Numerical and experimental analysis of a passive multilayer friction damper for seismic energy dissipation

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15 WCEE
LISBOA 2012

SUMMARY:
One of the methods to mitigate the effects induced by the seismic action on structures is friction based energy dissipation.
This paper presents the behavior of a multilayer frictional damper. The device consists of two mobile elements and other three fixed ones. Mobile elements are made of metal sheets that, on both sides, have ecologic friction material plates attached. Energy dissipation occurs at the contact surfaces between fixed elements and ecologic friction material pads. Calibration of device's dissipation capacity is achieved by means of high strength bolts. This paper presents experimental measurements of the steel-friction material couple frictional characteristics and the hysteretic characteristics of the damper. A numerical modeling of nonlinear behavior of the device has also been performed using ANSYS computer program.

Keywords: friction damper, experimental measurements, nonlinear numerical modeling

1. INTRODUCTION

Recent studies, (Vos et al., 2010) (Guha-Sapir et al., 2011), highlight the fact that earthquakes and earthquakes related phenomena (such as tsunamis) still account for a large percent of the overall fatalities and damages resulting from global natural disasters. These underline that the seismic protection of structures remains of crucial importance to civil engineers.

Earthquakes can damage structures or even cause their collapse. Traditional approach to seismic design is been based upon achieving a proper combination between strength and ductility of the structure in order to resist to imposed loads. Thus, the level of the structure security cannot be achieved because of the incapacity of designing method to consider uncertainties related to the earthquake action (Li and Li, 2008).

At the time being, beside the classic ductile design of structures, there are several other approaches of the seismic protection of buildings worldwide. They can be grouped into three broad areas (Soong, T.T. and Spencer Jr., B.F., 2002) as follows: base isolation, passive energy dissipation and active control.

The American standard FEMA 356 (2000) recommends considering the energy dissipation systems in a somewhat broader context than isolation systems. For the taller buildings (where isolation systems may not be feasible), energy dissipation systems should be considered as a design strategy when performance goals include the Damage Control Performance Range.

Due to its proven efficiency, seismic protection of buildings by means of additional devices that enhance the damping capacity of structures is gaining momentum within the engineering community worldwide. Friction dampers are often used because of their high-energy dissipation potential at relatively low cost and their ease to install and maintain (Toyooka et al, 2008).

The beginnings in field of earthquake protection of structures using frictional device are marked by the work of Pall (1979). Between 1985 and 2008 shaking table test performed on frame structures equipped with friction dampers showed the benefits resulting from their use (Filiatrault and Cherry,1986) (Kelly, Aiken and Pall, 1988) (Constantinou et al, 1991) (Aiken et al, 1993) (Mualla et al, 2003) (Amiri, Naghipoor and Jalali, 2008).

More recent a number of new and retrofitted buildings have been seismically enhanced using friction dampers (Soli et al, 2004) (Chandra et al, 2000) (Verganelakis and Pall, 2004) (Vail et al, 2004).

The present paper concerns to development of a new, rather inexpensive, friction damper. Experimental testing, regarding the behavior of device's frictional contacts, computational as well as experimental evaluation of the hysteretic behavior of the device is addressed.

2. GENERAL LAYOUT AND FEATURES OF THE DAMPER

The device was developed as a damping system that is to be inserted in structures at the floor level. It is designed to work in connection with tension only members that transmit the relative level displacement. The device is subject to a patent request (Stefancu, 2011a) filed to the Romanian State Office for Inventions and Trademarks. More details can be found in Stefancu (2011b).

This damper causes a frictional force by using the technology of the disk brake. Frictional energy dissipation occurs along four surfaces, hence leading to a number of advantages among which some could be considered: use of materials with a low maximum allowable working pressure, reduced wear effects, reduced frictional heating effects etc.

The energy induced in the building by seismic actions and transferred to device is mostly converted into thermal energy and the structure's response is decreased.

Composition of this damper along with an actual device is presented in Fig. 1 and Photograph 1.

