

EXTENDED ABSTRACTS

of the 5th International Congress of
Croatian Society of Mechanics
held in Seget Donji/Trogir, Croatia September 21-23, 2006

CIP - Katalogizacija u publikaciji Nacionalna
i sveučilišna knjižnica, Zagreb

UDK 531/534(063)

INTERNATIONAL Congress of Croatian Society of Mechanics (4
; 2003 ; Bizovac) Extended Abstracts of the
4th International Congress of Croatian Society of Mechanics held in
Seget Donji/Trogir, September 21-23, 2006. / edited by Franjo
Matejiček. Zagreb: Croatian Society of Mechanics, 2006.

Bibliografija iza svakog rada.

ISBN 953-96243-8-X.

Mehanika ~ Zbornik

430905170

Edited by Franjo Matejiček

Published by
CROATIAN SOCIETY OF MECHANICS

Head Office: Faculty of Mechanical Engineering and Naval Architecture
University of Zagreb Ivana Lučića 5, HR-10000 Zagreb, Croatia

Design by:
Franjo Matejiček

Printed by:
Grafika d.o.o. Osijek, Croatia

Issues: 150

ISBN 953-96243-8-X

All the papers are printed in their original form as were submitted by the authors.

5th INTERNATIONAL CONGRESS OF CROATIAN SOCIETY OF MECHANICS

ORGANIZING COMMITTEE

Franjo MATEJIČEK,	University of Osijek, President
Ivo ALFIREVIĆ,	University of Zagreb
Jasminka BIONDIĆ,	University of Zagreb
Nenad BIĆANIĆ,	University of Glasgow
Lidija FRGIĆ,	University of Zagreb
Dražan KOZAK,	University of Osijek
Zoran MRŠA,	University of Rijeka
Pavao MAROVIĆ,	University of Split
Jurica SORIĆ,	University of Zagreb
Vicko ŠIMIĆ,	University of Zagreb

Organiser of the Congress:

CROATIAN SOCIETY OF MECHANICS

Head Office: Faculty of Mechanical Engineering and Naval Architecture
University of Zagreb
Ivana Lučića 5, HR-10000 Zagreb, Croatia <http://www.csm.hr>

Secretary's Office

Jasminka Biondić
Faculty of Mechanical Engineering and Naval Architecture
University of Zagreb
Ivana Lučića 5, HR-10000 Zagreb, Croatia
Tel +385-1-61 68 540
Fax +385-1-61 68 187
e-mail: jasna.biondic@csm.hr

CENTRAL EUROPEAN ASSOCIATION FOR COMPUTATIONAL MECHANICS (CEACM)

Under the auspices of
Ministry of Science, Education and Sport of the Republic of Croatia

*The Extended Abstracts and Full-text Paper on CD-ROM was
published with the support of
Ministry of Science, Education and Sport of the Republic of Croatia*

Sponsored by:
University of Zagreb
Faculty of Mechanical Engineering and Naval Architecture in Zagreb
Faculty of Civil Engineering in Zagreb

University J.J. Strossmayer in Osijek
Mechanical Engineering Faculty in Slavonski Brod
Faculty of Civil Engineering in Osijek

University of Rijeka
Technical Faculty in Rijeka Faculty of
Civil Engineering in Rijeka

University of Split
Faculty of Electrical Engineering, Mechanical Engineering and
Naval Architecture in Split
Faculty of Civil Engineering and Architecture in Split

University of Dubrovnik
Engineering department in Dubrovnik

Croatian National Tourist Board
Zagreb

INTERNATIONAL SCIENTIFIC COMMITTEE

Ivo ALFIREVIĆ,	University of Zagreb
Nenad BIĆANIĆ,	University of Glasgow
Claudio BORRI,	University of Florence
René de BORST,	Delft University of Technology
Josip BRNIĆ,	University of Rijeka
Elmar FICKER,	Technical University München
Alessandro FREDDI,	University of Bologna
Lidija FRGIĆ,	University of Zagreb
Stjepan JECIĆ,	University of Zagreb
Vinko JOVIĆ,	University of Split
Michal KLEIBER,	Polish Academy of Sciences Warsaw
Vladimir KOMPIŠ,	University of Žilina
Dražan KOZAK,	University of Osijek
Ivica KOŽAR,	University of Rijeka
Wilfried B. KRÄTZIG,	Ruhr - University of Bochum
Željko LOZINA,	University of Split
Herbert A. MANG,	Vienna Technical University
Pavao MAROVIĆ,	University of Split
Franjo MATEJIČEK,	University of Osijek
Ante MIHANOVIĆ,	University of Split
Zoran MRŠA,	University of Rijeka
Osman MUFTIĆ,	University of Zagreb
Maks OBLAK,	University of Maribor
Eugenio OÑATE,	Polytechnic University of Catalonia
Roger J. OWEN,	University of Wales Swansea
Bernhard SCHREFLER,	University of Padua
Ivica SMOJVER,	University of Zagreb
Jurica SORIĆ,	University of Zagreb
Vicko ŠIMIĆ,	University of Zagreb
Leopold ŠKERGET,	University of Maribor
Boris ŠTOK,	University of Ljubljana
Zdravko TERZE,	University of Zagreb
Zdenko TONKOVIĆ,	University of Zagreb
Masao TOYODA,	University of Osaka
Zdravko VIRAG,	University of Zagreb
Michael P. WNUK,	University of Wisconsin - Milwaukee
Peter WRIGGERS,	University of Hannover

INTERNATIONAL REVIEW COMMITTEE

Ivo ALFIREVIĆ,	University of Zagreb
Josip BRNIĆ,	University of Rijeka
Mirko HUSNJAK,	University of Zagreb
Stjepan JECIĆ,	University of Zagreb
Ivica KOŽAR,	University of Rijeka
Željko LOZINA,	University of Split
Pavao MAROVIĆ,	University of Split
Franjo MATEJIČEK,	University of Osijek
Zoran MRŠA,	University of Rijeka
Maks OBLAK,	University of Maribor
Dragan PUSTAČIĆ,	University of Zagreb
Damir SEMENSKI,	University of Zagreb
Ivo SENJANOVIĆ,	University of Zagreb
Vladimir SIGMUND,	University of Osijek
Jurica SORIĆ,	University of Zagreb
Luka SOPTA,	University of Rijeka
Leopold ŠKERGET,	University of Maribor
Zdravko TERZE,	University of Zagreb
Goran TURKALJ,	University of Rijeka
Zdravko VIRAG,	University of Zagreb
Hinko WOLF,	University of Zagreb

PREFACE

The 5th International Congress of Croatian Society of Mechanics, which is organized by Croatian Society of Mechanics and Central European Association for Computational Mechanics (CEACM), is the fourth in the series of organized gathering of scientists and researchers in the field of general mechanics, mechanics of solids, mechanics of fluid and computational mechanics.

These Extended abstracts contains 2 invited lectures and 78 extended abstracts as well as attached CD-ROM with all of 78 session papers presented at the Congress in Hotel Medena in Seget Donji (Trogir) during the period September 21 – 23, 2006.

The overwhelming majority of papers deals with topics of computational mechanics but the papers from other areas of mechanics are included, too. Although, all contributing papers have been reviewed, they are printed and writhed from camera-ready copies supplied by the authors and the editors cannot accept responsibility for any errors or inconsistencies contained in the papers.

Editors and organizers are very grateful to all reviewers and participants for their contributions in making this Congress a success. They also wish to thank our sponsors for their financial support and all those who helped in any way in organizing this Congress.

Franjo Matejiček Zagreb,
September 2006.

4th International Congress of Croatian Society of Mechanics

CONTENTS

Nr.	Authors	Title	Page
193	Hrvoje Jasak	<i>Multi-physics Simulations in Continuum Mechanics</i>	3
192.	Dirk Lefeber, Björn Verrelst, Ronald Van Ham, Pieter Beyl, Bram Vanderborght, Michaël Van Damme, Joris Naudet, Rino Versluys	<i>Robotic Applications powered by Compliant Actuators</i>	13
102.	Chi-Min Liu, Chin-Hwa Kong	<i>Fully nonlinear equations for internal waves in a two-layer fluid</i>	31
103.	M. Vesenjak, L. Krstulović- Opara, A. Öchsner, Z. Ren, Ž. Domazet	<i>Experimental and Numerical Modelling of Open-Cell Cellular Structures</i>	33
104.	Gianpietro Del Piero, Giovanni Lancioni, Riccardo March	<i>A Variational Model for Fracture Mechanics</i>	35
105.	Jan Gajdošík, Jan Skoček, Jan Novák, Michal Šejnoha and Jan Zeman	<i>Homogenization of Solids Weakened by Cracks Averaging Techniques or Periodic Unit Cells</i>	37
106.	Jiří Šejnoha, Michal Šejnoha and Marie Kalousková	<i>Stochastic Modelling of Time Dependent Behaviour of Earth Structures</i>	39
107.	Michal Šejnoha, Marie Ka- lousková, Jan Zeman and Jan Novák	<i>Stochastic Approach to The Stability Analysis of Earth Slopes</i>	41
108.	Jan Sýkora, Jan Vorel, Michal Šejnoha, Jan Zeman and Jiří Šejnoha	<i>Multiscale Modelling of Masonry Structures – Synthesis of Constitutive Models and Scale Transition</i>	43
109.	Jan Vorel, Jan Sýkora, Jiří Še- jnoha and Michal Šejnoha	<i>Prediction of Homogenized Material Properties of Quarry Masonry from Micromechanics</i>	45
110.	Riko Rosman	<i>Cyclosymmetry in the Mechanics of Structures</i>	47
111.	Diana Šimić	<i>Optimizing Load-Bearing Capacity of Beams Sub-jected to Bending</i>	49
112.	Rudolf R. Pušenjak, Maks M. Oblak	<i>Symbolic Computation of Extended Lindstedt-Poincare Method with Multiple Time Scales</i>	51
113.	Torims T., Rudzītis	<i>The Three-Dimensional Roughness Parameters Model for Surfaces of Shipboard Diesel Engines Crankshaft Bearings</i>	53
114.	M. Ulbin, Z. Ren	<i>Virtual Terrain Modelling using GPS tracks</i>	55
115.	Zdravko Virag, Ivo Džijan and Severino Krizmanić	<i>Improved Symmetric Gauss-Seidel Method for Solving Sparse Linear Systems Appear in CFD</i>	57
116.	Jurij Avsec, Maks Oblak	<i>The Calculation of Thermophysical Properties for Solids on the Basis of Statistical Mechanics</i>	59

117. J.Rudzitis, J.Sherbergs, G.Konrads	<i>Calculation of Contact Area of Screw – and – Nut Drive Theard Surfaces</i>	61
118. T. Kroupa, R.Zemčík, V. Laš	<i>Progressive failure analysis of orthotropic plate loaded by transverse low-velocity impact</i>	63
119. D. Lanc, G. Turkalj, J. Brnić	<i>Beam Model for Creep Buckling Analysis</i>	65
120. Kozo Ikegami	<i>Strength Estimation of Adhesively Bonded Joint by Using the Adhesive Strength Law</i>	67
121. Vítězslav Adámek , František Valeš	<i>Non-stationary state of stress of a viscoelastic beam – dispersion and attenuation behaviour</i>	69
122. Z. Sharif Khodaei, J.Zeman	<i>Elastic modelling of functionally graded materials: one dimensional case</i>	71
123. J.Zeman,J. Novák, M.Šejnoha and J.Šejnoha	<i>Pragmatic multi-scale and multi-physics analysis of the Charles Bridge in Prague</i>	73
124. Jan Fuxa, Rostislav Kubala	<i>Multi-axial Static Strength Criterion Fit for Ductile Quasi-isotropic Materials</i>	75
125. Natálie Součková, Milan Matě- jka, Lukáš Popelka	<i>Experimental Investigation and Numerical Analyses of Flow on Airfoils with High Lift Devices and Spoilers</i>	77
126. Doleček Vlatko, Filipović Ivan, Bibić Dževad, Petrović Marin	<i>Importance of Crankshaft Torsional Vibration Sig- nals in Diagnostics of IC Engines</i>	79
127. Marta Sulyok-Selimbegović, Diana Šimić, Nataša Mrazović	<i>Reliability of the Steel Structure Under Wind Action</i>	81
128. Safet Isić, Vlatko Doleček, Isak Karabegović	<i>An Identification of Bifurcation Type Using Post- critical Motion Analysis</i>	83
129. Zlatan Kulenović	<i>Forces on a Dislocation Near a Crack in Piezoe- ramic Material</i>	85
130. Frane Vlak, Vedrana Cvitanić, Igor Duplančić	<i>Determination of the friction parameters using MATLAB interpolation surfaces</i>	87
131. Valdas Čaika, Mario Jelović and Dino Mandić	<i>Theoretical and Experimental Modeling of a Piezo- electric Common Rail Injector</i>	89
132. Vladimír Bella, Mariana Kuf- fová	<i>Fatigue characteristics of magnesium alloy AZ 91D at high - frequency cyclic loading</i>	91
133. V. Kompiš, M. Kompiš, M. Kaukič	<i>Method of Continuous Dipoles for Modeling Com- posite Materials with Microstructure</i>	93
136. T. Janda, M. Šejnoha	<i>Influence of Yield Function on Modelling of Me- chanical Response of Soil Materials</i>	95
137. D. Šimurda, T. Hyhlík, P. Šafařík	<i>Vortex Structures in Blade Cascades</i>	97
139. Ivica Kožar, Tino Medvidović	<i>Some remarks on displacement based dynamic measurements</i>	99
140. Marko Čanadija, Josip Brnić	<i>A Model for Cyclic Finite Strain Thermoplasticity</i>	101
141. Jindrich Petruska, Jan Bork- ovec	<i>Computational Simulation and Verification of Bolt Head Trimming Process</i>	103
142. Ivica Kožar, Neira Torić	<i>Modeling of Long Structures on Infinite Supporting</i>	105

143. Tomáš. Plachý, Michal Polák, Antonín. Rezek	<i>Monitoring of modal characteristic changes of a prestressed concrete element in dependence on dynamic cyclic loading</i>	107
144. I. Kožar, J. Ožbolt, B. Irhan and V. Travaš	<i>Contact Simulation During Nail Penetrating into a Concrete Block</i>	109
146. M. Borovinšek, Z. Ren	<i>Lattice modelling of metallic foams</i>	111
147. M. Borovinšek, Z. Ren	<i>Simulation of road safety barrier crash tests</i>	113
148. Hinko Wolf, Damir Semenski, Aleksandar Sušić	<i>Convergence of Eigenvalues of Monodromy Matrix of Piecewise Linear Oscillators</i>	115
149. J. Novák, M. Vokáč, , M. Šejnoha	<i>Experimental identification of Nonlinear Material Parameters of Regular Brick Masonry</i>	117
150. Zemčík R., Laš V., Měšt'ánek P.	<i>Numerical and experimental analysis of delamination of unidirectional composite material</i>	119
151. R. Valenta, J. Šejnoha, M. Šejnoha, T. Krejčí	<i>Accomplishment of Water Tightness and Volume Stability of Concrete Slabs</i>	121
153. G. Olmia, A. Freddi	<i>Shot peening technique for fatigue improvement of high strength steel gears</i>	123
155. Mozga N., Sudnieks F., Dreija Z, Filipov A.	<i>Flexible Manufacturing Technology of Assembly Based on Computer-Aided Design Systems</i>	125
156. Tomislav Svaguša	<i>The Factor Analysis of Prestressing Steel Cables one Component of Prestressed Concrete Beams</i>	127
157. Ante Mihanović, Željana Nikolić, Nikolina Živaljić	<i>Resonant Spectrum Analysis of Seismic and Wind Loading of Structure</i>	129
158. Ion Dumitru, Nicolae Faur, Adrian Cipleu	<i>Impact Fatigue – Research Domain for Further Development</i>	131
160. Lovre Krstulović-Opara and Ivan Perasović	<i>A comparison of measured data, code and fluid-structure interaction simulation of the wind flow around solar panels</i>	133
161. G. Bartoli, C. Borri, C. Costa, C. Mannini	<i>Time vs. frequency-domain analysis of single/multiple box girder bridge deck sections</i>	135
162. Goran Turkalj, Goran Vizen-tin, Josip Brnić	<i>Hybrid Beam Element for Stability Analysis of Semi-Rigid Frames</i>	137
163. Giangiacomo Minak and Daniele Ghelli	<i>Design of a Drop-weight machine for composite materials impact testing</i>	139
164. D. Pustaić and M. Lovrenić	<i>Analytical and Numerical Investigation of Crack Opening in Strain – Hardening Material</i>	141
166. Vjekoslav Damić, Majda Čohodar	<i>Dynamics of Flexible Multibody Systems using a Co-rotational approach</i>	143
168. Zoran Čarija, Zoran Mrša, Luka Dragović	<i>Turbulent flow simulation in Kaplan draft tube</i>	145
169. Vedrana Cvitanić, Željko Ložina, Frane Vlák	<i>Application of non-associated flow rule for sheet metal forming</i>	147
170. Marko Čavrak, Zoran Mrša, Goranka Štimac	<i>Regional Air Pollution Modelling</i>	149

171.	Siniša Družeta, Jerko Škifić, Bojan Crnković, Luka Sopta, Danko Holjević, Nelida Črnjarić-Žic, Senka Maćešić	<i>River Flood Lines Prediction with 1D Open Channel Model</i>	151
173.	L. Kranjčević, B. Crnković and N.Črnjarić-Žic	<i>Improved Implicit Numerical Scheme for one- Dimensional Open Channel Flow Equations</i>	153
174.	Jerko Škifić, Nelida Črnjarić- Žic, Bojan Crnković	<i>Efficient implementation of ENO scheme in water hammer wave modeling</i>	155
175.	Vedrana Kozulić, Blaž Go- tovac, Hrvoje Gotovac, Tonči Radelja	<i>Numerical solving of problems with high gradients using adaptive multiresolution method</i>	157
176.	Zeljane Ložina, Damir Vučina, Damir Sedlar	<i>Dynamic properties of structures and advanced al- gorithms for validation</i>	159
178.	Ante Agić, Budimir Mijović and Veljko Filipan	<i>The multiscale material behaviour model</i>	161
179.	Budimir Mijović and Ante Agić	<i>Numerical modelling of contact between elastic and rigid body</i>	163
180.	Igor Karšaj, Carlo Sansour, Jurica Sorić	<i>An Efficient Modelling of Anisotropic Large Strain Elastoplastic Deformation Responses</i>	165
181.	Tomislav Jarak, Jurica Sorić, Josip Hoster	<i>A robust meshless formulation for numerical analy- sis of shell-like structural components</i>	167
182.	N.Gubelj, D. Kozak, J. Pre- dan, P. Konjatić, M. Oblak, F. Matejiček, A. Sedmak, J. Vo- jvodić-Tuma	<i>Structural Integrity Assessment of T-Joints in Steam Pipelines</i>	169
183.	Marija Živić, Mario Šavar, Severino Krizmanić	<i>Effect of asymmetric density-temperature relation- ship of water on natural convection in rectangular enclosure</i>	171
184.	Ivica Skozrit, Zdenko Tonković, Ivo Alfrević	<i>J Estimation and Limit Load Analysis of Surface Cracked Tubes</i>	173
186.	N. Quinn, A. Ivankovic, A. Karac,	<i>An Investigation into Atherosclerosis Using the Fluid-Structure Interaction Technique</i>	175
187.	N. Murphy, A. Ivankovic	<i>An Investigation into Dynamic Crack Bifurcation in PMMA using a Cohesive Zone Model</i>	177
188.	N.Vulić, A. Šestan, V. Cvi- tanić	<i>Shafting Alignment Calculation and Validation Cri- teria</i>	179
189.	Zdravko Terze, Joris Naudet	<i>Projective Constraint Violation Stabilization Method for Multibody Systems on Manifolds</i>	181
190.	Joris Naudet, Dirk Lefebvre	<i>On the use of canonical momenta for real-time simu- lations</i>	183
191.	Vladimir Z. Sigmund, Tanja Kalman and Ivica Guljaš	<i>Nonlinear Wall Models for Earthquake Response Analysis</i>	185

Invited lectures

MULTI-PHYSICS SIMULATIONS IN CONTINUUM MECHANICS

Hrvoje Jasak

Keywords: Multi-physics, object-oriented, FVM, automatic mesh motion, fluid-structure, free surface flow.

1. Introduction

The era of commodity computing brings a profound change in the use of numerical analysis of continuum mechanics problems in science and engineering. Teraflop-rate computing facility, until recently available only to chosen few is becoming commonplace in an industrial setting. Range and complexity of problems where numerical methods are applied is also expanding. Here, massive reduction in the cost of solution allows the empirical or simplified models to be replaced by fundamental sets of equations.

Numerical simulation is also expanding beyond its traditional realm of structural analysis and fluid flow. In the first phase, this covers real material properties, chemically reacting flows, multi-phase and free surface flows, radiative heat transfer, electromagnetics *etc.* A natural extension of the above is coupled analysis of engineering components, covering a number of interacting phenomena. For example, a Fluid-Structure Interaction (FSI) simulation covers the physics of fluid flow and stress analysis, but the emphasis is on dynamic interaction between the two. Coupled nature of the problem brings its own challenges beyond the realm of “bread-and-butter” fluid flow or structural dynamics simulations.

In this paper, expanded role of numerical solvers in Computational Continuum Mechanics (CCM) will be examined from the point of view of solver development. We shall present a new way of implementing partial differential equations of continuum mechanics in numerical simulation software through the use of object-oriented programming. Desirable consequences of object-orientation include data protection and encapsulation, layered development and code re-use and facilitate side-to-side implementation of various physical models and discretisation methods. This will be followed by an overview of complex geometry support and model to model coupling framework. The paper is concluded with examples of coupled multi-model simulations. OpenFOAM [6, 10], an Open Source object-oriented C++ library for numerical simulations in continuum mechanics is the basis of the review and source of examples.

2. Multi-Physics Solver Design

Multi-physics problems vary widely in their nature, both in the physics involved and manner of coupling. The first step in a general multi-physics platform involves providing the required models in a “single-physics” mode. Issues of numerical implementation of continuum mechanics models in will thus be considered first in general terms.

Current generation of CCM solvers is written using the *functional approach* to software design. Here, data is available globally and undergoes a set of operations until a solution is reached. The software is broken up based on functionality and the data is available globally. For a large software project such as a modern CCM solver, this causes a number of problems in development, testing and deployment, as software complexity increases by its functionality cubed.

A drawback of functional programming is a pre-determined flow of execution: software architecture sets a “user scene” and provides a number of options. For example, a FVM fluid flow solver may provide options for combustion or multi-phase flow, but the “simulation scene” remains the same: space is represented by a single mesh, with fixed variable arrangement, a set of equations is in standard form, with a fixed choice of material properties, linear equation solvers *etc.* Even simple modifications, *e.g.* holding multiple meshes or solving two decoupled models side by side may be problematic.

Object-oriented (OO) software design attempts to break the complexity by creating smaller self-contained units, combining data and functions. The idea of object orientation is to analyse the problem at hand and recognise main objects that feature in it. Objects are then represented in software and combined to assemble the solver. Some crucial advantages come to mind. Firstly, isolated components with well-defined interfaces are implemented and tested in isolation, breaking inter-dependence. Secondly, components are implemented in a generic manner with re-use in mind and shared between various applications. Thirdly, complex components and top-level applications are assembled using low-level objects, making them smaller and easier to manage. Finally, encapsulating and protecting the data in smaller units facilitates debugging and maintenance.

We shall now examine the OO approach implemented in OpenFOAM.

