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Investigation of the Effects of Axial Force, Velocity and Cyclic Wear on the Characteristics of the Insulation Units in Friction Type Isolation Units with Curved Surfaces: An Experimental Study

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Abstract
The variation in friction coefficient encountered during a seismic event has an impact on the hysteretic behavior of Double Curved Surface Slider isolators. This variability is affected by axial load, sliding velocity, temperature rise at the sliding surface as well as the cyclic degradation. In this study, the variation of the friction coefficient is investigated both experimentally and numerically. Various full-scale static and dynamic tests of Double Curved Surface Slider were conducted at the TREES Lab at EUCENTRE (Pavia, Italy) which is particularly designed in accordance with the most innovative technologies due to its high-performance equipment, enabling conducting both dynamic and static experimental studies on full scale prototypes, and consequently minimizing the correlation and interpretation uncertainties with real structural conditions. Based on the test results three closed formed equations have been statistically derived which enable the calculation of coefficient of friction for each cycle considering velocity, axial force and temperature variations.
Keywords*:* Seismic isolators, coefficient offriction, velocity, axial load, thermal effects.

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I. Introduction

Although earthquakes are natural events that cannot be avoided, the damage incurred by them can be reduced. For many years, researchers have conducted extensive investigations in order to overcome the damaging effects of the earthquakes on the structures. Two basic methods have emerged to mitigate the effects of earthquakes, first method can be classified as "traditional technique" in which the earthquake effect is mitigated by increasing either the strength and stiffness or ductility of the structure. The other alternative method classified as "modern technique" aims to lower the earthquake demands on the structures through incorporation of energy dissipation devices such as damper and seismic isolation [1]. In a seismic isolation system, the superstructure is separated from the substructure using a relatively less stiff medium, which enables the superstructure to be decoupled from the motion of its base. The fundamental idea underlying base isolation is that the structures fundamental period at the isolation level is lengthened and the damping is increased which reduces the forces exerted to the superstructure though increased displacement at the isolation level [2, 3].

Over the century, seismic isolation has gradually become a widespread approach to prevent structures against earthquake damage. The idea of seismic isolation was originally reported in 1881 by a Japanese scientist, Kozo Kawai. However, present-day base isolation has developed in New Zealand starting from the 1960s [4, 5]. More recently, new seismic codes have been published, such as American [ASCE/SEI 7-10, 2010;

AASHTO, 2010], European [EN 15129, 2009], and Italian [NTC2008] that resulted in a larger number of applications, particularly for public and vital structures as well as for the retrofitting of existing structures.

There have been several articles published [3, 6, 7] that discuss different types of base isolation techniques and their predicted responses, such as roller bearings, rubber bearings, and frictional pendulum systems. Rubber-based isolation systems have the advantages of being simple to manufacture and model, having a high application frequency, and being one of the most cost-effective systems which provide simultaneous isolation and energy. However, they have drawbacks such as the necessity for extra damping depending on the material used, changing in mechanical characteristics over time, and the difficulty of identifying whether the lead core within the bearing is harmed following significant earthquakes. Similarly, the friction pendulum system offers most of thesebenefits with the ease of post-earthquake safety checks [8, 9].

Curve Surface Slider (CSS), also referred to as Friction Pendulum System (FPS) [10] has a number of advantages over other seismic isolation devices. Those advantages include the ability to provide an isolation period regardless of the mass of the structure, restoring capacity which is provided by the sliding surface itself as a result of the pendulum's working principle, high energy dissipation capacity that is governed by the tribological properties of the sliding materials, as well as their life span, durability characteristics, ease of construction and repeatability. [6, 10].

The FPS has been the subject of numerous experimental studies, as well as the development of many analytical models [11-13]. De Domenico et al., [14] carried out an experimental study to investigate the effects of frictional heating occurring on double-curved surface sliders on hysteretic behavior. Almazan and De la Llera [13] provided a mathematically formal explanation of the dynamic response of structures isolated by (FPS) which demonstrated that a mathematically "exact" model may be developed to account for significant deformation in isolators. According to experimental data, the friction coefficient is based on the load applied vertically, velocity, and temperature. These effects have been widely characterized and incorporated into consolidated boundary models and have been shown to influence the behavior of sliding systems during seismic shaking, temperature at the pendulum sliding surface is one of the most important parameters influencing the frictional response of FPS devices [15-19].