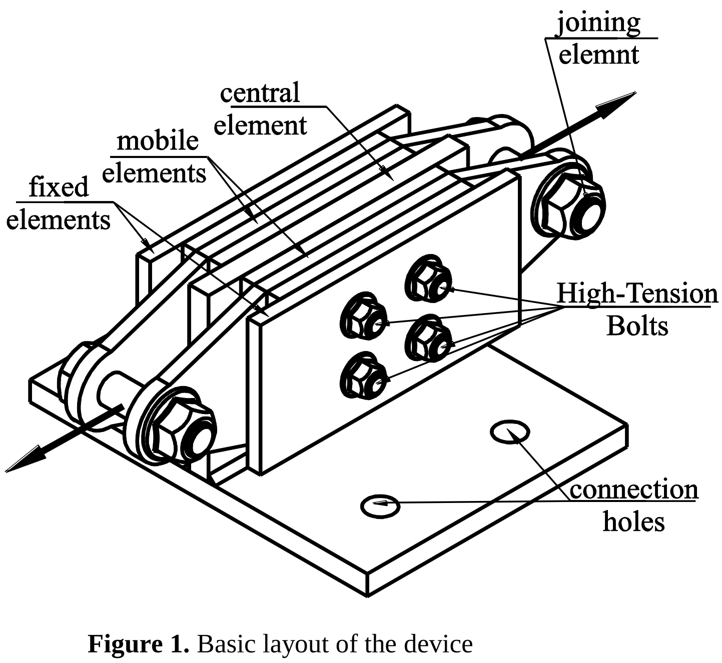


Figure 1. Basic layout of the device

Photo 1. The built device

Frictional energy dissipation results at the contact surface between the mobile elements and the fixed/central element. The pressure on the sliding surfaces is achieved by fastening the brake pad and

Zinc coated steel plate through high tension bolts. The joining elements are used to ensure a synchronic displacement of the mobile elements.

3. MATERIAL TESTING

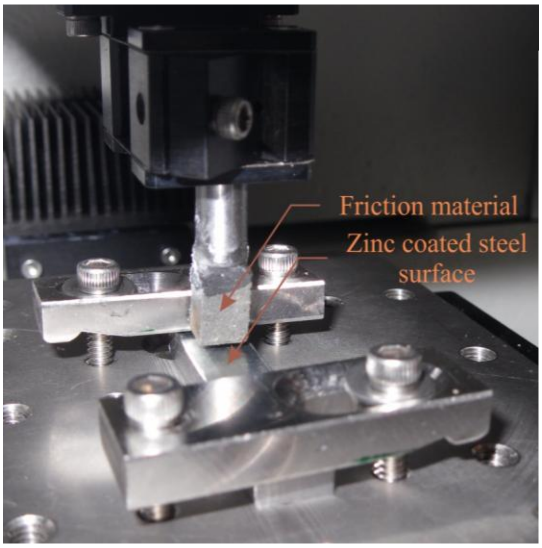
The damping behavior of the device is given by the interaction between the moving parts and the fixed ones. This interaction, in terms of damped energy, is best described by the friction coefficient.

It is known that, given the same contacting materials, the friction coefficient can vary when related to sliding velocity between the surfaces. According to Wen and Huang (2012) the friction coefficient varies with sliding velocity according to the formula:

$$\mu = (a + bU)e^{-cU} + d \tag{1}$$

where *U* is the sliding velocity; *a*, *b*, *c* and *d* are the constants to be determined by the material properties and the loading.

In accordance with the above statements, the experimental measurement sought to highlight the influence of sliding velocity on the value of the friction coefficient. Test speeds ranged between 6 and 30 mm/sec and given the travel distance corresponded to frequencies of 0.2 to 1 Hz. The test were carried out on a pin - flat surface scenario (as presented in Photo 2) and a loading of 10 N which provided a 0.4 MPa contact pressure. The test results came in the shape of friction coefficients and friction force graphs, as the one presented in Fig. 2.



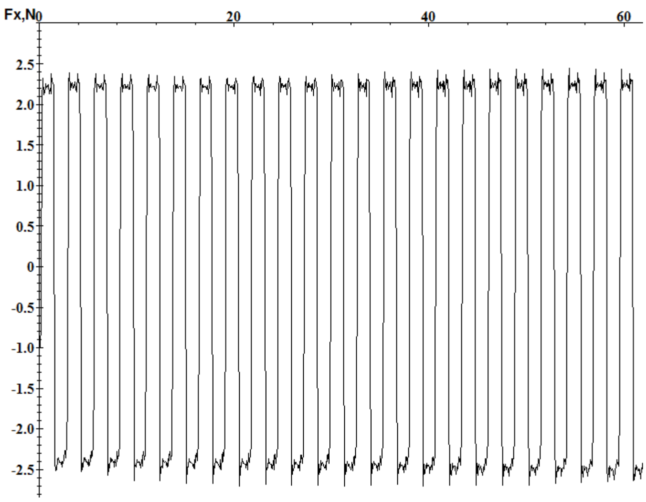


Photo 2. Experimental test setup

Figure 2. Friction force variation for a 12 mm/sec test speed

The average friction coefficients, as resulted from tests, are summarized in Table 1.