Implementing Continuum Models. Continuum mechanics has a common language: models are expressed as partial differential equations (PDE). If PDEs could be represented in software in a clearly recognisable form, implementation of complex models and inter-equation coupling would be made easy. With object orientation in mind, let us explore the software components that represent the incompressible turbulence kinetic energy equation:

$$\frac{\partial k}{\partial t} + \text{div}(\mathbf{u}k) - \text{div}[(\nu + \nu_t)\text{grad}(k)] = \nu_t \left[\frac{1}{2} (\text{grad}(\mathbf{u}) + \text{grad}(\mathbf{u})^t) \right] - \frac{\varepsilon_0}{k_0} \varepsilon \quad (1)$$

as the following implementation in code:

```
solve
(
    fvm::ddt(k)
  + fvm::div(phi, k)
  - fvm::laplacian(nu() + nut, k)
  == nut*magSqr(symm(fvc::grad(U)))
  - fvm::Sp(epsilon/k, k)
);
```

Figure 1: Software representation of the k equation in OpenFOAM.

The above represents a valid syntax in the C++ programming language [1] and individual terms of the equation can be easily identified without any further knowledge.

The main object in Eqn. (1) is an operator: *e.g.* time derivative, divergence or a gradient. An operator object cannot be implemented in isolation: we need a representation of space and time, fields and boundary conditions which support its functionality. We shall start the analysis from simple objects and combine them to reach the above.

Basic Components. Functional programming languages provide some basic building blocks to the programmer: integers, floating point numbers, strings, pointers and functions. In object orientation, one defines *classes*: utilising basic building blocks and combining data and functions, they provide functionality in a similar user-friendly syntax.

Continuum mechanics operates with fields of scalars, vectors and tensors, where a tensor is represented as a set of floating point numbers with associated algebra. The first object we can define is a *vector*. It contains 3 floating point numbers as data and associated functions, *e.g.* dot-product of two vectors, vector magnitude, addition and subtraction rules for a vector *etc.* A *tensor* is defined in a similar manner, with operations appropriate for its type.

The second component we can define in generic terms is a `List`: a collection of objects. In functionality, a list of objects is independent of the type it contains, indicating generic implementation.

We shall now define computational space and time. Space, represented by a mesh, is defined as a `List` of points (= vectors), faces and cells. A mesh class supports polyhedral cell handling and provides connectivity information, metrics (e.g. cell volumes) and associated mesh data. Along the time axis, simulation advances in a series of time steps. Here, relevant properties are the start and end time, time step size as well as various data management functionality, including input/output.

In most cases, numerical methods represent a field as a set of discrete values at prescribed points and a piecewise function defining spatial variation. Thus, operations will typically be repeated over all points. To avoid explicit looping, we can define a `Field` of objects as a `List` with associated algebra of the base type. Thus, multiplying a `Field` of vectors by a scalar implies the same operation on all members.

A special use of `Fields` is in boundary conditions. Here, a Dirichlet condition involves not only a list of values but also behaviour: fixed boundary values. A `patchField` class represents a boundary condition in generic terms, exposing the common behaviour. Evaluating a list of boundary conditions involves operating with the generic type and performing the action associated with the actual type. Specific conditions are derived from the generic form but implement specific behaviour, e.g. a Neumann condition or a symmetry plane.

Eqn. (1) is dimensionally homogenous, a fact we can use to facilitate checking. For this purpose, a `dimensionSet` class (mass, length, time, temperature, concentration, current and luminous intensity), with algebraic operations is built to dimensionally check all field and matrix operations.

`Fields` of variables are convenient, but insufficient for our purposes: a reference to spatial distribution is missing. Associating the values with a mesh, adding boundary conditions and a dimension set forms a `GeometricField`: a complete numerical representation of a field variable. Combinations of tensor rank and storage location indicate extensive code re-use: for example a cell-centred field of vectors in the FVM, `volVectorField` uses the same implementation as a `pointScalarField` (a scalar value at each mesh point) used for post-processing.

Implicit discretisation methods convert partial differential equations into sets of linear algebraic equations a solution of which is sought. Objects we can define are a sparse matrix (`lduMatrix`) and a generic linear equation solver `lduMatrixSolver`. Preconditioned Conjugate Gradient and Algebraic Multigrid [9] are examples of specific solver algorithms implemented in the library.

Discretisation Methods. All classes mentioned above exist without reference to a particular discretisation: they are independent and can be shared throughout the code. Implementation of a discretisation method consists only of discretisation-specific components and re-uses field algebra, boundary conditions and matrix support. Its components fall into three groups, illustrated on the Finite Volume (FV) discretisation in OpenFOAM.

Interpolation defines how to recover a solution away from computational points based on prescribed spatial variation, for example on cell faces;

Differentiation implements the explicit form of differential operators, e.g. given a scalar field ϕ a FV gradient operator will produce a vector field $grad(\phi)$. In Fig. 1, differential operators are denoted by `fvc::` (finite volume calculus);

Discretisation constructs a discrete representation of a differential operator in matrix form, denoted by `fvm::` (finite volume method).

Based on the above, we can revisit Fig. 1: `k` is a `volScalarField` and contains boundary conditions and a reference to a mesh. `fvm` operators produce a sparse matrix for each operator and matrices are summed up to form the discretised equation. The r.h.s. of the linear system is assembled via differentiation and field algebra operations. Matrix and field objects are dimensionally checked in algebra and calculus operations. Once assembled, the linear system is solved to obtain

field values for k .

Generic Interfaces. Objects listed above allow us to assemble a compact and legible solver code for a prescribed set of equations. In engineering use, this is not sufficient: one does not wish to repeat the exercise of code assembly every time a different turbulence model is selected by the user. In order to handle this, models are grouped under common interfaces, *virtual base classes*, defining behaviour in generic terms. Thus, a fluid flow solver would refer to a generic Reynolds averaged turbulence model and uses its interface without exposing the details. Selection of a particular turbulence model from a library of choices is performed at run-time. Each model is independent from others but answers to the same interface and is thus interchangeable from the point of view of consumer code.

This pattern appears throughout the software: boundary conditions, convection discretisation schemes, linear solvers, time advancement methods, material properties, physical models *etc.* behave in the same way and are implemented in identical manner.

3. Complex Geometry Support

In engineering practice, complex geometry is a rule rather than exception. Mesh generation remains a major issue: some recent solver developments, like polyhedral cell support [2, 4], aim to provide additional flexibility in meshing. Multi-physics, and FSI in particular, involves solution-dependent change of domain shape which needs to be handled through dynamic mesh support. We can recognise three modes of operation:

Automatic Mesh Motion, where the mesh is deformed through point motion. OpenFOAM implements a vertex-based automatic mesh motion solver [3, 8] with polyhedral cell support using a mini-element technique. Point motion is obtained by solving a motion equation, with boundary motion acting as a boundary condition [8]. Choice of motion equation and details of discretisation guarantee the mesh validity criteria will not be violated and mesh quality will be preserved as much as possible. Implemented mesh motion equations include a Laplace equation with several forms of variable diffusivity and a pseudo-solid equation [8].

Topological Changes. In cases of extreme boundary deformation, point motion may result in reduced mesh quality or the initial mesh may not accommodate the motion at all. In the absence of robust automatic meshing, this is handled through topological changes, where the mesh size or connectivity changes during the simulation. Topological change support is implemented in an OO manner. Lowest level changes, handled by the mesh allow us to add, modify a remove a point, face or a cell in the mesh. On top of this, pre-packaged topological change classes are implemented for typical cases, e.g. mesh layering and sliding interfaces. Finally, mesh templates are assembled and packaged into classes. Examples would include a generic mixer vessel mesh or an internal combustion engine mesh with moving piston and valves.

Automatic Mesh Generation. The ultimate way of handling topological changes is through automatic mesh generation. Here, local or global re-meshing is triggered by excessive mesh deformation or resolution requirements, providing flexibility for cases with global topology change. The technique relies on robust and locally controllable mesh generation, which is still an area of research. Polyhedral meshing [4], is a novel and promising technique in this direction.

4. Model-to-Model Coupling

Having established a framework for model implementation, we will turn to model coupling. Mainly due to software limitations, bulk of coupled FSI simulations today involves a combination of solvers, with management and synchronisation issues. For example, one code handles the fluid flow side, another is used for structural analysis and sometimes a third for coupling, data interpolation and simulation management. Even at first sight, this is complicated and unwieldily, and imposes limitations on the mode of coupling and problems in model setup.

In contrast, the object model presented above naturally allows for presence of multiple self-contained objects in the same environment. For example, multiple mesh support or multiple co-existing models are created by simply instantiating them. Moreover, various models and discretisation techniques rely on identical implementation of fields, matrices, linear solvers etc. allowing easy implementation of a range of coupling techniques.

Level of Coupling. Depending on the implied level of interaction between regions and physical models, one can identify four levels of coupling. The first, containing *one-way interaction* only is not of interest.

Explicit two-way coupling is typical when multiple software or discretisation methods are involved: simulations run side-by-side and exchange coupling information during the run. This approach formally operates in Picard iterations and fails even in modestly interacting problems.

Closer model-to-model coupling involves *matrix-level interaction*, where two physical models and model-to-model coupling terms are discretised separately and combined into a single linear system before the solution. This is an implicit variant of the above, with additional stability provided by the linear solver. Here, we can see the benefit of shared matrix and solver modules between multiple discretisation schemes.

Equation-level coupling [5] is based on the fact that continuum mechanics models originate from mass, momentum and energy balance models. Thus, a fluid-structure system may be considered as a single continuum governed by a single PDE, with different constitutive relations in various regions. Representing closest possible coupling, this level of interaction in fact represents a coupled physical model and may require special numerical techniques. Such techniques are particularly suitable for strong coupling or cases involving phase change and transition regions, *e.g.* melting and solidification.

Coupling Tools and Auxiliary Components. Successful handling of coupled simulations involves data transfer between various models and regions of space, either through data interpolation or in an implicit manner. Data manipulation tools for surface-to-surface interpolation, surface overlap detection, volume solution mapping etc. are available in the library. Shared mesh and field classes simplify the task: interpolation and sampling algorithms are written as generic tools and used as appropriate.

5. Open Source CCM Software

Use of commercial software in multi-physics is hindered by its closed design: licensing and deployment issues limit the access to software modules in source code and even to existing data and functions. This complicates all aspects of coupled simulations and, more generally, research based on commercial codes. Lack of research freedom, incomplete education of students and researchers, use of black box components, limitations of the “user coding” interface and problems with dissemination of results and implemented tools all stem from the closed software model. In the other extreme, complexity of developing a new solver from scratch in a research project is a limiting factor. Lead time increases immensely with increased software complexity and delays the actual creative work for a number of years.

Ironically, numerical software implements publicly available knowledge: implemented physical models and discretisation methods regularly originate from publicly funded research and are described in open literature. In other words, the knowledge is already in the public domain, but the implementation is proprietary.

Open Source Initiative [7] aims at promoting collaboration between users and programmers and is ideal in a research setting: scientific research, review and publishing already operates in this manner. Currently used at over 200 universities, research centres and commercial companies around the world, OpenFOAM [6] is a leading open source CCM project. Its capabilities are much wider than described above and include mesh generation, manipulation and conversion tools, particle tracking, massive parallelism, real material properties databases, data manipulation and analysis tools, pre- and post-processing *etc.*

6. Examples of Coupled Simulations

In this section, we shall present three simulations involving coupled physical models to illustrate the multiphysics capabilities of OpenFOAM.

Flow-Induced Deformation of a Solid. In the first example, we shall examine structural deformation of a solid forced by fluid flow. The flow is incompressible and laminar and couples with a linear elastic structural model in a large deformation formulation. Both models are discretised using second-order accurate unstructured FVM with mesh motion effects [8]. Explicit two-way coupling is used: fluid forces and the coupled boundary are mapped to the solid domain. On resolving the solid motion, deformation is mapped to the fluid domain, where it influences the flow through shape change. In order to accommodate boundary deformation, automatic mesh motion solver is used to deform the fluid mesh. Solid domain is deformed as a part of the solution algorithm in accordance with the model.

The geometry consists of a bent elastic pipe exposed to a pressure pulse and rigidly supported at the top. Under the load, Fig. 2, a pressure-induced wave travels along the pipe and superimposes on large-scale bending. Note the change in flow pattern and vibration induced flow detachment down stream from the bend. Results clearly show the importance of coupling both on the fluid flow and the stress state.

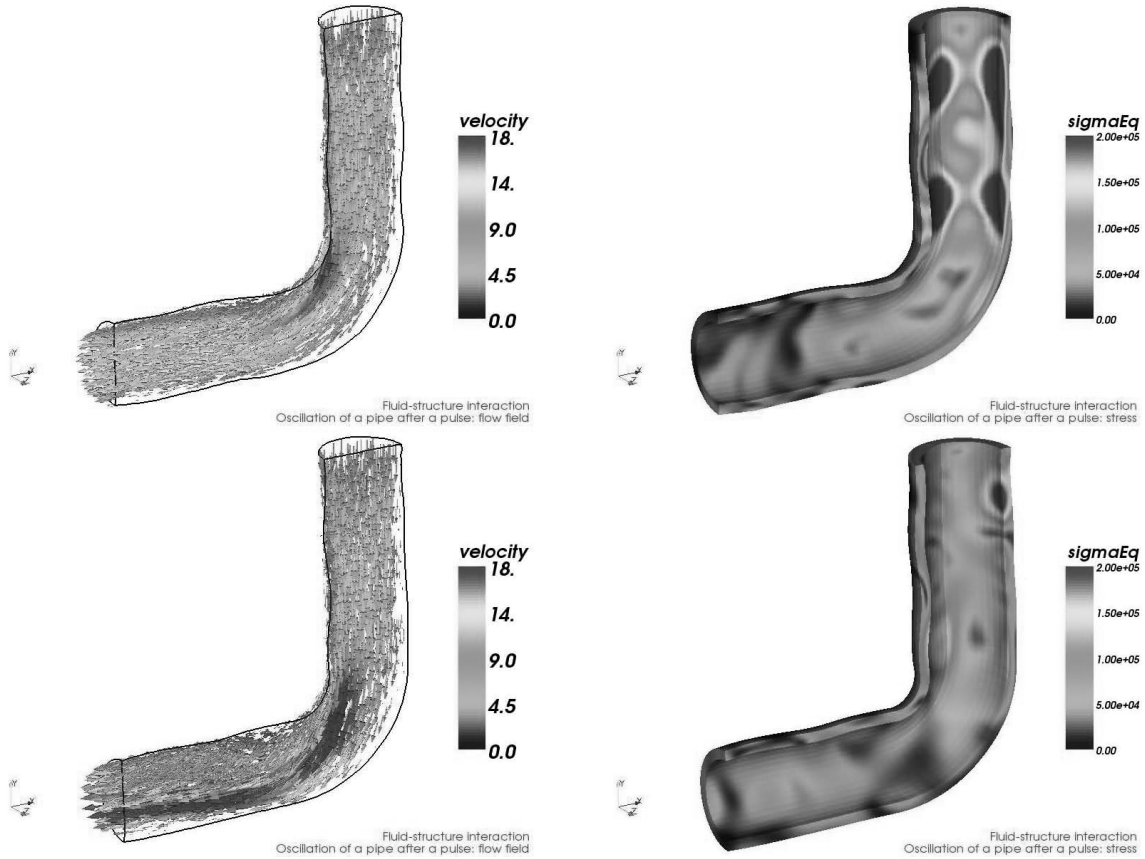


Figure 2: Deformation of a flexible pipe after a flow pulse, $t = 0,05$ s and $0,15$ s.
Left: velocity field; right: equivalent stress.

Coupled Fluid-Structure Problem: Falling Containers. In cases of strong interaction between the fluid and structure, explicit two-way coupling fails to produce a solution: iterative sequence within a time-step diverges due to coupling instability. The nature of coupling in such cases

requires the problem to be treated implicitly in its entirety.

Unlike the above, where fluid and structure were governed by their own equations, the mathematical model used in this example postulates that the complete domain represents a single continuum, governed by mass and momentum conservation law. Fluid and solid regions are marked via an indicator field γ and the constitutive relation for both are combined in a single expression based on γ . In order to achieve this, [5], constitutive law for a solid is written in terms of stress rate with deformation velocity as a primitive variable. Solution algorithm closely mimics standard pressure-velocity coupling techniques in fluid flows [5].

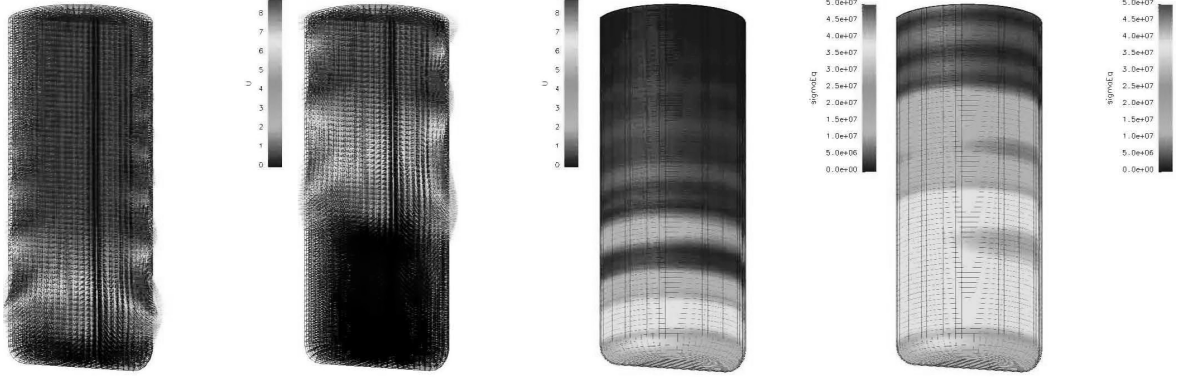


Figure 3: Simulation of a falling elastic container.
Left: velocity field; right: equivalent stress. From [5].

The case under consideration consists of a falling polyethylene container filled with incompressible liquid hitting a solid surface. The impact initiates the flow field which in turn forces the deformation of a container. Fig. 3 shows two snapshots of the flow field in the container and the equivalent stress distribution in the container.

Deforming Mesh Free Surface Flow Simulation: Rising Bubbles.

The final test case incorporates fluid-to-fluid coupling in free surface flows. Consider a case of a freely rising air bubble in water, with the free surface handled through mesh deformation. Both fluids are considered incompressible and in the laminar flow regime. The coupling on the surface consists of the kinematic (zero flux) and dynamic condition, with the pressure jump from surface tension [8]. The simulation is performed in a non-inertial reference frame attached to the bubble.

Coupled solver consists of a pressure-based flow solver, acting on both fluid regions and an automatic mesh motion solver with surface matching and interpolation tools. Boundary deformation is obtained from the free surface condition in fluid equations, where a double boundary condition of fixed pressure level and zero flux defines surface motion. Position of the free surface is iteratively adjusted to satisfy the flux condition by moving the mesh.

Fig. 4 shows the velocity field in the centre plane and deformed bubble shape at four time instances. Note the oscillating nature of interaction between the flow field and bubble shape.

In free surface flows, additional complications occur in the presence of surfactant chemicals, where the value of the surface tension coefficient depends on surface concentration of the pollutant. Limiting ourselves to the case of no absorption and desorption, surface concentration is modelled as passive scalar transport on a moving deforming surface, with the velocity field obtained from bulk flow. The coupled flow model now contains an addition component: a Finite Area solver [8] for passive scalar transport on a moving surface in 3-D.

Fig. 5 shows the velocity field around the surface coloured by local surfactant concentration. Locally variable surface tension has a profound effect on the flow pattern, bubble shape and terminal velocity, as well as its rising trajectory, shown in Fig. 6.

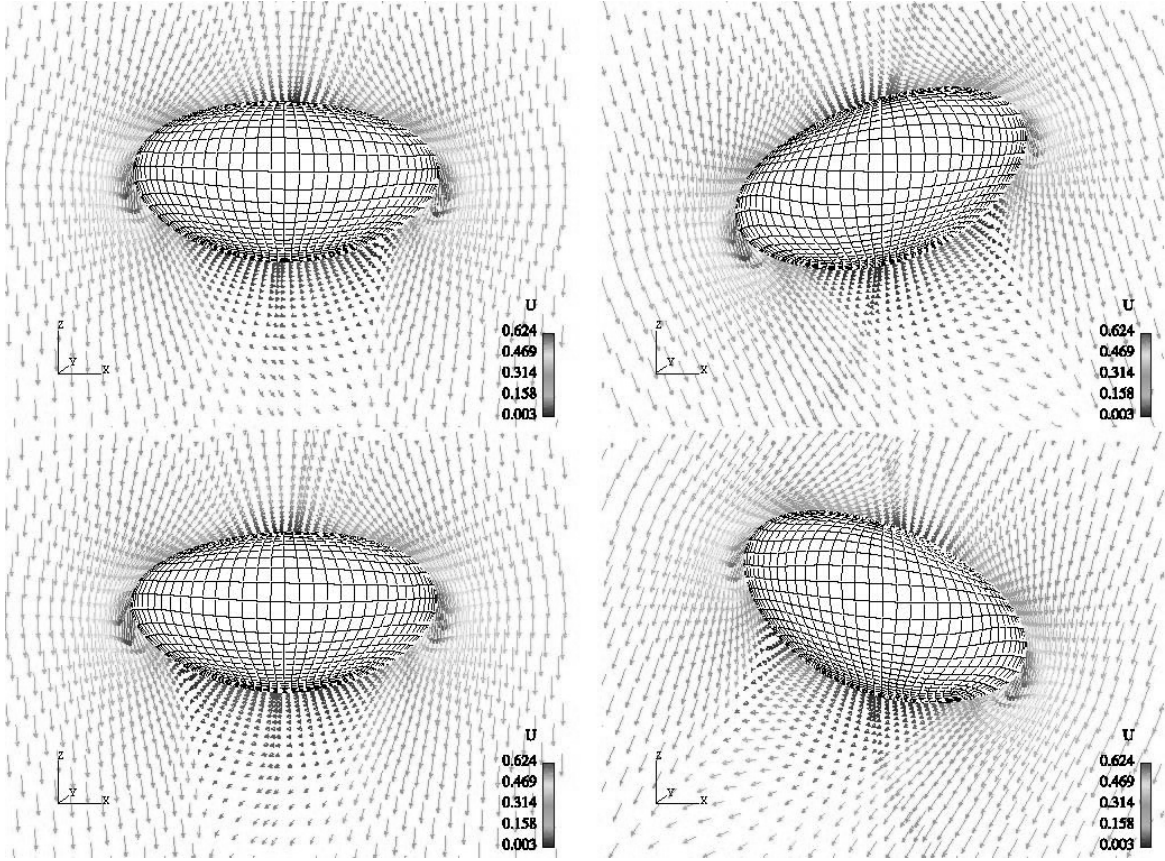


Figure 4: Deformed bubble shape and velocity field in the centre plane [8].

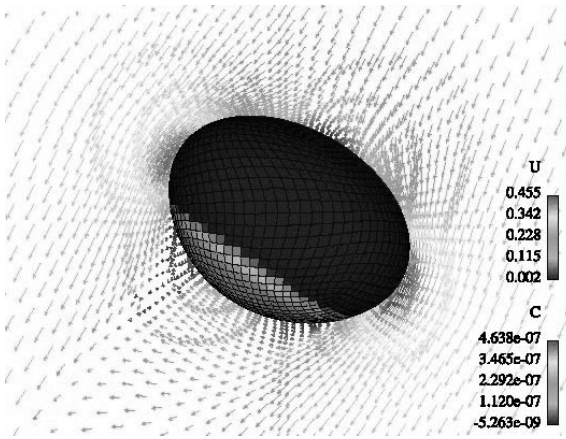


Figure 5: Surfactant concentration and bubble shape.

