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A simple tool for forecasting the mechanical response of anchored wire mesh panels

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Abstract. Anchored wire meshes are applied worldwide as protective structures against rockfall. The mechanical performance of a wire mesh is evaluated through laboratory tests, but these are generally not representative of the field conditions. This paper presents a simple tool to predict the force-displacement response of an anchored mesh panel to an out-of-plane load, extending the experimental standard punch test characteristic values (ISO 17745, ISO 17746, UNI 11437) to field conditions. Discrete element simulations are used to provide analytical relations that account for the effect of the different problem's variables on the ultimate resistance and maximum deflection of a mesh panel. The numerical results are subsequently used to define a master curve permitting the force-displacement response of a generic anchored mesh panel to be forecasted. Finally, the master curve is validated against ex-post simulations, and practical implications are discussed.

1. Introduction

The use of secured drapery systems has experienced a large growth in the last decades. Despite having been applied worldwide, strong unknowns about their field mechanical behavior still exist. To date, the mechanical behavior of a wire mesh is characterized by using laboratory procedures that are poorly representative of the in-field conditions, and therefore their results cannot be directly used in the design phase [1]. This highlights the need of a procedure that can provide a more realistic characterization of the field response of anchored wire meshes. The realization of large-scale field tests, even if they may be very informative, is difficult because of both technical and financial limitations. In this perspective, the recourse to a numerical approach represents a valid alternative. In the recent past, the discrete element method (DEM) has been efficiently used to model wire meshes from laboratory conditions [2, 3, 4, 5, 6] to large scale applications [7, 8, 9, 10, 11, 12, 1, 13].

In this work, discrete element simulations are used to analyze the force-displacement response of a mesh panel subject to a punching load. Firstly, the role of the specimen dimension and of the punching element size on the result of the laboratory punch test procedure is quantified (Sec. 3). Secondly, based on the results of a large set of simulations, a simple approach that allows the mechanical response of a generic anchored mesh panel to be forecasted is proposed. This will permit one to predict the entire force-displacement response of a mesh panel starting from the results of the standard laboratory punch test.

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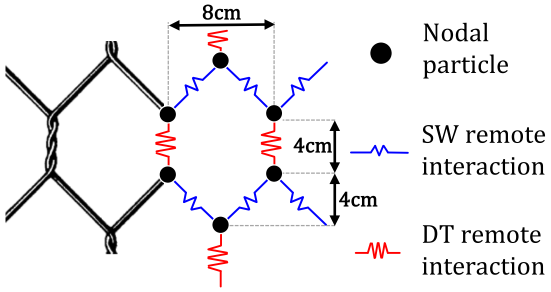

Figure 1. NWB description of the double-twisted wire mesh.

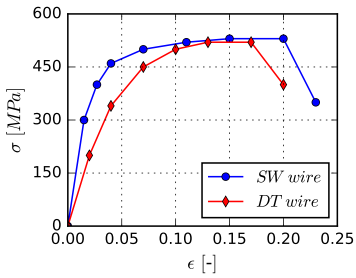

Figure 2. Tensile stress–strain relations of the single (SW) and double-twisted (DT) wires.

The standard hexagonal double-twisted wire mesh of nominal diameter 2.7mm is adopted as reference. The geometrical parameters that define the periodic cell of the mesh are reported in Fig 1. The numerical simulations presented in this paper are performed with the open-source code YADE [14].

2. Methodology

In this work, a node-wire based approach (NWB) is adopted for the numerical description of the wire mesh. This choice is made in order to reduce the computational cost with respect to a more refined cylinder-wire based description of the mesh; in the problem here considered (i.e. punch test), this simplification does not introduce significant differences in the mechanical response of the mesh [15, 16]. The mesh is therefore described as a set of spherical particles that are connected by long-range tensile interactions as shown in Fig 1. These interactions are defined by a piece-wise linear force-displacement function, which is computed from input stress-strain tensile relations [3]. The stress-strain curves used in the numerical model are reported in Fig. 2. The validation of the mesh model was presented in [1]. The mesh-punch contact parameters adopted in the simulations are: contact normal stiffness $k_n = 6.5 \times 10^5$ N/m, tangential contact stiffness $k_t = 0.4k_n$ and a contact friction coefficient $\mu_c = 0$. Details of the numerical model can be found in [1]. Finally, the time step is set equal to dt$= 3 \times 10^{-5}s$.