Table 1. Average friction coefficients

Velocity [mm/sec]	Friction coefficient
6	0.2516
12	0.2485
18	0.2877
24	0.3182
30	0.3329

The values of the friction coefficient, as resulted from testing (between 0.25 and 0.33) are consistent with the technical specifications as provided by the supplier of the friction material (between 0.2 and 0.45).

Analyzing the data presented in Table 1 one can notice that: the friction coefficient is velocity dependent (increasing as velocity increases) hence the behavior of the passive damper will be most likely velocity dependent.

4. NONLINEAR FINITE ELEMENT ANALYSIS OF THE DEVICE

When the response of a structure is proportional to the loading one terms this as being the case of linear behavior. This type of behavior can accurately describe a limited number of real cases. For the rest of them one expects the so-called nonlinear behavior. This arises from a number of causes (Cook, Malkus and Plesha, 1989), among which one could mention:

- part may come in/out of contact, contact areas may vary with loading, frictional interaction between moving elements may exist;
- displacement and/or rotations may be significant enough to require that the static equilibrium equations to be written for the deformed shape of the structure;
- materials may exhibit non-linear stress-strain relations;
- thermal conductivity may vary with temperature.

The analysis of the device has been performed using ANSYS. The nonlinear characteristic of the analysis is given both by the behavior of the materials: steel and friction pad material (Voiculescu, 2010) and by the frictional contact between moving elements.

In finite element analysis, the presence of two element does not necessary imply the existence of a stiffness relationship between them. To couple the element's stiffness matrices contact elements are required. These elements define the interaction between two or more sets of meshes.

ANSYS typically support four different contact algorithms: augmented Lagrangian, pure penalty, Multipoint constrain, and Lagrange multiplier method. The current analysis resorts to using only the Lagrangian approaches. Both methods relate to solving constrained optimization problems using variational techniques. For a frictionless contact the Lagrange multiplier and the augmented Lagrangian method starting points are respectively presented in eq. 2 and 3 (Crisfield, 1997):

$$L = \phi + \sum \lambda_N g_N \quad (2)$$

$$\phi = \phi_b + \sum_a \frac{1}{2} g^T C g \quad (3)$$

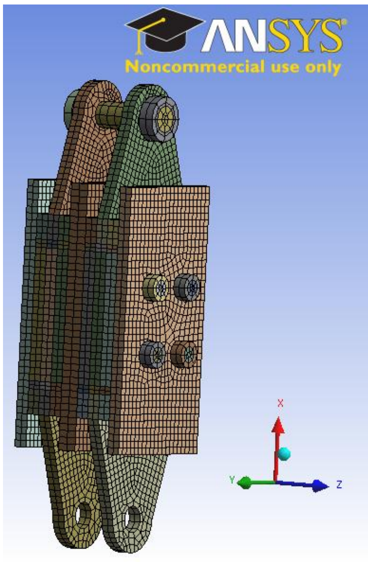
where L is the Lagrangian,is ϕ the total potential energy, λ_N 's are a set of Lagrangian multipliers relating to each relating to each of the contact elements, ϕ_b strain energy in the active contact elements, g contact gaps along normal and tangential direction, C is positive applied force.

More details about these algorithms and the steps that are taken so that they apply to frictional contacts are described by Chaudhary and Bathe (1986) and Simo and Laursen(1992).

The finite element analysis of the device has been performed on a modified damper (Fig. 3) – that would mach real testing conditions. Each bolt is preloaded with a 2 kN force so that the resulting contact pressure magnitude is comparable to the one from the material testing.

Loading frequencies are 0.2, 0.4, 0.6, 0.8 and 1 Hz. A value of 0.45 for the friction coefficient was used (in accordance to the technical specifications as provided by the supplier of the friction material).

A fixed support condition has been applied at the lower part and a displacement at of the joining element has been imposed.



