightening tests' list.

Flat washers	*Torque method*	15 tightening tests
	Combined Method	21 tightening tests
Disc Springs	*Torque method*	11 tightening tests
	Combined Method	11 tightening tests

The torque method has been applied adopting the procedure previously described, namely: during the first step, the target preload has been fixed equal to $0.75F_p=128.6kN$, while during the second one, the target preload force has been assumed equal to $1.10F_p=188.7kN$.
Conversely, the combined method has been executed applying to the specimens initially a tightening control phase analogous to the one performed with the torque method and, afterwards, a controlled part-turn rotation of the nut. In the case under study in this paper, as far as the total thickness of the specimens was equal to 84mm, the rotation of the nut was assumed equal to 90° according to the EN 1090-2 provisions.
In Fig.6 the results of the application of the two different procedures are delivered, representing in terms of *torque vs preload* and *preload vs time* a typical tightening session, both for the torque and for the combined method. In both cases, it is possible to note that the relationship between the torque and the preload is linear, with a slope corresponding approximately to the k-factor. Additionally, in the *preload vs time* curves, the two loading steps previously mentioned are easily recognisable, showing the approximate achievement of the target values. All the specimens have been tightened following these methodologies, and the results are herein critically discussed.



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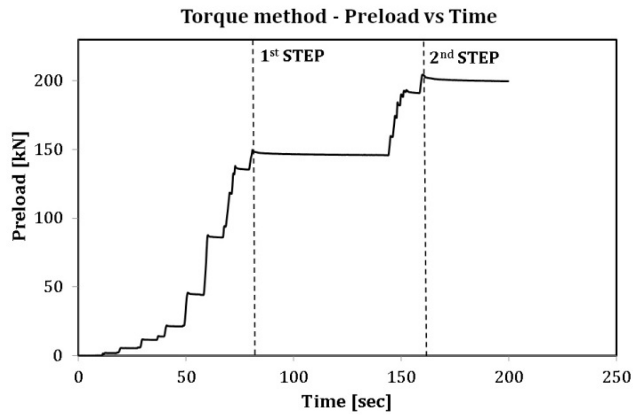

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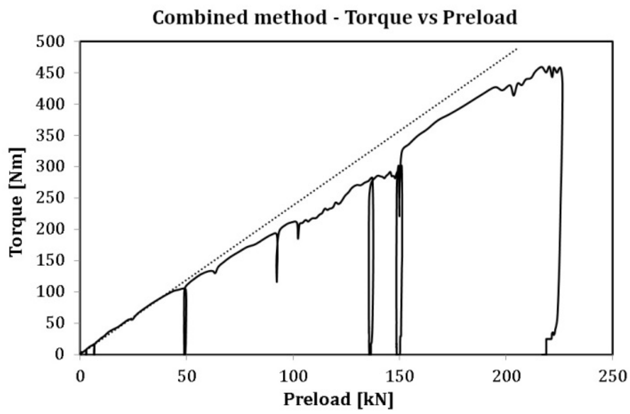

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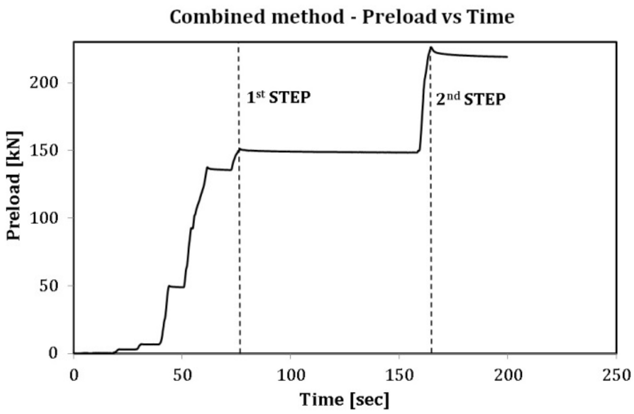

d)

Fig. 6 **–** Examples of tightening sessions

In order to assess the results, they have been collected into four charts delivered in Fig.7, while in Table 2 the main statistical parameters obtained for each group of tightening tests have been summarised. From the diagrams, it is possible to note some significant differences between the two tightening methods individuating some criticisms, especially with respect to the possible application to SFC dampers. First, a preliminary useful observation is that, the two procedures could assure, in all the tightening tests, a value of the pre-load higher than the minimum characteristic value used in design equal to 171.5kN. Therefore, they seemed to perform adequately from this point of view. Despite this, the results obtained with the torque and the combined method were significantly different (Fig.7). These differences, according to the authors, are essentially due to the different goals that the two methods want to achieve. In fact, as previously mentioned, the torque method (even though it has the criticisms previously evidenced) is executed basically under force control and is calibrated to assure that, in the same time, the characteristic value of the preload is higher than the nominal value used in design and that the 95% fractile of the preload is lower than the nominal bolt tensile strength.

