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# A MULTI-SCALE MODEL FOR CONTACT MECHANICS OF ROUGH SURFACES

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*Summary* An efficient methodology is proposed for the analysis of contact between rough surfaces. First, a micro-contact model is defined depending on surface topography and material properties. Second, the parameters issued from the model are integrated into a macro-scale numerical model following a specific method. An example of this method is presented to show its robustness. Comparing to a numerical model where surface topography is explicitly meshed, this strategy provides good results and saves considerably computation time.

## INTRODUCTION

Contact mechanics is of big interest in many mechanical systems such as friction brakes where the real contact area evolves constantly and influence drastically the system's performance. Indeed, contact interface is an important parameter to be considered because it plays a key role on many physical aspects such as stiffness, strength localisation, wear etc.

Consequently and given the scale where contact phenomena occur, it is nearly impossible to consider them in a macro-scale model without thinking on a multi-scale technique that transforms all the rules governing the micro-scale behaviour of contact interface into some sort of simplified laws that can be integrated into macro-scale models.

Accordingly, a multi-scale strategy is proposed to consider micro-contact phenomena's. First, for each considered contact phenomena, a micro-model is briefly described. Second, the multi-scale strategy of contact modelling is explained. Finally an example is presented and compared to a numerical model, where a real contact interface is explicitly modelled.

## MICRO-CONTACT MODELS

Considering a rigid plan making contact with an elastic body having a rough surface. Roughness can be seen as a random set of parabolic asperities (see fig.1). When contact occurs, surface asperities deform elastically and contact forces are distributed among them, and thus, contact stresses and displacements in each point are given by a combination of all contact forces. Contact zones are almost elliptic and pressure have a semi-elliptic shape. If the elastic body is considered is homogeneous, Hertz theory is used to describe pressure and displacements following an approach based on potential theory and developed in ([1],[2]). If the body have an elastic properties varying with the depth; the latter approach is no longer valid. For this case, a model have been developed using the Hankel transform [3] and transfer matrix techniques[5].

The latter models describe the elastic squeezing of asperities under normal forces. Now if the plan is sliding, shear stresses lead to an expansion of contact zones. Additionally, due to wear effect, a part of the debris issued from the interface are compacted and form a relatively smooth and flat contact zones, namely called "contact plateaus" (see fig.2), that have an arbitrary geometry. These zones are discretized into many square patches. The contact is mainly concentrated on these zones and the remaining asperities. The problem is solved using an optimization technique to minimize the system's energy ([4]), as well as the FFT technique to accelerate the solving scheme.

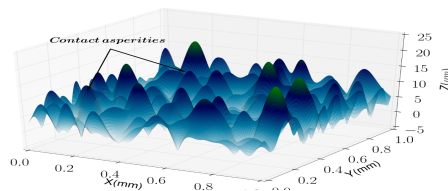


Figure 1: Schema of a surface containing asperities

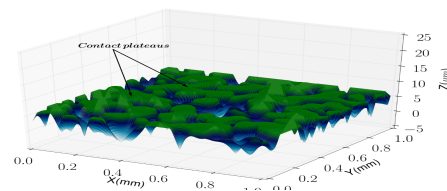


Figure 2: Schema of a surface containing plateaus

## MULTI-SCALE STRATEGY

The aim of this strategy is to integrate the microscopic behaviour of an interface in a macroscopic numerical model. Indeed surface roughness will not be represented physically in the model but only included via a parameter namely called contact stiffness  $K_n$ . Indeed, within finite element framework, contact problems are solved using regularization methods, such as the penalty method. More precisely, this technique consists on pairing up contact pressure  $p_n$  to the gap  $g_n$  by a stiffness  $K_n$ :  $p_n = K_n \cdot g_n$ . If the contact is perfect, theoretically  $g_n$  should be equal to zero, but practically,  $K_n$  is chosen to be very large to ensure a very small gap. However, this is not the case of real surfaces. In a real interface  $g_n$  evolves continuously considering

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roughness effect. To fulfil the strategy's objective, contact stiffness  $K_n$  will be chosen depending on surface roughness. Practically, contact surface is divided into many zones, where each zone has its proper geometry and roughness. On the one hand, micro-contact models give an information on how real displacement is related to contact pressure, and on the other hand, one can calculate the elastic displacement corresponding to perfect contact situation. The gap is then deduced by a simple subtraction of these displacements from real displacement, and automatically stiffness is calculated by derivation. This operation is done for all surface zones so that each zone has its local stiffness.