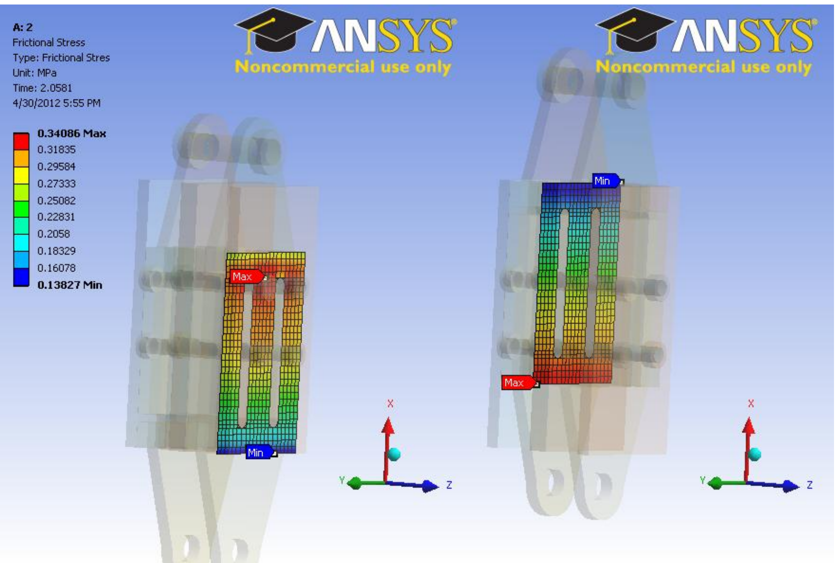


Figure 3. Analyzed device **Figure 4.** Frictional stress distribution at the outer and inner contact surface

The results of the analysis are summarized in Table 2.

Table 2. Results of the analyses

Loading frequency		0.2 Hz	0.4 Hz	0.6 Hz	0.8 Hz	1 Hz
Maximum displacement amplitude [mm]		24.97	25	25	25.016	25.016
Overall Friction force [N]		14218	14247	14248	13660	13659
Frictional Stress outer surface	Minimum [MPa]	0.25718	0.25085	0.25076	0.25583	0.25552
	Maximum [MPa]	0.35383	0.3262	0.33443	0.3573	0.35843
Frictional Stress	Minimum [MPa]	0.28184	0.28159	0.28175	0.21361	0.21341
	Maximum [MPa]	0.35959	0.35789	0.36218	0.36874	0.36931
Normal Stress - X axis	Minimum [MPa]	-157.19	-157.51	-155.92	-211	-201.8
	Maximum [MPa]	160.38	158.83	156.25	162.1	162.14
Normal Stress - Z axis	Minimum [MPa]	-163.26	-160.86	-161.39	-157.08	-156.93
	Maximum [MPa]	165.43	162.69	163.31	163.87	163.98
Contact Pressure outer surface	Minimum [MPa]	0.5715	0.55745	0.55723	0.56851	0.56782
	Maximum [MPa]	0.7863	0.72489	0.74318	0.79399	0.7965
Contact Pressure inner surface	Minimum [MPa]	0.62817	0.63681	0.63693	0.47468	0.47425
	Maximum [MPa]	0.79909	0.80519	0.80484	0.84545	0.84086
Shear Stress XY	Minimum [MPa]	-57.762	-57.038	-57.448	-123.46	-122.43
	Maximum [MPa]	57.295	57.239	57.671	106.89	110.35

Caution must be taken when using the result of the analysis because, the maximum value are influence by:

- induced high accelerations when changing the displacement direction that lead to false peak stresses;
- stress concentration due to contact type – Hertzian contact;
- algorithm used for the contact between the joining element and the moving parts. For first three frequencies an augmented Lagrange with 0.01 contact stiffness has been used while for the last to a normal Lagrange has been utilized.

The force - displacement relations, as resulting from the analysis are presented in Fig. 5.