3. Laboratory punch test

The standard punch test configuration (ISO 17745, ISO 17746, UNI 11437) is worldwide adopted by manufacturers in order to provide characteristic values (i.e. maximum punching force and panel's deflection at failure) to quantify the mesh mechanical performance. Nevertheless, other configurations adopting different dimensions of the punching element and/or of the mesh specimen, are used in practice; therefore, comparison of the data reported in the technical literature may be difficult. In the perspective of providing simple analytical relations permitting one to extend the experimental results to a "common" configuration, a parametric analysis is performed. The panel's side length $L^l = 3.0$m and the punch dimension $D^l = 1.0$m required by the ISO standards, as well as the thus obtained characteristic values ($F_l^* = 73.3$kN and $\delta_l^* = 0.60$m), are adopted as reference to normalize the numerical results.

The effect of the punching element dimension is investigated performing a set of simulations in which the punching element dimension D is varied ranging from 0.1m to 2.5m, with an incremental step of 0.1m (the limit value $D = 0.1$m is related to the mesh opening size). Furthermore, the effect of the mesh specimen dimension is investigated performing a set of simulations in which the panel dimension L is varied in the range 2.0m-4.5m (only square panel

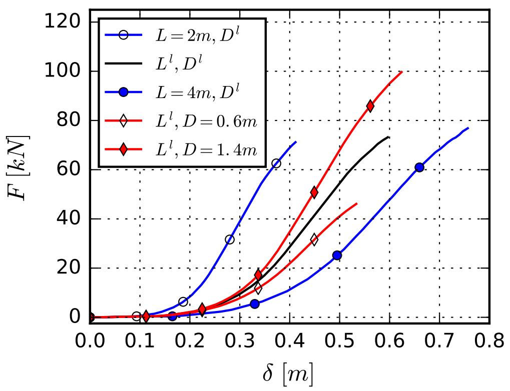

Figure 3. Force-displacement curves for different values of D and of L in the laboratory test configuration ($L^l = 3.0$m, $D^l = 1.0$m).

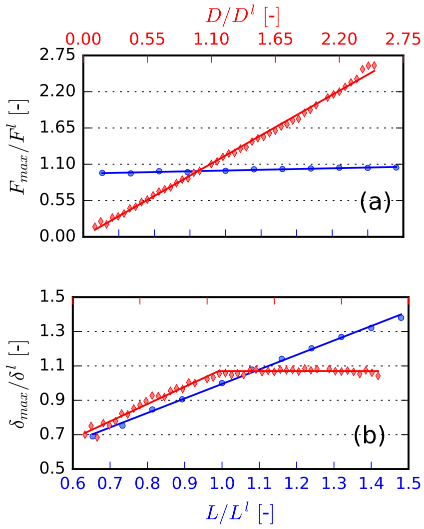

Figure 4. Effect of D (●) and of L (♦) in terms of: **(a)** maximum force F_{max} and **(b)** deflection at failure δ_{max}.

were considered), with an incremental step of 0.25m.

The effect of a variation of D on the mechanical response of the mesh is reported in Fig. 3, while the trends of the force at failure F_{max} and the related deflection δ_{max} with D are reported in Fig. 4a and Fig. 4b respectively. The force value at failure shows a monotonic increment with the punching element size, while the deflection value at failure firstly increases with D, then it stabilizes to a constant value. This is related to the finite size of the mesh specimen, and should be related to a threshold value of the ratio D/L as discussed in [1]. A threshold value $D/L = 0.4$ was observed independently of the considered value of the panel dimension L; the results referring to the cases $L \neq 3$m are here omitted for the sake of brevity.

