EUROMECH Newsletter 29

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President's Address

2006 is an election year for the EUROMECH Council, which is the governing body of our Society. Four new members will be elected during the autumn. The Advisory Board (a list of current members is available on the web at www.euromech.org) will prepare a list of candidates for whom members of EUROMECH may vote in September-October 2006. Suggestions for candidates may be made to any member of the Advisory Board. If you wish to suggest a candidate, please make sure that he/she is willing to serve on the Council for six years. Please also supply a one page Curriculum Vitae and a complete address. The final choice of candidates will reflect both the need for some continuity with the remaining Council members and for a suitable distribution over the different countries in Europe. A sufficient representation of the different disciplines within fluid and solid mechanics should also be ensured.

Candidates will prepare biographical notes, which will be published in the Newsletter before the elections. The new members of the Council will take office on 1 January 2007.

Many of you will have noticed that EUROMECH has a brand new website (www.euromech.org), which has been expertly and tastefully designed and put together by our Secretary–General Bernhard Schrefler and Ms Sara Guttilla, Science Officer at CISM in Udine. You will find on the site up-to-date information regarding all the activities and events sponsored by the Society (colloquia, series of major conferences, prizes and awards, EUROMECH fellows, and more) as well as the possibility of applying for membership. Please do not hesitate to contact the Secretary-General if you wish to suggest new items to be included. We intend to use the site as an effective means to interact with all EUROMECH members.

We are also pleased to announce that thanks to the efforts of the Treasurer Wolfgang Schröder, EUROMECH has now officially been registered as a non-profit society under German law, based in Aachen, right in the centre of Europe! This statute will significantly facilitate applying for grants in order to support our activities.

Patrick Huerre

President, EUROMECH

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Prize Paper: Optimal shapes of a beam under parametric excitation

Alexei A. Mailybaev won the EUROMECH Young Scientist Prize, awarded at the fifth EUROMECH Non-linear Dynamics Conference Eindhoven, August 2005

Alexei A. Mailybaev[1], Hiroshi Yabuno[2] and Hiroyuki Kaneko[3]

Abstract

Straight elastically supported beams of variable width under the action of a periodic axial force are considered. Two shape optimization problems for reducing parametric resonance zones are studied. In the first problem, the minimal (critical) amplitude of the excitation force is maximized. In the second problem, the range of resonant frequencies is minimized for a given parametric resonance zone and a fixed amplitude of excitation. These two optimization problems are proved to be equivalent in the case of small external damping and small excitation force amplitude. It is shown that optimal designs have strong universal character, i.e. they depend only on the natural modes involved in the parametric resonance and boundary conditions. Optimal beam shapes are found for different boundary conditions and resonant modes. Experiments for uniform and optimal simply supported elastic beams have been conducted, demonstrating good agreement with theoretical prediction.

1. Introduction

Shape optimization of beams under stability criteria is an interesting and important topic from both theoretical and practical points of view. On one hand, this problem requires sophisticated analytical and numerical methods for sensitivity analysis and knowledge onof bifurcation and singularity theories. On the other hand, the correct formulation of the problem, including minimal width constraints and analysis of nonlinear behavior, may strongly influence the practical use of optimal designs. We refer the reader to the surveys on optimal beam problems for the case of static loads [8] and for the case of follower loads [4]. In [6] beams were optimized under several natural frequency constraints. The optimal shape of a pipe conveying fluid was investigated in [3]. Optimal shapes of a beam, minimizing the critical

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excitation amplitudes of primary parametric resonance zones, were analyzed numerically in [9]. Very few experimental works on optimal beams are available in the literature, see [3].

In this paper, we consider beams of rectangular cross-section with constant thickness and variable width. The length and total volume of the beam are fixed. Elastic supports at both ends are considered, which, in particular, include the cases of simply supported and clamped boundary conditions. Parametric resonance of the beam under periodic axial force excitation is studied. Two optimization problems are considered. In the first problem, the minimal (critical) amplitude of the excitation force is maximized. In the second problem, the range of resonant excitation frequencies is minimized for a given parametric resonance zone and a fixed amplitude of excitation. We prove that these two optimization problems are equivalent in the case of small external damping and small excitation force amplitude. Moreover, we show that the optimal designs do not depend on the damping coefficient, or the value of the excitation amplitude in the second problem, or the resonance number. All these properties reveal a clear universal character of optimal beam shapes, which is of great importance for practical use of optimal designs.

Optimal beam shapes are found numerically for different boundary conditions and resonant modes. Experiments on simple parametric resonance of the first mode are conducted for the uniform and optimal simply supported beams. The reduction of the resonance zone for the optimal beam is shown to be in very good agreement with theoretical prediction.

2. Basic Equations

Let us consider a straight elastic beam of rectangular cross-section having a length l, a thickness h, and a variable width $s(x)$, see Fig. 1. The beam is loaded by a periodic axial force $p(t) = p_0 + \delta\varphi(\Omega t)$, where δ and Ω are the amplitude and frequency of parametric excitation, respectively, and $\varphi(t)$ is a periodic function of period 2π. We study small vibrations of the beam in the xy-plane, see Fig. 1. A deflection of the beam at the position x and time t is denoted by $w(x,t)$. It is assumed that the beam is externally damped, and the damping force at the position x is proportional to the velocity and width of the beam with the external damping coefficient γ.

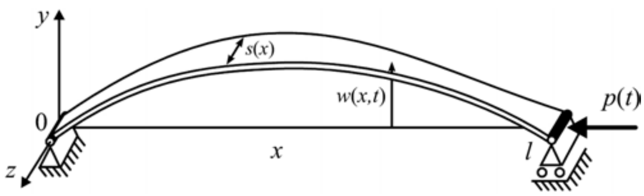

Fig. 1: Beam of variable width under parametric excitation.

The dimensionless equation of motion of the beam has the form

$$(s(x)w'')'' + p(t)w'' + \gamma s(x)\dot{w} + s(x)\ddot{w} = 0, \tag{1}$$

where the dimensionless variables and parameters are

$$t^* = \frac{h}{l^2}\sqrt{\frac{E}{12\rho}}\,t, x^* = \frac{x}{l}, w^* = \frac{w}{l}, s^* = \frac{hl}{V_0}s, \gamma^* = \frac{l^2}{h^2}\sqrt{\frac{12}{E\rho}}\,\gamma,$$
$$p_0^* = \frac{12l^3}{h^2EV_0}p_0, \delta^* = \frac{12l^3}{h^2EV_0}\delta, \Omega^* = \frac{l^2}{h}\sqrt{\frac{12\rho}{E}}\,\Omega, \tag{2}$$
$$p^*(t^*) = p_0^* + \delta^*\varphi(\Omega^* t^*),$$

The dot $(\dot{})$ and prime $(')$ represent the derivatives with respect to t^* and x^*, respectively (in equation (1) and hereafter, the asterisk $(^*)$ is omitted. Here E is the Young modulus, ρ is the density, and $V_0 = h\int_0^l s(x)dx$ is the total volume of the beam.

The external damping term $\gamma s(x)\dot{w}$ in equation (1) takes into account the variation of the damping force due to the change of beam width. The damping coefficient γ depends on the viscosity and density of the external medium (gas or fluid) and on the frequency of beam vibrations, see e.g. [1]. Within a fixed parametric resonance zone, the range of frequency of beam vibrations is small. Thus, the damping coefficient γ can be assumed to be constant for each particular resonance zone.

We consider the case of a beam elastically supported at both ends, having the boundary conditions

$$w(0,t) = w(1,t) = 0, (-c_1w' + sw'')_{x=0} = (c_2w' + sw'')_{x=1} = 0, \tag{3}$$

where $c_1 \geq 0$ and $c_2 \geq 0$ are dimensionless elastic coefficients of the supports. The limit cases $c_1 = c_2 = 0$ and $c_1 = c_2 = \infty$ correspond to simply supported and clamped-clamped beams, respectively.

The total volume of the beam is assumed to be fixed, which yields the dimensionless condition

$$\int_0^1 s(x)dx = 1. \tag{4}$$

The width function $s(x)$ can be constrained from below

$$s(x) \geq s_{\min}, \tag{5}$$

where $0 \leq s_{\min} < 1$ is the minimum allowed beam width.

Let us consider free vibrations of the beam without periodic excitation ($\delta = 0$) or damping ($\gamma = 0$), and assume that $p_0 < p_{cr}$, i.e., the constant compressive force p_0 is lower than the static critical force of the beam p_{cr}. Looking for a solution of (1) in the form $w(x,t) = u(x)\exp(i\omega t)$, we obtain the equation

$$(s(x)u'')'' + p_0 u'' = \omega^2 s(x) u \tag{6}$$

with the boundary conditions

$$u(0) = u(1) = 0, (-c_1 u' + s u'')_{x=0} = (c_2 u' + s u'')_{x=1} = 0 \tag{7}$$

determining the eigenfrequencies ω and corresponding eigenmodes $u(x)$ of the beam prestressed by a constant force p_0.

3. Parametric Resonance Zones

Let $0 < \omega_1 < \omega_2 < \cdots$ be eigenfrequencies of free vibration of the beam without parametric excitation and damping (all the eigenfrequencies are assumed to be distinct). The corresponding eigenmodes $u_1(x), u_2(x), \ldots$ determined by equation (6) with boundary conditions (7) can be chosen satisfying the normalization conditions

$$\int_0^1 s u_i^2 dx = 1, \quad i = 1, 2, \ldots. \tag{8}$$

If small parametric excitation is applied, the trivial equilibrium of the beam, $w(x,t) \equiv 0$, may become unstable (the parametric resonance occurs). For beams of variable cross-section, the parametric resonance may happen only if the excitation frequency Ω is close to the resonant value $\Omega_0 = (\omega_i + \omega_j)/k$ for some eigenfrequencies ω_i, ω_j and a positive integer k, see e.g. [5]. The cases $i = j$ and $i \neq j$ are called simple and summed combination resonances, respectively. Difference combination resonances corresponding to the excitation frequencies $\Omega_0 = (\omega_i - \omega_j)/k$, $i > j$, do not occur. For small excitation amplitudes δ and damping coefficients γ, we find the parametric resonance zones for system (1), (3) using the results of [5] (see also [7, Chap. 11]) as

$$\gamma^2 - \frac{b_{ij}^2 (a_k^2 + b_k^2)}{4\omega_i \omega_j} \delta^2 + k^2 \left(\Delta\Omega - \frac{c_0 (\omega_i b_{jj} + \omega_j b_{ii})}{2k\omega_i \omega_j} \delta \right)^2 \leq 0, \tag{9}$$

where

$$\Delta\Omega = \Omega - \Omega_0, b_{ij} = -\int_0^1 u_i u_j dx, c_0 = \frac{1}{2\pi} \int_0^{2\pi} \varphi(t) dt, \tag{10}$$

$$a_k = \frac{1}{\pi} \int_0^{2\pi} \varphi(t) \cos(kt) dt, b_k = \frac{1}{\pi} \int_0^{2\pi} \varphi(t) \sin(kt) dt. \tag{11}$$

Inequality (9) represents the first order approximation of the stability domain valid for small values of δ and γ.

If the damping coefficient $\gamma > 0$ is fixed, the minimal absolute (critical) value of the excitation amplitude δ, at which the parametric resonance may occur, is (see [5])

$$\delta_{res}(\gamma) = 2\gamma\sqrt{\frac{\omega_i\omega_j}{b_{ij}^2(a_k^2 + b_k^2)}}. \qquad (12)$$

If $\delta > \delta_{res}$, the instability occurs for excitation frequencies belonging to a certain interval. The width of this resonant interval equals

$$\Delta\Omega_{res}(\delta,\gamma) = \frac{1}{k}\sqrt{\frac{b_{ij}^2(a_k^2 + b_k^2)}{\omega_i\omega_j}\delta^2 - 4\gamma^2}. \qquad (13)$$

4. Optimization Problems

Let us fix the value of the damping coefficient $\gamma > 0$, and consider the shape of the beam maximizing the critical excitation amplitude δ_{res} given by expression (12). Since the quantities a_k and b_k do not depend on $s(x)$, the optimal shape can be found by the minimization of the objective functional

$$\Phi(s) = \frac{|\, b_{ij}\, |}{\sqrt{\omega_i\omega_j}} = \frac{1}{\sqrt{\omega_i\omega_j}}\left|\int_0^1 u'_i\, u'_j\, dx\right| \to \min. \qquad (14)$$

The shape $s(x)$ appears in expression (14) implicitly through the eigenfrequencies ω_i, ω_j and corresponding eigenmodes $u_i(x)$, $u_j(x)$, which are defined by equation (6) and boundary conditions (7). The critical excitation amplitude is expressed using the functional Φ as $\delta_{res} = 2\gamma/(\Phi\sqrt{a_k^2 + b_k^2})$.

Let us consider the width $\Delta\Omega_{res}$ of the interval of resonant excitation frequencies for a specific resonance zone and fixed parameters $\gamma > 0$ and $\delta > \delta_{res}(\gamma)$. This width is given by formula (13). As a second optimization problem, consider the shape $s(x)$ of the beam which minimizes $\Delta\Omega_{res}$. Using the functional Φ defined in (14), we obtain $\Delta\Omega_{res} = \frac{1}{k}\sqrt{(a_k^2 + b_k^2)\Phi^2\delta^2 - 4\gamma^2}$. Again, we arrive at the minimization problem (14).

The analytical formulation of the optimization problem is to find the shape $s(x)$ minimizing the functional $\Phi(s)$ under the constant volume constraint (4). One can see that the optimal shape of the beam depends only on the modes involved in the resonance (indexes i and j), on the constant compressive force p_0, and on the elastic support coefficients c_1 and c_2. This optimal shape has a universal character, since it solves both the maximization problem for the critical excitation amplitude $\delta_{res}(\gamma) \to \max$ with an arbitrary fixed damping coefficient γ and the minimization problem for the resonant frequency range $\Delta\Omega_{res}(\delta,\gamma) \to \min$ with arbitrary fixed δ and γ. The optimal beam shape does not depend on the resonance number *k* and on the form of the periodic excitation function $\varphi(t)$. We note that these conclusions are based on the first order approximation of the resonance zone (9). Hence, they are valid for small δ and γ.