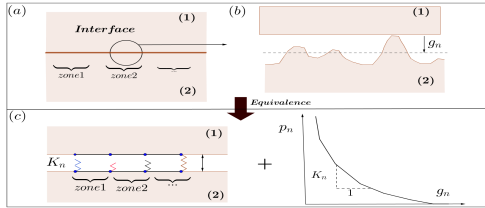


Figure 3: Schema of the proposed strategy

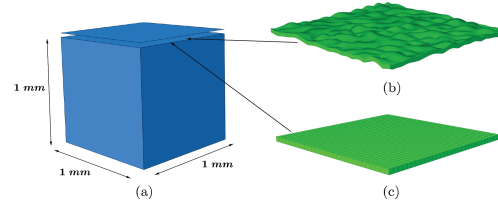


Figure 4: Numerical models

### EXAMPLE : CONSIDERATION OF ASPERITIES

Considering a homogeneous elastic cube and a rigid plan (see fig.4(a)). The plan is moved vertically against the cube with a given vertical motion  $\delta$ . Roughness is modelled by asperities. Two numerical models are proposed:

- Numerical reference model: A first model where asperities are explicitly meshed (see fig.4(b));
- Embedded numerical model: In the second model, the interface is flat. The surface is divided into  $25 \times 25$  square zones. For each zone, the contact interface is embedded with a local contact stiffness using a semi analytic model considering elliptic asperities and the strategy described above (see fig.4(c)).

In fig.5, the evolution of contact mean pressure vs  $\delta$  is presented for both numerical models and micro-model. The results obtained from embedded models match with the reference model and the semi analytic micro-model. In fig.6, contact pressure distribution is shown for different models. In fig.(a) and (b), it is clear that the number of contact zones, their shape and their orientation are consistent with the numerical reference model. In the embedded models (fig.6(c)), contact stresses distribution correspond to the contacting zones and depend on the size of each zone and its geometry. These zones are the ones where asperities are in contact, hence each zone describes macroscopically the micro mechanical behaviour of the asperities standing inside it. As regards the computational time, It has been reduced significantly. For the numerical reference model, the computational time is 3h whereas it does not exceed 15min for the embedded models and 10min for the semi-analytic model.

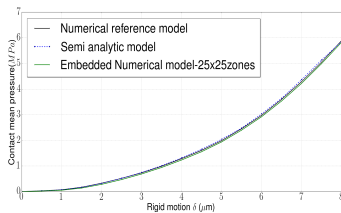


Figure 5: Mean pressure vs displacement

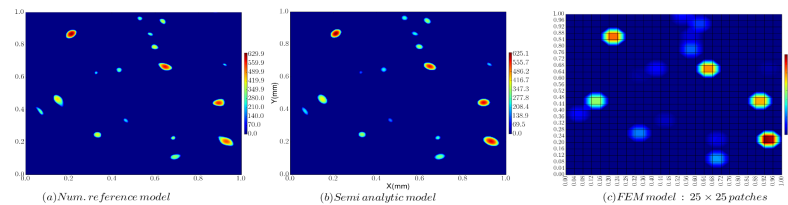


Figure 6: Contact pressure distribution (MPa) for different models

### CONCLUSIONS

In conclusion, various contact models of rough surfaces have been presented. Also a strategy to integrate these models into macroscopic models have been proposed. Numerical analysis shows consistent results and the computational time have been reduced remarkably.

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