The influence of a variation of L on the force-displacement response of the mesh is shown in Fig. 3. Larger mesh panels have to undergo a greater deflection before actively contrasting the punch displacement. Furthermore, they also show a lower out-of-plane stiffness; for instance, by referring to the stiffness parameter k_{75} as defined in [1]: $k_{75} = 247$kN/m for $L = 2$m, $k_{75} = 193$kN/m for $L = 3$m, $k_{75} = 170$kN/m for $L = 4$m. The influence of a variation of the sample dimension on the characteristic values F_{max} and δ_{max} is shown in Fig. 4a and Fig. 4b respectively. It is evident that a variation of L has a significant effect on the panel deflection, while its influence on the maximum force is almost negligible. The effect of L on F_{max} and δ_{max} can be estimated through Eqs. 1-2, which are obtained from a linear least squares fit of the numerical data. Similar relations were obtained in [1] in order to quantify the effect of the punching element dimension.

$$F_{max} = \left(0.89 + 0.11\frac{L}{L^l}\right) F_l^* \tag{1}$$

$$\delta_{max} = \left(0.15 + 0.85\frac{L}{L^l}\right) \delta_l^* \tag{2}$$

Comparing the effect of the geometrical parameters D and L on the mesh panel mechanical response, it is interesting to note how the maximum punching force is mostly controlled by the

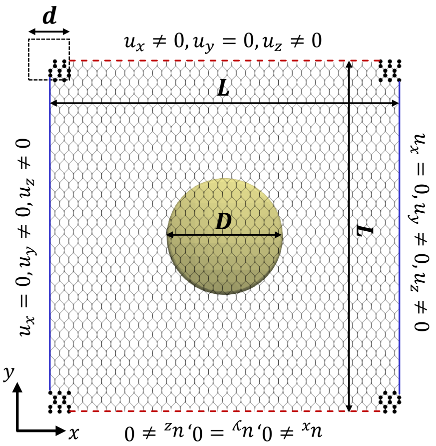

Figure 5. Top view of the numerical model of the anchored punch test. Bold points indicate the fixed nodal particles.

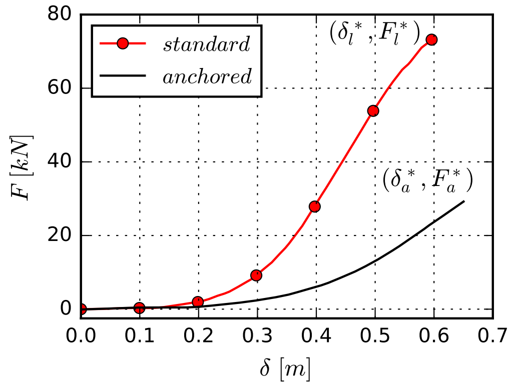

Figure 6. Comparison of the mechanical response obtained from the standard and the anchored punch test configurations.

dimension of the punching element; instead, the deflection at failure, is controlled by both the dimensions D and L at first, then only by the panel dimension when the ratio D/L is greater than a threshold value (i.e. $D/L > 0.4$).

4. Towards field conditions: anchored mesh panel punch test

The standard punch test configuration is not representative of the field conditions, especially concerning the boundary conditions imposed to the mesh panel; indeed, the laboratory specimen is fixed along its outer boundary, while in field applications the mesh is only locally fixed to the slope face by anchor plates. This may induce an overestimation of the field puncturing resistance [1], therefore the mechanical response obtained through the laboratory test cannot be directly used to characterize the field behavior of the mesh. In a recent work, Pol et al. [1] have defined a periodic configuration, in a discrete element framework, permitting to characterize the mechanical response of an anchored mesh panel. This configuration provides a lower bound estimation of the punching resistance of an anchored mesh panel [1]. In this work, the same configuration is adopted; a sketch of the geometry of the numerical model is presented in Fig. 5 for the sake of clarity. A square mesh panel of side length L is adopted and a dome-shaped punching element of diameter D is used. Periodic boundary conditions are imposed along the panel's edges, while the anchor plates are simulated by fixing all the degrees of freedom of the nodal particles that are ideally intercepted by them (bold points in Fig. 5). In order to respect the periodicity of the problem, half of the anchor plate side length (i.e. $d/2$) is considered in the model. The numerical parameters are the ones reported in Sec. 2.