5. Optimal Shapes

In this section, we analyze optimal shapes of the beam minimizing the size of simple resonance zones ($i = j$). For numerical computation of eigenfrequencies and eigenmodes we used the finite difference method [2], where the beam was divided into $n = 100$ equal parts. The gradient method was used for beam optimization, where derivatives of the objective function, eigenfrequencies, and eigenmodes were evaluated for the discrete system. Comparing the results with the $n = 200$ elements approximation, we estimate the relative error to be about 0.1% for the minimum value of objective function Φ and 0.2% for the shape function $s(x)$. As the initial shape, we consider the uniform beam with $s(x) \equiv 1$. For some cases, the global search with over 1000 random nonsymmetric initial shapes was performed by using the $n = 20$ elements approximation. The computations showed that the derived optimal designs correspond to the global minimum of the objective function.

For a simply supported beam ($c_1 = c_2 = 0$) and zero static compressive force ($p_0 = 0$), the optimal shape corresponding to the first mode ($i = 1$) is shown in Fig. 2(a) by a thin solid line. Here we did not use the minimum width constraint ($s_{\min} = 0$). The values of the objective function for the uniform and optimal beams are equal to $\Phi_{uniform} = 1$ and $\Phi_{optimal} = 0.791$, respectively. Thus, the objective function describing the size of simple resonance zones corresponding to the first mode is decreased by 20.9%.

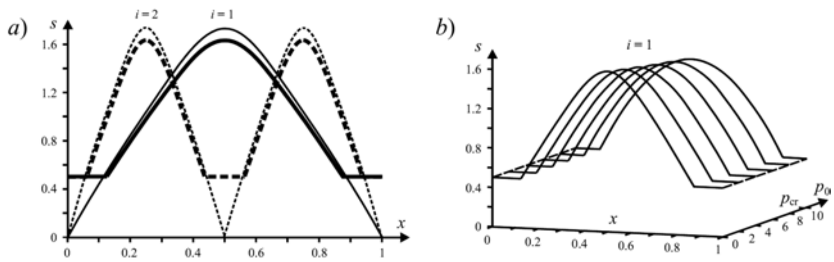

Fig. 2: Optimal shapes of a simply supported beam.

The optimal shape corresponding to the second mode ($i = 2$) is shown in Fig. 2(a) by a thin dashed line. The optimal beam has zero widths at the supports and in the middle. Zero width in the middle of the beam can be regarded physically as a hinge connecting two parts of the beam. In this case, however, the deflection function $w(x,t)$ and the eigenmode $u(x)$ can have a discontinuity of the slope at $x = 0.5$. Such a beam becomes a mechanism losing stability at zero force. Therefore, the optimal shape of the beam corresponding to the second mode is physically irrelevant. In order to avoid the situation when the width vanishes at some points, we restrict the minimum beam width by using constraint (5) with $s_{\min} = 0.5$. The optimal beam shapes for the first and second modes are shown in Fig. 2(a) by bold solid and bold dashed lines, respectively. The relative reduction of the objective functional equals 17.67% and 17.74% for the first and second modes, respectively. The derived solutions are physically correct and have practical form.

Dependence of the optimal shape on the static compressive force p_0 for the first mode is shown in Fig. 2(b), where the minimal width constraint $s_{\min} = 0.5$ is used ($p_{cr} = \pi^2$ is a critical static force). One can see that the optimal shapes depend weakly on the compressive force p_0.

The results for a clamped-clamped beam ($c_1 = c_2 = \infty$) and zero static compressive force ($p_0 = 0$) are presented in Fig. 3(a). The optimal shapes for the first and second modes ($i = 1$ and 2) are shown by solid and dashed lines, respectively. The thin and bold lines correspond to the unconstrained ($s_{\min} = 0$) and constrained ($s_{\min} = 0.5$) beam widths. Compared to the uniform beam, the reduction of the objective functional is 18.63% and 18.80% for the optimal beams with $s_{\min} = 0.5$ corresponding to the first and second modes, respectively.

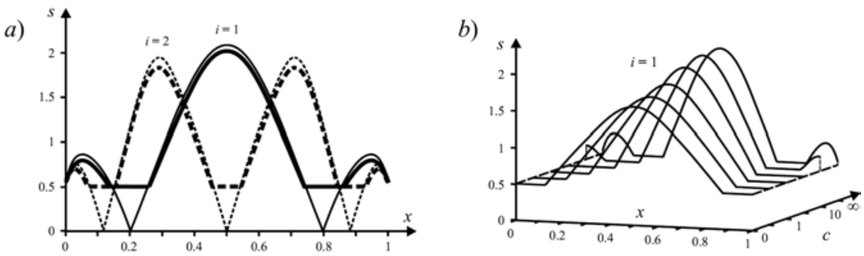

Fig. 3: Optimal shapes: (a) clamped-clamped beam, (b) elastically supported beams for different elastic support coefficients

Finally, let us consider elastically supported beams with equal elastic coefficients of supports $c = c_1 = c_2$. Dependence of optimal beam shapes on the elastic coefficient c is presented in Fig. 3(a), where the optimization was carried out for the first mode ($i = 1$), zero static compressive force $p_0 = 0$, and the width constraint $s_{\min} = 0.5$. The considerable dependence of the optimal shape on the elastic coefficient c is observed.

6. Experiments

Experiments were conducted for the uniform and optimal simply supported beams. The optimization is carried out for simple resonance zones of the first mode (lowest natural frequency) with zero static compressive force $p_0 = 0$ and the minimal width constraint $s_{\min} = 0.5$, which yields the optimal design shown in Fig. 2(a) by a bold solid line.

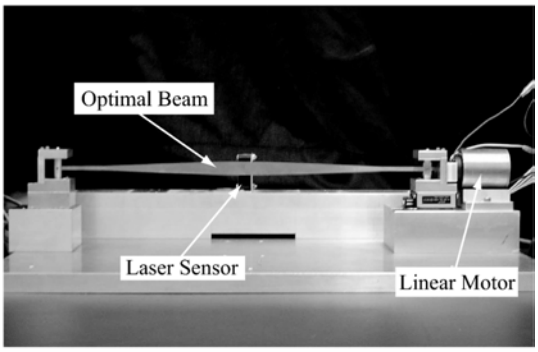

Fig. 4: Experimental arrangement

Uniform and optimal phosphor-bronze beams were used in the experiments. A photograph of the experimental arrangement is shown in Fig. 4. The parameters of the uniform beam are:

$$m = 0.0322\,kg, \quad l = 0.45\,m, \quad h = 0.0008\,m,$$
$$s_{uniform} = 0.01\,m, \quad \frac{\omega_{uniform}}{\pi^2} = 4.098\,s^{-1}, \tag{15}$$

where $\omega_{uniform}/(2\pi) = 6.437\,Hz$ is the experimentally measured first natural frequency of the beam. The dimensionless external damping coefficient $\gamma = 0.184$ is obtained by matching the theoretically computed decrement of free oscillations of the uniform beam with the experimental data. The optimal beam has the same mass, length, and thickness.

The derived experimental boundaries of parametric resonance zones in the plane of dimensionless frequency and amplitude of the excitation force are shown in Fig. 5(a). One can observe very good agreement of the experimental data with theoretical prediction.

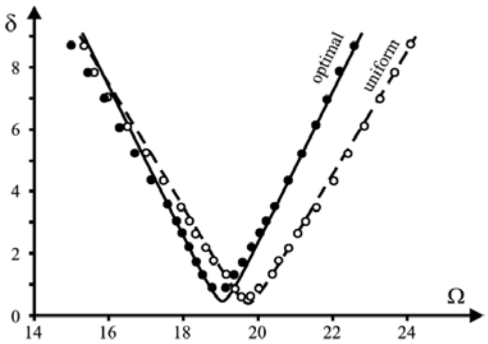

Fig. 5: Boundary of the parametric resonance zone: uniform beam (° experimental, $---$ theoretical) and optimal beam (• experimental, $——$ theoretical)

7. Conclusions

In this paper, we studied optimal shapes of elastic beams under the action of a periodic axial force. Two optimization problems have been considered: the maximization of the minimal (critical) excitation amplitude and the minimization of the interval of resonant excitation frequencies for a fixed excitation force amplitude. It is shown that these two problems lead to the same objective function, depending only on the frequencies and modes of free vibrations of the beam. Optimal beam shapes are found for different boundary conditions and resonant modes. The gain of the optimization is about 17–22% for the objective function depending on the boundary conditions and width constraints. The experiments conducted for simply supported uniform and optimal beams showed good agreement with theoretical predictions.

From a practical point of view, the important feature of an optimal design is its weak sensitivity to changes in different parameters. We proved that the optimal beam shapes obtained in the paper do not depend on the amplitude and shape of the periodic force function, or the resonance number, or the external damping coefficient. Thus, the optimal beams possess a strong universality property, and depend only on the boundary conditions and natural modes involved in the parametric resonance.

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Prize Paper: Simulation of non-smooth mechanical systems with many unilateral constraints

Christian Studer won the EUROMECH Young Scientist Prize,
awarded at the fifth EUROMECH Non-linear Dynamics Conference
Eindhoven, August 2005

Christian Studer Christoph Glocker

Abstract

The two main methods used to simulate non-smooth dynamical systems are the event-driven and time-stepping approach. Systems with many non-smooth constraints can only be handled by time-stepping methods. An example of a simple time-stepping method is the Moreau midpoint rule. Instead of accelerations, velocity updates are calculated. This leads to inclusions describing the discretized system. The inclusions are turned into non-linear equations and solved by an iterative method. This procedure is known as the Augmented Lagrangian Method. In the paper we will use the following non-smooth constraints: unilateral contacts, planar friction contacts and spatial friction contacts. Time-stepping methods are well suited to systems with many unilateral constraints. We will show an example of a thousand balls falling into a funnel. Because common time-stepping algorithms require small time steps, they are not advisable for systems with just a few non-smooth constraints. A proposal for a time step adjustment is given at the end of the paper.

Key words

non-smooth dynamical systems, time-stepping, setvalued force laws, event-driven methods, Augmented Lagrangian

1. Introduction

A ball falling to the ground is a simple example of a non-smooth mechanical system with one unilateral contact. In a planar modelling, the ball's three degrees of freedom are reduced to two when the ball touches the ground. If we additionally consider friction, then the degrees of freedom are reduced to one in the case of sticking. Thus, we have different equations of motion for these different contact configurations. In case of an impact, an impact law must be applied. Event driven methods [6] simulate non-smooth systems by separating the motion into smooth parts and switching points at which the system equations are changed. These methods are only suited to systems with a few contacts. Time-stepping methods [8], [11] calculate velocity updates instead of accelerations. As a consequence, contact behaviour and impact can

be treated by the same equations. The non-smooth constraints are modelled by set-valued force laws [7]. Time-stepping methods allow a robust simulation of dynamical systems with many unilateral contacts. As an example we show a thousand balls falling down a funnel. Time-stepping methods require an overall fixed small time step to resolve the switching points of the system. Thus mechanical systems with few contacts must also be treated with a very small time step. A way out of this problem is a time step adjustment.
The discrete formulation of a non-smooth system can be written as inclusions. These inclusions are turned into non-linear equations, which can be solved by an iterative method. This procedure is known as the Augmented Lagrangian Method[1], [10].

2. Description of a non-smooth system

A non-smooth mechanical system without impacts but with n non-smooth constraints can be described by the equation of motion and n set-valued force laws. In the case of an impact, an impact equation together with n set-valued impact laws has to be stated [6], [7], [8]. For simplicity we will exclude an explicit dependency of the dynamical system on time t.

2.1. Equation of motion

The equation of motion is

$$\begin{aligned} \dot{\mathbf{u}} &= \mathbf{M}^{-1}(\mathbf{h} + \sum_{i=1}^{n} \mathbf{W}_i \lambda_i), \\ \dot{\mathbf{q}} &= \mathbf{u} \end{aligned} \tag{1}$$

We denote by $\mathbf{M} = \mathbf{M}(\mathbf{q}(t))$ the positive definite mass matrix, by $\mathbf{q} = \mathbf{q}(t)$ the generalized coordinates and by $\mathbf{u} = \mathbf{u}(t)$ the generalized velocities. All bilateral constraints of the system are taken into account by the generalized coordinates q (Lagrange II). The vector $\mathbf{h} = \mathbf{h}(\mathbf{q}(t), \mathbf{u}(t))$ contains all external and gyroscopic forces. The set-valued force laws are considered by a Lagrange I formulation of the equation of motion. We denote the contact forces of the i-th non-smooth constraint by $\lambda_i = \lambda_i(t)$ and the corresponding generalized force directions by $\mathbf{W}_i = \mathbf{W}_i(\mathbf{q}(t))$.