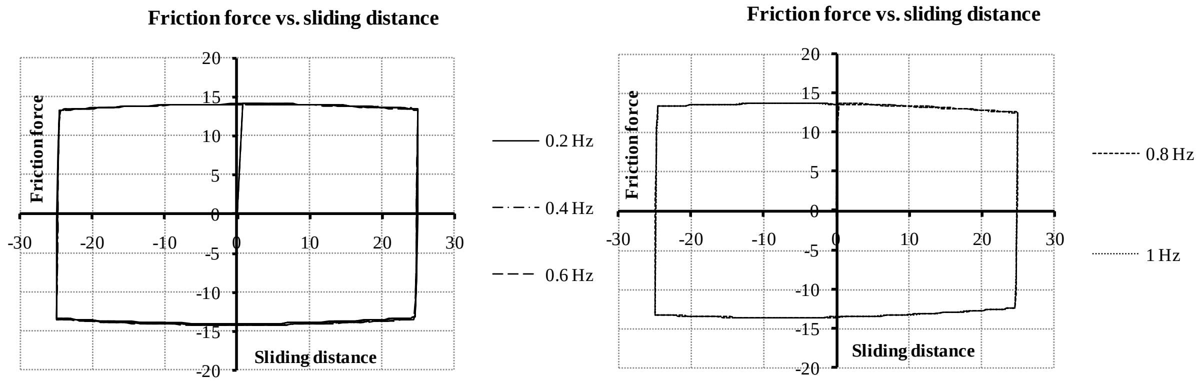


Figure 5. Force displacement relation – as resulted from the analysis

The difference between the two hysteretic curves is due to the contact algorithm used, that lead to greater friction force.

5. EXPERIMENTAL EVALUATION OF DEVICE’S DAMPING CAPACITY

A model of the device has been subjected to cyclic loading. The test setup as well as a detail view of the device is presented in Photo 3.

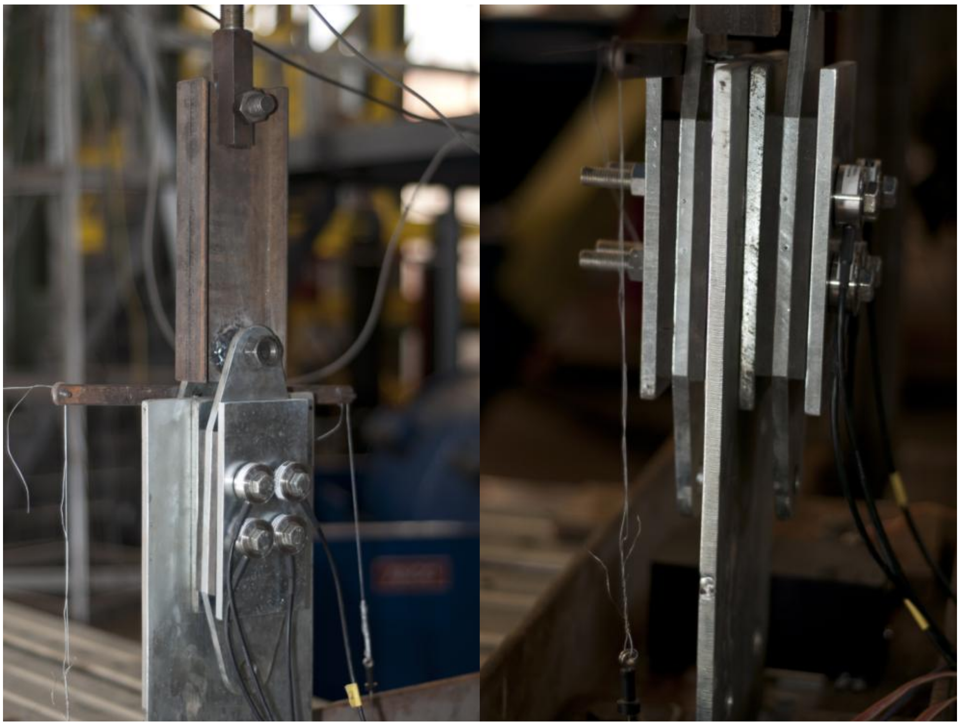


Photo 3. Test setup

Due to testing restriction, a modified device has been used. The damper is similar to the one used for the nonlinear finite element analysis. Displacement transducers were used to monitor mobile element’s movement. Load washer ensured that the initially applied pretension force was 2 kN. A force cell, placed at the upper placed on the moving piston, measured the overall force that would produce the movement of the mobile elements – the friction force. The hysteretic curves as resulted from test are presented in Fig. 6.