Conversely, the combined method is a displacement control procedure, able to guarantee that the nominal value of the bolt preload is attained. However, even though it is very effective in achieving this objective, it does not provide a clear control on the applied upper bound value of the pre-load, which is based only on an empirical relationship between the part-turn rotation of the nut and the thickness of the bolted assembly.

The varied accuracy of the two methods is very clear from the results represented in Fig.7. In fact, for the torque method, under the assumption of a normal distribution, it resulted in the lower and upper bound fractiles of the preloads being always contained in between the bound values represented by the bolt's yield strength and nominal resistance. This happened both, for assemblies employing flat washers, and for those equipped with disc springs. Conversely, for the combined method, it is easy to note from Fig.7 that while, on the one hand, the minimum preload was always achieved, on the other hand, in many cases the obtained pre-load was greater than the nominal bolt tensile strength, with the consequent risk to over-load the bolts.

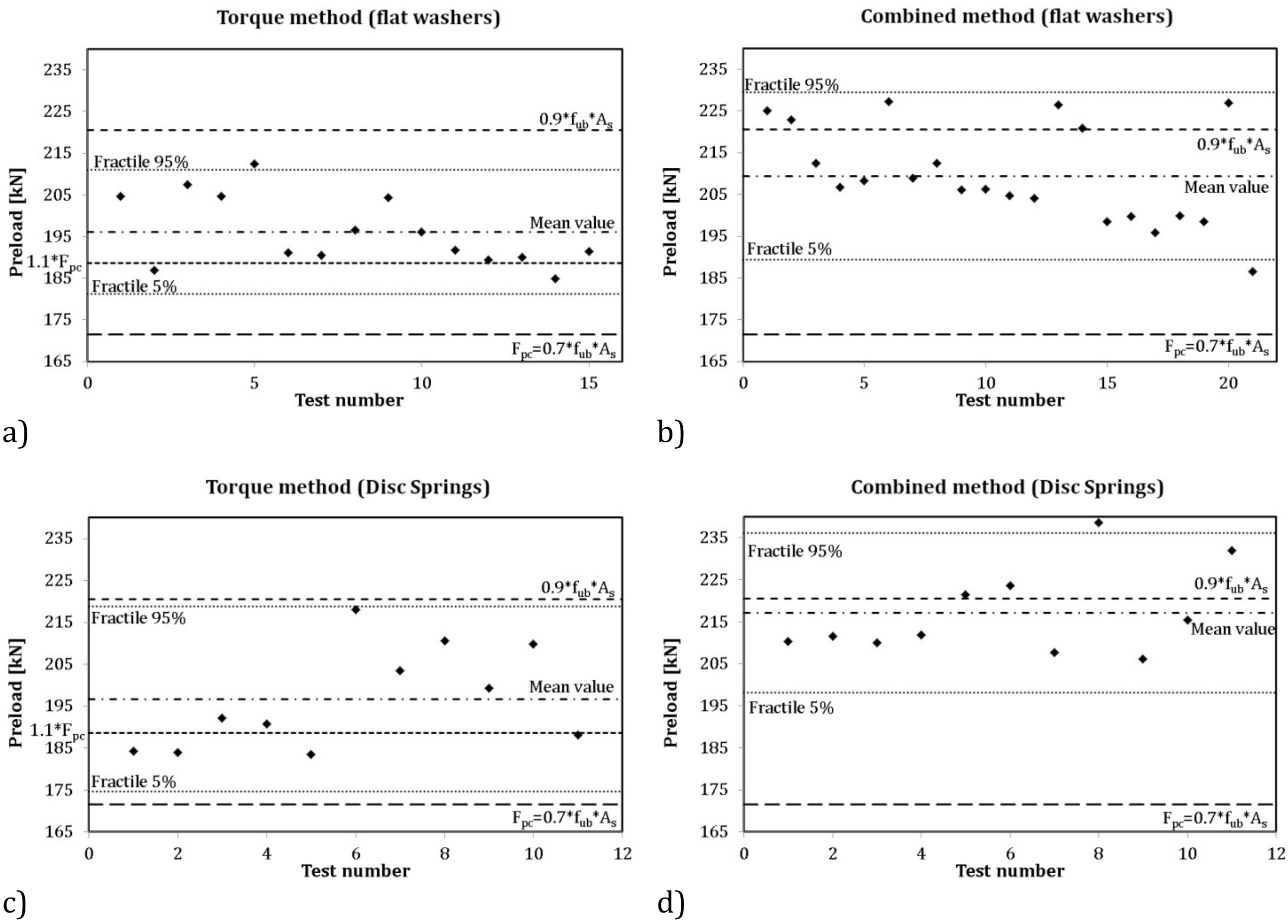

Fig. 7 – Summary of the tightening tests

In terms of statistical parameters, both methods provided low values for the coefficient of variation (lower than 0.06), complying with the minimum requirements of EN 1090-2, but the tightening forces applied with the combined method were characterised by an upward shift of the mean value (very close to the nominal bolt tensile strength) and of the upper and lower bound fractiles. Particularly with the combined method, in both cases (with or without disc springs), the upper bound fractile exceeded the minimum bolt yielding resistance. Additionally, in the case of the torque method, the presence of the disc springs did not seem influent on the response of the assembly, while for the combined method the presence of the disc springs provided a further upward shift of the mean value. This is probably because the procedure suggested by EN1090-2 has been extended straightforwardly to assemblies with disc springs, adopting the same angle of rotation of the nut (90°). Supposedly, with disc springs, due to the difference in the stiffness of the assembly, a recalibration of the part-turn of the nut should be performed. This should be investigated in greater detail in further analyses. Furthermore, this result may be also related to the fact that these disc springs are usually sold without "pre-setting" as, instead, suggested by [50] in application to AFC of SHJs. Additional burdens that seem to characterize the behaviour of the disc springs standardised according to DIN6796 are that they have a limited load capacity (which usually leads to adopt more disc springs in parallel), a low maximum deflection and edges which are not smooth and round, leading to a considerable energy dissipation while squashing them. All these factor, substantially, lead to a behaviour of the disc spring which is not actually elastic.