In Fig. 6 the mechanical response obtained in the standard punch test configuration is compared with the one obtained considering the periodic configuration above described, characterized by $D = 1.0$m, $L = 3.0$m and $d = 0.32m$. It is evident that adopting a more realistic schematization of the field boundary conditions leads to a reduction of the punching force that mesh panel can support, as well as to a lower out-of-plane stiffness with respect to the one observed in the standard laboratory procedure. As stated in [1] the ratios between the values of force and deflection at failure observed in the standard and in the anchored ($F_a^* = 29.1$kN, $\delta_a^* = 0.65$m) configurations may be used to a first estimation of the variation of the mesh characteristics values moving from laboratory to field conditions: $\alpha_F = F_a^*/F_l^* = 0.40$, $\alpha_\delta = \delta_a^*/\delta_l^* = 1.08$. It should be noted that the values of these coefficients may vary if a different test geometry and/or a different mesh are adopted.

4.1. Master curve

In the perspective of defining a master curve that permits reconstruction of the entire force-displacement response of an anchored mesh panel to an out-of-plane loading conditions a large number (>300) of numerical simulations was performed. In each simulation a different set of the geometrical variables of the problem (i.e. d, L, D) is used, the specific values being randomly chosen in a reasonable wide range: $0.16\text{m} \leq d \leq 0.48\text{m}$, $1.25\text{m} \leq L \leq 4.5\text{m}$, $0.1\text{m} \leq D \leq 2.5\text{m}$. Subsequently, the F-δ curves are normalized with respect to the related F_{max} and δ_{max} values, showing that they collapse inside a narrow envelope (filled area in Fig. 7), and thus that the definition of a master curve is possible. From each normalized curve ($\tilde{F}$-$\tilde{\delta}$ curve) a "new" curve composed by a set of equispaced points along the normalized displacement axis is derived through interpolation of the original data. Finally, a master curve is derived by fitting the "new" data set using a non-linear least square regression model (model function $f(x) = ax^3 + bx^2 + cx$). The thus obtained fitting curve (i.e master curve) is given by:

$$\tilde{F}\left(\tilde{\delta}\right) = 1.312\tilde{\delta}^3 - 0.385\tilde{\delta}^2 + 0.073\tilde{\delta} \tag{3}$$

In Fig. 7 the master curve is compared with the envelope of the normalized force-displacement curves obtained from the numerical simulations. The master curve is well representative of the trend of the force-displacement response.

4.2. Forecasting methodology

In order to forecast the mechanical response of a given anchored mesh panel the master curve defined in Sec. 4.1 has to be coupled with a methodology that permits estimation of the force and deflection values at failure characteristic of the considered mesh system. In this perspective, the analytical relations defined in [1] are adopted (see Appendix). They permits extension of the characteristic values deriving from the standard punch test (i.e. F_l^*, δ_l^*) to a generic mesh system (i.e. F_a, δ_a). The thus obtained forecasting method (referred as $\hat{F}(\delta)$ in the following) can be summarized with the following three steps procedure:

- the reference values F_a^* and δ_a^* are computed from the values F_l^* and δ_l^* obtained from the standard laboratory characterization (e.g. ISO punch test) of the considered mesh type through the application of the coefficient α_F and α_δ;
- the characteristic values F_a and δ_a of the mesh system considered in the analysis are computed through the application of Eqs. A.1-A.2. The effect of the relative position between the punching element and the anchors can instead be estimated by using the contour plots reported in Appendix (see Fig. A1);
- the entire force-displacement behavior of the mesh system is forecasted by rescaling the master curve with the characteristic values F_a and δ_a computed in the previous step.