2.2. Set-valued force laws

The set-valued force laws are described by normal cones N_C to a set C. The following sets C are used

$$\begin{aligned} C &= \mathbb{R}_0^+, \\ C &= S_2(a) = \{x \in \mathbb{R} \mid \, | x | \leq a\}, \\ C &= S_3(a) = \{\mathbf{x} \in \mathbb{R}^2 \mid \, || \mathbf{x} || \leq a\}. \end{aligned} \tag{2}$$

The normal cones for these selected sets C are

$$\begin{aligned}
N_{\mathbb{R}_0^+}(x) &= \begin{cases} 0 & if \quad x \in \mathbb{R}_0^+/\partial\mathbb{R}_0^+ \\ -\mathbb{R}_0^+ & if \quad x \in \partial\mathbb{R}_0^+ \end{cases} \\
N_{S_2(a)}(x) &= \begin{cases} 0 & if \quad x \in S_2(a)/\partial S_2(a) \\ \mathbb{R}_0^+ \dfrac{x}{\mid x \mid} & if \quad x \in \partial S_2(a) \end{cases} \\
N_{S_3(a)}(\mathbf{x}) &= \begin{cases} 0 & if \quad \mathbf{x} \in S_3(a)/\partial S_3(a) \\ \mathbb{R}_0^+ \dfrac{\mathbf{x}}{\mid\mid \mathbf{x} \mid\mid} & if \quad \mathbf{x} \in \partial S_3(a). \end{cases}
\end{aligned} \tag{3}$$

For each contact we define a displacement function $g_i = g_i(\mathbf{q}(t))$. The time derivative $\dot{g}_i$ of this function is $\gamma_i = \gamma_i(\mathbf{q}(t), \mathbf{u}(t))$ almost everywhere. We introduce the set P_N of all unilateral contacts, the set P_{T2} of all planar friction contacts and the set P_{T3} of all spatial friction contacts. The contact i can either be a unilateral contact $i \in P_N$, a planar friction contact $i \in P_{T2}$ or a spatial friction contact $i \in P_{T3}$. In the case of a unilateral contact, the function g_i is the normal distance between the contact points, and γ_i is the relative contact velocity in normal direction. The corresponding set-valued force law on displacement and velocity level is

$$\begin{aligned}
-g_i & & \in N_{\mathbb{R}_0^+}(\lambda_i) \qquad & i \in P_N, \\
-\gamma_i &= -\mathbf{w}_i^\top \dot{\mathbf{q}} & \in N_{\mathbb{R}_0^+}(\lambda_i) \qquad & i \in P_N.
\end{aligned} \tag{4}$$

Note that the set-valued force law of a unilateral contact formulated on velocity level is only applicable if $g_i = 0$. For a planar friction contact, the displacement function g_i is the tangential displacement and γ_i is the relative contact velocity in tangential direction. The corresponding set-valued force law is defined on velocity level

$$-\gamma_i = -\mathbf{w}_i^\top \dot{\mathbf{q}} \in N_{S_2(a_i)}(\lambda_i) \qquad i \in P_{T2}. \tag{5}$$

For spatial friction, the relative tangent contact velocity γ_i and the contact forces λ_i are planar vectors in the tangent plane. They can be described in an arbitrary tangential coordinate system $(\mathbf{t}_1, \mathbf{t}_2)$. The set-valued force law on velocity level is

$$-\gamma_i = -\mathbf{W}_i^\top \dot{\mathbf{q}} \in N_{S_3(a_i)}(\lambda_i) \qquad i \in P_{T3}. \tag{6}$$

The scalar $a_i = a_i(t)$ is the maximum friction force in the i-th friction contact. Note that we treat the unilateral contact $j \in P_N$ and the corresponding friction contact $i \in P_{T2,3}$ as two contacts, so $a_i = \mu\lambda_j$. Introducing the maximum friction force a_i instead of $\mu\lambda_j$ allows more flexibility.

2.3. Impact equation

In the case of an impact, the equation of motion (1) is not applicable due to the velocity jump from the pre-impact velocity $\mathbf{u}^-$ to the post-impact velocity $\mathbf{u}^+$. This jump requires infinitely large contact forces λ_i. Therefore, we integrate the equation of motion over the instantaneous impact time $\int_{t^-}^{t^+}$ and arrive at the impact equations

$$\begin{aligned} \mathbf{u}^+ - \mathbf{u}^- &= \mathbf{M}^{-1} \sum_{i=1}^{n} \mathbf{W}_i \Lambda_i, \\ \mathbf{q}^+ - \mathbf{q}^- &= 0. \end{aligned} \tag{7}$$

The impulsive forces $\mathbf{\Lambda}_i$ are obtained by integrating the infinitely large contact forces $\boldsymbol{\lambda}_i$ over the instantaneous impact time.

$$\begin{aligned} \mathbf{\Lambda}_i &= \int_{t^-}^{t^+} \boldsymbol{\lambda}_i \, dt, \\ A_i &= \int_{t^-}^{t^+} a_i \, dt. \end{aligned} \tag{8}$$

A more detailed explanation can be found in [8]. Further, we have to define impact laws to obtain a relation between the impulsive forces Λ_i and the relative kinematics.

2.4. Newton's impact law

The Newton impact law for a unilateral contact, a planar friction contact and a spatial friction contact can be expressed in the normal cone formulations:

$$\begin{aligned} -(\gamma_i^+ + \varepsilon_i \gamma_i^-) &\in N_{\mathbb{R}_0^+}(\Lambda_i) & i \in P_N, \\ -(\gamma_i^+ + \varepsilon_i \gamma_i^-) &\in N_{S_2(A_i)}(\Lambda_i) & i \in P_{T2}, \\ -(\boldsymbol{\gamma}_i^+ + \varepsilon_i \boldsymbol{\gamma}_i^-) &\in N_{S_3(A_i)}(\mathbf{\Lambda}_i) & i \in P_{T3}. \end{aligned} \tag{9}$$

The relative contact velocity before the impact is γ_i^-, the relative contact velocity after the impact is γ_i^+. The restitution coefficient ε_i lies between zero and one. Note that the impact law for a unilateral contact is only valid if $g_i = 0$.

2.5. Event-driven methods

It is common to simulate non-smooth systems using event-driven methods. The idea is to detect all points in time for which the dynamical system changes, such as points at which an impact occurs, or at which a friction contact starts to slide. Points at which such events happen are called switching points. The period between these switching points can be described by a common smooth formulation and can be treated classically. Reaching a switching point, the further behaviour of the system is analyzed with the methods of non-smooth mechanics. When an impact occurs, the impact equations (7) and the impacts laws (9) are evaluated to obtain the new initial conditions. Analyzing the equation of motion (1) and the set-valued force laws (4-6), the further contact state can be determined and a new smooth formulation for the further integration can be stated. Event driven methods are complicated, because the mechanical system has to be rearranged for the different smooth periods. The treatment of a system with many switching points or even accumulating switching points by event-driven methods is not advisable.

2.6. Remarks

The formulation of the set-valued force law of a unilateral contact using velocity level is only valid if the contact is closed, that is $g_i = 0$. An open unilateral contact is not allowed. We will call all closed unilateral contacts "active contacts". The friction contact is defined by velocity level, so that no further restriction has to be made. Note that consideration of a friction contact whose maximum friction force a_i is equal to zero is a waste of computing power.

3. Discretization

In this section we will show one possible discretization of a non-smooth mechanical system. The so called time-stepping methods merge the equation of motion and the impact equation. We will discuss the time-stepping method known as Moreau's midpoint rule [11].

3.1. Discrete equation of motion

The equation of motion (1) is integrated over a finite time interval $\Delta t = t_E - t_B$:

$$\begin{aligned} \mathbf{u}_E - \mathbf{u}_B &= \int_{t_B}^{t_E} \mathbf{M}^{-1}(\mathbf{h} + \sum_{i=1}^{n} \mathbf{W}_i \boldsymbol{\lambda}_i \, dt), \\ \mathbf{q}_E - \mathbf{q}_B &= \int_{t_B}^{t_E} \mathbf{u} \, dt. \end{aligned} \tag{10}$$

By the index B we denote the velocity and displacement at t_B, the index E stands for the time t_E. We define the impulsive forces [4]

$$\begin{aligned} \widehat{\boldsymbol{\Lambda}}_i &= \int_{t_B}^{t_E} \boldsymbol{\lambda}_i \, dt, \\ \widehat{A}_i &= \int_{t_B}^{t_E} a_i \, dt. \end{aligned} \tag{11}$$

[4]Normally only the instantaneous integral (8) is called impulsive force. In this paper we will expand the term "impulsive force" to the finite integral (10), which is a discretization of (8)

The mass matrix $\mathbf{M}$ and the matrix of the generalized force directions $\mathbf{W}_i$ are treated as constant in the time interval $[t_A, t_B]$. By the index M we denote the midpoint time. We define

$$\begin{aligned} \mathbf{M}_M &= \mathbf{M}(\mathbf{q}_B + \frac{\Delta t}{2}\mathbf{u}_B), \\ \mathbf{W}_{Mi} &= \mathbf{W}_i(\mathbf{q}_B + \frac{\Delta t}{2}\mathbf{u}_B). \end{aligned} \quad (12)$$

With these assumptions we obtain

$$\begin{aligned} \mathbf{u}_E - \mathbf{u}_B &= \mathbf{M}_M^{-1}(\int_{t_B}^{t_E} \mathbf{h}\,dt + \sum_{i=1}^{n} \mathbf{W}_{Mi}\widehat{\mathbf{\Lambda}}_i), \\ \mathbf{q}_E - \mathbf{q}_B &= \int_{t_B}^{t_E} \mathbf{u}\,dt. \end{aligned} \quad (13)$$

The two remaining integrals are approximated as

$$\begin{aligned} \int_{t_B}^{t_E} \mathbf{h}\,dt &= \mathbf{h}_M \Delta t, \\ \int_{t_B}^{t_E} \mathbf{u}\,dt &= \frac{\mathbf{u}_E + \mathbf{u}_B}{2}\Delta t, \end{aligned} \quad (14)$$

with

$$\mathbf{h}_M = \mathbf{h}(\mathbf{q}_B + \frac{\Delta t}{2}\mathbf{u}_B, \mathbf{u}_B). \quad (15)$$

Thus the discrete form of the equation of motion according to Moreau's midpoint rule is

$$\begin{aligned} \mathbf{u}_E &= \mathbf{u}_B + \mathbf{M}_M^{-1}(\mathbf{h}_M \Delta t + \sum_{i=1}^{n} \mathbf{W}_{Mi}\widehat{\mathbf{\Lambda}}_i), \\ \mathbf{q}_E &= \mathbf{q}_B + \frac{\mathbf{u}_B + \mathbf{u}_E}{2}\Delta t. \end{aligned} \quad (16)$$

For $t_E \to t_B$ we obtain in the case of an impact the impact equation (7). In the case of no impact, $\mathbf{u}_E = \mathbf{u}_B$ and $\mathbf{q}_E = \mathbf{q}_B$.

3.2 Discrete set-valued impulsive force laws

In this subsection we will set up the discrete set-valued impulsive force laws. These laws should cover both set-valued force laws (4-6) and impact laws (9). We state

$$\begin{aligned} -(\gamma_{Ei} + \varepsilon_i \gamma_{Bi}) &\in N_{\mathbb{R}_0^+}(\hat{\Lambda}_i) & i &\in P_N, \\ -(\gamma_{Ei} + \varepsilon_i \gamma_{Bi}) &\in N_{S_2(\hat{\Lambda}_i)}(\hat{\Lambda}_i) & i &\in P_{T2}, \\ -(\boldsymbol{\gamma}_{Ei} + \varepsilon_i \boldsymbol{\gamma}_{Bi}) &\in N_{S_3(\hat{\Lambda}_i)}(\widehat{\mathbf{\Lambda}}_i) & i &\in P_{T3}, \end{aligned} \quad (17)$$

where $\widehat{\mathbf{\Lambda}}_i$ is the integral of the contact force (11). We detect active unilateral contacts by a discrete form of $g_i = 0$:

$$g_{iM} = g_i(\mathbf{q}_B + \mathbf{u}_A \frac{\Delta t}{2}) < 0. \tag{18}$$

For $t_E \to t_B$ we obtain in the case of an impact the impact laws (9) by replacing γ_{Ei} with γ_i^+ and γ_{Bi} with γ_i^-. In the case of no impact we determine the set-valued force laws (4-6) using the velocity level. According to the mean value theorem we write

$$\begin{aligned} \lim_{t_E \to t_B} \widehat{\mathbf{\Lambda}}_i &= \lim_{t_E \to t_B} \int_{t_B}^{t_E} \boldsymbol{\lambda}_i \, dt \\ &= \boldsymbol{\lambda}_{Bi} \lim_{\Delta t \to 0} \Delta t = \boldsymbol{\lambda}_{Bi} \, dt. \end{aligned} \tag{19}$$

Using $dt > 0$ we obtain

$$\begin{aligned} N_{\mathbb{R}_0^+}(\lambda_{Bi} \, dt) &= N_{\mathbb{R}_0^+}(\lambda_{Bi}), \\ N_{S_2(a_{Bi}dt)}(\lambda_{Bi} \, dt) &= N_{S_2(a_{Bi})}(\lambda_{Bi}), \\ N_{S_3(a_{Bi}dt)}(\boldsymbol{\lambda}_{Bi} \, dt) &= N_{S_3(a_{Bi})}(\boldsymbol{\lambda}_{Bi}). \end{aligned} \tag{20}$$

For $t_E \to t_B$ the relative velocity γ_{Bi} is identical to γ_{Ei}

$$\gamma_{Ei} + \varepsilon_i \gamma_{Bi} = (1 + \varepsilon_i)\gamma_{Bi}. \tag{21}$$

A multiplication of a cone by a positive scalar does not change the cone. It holds that

$$\frac{1}{1 + \varepsilon_i} N_C(\boldsymbol{\lambda}_{Bi}) = N_C(\boldsymbol{\lambda}_{Bi}). \tag{22}$$

with the non-negative restitution coefficient $\varepsilon_i \in [0,1]$. Thus, for $t_E \to t_B$ and no impact, the set-valued impulsive force laws (17) become

$$\begin{aligned} -\gamma_{Bi} &\in N_{\mathbb{R}_0^+}(\lambda_{Bi}) & i \in P_N, \\ -\gamma_{Bi} &\in N_{S_2(a_{Bi})}(\lambda_{Bi}) & i \in P_{T2}, \\ -\gamma_{Bi} &\in N_{S_3(a_{Bi})}(\boldsymbol{\lambda}_{Bi}) & i \in P_{T3}, \end{aligned} \tag{23}$$

The criterion for the detection of an active unilateral contact (18) is

$$g_{Bi} = g_i(\mathbf{q}_\mathrm{B}) < 0. \tag{24}$$

The laws (23) and (24) are the set-valued force laws (4-6) defined by velocity level.