Table 2. Tightening tests – Summary of statistical results.

		***μ* [kN]**	***σ* [kN]**	***CV***	***Fractile 5% [kN]***	***Fractile 95% [kN]***	**F_p [kN]**	**$1,1F_p$ [kN]**
Flat Washers	*Torque Method*	196.14	8.19	0.04	181.24	211.04	171.50	188.65
	Combined Method	209.44	11.35	0.05	189.41	229.47	171.50	-
Disc Springs	*Torque Method*	196.73	10.53	0.05	172.98	214.25	171.50	188.65
	Combined Method	217.15	10.03	0.05	198.16	236.15	171.50	-

As a conclusion, from the developed analyses, it seems that the torque method provided reliable results, fully complying with the EN 1090-2 requirements, both in case of traditional HV assemblies and in case of HV assemblies with disc springs. Nevertheless, it must be underlined that, as already specified before, a good threads quality is a fundamental feature for the consistent application of this methodology. Conversely, the combined method, even though on the one hand provided a reliable response in terms of achievement of the minimum preload, on the other hand showed to be more empirical, leading, in some cases, to values of the tightening force being very close to or exceeding the bolt's yielding resistance. Additionally, it was evident from the experimental data that the part-turn rotations currently indicated in EN1090-2 should not be extended straightforwardly to assemblies with disc springs. In fact, when disc springs are employed, due to the different stiffness of the assembly, a recalibration of the part-turns should be previously performed.

3.3 Short-term and mid-term relaxation tests

In order to investigate the possible short- and medium-term relaxation effects of bolted assemblies, specific tests have been conducted on specimens, in some cases employing disc springs and HV washers, in other cases only with standard HV washers. The tests were performed similarly to the tightening tests. Therefore, the specimens were tightened and the preload was monitored for a period of time. No tensile force was applied to the specimens. Some tests were performed by tightening according to the torque method and others using the combined method. In 24 cases, the preload was monitored for a period of at least 18 hours (short-term tests) and in 4 cases, the bolts' force was recorded for a minimum of 30 days (mid-term tests). In Table 3, the typology of short-term tests performed are reported. The results, which are not reported here in extensive detail for reasons of brevity, demonstrate that there is no significant correlation between the tightening method and the bolts' loss of tension, while it was possible to observe a very strong difference comparing the behaviour of the assemblies with or without disc springs. In particular, as shown afterwards, when disc springs were included in the assembly, the loss of preload was significantly higher, even though the experimental data were characterised by a lower dispersion.

Table 3. Short term relaxation tests list.