Experimental studies [14, 16] demonstrated that the heating effect causes the friction coefficient to drop as the device is subjected to several cycles, and this phenomena can be quite relevant when reversed cyclic motion occurs, yet it is essentially negligible at low velocities. Another experimental study conducted by Furinghetti et al. [12] on the double-curved surface isolation exposed to bidirectional and radial sliding movements, showed that the sliding speed determines the decay of the friction coefficient, but the vertical load has a minimal influence that may be ignored at higher load levels (e.g. for the materials in the research, at levels generating contact pressures greater than 30 MPa, however the effect is significantly dependent on the chemical composition of the sliding surface). Furthermore, in the literature, there are few experimental studies about the FPS and they mostly focus on the single friction pendulum system. Therefore, additional experimental studies are required for proper understanding of the characteristics that affect the friction coefficient in double curved surface sliders (DCSS).

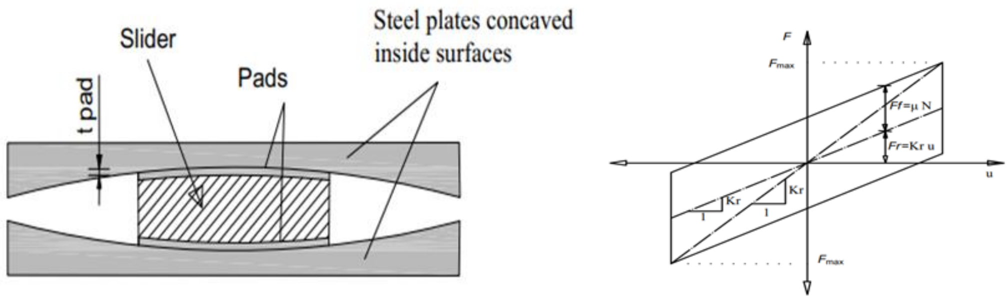
II. Hysteretic Behavior Of Friction Isolators

Fundamental equations of determining the dynamic and the hysteretic behavior of frictional sliding bearings, which would be important in the sequel of this paper are summarized here. The DCSS device, as shown in Fig. 1a, consists of two steel plates with concave inside surfaces that are covered by a stainless-steel sheet with minimum (which generally has a thickness of 2.5mm). A slider (usually made of steel) with exterior convex surfaces, which is equipped with pads of a particular sliding material at the convex surfaces, is placed between these two steel plates. Generally, two primary parameters affect the device's mechanical behavior: the tribological characteristics of these sliding components at the interface and the radius of the curvature R of the two opposing sets of curved surfaces (one convex and the other concave) [14].

The basic model for understanding DCSS's mechanical behavior is considers a constant vertical load and a coefficient of sliding friction (simplified theory of Coulomb). The DCSS combines the re-centering action caused by the curvature of the sliding surface with the energy dissipation caused by friction in a single device. As a result, when the restoring and frictional forces are added together, the total horizontal force provides the rigid-plastic hysteretic behavior [20] presented in Fig. 1b.

$$F = F_r + F_f = \frac{N}{R_{eff}\ cos\o} u + \mu N\ \mathrm{sgn}(\dot{u}). \quad \textbf{(1)}$$

where Fr represents the re-centering force, sgn(·) is the signum function, thus the sign of $F_f = \pm\mu N$ (the frictional force) depends on $\dot{u}$ (velocity), μ is the coefficient of sliding friction, u indicates displacement magnitude, N represents the load acting vertically on CSS, Ø is the rotation angle, and R_{eff} is the pendulum effective radius.



(a) (b)

Fig. 1.(a) Double curved slider; (b) force-displacement (hysteresis) diagram [14]

As aforementioned, the friction coefficient is affected by temperature rise at the sliding surface, velocity, vertical load, as well as environmental and cyclic degradation effects. Based on the dynamic tests conducted it has been determined that among these factors, the temperature increase at the sliding surfaces is the main and most significant source of fluctuations in the friction coefficient.