A conceptual scheme of this procedure is shown in Fig. 8 for the sake of clarity. The validation of the $\hat{F}(\delta)$ method above described is provided by comparing the predicted F-δ curves with the one obtained from ex-post simulations. The mechanical response of five test cases are compared in Fig. 7b-f; the problem's variables are randomly chosen (sampled from a uniform distribution) in the same range used for the definition of the master curve; these are specified at the top of the related figure and in Tab. 1. The mechanical response obtained through the $\hat{F}(\delta)$ methodology and the one deriving from the ex-post simulations are in good agreement for all of the considered cases. A very precise prediction of the entire force-displacement behavior is provided by the application of the forecasting procedure when a centered punching element is considered (see Fig. 7b-d). Instead, when introducing also an eccentricity of the punching element position, with respect to the center of the mesh panel (e_x and e_y in Fig. 8), the predicted F-δ curve slightly deviates from the simulated one (see Fig. 7e-f); however, the differences between the two curves

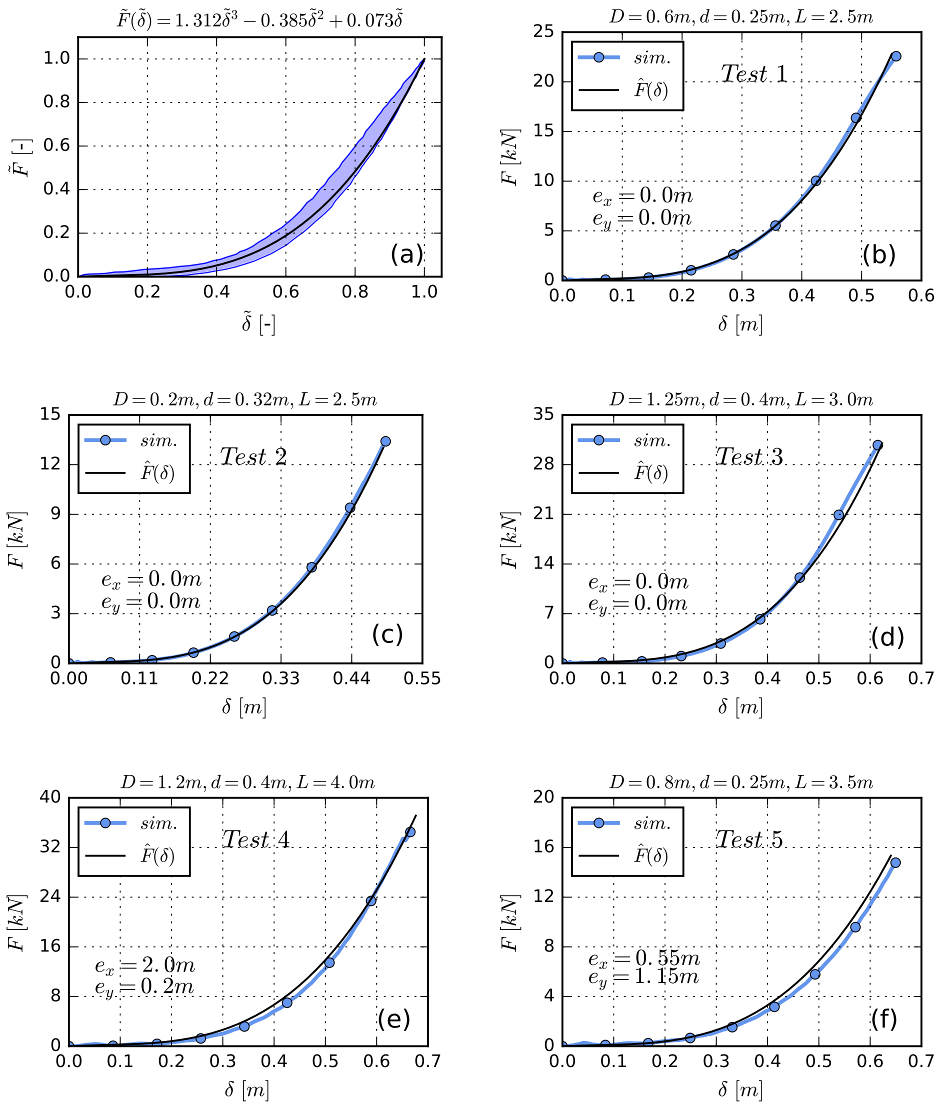

Figure 7. **(a)** Envelope of the normalized F-δ curves. **(b)-(f)** Comparison of the panel mechanical response obtained from ex-post simulations and from the forecasting method $\hat{F}(\delta)$.