Flat Washers	*Torque method*	6 tightening tests
	Combined Method	6 tightening tests
Disc Springs	*Torque Method*	6 tightening tests
	Combined Method	6 tightening tests

In Table 4, the results of the tests performed on specimens with HV flat washers are summarised. For the sake of simplicity, even though the loss of tension was monitored continuously during the tests, they are reported in this table in correspondence of precise instants, namely at 1h, 6h, 12h and 18h. For each time instant, the losses are reported summarising the results of the twelve tightening tests in terms of statistical parameters. It is evident from Table 4 that in each time instant the coefficient of variation is very high, underlining the strong aleatory nature of the phenomena, while the average loss varies from about 4% to 5.3% at 1h and 18h from the initial tightening, respectively. In the last column of the table, the expected loss at 50 years (reference life-time of the structure) is estimated through extrapolation of the data (according to EN 1090-2), considering the possibility to define, starting from the test data, an equation providing the loss over the time. To this scope, four regression curves are proposed, normalising the loss with respect to the loss of initial bolt tension that occurred at different time instants (1h, 6h, 12h, and 18 h) (Fig.8). These regressions can be used as rapid tools to evaluate the bolts' loss of initial tension over time. Additionally, they can also be interpreted from a statistical standpoint, assuming as a random variable the loss at 1h, 6h, 12h, and 18h, with the mean values and coefficients of variations reported in Table 4. Therefore, the regressions can be expressed in the following way:

$$\frac{p_{\%}\,(t)}{p_{\%,\#h}} = c_1 \ln(t) + c_2 \tag{1}$$

where $p_{\%}\,(t)$ is the loss of preload at time t (in hours), c_1 and c_2 are two constants calibrated on the experimental data by means of least square regression, and $p_{\%,\#h}$ is the loss of preload at time #, with # equal to 1h, 6h, 12h, or 18h. Based on the obtained data, according to the procedure provided by EC0, $p_{\%,\#h}$ is a variable that can be assumed normally distributed with mean value equal to the mean value (reported in the second column of Table 4) and coefficient of variation estimated from the sample (reported in the fourth column of Table 4). In this way, the mean, lower bound and upper bound fractiles of the loss used to normalise the regression can be determined as follows:

$$p_{\%\#h,5\%fractile} = p_{\%\#h,mean}(1 - \gamma \times CV) \tag{2}$$

$$p_{\%\#h,95\%fractile} = p_{\%\#h,mean}(1 + \gamma \times CV) \tag{3}$$

where γ accounts for the narrowness of the sample and can be expressed as:

$$\gamma = \left(1 + \frac{1}{n}\right)^{0.50} t_{\alpha,n} \tag{4}$$

where n is the numerosity of the sample, and $t_{\alpha,n}$ is the quantile of the t-student's distribution with α=0.05.

Table 4. Short term relaxation tests results (Flat washers).

	μ [%]	**σ [%]**	**CV**	**Fractile 5% [%]**	**Fractile 95% [%]**	**Expected loss at 50 years (Regression curve)**			**c₁**	**c₂**
						5%	*Mean*	*95%*		
1h	*4.03%*	*1.01%*	*25.12%*	*2.22%*	*5.83%*	*5.37%*	*9.72%*	*14.07%*	*0.1078*	*1.0132*
6h	*4.79%*	*1.35%*	*28.24%*	*2.38%*	*7.20%*	*4.77%*	*9.60%*	*14.43%*	*0.0894*	*0.842*
12h	*5.09%*	*1.56%*	*30.71%*	*2.30%*	*7.87%*	*4.33%*	*9.55%*	*14.78%*	*0.0838*	*0.7893*

18h	*5.26%*	*1.70%*	*32.36%*	*2.23%*	*8.30%*	*4.03%*	*9.53%*	*15.03%*	*0.0808*	*0.7609*

The results of the regression curves are summarised in Table 4 with the correlation coefficients reported directly in Fig.8. In general, the results of the tests demonstrate that for bolt assemblies with normal washers, the loss occurring in 50 years, on average, is approximately equal to 10%, and that half of the total loss occurring during the life-time of the bolted assemblies occurs in the first 12h.