An experimental study carried out by De Domenico, D., et al. [14] confirms that the rise in temperature ΔT and the associated cyclic effects can influence the entire hysteretic behavior of friction isolators. Moreover, numerical investigations on the cyclic response of FPS devices accounting for the rise in temperature at the sliding surfaces and its impact on the coefficient of frictionare presented in [18, 21, 22]. In such studies, the effect of the displacement path on the frictional heat flux created at the sliding surfaces was studied, and the pertinent impact on the force-displacement and damping properties of the isolation unit was described. Several bidirectional orbits, such as elliptical, circular, and "8-shaped" orbits were compared with unidirectional motion. Eventually, it is anticipated that friction decay will have an impact on the overall response of a structural system separated by FPS devices by causing a reduction in reaction forces at the supports but an increase in the maximum displacement. In addition to the self-heating temperature rise at the sliding surfaces, ambient temperature may affect the friction coefficient as well. Dolce, M., et al. [19], emphasize that when the air temperature rises, the sliding friction coefficient reduces, and the rate of decrease is faster when low-to-medium temperature transformations are taken into account rather than moderate to-high temperatures.

Velocity is another factor that affects the coefficient of friction. Studies [23, 24] show that as velocity increases, the sliding value of the friction coefficient rises until it reaches a constant value (fig 2). The value of the constant velocity is given between 10 cm/s and 20 cm/s [23]. Moreover, Fujita, r., et al., [25] have concluded that there might be considerable variation in the friction coefficient at low velocity (less than 50 cm/s or so) however, the velocity effect is relatively small at high velocities and it could be ignored for calculation of maximum displacement demands at strong earthquake motions.

Numerous studies have been published in the literature regarding how the applied load affects the coefficient of friction [19, 26, 27]. Dolce et al., [19] investigated the vertical load effects on steel PTFE interfaces and indicated that as the pressure increases, the sliding friction coefficient decreases. The decreasing rate, however, depends on both air temperature and velocity, the rate increases when the air temperature drops and the velocity rises.

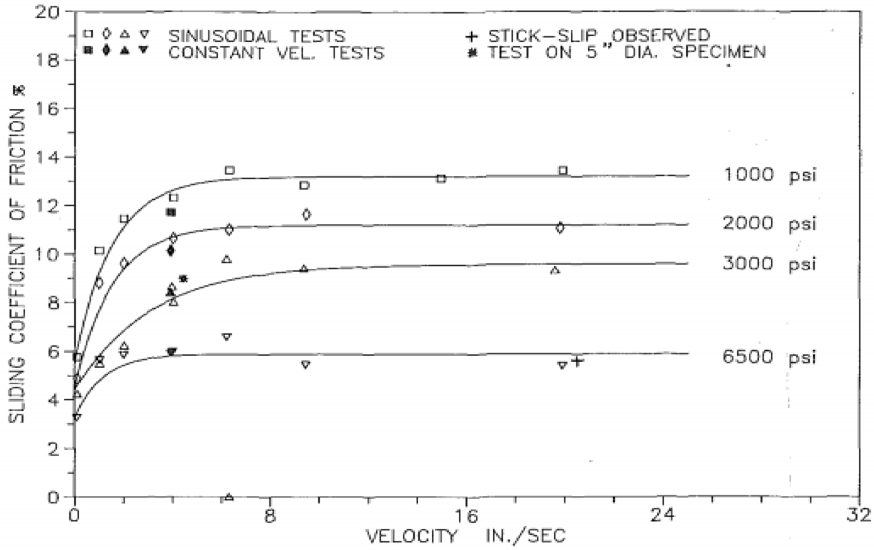


Fig. 2. Sliding friction coefficient variation with velocity of Glass-Filled Teflon at 25% for sliding parallel to Lay [23].