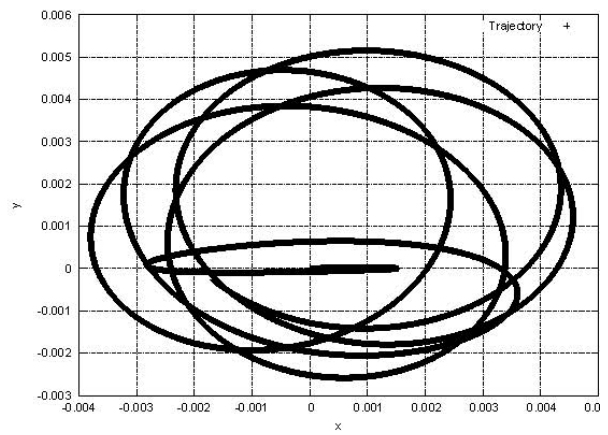


Figure 6: Rising trajectory of the bubble centroid in the $x-y$ plane.

7. Summary

Expanding computing power available today allows us to tackle numerical simulation problems in engineering design, beyond the traditional separation between fluid flow and structural analysis. Multi-physics simulations provide increased fidelity by capturing the dynamics of interaction between physical models but also bring new challenges. This involves not only numerical issues of model-to-model coupling but also bring up problems of software design and simulation setup.

In this paper we have reviewed the software design and coupling issues arising from multi-physics simulations in Computational Continuum Mechanics. Object-oriented approach is offered as an alternative to traditional functional implementation. Main objects present in the simulation are represented in software as classes, consisting of protected data and functions operating on it, and combine to create higher-level objects. This provides higher level of flexibility and allows us to implement and couple various physical models and coupling algorithms with more freedom.

The object model described in the paper is implemented in the Open Source C++ library OpenFOAM [6]. The library also contains various coupling and data interpolation tools and implements a number of discretisation methods and a wealth of physical models. Its capabilities in multi-physics simulations are illustrated on three coupled problems, including weakly and strongly coupled fluid structure interaction and a coupled free surface simulation with surfactant effects.

Acknowledgement

The author acknowledges the contribution of dr. sc. Željko Tuković of the Faculty Mechanical Engineering and Naval Architecture, University of Zagreb, Croatia and dr. Aleksandar Karač of the University of Zenica, Bosnia and Herzegovina and University College Dublin, Republic of Ireland in the preparation of the material. Special mention is due to the research group of prof. Alojz Ivanković, University College Dublin, for their contribution to the structural analysis and FSI capabilities of OpenFOAM. Thanks are also due to all contributors to the OpenFOAM project, which has made this work possible.

References

- [1] Programming languages – C++. ISO/IEC Standard 14822:1998, 1998.
- [2] H. Jasak. Error analysis and estimation in the Finite Volume method with applications to fluid flows. PhD thesis, Imperial College, University of London, 1996.
- [3] H. Jasak and Ž. Tuković. Automatic mesh motion for the unstructured finite volume method. Submitted to J. Comp. Physics, Feb 2004.
- [4] F. Juretić. Error Analysis in Finite Volume CFD. PhD thesis, Imperial College, University of London, 2004.
- [5] A. Karač. Drop impact of fluid-filled polyethylene containers. PhD thesis, Imperial College, University of London, 2003.
- [6] OpenFOAM project web pages. <http://www.openfoam.org>, 2004.
- [7] Open Source initiative web pages. <http://www.opensource.org>.
- [8] Ž. Tuković. Finite Volume Method on Domains of Varying Shape. PhD thesis, Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, 2005. In Croatian.
- [9] J.M. Weiss, J.P. Maruszewski, and W.A. Smith. Implicit solution of preconditioned Navier-Stokes equations using algebraic multigrid. AIAA Journal, 37(1):29–36, 1999.
- [10] H.G. Weller, G. Tabor, H. Jasak, and C. Fureby. A tensorial approach to computational continuum mechanics using object orientated techniques. Computers in Physics, 12(6):620 – 631, 1998.

Abstract

In the pursuit of improved performance of engineering equipment through numerical simulation, it is often necessary to account for interaction between physical models previously studied in isolation. Multi-physics simulations bring new complexity in supporting multiple models within a single simulation, data mapping, coupling algorithms and problem setup. This forces us to re-think the issues of model representation and software design.

In this paper, we shall present a novel framework for model implementation in Computational Continuum Mechanics through object-oriented software design. Ease of implementation is achieved by mimicking the form of the partial differential equation in software. This approach, implemented in OpenFOAM, has proven sufficiently elegant and efficient for a general purpose numerical solver. The library supports 3-D polyhedral meshes with mesh motion support and contains data interpolation tools as needed for multi-physics simulations. Shared components and generic tools make it ideal for handling complex coupled cases. Coupling approach and library capabilities are illustrated on three simulations involving interacting physical models and fluid-structure interaction.

Hrvoje Jasak

Wikki Ltd. 33 Palmerston House, 60 Kensington Place, London, W8 7PU, United Kingdom;
Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Ivana Lučića 5, Zagreb, Croatia; E-mail: h.jasak@wikki.co.uk

Robotic Applications powered by Compliant Actuators

Dirk Lefeber, Björn Verrelst, Ronald Van Ham, Pieter Beyl, Bram Vanderborght, Michaël Van Damme, Joris Naudet, Rino Versluys

Keywords: Compliant actuation, adaptable compliance, assistive (medical)robotics

Abstract – Variable compliant actuation principles emerge in modern robotic applications. This type of actuation provides some specific features to a robotic system, such as chock absorption, safer human/robot interaction and lower energy consumption. A survey on existing variable compliant actuator types is given, followed by a detailed description of two actuation principles developed in our research group: an antagonistic setup of two Pleated Pneumatic Artificial Muscles (PPAM) and the Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator (MACCEPA). Subsequently, several robotic implementations of these two actuators are discussed briefly.

1. Introduction

Currently, robotics is one of the fields undergoing great scientific innovation and expansion. Automation, via robotic applications, in production industries has grown increasingly. In particular, multifunctional manipulators executing several industrial assembly tasks, i.e. large, fixed-base industrial robots executing “pick-and-place” tasks, with control strategies focused on precision end-effect positioning. Additionally, high precision robotic applications in the framework of medical (surgical) and microchips production are needed especially regarding high precision position control. Electrical motors combined with adapted reduction elements envelop the most commonly used actuator technology at the moment. The developed control algorithms are specifically optimized for position control, possibly in combination with force control. New developments in the area of robotics, however, have currently allowed for the development of multi-legged mobile robots and service robots oriented towards closer human/robot interaction. Especially in Japan, where a large number of corporations each work on a private humanoid robot, strong developments steer towards new applications in social service, domotics, rehabilitation and prosthesis. Important is that these domains require distinct features from the robotic system, such as special dynamical properties for reasons of chock absorption, energy consumption, and flexible and safe human/robot interaction.

Present robots mainly are actuated with electrical drives, with the required reduction units, since this technology is well known. However, the use of these reduction units renders the joint to be more rigid, encompassing several disadvantages vis-à-vis applications of human/robot interactions. In this manner, the structure of the reduction elements is heavily burdened through jolting of highly reflected inertia values. Furthermore, this manner of actuation leaves small room for “soft” robot/human interaction and complicates the exploitation of the system’s natural dynamics. The latter encompasses generating robot movements closely associated to the natural, passive movement of the system. However, this requires compliant joints of which ideally stiffness can be

manipulated. In a similar way, humans set up their joint stiffness, through relaxation of the muscles, according to the action or task performed. One of the positive consequences of movement within the natural dynamics of the system is the decreased energy consumption, especially important for mobile robots that require an onboard energy source. If the desired movements fall within passive dynamics, certain dynamic limits of the actuator/application frame can moreover shift.

One way to deal with compliance control is by implementing active impedance control using standard electrical actuation. With this kind of control the stiffness is set artificially, but the response time of these systems is fairly low due to several limitations such as sensor resolution, motor control bandwidth, sensor noise, etc.... Moreover, the exploitation of natural dynamics, by means of energy storage in an elastic element, and reduction of shock effects is not possible with such an approach. Reduction of impact effects is essential first for protection with respect to the robot structure itself, e.g. during touch-down impacts of multi-legged robots, and secondly for protection of humans, close to the robot, during unplanned interactions. An alternative for the implementation of compliance control, which tackles the disadvantages previously mentioned, is the use of actuation with inherent mechanical impedance control.

In the next sections an overview of existing variable passive compliant actuator types is given, followed by a detailed description of two actuation principles developed by the Robotics & Multibody Mechanics Research Group: an antagonistic setup of two Pleated Pneumatic Artificial Muscles (PPAM) and the Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator (MACCEPA). Subsequently, several robotic implementations with these two actuators are discussed briefly.

2. Brief Survey on Variable Compliant Actuators

An important contribution in the research towards soft actuators has been given by Pratt et al. with the development of the "Series Elastic Actuator" (SEA) [1]. It consists of a motor drive in series with a spring. The SEA is a linear actuator which uses a linear spring coil, but the same principle can be implemented using a torsional spring to create a rotational SEA. Numerous implementations have been made with such structures, e.g. the planar bipedal robot "Spring Flamingo" [2], the hopping monoped at ARL [3] and the DM² actuation principle [4] for manipulators, focussing on safety. The main disadvantage for the SEA principle is the compliance being constant. Currently, more and more research is focussing on actuators with adaptable compliance.

There are several ways to implement variable passive compliance, which can be categorized in two main groups [5]. The first group of adaptable compliant actuators can be described as *an antagonistic setup of 2 non-linear springs*. The idea is that two actuators with a non-adaptable compliance, characterized by a non-linear force-displacement course, are coupled antagonistically, which means that they work against each other. By controlling both actuators, the compliance and position of this antagonistic setup can be set. A second group uses the *structural Stiffness Control*. By varying the dimensional properties of e.g. a (leaf) spring, like length or moment of inertia, the stiffness constant of a spring can be adapted. For example, when an elastic beam is clamped at one side, the other end will behave compliant.

In the first group, *antagonistic setup of 2 non-linear springs*, several different designs can be mentioned. It is easily shown that non-linear springs have to be used to achieve variable compliance, the use of linear springs does not allow for compliance adaptation. English and Russell [6] describe theoretically in detail an antagonistic setup of two "quadratic" springs, for such a setup the stiffness and deflection are decoupled (linear torque to angle relation at fixed compliance). The "Biologically Inspired Joint Stiffness Control" [7] is a practical implementation using a complex mechanism in an attempt to create such "quadratic" springs in an antagonistic setup. Another approach for a non-linear antagonistic spring setup involves the "Actuator with Mechanically Adjustable Series Compliance" (AMASC) [8]. It has fibreglass springs with a large energy storage

capacity. The mechanism has two motors, one for moving the position and the other for controlling the stiffness. But the complete system is rather complex and heavy. A more compact design is the novel electro-mechanical “Variable Stiffness Actuation” (VSA) motor [9] of the university of Pisa, which has been designed for safe and fast physical human/robot interaction. It belongs to the category of antagonistic setup but it is less obvious than in the other designs. A timing transmission belt connects nonlinearly the main shaft to an antagonistic pair of actuator pulleys connected to position-controlled back-drivable DC motors. The belt is tensioned by springs. Concordant angular variations control displacements of the main shaft, while the opposite variations of the two DC motors generate stiffness variations. Instead of using mechanical springs several implementations involve pneumatics. The compressibility of air for a closed pneumatic cylinder typically generates a non-linear spring characteristic. Now, instead of using standard pneumatic cylinders many research groups implement pneumatic artificial muscles in an antagonistic setup, for which McKibben Artificial muscle [10] types are widely used. The last decades a lot of implementations, using pneumatic artificial muscles for robotics applications, have been made [11, 12, 13, 14, 15, 16], although not all of them specifically focus on compliance variation. An example of the latter has been developed by Bicchi and Tonietti [17], and a discussion will be given in section 3 on the antagonistic compliance implementation with the novel Pleated Pneumatic Artificial Muscles (PPAM) [18], developed by our research group.

As an alternative to the antagonistic setup of two non-linear springs, variations in stiffness can also be achieved through manipulation of the effective structure of a spring, also called *Structure Controlled Stiffness* as discussed clearly by Hollander and Sugar [19]. E.g. bending a leaf spring is a form of storing energy. However, apart from simple loading (energy storage) and unloading (energy return) of the leaf spring, an extra step of altering the leaf spring’s effective stiffness is introduced. To understand the basic concepts of a SCS actuator, consider the small deflection beam equation [19]:

$$M = \left(\frac{EI}{L} \right) \cdot \theta \quad (1)$$

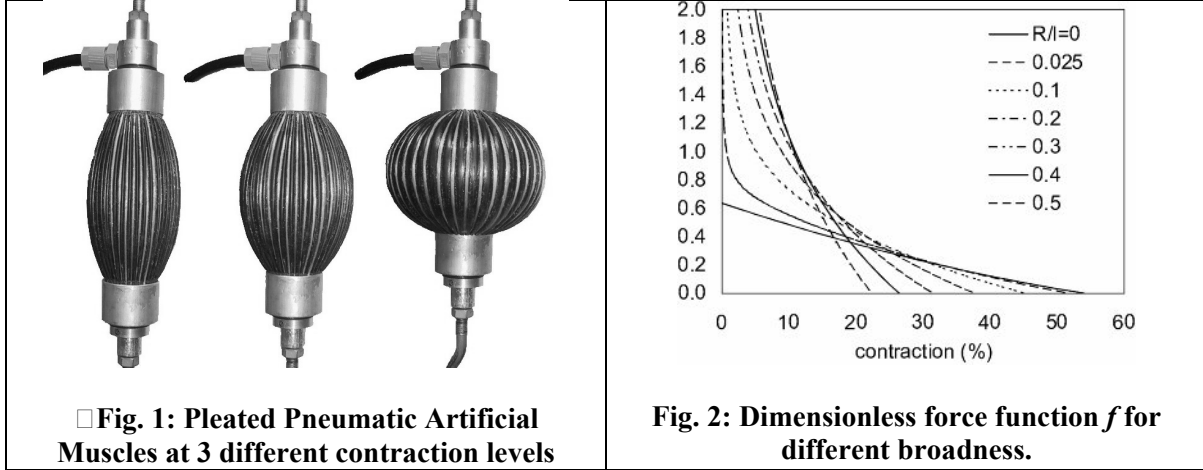
M is the bending moment, E the material modulus, I the cross-sectional moment of inertia, L the effective beam length, and θ is the angle of bending or slope. In this representation of bending, the term EI/L represents the bending stiffness. In order to control the stiffness of the structure, the main two parameters to be altered are the effective length of the spring L , and the cross-sectional inertia I . The latter for example can be changed by merging several sheets of elastic material. The idea is to increase the stiffness of such element by pressing the sheets together, for which several methods can be applied. In [20] electrostatic forces are imposed, while in [21] vacuum is used. Another very simple approach to change the rectangular cross-sectional inertia of a spring leave with respect to the deflection angle is to rotate the leave around its longitudinal axis. Hollander et al. applied this idea for a wearable orthosis, in which the rotating leave spring is packed inside a normal coil spring to avoid buckling [19]. The bending stiffness in equation (1) can also be changed by varying the actual length of a leave spring. An interesting implementation of this idea has been done for the “Wendy” robot [22] at the Sugano lab with the so called “Mechanically Impedance Adjuster” [23]. The “Jack Spring” actuator [24] on the other hand, uses changes in active length of a normal spring coil to make the compliance variable.

Another novel type of passive variable compliant actuation mechanism has been developed by our research group, the Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator (MACCEPA). This type actuator falls outside the two categories mentioned earlier, since it uses a specific geometrical positioning of one compliant spring element. A description of this actuator is given in section 4.

3. Antagonistic setup with Pleated Pneumatic Artificial Muscles

A. Pleated Pneumatic Artificial Muscle

A pneumatic artificial muscle is essentially a membrane that will expand radially and contract axially when inflated, while generating high pulling forces along the longitudinal axis. Different designs have been developed. The best known is the so called McKibben muscle [10]. This muscle contains a rubber tube which expands when inflated, while a surrounding netting transfers tension. Hysteresis, due to dry friction between the netting and the rubber tube, makes control of such a device rather complicated. Typical for this type of muscle is a threshold level of pressure before any action can take place. The main goal of the new Pleated Pneumatic Artificial Muscle design (PPAM) [18] was to avoid friction, thus making control easier while avoiding the threshold. This was achieved by arranging the membrane into radially laid out folds that can unfurl free of radial stress when inflated. Tension is transferred by stiff longitudinal fibres that are positioned at the bottom of each crease.



A photograph of the inflated and deflated state of the Pleated Pneumatic Artificial Muscle is given in Fig. 1. If the influence of elasticity of the high tensile strength material used for the fibres is omitted, the characteristic for the generated force is given by:

$$F = pl^2 f(\varepsilon, R/l) \quad (2)$$

where p is the applied gauge pressure, l the muscle's full length, R its unloaded radius and ε the contraction. The dimensionless function f , which depends only on contraction and geometry, is given for different values of broadness R/l in Fig. 2.

The thicker the muscle, the less it contracts and the higher the forces it generates. Contraction can reach up to 54% in a theoretical case with $R/l=0$. At low contraction forces are extremely high causing excessive material loading, and the generated forces drop too low for large contraction. Thus contraction is bounded between two limits, 5 and 35%, in practise.

The graph in Fig. 3 gives the generated force for different pressures of a muscle with initial length 11cm and unloaded diameter 2.5cm. Forces up to 5000N can be generated with gauge pressure of only 3bar while the device weighs about 100g.

B. Antagonistic Muscle Setup: Generated Torque

Pneumatic artificial muscles can only pull. In order to have a bi-directionally working revolute joint one has to couple two muscles antagonistically. The muscle connections -pull rods and lever mechanism- are designed such that the muscle's highly non-linear force-length characteristic is transformed to a more flattened torque-angle characteristic. Fig. 4 shows the straightforward connecting principle.

Depending on the desired function of the joint, the dimensions can be chosen in order to meet the needs of the specified joint function, not only in torque levels but also in range of motion. The torque characteristics of e.g. the pendulum setup discussed in section IV are shown in Fig. 5.

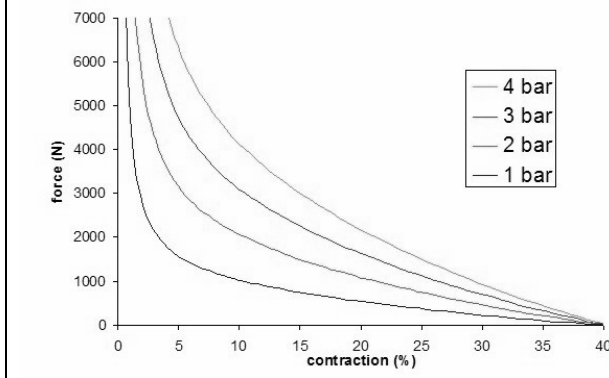


Fig. 3: Generated force of a muscle with initial length 11cm and unloaded diameter 2.5cm.

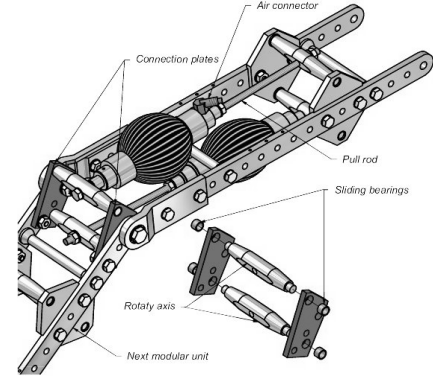


Fig. 4: Straightforward connection of the antagonistic muscle setup.

Both torques are given for different pressure values. Taking into account equation (2) and if r_1 and r_2 define the leverage arm of the agonist and antagonist muscle respectively, the joint torque (T) is given by following expression:

$$T = T_1 - T_2 = p_1 l_1^2 r_1 f_1 - p_2 l_2^2 r_2 f_2 = p_1 t_1(\theta) - p_2 t_2(\theta) \quad (3)$$

with p_1 and p_2 the applied gauge pressures in agonist and antagonist muscles respectively which have lengths l_1 and l_2 . The dimensionless force functions of both muscles are given by f_1 and f_2 . The functions t_1 and t_2 , in equation (3), are determined by the choices made during the design phase and depend on the joint angle θ . Thus joint position is influenced by weighted differences in gauge pressures of both muscles.

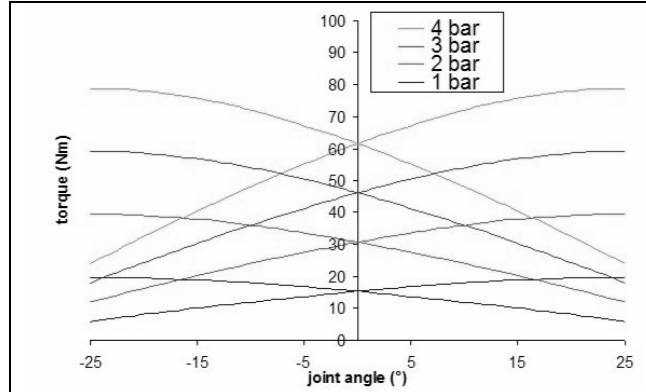


Fig. 5: Generated torques by each muscle of an antagonistic setup at different pressure levels

C. Antagonistic Muscle Setup: Compliance

The PPAM has two sources of compliance: gas compressibility and the dropping force to contraction characteristic. The latter effect is typical for pneumatic artificial muscles while the first is similar to standard pneumatic cylinders. Joint stiffness, the inverse of compliance, for the considered revolute joint can be obtained by the angular derivative of the torque characteristic in equation (3):

$$K = \frac{dT}{d\theta} = \frac{dp_1}{d\theta} t_1 + p_1 \frac{dt_1}{d\theta} - \frac{dp_2}{d\theta} t_2 - p_2 \frac{dt_2}{d\theta} \quad (4)$$

The terms $dp_i/d\theta$ represent the share in stiffness of changing pressure with contraction, which is determined by the action of the valves controlling the joint and by the thermodynamic processes taking place. If the valves are closed and if we assume polytropic compression/expansion the pressure changes inside the muscle are a function of volume changes:

$$P_i V_i^n = P_{i_0} V_{i_0}^n \quad \text{with} \quad P_i = p_i + P_{atm} \quad (5)$$

leading to:

$$\frac{dp_i}{d\theta} = -n(P_{atm} + p_{i_0}) \frac{V_{i_0}^n}{V_i^{n+1}} \frac{dV_i}{d\theta} \quad (6)$$

With P_i , V_i the absolute pressure and volume of muscle i , P_{i_0} the absolute initial pressure, V_{i_0} the initial volume when muscle i was closed and p_i , p_{i_0} the gauge pressure and initial gauge pressure. n is the polytropic index and P_{atm} the atmospheric pressure.