3.3 Time-stepping inclusions

By defining global vectors of all contact velocities, impulsive forces and contact force directions

$$\gamma = (\gamma_1^T ... \gamma_n^T)^T,$$
$$\widehat{\boldsymbol{\Lambda}} = (\widehat{\boldsymbol{\Lambda}}_1^T ... \widehat{\boldsymbol{\Lambda}}_n^T)^T, \tag{25}$$
$$\boldsymbol{W}_M = (\boldsymbol{W}_{M1} ... \boldsymbol{W}_{Mn})$$

we can rearrange the first equation of (16) to

$$\gamma_E + \varepsilon\gamma_B = \boldsymbol{G}\widehat{\boldsymbol{\Lambda}} + \boldsymbol{c}. \tag{26}$$

The matrix G and the vector c are

$$\boldsymbol{G} = \boldsymbol{W}_M^\top \boldsymbol{M}_M^{-1} \boldsymbol{W}_M,$$
$$c = (\boldsymbol{I} + \varepsilon)\gamma_B + \boldsymbol{W}_M^\top \boldsymbol{M}_M^{-1} \boldsymbol{h}_M \Delta t. \tag{27}$$

The matrix ε is a diagonal matrix with the entries ε_i. The matrix **G** is positive definite if all active force directions $\mathbf{W}_{Mi}$ are independent. This means that the active constraints cannot cause an underdetermined system in any contact configuration. Otherwise, the matrix **G** is only positive semidefinite. The diagonal entries of **G** are larger than zero because the mass matrix $\mathbf{M}_M$ is assumed to be positive definite. The equation (26) merged with the discrete set-valued impulsive force laws (17) gives n inclusions describing the n individual contacts. The inclusion for a unilateral contact is

$$\sum_{j=1}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j + N_{\mathbb{R}_0^+}(\widehat{\Lambda}_i) \ni -c_i \qquad i \in P_N. \tag{28}$$

For a planar friction contact we obtain

$$\sum_{j=1}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j + N_{S_2(\widehat{A}_i)}(\widehat{\Lambda}_i) \ni -c_i \qquad i \in P_{T2}. \tag{29}$$

A spatial friction contact $i \in P_{T3}$ is described by

$$\sum_{j=1}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j + N_{S_3(a_i)}(\widehat{\boldsymbol{\Lambda}}_i) \ni -c_i \qquad i \in P_{T3}. \tag{30}$$

By stating the appropriate inclusion for each contact, the system is completely described.

3.4 Algorithm

The time-stepping method can be stated as follows [11]

i. Calculate the position

$$\boldsymbol{q}_M = \boldsymbol{q}_B + \boldsymbol{u}_B \frac{\Delta t}{2}. \tag{31}$$

ii. Define all closed unilateral contacts i

$$i \in \{j \in P_N \mid g_j(\boldsymbol{q}_M) < 0\} \tag{32}$$

as active contacts on velocity level. Because the physical level of the friction contacts is determined by the velocity level, all friction contacts can be viewed as active. Note that considering a friction contact $i \in P_{T2,3}$ belonging to a non-active unilateral contact $j \in P_N$ is a waste of computing power, because the maximum impulsive friction force $\hat{A}_i = \mu \hat{\Lambda}_j$ is zero. Such a friction contact can be regarded as non-active friction contact.

iii. Set up the generalized force directions $\mathbf{W}_{Mi}$ (12) for all active contacts.

iv. Calculate the vector $\mathbf{h}_M$ (15) and $\mathbf{M}_M$ (12). Calculate $\mathbf{G}$ and $\mathbf{c}$ according to (27).

v. Solve the time-stepping inclusions (28-30). As result we obtain the impulsive forces $\hat{\mathbf{\Lambda}}$ and thus the velocities $\mathbf{u}_E$ (16).

vi. Calculate the end position $\mathbf{q}_E$

$$\boldsymbol{q}_E = \boldsymbol{q}_B + \frac{\boldsymbol{u}_E + \boldsymbol{u}_B}{2} \Delta t = \boldsymbol{q}_M + \boldsymbol{u}_E \frac{\Delta t}{2}. \tag{33}$$

3.5 Remarks

The main idea of the time-stepping methods involves using the impulsive forces $\hat{\mathbf{\Lambda}}_i$ instead of the forces $\boldsymbol{\lambda}_i$. This enables us to treat impacts and smooth motion by the same discrete equations. Instead of accelerations, velocity updates are calculated.

Time Stepping methods can be viewed as event-driven methods in which the switching points are not instantaneous but have a finite "time-length". In addition, time-stepping methods can change their contact state automatically, and no different equations of motion have to be formulated for the different periods between the switching points. The time step should be chosen to be very small, in order to resolve the switching points. Systems with many non-smooth constraints switch permanently and require an overall small time step, so that a fixed time-stepping method is appropriate. However, systems with few non-smooth constraints may have long periods in which no switching point occurs. In this case, the use of a fixed small time step is a waste of computing power. This problem can be bypassed by using a variable time step.

The combination of the equation of motion (1) and the set-valued force laws (4-6) forms a differential algebraic system (DAE). The set-valued force laws on displacement level cause an index 3 DAE, the formulation on velocity level results in an index 2 DAE. The use of the impulsive forces $\hat{\mathbf{\Lambda}}_i$ makes the solution of this index 2 DAE possible.

Various time-stepping methods [5], [9], [14] differ in the approximation of the integrals (14). Also the discrete set-valued impulsive force laws (17) are formulated in a different manner. Formulation of the set-valued impulsive force laws by velocity level allows for a good consolidation of impact laws (9) and set-valued force laws (4-6). A disadvantage is the drift in the unilateral contacts caused by numerical errors. Formulation of a unilateral contact by displacement level eliminates the drift problem, but consideration of the impact law with $\varepsilon > 0$ becomes cumbersome. The formulation on displacement level results in an index 3 DAE, whose solution is a problem for $t_E \to t_B$. Some time-stepping methods merge velocity and displacement levels.

4. Solving the time-stepping inclusions

The time-stepping inclusions (28-30) can be solved in various ways. One way is to transform the inclusions in linear and non-linear complementarity problems [6], [8], [4]. We will focus on another method which transforms the inclusions into non-linear equations. This can be interpreted as stating the adequate conditions for the saddle point of the Augmented Lagrangian, as either exact regularization of the set-valued impulsive force laws, or successive solutions of the individual inclusions [1], [1], [10], [11]. We will focus on the latter interpretation. For $r > 0$, the simple inclusion [12]

$$\boldsymbol{x} + rN_C(\boldsymbol{x}) \ni \mathbf{b} \tag{34}$$

is equal to

$$\boldsymbol{x} = prox_C(\boldsymbol{b}). \tag{35}$$

The $prox_C$ function describes the projection on the set C, that is the point $\boldsymbol{x} = prox_C(\boldsymbol{b})$. is the nearest point to b in the set C (proximal point). The $prox_C$ functions for the selected sets C are:

$$\begin{aligned}
prox_{\mathbb{R}_0^+}(b) &= \begin{cases} b & if \quad b \in \mathbb{R}_0^+ \\ 0 & if \quad b \notin \mathbb{R}_0^+ \end{cases} \\
prox_{S_2(a)}(b) &= \begin{cases} b & if \quad b \in S_2(a) \\ a\dfrac{b}{\mid b \mid} & if \quad b \notin S_2(a) \end{cases} \\
prox_{S_3(a)}(\mathbf{b}) &= \begin{cases} \mathbf{b} & if \quad \mathbf{b} \in S_3(a) \\ a\dfrac{\mathbf{b}}{\mid\mid \mathbf{b} \mid\mid} & if \quad \mathbf{b} \notin S_3(a). \end{cases}
\end{aligned} \tag{36}$$

The relation between the inclusion (34) and the non-linear equation (35) can be shown by writing the normal cone $N_C(\boldsymbol{x})$ as a subdifferential of the indicator function $\Psi_C(\boldsymbol{x})$. The inclusion (34) becomes

$$0 \in \frac{1}{r}(\boldsymbol{x} - \mathbf{b}) + \partial\Psi_C(\boldsymbol{x}). \tag{37}$$

Integrating (37) leads to

$$\arg\min_{x} \frac{1}{2r} \, || \, \boldsymbol{x} - \mathbf{b} \, ||^2 \, + \Psi_C(\boldsymbol{x}) \quad \forall \boldsymbol{x}. \tag{38}$$

This unconstrained optimization problem can be turned into a constrained optimization problem

$$\arg\min_{x} \frac{1}{2r} \, || \, \boldsymbol{x} - \mathbf{b} \, ||^2 \quad \forall \boldsymbol{x} \in C. \tag{39}$$

The solution $\mathbf{x}$ of this constrained optimization problem (39) is the nearest point to $\mathbf{b}$ in the set C. The relation between (34) and (35) can also be interpreted as the solution of one single non-smooth constraint. With the help of (34) and (35), the time-stepping inclusions (28-30) can be turned into non-linear equations.
The inclusion for a unilateral contact (28) can be written as

$$-\sum_{\substack{j=1 \\ j \neq i}}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j - c_i \in G_{ii} \widehat{\Lambda}_i + N_{\mathbb{R}_0^+}(\widehat{\Lambda}_i). \tag{40}$$

Since $\alpha_i = G_{ii} > 0$ is a scalar, we obtain

$$\begin{aligned} \hat{\Lambda}_i &= prox_{\mathbb{R}_0^+}(J_i(\hat{\boldsymbol{\Lambda}})), \\ J_i(\hat{\boldsymbol{\Lambda}}) &= -\frac{1}{\alpha_i}(\sum_{\substack{j=1 \\ j \neq i}}^{n} \boldsymbol{G}_{ij} \hat{\boldsymbol{\Lambda}}_j + c_i). \end{aligned} \tag{41}$$

The impulsive force $\hat{\boldsymbol{\Lambda}}_i$ appears only on the left hand side of (41). Thus it can be computed instantaneously if all other forces $\hat{\boldsymbol{\Lambda}}_j$ are known. The quotient $\frac{\boldsymbol{G}_{ij}}{\alpha_i}$ shows the direct influence of the j-th contact on the unilateral contact i.
The non-linear equation for a planar friction contact (29) follows when replacing the set $\mathbb{R}_0^+$ by $S_2(a_i)$ in the non-linear equation (41).
The inclusion of a spatial friction constraint (30) can be reformulated as

$$-\sum_{\substack{j=1 \\ j \neq i}}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j - \boldsymbol{c}_i \in \boldsymbol{G}_{ii} \widehat{\boldsymbol{\Lambda}}_i + N_{S_3(\hat{A}_i)}(\widehat{\boldsymbol{\Lambda}}_i). \tag{42}$$

Since the matrix $\boldsymbol{G}_{ii}$ is not a scalar, it has to be splitted into a scalar $\alpha_i > 0$ and a remaining part $\boldsymbol{B}$

$$\boldsymbol{G}_{ii} = \alpha_i \mathbf{I} + \mathbf{B}. \tag{43}$$

The inclusion (30) becomes

$$\begin{aligned} &-\sum_{\substack{j=1 \\ j \neq i}}^{n} \boldsymbol{G}_{ij} \widehat{\boldsymbol{\Lambda}}_j - \boldsymbol{c}_i - \boldsymbol{B} \widehat{\boldsymbol{\Lambda}}_i \in \\ &\in \alpha_i \widehat{\boldsymbol{\Lambda}}_i + N_{S_3(\hat{A}_i)}(\widehat{\boldsymbol{\Lambda}}_i). \end{aligned} \tag{44}$$

Using (35) we obtain the non-linear equation

$$\hat{\mathbf{\Lambda}}_i = prox_{S_3(\hat{A}_i)}(\boldsymbol{J}_i(\hat{\mathbf{\Lambda}})),$$
$$\boldsymbol{J}_i(\hat{\mathbf{\Lambda}}) = -\frac{1}{\alpha_i}(\sum_{\substack{j=1 \\ j \neq i}}^{n} \boldsymbol{G}_{ij}\hat{\mathbf{\Lambda}}_j + \boldsymbol{c}_i + \boldsymbol{B}\hat{\mathbf{\Lambda}}_i). \tag{45}$$

Note that the impulsive force $\hat{\mathbf{\Lambda}}_i$ appears on both sides of equation (45). The spatial friction contact i can not be solved by one projection if all other impulsive contact forces $\hat{\mathbf{\Lambda}}_j$ are known. The remainder matrix $\boldsymbol{B}$ depends on the choice of the the tangential unit vectors $\boldsymbol{t}_1$ and $\boldsymbol{t}_2$ of the spatial friction contact.

4.1 Solving the non-linear equations

The non-smooth dynamical system is thus described by n non-linear equations (41) and (45). We will discuss only the unilateral contact and the spatial friction contact, because the non-linear equation of a planar friction contact is very similar to the non-linear equation of a unilateral contact. The non-linear equations (41) and (45) can be solved by a Newton-Raphson iteration method [3] or any other fixed point iteration method. We will focus on a Jacobi and a Gauss Seidel like iteration method [3], [13].

4.1.1 Jprox method

One possible iterative instruction to solve the non-linear equations (41) and (45) is

$$\begin{aligned} \hat{\Lambda}_i^{\nu+1} &= prox_{\mathbb{R}_0^+}(J_i(\hat{\mathbf{\Lambda}}^\nu)) \qquad i \in P_N \\ \hat{\mathbf{\Lambda}}_i^{\nu+1} &= prox_{S_3(\hat{A}_i)}(\boldsymbol{J}_i(\hat{\mathbf{\Lambda}}^\nu)) \quad i \in P_{T3} \end{aligned} \tag{46}$$

Note that the instruction $\hat{\mathbf{\Lambda}}_i^{\nu+1} = J_i(\hat{\mathbf{\Lambda}}^\nu)$ is the Jacobi iteration instruction to solve the linear system

$$\boldsymbol{G}\hat{\mathbf{\Lambda}} + \boldsymbol{c} = 0. \tag{47}$$

The instruction $\hat{\mathbf{\Lambda}}_i^{\nu+1} = J_i(\hat{\mathbf{\Lambda}}^\nu)$ can be viewed as a Jacobi-relaxation instruction for two components h and $(h+1)$ at once. Thus the instruction (46) is the Jacobi method combined with a projection. We will call this method the Jprox method, because it combines the Jacobi method (J method) with the prox function. The Jprox method solves the individual contacts, using the assumption that all other impulsive contact forces $\hat{\mathbf{\Lambda}}_j^\nu$ are known. The calculation of $\hat{\mathbf{\Lambda}}_i^{\nu+1}$ does not make use of the fact that $\hat{\mathbf{\Lambda}}_{i-1}^{\nu+1}$ is already known.