III. Purpose Of Study

Since current design methodologies and commercial analysis software cannot simultaneously incorporate the effects of velocity, temperature rise at the sliding surface, axial load, and environmental effects on the friction coefficient, the current practice requires a boundary analysis with lambda coefficient which include the maximum variation is required. This, oversimplification gives rise to a remote approach and unconservative results. Within the scope of this study, experimental dynamic studies have been carried out to develop a friction model that includes the effects of all parameters are considered which can be included in commercial software, eliminating the execution of boundary analyses and conveys calculation of more accurate displacement and force demands on the substructure and on the isolation layer.

IV. Experimental Setup

Full-scale prototypes of double curved surface sliders were tested at the TREES Lab at EUCENTRE (Pavia, Italy). In order to minimize the correlation and interpretation uncertainties with real structural conditions and enable both dynamic and static experimental study on full scale prototypes, the EUCENTRE experimental facility has been particularly constructed in accordance with the most modern technologies.

The TREES Lab features four experimental facilities: the Strong Floor-Reaction Wall System for full-scale pseudo-static and pseudo-dynamic tests; the high performance uniaxial Shaking Table; the bi-axial Bearing Tester System for testing of full-scale bearing and isolation devices, with high dynamic and force capabilities; the mobile unit, with the most advanced tools for in situ non-destructive tests and for fast seismic vulnerability analyses.

The static and dynamic experimental evaluations of isolation and dissipation devices are performed with the Bearing Tester Machine. Under various static axial loads of up to 40.000 kN, the base table (1.7 m x 4.3 m) enables the vertical, longitudinal, roll, pitch, and yaw degrees of freedom. The BTS Controller is a digital real-time controller that delivers PID closed loop control with a delta-p feedback signal. It is made up of a MTS console assembly, cabling, and control software. The performance characteristics of bearing testers are given in Table 1.

Table 1 Bearing Tester Performance Characteristics.

Base Table	1.7 m x 4.3 m
DOF	Longitudinal, Vertical, Roll, Pitch, Yaw
Peak Displacement	Horiz: ±500 mm, Vert.: ±70 mm
Peak Velocity	Horiz: ±2.200 mm/s, Vert.:±250 mm/sec (stroke dependent)

Peak Force	Horiz: 2.000 kN, Vert: 40.000 kN
Peak Overturning Moment	20.000 kNm
Working Frequency range	0÷20 Hz
Max Flow Rate	27.000 lit/min

The hydraulic power supply is composed of eight hydraulic power supply units, each capable of supplying 170 lpm at 280 bar working pressure for a total flow of 1360 lpm, and 5 power storage banks, each with two 45-liter piston accumulators and six 30-liter gas bottles for a total piston accumulator volume of 450 liters and a total bottle volume of 900 liters.

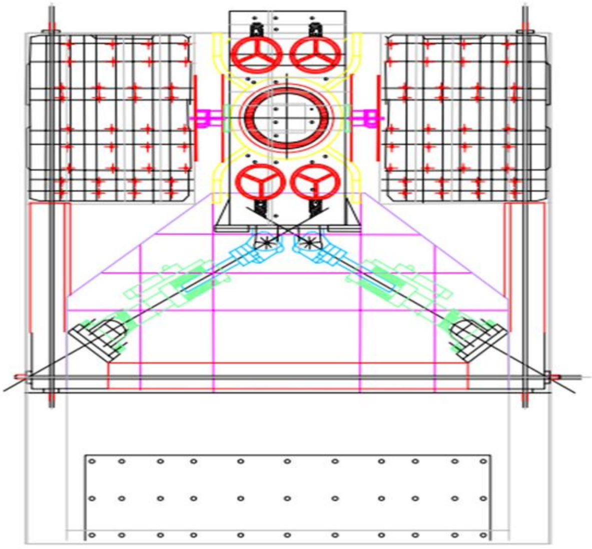


Fig. 3. The bearing tester (plan scheme).

Proposed Equations and Discussuon of Results
In this chapter, experimental results on friction type isolators are examined. Experiments on isolators were carried out for different velocity and pressure values. Results are available for each cycle. The total number of results obtained is 1774.