Test	Problem's variables					DEM results		Eqs. A.1-A.2	
#	d [m]	L [m]	D [m]	e_x [m]	e_y [m]	F_{max} [kN]	δ_{max} [m]	F_a [kN]	δ_a [m]
1	0.25	2.5	0.60	0.00	0.00	22.6	0.56	22.8 (+1%)	0.55 (-2%)
2	0.32	2.5	0.20	0.00	0.00	13.4	0.49	13.0 (-3%)	0.49 (<1%)
3	0.40	3.0	1.25	0.00	0.00	30.9	0.62	31.0 (<1%)	0.62 (<1%)
4	0.40	4.0	1.20	2.00	0.20	34.9	0.67	37.1 (+6%)	0.68 (+1%)
5	0.25	3.5	0.80	0.55	1.15	14.9	0.65	15.3 (+3%)	0.64 (-2%)

Table 1. Comparison of the values of F_{max} and δ_{max} obtained from ex-post simulations and through the Eqs. A.1-A.2 (values in brackets estimate the error in the predicted values).

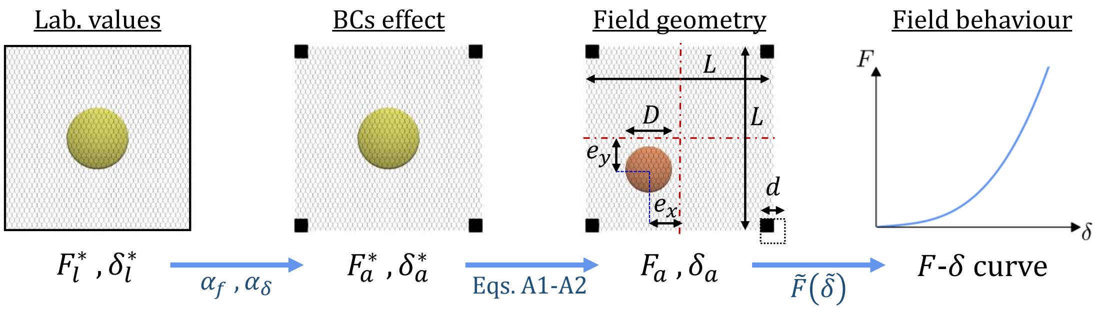

Figure 8. Conceptual scheme of the workflow of forecasting method $\hat{F}\,(\delta)$.

remain small. In light of the obtained results, the proposed method appears as a promising tool for forecasting the force-displacement behavior of an anchored mesh panel, starting from the only knowledge of the characteristic values of a standard laboratory punch test. A further result is related to the confirmation of the validity of the relations given in [1]. Even if, the latter were defined from a parametric analysis in which each parameter was assumed as independent, their sequential application permits precise prediction of the values at failure of a mesh system characterized by a generic combination of the geometrical parameters d, L, D, e_x and e_y. Minor differences are in fact observed between the predicted F_a and δ_a values and the values at failure obtained from the numerical simulations as reported in Tab. 1.

5. Conclusion

In this work, discrete element simulations have been used to define a simple tool for forecasting the mechanical response of anchored mesh panels subjected to a punching load. Initially, the effect of the specimen dimension in standard laboratory punch test conditions was analyzed. Analytical relations, complementary to ones proposed in a previous work [1], were derived. These relations permit extension of the results obtained in different configuration to a "common" one.

The data obtained from a large set of numerical simulations were used in order to derive a master curve that allows the trend of the force-displacement response of an anchored mesh panel to be reconstructed. A simple tool permitting the prediction of the mechanical response (i.e. F-δ curve) of a generic anchored mesh panel was defined by associating to the master curve the analytical relations defined in [1]. However, a further validation against experimental data on anchored panels is required to prove the robustness of the method. The approach is implemented in an online tool (http://geotechlab.dicea.unipd.it/codes/design-tool-for-drapery-mesh).