Taking the torque characteristics as an example, the following reasoning can be made for closed muscles: an increase of the angle θ results in an increase of the torque generated by one muscle (Fig. 5) while its volume decreases (at -25° this muscle is completely bulged and generates low forces). Thus $dt_1/d\theta > 0$ and $dV_1/d\theta < 0$. For the other muscle the actions will be opposite. Combining equations (4), (5) and (6) with this information gives:

$$K = k_1 p_{1_0} + k_2 p_{2_0} + k_{atm} P_{atm} \quad (7)$$

with:

$$k_i = n t_i \frac{V_{i_0}^n}{V_i^{n+1}} \left| \frac{dV_i}{d\theta} \right| + \frac{V_{i_0}^n}{V_i^n} \left| \frac{dt_i}{d\theta} \right| > 0 \quad i = 1, 2 \quad (8)$$

$$k_{atm} = k_1 + k_2 - \left| \frac{dt_1}{d\theta} \right| - \left| \frac{dt_2}{d\theta} \right|$$

The coefficients k_1 , k_2 , k_{atm} are a function of the angle and are determined by the geometry parameters of the joint and muscles. From equation (7) the conclusion is drawn that a passive spring element is created with an adaptable stiffness controlled by the weighted sum of both initial gauge pressures when closing the muscle. Since stiffness is depending on a sum of gauge pressures while position is determined by differences in gauge pressure, the angular position can be controlled while setting stiffness.

D. Antagonistic Muscle Setup: Trajectory Control with Compliance Setting

This paragraph discusses a joint trajectory tracking controller for the antagonistic setup of which the separate units are depicted in Fig. 6. The tracking controller is divided into a computed torque controller, a delta-p unit and a pressure bang-bang controller. And in parallel compliance setting can be provided by the compliance controller depending on the application. The computed torque controller calculates the required joint torque based on the robot dynamics. The delta-p unit translates the calculated torques into desired pressure levels for the two muscles of the antagonistic set-up. Finally the bang-bang pressure controller determines the necessary valve actions to set the correct pressures in the muscles.

Computed Torque

In the first module, the joint torques are calculated using the popular computed torque technique consisting of a feedforward part and a PID feedback loop. The specific formulation of the equations depend on the robotic application.

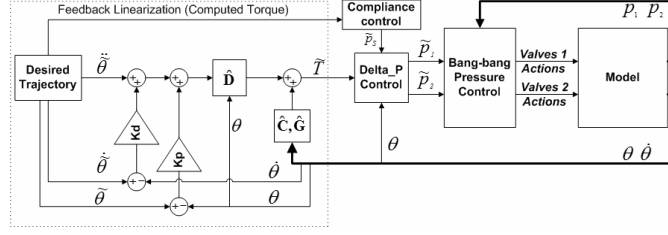


Fig. 6: Trajectory tracking control scheme

Delta-p Unit

The computed torque $\tilde{T}$ is fed to the delta-p control unit, which calculates the required pressure values to be set in the muscles. These two gauge pressures are generated as follows:

$$\begin{aligned}\tilde{p}_1 &= \frac{\tilde{p}_s}{\hat{i}_1(\theta)} + \Delta\tilde{p} \\ \tilde{p}_2 &= \frac{\tilde{p}_s}{\hat{i}_2(\theta)} - \Delta\tilde{p}\end{aligned}\quad (9)$$

with p_s a parameter that is used to influence the sum of pressures and consequently the joint stiffness, and Δp a parameter that influences the difference in pressure of the two muscles in order to control the generated torque. The functions $\hat{i}_i(\theta)$ ($i=1,2$) represent the torque characteristics of the antagonistic muscle setup and are calculated with estimated values of the muscle force functions and geometrical parameters. Expression (3) allows to link the required torque to the required pressure values in the muscles:

$$\tilde{T} = \tilde{p}_1 \hat{i}_1(\theta) - \tilde{p}_2 \hat{i}_2(\theta) = (\hat{i}_1(\theta) + \hat{i}_2(\theta)) \Delta\tilde{p} \quad (10)$$

If the calculated pressure values $\tilde{p}_i$ ($i=1,2$) of equations (9) are set in the muscles, the generated torque depends only on $\Delta\tilde{p}$ and is independent of the joint stiffness parameter p_s , in case the modelling would be perfect. This means that joint stiffness is changed without affecting the joint angular position. Feeding back the knee angle θ and introducing the torque $\tilde{T}$, expression (10) can be used to determine the required $\Delta\tilde{p}$ for tracking purpose:

$$\Delta\tilde{p} = \frac{\tilde{T}}{\hat{i}_1(\theta) + \hat{i}_2(\theta)} \quad (11)$$

The delta-p unit is actually a feedforward calculation from torque level to pressure level, using the kinematic model of the muscle actuation system. The calculated $\Delta\tilde{p}$ affects the torque required to track the desired trajectory, while p_s is introduced to determine the sum of pressures which influences the stiffness of the joint. The p_s parameter is determined by the compliance control block and depends on the application. Currently, this compliance controller is used to calculate a suitable p_s in order to exploit natural dynamics, by continuously tuning the compliance of the joint to the required compliance of an imposed trajectory (see section 5A).

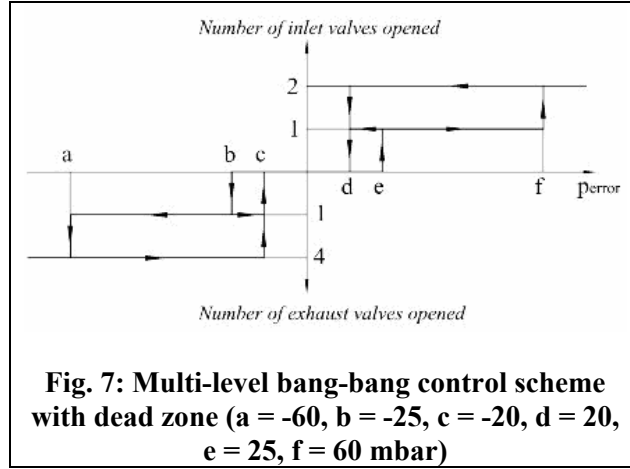
Bang-bang Pressure Controller

For some applications (e.g. biped, lightweight robot arm) the weight of the valves controlling the muscles should be taken as low as possible. But since most pneumatic systems are designed for fixed automation purposes where weight is not an issue at all, most off-the-shelf proportional valves are far too heavy for this application. Thus a proper design of the pressure control is of great importance. In order to realize a lightweight rapid and accurate pressure control, fast switching on-off valves are used. The pneumatic solenoid valve '821 2/2 NC' made by Matrix weighs only 25g.

With their reported opening times of about 1ms and flow rate of 180 Stdl/min, they are about the fastest switching valves of that flow rate currently available.

To pressurize and depressurize the muscles, which have a varying volume up to 400ml, a number of these small on-off valves are placed in parallel. Obviously the more valves used the higher the electric power consumption, the price and also the weight will be. Simulations of the pressure control of a constant volume resulted in a choice of two inlet and four outlet valves. The different number between inlet and outlet comes from the asymmetric pressure conditions between inlet and outlet, combined with the aim to create equal muscle's inflation and deflation times.

The pressure control in a volume is achieved with a bang-bang controller with various reaction levels depending on the difference between measured and desired pressure (Fig.7). If this difference is large, two inlet or four outlet valves, depending on the sign of the error, are opened. If this difference is small, only one valve is switched and when the error is within reasonable limits no action is taken, leaving the muscle closed. More detailed information on the valve system can be found in reference [25].



4. Mechanically Adaptable Compliance and Controllable Equilibrium Position Actuator (MACCEPA)

Considering the possible drawbacks of a pneumatic system, namely the high cost of pressurized air production and supply requirements for autonomy, our research group has recently developed another compliant actuator type: the Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator (MACCEPA). The primary design goals are a straightforward mechanical concept and a straightforward relation between the mechanical parameters and the control parameters.

Fig. 8 shows a schematic drawing of the MACCEPA prototype design depicted in Fig. 9. Three bodies are pivoting around a common rotation axis in point a , of which the smallest one is the lever arm ab . A spring is attached between a fixed point b on the lever arm and a cable running around a fixed point c on the right body to a pre-tension mechanism. The angle ϕ between the lever arm and the left body, is set by an electrical actuator, which in fact sets the equilibrium position. A second actuator, either a rotational or a linear actuator, pulling the cable connected to the spring, sets the pre-tension of the spring. The pre-tension will vary the torque at a certain angle α , in this way determining the spring constant of an equivalent torsion spring. As a result both the equilibrium position and the joint's compliance can be adapted independently by means of two control parameters, namely the angle ϕ and the amount of pre-tension in the linear spring respectively, each of them being set by a separate servo motor.

Equation (12) calculates the generated torque, which is independent of the angle ϕ and set by lengths B and C , spring stiffness constant k and the pre-tension variable P , which is the extension of the spring caused by pre-tensioning. The actuator torque is a result of the combined effect of pre-

tension and the length change of bc [26]. For C/B ratio's greater than 5, the torque-angle characteristic of the MACCEPA becomes quasi linear for a large range of α ($-60^\circ < \alpha < 60^\circ$)

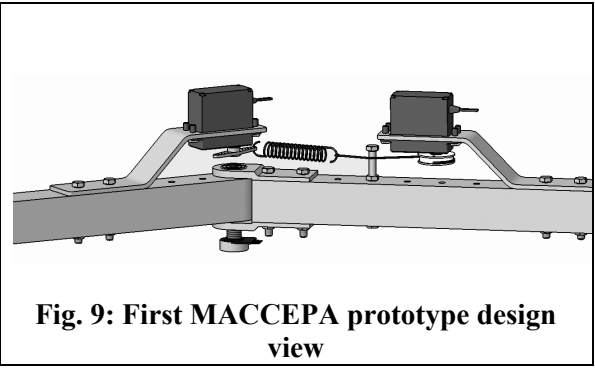
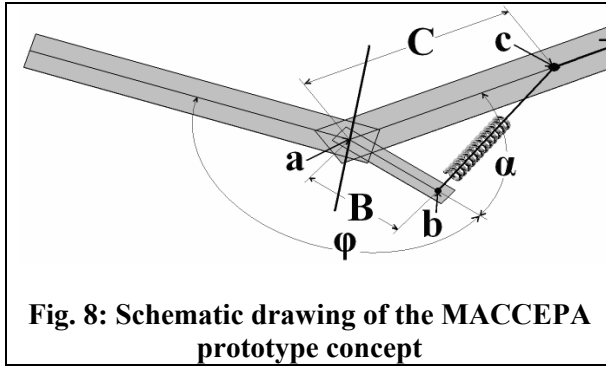


Fig. 10 depicts the torque characteristics for different pre-tension length values P and a specific set of B , C and k values.

$$T = k.B.C.\sin\alpha.\left(1 + \frac{P - |B - C|}{\sqrt{B^2 + C^2 - 2.B.C.\cos\alpha}}\right) \quad (12)$$

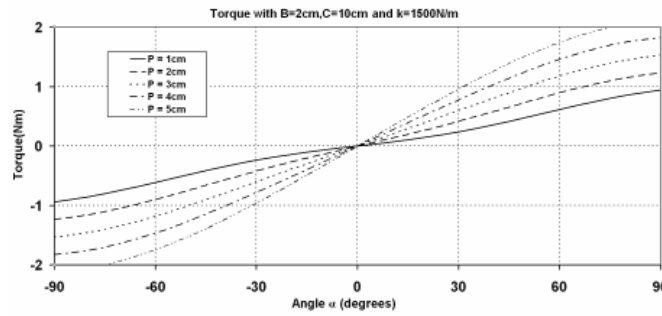


Fig. 10: Generated torque as a function of α at different pre-tension length values P for a MACCEPA with $B=0.02m$, $C=0.1m$, $k=1500N/m$

Within the quasi-linear working range linearization of equation (12) yields the linearized torque formula (13) equivalent to the angle-torque relation of a linear torsion spring with an adaptable spring constant.

$$T = \alpha.\mu.P \quad \text{with} \quad \mu = \frac{k.B.C}{|B - C|} \quad (13)$$

In this way the requirement of a straightforward control law is clearly met. Unlike the PPAM powered rotational actuator, the control signals for the equilibrium position and the compliance are independent of the actuator's actual position, which further simplifies the controller. Furthermore, standard servo motors are used for positioning of the equilibrium pose and tensioning the spring. Video material showing the working principle and behaviour of the MACCEPA is available on the website <http://mech.vub.ac.be/maccepa>.

5. Variable Compliant Actuators for Bipedal robots

In bipedal walking robot design the mechanisms behind human walking evidently serve as the primary source of inspiration. Joint compliance plays a crucial role in human locomotion both to reduce energy consumption and adapt gait patterns.

While running tendons and ligaments act as spring-like elements, storing and releasing elastic strain energy and in this way reducing the work required by the muscles [27]. Because the movement of the centre of mass during running is similar to a bouncing ball, running is often referred to as *bouncing gait*. This makes it possible for the human body to operate at an efficiency of 40-50%, while the maximum efficiency of a contracting muscle is only about 25%. The global compliance of the legs is used to adapt gait parameters. Increasing the compliance, for instance, results in higher stride frequencies [28].

While walking, different energy saving mechanisms are called upon, of which the most important one is often referred to as the *inverted pendulum mechanism*. About 60% of the mechanical energy, required to lift and accelerate the centre of mass, is being conserved [29]. This observation has led to the development of passive walking mechanisms, the so called *passive walkers* [30]. Joint compliance reduces the impact at heel strike during walking and induces passive motion, in turn reducing motor control and muscle work.

In summary, bipedal walking robots clearly benefit from compliant actuation because of the lowered reflected inertia and the energy storage capability. Not only because it results in energy-efficient and human-like locomotion, but also for a higher level of safety in human-robot interaction.

Our research group has designed and built two bipedal walking robots: “*Lucy*”, actuated by PPAM’s and “*Veronica*”, powered by MACCEPA’s. In the next sections a brief description is given of the two biped projects and the concepts and strategies involved.

A. The Biped “Lucy” Actuated by PPAM’s

“*Lucy*” (Fig. 11) is a two dimensional walking robot with two articulated legs and an armless body. Each joint module, namely the ankle, the knee and the hip joint, is powered by an antagonistic pair of pleated pneumatic artificial muscles. As the robot’s walking motion is restricted to the sagittal plane a guiding mechanism has been incorporated, preventing the robot from falling sideward. A treadmill is used for long distance walking and its speed is controlled according to the biped’s walking speed.

The goal of the project is to achieve a lightweight bipedal walking robot able to walk in a dynamically stable way while exploiting the adaptable compliance of the pleated pneumatic artificial muscles in order to reduce energy consumption and control efforts [31]. A model-based joint trajectory tracking control strategy is used in combination with real-time planar trajectory generation.

The joint trajectory tracking controller combines a computed torque controller, a Δp -unit and a pressure bang-bang controller similar to the one discussed in section II. The computed torque controller calculates the required joint torques out of the robot’s dynamics. This dynamic model is different for the single and the double support phase. In single support the robot has 6 DOF, in double support the number of DOF is reduced to 3, which makes the system over-actuated. The Δp -unit translates the calculated joint torques into the required pressure levels of the two muscles of each antagonistic set-up. The bang-bang controller determines the control signals to be sent to the valves, setting the required pressures in the muscles. The trajectory generator and both the computed torque and Δp -units are implemented on a central PC, the bang-bang controller is locally implemented on micro-controller units, resulting in a global control frequency of 2000Hz. For more information about this control architecture the reader is referred to [32].

The current trajectory generator is based on the inverted pendulum approximation, which models the robot as a single point mass. It keeps the zero moment point in the ankle point during the single support phase, it provides a smooth transition of the ZMP from the rear ankle point to the front ankle point during the double support phase and it allows the step-length, intermediate foot lift and velocity to be chosen differently for each step. The hip is not moving on a straight line as in most of the inverted pendulum based methods, since the strategy uses quasi-stretched knees for more

naturally looking walking and reduced knee torques. With this strategy the robot is able to walk with different step-lengths and at different speeds, having a maximum speed of 0.11m/s.

Besides a trajectory generator and tracking controller, a compliance controller is to be developed to exploit the benefits of adaptable compliance. For this purpose an online compliance adaptation algorithm has been developed and verified on a single pendulum test setup [33], depicted in Fig. 12. The algorithm aims at tuning the stiffness of the PPAM's in such a way that it fits the natural stiffness of the desired trajectory, which is calculated as the derivative of the torque necessary to track the desired trajectory with respect to the joint angle. This approach, in combination with trajectory tracking, will result in semi-passive walking. Experimental validation of the single DoF joint shows the effectiveness of the algorithm, however, additional testing on multi-joint systems is necessary. Moreover the algorithm calculates a local optimum in energy-efficiency, bounded by the underlying concept of the tracking controller and the trajectory generator. Video material is available on the website of the “Lucy” (<http://lucy.vub.ac.be>).



Fig. 11: Bipedal walking robot “Lucy” actuated by PPAM’s

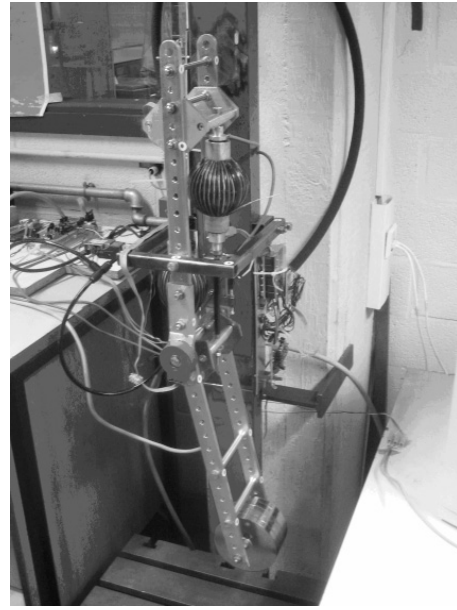


Fig. 12: One DoF pendulum actuated by PPAM’s.

B. The Biped “Veronica” Actuated by MACCEPA’s

“Veronica” (Variable joint Elasticity Robot with a Neuro-Inspired Control Approach) is a planar biped, like “Lucy”, with 6 degrees of freedom (Fig. 13) [34]. The hip, knee and ankle joints of the legs are pin joints, each actuated by a *Slimline* MACCEPA [34] (Fig. 14). This MACCEPA-type actuator, consisting of two servo motors incorporated in the link structure, is a more compact variant of the first prototype.

The goal of this project is twofold: the application is meant both to serve as a test-bed for the MACCEPA and to verify a novel control approach, different from the one implemented in “Lucy” and largely based on the concept of passive walking. In unactuated passive walking the cyclic motion is typically sustained by gravitation on a shallow slope and its frequency is determined by the mechanical properties of the passive walker. Applying this concept to level-ground walking, one needs to take into account friction and impact losses and therefore introduce actuation. Most of the existing walking robots, based on the concept of passive walking, have fixed compliance properties [35] or a small number of different compliance settings [36], resulting in one single stride frequency.

The aim of this biped project is to expand the range of natural walking motions by means of adaptable compliance. The natural joint trajectories of the system can be adapted at will by systematically setting the equilibrium positions and compliance values of the different joints, resulting in a variety of semi-passive gait patterns. For this purpose the required gait pattern is divided into distinct phases (e.g. swing leg phase, stance leg inverted pendulum motion phase,...), each of them corresponding with a natural trajectory and consequently a set of control parameters. At phase transition, the control parameters associated with a particular phase are set and the system performs the required natural motion accordingly.

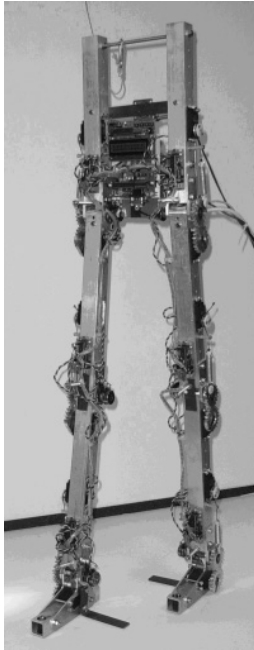


Fig. 13: Bipedal walking robot “Veronica” actuated by MACCEPA’s

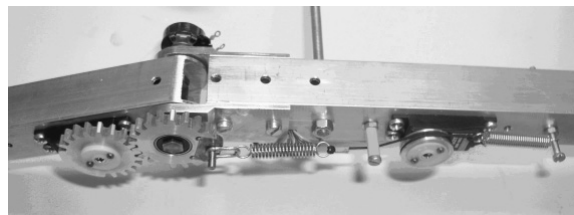


Fig. 14: *Slimline* MACCEPA variant

The approach described above is referred to as *Controlled Passive Walking*. Fig. 15 illustrates the concept, applied to a single joint, actuated by a *slimline* MACCEPA, of which the upper link is considered fixed. Three situations *A*, *B* and *C* are depicted, all of them with the same settings of compliance and equilibrium position, but with different lower link positions. *B* shows the state in which no external torque is applied to the joint. By means of an external torque the link position can be changed from the equilibrium position to the position in state *A*. If the link, having zero angular velocity, is released from this state, i.e. by removing the external torque, it will swing, in the absence of friction, to *C*, the mirrored state of *A*. In *C* the link has zero angular velocity and the experiment can be repeated. Without intervention, one obtains an oscillating spring-mass-system.

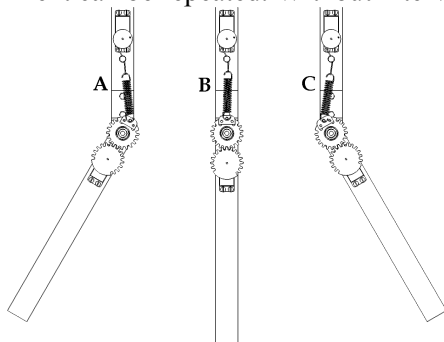


Fig. 15: *Slimline* MACCEPA actuated joint in three different states *A* (-30°), *B* (0°), *C* (30°)



Fig.16: Soft robotic arm small-scale model

The time elapsed between A and C depends both on the link's moment of inertia and on the joint's compliance, set by the MACCEPA. The end position in state C depends on the equilibrium position of the MACCEPA. Given a starting position, one can achieve a desired end position in a desired time within a certain range simply by setting the compliance and the equilibrium position at the beginning of the swing phase.

Some preliminary walking experiments on “*Veronica*” show promising results. In these experiments control parameters were found by trial-and-error. Through observation of slow motion replays of the walking motion, the different parameters were tuned. The duration of the swing phase for instance was easily adjusted by changing the compliance of the hip joint. Video material of the walking experiments, based on this intuitive control method, is available on the website <http://mech.vub.ac.be/maccepa>.

6. Variable Compliant Actuators in Assistive Technology and Rehabilitation

Compliant actuator technology has another broad field of interest: assistive devices for use in industry and in non-industrial sectors, like healthcare.

A. Assistive Manipulator Technology

The use of compliant actuation enables new assistive manipulator designs, combining operator safety, low weight, ease of operation, ergonomics and low cost. The goal of our “*soft arm*” research project is to develop a proof-of-concept manipulator actuated with PPAM's that can be used in direct contact with an operator, without expensive force or torque sensors and without user interaction through control elements (such as joysticks). For this purpose a small-scale manipulator model, consisting of two links in inverse elbow configuration and actuated by PPAM's, has been developed (Fig. 16). The proposed concept of robot-assisted manipulation is based on both the manipulator and the operator holding and handling the load at the same time. This direct interaction between operator and manipulator allows the operator to position the load accurately while the weight is being carried by the manipulator. Using the muscles' gauge pressures in combination with joint position measurement to estimate the interaction forces of the operator, the system should be able to predict and generate the desired end effector path. In this way carrying most of the payload. Operator safety relies both on hardware properties, such as joint compliance and lightweight design, as on control strategy.

The use of PPAM's as compliant actuators and the highly non-linear dynamics of the system make controlling it a complex task. A PID controller with gravity compensation has been implemented for preliminary testing of the manipulator setup. This controller, however, is inadequate for tracking purposes. Instead, a sliding mode controller has been proposed and tested. For a more detailed description of the sliding mode controller the reader is referred to [37]. The gauge pressures, in turn, are set by off-the-shelf proportional pressure regulating valves with internal PID controllers.