4.1.2 JORprox method

The Jprox method (46) is altered in the sense that the underlying linear equation system (47) is solved by the Jacobi relaxation method (JOR). Therefore, we rearrange the inclusion of a unilateral contact (28) by adding $\frac{\hat{\Lambda}_i}{r_i}$ on both sides:

$$-\sum_{i=1}^{n} \boldsymbol{G}_{ij}\hat{\boldsymbol{\Lambda}}_j - c_i + \frac{\hat{\Lambda}_i}{r_i} \in \frac{\hat{\Lambda}_i}{r_i} + N_{\mathbb{R}_0^+}(\hat{\Lambda}_i) \tag{48}$$

The same can be done for the inclusion of a spatial friction contact (30).
The corresponding non-linear equation for a unilateral contact $i \in P_N$ is

$$\begin{aligned} \hat{\Lambda}_i &= prox_{\mathbb{R}_0^+}(JOR_i(\hat{\boldsymbol{\Lambda}})), \\ JOR_i(\hat{\boldsymbol{\Lambda}}) &= \hat{\Lambda}_i - r_i \left(\sum_{j=1}^{n} \boldsymbol{G}_{ij}\ \hat{\boldsymbol{\Lambda}}_j^{\nu}\ + c_i\right) \end{aligned} \tag{49}$$

For the spatial friction contact $i \in P_{T3}$ we obtain

$$\begin{aligned} \hat{\Lambda}_i &= prox_{S_3(\hat{A}_i)}(\boldsymbol{JOR}_i(\hat{\boldsymbol{\Lambda}})), \\ \boldsymbol{JOR}_i(\hat{\boldsymbol{\Lambda}}) &= \hat{\boldsymbol{\Lambda}}_i - r_i \left(\sum_{j=1}^{n} \boldsymbol{G}_{ij}\ \hat{\boldsymbol{\Lambda}}_j^{\nu}\ + c_i\right) \end{aligned} \tag{50}$$

The corresponding iterative instructions are

$$\begin{aligned} &\hat{\Lambda}_i^{\nu+1} = prox_{\mathbb{R}_0^+}(JOR_i(\hat{\boldsymbol{\Lambda}}^{\nu})), i \in P_N \\ &\hat{\boldsymbol{\Lambda}}_i^{\nu+1} = prox_{S_3(\hat{A}_i)}(\boldsymbol{JOR}_i(\hat{\boldsymbol{\Lambda}}^{\nu})), i \in P_{T3} \end{aligned} \tag{51}$$

We will call the iterative instructions (51) the JORprox method. The JORprox method consists of a JOR iteration combined with a projection. The factor r_i contains the relaxation parameter. Note that the instruction in (50) requires one r_i, although two components are iterated at once. That means that the relaxation parameters for both components are not independent. It can be shown that in case of dry friction the JORprox iteration converges if the underlying JOR method does. Convergence of the JOR method can only be guaranteed if the matrix $\boldsymbol{G}$ is strictly diagonal dominant. An optimal convergence can be achieved by minimizing the Lipschitz constant. Using the maximum norm we obtain an optimal r_i

$$r_i = \frac{1}{G_{ii}}. \tag{52}$$

Using this r_i we arrive at the original Jprox method. In the case of a spatial friction contact, $\boldsymbol{G}_{ii}$ is a matrix and the largest of the two diagonal elements can be chosen. Note that if the matrix $\boldsymbol{G}$ is not strictly diagonal dominant, then convergence cannot be guaranteed. For a small r_i the iteration might still converge because the Lipschitz constant is still near to one. A good empirical criterion is

$$r_i = \frac{1}{\sum_{k=1}^{m} | G_{hk} |}. \tag{53}$$

The index m denotes the dimension of the matrix $\boldsymbol{G}$ which is not equal to the number of constraints n, because the spatial friction contacts require two rows in $\boldsymbol{G}$. The index h denotes the row in $\boldsymbol{G}$ belonging to the i-th constraint. In case of a spatial friction constraint, h is chosen in a way that r_i becomes minimal. If the diagonal elements in $\boldsymbol{G}$ predominate, then the criteria (52) and (53) become similar.

4.1.3 SORprox method

An iterative instruction based on the Gauss Seidel relaxation method (SOR) is

$$\begin{aligned} \hat{\Lambda}_i^{\nu+1} &= prox_{\mathbb{R}_0^+}(SOR(\hat{\boldsymbol{\Lambda}}^{\nu+1}, \hat{\boldsymbol{\Lambda}}^{\nu}), i \in P_N \\ \hat{\boldsymbol{\Lambda}}_i^{\nu+1} &= prox_{S_3(\hat{A}_i)}(\boldsymbol{SOR}(\hat{\boldsymbol{\Lambda}}^{\nu+1}, \hat{\boldsymbol{\Lambda}}^{\nu})), i \in P_{T3} \end{aligned} \tag{54}$$

with the iterative instructions of the Gauss Seidel relaxation method for a linear system (47):

$$\begin{aligned} & SOR(\hat{\boldsymbol{\Lambda}}^{\nu+1}, \hat{\boldsymbol{\Lambda}}^{\nu}) = \\ & = \hat{\Lambda}_i^{\nu} - r_i (\sum_{j=1}^{j<i} \boldsymbol{G}_{ij} \hat{\boldsymbol{\Lambda}}_j^{\nu+1} + \sum_{j=i}^{n} \boldsymbol{G}_{ij} \hat{\boldsymbol{\Lambda}}_j^{\nu} + c_i) \\ & \boldsymbol{SOR}(\hat{\boldsymbol{\Lambda}}^{\nu+1}, \hat{\boldsymbol{\Lambda}}^{\nu}) = \\ & = \hat{\boldsymbol{\Lambda}}_i^{\nu} - r_i (\sum_{j=1}^{j<i} \boldsymbol{G}_{ij} \hat{\boldsymbol{\Lambda}}_j^{\nu+1} + \sum_{j=i}^{n} \boldsymbol{G}_{ij} \hat{\boldsymbol{\Lambda}}_j^{\nu} + \boldsymbol{c}_i), \end{aligned} \tag{55}$$

We will call method (54) the SORprox method. Different to the JORprox method, the calculation of $\hat{\boldsymbol{\Lambda}}_i^{\nu+1}$ makes use of the fact that $\hat{\boldsymbol{\Lambda}}_{i-1}^{\nu+1}$ is already known. Convergence of the SOR method can be guaranteed for positive definite matrices **G**. It is presumable that for dry friction the SORprox method has the same convergence criterion as the SOR method, meaning that the linear part (55) will determine convergence.

4.2 Remarks

An important aspect is the uniqueness of the solution of the non-linear equations (41) and (45). Consider a system with two unilateral contacts

$$\begin{aligned} \hat{\boldsymbol{\Lambda}}_1 &= prox_{\mathbb{R}_0^+}(-\frac{1}{G_{11}}(G_{12}\hat{\boldsymbol{\Lambda}}_2 + c_1)), \\ \hat{\boldsymbol{\Lambda}}_2 &= prox_{\mathbb{R}_0^+}(-\frac{1}{G_{22}}(G_{21}\hat{\boldsymbol{\Lambda}}_1 + c_2)). \end{aligned} \tag{56}$$

We assume that the contact force directions $\boldsymbol{w}_1$ and $\boldsymbol{w}_2$ are not independent. Thus, the matrix **G** is only positive semidefinite and not positive definite. If both $prox_{\mathbb{R}_0^+}$ in (56) do not project on the set $\mathbb{R}_0^+$, then the system (56) is underdetermined. The impulsive forces $\hat{\Lambda}$ cannot be determined uniquely. Note that in this case the motion is in many cases unique. It is therefore sufficient to find one possible solution for the impulsive forces. The non-unique impulsive forces represent an arbitrary inner tension state.
A non-regular matrix **G** does not necessarily induce non-unique impulsive forces. If, for example the $prox_{\mathbb{R}_0^+}$ from the second contact projects on its set $\mathbb{R}_0^+$, then the system (56) has a unique solution. Thus, if active contacts *can* act on the system in a way that it becomes underdetermined, then the matrix **G** is positive semidefinite and not positive definite. The matrix **G** is no longer invertible. The solution for the impulsive forces $\hat{\boldsymbol{\Lambda}}$ might be non-unique, but not necessarily, because the $prox_C$ functions may eliminate dependent rows in **G**. As a consequence of the non-regularity of **G**, one cannot expect general convergence of the SORprox method.

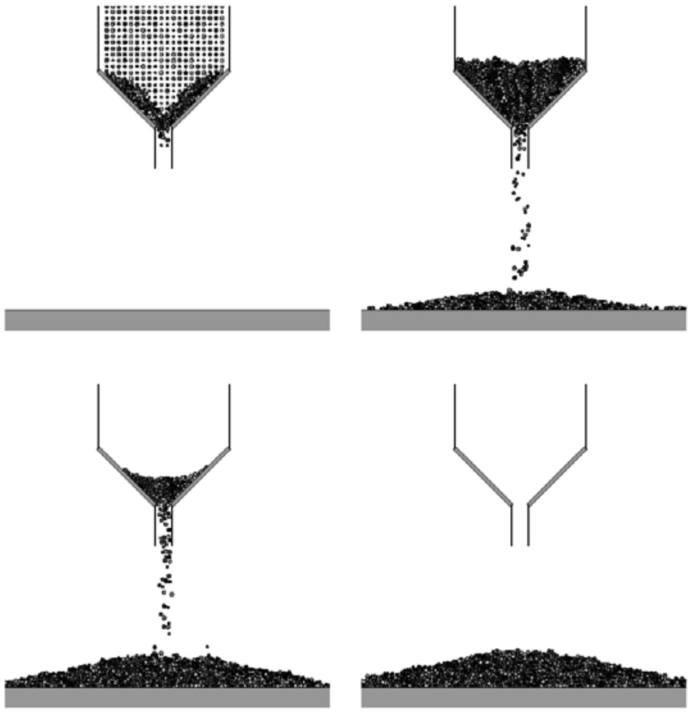

Fig. 1: A thousand balls falling through a funnel. Simulation of about half a million possible unilateral and friction contacts with Moreau's midpoint rule. The time step is fixed and small. The restitution and friction coefficients are: $\varepsilon_N = 0.3, \varepsilon_T = 0, \mu = 0.3$

5. Systems with many unilateral constraints

An example of a system with many unilateral contacts is shown in Figure 1. A thousand balls are thrown into a funnel. About half a million possible unilateral contacts and half a million possible friction contacts are modelled. It is nearly impossible to simulate this problem with event-driven methods because there are so many possible contact configurations and switching points. Note that event-driven methods cannot treat accumulative switching points. An example of an accumulation point is a ball falling on the ground. There will be infinitely many impacts in a finite time. An event-driven method detects all these infinitely many impacts and is therefore not able to pass the accumulation point.

A time-stepping method with a small fixed time step Δt is, however, suitable for such problems. The algorithm is very robust. An overall small time step is reasonable, because there are so many switching points. The size of the time step governs the resolution of the switching points. The result is not necessarily very accurate for an individual event, but we are more interested in the overall behaviour of the system than in the particular motion of one single ball.

6. System with few unilateral constraints

A time-stepping method with a fixed time step causes unessential computional effort if it is used to simulate systems with a few non-smooth constraints., The simulation has to be much more accurate, because not just the overall behaviour is of interest for such systems. Thus, the time step has to be chosen to be very small to resolve the switching points. The smooth part of the motion must also be integrated using this small time step, which is a waste of computing power.

We suggest a time-stepping method with variable step length. The system is described by the non-linear equations (41) and (45). If a unilateral contact is closed, then the $prox_{\mathbb{R}_0^+}$ function does not project onto the set $\mathbb{R}_0^+$. An open unilateral contact can be detected by a $prox_{\mathbb{R}_0^+}$ that does project on the set $\mathbb{R}_0^+$. In general, a switching point of a contact can be detected by analyzing the behaviour of the $prox_C$ function. If a $prox_C$ function projects on the set C during the time step t_1 and does not project on the set C in the following time step t_2, then a switching point is detected during the time steps t_1 and t_2. Of course, this is also the case if the $prox_C$ function changes from "non-project" to "project". The time step is reduced and the calculation is redone, until a minimal time step length is reached. Thus, the switching point is located with nested intervals. The procedure is shown in Figure 2.

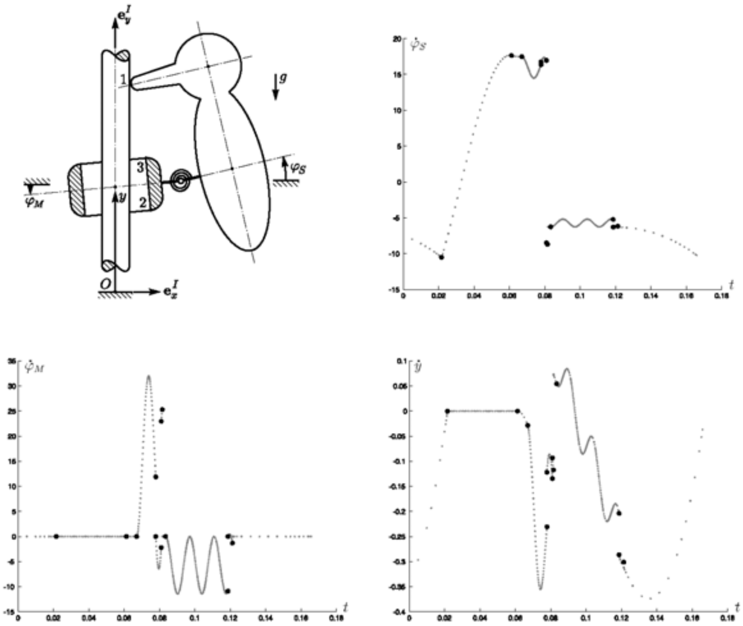

Fig. 2: Moreau's midpoint time-stepping method with variable step length tested on the woodpecker toy. The switching points are plotted as solid circles.