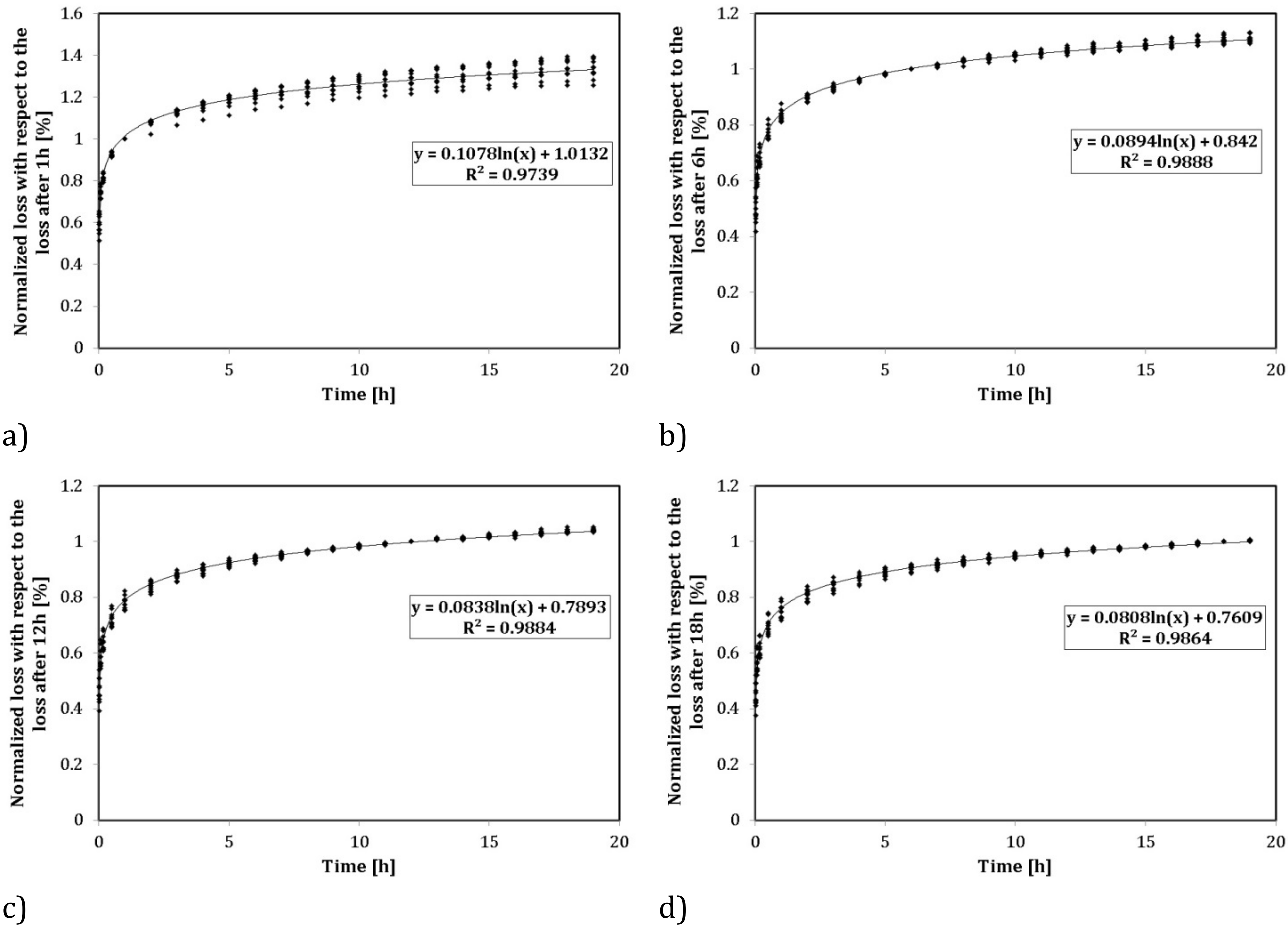

Fig. 8 **–** Regression curves of short term relaxation tests (Flat washers) normalised with respect to the loss occurred in fixed time instants a) 1h; b) 6h; c) 12h; d) 18h.

For the specimens equipped with disc spring, the same number of tests and the same analysis of the data have been carried out. The results are reported in Table 5 and Fig.9. In this case, it is easy to note from Table 5 that in the assemblies with disc springs the observed loss of tension was always greater but, as stated, was also characterised by a lower variability. In fact, the coefficient of variation evaluated at different time instants was, in this case, equal to about 8%. As before, in order to have a fast tool to estimate the bolts' loss of tension over time, regression analyses of the data have been carried out, normalising the regression curves with the loss occurring at 1h, 6h, 12h, and 18h. The loss of tension occurred at different time instants with the mean values and coefficients of variations reported in Table 5. The analysis of the data points out that in assemblies with disc springs, the loss of preload estimated at 50 years is, on average, of about 27% and that half of this loss occurs in the first 12h.