Examples of the soft arm's behaviour and the underlying effects of compliance are available on the website http://mech.vub.ac.be/multibody/topics/soft_arm.htm.

B. Medical Assistive Device: Prosthesis

Compliant actuators seem well suited to meet the specific requirements of assistive devices for the disabled, for instance in the fields of prosthetics, orthotics and rehabilitation [19][38][39]. Safety is of primary concern in wearable robotic systems and medical rehabilitation robotics, but ergonomics, design complexity, size, weight and design functionality are equally important and moreover interdependent. In the case of lower limb prostheses, orthoses and step rehabilitation

devices the adaptable compliance is an additional advantage in mimicking human gait, as discussed in the section on bipedal walking robots.

The goal of our “*IPPAM*” project (Intelligent transtibial prosthesis actuated by Pleated Pneumatic Artificial Muscles) is to develop a transtibial prosthesis for below-knee amputees, able to adapt to the intentions of the wearer and to changing walking surface properties. The current state-of-the-art in prosthetics is limited to passive systems without compliant elements or with fixed compliance properties [40][41]. These restrictions might lead to an unnatural, uncomfortable gait and even result in medical pathologies on a longer term. Introducing variable compliant actuation enables specific active assistance, for instance at the toe-off phase and during transition of different modes, like starting and stopping, and provides a means to change the ankle joint’s compliance according to the stride frequency and the compliance characteristics of the ground surface [42][43]. The PPAM’s high force-to-weight ratio is an advantage towards lightweight and compact design. Currently, the project is in the mechanical design phase covering the PPAM type selection, the structural design, stress sizing and dynamics simulation. The next phase will focus on the measurement system and providing the basis for the prosthesis’ intelligence. Issues of pneumatic power generation and autonomy won’t be addressed extensively, as the first prototype will above all be a proof-of-concept for the integration of variable compliant actuation in prosthetics. After hardware completion and preliminary testing, the prosthesis’ performance will be evaluated on a patient group in lab settings both with objective and subjective outcome measures.

C. Rehabilitation Assistive Device

The compliant actuator technologies presented in this paper are not confined to the applications outlined in the previous sections. Adaptable compliance brings added value to other existing applications and creates new functionalities. A recent project, in start-off phase, focuses on the development of a PPAM actuated step rehabilitation robot (Fig. 17) assisting patients with gait problems, spinal cord injury patients for instance, recovering normal walking abilities. The compliant actuators provide enhanced ergonomics and passively absorb possible spasmodic movements. The adaptable compliance serves as a tool to alter the gait pattern and gradually change the level of assistance during therapy.



Fig. 17. Artistic view of the step rehabilitation robot actuated by PPAM’s

On the other hand, the PPAM’s high force-to-weight ratio enables the actuated orthosis to provide full bodyweight support. This makes for a small step towards a fully autonomous exoskeleton. Existing exoskeletons however are partially assistive devices of which the control performance relies on the postural stability and motor control of the wearer [44].

In general, variable passive compliant actuation creates an interesting approach to many emerging robotic technologies, both in industrial and non-industrial environments, where robots and human beings are in close interaction and accurate positioning or precise trajectory tracking is often

secondary.

7. Conclusion

This paper discussed the importance of variable passive compliant actuation. A survey on existing variable passive compliant actuator designs was given and two compliant actuator technologies, both developed within the Robotics & Multibody Mechanics Research Group at Vrije Universiteit Brussel, were discussed in detail: the pleated pneumatic artificial muscle (PPAM) and the mechanically adaptable compliance and controllable equilibrium position actuator (MACCEPA). An overview was given of the different research projects in which these compliant actuator types are incorporated and play a crucial role: bipedal walking robots “Lucy” and “Veronica”, a soft robotic manipulator and an intelligent transtibial prosthesis. Also a future research project in the field of step rehabilitation robotics was introduced.

Acknowledgement

B. Vanderborght is a Phd student with a grant from the Fund for Scientific Research-Flanders (Belgium)(FWO). “Lucy” has been built with the financial support of the Research Council (OZR) of Vrije Universiteit Brussel.

The “MACCEPA” research is supported by the LINK project of the IWOIB of the Brussels Capital Region (Belgium)

The “soft arm” project is supported by the Fund for Scientific Research-Flanders (Belgium, FWO) and by the Research Council (OZR) of Vrije Universiteit Brussel.

The “IPPAM” project is supported by the Fund for Scientific Research-Flanders (Belgium, FWO).

REFERENCES

- [1] Pratt, G. A. and Williamson, M. M., Series elastic actuators, *Proc IEEE-IROS Conference*, Pittsburg, USA, pp. 399-406, 1995.
- [2] Pratt, J. E. and Pratt, G. Exploiting natural dynamics in the control of a planar bipedal walking robot, *Proc of the 36th Annual Allerton Conf on Communication, Control, and Computing*, Monticello, Illinois, 1998.
- [3] Gregorio, P., Ahmadi, M. and Buehler, M., Design, control, and energetics of an electrically actuated legged robot, *IEEE Transactions on Systems, Man and Cybernetics* 27B(4): 626-634, 1997.
- [4] Zinn, M., Khatib, O., Roth, B. and Salisbury, K., A New Actuation Concept for Human-Friendly Robot Design, *IEEE Robotics and Automation Magazine*, June, pp. 12-21, 2004.
- [5] Van Ham, R. *Compliant Actuation for Biological Inspired Bipedal Walking Robots*, PhD thesis, Vrije Universiteit Brussel, 2006.
- [6] English, C. E. and Russel, D., Mechanics and stiffness limitations of a variable stiffness actuator for use in prosthetic limbs, *Mechanism and Machine Theory*, vol. 34, pp. 7-25, 1999.
- [7] Migliore, S. A., Brown, E. A. and DeWeerth, S. P. Biologically inspired joint stiffness control, *Proc IEEE Int Conf on Robotics and Automation, Barcelona, Spain, April*, pp. 4519-4524, 2005.
- [8] Hurst, J. W., Chestnutt, J. and Rizzi, A., An actuator with physically variable stiffness for highly dynamic legged locomotion, *Proc of the IEEE Int Conf on Robotics and Automation, New Orleans, USA*, pp. 4662-4667, 2004.
- [9] Bicchi, A. and Tonietti, G. Fast and soft arm tactics: Dealing with the safety-performance trade-off in robot arms design and control, in *IEEE Robotics and Automation Magazine*, vol. 11, pp. 22-33, 2004.
- [10] Schulte, H. F. The characteristics of the McKibben artificial muscle, *The Application of External Power in Prosthetics and Orthotics, number Publication 874*, National Academy of Sciences National Research Council, Lake Arrowhead, pp. 94-115, 1961.
- [11] Berns, K., Albiez, J., Kepplin, V. and Hillenbrand, C., Airbug insect like machine actuated by fluidic muscle, *Proceedings of the 4th International Conference on Climbing and Walking Robots, from Biology to Industrial Application*, Karlsruhe, Germany, pp. 237-244, 2001.
- [12] Davis, S. T. and Caldwell, D. G., The bio-mimetic design of a robot primate using pneumatic artificial muscle actuators, *Proc 4th Int Conf on Climbing and Walking Robots*, Karlsruhe, Germany, pp. 197-204, 2001.
- [13] Kingsley, D. A., Quinn, R. D. and Ritzmann, R. E. A cockroach inspired robot with artificial muscles, *Proceedings of the International Symposium on Adaptive Motion of Animal and Machines*, Kyoto, Japan, 2003.
- [14] Raparelli, T., Beomonte Zobel, P. and Durante, F., The design of a 2-dof robot for functional recovery therapy driven by pneumatic muscles, *Proceedings of the 10th International Workshop on Robotics in Alpe Adria Danube Region*, Vienna, Austria, 2001.
- [15] van der Linde, R. Q., Design, analysis and control of a low power joint for walking robots, by phasic activation of mckibben muscles., *IEEE Transactions on Robotics and Automation* 15(4): 599-604, 1999.
- [16] Wisse, M., Biologically inspired biped design, *Symposium of the Int. Soc. for Postural and Gait Research*, Maastricht, The Netherlands, pp. 900-903, 2001.

- [17] Bicchi, A. and Tonietti, G., Design, Realization and Control of Soft Robot Arms for Intrinsically Safe Interaction with Humans, *Proc. IARP/RAS Workshop on Technical Challenges for Dependable Robots in Human Environments*, pages 79-87, October, 2002.
- [18] Daerden F. & Lefeber D., *The concept and design of pleated pneumatic artificial muscles*, *International Journal of Fluid Power*, 2(3):41-50, 2001.
- [19] Hollander, K.W. and Sugar, T.G., Concepts for compliant actuation in wearable robotic systems, *US-Korea Conference*, North Carolina, August, 2004.
- [20] O. Tabata, S. Konishi, P. Cusin, Y. Ito, F. Kawai, S. Hirai, and S. Kawamura. Microfabricated Tunable Bending Stiffness Device. *Proc. of the 13th Annual Int. Conf. On Micro Electro Mechanical Systems*, Miyazaki, 1999.
- [21] S. Kawamura, T. Yamamoto, D. Ishida, T. Ogata, Y. Nakayama, Tabat and S. Sugiyama. Development of Passive Elements with Variable Mechanical Impedance for Wearable Robots. *Proceedings of the 2002 IEEE International Conference on Robotics & Automation*, Albuquerque, New Mexico, April, pp. 462-467, 1997.
- [22] Morita, T., Iwata, H., and Sugano, S., Development of Human Symbiotic Robot: WENDY, *Proc IEEE Int Conf on Robotics and Automation*, Detroit, Michigan, May, pp. 462-467, 1999.
- [23] Morita, T. and Sugano, S. Development and Evaluation of Seven-D.O.F. MIA ARM, *Proc IEEE Int Conf on Robotics and Automation*, Albuquerque, New Mexico, April, pp. 462-467, 1997.
- [24] Hollander, K.W., Sugar, T.G., and Herring, D.E., Adjustable Robotic Tendon using a 'Jack Spring'™, *Proc. IEEE International Conference on Rehabilitation Robotics*, Chicago, IL, USA, June, pp. 113-118, 2005
- [25] Van Ham R., Verrelst B., Daerden F., Vanderborcht B., and Lefeber D., Fast and accurate pressure control using on-off valves, *International Journal of Fluid Power*, vol. 6, March, pp. 53-58, 2005.
- [26] Van Ham R., Vanderborcht B., Van Damme M., Verrelst B., & Lefeber D., *MACCEPA: the Actuator with adaptable compliance for dynamic walking bipeds*, *CLAWAR 2005: 8th International Conference on Climbing and Walking Robots and the Support Technologies for Mobile Machines*, 12-15 September 2005 - London, U.K., p 759-766.
- [27] Alexander R. McN., *Energy-saving mechanisms in walking and running*, *J. exp. Biol.* 160, 55-69 (1991).
- [28] J.W. Hurst and A. Rizzi, Physically Variable Compliance in Running, *CLAWAR*, Springer-Verlag, www.springeronline.com, September, 2004.
- [29] R.R. Neptune, F.E. Zajac, S.A. Kautz, Muscle mechanical work requirements during normal walking: the energetic cost of raising the body's center-of-mass is significant, *Journal of Biomechanics* 37, 817-825, 2004.
- [30] T. McGeer, Passive dynamic walking, *Intern. J. Robot. Res.*, 9(2):62-82, April 1990.
- [31] Verrelst B., Van Ham R., Vanderborcht B., Daerden F. & Lefeber D., The Pneumatic Biped "LUCY" Actuated with Pleated Pneumatic Artificial Muscles, *Autonomous Robots* 18, pp.201-213, 2005.
- [32] Verrelst B., Vermeulen J., Vanderborcht B., Van Ham R., Naudet J., Lefeber D., Daerden F. & Van Damme M., Motion Generation and Control for the Pneumatic Biped "Lucy", *International Journal of Humanoid Robotics*, Vol. 3, No. 1, pp. 1-35, 2006.
- [33] Vanderborcht B., Verrelst B., Van Ham R., Van Damme M., Lefeber D., Meira Y Duran B. & Beyl P., Exploiting natural dynamics to reduce energy consumption by controlling the compliance of soft actuators, *The International Journal of Robotics Research (IJRR)*, Volume 25 Issue 4, april 2006, pp. 343-358.
- [34] Ronald Van Ham, Bram Vanderborcht, Björn Verrelst, Michaël Van Damme & Dirk Lefeber, MACCEPA: the Mechanically Adjustable Compliance and Controllable Equilibrium Position Actuator used in the 'Controlled Passive Walking' biped Veronica, *Proceedings of the 15th International Symposium on Measurement and Control in Robotics*, 08-10 November 2005, Brussels, Belgium.
- [35] Pratt, J. E. & Pratt, G. A., Exploiting Natural Dynamics in the Control of a Planar Bipedal Walking Robot, *Proceedings of the Thirty-Sixth Annual Allerton Conference on Communication, Control, and Computing*, pp. 739-748, 1998.
- [36] M. Wisse and J. van Frankenhuyzen, Design and construction of mike; a 2d autonomous biped based on passive dynamic walking, *Proc., Conference on Adaptive Motion of Animals and Machines, AMAM*, Kyoto, Japan, Paper number WeP-I-1, 2003.
- [37] M. Van Damme, R. Van Ham, B. Vanderborcht, F. Daerden and D. Lefeber, *Design and Control of a "Soft" 2-DOF Planar Pneumatic Manipulator*, *Proceedings of the 15th International Symposium on Measurement and Control in Robotics*, 08-10 November, Brussels, Belgium, 2005.
- [38] Wright, P.J. Virk, G.S. Gharooni, S.C. Smith, S.A. Tylor, R.I. Bradshaw, S. Tokhi, M.O. Jamil, F. Swain, I.D. Chappell, P.H. Allen, R., Powering and Actuation Technology for a Bio-Robotic Walking Orthosis, In: *Armada M A, Gonzalez de Santos P and Tachi S (eds.) Measurement and Control in Robotics, Produccion Grafica Multimedia*, ISBN 84 6079 693 0, pp.133-138, 2003.
- [39] Riener, R. Frey, M. Bernhardt, M. Nef, T. Colombo, G., Human-centered rehabilitation robotics, *9th International Conference on Rehabilitation Robotics*, 2005. *ICORR 28 June-1 July*, pp.319 - 322, 2005.
- [40] Geil, M. D., Energy loss and stiffness properties of dynamic analysis elastic response prosthetic feet, *Journal of Prosthetics and Orthotics*, 13(3):70-73, 2001.
- [41] Hafner, B. J., Sanders, J. E., Czerniecki, J. and Ferguson, J., Transtibial energy-storage-and-return prosthetic devices: a review of energy concepts and a proposed nomenclature, *Journal of Rehabilitation Research and Development*, 39(1):1-11, 2002.
- [42] Ferris, D. P., and C. T. Farley., Interaction of leg stiffness and surface stiffness during human hopping, *J. Appl. Physiol.* 82: 15-22, 1997.
- [43] Hansen AH, Childress DS, Miff SC, Gard SA, Mesplay KP, The human ankle during walking: implications for design of biomimetic ankle prostheses, *J Biomech.* 2004 Oct;37(10):1467-74.
- [44] H. Kazerooni, Jean-Louis Racine, Lihua Huang, and Ryan Steger, On the control of the Berkeley Lower Extremity Exoskeleton (BLEEX), *Proceedings of the 2005 IEEE International Conference on Robotics and Automation*, Barcelona, Spain, April 2005

Prof. dr. ir. Dirk Lefeber

Vrije Universiteit Brussel, Department of Mechanical Engineering, Pleinlaan 2, 1050 Brussels, Belgium

Tel +32 2 629 28 64, Fax +32 2 629 28 65, E-mail dlefeber@vub.ac.be

Session lectures

FULLY NONLINEAR EQUATIONS FOR INTERNAL WAVES IN A TWO-LAYER FLUID

Chi-Min Liu, Chin-Hwa Kong

Keywords: Boussinesq equations, internal waves, two-layer fluid

Abstract

A new model provided for simulating internal waves in a two-layer fluid system is established in this study. Analogous to methods analyzing Boussinesq equations for surface waves, a set of fully nonlinear Boussinesq equations for internal waves is consequently derived. Using the technique of the Padé approximation one can determine the optimal model. Some wave properties of present study are compared with those of existing results to verify the validity of present model in deep configuration. Certainly, present model can be reduced the Boussinesq model for surface waves.

Concluding Remarks and Future Works

The fully nonlinear model equations, namely the enhanced Boussinesq equations, for internal waves in a two-fluid system are derived in this study. By incorporating the Padé approximants in the exact linear dispersion relationship, one can determine the depth parameters z_u and z_l , and thus obtain the optimal models. Present models are expected to be applied not only in coastal region but in deep oceans. Furthermore, a higher-order set of model equations is expected to achieve the superior accuracy of wave properties in deep water configuration in the future.

Acknowledgement

This study is supported by National Science Council of Taiwan with grant number, NSC-94-2611-E-270-001, which is greatly acknowledged.

References

- [1] Benjamin, T. B., "Internal waves of finite amplitude and permanent form", *J. Fluid Mech.*, Vol.25, pp.241-270, 1966.
- [2] Benjamin, T. B., "Internal waves of permanent form of great depth", *J. Fluid Mech.*, Vol.29, pp.559-592, 1967.
- [3] Chen, Y., and P. L.-F. Liu, "Modified Boussinesq equations and associated parabolic models for water wave propagation", *J. Fluid Mech.*, Vol.288, pp.351-381, 1995.
- [4] Choi, W., and R. Camassa, "Weakly nonlinear internal waves in a two-fluid system", *J. Fluid Mech.*, Vol.313, pp.83-103, 1996.
- [5] Choi, W., and R. Camassa, 1996 "Long internal waves of finite amplitude", *Phys. Rev. Lett.*, Vol.77 (9), pp.1759-1762, 1996.
- [6] Choi, W., and R. Camassa, "Fully nonlinear internal waves in a two-fluid system", *J. Fluid Mech.*, Vol.396, pp.1-36, 1999.
- [7] Davis, R. E., and A. Acrivos, 1967 "Solitary internal waves in deep water", *J. Fluid Mech.*, Vol.29, pp.593-607, 1967.

- [8] Evans, W. A. B., and M. J. Ford, "An integral equation approach to internal (2-layer) solitary waves", *Phys. Fluids*, Vol.8, pp.2032-2047, 1996.
- [9] Lamb, H., *Hydrodynamics*, Cambridge Univ. Press, New York, 1932.
- [10] Koop, C. G., and G. Butler, "An investigation of internal solitary waves in a two-fluid system", *J. Fluid Mech.*, Vol.112, pp.225-251, 1981.
- [11] Kubota, T., D. R. S. Ko, and L. D. Dobbs, "Propagation of weakly nonlinear internal waves in a stratified fluid of finite depth", *AIAA J. Hydrodyn.*, Vol.12, pp.157-165, 1978.
- [12] Madsen, P. A., and H. A. Schäffer, "Higher-order Boussinesq-type equations for surface gravity waves: derivation and analysis", *Phil. Trans. R. Soc. Lond. A*, Vol.356, pp.3123-3184, 1998.
- [13] Matsuno, Y., "A unified theory of nonlinear wave propagation in two-layer fluid systems", *J. Phys. Soc. Jpn.*, Vol.62, pp.1902-1016, 1993.
- [14] Matsuno, Y., "Higher-order nonlinear evolution for interfacial waves in a two-layer fluid system", *Phys. Rev. E*, Vol.49, pp.2091-2095, 1994.
- [15] Nwogu, O., "Alternative form of Boussinesq equations for nearshore wave propagation", *J. Waterways, Port, Coast., Ocean Engng., ASCE* Vol.119, pp.618-638, 1993.
- [16] Ono, H., "Algebraic solitary waves in stratified fluids", *J. Phys. Soc. Jpn.*, Vol.39, pp.1082-1091, 1975.
- [17] Osborne, A. R., and T. L. Burch, "Internal solitons in the Andaman Sea", *Science*, Vol.208, pp.451-460, 1980.
- [18] Umeyama, M., "Third-order Stokes internal waves for a density stratified two-layer fluid", *Mem. Tokyo Met. Univ.*, 50, 120-136, 2000.
- [19] Umeyama, M., "Experimental and theoretical analyses of internal waves of finite amplitude", *J. Waterways, Port, Coast., Ocean Engng., ASCE* Vol.128(3), pp.133-141, 2002.
- [20] Wei, G., J. T. Kirby, S. T. Grilli, and R. Subramanya, "A fully nonlinear Boussinesq model for surface waves. Part 1. Highly nonlinear, unsteady waves", *J. Fluid Mech.*, Vol.294, pp.71-92, 1995.

Chi-Min Liu, Assistant Professor,
ChienKuo Technology University,
1, Jieshou N. Rd., Changhua City,
TAIWAN.
Tel: +886-4-7111111 ext.2505
Email: cmliu@ctu.edu.tw

Chin-Hwa Kong, Professor,
Department of Engineering Science and Ocean Engineering, National Taiwan University
73 Chow Shan Rd., Taipei,
TAIWAN.
Tel: +886-2-33665753
Email: chkong@ntu.edu.tw

EXPERIMENTAL AND NUMERICAL MODELLING OF OPEN-CELL CELLULAR STRUCTURES

M. Vesenjak, L. Krstulović-Opara, A. Öchsner, Z. Ren, Ž. Domazet

Keywords: cellular structures, open cells, experiments, computational modelling, dynamic loading

ABSTRACT

The paper describes the behaviour of regular open-cell cellular structures under dynamic compressive loading and the influence of the pore filler. Such materials, with their high energy absorption, are of particular importance as crash absorbers where impact energy is absorbed and dissipated. Open-cell cellular specimens made of aluminium alloy and polymer have been experimentally tested with aim to determine their stiffness, and for validation of the computational models. The quasi-static and dynamic compressive loading has been applied to the open-cell cellular structures in order to evaluate the strain rate sensitivity (Fig. 1).

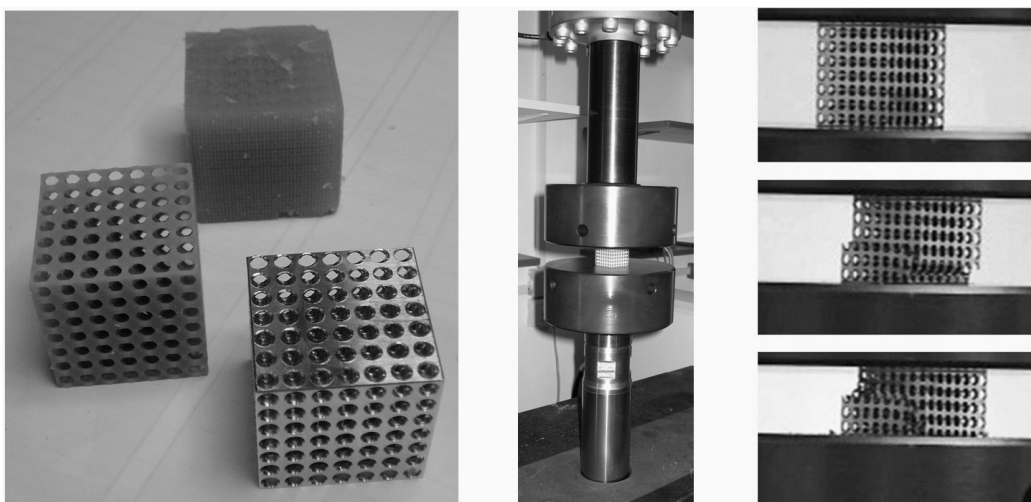


Figure 1. Specimens and experimental testing

Additionally, cellular specimens with filler material have been subjected to compressive loading. The specimens with the filler material have shown that the observed brittle behaviour at rupture has been reduced and the energy absorption increased (Fig. 2). Such structures are also very strain rate dependent as shown in Figure 2.