The Moreau midpoint time-stepping method with a variable step length has been tested on a woodpecker toy. The woodpecker toy consists of a pole, a sleeve with a hole that is slightly larger than the diameter of the pole, a spring, and the woodpecker. In operation, the woodpecker moves down the pole, performing some kind of pitching motion which is controlled by the sleeve [8]. The results are shown in Figure 3. The time step is reduced if the algorithm detects a switching point or if the integration error becomes too big. The switching points are plotted as solid dark circles.

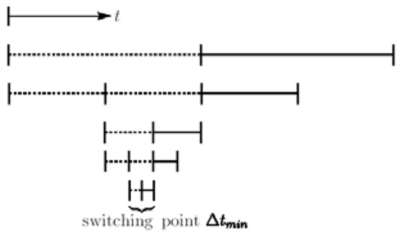

Fig. 3: Localization of a switching point. By comparing the behaviour of the $prox_C$ functions in consecutive time steps, a switching point can be detected. A time step for which the $prox_C$

function does "not project" is plotted dashed, a time step for which the $prox_C$ function "project" is plotted solid. If a switching point is localized, then the time step is reduced until a minimal time step Δt_{min} is reached.

7. Conclusions

Time-stepping methods are very suitable for simulation of mechanical systems with many non-smooth constraints, as demonstrated in section 5. Small systems with few constraints can be handled by time-stepping methods with a variable time step length. This enables an accurate localization of switching points. An example was given in section 6. In further development, extrapolation methods should be used to increase the order of the integration during the smooth parts of the motion.

The time-stepping inclusions which describe the discrete system are represented by non-linear equations. For each contact, one specific equation which describes the contact behaviour is formulated. By this approach, spatial friction situations can be treated in a compact way. The non-linear equations are solved iteratively by a Jacobi or a Gauss Seidel like procedure.

Contact		from	to
Friction contact	2	slip	stick
Friction contact	2	stick	slip
Unilateral contact	2	closed	open
Unilateral contact	3	open	closed
Unilateral contact	3	closed	open
Unilateral contact	1	open	closed
Unilateral contact	1	closed	open
Unilateral contact	3	open	closed
Unilateral contact	3	closed	open
Unilateral contact	2	open	closed
Unilateral contact	2	closed	open
Unilateral contact	2	open	closed

Table 1: The different switching incidents of the woodpecker toy. The corresponding switching points are shown in Figure 2

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EUROMECH Fellows: Nomination Procedure

The EUROMECH Council has pleasure in announcing the introduction of the category of **EUROMECH Fellow**, starting in 2005. The status of Fellow is awarded to members who have contributed significantly to the advancement of mechanics and related fields. This may be through their original research and publications, or their innovative contributions in the application of mechanics and technological developments, or through distinguished contribution to the discipline in other ways.

Election to the status of Fellow of EUROMECH, the European Mechanics Society, will take place in the year of the appropriate EUROMECH Conference, EFMC or ESMC respectively. The number of fellows is limited in total (fluids and solids together) to no more than one-half of one percent of the then current membership of the Society.

Nomination conditions:

- The nomination is made by **two sponsors** who must be members of the Society;
- Successful nominees must be members of the Society;
- Each nomination packet must contain a completed Nomination Form, signed by the two sponsors, and no more than four supporting letters (including the two from the sponsors).

Nomination Process:

- The nomination packet (nomination form and supporting letters) must be submitted before 15 January in the year of election to Fellow (the year of the respective EFMC or ESMC);
- Nominations will be reviewed before the end of February by the EUROMECH Fellow Committee;
- Final approval will be given by the EUROMECH Council during its meeting in the year of election to Fellow;
- Notification of newly elected Fellows will be made in May following the Council meeting;
- The Fellow award ceremony will take place during the EFMC or ESMC as appropriate.

Required documents and how to submit nominations:

Nomination packets need to be sent before the deadline of 15 January in the year of the respective EFMC or ESMC to the President of the Society. Information can be obtained from the EUROMECH web page **www.euromech.org** and the Newsletter. Nomination Forms can also be obtained from the web page or can be requested from the Secretary-General.

NOMINATION FORM FOR FELLOW

NAME OF NOMINEE:……….………………………………………………………….…

OFFICE ADDRESS:……………………………………………………………………....

……………………………………………………………………………………………….

……………………………………………………………………………………………….

EMAIL ADDRESS:…………………………………………………………………………..

FIELD OF RESEARCH: ……………………………………………………………………..

Fluids: ☐ Solids: ☐

NAME OF SPONSOR 1: ..………...…………………………………………………………

OFFICE ADDRESS:…………………………………………………………………………..…

……………………………………………………………………………………………….

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EMAIL ADDRESS:………………………………………………………………………………

SIGNATURE & DATE: ………………………………………………………………………

NAME OF SPONSOR 2: ..………...………………………………………………………….

OFFICE ADDRESS:………..………….……………………………………….………..….

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EMAIL ADDRESS:……………………………………………………………………………….

SIGNATURE & DATE: ……………………………………………………………………….

SUPPORTING DATA

- Suggested Citation to appear on the Fellowship Certificate (30 words maximum);
- Supporting Paragraph enlarging on the Citation, indicating the Originality and Significance of the Contributions cited (limit 250 words);
- Nominee's most Significant Principal Publications (list at most 8);
- NOMINEE'S OTHER CONTRIBUTIONS (invited talks, patents, professional service, teaching etc. List at most 10);
- NOMINEE'S ACADEMIC BACKGROUND (University Degrees, year awarded, major field);
- NOMINEE'S EMPLOYMENT BACKGROUND (position held, employed by, duties, dates).

SPONSORS DATA

Each sponsor (there are two sponsors) should sign the nomination form, attach a letter of recommendation and provide the following information:

- Sponsor's name;
- Professional address;
- Email address;
- Eponsor's signature/date.

ADDITIONAL INFORMATION

Supporting letters (no more than four including the two of the sponsors).

TRANSMISSION

Send the whole nomination packet to:
Professor Patrick Huerre
President EUROMECH
Laboratoire d'Hydrodynamique, École Polytechnique
91128 Palaiseau Cedex, France
E-mail: huerre@ladhyx.polytechnique.fr

EUROMECH Prizes: Nomination Procedure

Fluid Mechanics Prize
Solid Mechanics prize

Regulations and Call for Nominations

The *Fluid Mechanics Prize* and the *Solid Mechanics Prize* of EUROMECH, the European Mechanics Society, shall be awarded on the occasions of Fluid and Solid conferences for outstanding and fundamental research accomplishments in Mechanics.
Each prize consists of 5000 Euros. The recipient is invited to give a Prize Lecture at one of the European Fluid or Solid Mechanics Conferences.

Nomination Guidelines:
A nomination may be submitted by any member of the Mechanics community. Eligible candidates should have undertaken a significant proportion of their scientific career in Europe. Self-nominations cannot be accepted.
The nomination documents should include the following items:

- A presentation letter summarizing the contributions and achievements of the nominee in support of his/her nomination for the Prize,;
- A curriculum vitae of the nominee,
- A list of the nominee's publications,
- At least two letters of recommendation.

Five copies of the complete nomination package should be sent to the Chair of the appropriate Prize Committee, as announced in the EUROMECH Newsletter and on the Society's Web site www.euromech.org Nominations will remain active for two selection campaigns.

Prize committees
For each prize, a Prize Committee, with a Chair and four additional members shall be appointed by the EUROMECH Council for a period of three years. The Chair and the four additional members may be re-appointed once. The committee shall select a recipient from the nominations. The final decision is made by the EUROMECH Council.

Fluid Mechanics Prize

The nomination deadline for the Fluid Mechanics prize is **15 January in the year of the Fluid Mechanics Conference**. The members of the *Fluid Mechanics Prize and Fellowship Committee* are:

- I.D. Abrahams
- H.H. Fernholz (Chair)
- P. Huerre
- D. Lohse
- W. Schröder

Chairman's address

Professor H.H. Fernholz
Hermann-Föttinger-Institut für Strömungsmechanik
Technische Universität Berlin
Müller-Breslau Strasse 8
D - 10623 Berlin, Germany
Tel. : +49-30-3142-2693
Fax : +49-30-3142-1101
Email: fernholz@pi.tu-berlin.de

Solid Mechanics Prize

The nomination deadline for the Solid Mechanics prize is **15 January in the year of the Solid Mechanics Conference**. The members of the *Solid Mechanics Prize and Fellowship Committee* are:

- A. Benallal
- I. Goryacheva
- H.M. Jensen
- F.G. Rammerstorfer (Chair)
- B. A. Schrefler

Chairman's address

Professor F.G. Rammerstorfer
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Tel. : +43-1-58801-31700
Fax : +43-1-58801-31799
Email: ra@ilsb.tuwien.ac.at

EUROMECH Conferences in 2006 and 2007

The general purpose of EUROMECH conferences is to provide opportunities for scientists and engineers from all over Europe to meet and to discuss current research. Europe is a very compact region, well provided with conference facilities, and this makes it feasible to hold inexpensive meetings.
The fact that the EUROMECH Conferences are organized by Europeans primarily for the benefit of Europeans should be kept in mind. Qualified scientists from any country are of course welcome as participants, but the need to improve communications within Europe is relevant to the scientific programme and to the choice of leading speakers.

A EUROMECH Conference on a broad subject, such as the ESMC or the EFMC, is not a gathering of specialists all having the same research interests, and much of the communication which takes place is necessarily more in the nature of the imparting of information than the exchange of the latest ideas. A participant should leave a Conference knowing more and understanding more than on arrival, and much of that gain may not be directly related to the scientist's current research. It is very important therefore that the speakers at a Conference should have the ability to explain ideas in a clear and interesting manner, and should select and prepare their material with this expository purpose in mind.

EMMC9
9th EUROMECH-MÉCAMAT Conference
DATES: 9-12 May 2006
LOCATION: Moret Sur Loing, France
CONTACT: Jacques Besson, Ecole Nationale des Mines de Paris, France
E-MAIL: jacques.besson@ensmp.fr
WEBSITE: http://www.mat.ensmp.fr/EMMC9/

EFMC6
6th European Fluid Mechanics Conference
DATES: 26 – 30 June 2006
LOCATION: KTH, Stockholm, Sweden
CONTACT: efmc6@mech.kth.se
WEBSITE: http://www2.mech.kth.se/efmc6/

ESMC6

6th European Solid Mechanics Conference

DATES: 28 August - 1 September 2006

LOCATION: Budapest University of Technology and Economics (BUTE), Budapest, Hungary

CONTACT: Prof. Gábor Stépán – chairman; Dr. Ádám Kovács – secretary, BUTE Department of Applied Mechanics, 1521 Budapest, P.O. Box 91

Fax: +36 1 463 3471

E-MAIL: esmc2006@mm.bme.hu

WEBSITE: http://esmc2006.mm.bme.hu

EMMC10

10th EUROMECH-MÉCAMAT Conference

DATES: Autumn 2007

LOCATION: Warsaw, Poland

CONTACT: W.K.Nowacki, IPPT-Polish Academy of Sciences

E-MAIL: wnowacki@ippt.gov.pl

WEBSITE:

EETC11

11th EUROMECH European Turbulence Conference

DATES: 25 – 28 June 2007

LOCATION: Faculty of Engineering of the University of Porto
Porto, Portugal

CONTACT: etc11@fe.up.pt.

WEBSITE: http://www.fe.up.pt/etc11

Third Announcement

9th EUROMECH-MÉCAMAT Conference – EMMC9

Local approach to fracture
9–12 May 2006
Moret Sur Loing, France
http://www.mat.ensmp.fr/EMMC9

This conference is the ninth in a new series of European Mechanics of Materials Conferences to be held under the auspices of the European Mechanics Society and the French Society for Mechanics of Materials. These EUROMECH-MÉCAMAT Conferences continue the tradition of past MECAMAT International Seminars.

The purpose of the meeting is to bring together specialists in experimental, modeling and simulation techniques devoted to the analysis of macroscopic fracture based on the description of microscopic mechanisms. Various aspects such as the following are concerned:

- Ductile fracture of metals;
- Brittle fracture of metals;
- Ductile to brittle transition;
- Creep rupture;
- Fracture of polymers and elastomers;
- Experimental fracture mechanics;
- Constitutive models;
- Micromechanical modelling;
- Scale effects;
- Computational fracture mechanics;
- Load history effect (including WPS).

The Conference will include presentation in oral or poster form. Abstracts of about 500 words were invited before September 1, 2005. They were expected to contain the title of the communication, full names and addresses of the authors, objectives of the study, methods employed, and the most significant results. Submission of the abstract by e-mail (PDF format) was recommended. Notification of acceptance was sent to authors by December 15, 2005. A six-page paper was due before March 15, 2006. Copies of all these papers will be available in book-form as pre-prints of the proceedings, on the first day of the conference. Full size refereed papers will be published later as a special issue of Engineering Fracture Mechanics. These papers should be sent before December 31, 2006. Instructions concerning the format of the papers will be available on the conference web page.

Co-Chairmen : J. Besson, D. Steglich, D. Moinereau

Conference Secretariat: V Diamantino

E-mail: emmc9@mat.ensmp.fr

Second Announcement
6th European Fluid Mechanics Conference – EFMC6

26–30 June 2006
KTH, Stockholm, Sweden
http://www2.mech.kth.se/efmc6/

The 6th European Fluid Mechanics Conference, organized by EUROMECH (the European Mechanics Society), will take place at KTH, Royal Institute of Technology, Stockholm, 26–30 June 2006.

The conference aims to provide an international forum for the exchange of information on all aspects of fluid mechanics, including instability and transition, turbulence, multiphase and non-Newtonian flows, bio-fluid mechanics, reacting and compressible flows, numerical and experimental methods, geophysical flows etc., as well as all types of fluid mechanics applications.