Three equations to obtain friction coefficient for each cycle are proposed.Linear least squares method with Bi-square method by MATLAB software is used to obtain equations from experimental data. General function of the method, which is a 5th degree polynomial equation with two unknowns, is given Eq. (2):

$$F(P,V)_n = a_{00} + a_{10}P + a_{01}V + a_{20}P^2 + a_{11}PV + a_{02}V^2 + a_{30}P^3 + a_{21}P^2V + a_{12}PV^2 + a_{03}V^3 + a_{40}P^4 + a_{31}P^3V + a_{22}P^2V^2 + a_{13}PV^3 + a_{04}V^4 + a_{50}P^5 + a_{41}P^4V + a_{32}P^3V^2 + a_{23}P^2V^3 + a_{14}PV^4 + a_{05}V^5 . \quad \textbf{(2)}$$

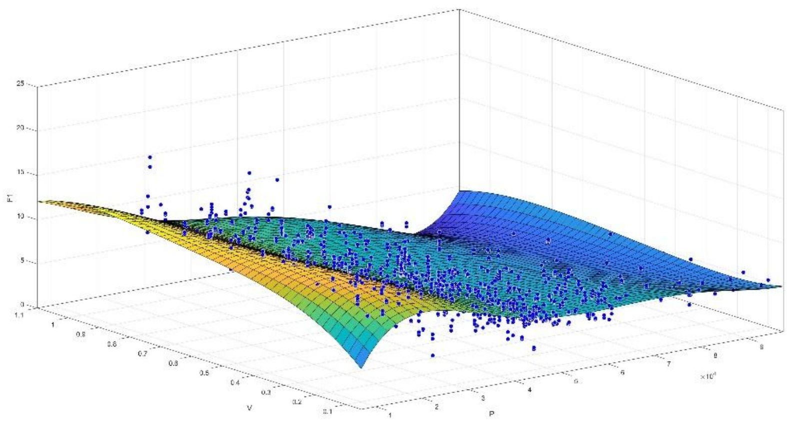
where P is the vertical load, V is velocity and $a_{mn}$is a constant that changes for each loop. Constant values are tabulated in Table 2.

Table 2 Constant Values for Proposed Equations on Each Cycle.

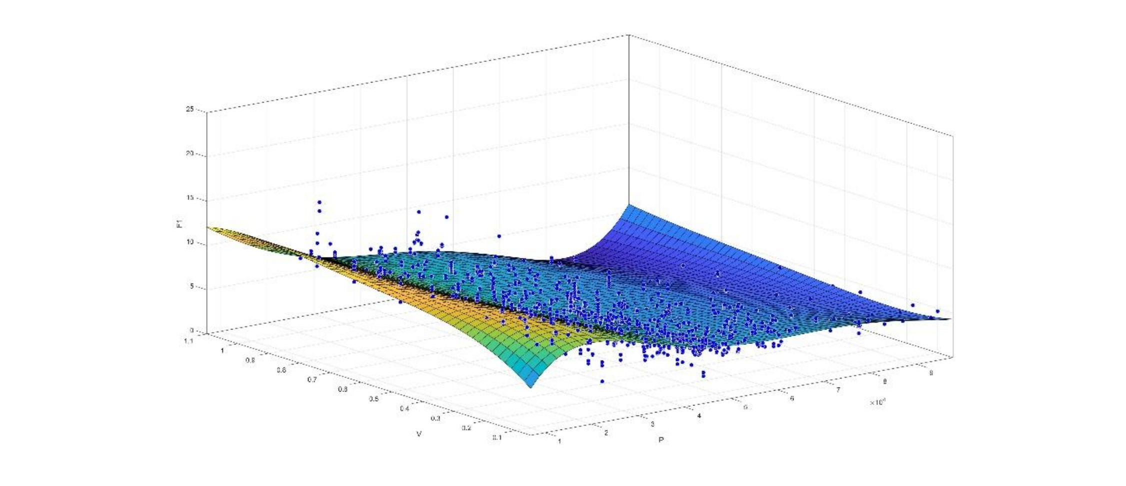
Constant	Cycle 1	Cycle 2	Cycle 3
a_{00}	-7.562500	-4.639000	-1.903900
a_{10}	0.002038	0.0016425	0.0013235
a_{01}	59.02000	54.82000	45.77500
a_{20}	-8.9E-08	-7.12E-08	-5.81E-08
a_{11}	-0.003287	-0.003339	-0.002972