Further investigations of the filler influence were carried out with extensive dynamic nonlinear computational simulations (Fig. 3). Computational results of the open-cell cellular structure have shown that the influence of the filler increases with increasing the relative density.

The developed computational models provide for better understanding of behaviour of such materials and development of new approaches for design of open-cell cellular structures.

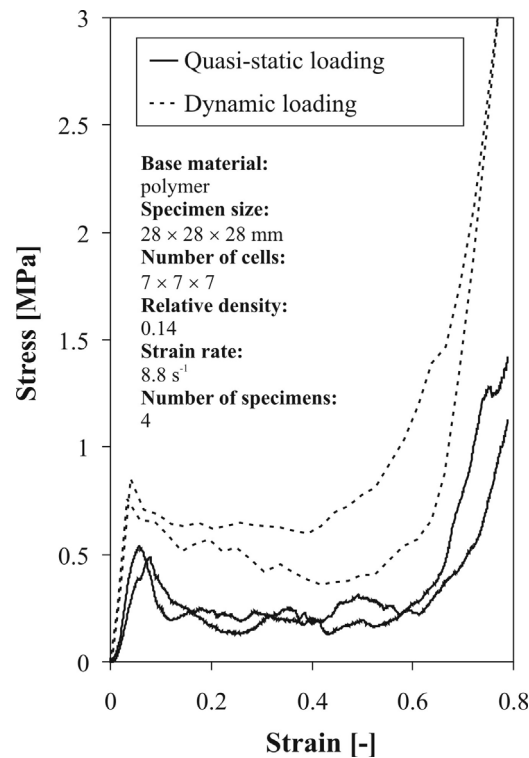


Figure 2. Behaviour of polymer cellular structure with filler

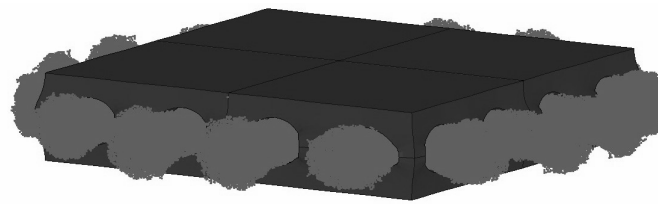


Figure 3. Computational model of cellular structure with filler

References

- [1] Banhart J., "Manufacture, characterisation and application of cellular metals and metallic foams", Progress in materials science (2001), vol. 46, no. 6, pp. 559-632.
- [2] Gibson, L.J., Ashby, M.F., "Cellular solids: Structure and properties", Cambridge University Press, Cambridge, 1997.
- [3] Öchsner, A., "Experimentelle und numerische Untersuchung des elasto-plastischen Verhaltens zellulärer Modellwerkstoffe", VDI Verlag GmbH, Düsseldorf, 2003.
- [4] Drstvenšek, I., "Rapid prototyping and innovative manufacturing network", Maribor, Available Online: <http://www.rapiman.mojsevis.net/> [7.1.2005].
- [5] Vesenjāk, M., "Computational modelling of cellular structures under impact loading", PhD Thesis, University of Maribor, Faculty of Mechanical Engineering, Maribor, 2006.
- [6] Vesenjāk, M., Öchsner, A., Ren, Z., "Behaviour of closed-cell foams under impact", 2. Workshop on Advanced Computational Engineering Mechanics, Erlangen, 2005, pp. 195-204.
- [7] Hallquist, J.O., "Keyword manual", Livermore Software Technology Corporation, Livermore, 2003.
- [8] Liu, G.R., Liu, M.B., "Smoothed particle hydrodynamics: a meshfree particle method", World Scientific, Singapore, 2003.

Corresponding author:

Matej Vesenjāk, Dipl. Ing. Mech. Eng.

University of Maribor, Faculty of Mechanical Engineering, Smetanova ul. 17, 2000 Maribor, Slovenia,

A Variational Model for Fracture Mechanics

Gianpietro Del Piero, Giovanni Lancioni, Riccardo March

Keywords: Fracture, Crack Mechanics, Variational Calculus, Finite Elements

1. The mathematical model

The object of the present communication is a re-consideration of the results obtained by Bourdin, Francfort and Marigo in their paper [1]. In it, the problem of the formation and growth of fracture in an elastic body subjected to monotonically increasing loading is solved numerically by means of an ad hoc finite element technique.

According to the formulation given in [2], the problem of the fracture of an elastic body is reduced to the minimum problem for an energy functional consisting of two terms, a bulk energy, which is assumed to be a quadratic function of the displacement gradient, just as in infinitesimal elasticity, and a surface energy, which is assumed to be proportional to the area of the fracture surface, just as in Griffith's theory of fracture.

The solution technique followed in [1] is based on a regularization procedure due to Ambrosio and Tortorelli [3], and consisting in approximating the jumps of the displacement field at the fracture surface by a sequence of smooth scalar fields. Each term in the sequence provides a regularized energy functional, and the global minima of these functionals converge to the solution of the fracture problem in the specific variational sense provided by the Γ -convergence theory.

In the present communication, the quadratic energy functional of infinitesimal elasticity is replaced by a fully non-linear functional. In particular, the dependence of the bulk energy density on the determinant of the deformation gradient is taken into account. This is a rather delicate affair, since this requires an *ad hoc* gamma-convergence result, not yet fully available, though the lower semicontinuity of the functional has been proved recently [4].

2. Mechanical aspects

With respect to classical models for fracture, the model developed in [2] has a distinctive property: it does not assume the existence of pre-cracked or weakened regions to determine the fracture onset. Also, both stress concentration intensities and pre-determined fracture modes play no role in the model. In the very spirit of Griffith's theory, fracture takes place when, at increasing values of the loading parameters, the formation of a crack becomes energetically more convenient than a further increase of elastic deformation. In this respect, as recognized by leading researchers in the area of fracture mechanics, fracture can nowadays be regarded as a peculiar form of elastic instability.

The model is almost completely elastic, the only element of irreversibility being the assumption, introduced at the level of the numerical solution algorithm, that a fracture cannot heal. Thus, under the action of the loading parameters the size of the fractured region can only grow. An important consequence of this assumption is the fundamental role played by the local energy minimizers, as opposed to global minimizers which, almost exclusively, form the object of the classical theories.

3. Numerical simulations

Some two-dimensional simulations, based on an algorithm for incremental finite-element discretization, were provided in [1]. In the present communication, the same simulations are made on the modified energy functional. They reveal two main points in which the results of [1] are decidedly improved by taking into account the dependence of the bulk energy on the determinant:

1. The possibility of interpenetration of matter under compressive loads, implicit in the formulation given in [1], is eliminated. This is done by attributing very large values to the bulk energy when the determinant of the deformation gradient is close to zero.

2. The symmetric behavior in tension and compression, which in [1] is a consequence of the quadratic structure of the functional, disappears as well. When assuming a quadratic bulk energy, one has the undesired consequence that the shape of a minimizing configuration is insensitive to the sign of the applied load. In particular, one gets the same fracture response in tension and in compression. In this respect, the introduction of the dependence on the determinant provides the way for a clear distinction between the two types of response. In fact, in our two-dimensional simulations we obtain that fracture is concentrated on a line (cleavage fracture) when the specimen is subject to tension, and diffused on a two-dimensional cracked zone when the specimen is compressed.

References

- [1] Bourdin B., G.A. Francfort, J.J. Marigo, “Numerical experiments in revised brittle fracture”, *J. Mech. Phys. Solids*, 48, 2000, pp. 797-826,
- [2] Francfort G.A., J.J. Marigo, “Revisiting brittle fracture as an energy minimization problem”, *J. Mech. Phys. Solids*, 46, 1998, pp. 1319-1342,
- [3] Ambrosio L., V.M. Tortorelli, “Approximation of functionals depending on jumps by elliptic functionals via Γ -convergence”, *Comm. Pure Appl. Math.*, 43, 1990, pp. 999-1036,
- [4] Fusco N., C. Leone, R. March, A. Verde, “A lower semicontinuity result for polyconvex functionals” *Proc. Roy. Soc. Edinburgh, Sect. A, Math.*, to appear.

Gianpietro Del Piero

Dipartimento di Ingegneria, Universita' di Ferrara, Via Saragat 1, Ferrara, Italy <dlpgpt@unife.it>

Giovanni Lancioni

Dip. di Costruzioni e Strutture, Università Politecnica delle Marche, Ancona, Italy <g.lancioni@univpm.it>

Riccardo March

Istituto per le Applicazioni del Calcolo, CNR, Roma, Italy <r.march@iac.cnr.it>

HOMOGENIZATION OF SOLIDS WEAKENED BY CRACKS - AVERAGING TECHNIQUES OR PERIODIC UNIT CELLS

Jan Gajdošík, Jan Skoček, Jan Novák, Michal Šejnoha and Jan Zeman

Jointed rocks represent one particular class of solids weakened by cracks or more correctly in the spirit of geotechnical engineering by joints. In any case, these joints represent discontinuities in otherwise homogeneous solid. Owing to generally large dimensions of the analyzed problem (large underground caverns, deep seated tunnels, etc.) it is possible to treat joints, having sizes even of the order of meters, from the micromechanics point of view [2].

The possibility of using the micromechanics based tools for the analysis of highly jointed rocks is the main motivation for the presented work. Such analysis is well presented in the context of multi-scale solution technique. The major advantage of the multi-scale modeling is no need for constructing a macroscopic nonlinear constitutive law. Instead, a detailed analysis on micro-scale that properly accounts not only for actual microstructure configuration but also for essential damage mechanisms governing the macroscopic nonlinear response serves as the input for the subsequent analysis on macroscale. The two problems on both scales are solved simultaneously, but generally in the uncoupled manner [2, 3]. Combining the nonlinear behavior on the level of joints with the possible evolution of critical discontinuity on macroscale, brings, apart from numerical implementation, no additional difficulties. Such discontinuity can be introduced, e.g. in the context of partition of unity method (PUM) [1].

Although the concept of PUM is briefly introduced, the main objective remains the homogenization of solids with relatively high density of cracks (joints). Two particular approaches are available in this regard. The first approach takes advantage of well known averaging techniques that include Dilute approximation, Self consistent method, Cai-Horii and Mori-Tanaka methods. Details on the application of the above methods to cracked solids can be found in [5, 4, 6]. Although relatively simple their extension to account for nonlinear effects is not straightforward. Therefore, a formulation that exploits the concept of periodic unit cells is also addressed.

There are several ways to generate such a unit cell. If the knowledge of true microstructure is known, it is possible to proceed along the same lines as discussed in [7]. Otherwise a randomness of crack patterns needs to be assumed. Several such distributions are evident in Fig. 1. The large RVE then may substitute the true microstructure in the derivation of smaller unit cells having similar material statistics [7]. The question of the minimum number of cracks within a PUC is, however, still open and is the subject of current research. Another drawback of this approach is a lack of microstructural information currently limited to crack density on the one hand and a number of free parameters (PUC dimensions, number of cracks, crack size) to generate a PUC with the desired crack density on the other hand. Fig. 1(a),(b) shows two different crack patterns having the same crack density. Whether these two crack patterns will possess the same effective properties is another issue that deserves attention.

Finally note that the present approach is limited to fixed crack density. Extension accounting for an increase of crack density due to progressive damage is not straightforward as it would require remeshing and some sort of transfer of state variables from the old mesh to the new one. The crack density growth would bring, on the other hand, no extra complexities when using simple averaging methods. In view of this, these methods still deserve our attention.

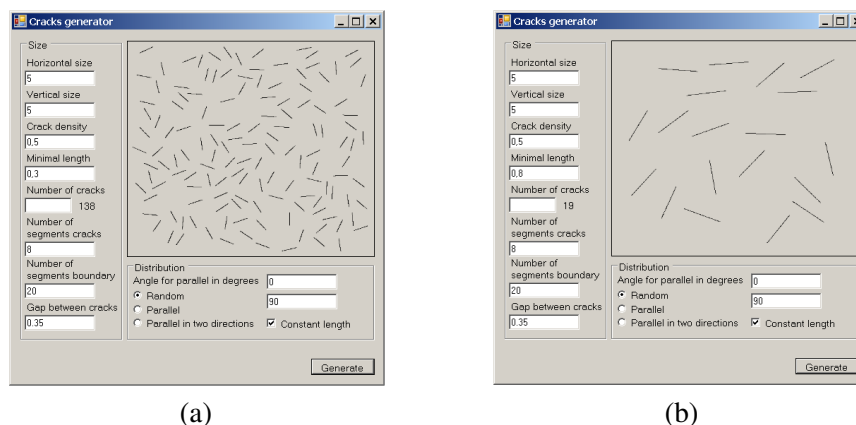


Figure 1. Two possible arrays of randomly distributed cracks of various sizes but the same crack density

Acknowledgement

The financial support provided by the project CEZ MSM 6840770003 and partially also by the GACR grant 106/03/H150 is gratefully acknowledged.

References

- [1] Audy, M., Krček, J., Zeman, J. and Šejnoha, M., “Constant strain triangular element with embedded discontinuity based on partition of unity”, *Building Research Journal*, 2002.
- [2] Cai, M. and Horii, H., “A constitutive model and FEM analysis of jointed rock masses”, *Int. J. Rock. Mech. Min. Sci. & Geomech.*, 30(4), 1993, pp. 351–359
- [3] Fish, J., Yu, Q. and Shek, K., “Computational damage mechanics for composite materials based on mathematical homogenization”, *International Journal for Numerical Methods in Engineering*, 45(11), 1999, pp. 1657–1679
- [4] Kachanov, M., “Effective elastic properties of cracked solids: critical review of some basic concepts”, *Applied Mechanics Review*, 45(8), 1992, pp. 304–335
- [5] Laws, N. and Dvorak, G.J., “The effect of fiber breaks and aligned penny-shaped cracks on the stiffness and energy release rates in unidirectional composites”, *International Journal of Solids and Structures*, 23(9), 1987, pp. 1269–1283
- [6] Nemat-Nasser, S. and Horii, M., “Micromechanics: overall properties of heterogeneous materials”, second ed., Elsevier Science B.V., North-Holland, 1999
- [7] Zeman, J. and Šejnoha, M., “Numerical evaluation of effective properties of graphite fiber tow impregnated by polymer matrix”, *Journal of the Mechanics and Physics of Solids*, 49(1), 2001, pp. 69–90

Ing. Jan Gajdošík

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.gajdosik@fsv.cvut.cz

Jan Skoček

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:honzaskocek@seznam.cz

Ing. Jan Novák

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.novak@cml.fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:sejnom@fsv.cvut.cz

Ing. Jan Zeman, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:zemanj@cml.fsv.cvut.cz

STOCHASTIC MODELING OF TIME DEPENDENT BEHAVIOR OF EARTH STRUCTURES

Jiří Šejnoha, Michal Šejnoha and Marie Kalousková

Together with the available computational power the driving engines contributing to a rapid evolution of new solution strategies for the design and verification of earth structures include unfortunately also catastrophic failures of these structures. Dump slopes or earth dams fit well within the category of structures prone to such failures. In this context the probabilistic analysis receives a considerable attention, especially when combined with risk analysis. The risk, presented as the value of the probability of failure multiplied by the cost spent on removing the consequences of failure, is much a stronger argument for the selection of even more expensive technologies for the construction than just the probability of failure itself. This field of expertise, however, is not the main topic of this paper. Instead, we limit our attention to the evaluation of the first variable in this equation, the probability of failure, but unlike in the accompanying paper [2] we shall consider this task in much broader sense by adding the factor of time. Nevertheless, the stepping stone in our analysis will be once again evaluation of the factor of safety FS through the solution of the slope stability problem as described in [2].

At minimum the stability analysis of earth structures when presented in the framework of Mohr-Coulomb failure criterion [3] requires the knowledge of four parameters, which are often assumed to have a random character. They are essentially random functions, which vary both in time and space. The material parameters subjected to random variation are the self-weight $\gamma = \gamma(x, y)$, effective strength parameters $c = c(x, y)$ and $\varphi = \varphi(x, y)$, and pore pressure $p = p(x, y, t)$. Owing to the complexity of the underlying numerical analysis we settle henceforth for only two random variables, namely the cohesion c and the angle of internal friction φ . The basic statistical representation of these parameters is given in [2]. The principle objective of this contribution is concerned with the situation, in which the conditions governing the slope stability analysis change with time. A typical example is an earth dam with the possibility of gradual change of ground water table (GWT) during filling or due to flooding. Heavy rains leading to a sudden change in water pore pressure are also usually responsible for catastrophic failure of dump slopes. Predicting such failures is therefore of paramount importance. Valuable information in this regard may provide the (instantaneous) *failure rate* [1] (sometimes referred to as *hazard function*). An illustrative example is used to propose a simple, but rather appealing procedure, for estimation of these quantities. In case of an artificial dam subjected to gradual change in ground water table.

Evaluating the stability condition of the working system (earth dam) using the concept of factor of safety the reliability condition can be presented, regardless of the geometrical model, in the form

$$FS > \overline{FS} \geq 1, \quad (1)$$

where $\overline{FS}$ represents a certain value of the factor of safety specified, e.g., by regional standards that must be exceeded. The probability of failure is then provided by

$$p_f(t) = P\{FS(t) \leq \overline{FS}\} = F_{FS}(\overline{FS}). \quad (2)$$

The reliability $R(t)$ of the system is then expressed in the form

$$R(t) = P\{FS > \overline{FS}(t)\} = 1 - F_{FS}(\overline{FS}(t)). \quad (3)$$

Considering the conditional probability that the failure occurs in the time interval $(t + dt)$ providing the system was working properly at time t allows us to write the instantaneous failure rate (intensity of failure) $\lambda(t)$ in the form

$$\lambda(t) = -\frac{dR(t)}{R(t)}. \quad (4)$$

Eq. (4) can be rewritten with the help of Eq. (3) to get

$$\lambda(t) dt = \frac{dF_{FS}(\overline{FS}(t))}{1 - F_{FS}(\overline{FS}(t))} = \frac{f_{FS}(\overline{FS}(t))}{1 - F_{FS}(\overline{FS}(t))} \frac{d\overline{FS}}{dt} dt. \quad (5)$$

Finally, integrating Eq. (5) yields

$$\int_0^t \lambda(\tau) d\tau = -\ln \frac{1 - F_{FS}(\overline{FS}(t))}{1 - F_{FS}(\overline{FS}(0))} = -\ln \frac{R(t)}{R(0)} = -\ln R(t) = \Lambda(t), \quad (6)$$

where $\Lambda = \Lambda(t)$ is the accumulated intensity of failure. To arrive at Eq. (6) we took advantage of the fact that $R(0) = 1$, which in turn is guaranteed by the general property of the distribution function $F(0) = 0$. The failure rate (or accumulated intensity of failure) as a quantity giving some insight to the reliability of the well performing system can find its application particularly if the in situ monitoring is available. In particular, suppose that such a monitoring can estimate the drop of the factor of safety FS or, by another name a shift of the real distribution function to the left by ΔFS in time interval $\Delta t = t_2 - t_1$. The failure rate at the given time t_1 can be then found from Eq. (5) as

$$\lambda(t_1) = \frac{f_{FS}(\overline{FS}(t_1))}{1 - F_{FS}(\overline{FS}(t_1))} \frac{\Delta \overline{FS}}{\Delta t}. \quad (7)$$

Acknowledgement

The financial support provided by the grant No. 1ET410430516 awarded by the Czech Academy of Sciences and by the project No. 1M6840770001 awarded by the Ministry of Education, Youth and Sports of the Czech Republic, within activities of the CIDEAS research center, is gratefully acknowledged.

References

- [1] Taylor, L. D., "Probability and mathematical statistics", HARPER & ROW PUBLISHERS, New York, 1974
- [2] Šejnoha, M., Kalousková, M., Zeman, J. and Novák, J., "Stochastic approach to the stability analysis of earth slopes", In: ICCSM 06, Split, Croatia, 2006
- [3] Potts, D. M. and Zdravkovič, L., "Finite element analysis in geotechnical engineering – theory", Thomas Telford, London, 1999

Prof. Ing. Jiří Šejnoha, DrSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354492, Fax:+420-2-24310775, e-mail:sejnoha@fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354494, Fax:+420-2-24310775, e-mail:sejnom@fsv.cvut.cz

Ing. Marie Kalousková, CSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-243589, Fax:+420-2-24310775, e-mail:kalouskova@fsv.cvut.cz

STOCHASTIC APPROACH TO THE STABILITY ANALYSIS OF EARTH SLOPES

Michal Šejnoha, Marie Kalousková, Jan Zeman and Jan Novák

Deterministic approach to the analysis of engineering problems is justified merely in cases where input data earn such a respect that the expected results are assumed to be more or less “unique”. Unfortunately, such a situation hardly ever occurs in practice especially in geotechnical applications where input data often acquire a label of being vague rather than well defined. This unpleasant fact is usually blamed for a lack of time when conducting a suitable experimental program. Nevertheless, the source for uncertainties in material input data can be even less obscure particularly when considering dump slopes consisting of mixture of waste material. Inability in describing the material by a single set of parameters thus naturally opens a way to the field of statistics. Accepting a random character of input data clearly requires considerably different approach to the solution of the problem of stability of earth slopes when compared to purely deterministic analysis introduced. Stochastic approach to the modeling of failure of earth structures is the prominent topic of this paper. We present the basic theoretical grounds pertinent to the modeling of slope stability problems in the framework of finite element method and stochastic environment including estimates of the probability of failure. Application to the topic of reliability is presented in the accompanying paper [1].

Solution strategy of the slope stability problem when introduced in conjunction with the finite element draws on the assumption that the forces generated by the self weight represent the only source of loading and are applied in a single increment to an initially stress-free slope. It has been argued in [2] that the predicted factor of safety derived from simple plasticity models (Mohr Coulomb, Drucker-Prager) is insensitive to the form of gravity application, thus justifying the use of gravity *turn-on* procedure. As typical for soil stability problem relating the critical strength parameters of a soil $\bar{c}, \bar{\varphi}$ to the actual strength parameters c, φ , allows us to introduce the factor of safety FS in the form

$$\bar{c} = \frac{c}{FS}, \quad \bar{\varphi} = \frac{\varphi}{FS}. \quad (1)$$

Adopting Eq. (1) within the context of finite element method leads to the *shear strength reduction technique* offering the searched factor of safety as the factor by which the original shear strength parameters c, φ must be reduced to achieve the slope instability. Such an approach is therefore consistent with conventional limit equilibrium methods, particularly in a sense of defining the factor of safety as the ratio of restoring and driving forces (moments)

$$FS = \frac{R(\bar{c}, \bar{\varphi})}{E(\bar{\gamma})}, \quad (2)$$

Unlike the deterministic approach that offers only a single value of the factor of safety for a given set of parameters provided by (2), the stochastic approach evaluates the reliability of a structure in terms of the probability of failure p_f

$$p_f = P\{FS = \frac{R(c, \varphi)}{E(\gamma)} < 1\}, \quad (3)$$

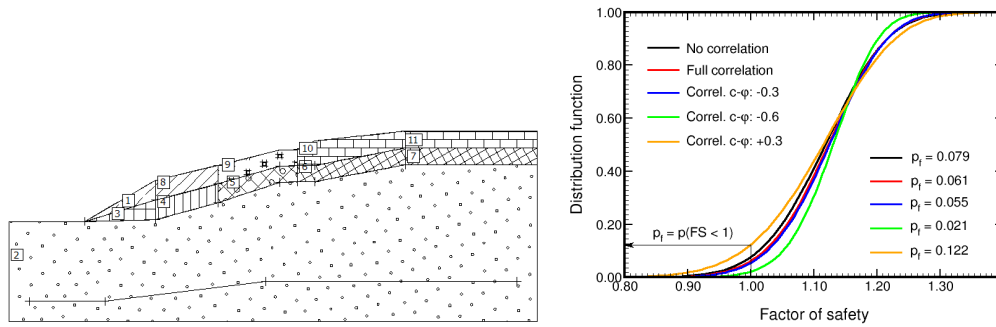


Figure 1. Slope geometry and resulting distribution functions of FS

where variables R and E represent in both equations restoring and driving forces, respectively. The major advantage of Eq. (3) appears in the fact that we do not have to deal with the exact values of input parameters, but rather with their probable range. The random material parameters are then mostly presented in terms of probability distributions.