Eight prominent scientists have already accepted the invitation to give keynote lectures in their respective fields of expertise. These are (in alphabetical order):

- Gustav Amberg (Sweden) - Fluid mechanics of phase change;
- Stephan Fauve (France) - Generation of magnetic fields by turbulent flows of liquid metals;
- Sascha Hilgenfeldt (USA) - The power of bubbles: Unconventional micro-fluidics;
- Rich Kerswell (UK) - Progress in Reynolds' problem: transition to turbulence in pipe flow;
- Hilary Ockendon (UK) - Continuum models in industrial applications;
- Norbert Peters (Germany) - Combustion;
- Jens Sørensen (Denmark) - Wind turbine wake structures;
- Sandra M. Troian (USA) - Microfluidic actuation and sensing for open architecture systems: Fundamentals to applications.

In addition to these 8 invited lectures and one lecture by the EUROMECH Fluid Mechanics Prize winner (not yet selected), contributions were solicited from the worldwide fluid mechanics research community. The paper selection was made by the EUROMECH Fluid Mechanics Conference Committee on the basis of extended abstracts submitted before the end of 2005. For further information please visit the conference website.

Enquiries should be sent to **efmc6@mech.kth.se**

Second Announcement
6th European Solid Mechanics Conference – ESMC6

28 August - 1 September 2006
Budapest University of Technology and Economics
Budapest, Hungary
http://esmc2006.mm.bme.hu

The 6th European Solid Mechanics Conference (ESMC 2006) will be held at the Budapest University of Technology and Economics (BME), Hungary, 28 August - 1 September, 2006 under the auspices of the European Mechanics Society (EUROMECH).

The conference aims to provide an international forum for the exchange of information on all aspects of solid mechanics, including Continuum Mechanics
(*General theories, Elasticity, Plasticity, Multi-field problems*), Materials Mechanics (*Damage and fracture, Viscoelastic materials and systems, Composites, Contact problems*), Structural Mechanics (*Beam structures, Plates and shells, Stability, Structural optimization*), Dynamics (*Kinematics, Multibody systems, Vibrations, Nonlinear dynamics*), Computational and experimental methods

The following scientists have already accepted the invitation to give keynote lectures in their respective fields of expertise:

- Werner Schielen (Universität Stuttgart, Germany) - Dynamics
- Gerhard A. Holzapfel (KTH Stockholm, Sweden) - Biomechanics
- Jean-Jacques Marigo (Université Paris 13, France) - Fatigue/Fracture
- Paul van Houtte (KU Leuven, Belgium) - Plasticity/Damage
- Alexander B. Movchan (University of Liverpool, UK) - Stability
- Nikita Morozov (St. Petersburg State University, Russia) - Micromechanics
- Dick van Campen (Eindhoven, The Netherlands) - Nonlinear Dynamics

In addition to these invited lectures and one lecture by the EUROMECH Solid Mechanics Prize winner (not yet announced), contributions were solicited from the worldwide solid mechanics research community. The paper selection was made by the EUROMECH Solid Mechanics Conference Committee on the basis of extended abstracts submitted before the end of 2005.For further information please visit the conference website.

Enquiries should be sent to **esmc2006@mm.bme.hu**

Second Announcement and Call for Papers
11th EUROMECH European Turbulence Conference
ETC11

25–28 June 2007
Faculty of Engineering of the University of Porto, Portugal
http://www.fe.up.pt/etc11

The 11th EUROMECH European Turbulence Conference (ETC11), organized by the EUROMECH - European Mechanics Society, will take place at the Faculty of Engineering of the University of Porto (FEUP) in Porto, Portugal.

The conference aims to provide an international forum for exchange of information on most fundamental aspects of turbulent flows, including instability and transition, intermittency and scaling, vortex dynamics and structure formation, transport and mixing, turbulence in multiphase and non-Newtonian flows, reacting and compressible turbulence, acoustics, control, geophysical and astrophysical turbulence, and large-eddy simulations and related techniques, MHD turbulence and atmospheric turbulence.

Following the established tradition, the conference programme will comprise 8 invited talks (two per day), selected papers and poster sessions.

Contributions are solicited from the worldwide turbulence research community.

The paper selection will be made by the EUROMECH Turbulence Conference Committee on the basis of two-page abstracts submitted via the conference webpage, at **www.fe.up.pt/etc11** by 6 October 2006.

All accepted papers and posters will appear in a conference proceedings to be distributed among the participants. A smaller set of papers may be published after the conference in a special issue of a scientific journal. For further information and updates please visit the conference website or contact the organizers at **etc11@fe.up.pt**.

EUROMECH Colloquia in 2006 and 2007

EUROMECH Colloquia are informal meetings on specialized research topics. Participation is restricted to a small number of research workers actively engaged in the field of each Colloquium. The organization of each Colloquium, including the selection of participants for invitation, is entrusted to a Chairman. Proceedings are not normally published. Those who are interested in taking part in a Colloquium should write to the appropriate Chairman. Number, Title, Chairperson or Co-chairperson, Dates and Location for each Colloquium in 2006, and preliminary information for some Colloquia in 2007, are given below.

EUROMECH Colloquia in 2006

470. Recent Development in Magnetic Fluid Research
Chairman: Dr. Stefan Odenbach, ZARM, University of Bremen, Am Fallturm, D-28359, Bremen, Germany
Phone: +49-(0)421 2184 785, Fax: +49-(0)421 2182 521
E-mail: odenbach@zarm.uni-bremen.de
Co-chairman: Prof. Dr. Elmars Blums, Institute of Physics, University of Latvia, Salaspils, Latvia
Euromech contact person: Prof. Wolfgang Schröder
Date and location: 27 February-1 March 2006, Dresden, Germany
http://www.zarm.uni-bremen.de/2forschung/ferro/conferences/Euromech06/euromech_colloquium_470.htm

475. Fluid Dynamics in High Magnetic Fields
Chairman: Prof. A. Thess, Department of Mechanical Engineering, Ilmenau, University of Technology, P.O. Box 100 565, D-98684, Ilmenau, Germany
Phone: +49-(0)3677 69 2445, Fax: +49-(0)3677 69 1281
E-mail: thess@tu-ilmenau.de
Euromech contact person: Prof. Jorge Ambrosio
Date and location:1-3 March 2006, Ilmenau, University of Technology, Germany
http://www4.tu-ilmenau.de/mfd/euromech2006.html

476. Real-time Simulation and Virtual Reality Applications of Multibody Systems
Chairman: Prof. J. Cuadrado, Escuela Politecnica Superior, Universidad de La Coruña, Mendizabal s/n 15403 Ferrol, Spain
Phone: +34-9813 37400 ext. 3873, Fax: +34-9813 37410
E-mail: javicuad@cdf.udc.es

Co-chairman: Prof. W. Schiehlen, Institute B of Mechanics, University of Stuttgart, Germany
Euromech contact person: Prof. Jorge Ambrosio
Date and location: 13-16 March 2006, Ferrol, Spain
http://lim.ii.udc.es/events/euromech476/

477. Particle-laden Flow. From Geophysical to Kolmogorov Scales
Chairman: Prof. B.J. Geurts, Mathematical Sciences, University of Twente, P.O. Box 217, 7500 AE Enschede, The Netherlands
Phone: +31-(0)48 94125, Fax: +31-(0)48 94833
E-mail: b.j.geurts@utwente.nl
Co-chairman: Prof. Dr. H.J.H. Clercx, Eindhoven University of Technology, and Dr. W.S.J. Uijttewaal, Delft University of Technology, The Netherlands.
Euromech contact person: Prof. Detlef Lohse
Date and location: 21-23 June 2006, University of Twente, The Netherlands
http://wwwhome.math.utwente.nl/~geurtsbj/workshops/euromech_477/

478 Non-equilibrium Dynamical Phenomena in Inhomogeneous Solids
Chairman: Prof. Juri Engelbrecht, Centre for Nonlinear Studies, Institute of Cybernetics, Tallinn University of Technology, Akadeemia tee 21, 12618 Tallinn, Estonia
E-mail: je@ioc.ee
Co-chairman: Prof. Gerard A. Maugin
Euromech contact person: Prof. Ahmed Benallal
Date and location: 13-16 June 2006, Tallinn University of Technology, Estonia
http://greta.cs.ioc.ee/~berez/euromech478/

479. Numerical Simulation of Multiphase Flow with Deformable Interfaces
Chairman: Prof. Bendiks Jan Boersma, Laboratory for Aero and Hydrodynamics, Mekelweg 2, 2628 CD Delft, The Netherlands
E-mail: b.j.boersma@wbmt.tudelft.nl
Co-chairman:
Euromech contact person: Prof. Detlef Lohse
Date and location: 14-16 August 2006, "De Pier", Scheveningen, The Netherlands
http://www.ahd.tudelft.nl/~emil/euromech/

480. High Rayleigh Number Convection
Chairman: Prof. Detlef Lohse, University of Twente, The Netherlands
E-mail: lohse@tnw.utwente.nl
Co-chairman:
Euromech contact person: Prof. Hans H. Fernholz
Date and location: 4-8 September 2006, Trieste

484. Wave Mechanics and Stability of Long Flexible Structures Subjected to Moving Loads and Flows
Chairman: Prof. Andrei V. Metrikine, TU Delft, Faculty of Civil Engineering and Geosciences, PO Box 5048, 2600 GA, Delft, The Netherlands
E-mail: a.metrikine@citg.tudelft.nl
Co-chairman: Prof. L. Fryba and Prof. E. de Langre
Euromech contact person: Prof. Irina Goryacheva
Date and location: 19-22 September 2006, TU Delft, The Netherlands
http://www.euromech484.nl/

485. Durability of Composite Materials
Chairman: Prof. Antonio Torres Marques, Departamento de Engenharia Mecanica e Gestao Industrial, Faculdade de Engenharia de Universidade do Porto, Rua Dr. Roberto Frias, 4200-465 Porto, Portugal
E-mail: marques@fe.up.pt
Co-chairman: Prof. Albert Cardon
Euromech contact person: Prof. Jorge Ambrosio
Date and location: 18-21 July 2006, Hotel Santa Luzia, Viana do Castelo, Portugal

486. Deformation and Fracture Processes in Paper and Wood Materials
Chairman: Prof. Per A. Gradin, Department of Solid Mechanics, Mid Sweden University, Sweden
E-mail: per.gradin@miun.se
Co-chairman: Prof. Tetsu Uesaka
Euromech contact person: Prof. Ahmed Benallal
Date and location: 12-15 June 2006, University of Sundsvall, Sweden
http://www.miun.se/fscn/euromech486

487. Structure Sensitive Mechanics of Polymer Materials-Physical and Mechanical Aspects
Chairman: Prof. Yves Remond, Institut de Mecanique des Fluides et de Solids UMR 7507 ULP, 67000 Strasbourg, France
E-mail: remond@imsfs.u-strabg.fr
Co-chairman: Prof. Stanislav Patlazhan
Euromech contact person: Prof. Ahmed Benallal
Date and location: 10-13 October 2006, Strasbourg, France

483. Non-linear Vibrations of Structures

Chairman: Prof. P.L. Ribeiro, IDMEC/DEMEGI, Faculdade de Engenharia
Universidade do Porto, Rua Doutor Roberto Frias, 4200-465 Porto, Portugal
Phone: +351 22 508 1713; Fax: +351 22 508 1445
E-mail: pmleal@fe.up.pt
Co-chairman: *to be decided*
Euromech contact person: Prof. J. Ambrosio
Date and location: 5-7 September 2007, University of Porto, Portugal

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ERRATA CORRIGE (NL28)

Colloquium "Thermomechanics of Non-Homogeneous Structures", was incorrectly inserted among the programmed Colloquia for 2006. The Colloquium will not take place.

EUROMECH Colloquium 464b

"Wind Energy"

4-7 October 2005, Oldenbourg, Germany

Chairperson: Prof. Dr. Joachim Peinke
Co-Chairperson: Prof. Dr.-Ing. Peter Schaumann

In order to tackle the problems and reservations in the interdisciplinary community of wind energy scientists, ForWind, the centre for wind energy research at the Universities of Oldenburg and Hanover, arranged the EUROMECH Colloquium 464b - Wind Energy, which was held at the Carl von Ossietzky University of Oldenburg. The central aim of this colloquium was to bring together the previously separate communities of wind energy scientists and those who do fundamental research in mechanics. Wind energy is a challenging task in mechanics, but well focused research will find important applications in wind energy conversion.

More than 80 experts from 16 countries from all over the world attended the meeting, confirming the need and the concept of this colloquium. The 46 oral and 28 poster presentations were grouped in the following topics:

- Wind climate and wind field;
- Gusts, extreme events and turbulence;
- Power production and fluctuations;
- Rotor aerodynamics;
- Wake effects;
- Materials, fatigue and structural health monitoring.

Phenomenological approaches based mainly on experimental and empirical data as well as advanced fundamental mathematical scientific approaches were presented. These spanned the range from reliability investigations to new CFD codes for turbulence models and Levy statistics of wind fluctuations.

During the meeting, a clearer understanding emerged about the fundamental research that is essential for future developments in wind energy. There is a need for:

- A better understanding of the marine atmospheric boundary layer, ranging from mean wind profiles to highly resolved turbulence properties. These questions need further measurements as well as appropriate numerical simulationand physical models. A proper and detailed wind field description is indispensable for correct power and load modelling;

- CFD simulations for wind profiles and rotor aerodynamics with advanced methods (aero elastic codes) that include experimental details on the dynamic stall phenomenon as well as near and far field rotor wakes;
- A site independent description of wind power production taking into account turbulence induced fluctuations;
- Recognition of the material loads on different components in a WEC and potential fatigue due to the high number of cycles within such complex machines;
- An advanced numerical hybrid model for a 3D simulation of a WEC, taking into account wind and wave loads as well as all effects of operation in a so called 'integrated' model.

Many intensive discussions on these and other topics took place between participants from different disciplines during coffee and lunch breaks and also during social events, which included a city reception and the colloquium dinner. Positive feedback about the meeting's scientific and social aspects encouraged the scientific committee to decide to have annual follow-up meetings alternately organized by DUWIND, Risø and ForWind. Participants agreed that the scientific interdisciplinary cooperation and international collaboration should be intensified and are grateful to EUROMECH for making this colloquium possible.