Table 5. Short term relaxation tests results (Disc Springs).

	μ [%]	**σ [%]**	**CV**	**Fractile 5% [%]**	**Fractile 95% [%]**	**Expected loss at 50 years (Regression curve)**			**c_1**	**c_2**
						5%	*Mean*	*95%*		
1h	*10.71%*	*0.91%*	*8.51%*	*9.09%*	*12.34%*	*22.98%*	*27.09%*	*31.19%*	*0.1181*	*0.9943*
6h	*12.98%*	*1.00%*	*7.72%*	*11.20%*	*14.77%*	*23.33%*	*27.05%*	*30.77%*	*0.0973*	*0.82*
12h	*13.76%*	*1.03%*	*7.46%*	*11.93%*	*15.60%*	*23.44%*	*27.04%*	*30.63%*	*0.0917*	*0.7732*
18h	*14.19%*	*1.04%*	*7.32%*	*12.34%*	*16.04%*	*23.52%*	*27.05%*	*30.58%*	*0.089*	*0.75*

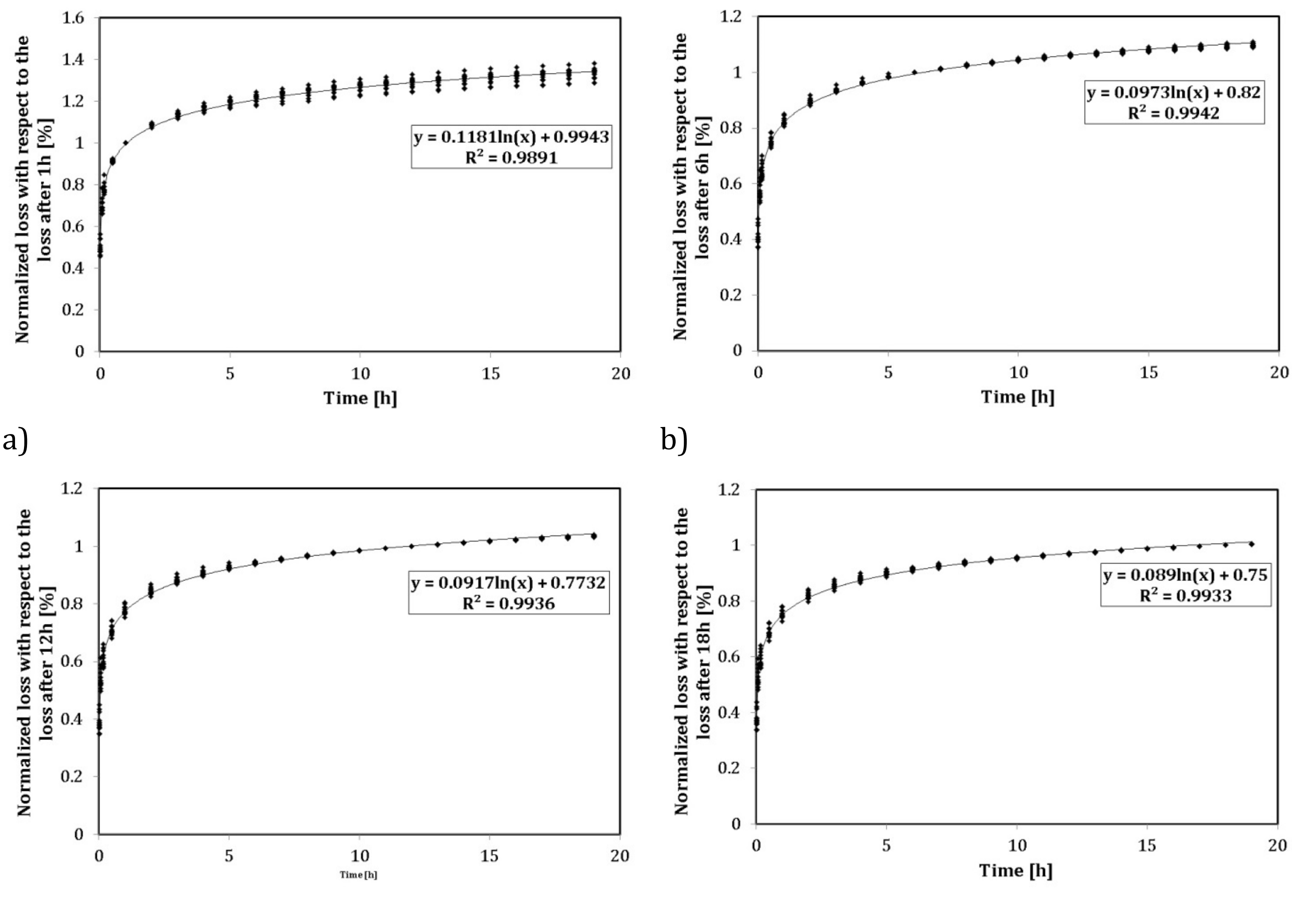

Fig. 9 – Regression curves of short term relaxation tests (Disc Springs) normalised with respect to the loss occurred in fixed time instants a) 1h; b) 6h; c) 12h; d) 18h.