Given the probability distribution of input parameters allows us to draw random numbers from these distributions and create a sampled space of sets of input parameters. These sets are then utilized in a certain simulation process to arrive at derived output parameters, e.g., derived set of factors of safety to be fitted with the most appropriate probability distribution allowing for subsequent evaluation of the desired probability of failure. Application of this approach to a dump slope appears in Fig. 1. It is worth to note, however, that not the probability of failure itself as listed in Fig. 1(b), but rather the whole distribution function should be examined when evaluating the reliability of the working system.

Acknowledgement

The financial support provided by the project CEZ MSM 6840770003 and partially also by the GAČR grant No. 103/04/1134, and by the grant No. 1ET410430516 awarded by the Czech Academy of Sciences is gratefully acknowledged.

References

- [1] Šejnoha, M., Šejnoha, J. and Kalousková, M., “Stochastic modeling of time dependent behavior of earth structures”, In: ICCSM 06, Split, Croatia, 2006
- [2] Griffiths, D. V., “Stability analysis of highly variable soils by elastoplastic finite element”, International Journal for Numerical Methods in Engineering, 50, 2001, pp. 2667–2682

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354494, Fax: +420-2-24310775, e-mail: sejnomo@fsv.cvut.cz

Ing. Marie Kalousková, CSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-243589, Fax: +420-2-24310775, e-mail: kalouskova@fsv.cvut.cz

Ing. Jan Zeman, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354482, Fax: +420-2-24310775, e-mail: zemanj@cml.fsv.cvut.cz

Ing. Jan Novák

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354472, Fax: +420-2-24310775, e-mail: jan.novak@cml.fsv.cvut.cz

MULTI-SCALE MODELING OF MASONRY STRUCTURES - SYNTHESIS OF CONSTITUTIVE MODELS AND SCALE TRANSITION

Jan Sýkora, Jan Vorel, Michal Šejnoha, Jan Zeman and Jiří Šejnoha

While the robust commercial software ATENA has been successfully used in the modeling of complex historical structures in totally uncoupled framework with the transition of scales solely on the basis of homogenized material parameters, its applicability when linking scales in a coupled manner becomes rather cumbersome. Direct use of available commercial codes is further precluded by the inherited quasi-brittle behavior of masonry structures with expected localization of inelastic deformation on both scales. A new more competitive computational tool that allows for better prediction of the behavior of complex materials or structures across all relevant length scales is therefore needed. Motivation for the proposed is evident from Fig. 1 showing large masonry structure with possibly periodic geometry on the level of constituents, further referred to as mesoscale, a single localized macroscopic crack running through both phases and a corresponding periodic crack pattern resulting from the mesoscopic analysis using the first-order periodic homogenization.

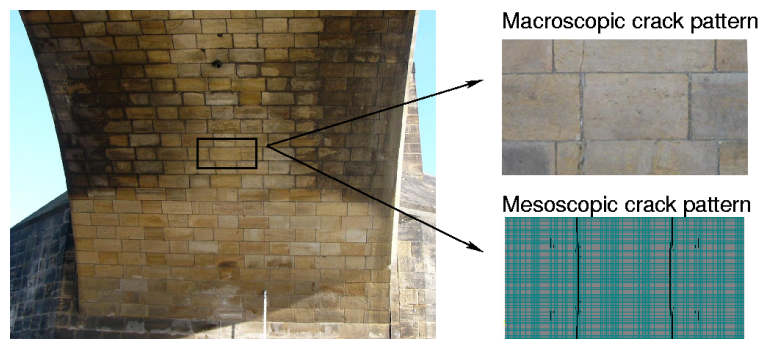


Figure 1. Motivation for multi-scale analysis

The stepping stone in the proposed approach is the concept of meso-macro modeling framework developed in [1] for regular masonry structures. In his approach the scale transition accounting for regularization on macroscale is performed through the introduction of macroscopic shear band. Although very promising, this approach shows a number of difficulties on the numerical implementation side. Also note a single discontinuity rather than crack band developed on the macroscale, Fig. 1.

Thus, with reference to [3], we offer an alternative approach based on continuous-discontinuous approach similar to one applied in [2] to ECC composites. While the smeared crack model is assumed for the analysis of intricate geometries on meso-scale using the concept of statistically equivalent periodic unit cells, the partition of unity method (PUM) is called to link the estimated distributed damage (owing to the assumed periodic boundary conditions) to discrete cracks visible on the macroscale, Fig. 1.

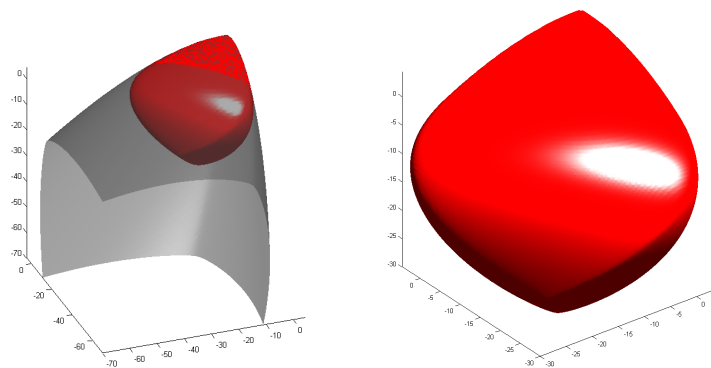


Figure 2. 3D visualization of the yield surface in the principal stress space

Although the extraction of the macroscopic material behavior from the detailed representation at a lower scale constitutes the most important part of the multi-scale computational technique, its success in the prediction of the true behavior is highly influenced by the proper representation of the material response on the level of individual phases, bricks and mortar. Here, the attention is limited to the formulation of the new constitutive model capable of representing the material failure in shear and confining pressure. Graphical representation of the proposed smooth cap model appears in Fig. 2. This model is expected to be implemented in the ATENA finite element code [4].

Acknowledgement

The financial support provided by the project CEZ MSM 6840770003 and partially also by the GAČR grants 106/03/H150 and GACR 103/04/1321 is gratefully acknowledged.

References

- [1] Massart, T. J., “Multi-scale modeling of damage in masonry structures”, Ph.D. thesis, Technische Universiteit Eindhoven, 2003, 159 pages
- [2] Simone, A., “Continuous-Discontinuous Modelling of Failure”, Ph.D. thesis, Delft University Press, 2003, 172 pages
- [3] Jirásek, M., “Comparative study on finite elements with embedded discontinuities”, *Computer Methods in Applied Mechanics and Engineering*, 188, 2000, pp. 307–330.
- [4] Červenka, V., Jendele, L. and Červenka, J., “ATENA Program Documentation. Part 1: Theory”, Červenka Consulting, Prague, 2003, <http://www.cervenka.cz/Web>

Ing. Jan Sýkora

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.sykora.1@fsv.cvut.cz

Ing. Jan Vorel

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.vorel@fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:sejnom@fsv.cvut.cz

Ing. Jan Zeman, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:zemanj@cml.fsv.cvut.cz

Prof. Ing. Jiří Šejnoha, DrSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:sejnoha@fsv.cvut.cz

PREDICTION OF HOMOGENIZED MATERIAL PROPERTIES OF QUARRY MASONRY FROM MICROMECHANICS

Jan Vorel, Jan Sýkora, Jiří Šejnoha and Michal Šejnoha

Rehabilitation of complex historical structures is at the current forefront of engineering interest. Examples of the day include the Charles Bridge in Prague and the St. Barbora cathedral in Kutná Hora. An accurate prediction of the behavior of such structures under complex macroscopic loading program with the main objective of offering the most suitable way of reconstruction requires the knowledge of homogenized effective properties including effective fracture energy of masonry structures with often irregular distribution of stone blocks. While a carefully prepared experimental program may provide reliable estimates of the required quantities for masonry systems with regular arrangement of blocks, the quarry masonry are best analyzed with the help of classical micromechanics tools. Classification of various approaches to the analysis of structural masonry can be found, e.g. in [2, 3]

Owing to its complexity when applied to large real world masonry structures the prominent uncoupled multi-scale analysis [1], where the output from the analysis on meso-scale (level of individual phases) serves as the input for the analysis on macro-scale and vice versa, is often further simplified and the two analyses are performed entirely independently. Here, the concept of periodic unit cell (PUC) appears as a suitable method of attack when deriving the homogenized material properties needed in the subsequent macroscopic analysis of the whole structure [5]. The typical quasi-brittle response of the two basic building blocks (the stone and mortar phases) with inherent strain softening further accompanied by localization of inelastic strains may rise some doubts on the applicability of homogenization schemes based on periodic fields. It was postulated, however, in [6] that if limiting our attention solely to the determination of effective (macroscopic) material properties the periodic homogenization employing the smallest PUC as a suitable representative volume element provides reasonable estimates of the desired material parameters including the macroscopic fracture energy. The morphology of a meso-scopic sample can be obtained either from the test pits which are excavated, e.g. from the body of Charles Bridge in Prague, or from digitized photographs converted into bit maps and eventually discretized using the mesh of finite elements applied to both basic materials.

Nevertheless, caution must be paid when deriving the effective material properties of quarry masonry. In particular, it was suggested from experimental observations that the air bubbles in the pores near the surface of dry stones may considerably impair interfacial properties along the interface transition zone (ITZ). Proper treatment of ITZ thus appears crucial in deriving sufficiently accurate results. To address this task, the Mohr-Coulomb material model is applied in the present study in conjunction with the contact elements to grasp this phenomenon. The behavior of the remaining two phases (mortar and stone blocks) is treated using the smeared crack model implemented in the ATENA commercial code [4].

Calibration of the model is a very important task from both the theoretical and practical points of view. An experimentally tested quarry masonry sample serves as a suitable source for optimizing the material data, especially for contact elements in the ITZ. Loading in compression was selected because in this particular case the satisfactory correspondence of the computationally obtained results with experimental outputs is rather difficult to achieve.

There is a variety of techniques how to optimize the input data. One approach is very simple and starts from a set of input parameters based on "trial and error" procedure, e.g. see [7]. In this contribution another way which stands to benefit from sets of randomly generated input data is used, of course under the assumption that probability density distributions of individual input parameters can be well estimated. Standard simulation techniques, such as the Monte Carlo or Latin Hypercube Sampling methods, are very suitable tools for data generation. The calculated loading paths are compared with that obtained experimentally. The least square method applied to the difference between the calculated and measured loading force (the test is controlled by the displacement) then yields the optimized model data.

Finally, the loading paths for the PUC with the ITZ and the corresponding material properties are computed. These load-strain curves are then used to determine the fracture energy for different directions. There are two possible ways available how to obtain these values demonstrating the capability of using both the RILEM recommendation and the incremental solution.

Acknowledgement

This outcome has been achieved with the financial support of the Ministry of Education, Youth and Sports, project No. 1M680470001, within activities of the CIDEAS research centre. In this undertaking, theoretical results gained in the projects GAČR 103/04/1321 and 103/06/H150 were partially exploited.

References

- [1] Fish, J. and Shek, K., "Multiscale Analysis of Large-Scale Nonlinear Structures and Materials", *International Journal for Computational Civil and Structural Engineering*, 1(1), 2000, pp. 79–90
- [2] Rots, J., "Structural Masonry - An Experimental/Numerical Basis for Practical Design Rules", Balkema, Rotterdam, 1997
- [3] Lourenço, P.B., "Computational Strategies for Masonry Structures", Delft University Press, 1996
- [4] Červenka, V., Jendele, L. and Červenka, J., "ATENA Program Documentation. Part 1: Theory", Červenka Consulting, Prague, 2003, <http://www.cervenka.cz/Web>
- [5] Šejnoha, J., Bittnar, Z., Šejnoha, J., Zeman, J., Novák, J. and Janda, Z., "Stress and damage analysis of the Charles Bridge in Prague". Technical report, Mott MacDonald Prague Ltd., 2005, 159 pages (In Czech).
- [6] Novák, J., Šejnoha, M. and Zeman, J., "On Representative Volume Element Size for Analysis of Masonry Structures", *Proceedings of The Tenth International Conference on Civil, Structural and Environmental Engineering Computing*, Edited by B.H.V. Topping, Rome-Italy, 2005
- [7] Sýkora, J., "Diploma work - Computational modeling of quarry masonry", Praha, 76 pages, 2005, (in Czech)

Ing. Jan Vorel

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.vorel@fsv.cvut.cz

Ing. Jan Sýkora

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, e-mail:jan.sykora.1@fsv.cvut.cz

Prof. Ing. Jiří Šejnoha, DrSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354492, Fax:+420-2-24310775, e-mail:sejnoha@fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354494, Fax:+420-2-24310775, e-mail:sejnom@fsv.cvut.cz

CYCLOSMMETRY IN THE MECHANICS OF STRUCTURES

Riko Rosman

Keywords: cyclosymmetry, symmetry operations, symmetry

Symmetric engineering structure-systems, for simplicity structures in the following, can be 1) mirror-symmetric, 2) cyclo- or rotation symmetric and/or 3) point- or central symmetric. The corresponding symmetry elements are 1) the symmetry plane in the case of spatial and the symmetry axis in the case of plane structures, 2) the axis of cyclo- or rotation symmetry and 3) the symmetry point or center.

With respect to the symmetry elements symmetry operations, symmetry transformations or isometries can be performed. A symmetry operation is an imagined movement of the structure which brings it into an equivalent configuration indistinguishable from the initial one. The self-coincidence implies not only the geometric but also the structural, especially stiffness properties. Analogous statements apply also for actions onto the structure, usually loads and the corresponding response quantities. A special kind of symmetric are anti- or skewsymmetric actions and response quantities; they differ only in their orientation or sign.

General definitions are: 1) A load acting onto a symmetric structure is symmetric, if a symmetry operation corresponding to the structure brings it into an equivalent configuration. 2) A load acting onto a symmetric structure is antimetric, if a symmetry operation corresponding to the structure and a sign reversal brings it into an equivalent configuration.

On the basis of experience the following axioms can be formulated: 1) Symmetric loads acting onto a symmetric structure produce symmetric reactions, internal forces and displacements. 2) Antimetric loads acting onto a symmetric structure produce antimetric reactions, internal forces and displacements [1, 2, 3].

Nonsymmetric actions onto the structure can be split into symmetric and antimetric components and their effects superposed.

Using the given general definitions and axioms mechanical analyses of symmetric structures can be significantly simplified. The structures degree of static or kinematic redundancy often is much smaller and often an indeterminate task becomes a determinate one. Often it is obvious that some response quantities vanish at a symmetry plane or axis.

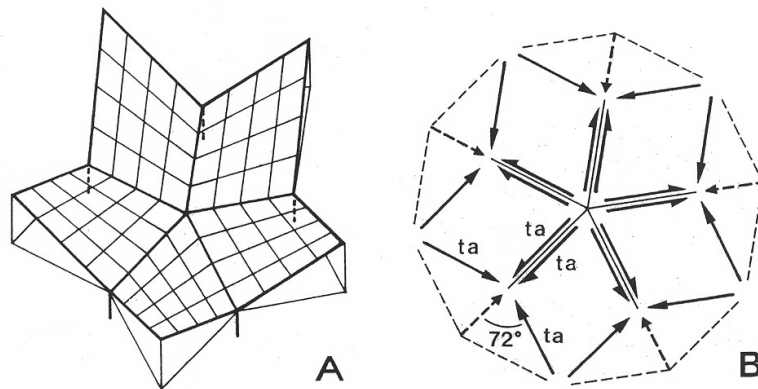
Symmetry in general, but especially cyclosymmetry is a highly desirable property of engineering structures. This is due to the favourable response behaviour of symmetric structures when subjected to the usual actions, the simplicity of their static and dynamic analyses, their aesthetic advantages, their easier construction and a better economy as compared to that of nonsymmetric structures.

The symmetry operations of cyclosymmetric structures are rotations about the symmetry axis. A n -fold cyclosymmetric structure has a n -fold axis of symmetry. The maximal value of n in the expression $2\pi/n$ for the rotation angle is the maximal number of possible symmetry operations, so that the possible rotation angles are $2\pi/n$, $2(2\pi/n)$, $3(2\pi/n)$, $n(2\pi/n) = 360^\circ$. In the

limit case $n = \infty$ the system is rotation-symmetric.

FIGURE 1A shows a roof structure consisting of five, in the general n shells having the form of hyperbolic paraboloids. Each shell has two interior horizontal edges and two exterior edges rising outwards. In the plan the structure has the form of a Pentagram. The shells are supported along its edges by boundary bars the interior ones forming a star-like bar system. This is supported by five columns.

The structure is 5-fold cyclosymmetric with respect to its central vertical axis. The load case own weight and uniform snow load has the same kind of symmetry. Hence, all shells, all bars and all columns are equally stressed. For example, if a denotes an edge's length and t the shear flow along the edges, all interior bars of the bar system are acted upon by forces at (Figure 1B), so that they are partly compressed, partly tensioned.



References

- [1] Glockner, G., "Symmetry in Structural Mechanics", Journal of the Structural Division, Vol. 99, 1973, pp. 71-88
- [2] Rosman, R., "Statik zyklisch symmetrischer Ring- und Rostrahmen", Bautechnik, Vol. 62, 1985, pp. 349-356
- [3] Rosman, R., "Beitrag zur Statik der freitragenden Podesttreppe", Wissenschaftliche Zeitschrift der TU Dresden, Vol. 31, 1982, pp. 105-109
- [4] Wilkinson, P., "Marketing in the Engineering", Marketing London UK, 1993

Riko Rosman, Academician

Faculty of Architecture, Šenoina 8, HR-10 000 Zagreb, Croatia,

Telephone ++ 385 1 4839838, Telefax: ++ 385 1 482 80 3879, e-mail : rosman@hazu.hr

OPTIMIZING LOAD-BEARING CAPACITY OF BEAMS SUBJECTED TO BENDING

Diana Šimić Ph.D. (Civ.Eng.)

Keywords: pre-bending, built-up beam, stresses, strains, zero deflection, load, increase in capacity

Extended abstract

The behaviour of built-up beams resting on two supports, and manufactured based on pre-bending technology, is described. The n number of beams, all of them made of similar material, is considered. The beams are of rectangular cross section, of height h and width b , of equal length, and are freely placed one upon another, as shown in Figure 1. During pre-bending of beams of height h , the direction of load is opposite to the direction of load to be imposed during service life, cf. Figure 1.a. In such prestressed state the beams of height h are linked together with coupling devices so as to form a built-up beam that behaves as a monolithic beam of height $H = n \cdot h$, and are then freed from external load, cf. Figure 1.b. The distribution of total normal stresses in cross section of a built-up beam, after load has been relaxed during pre-bending, is presented in Figure 1.c. By linking together n beams of height h , i.e. by forming a built-up beam of height $H = n \cdot h$, the moment of inertia and section modulus of the cross section are increased by n^2 or by n times when compared to the previous state in which n beams of height h are freely placed one above another without coupling devices.

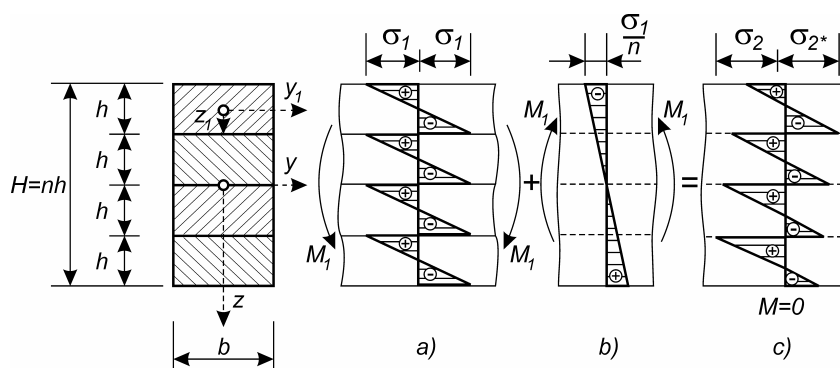


Figure 1 Diagram of normal stresses during the pre-bending procedure: a) during application of load, b) after relaxation of load, c) total stress after relaxation of load

The following applies to Figures 1.c. and 2.a.:

$$\sigma_2 = \sigma_1 \cdot \left(1 - \frac{1}{n}\right), \quad \sigma_2^* = \sigma_1 \cdot \left(1 + \frac{n-2}{n^2}\right) \quad (1)$$

The diagram of normal stresses in a built-up beam is presented in Figure 2.b for a given external load during service life. After superposition of stresses presented in Figure 2.a and 2.b we obtain the state of stress presented in Figure 2.c, which is the state of total stresses in the hazardous cross section of the beam subjected to a given load during service life.

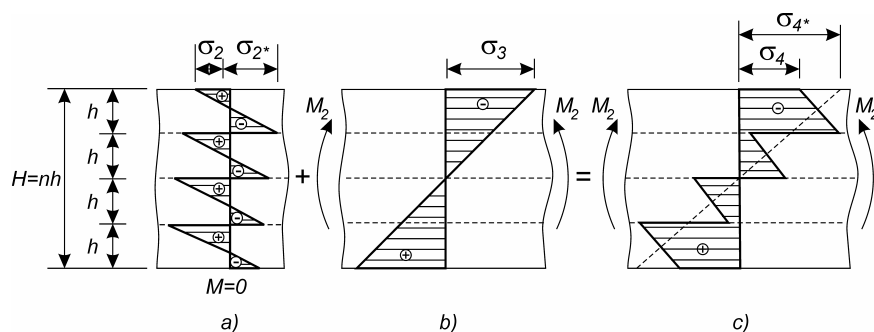


Figure 2 Normal-stress diagrams in a built-up beam

In the Figure 2.c we have:

$$\sigma_4 = \sigma_3 - \sigma_1 \cdot \left(1 - \frac{1}{n}\right), \quad \sigma_4^* = \sigma_3 \cdot \left(1 - \frac{2}{n}\right) + \sigma_1 \cdot \left(1 + \frac{n-2}{n^2}\right) \quad (2)$$

The relationship between maximum stresses during pre-bending and maximum bending stresses during application of a given service load is obtained from the condition $\sigma_4 \geq \sigma_4^*$:

$$\sigma_1 \leq \sigma_3 \cdot \frac{n}{n^2 - 1} \quad (3)$$

The following is obtained from the strength condition $\sigma_4 \leq \sigma_{dop}$:

$$\sigma_3 \leq \frac{n+1}{n} \cdot \sigma_{dop} \quad (4)$$

The allowable maximum stress during pre-bending is obtained from expressions (3) and (4):

$$\sigma_1 \leq \frac{1}{n-1} \cdot \sigma_{dop} \quad (5)$$

It can be concluded that during pre-bending we can obtain, not only a greater bearing capacity of the built-up beam, but also the zero beam deflection under a given load during service life

References

- [1] Bosin, G., Rodsky, M., Wolkoff, H. "Statics and Strength of Materials", Mc Graw-Hill, New York, 1988.
- [2] Kinsky, R., Engineering Mechanics and Strength of Materials, Mc Graw-Hill, Sydney, 1986.
- [3] Šimić, V., Otpornost materijala I, 2nd edition, Školska knjiga, Zagreb, 2002.
- [4] Ugural, A., Fenster, S., Advanced Strength and Applied Elasticity, Elsevier, New York, 1987.