EUROMECH Colloquium 469

"LES of Complex Flows"

6-8 October 2005, Dresden, Germany

Chairperson: Prof. Nikolaus Adams
Co-Chairperson: Prof. Michael Manhart

The Colloquium was held to explore the application of Large Eddy Simulation (LES) to complex flows and the resulting need for development of appropriate numerical methods to cope with these challenges.

The importance of the colloquium was reflected in the participation of 50 scientists from research institutions all over the world (Europe, 43; USA, 5; Japan, 2). The relevance of the colloquium topic was also emphasised through the presentations and participation of industrial software developers and users.

Due to the strong response to the call for abstracts, the programme committee organised a poster session with 10 poster presentations, to accommodate the broad spectrum of contributed papers. The conference presentations were scheduled at 25 minute intervals, in order to allow ample time for scientific discussion of the presented results. 32 oral presentations were given during the two and a half days of the colloquium. All contributions were collected in a book of abstracts that was given to participants. There were also three invited talks by senior researchers in the field of LES, who have made seminal contributions to the subject.

The topics covered by the participants included:

- The application of LES to complex geometric configurations;
- The coupling of LES with other numerical methods (i.e. RANS);
- LES in compressible flows, wall modelling and combustion related problems;
- Development of improved numerical methods.

The presented work will be made available to the broader scientific community through the publication of a special issue of "Theoretical and Computational Fluid Dynamics".

EUROMECH Colloquium 471

"Turbulent Convection in Passenger Compartments"

13-14 October, 2004, German Aerospace Centre, Göttingen

Chairperson: Prof. Claus Wagner
Co-Chairperson: Prof. Andre Thess

The objective of the colloquium was to discuss physical phenomena that are relevant to turbulent mixed convection and to allow for an exchange of ideas on numerical methods and experimental techniques needed to investigate flow and temperature distributions in passenger compartments and their effect on the passenger. Forced and buoyant convection in the passenger compartments of automobiles, trucks, trains and in aircraft cabins controls the thermal comfort of passengers, the air quality and the transport of pollutants and pathogens. There is a high industrial demand for a better understanding of turbulent convection in passenger compartments and for reliable computational methods, accurate measurement techniques and proper test facilities.

About 30 participants from Europe and the USA and attended the colloquium. There were 16 presentations, including two keynote lectures. These were given by Paul F. Linden from the University of California in San Diego, USA entitled "Ventilation flow patterns, their control and the consequences for comfort in enclosed spaces" and by Alexander Orellano from Bombardier Transportation in Henningsdorf, Germany, entitled "Towards virtual testing of Thermal Comfort".

On the first day, 8 lecturers presented fundamental aspects of ventilation in enclosed spaces, and the numerical methods and experimental techniques used. In summary:

- Fundamental investigations are conducted using techniques like Direct Numerical Simulations and Large-Eddy Simulation (LES) in simplified computational domains;
- Computational Fluid Dynamics (CFD) methods which solve the Reynolds-averaged Navier-Stokes (RANS) equation together with statistical turbulence models are used to predict the airflow in more complex compartments;
- Experimental investigations utilizing sophisticated laser based measurement techniques like Particle Image Velocimetry (PIV) may be used to verify computer based forecasts;
- Participants experienced a live PIV measurement in DLR's cabin model of the long-distance aircraft A380, which was constructed to perform

flow and acoustic measurements in a realistic environment.

On the second day, another 8 lectures were given by representatives of the aircraft manufacturers Boeing and Airbus, the rail transportation industry and researchers from universities and other research institutions . These focused on applied work to improve the thermal comfort in real aircraft cabins and trains. It was shown that complex flow phenomena have to be studied numerically and experimentally in order to fulfil challenging thermal requirements for aircraft cabin and train compartments and to understand how pathogens and other contaminants are transported within these compartments. In the final discussion, participants agreed that the colloquium had been very successful and that they were interested in meeting again in the future. All thanked Euromech for supporting the meeting and for making it possible.

EUROMECH Colloquium 473

"Fracture of Composite Materials"

27-29 October, 2005, Porto, Portugal

Chairperson: Prof. António Torres Marques
Co-Chairpersons: Claude Bathias, Paulo Tavares de Castro, Alfredo Balacó de Morais

44 papers were selected for presentation at the colloquium. Accepted abstracts were made available to the community through the Colloquium web page. Authors were offered the opportunity to provide full-length papers, in order to support their presentations and for submission to "Engineering Fracture Mechanics". The colloquium took place in the conference room of Hotel Infante Sagres. The average number of participants in the room was 38. A book of abstracts was distributed to all participants during registration.

There were 10 sessions during the 2½ days of the colloquium. Each morning and afternoon had two sessions separated by a 30 minute coffee-break. Each session included 4-5 presentations of 20 minutes each, followed by discussion coordinated by the session chairman, selected from the members of the Scientific Committee and senior researchers attending the meeting. The first and final presentations were invited lectures of 30 minutes each. At the end of the Colloquium, a one-hour session concerning future research activities in Fracture Mechanics of Composites took place. Several ideas were discussed and future interaction was agreed.

Each author of a paper at the colloquium was asked to submit a revised version for possible publication in "Engineering Fracture Mechanics".. A special edition of the journal is expected to be published in 2006.

Grants to young participants
The following participants were offered a €600 grant, according to EUROMECH regulations:

- Edgars Sparnins, Institute of Polymer Mechanics, University of Latvia, Aizkraukles iela 23, LV-1006 Rīga, Latvia, "Statistical features of the transverse ply cracking in cross-ply laminates"; e-mail: sparns@pmi.lv;
- Fracinsco Evangelista-Junior, Department of Transportation Engineering (DET), Federal University of Ceará (UFC), Fortaleza, Brazil, "Cohesive Crack Modeling and Multi-Scale Analysis of Asphaltic Matrix Composites"; e-mail: fejr@det.ufc.br;
- Masahito Ueda, Tokyo Institute of Technology, 2-12-1 O-okayama Meguro, Tokyo, 1528552, Japan, "Delamination monitoring of CFRP laminate using two-stage electric potential change method with equivalent electric conductivity"; e-mail: mueda@ginza.mes.titech.ac.jp.

EUROMECH Colloquium 474

"Material Instabilities in Coupled Problems"

30-31 August, 2005, Troyes, France

Chairperson: Prof. A. Benallal
Co-Chairperson: Prof. D. Bigoni

Euromech Colloquium 474 " Material Instabilities in Coupled Problems" was organized within the framework of the French Congress of Mechanics (Congrès Français de Mécanique-CFM) following discussion between Patrick Huerre (President of Euromech) and the CFM organizing and scientific committees. This was the first time that a Euromech Colloquium had been organized in such a way in collaboration with another institution. There were 23 lectures of 30 minutes duration each, given by invited scientists from Europe and overseas.

Material instabilities are an important topic in the understanding of localisation and rupture phenomena. While there has been extensive work in this field of solid mechanics, there is a need to include in the localisation process various physical phenomena and their coupling to the mechanical behaviour of the solid. The main objective of the colloquium was to address the various coupling mechanisms and their effects in the developments and growth of instabilities, which lead to localisation and failure in solids in the plastic and viscoplastic regimes. The following topics were considered in the colloquium:

- Thermo-mechanical couplings;
- Hydro-mechanical couplings;
- Chemo-mechanical couplings;
- Bio-mechanical couplings;
- Microstructural and phase transformation couplings.

Other devices, such as electro-mechanical couplings, were noted during the colloquium. The colloquium covered theoretical, experimental and numerical investigations. Materials discussed included:

- Composites;
- Metallic alloys;
- Polymers;
- Shape memory alloys;
- Geomaterials;
- Brain tissues;
- Concretes..

Instabilities analysed included:

- Catastrophic landslides;
- Earthquakes;
- Failure of cementitious materials;
- Sheet metal forming and machining;
- Corrosion;
- Crack propagation due to oxidation.

.The talks clearly showed the complexity of the topics and especially the modelling of coupled phenomena, on which the development of material instabilities are based. Further, it was often stressed how these material instabilities are sensitive to modelling. Experimental results are becoming available, but there is a need for additional experiments, both to improve the modelling and to validate the theoretical criteria that are used to predict the emergence and development of material instabilities.

Even though it was impossible to cover all types of coupling in a two-day meeting, it was felt that electro-magnetic couplings were under-represented at the colloquium. Also, it was suggested that various practical situations exist where more than two coupled phenomena are present.

The similarity of tools used in different situations, and the opportunities for cross-fertilisation, were noted as important reasons for further colloquia. It was agreed in the conclusion to the colloquium that other meetings are needed to discuss all aspects of coupled phenomena. It is recommended that another Euromech colloquium in the field should be held by 2008, in addition to other conferences on the topic that are already planned.

EUROMECH Colloquium 476

"Real-time simulation and virtual reality applications of multibody systems"

13-16 March, 2006, Ferrol, Spain

Chairperson: Prof. J. Cuadrado
Co-Chairperson: Prof. W. Schiehlen

EUROMECH Colloquium 476 "Real-time simulation and virtual reality applications of multibody systems" was held at the University of La Coruna, Ferrol, Spain. The colloquium falls within the general field of multibody dynamics. It was attended by 58 participants from 17 countries, including the USA, Canada, Japan, South Korea and India. Although more than half of the delegates were multibody dynamicists, specialists from other fields like robotics, haptics, vehicle dynamics, computer science, virtual reality and biomechanics were also present. 40 lectures were delivered, organized into 13 sessions.

Real-time simulation of the dynamics of multibody systems is generally required for applications with human- and/or hardware-in-the-loop. Three main lectures were delivered on virtual reality technologies, haptic interfaces, and hardware-in-the-loop applications, respectively.

A relevant part of the presentations was devoted to the development of efficient formulations and related numerical issues, which are key aspects when seeking real-time performance in the simulation of multibody system dynamics. All the problem stages, i.e. modelling, formulation of the equations of motion, numerical integration and implementation, were addressed, since all of them have significant influence on the performance of the simulation. In the case of flexible bodies, attention was also paid to model reduction techniques.

The huge increase in capabilities that virtual reality provides to multibody simulation through its two main features, immersion and interaction, was addressed too in the Colloquium. Several applications recently developed were shown, and a whole session was devoted to car and machinery simulators. Two sessions dealt with the contact problem, always a relevant issue in multibody dynamics, but connected in this case to the use of haptic systems, hardware devices which introduce the sense of touch in the simulation, and enable the user to exert/perceive forces. Two more sessions were dedicated to control problems, since they can often benefit from the real-time simulation of the mechanical systems to be controlled, either through their simplified or detailed models.

Participants had the opportunity to discuss the latest results, with work still in progress being presented at the meeting. Interaction was encouraged during the colloquium by accommodation in a hotel within a ten-minute walk from the venue, and by the daily social program.

The colloquium showed that the multibody dynamics community is active, and that new topics are consolidating inside it, in close cooperation with several neighbouring disciplines. It has been planned that selected contributions will be published as papers in a special issue of "Multibody System Dynamics ".

Objectives of EUROMECH, the European Mechanics Society

The Society is an international, non-governmental, non-profit, scientific organisation, founded in 1993. The objective of the Society is to engage in all activities intended to promote in Europe the development of mechanics as a branch of science and engineering. Mechanics deals with motion, flow and deformation of matter, be it fluid or solid, under the action of applied forces, and with any associated phenomena. The Society is governed by a Council composed of elected and co-opted members.

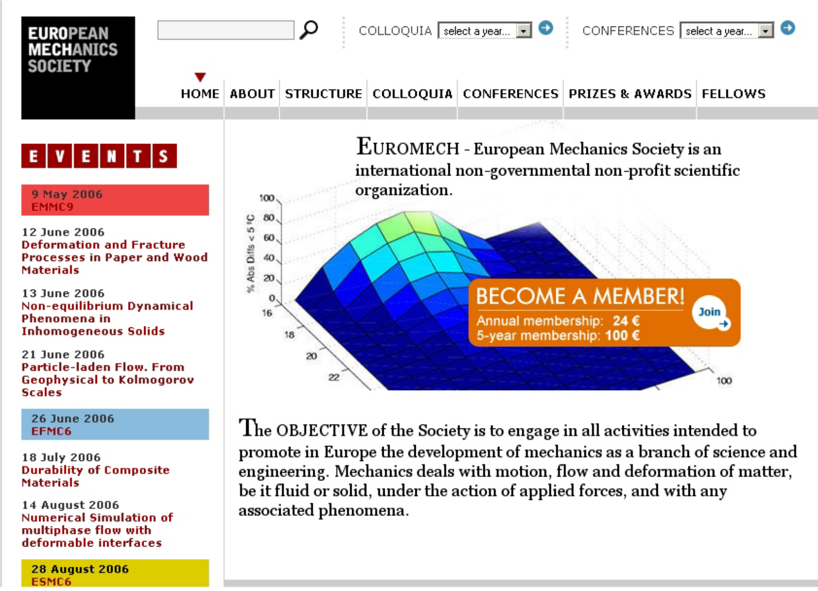
Activities within the field of mechanics range from fundamental research on the behaviour of fluids and solids to applied research in engineering. The approaches used comprise theoretical, analytical, computational and experimental methods. The Society shall be guided by the tradition of free international scientific co-operation developed in EUROMECH Colloquia.

In particular, the Society will pursue this objective through:

- The organisation of European meetings on subjects within the entire field of mechanics;
- The establishment of links between persons and organisations including industry engaged in scientific work in mechanics and in related sciences;
- The gathering and dissemination of information on all matters related to mechanics;
- The development of standards for education in mechanics and in related sciences throughout Europe.

These activities, which transcend national boundaries, are to complement national activities.

The Society welcomes to membership all those who are interested in the advancement and diffusion of mechanics. It also bestows honorary membership, prizes and awards to recognise scientists who have made exceptionally important and distinguished contributions. Members may take advantage of benefits such as reduced registration fees to our meetings, reduced subscription to the European Journal of Mechanics, information on meetings, job vacancies and other matters in mechanics. Less tangibly but perhaps even more importantly, membership provides an opportunity for professional identification; it also helps to shape the future of our science in Europe and to make mechanics attractive to young people.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26 June 2006 EFMC6
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