Constant	Cycle 1	Cycle 2	Cycle 3
a_{02}	-83.30000	-88.58000	-70.92150
a_{30}	1.687E-12	1.323E-12	1.076E-12
a_{21}	7.882E-08	7.941E-08	7.171E-08
a_{12}	0.0022865	0.0027342	0.0023599
a_{03}	55.010000	79.005000	60.365000
a_{40}	-1.46E-17	-1.11E-17	-8.96E-18
a_{31}	-9.26E-13	-9.09E-13	-8.27E-13
a_{22}	-1.73E-08	-2.3E-08	-1.98E-08
a_{13}	-0.001341	-0.001762	-0.001509
a_{04}	7.500000	-20.400000	-11.295000
a_{50}	4.686E-23	3.485E-23	2.758E-23
a_{41}	3.951E-18	3.82E-18	3.488E-18
a_{32}	7.607E-14	8.832E-14	7.896E-14
a_{23}	1.01E-09	4.304E-09	3.232E-09
a_{14}	0.0004403	0.0005135	0.0004528
a_{05}	-17.565000	-5.835000	-7.400000

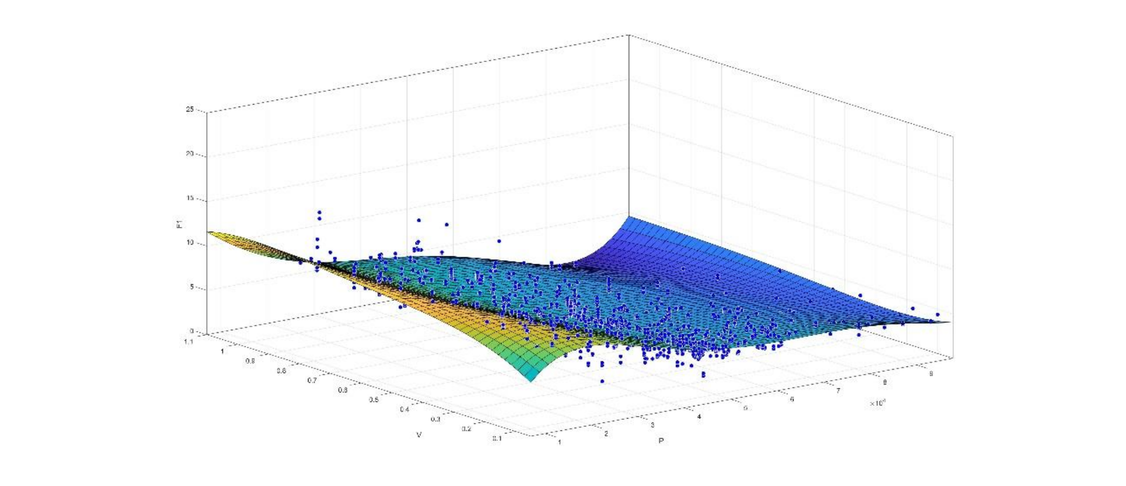
The graphs of the polynomial functions are given in Fig. 4. It is observed that the friction coefficient decays in each cycle due to factors such as heat, environmental effects and cyclic degradation.



(a) Cycle 1



(b) Cycle 2



(c) Cycle 3

Fig. 4. Function Graphs

When the proposed equations are tested with experimental data, it has been observed the rate of error is different for each cycle. The maximum error rates calculated for each equation are given in Table 3.

Table 3 Error rates for equations.

Cycle 1	Cycle 2	Cycle 3
%39.99	%39.11	%38.71

V. Conclusion And Further Studies

In this study, equations have been proposed to get friction coefficient for friction type isolators. Axial load and velocity are used as inputs in the equations. It is thought that the use of the suggested equations for the friction coefficient will help to obtain more accurate analysis results and reduce the analysis time, due to the exclusion of boundary analysis requirement. Following this study, a new isolator model will be programmed for the OPENSEES software.

Statements and Declarations

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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