In order to better clarify the role of time-dependent effects on the bolts' pre-load, other four tests have been performed: two on equal specimens with flat washers and two on equal specimens with disc springs. These tests have been extended over a period of 30 days monitoring continuously the pre-load. The results of the tests are shown in Fig.10, where the dashed lines report the results of the tests with flat washers and the continuous lines represent those of the specimens with disc springs. Again, it is immediate to note from Fig.10 that the specimens employing disc springs showed a higher loss of preload since the beginning. This is probably related to the higher deformability of the assembly. In fact, the higher deformability of the assembly seems to provide, at installation, a higher elastic return of the tightening force. This may, obviously, also depend on the installation procedure that has been carried out; in the current case – manually – with a torque wrench. The same results represented in Fig.10, are summarised in Table 6, reporting the losses in different time instants, namely at 1h, 6h, 12h, 18h, 15 days, and 30 days. Additionally, in this table, for every test an estimate of the loss of preload at 50 years is reported, performing several regression analyses using the data acquired from the beginning of the test up to the fixed time instants.

This is to check on the approximation obtained in the estimate of the 50-years loss using the data coming from the short-term tests, with respect to the estimate made using the data coming from the mid-term tests. From the results reported in Table 6, several aspects can be noted. First, as also previously observed, the loss of initial tension with disc springs was evidently higher since the beginning and it did not stabilise as fast as it did for specimens with flat washers. In fact, for specimens with flat washers, the regression curves became practically stable after 12-18h from the tightening, while for specimens employing disc springs, the loss was not completely stabilised even after 30 days. This is evidenced from the extrapolations of the 50-year loss. In fact, for specimens with flat washers the estimate of the loss of initial tension did not change significantly, if considering the data at 6h or 30 days (the regression curve is stabilised already after 6h and does not vary significantly adding data of the next 29days and 18h). Conversely, for specimens with disc springs the estimate of the loss of preload varies significantly considering the data at 18h or the data at 30 days (the regression curve is not stable and tends to stabilise after a longer period of time because the slope of the curve tends to soften over the time).

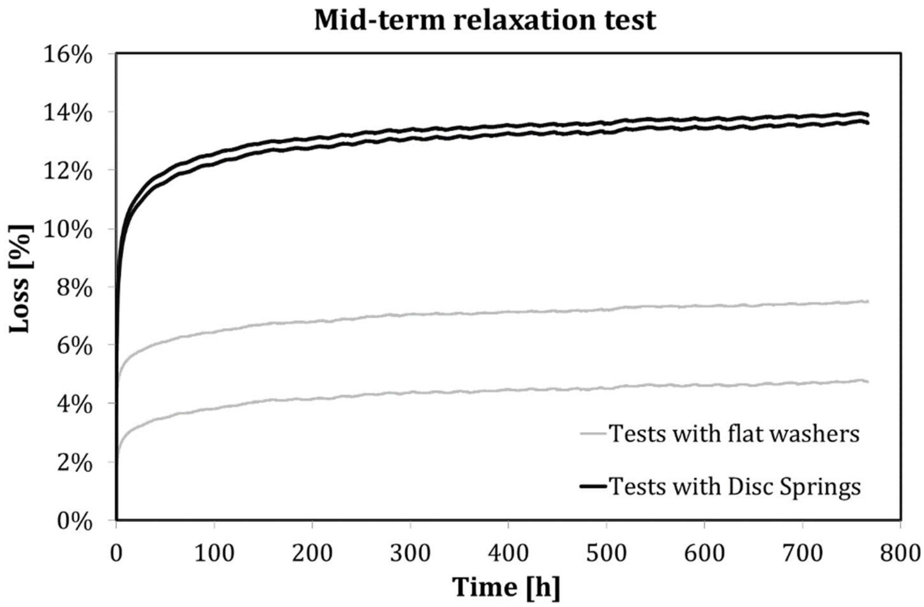

Fig. 10 **–** Medium-term relaxation tests results

Therefore, from the obtained results it seems that short-term tests provide accurate results in estimating the loss of bolts' tension for assemblies with flat washers even after 6 hours and, additionally, the estimate with short-term tests is also slightly on the safe side. Conversely, for assemblies with disc springs, as far as the loss does not stabilise rapidly, it seems that the short-term tests are not accurate, and too conservative to estimate the loss over 50-years. Overall, in all the tests, both with disc springs and with normal washers, about 70% of the loss expected to occur in 50 years ended in 30 days.