In the perspective to move towards a displacement-based design methodology, the proposed approach represents a fundamental tool permitting prediction of the entire force-displacement behavior of an anchored mesh panel, starting from the laboratory characteristic values (i.e. punch test results). Alternatively, it can be associated to the standard design methodologies (based on limit equilibrium method) in order to estimate the expected maximum deformations of the mesh system in serviceability conditions (i.e. hybrid approach).

A test configuration (Sec. 4) representative of a multi-block loading condition was considered since it provides a lower bound value of the mesh resistance; therefore, this represents a conservative choice in a design perspective. Further investigations are ongoing to extend the proposed approach to the case of a localized block acting on a large mesh system. The results presented in this work refer to the adopted mesh type; nevertheless, the proposed approach can be extended to other mesh types and test configurations in a very straightforward manner.

Appendix

The analytical relations permitting estimation of the effect of the problem's geometrical variables on the characteristic values F_a and δ_a defined in [1] are here condensed in Eqs. A.1-A.2. The reference values are: $F_a^* = 29.1\text{kN}$, $\delta_a^* = 0.65\text{m}$, $D^* = 1.0\text{m}$, $d^* = 0.32\text{m}$, $L^* = 3.0\text{m}$. The effect of the punching element eccentricity (i.e. e_x, e_y) is estimated through the contour plots reported in Fig. A1. The latter are derived normalizing the data reported in [1] on the panel dimension.

$$\begin{cases} F_a = \left[2.47\frac{D}{D^*}\left(0.48\frac{d}{d^*}+0.45\right)\left(0.16\frac{L}{L^*}+0.84\right)\right]F_a^* & \text{if } D \lesssim d,\ \ d \geq 0.16m \\ F_a = \left[\left(0.06\frac{D}{D^*}+0.94\right)\left(0.48\frac{d}{d^*}+0.45\right)\left(0.16\frac{L}{L^*}+0.84\right)\right]F_a^* & \text{if } D \gtrsim d,\ \ d \geq 0.16m \end{cases} \tag{A.1}$$

$$\begin{cases} \delta_a = \left[0.98\frac{L}{L^*}\left(0.72\frac{D}{D^*}+0.78\right)\left(0.02\frac{d}{d^*}+0.98\right)\right]\delta_a^* & \text{if } D \lesssim d,\ \ d \geq 0.16m \\ \delta_a = \left\{0.98\frac{L}{L^*}\left[0.03\left(\frac{D}{D^*}\right)^2-0.16\frac{D}{D^*}+1.13\right]\left(0.02\frac{d}{d^*}+0.98\right)\right\}\delta_a^* & \text{if } D \gtrsim d,\ \ d \geq 0.16m \end{cases} \tag{A.2}$$

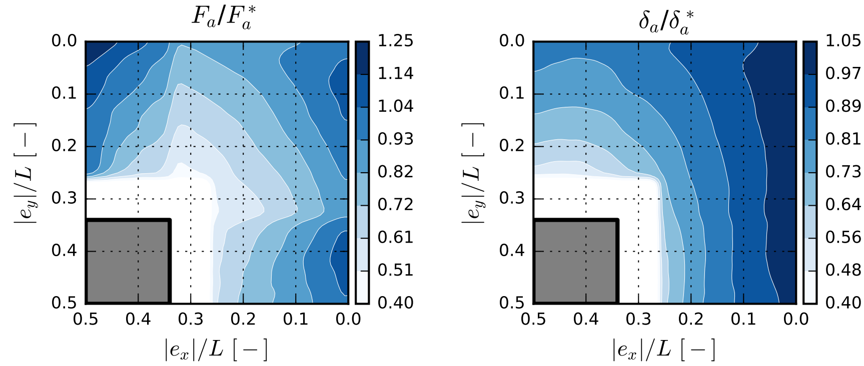

Figure A1. Effect of the punching element eccentricity e_x, e_y.

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