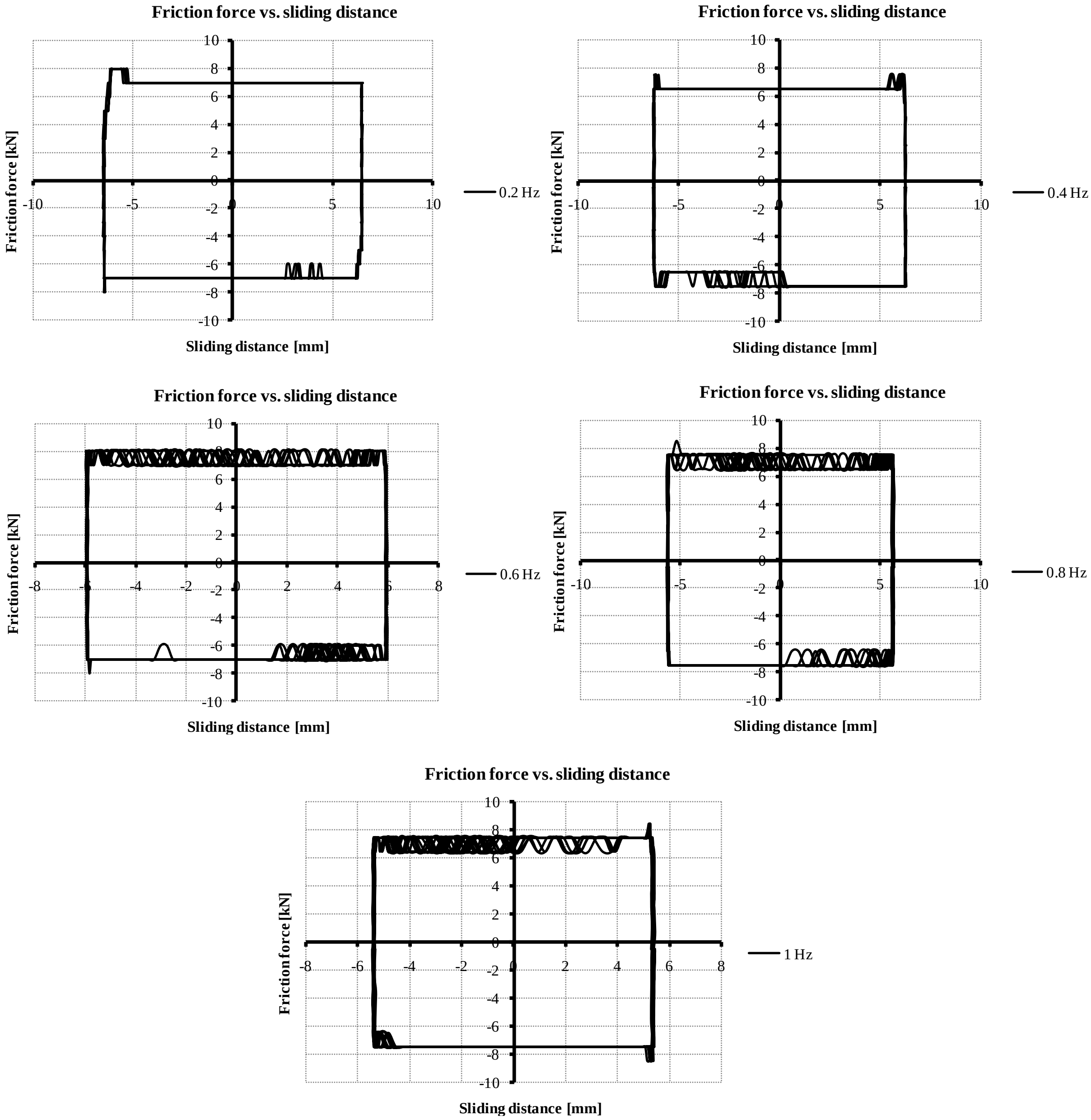


Figure 6. Force displacement relation – as resulted from tests

6. CONCLUDING REMARKS

The experimental tests highlight that the friction force does vary with speed. Its average value is as follows: 6.6 kN for 0.2 Hz, 8.2 kN for 0.4 Hz, 9.3 kN for 0.6 Hz, 8.7 kN for 0.8 Hz and finally 8.2 kN for 1 Hz. The difference between the tests performed on the materials and the ones performed on the device is that the friction coefficient does not follow the same ascending path.

This may be caused by wear or other friction related phenomenon. Further test will be performed to monitor the variation of normal force during the device's operation.

In the global context of earthquake protection of buildings, the proposed device, that is both cheap to manufacture and maintain, can be a reliable alternative to the classic approach of seismic design of structures.

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