Table 6. Medium term relaxation tests results.

			1h	6h	12h	18h	15days	30days
Flat washers	***Test 1***	*Loss of preload after # hours/days*	4.35%	5.21%	5.50%	5.66%	7.06%	7.40%
		50 years' loss estimated using the data obtained up to #hours/days	15.74%	11.68%	11.04%	10.68%	10.36%	10.41%
	Test 2	*Loss of preload after # hours/days*	1.99%	2.71%	2.98%	3.12%	4.37%	4.68%
		50 years loss estimated using the data obtained up to #hours/days	5.00%	6.35%	6.75%	6.74%	7.48%	7.33%
Disc Springs	***Test 3***	*Loss of preload after # hours/days*	6.78%	9.36%	10.19%	10.62%	13.13%	13.54%
		50 years loss estimated using the data obtained up to #hours/days	17.54%	23.37%	23.77%	23.52%	19.42%	18.83%
	Test 4	*Loss of preload after # hours/days*	7.01%	9.66%	10.50%	10.91%	13.43%	13.84%
		50 years loss estimated using the data obtained up to #hours/days	18.03%	24.01%	24.28%	24.03%	20.32%	19.22%

4. CONCLUSIONS

In this paper, an experimental analysis regarding the accuracy of the tightening procedures proposed by EN 1090-2 and the influence of time-related relaxation effect over the pre-load has been carried out. In the experimental analysis, different bolt assemblies have been tested considering also the possibility to employ the standardised type of European disc springs. Based on the developed work the following conclusions can be drawn:

- The torque method as currently codified – even though as already evidenced by Berenbak [42] presents some criticisms and should be improved – seems sufficiently accurate. In fact, with the torque procedure, in all the tightening tests, the bolt target preload was matched obtaining the expected accuracy. The accuracy did not vary significantly with the type of bolt assembly. In fact, analysing the experimental data statistically revealed that the torque method can be applied to both standard HV assemblies and to assemblies employing disc springs not changing the accuracy;
- The combined method has proved to be accurate in achieving the nominal preload. However, despite this, it demonstrated to be more empirical and not accurate enough in limiting the bolt preload below the nominal value of the yield resistance. In fact, following the currently codified procedure, in many cases a preload exceeding the nominal bolt tensile strength was applied. This may be due to an imprecise definition

of the part-turn rotations indicated in the EN 1090-2. Additionally, the tests performed have shown that the part-turn rotations suggested by the EN 1090-2 cannot be straightforwardly extended to assemblies with disc springs. This is because, due to the higher deformability of the assembly, a different part-turn rotation should be applied to the assembly. Therefore, a recalibration of the part-turns should be made in order to extend the combined method to assemblies with disc springs;

- The short-term tests have revealed that the bolts' loss of initial tension is a relevant effect that deserves to be accounted for in design procedures. To this scope, regression curves of the experimental data have been provided. Overall, the short-term experimental tests have shown that with normal assemblies the loss of preload in 18h is equal, on average, to about 5%, while for assemblies with disc springs it reaches an average value of 14% in 18h. The extrapolation of short-term data led to an estimate of the average loss over 50 years of about 10% and 27% for assemblies with normal HV washers and disc springs, respectively. In both cases, it was observed that the loss of preload occurring in 18h is about 50% of the total estimated to occur in 50 years;
- In order to generalise the obtained results, the possibility to employ different configurations of disc springs or customised pre-set disc springs should also be investigated. In fact, it was demonstrated in this work that the European standardised disc springs are not effective in maintaining the initial pre-load because, basically, they do not behave elastically. Additionally, they have, as already stated in the paper, a low load capacity and maximum deflection as well as edges which are not smooth and round, leading to a significant energy dissipation while squashing them. In this regard, further experimental analyses will be undertaken by the authors in the future research activities;
- Overall, from the experimental data, it was observed that both, in case of assemblies with disc springs and normal washers, about 70% of the loss that is expected to occur in 50 years ends in 30 days.

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