Diana Šimić; Ph.D.(Civ.Eng.), Assistant Professor, Faculty of Civil Engineering, University of Zagreb, Department of Technical Mechanics, Kačićeva 26, 10000 Zagreb, Croatia
Phone: +385 1 4639 426; Fax: +385 1 4828 049; e-mail: dianas@grad.hr

SYMBOLIC COMPUTATION OF EXTENDED LINDSTEDT- POINCARÉ METHOD WITH MULTIPLE TIME SCALES

Rudolf R. Pušenjak, Maks M. Oblak

Keywords: Lindstedt-Poincaré method, time scales, cubic nonlinearities, symbolic computation

1. Introduction

Extended Lindstedt-Poincaré (EL-P) method with multiple time scales recently gained much attention among researchers due to possibility of computing nonlinear almost periodic oscillations. For this purpose, multiple time scales are introduced by means of nonlinear frequencies of the dynamical system, which can be incommensurate in general. When response of nonlinear dynamical system is expressed in form of power series of perturbation parameter, then collecting of all terms of like powers generate sequence of linear partial equations, which must be solved successively. In solving subsequent equations, secular terms appear, which must be eliminated. Elimination of secular terms leads to the solvability conditions of various orders. Symbolic computation of set of obtained equations with growing complexity is proposed to automatically derive the solvability conditions of several orders and to compute coefficients, which determine solutions of particular linear partial differential equations.

2. Extended Lindstedt-Poincaré method

EL-P method is applied to N -DOF dynamical systems with cubic non-linearities:

$$\frac{d^2 u_n}{dt^2} + \varepsilon c_n \frac{du_n}{dt} + \omega_{ln}^2 u_n + \varepsilon \sum_{m=1}^N \sum_{p=1}^N \sum_{q=1}^N \Gamma_{mpq} u_m u_p u_q = \sum_{k=0}^{\infty} \varepsilon^k p_{nk} \cos \omega t, \quad n=1,2,\dots,N \quad (1)$$

where ε is used as small positive perturbation parameter. Extension of the Lindstedt-Poincaré method with multiple time scales can be used to disseminate range of computation from periodic to almost periodic oscillations. Almost periodic oscillations are in general composed from incommensurate frequencies ω_n , ($n=1,2,\dots,N$) which define N time scales through relationship:

$$\tau_n = \eta \omega_n t, \quad (n=1,2,\dots,N), \quad \omega t = \sum_{l=1}^N a_l \tau_l \quad (2)$$

Due to time scales, usual time derivatives d/dt and d^2/dt^2 are replaced by differential operators:

$$\frac{d}{dt} = \eta \sum_{i=1}^N \omega_i \frac{\partial}{\partial \tau_i}, \quad \frac{d^2}{dt^2} = \eta^2 \sum_{i=1}^N \sum_{j=1}^N \omega_i \omega_j \frac{\partial^2}{\partial \tau_i \partial \tau_j} \quad (3.a,b)$$

The incommensurate frequencies ω_n are non-linear frequencies, which can be expressed about linear frequencies ω_{n0} of the system by means of power series:

$$\omega_n = \omega_{n0} + \sum_{k=1}^{\infty} \varepsilon^k \omega_{nk} \quad (4)$$

and equation (1) is replaced by nonlinear partial differential equations:

$$\eta^2 \sum_{i=1}^N \sum_{j=1}^N \omega_i \omega_j \frac{\partial^2 u_n}{\partial \tau_i \partial \tau_j} + \eta c_n \sum_{i=1}^N \omega_i \frac{\partial u_n}{\partial \tau_i} + \omega_{ln}^2 u_n + \varepsilon \sum_{m=1}^N \sum_{p=1}^N \sum_{q=1}^N \Gamma_{mpq} u_m u_p u_q = \sum_{k=0}^{\infty} \varepsilon^k p_{nk} \cos \left(\sum_{l=1}^N a_l \tau_l \right), \quad n=1,2,\dots,N \quad (5)$$

Using solution in the form of power series expansion:

$$u_n = \sum_{k=0}^{\infty} \varepsilon^k u_{nk}(\tau_1, \tau_2, \dots, \tau_N) \quad (6)$$

perturbation process leads to sequence of linear partial differential equations, which are solved successively. In this process, symbolic computation is performed in Mathematica[®] 5.2, which enables automatic computation of solvability conditions and unknown coefficients.

3. Almost periodic oscillations of undamped clamped-hinged beam

Symbolic computation of solvability conditions and subsequent determination of unknown coefficients is presented for undamped, free vibrating hinged-clamped beam, which can exhibit almost periodic oscillations. Because in this case no additional relation between time scales exist, solvability conditions can be symbolically generated in Mathematica[®] 5.2 by using program of the form:

```
Do[solv[n]=Coefficient[rhsequation[n],{Cos[τn]}];
Print[solv[n][[1]]=0],{n,1,2}];
```

where **rhsequation[n]** denotes right hand side of linear partial differential equation on the particular stage of the perturbation process. On the Fig. 1 results of EL-P method and comparison with results of IHB method are shown for both modes of almost periodic oscillations, where good agreement is established.

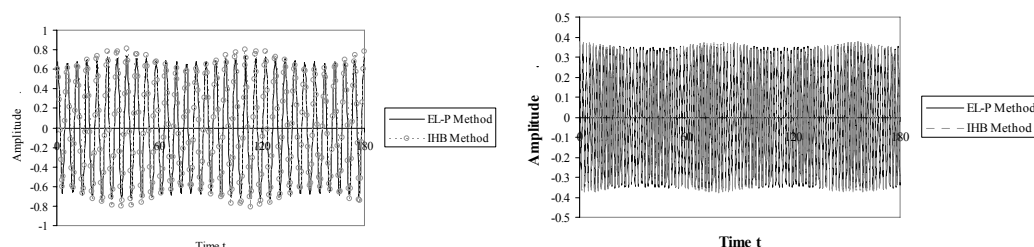


Figure 1. Almost periodic oscillations of undamped clamped-hinged beam

5. Conclusion

EL-P Method with multiple time scales is presented for symbolic computing almost periodic oscillations as well as for construction of fundamental resonance of undamped clamped-hinged beam. The results of almost periodic oscillations are compared by results of IHB Method, where good agreement is obtained. The presented method can be applied also for construction of other types of resonances such as superharmonic, subharmonic and combination resonances.

References

- [1] Lau, S. L., Cheung, Y.K., Chen, S., "Alternative Perturbation Procedure of Multiple Scales for Nonlinear Dynamics Systems," J. Appl. Mech., Vol. 56, 1989, pp. 667-675.
- [2] Nayfeh, A. H., Mook, D. T., "Nonlinear oscillations", John Wiley and Sons, New York, 1979
- [3] Pušenjak, R. R., Oblak, M. M., "Incremental harmonic balance method with multiple time variables for dynamical systems with cubic non-linearities," Int. j. num. meth. eng., Vol. 59, No.2, 2004, pp. 255-292.

Rudolf R. Pušenjak, PhD

University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, 2000 Maribor, Slovenia, +386 2 220 7801, rudi.pusenjak@uni-mb.si

Maks M. Oblak, PhD

University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, 2000 Maribor, Slovenia, +386 2 220 7801, maks.oblak@uni-mb.si

The Three-Dimensional Roughness Parameters Model for Surfaces of Shipboard Diesel Engines Crankshaft Bearings

T. Torims, J. Rudzitis

Keywords: three-dimensional, roughness model, roughness parameters, crankshaft bearings

Currently technologies available for ship repair enterprises are sufficient to ensure these requirements, however, they are very time consuming and consequently expensive. Therefore comprehensive research of the shipboard diesel engines crankshaft journals surfaces machining has been done. This new technological approach allowed to improve technological processes and to identify respective surface roughness parameters. Moreover, it allows performing a crankshaft journal grinding inside the housing without removing it from engine. Solving problems related with surface accuracy it is possible to considerably improve the crankshaft machining process as well as performance of maintenance operations and consequently overall quality of repair works.

In the above mentioned research the model of crankshaft journals surface roughness was determinate. This mathematical model of the surface roughness should be complete enough to describe the real surface and in the same time simple enough to be practically applicable. It was defined that every type of mechanical machining and surface creation process has their own, unique surface roughness topography. However, all mechanically processed surfaces principally can be divided into two groups: isotropic and anisotropic surfaces. Additionally to this classification, surface roughness mathematical functions can be divided into three groups, namely, with regular character, irregular character and mixed type. The shipboard diesel engines crankshaft journals machining with the mechanical grinding method, can be looked as anisotropic surface with irregular character of surface roughness. This kind of surface can be described by the normal distribution (Gaussian) law. It was determinate, that the optimal microtopographical model of the particular surface is composed by the following parameters: roughness height parameter R_{aT} and two spacing parameters S_{m1} and S_{m2} . It is important to note that the measurement basis for these roughness parameters is a mean plane of overall roughness.

The formulas of the surface roughness parameters R_{aT} , S_{m1} and S_{m2} were determinate. The microtopographical surface roughness height parameter:

$$R_{aT} = \frac{1}{A} \iint_{\Omega} |h(x, y)| dx dy \quad (1)$$

where: A – area of the viewed surface, mm²;
 $h(x, y)$ – surface roughness (points) deviation from the mean plane,
in coordinates x and y .

Topographical spacing parameters S_{m1} and S_{m2} :

$$S_{m1,2} = \frac{2}{n(0)_{1,2}} \quad (5)$$

where: $n(0)_{1,2}$ – number of „zeros” within one length unit, when the profile crosses the mean plane in the two perpendicular directions to each other.

Nevertheless, it is important to note that very often surface roughness measuring equipment or specific measuring conditions (as it is in the case of crankshaft bearings) do not allow reading of the microtopographical parameters R_{aT} , S_{m1} and S_{m2} . Therefore, to obtain microtopographical values

of above mentioned parameters the discrete method were applied, by breakdown of observing surface in to the several separate profiles. Thus calculations of the microtopographical values of R_{aT} , S_{m1} and S_{m2} have been done using discrete values recorded in two dimension profile. For this purpose the following formulas were determinate:

$$R_{aT} = \frac{1}{MN} \sum_{j=1}^N \sum_{i=1}^M R_a(i, j) \quad (6)$$

$$S_{m1} = \frac{1}{MN} \sum_{j=1}^N \sum_{i=1}^M S_{m1}(i, j) \quad (7)$$

$$S_{m2} = \frac{1}{MN} \sum_{j=1}^N \sum_{i=1}^M S_{m2}(i, j) \quad (8)$$

where: M – number of measurements within single profile;
 N – number of profiles examined;
 i – the examined profile;
 j – the measurement within profile.

Note: measurement S_{m1} and S_{m2} should be done in the perpendicular directions to each other. Naturally, these formulas allow making transition from discrete values of R_{aT} , S_{m1} and S_{m2} measured in profile to the surface roughness microtopographical parameters.

The research of the three-dimensional roughness parameters model for surfaces of shipboard diesel engines crankshaft bearings allows obtaining the following conclusions:

The model of microtopographical surface roughness is based on mean arithmetical deviation of surface - R_{aT} and two spacing parameters S_{m1} and S_{m2} - surface spacing parameters between the peaks. These two spacing parameters are measured in the perpendicular directions to each other. Moreover, this model fully complies with crankshaft grinding technology and its parameters can be obtained in practical work.

The microtopographical surface roughness parameters necessary to characterise crankshaft journals were developed. The formulas were provided allowing switch from discrete profile surface roughness parameters to the microtopographical parameters R_{aT} , S_{m1} and S_{m2} .

The experimental researches with the technological equipment have been made. Thus determinate the real impact of technological regimes to the microtopographical surface roughness parameters R_{aT} , S_{m1} and S_{m2} . Results achieved during these experiments were analysed multifactorially and this analysis allowed to determinate the empirical formulas for calculation of above mentioned parameters. Obtained formulas exactly reflect to the elaborated equipment for grinding of crankshafts journals. Therefore these formulas will be applicable to the ships' engines repairs, when the impact of the technological regimes needs to be evaluated.

Acknowledgement

This work has been partly supported by the European Social Fund within the National Programme, Support for the carrying out of doctoral study programmes and post-doctoral research projects, Support for the development of doctoral studies at Riga technical University.

Toms Torims, Dr. Sc. Ing.

Riga Technical University, Faculty of Transport and Mechanical Engineering, Institute of Mechanical Technology, Ezermalas street 6, Riga, LV – 1006, Latvia, Tel.: +371 7089701, Fax: +371 7089701
e-mail: ttorims@gmail.com

Janis Rudzitis, Dr. habil. Sc. Ing., Professor

Riga Technical University, Faculty of Transport and Mechanical Engineering, Institute of Mechanical Technology, Ezermalas street 6, Riga, LV – 1006, Latvia, Tel.: +371 7089701, Fax: +371 7089701
e-mail: arai@acad.latnet.lv

VIRTUAL TERRAIN MODELING USING GPS TRACKS

M. Ulbin, Z Ren

Keywords: terrain modelling, surface smoothing, GPS tracks

1. Introduction

For computational simulations of wheeled vehicle behaviour with random movement on known terrain it is necessary to define the virtual terrain model. This paper presents a method for creation of virtual terrain by using the data tracks recorded by handheld GPS device as cheap alternative to using photogrammetry and laser altimetry [1]. Discrete points from GPS tracks are first triangulated to form the initial surface representation. Grid mesh of different polygon densities is then mapped on triangulated surface, which is further smoothed by using the Overhauser splines. The later are necessary for creating additional points and moving averages for smoothing the peaks. The developed computer program enables export of terrain data in different formats for the purpose of computer simulation. The paper concludes with practical example, where the real terrain is scanned by the GPS device and then converted to virtual terrain by applying the developed methods.

2. Measuring the terrain

Terrain was measured using a Garmin GPSmap 60 CS GPS device without differential device [3]. Recorded track was intentionally very small and it consisted of only 143 points for the 1,5 km long track which covered about 1 ha of terrain.

3. Terrain modelling

Terrain was modelled using developed computer program MODTER. Result of terrain measurement was a cloud of points, which required surface interpolation.

3.1 Triangulation

Delauney triangulation [2, 5] was used in the next step of modelling virtual terrain. A cloud of measured points was converted to triangular polygons.

3.2 Virtual terrain smoothing

The first method is converting the triangular mesh into uniform rectangular grid with arbitrary resolution. The only possibility to enhance quality of virtual terrain, apart from better initial measurement, is creating additional points on the surface. The Overhauser splines [6] defined by existing points were used for that purpose. The moving averages method was used for smoothing the peaks which resulted from numerical errors [4].

3.3 Computer program MODTER

The program can read recorded track data from GPS devices or DEM (Digital Elevation Map) format. After appropriate modelling the virtual terrain data can be exported to either VRML (Virtual Reality Modeling Language) or ADAMS format for using in the simulation.

4. Conclusion

Modelling virtual terrain using GPS track data presents a cheap and effective way for building dynamic simulation environment. The quality of the terrain is mainly dependent on number of measured points, but using techniques for adding virtual points and smoothing surface makes it possible to create the virtual environment with minimal number of measured points. Developed techniques could be useful also for other areas of interest like mapping and civil engineering.



Figure 1. Measured terrain

Measured terrain from figure 1 was converted to virtual terrain in figure 2. It proved to be at sufficient quality and it is easy to measure even several times larger areas of terrain. In practice terrain data is measured with GPS device build in the vehicle, while at the same time the dynamic response of the vehicle is recorded.

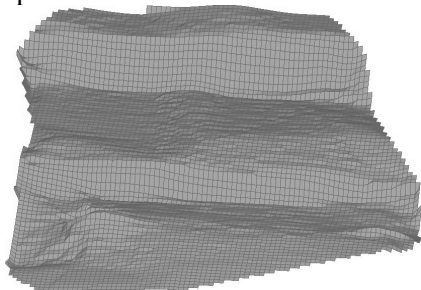


Figure 2. Smoothed virtual terrain

Acknowledgement

Financial support through the research program "Knowledge for safety and peace 2004-2010", 3311-04-828023, M2-0023 (MORS, MŠZŠRS, ARRS) is gratefully acknowledged.

References

- [1] Cowen, D.J., Jensen, J.R., Hendrix, Chad, Hodgson, M.E., and Schill, S.R, "A GIS-assisted rail construction econometric model that incorporates LIDAR data" Photogrammetric engineering and Remote Sensing, v. 66, no. 11, p. 1323-1328, 2000.
- [2] Ilfick, M. H., "Contouring by Use of a Triangular Mesh", Cartographic Journal, Volume 16, pp 24-28, 1979.
- [3] Garmin GPSmap 60 CS, "Owner's Manual", Rev. C, Sep, 2004
- [4] Kenney, J. F. and Keeping, E. S. "Moving Averages." §14.2 in Mathematics of Statistics, Pt. 1, 3rd ed. Princeton, NJ: Van Nostrand, pp. 221-223, 1962.
- [5] S.W.Sloan, G.T.Houlsby, "An Implementation of Watson's Algorithm for Computing 2-D Delauney Triangulations", Advanced Engineering Software, Volume 6, Number 4, 1984
- [6] Brewer, J., Anderson, D. C., "Visual interaction with Overhauser curves and surfaces", Computer Graphics, Vol. 11, No. 2, 1977, pp. 132-137.

doc. dr. Miran Ulbin

University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, 2000 Maribor, Slovenia, ulbin@uni-mb.si

prof. dr. Zoran Ren

University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, 2000 Maribor, Slovenia, ren@uni-mb.si

IMPROVED SYMMETRIC GAUSS-SEIDEL METHOD FOR SOLVING SPARSE LINEAR SYSTEMS APPEAR IN CFD

Zdravko Virag, Ivo Džijan and Severino Krizmanić

Keywords: Sparse linear systems, Iterative method, Symmetric Gauss-Seidel method

A typical subtask in Computational Fluid Dynamics (CFD) is to solve the system of linear algebraic equations $Ax = b$, where A is sparse coefficient matrix, x is solution vector and b is right hand side vector. In finite volume method for incompressible fluid flow three different types of the coefficient matrix appear: type I corresponds to the case of solving the Poisson equation with von Neumann boundary conditions (referred as pressure equation), type II to solving of the Laplace equation with Dirichlet boundary conditions and type III to the case of under-relaxed convection-diffusion equation. The aim of the paper is to propose two new variants of iterative solvers and to compare them with the some most frequently used Krylov space methods in terms of CPU time required to achieve prescribed solution accuracy. The new proposed methods SGS3 and SGS2 are based on the Symmetric Gauss-Seidel method in which the coefficient matrix A is approximated as $(L + D)(I + D^{-1}U)$, where L , D and U are lower triangle, diagonal and upper triangle part of original matrix A , respectively, I is identity matrix. The following table summarizes the generic algorithms for the two methods.

<i>The SGS3 method</i>	<i>The SGS2 method</i>
Initialize: $r^0 = b - Ax^0$, $\delta = bb$, $\Delta x^0 = 1$, $u^0 = A\Delta x^0$ Repeat while $rr / \delta < \text{tolerance}$: solve: $(L + D)(I + D^{-1}U)f = r^{(v-1)}$ compute: $v = Af$ solve: $(L + D)(I + D^{-1}U)s = r^{(v-1)} - v$ compute: $w = As$ solve: $\begin{vmatrix} u^{(v-1)}u^{(v-1)} & vu^{(v-1)} & wu^{(v-1)} \\ u^{(v-1)}v & vv & wv \\ u^{(v-1)}w & vw & ww \end{vmatrix} \cdot \begin{vmatrix} \alpha \\ \beta \\ \gamma \end{vmatrix} = \begin{vmatrix} r^{(v-1)}u^{(v-1)} \\ r^{(v-1)}v \\ r^{(v-1)}w \end{vmatrix}$ compute: $\Delta x^{(v)} = \alpha \Delta x^{(v-1)} + \beta f + \gamma s$, $x^{(v)} = x^{(v-1)} + \Delta x^{(v)}$ compute: $u^{(v)} = \alpha u^{(v-1)} + \beta v + \gamma w$ $r^{(v)} = r^{(v-1)} - u^{(v)}$	Initialize: $r^0 = b - Ax^0$, $\delta = bb$, $\Delta x^0 = 1$, $u^0 = A\Delta x^0$ Repeat while: $rr / \delta < \text{tolerance}$: solve: $(L + D)(I + D^{-1}U)f = r^{(v-1)}$ compute: $v = Af$ solve: $\begin{vmatrix} u^{(v-1)}u^{(v-1)} & vu^{(v-1)} \\ u^{(v-1)}v & vv \end{vmatrix} \cdot \begin{vmatrix} \alpha \\ \beta \end{vmatrix} = \begin{vmatrix} r^{(v-1)}u^{(v-1)} \\ r^{(v-1)}v \end{vmatrix}$ compute: $\Delta x^{(v)} = \alpha \Delta x^{(v-1)} + \beta f$, $x^{(v)} = x^{(v-1)} + \Delta x^{(v)}$ compute: $u^{(v)} = \alpha u^{(v-1)} + \beta v$ $r^{(v)} = r^{(v-1)} - u^{(v)}$

Here, we are going to apply the new proposed methods, the preconditioned PGMRES(5) method [2] and the preconditioned PTFQMR method [1] for solving linear systems for all three types of matrix, at three different grid sizes (different numbers of unknowns).

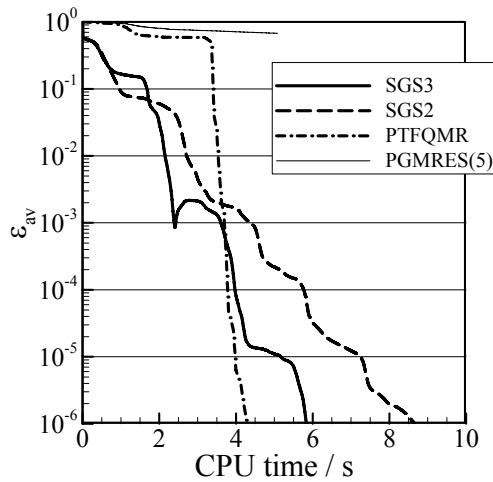


Fig. 1 Results for the matrix of type I

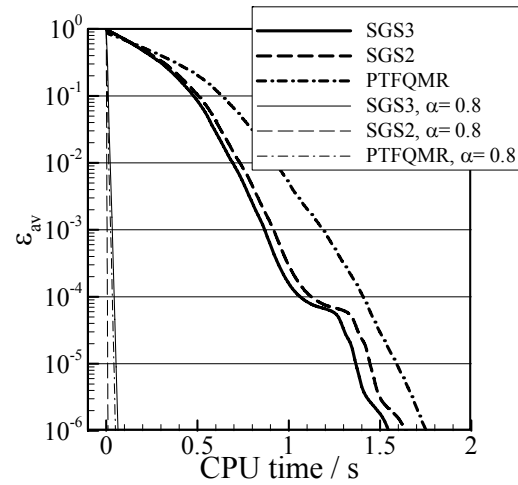


Fig. 2 Results for the matrix of type II and III

Fig. 1 shows the average solution error as a function of CPU time when solving set of 39547 equations with the coefficient matrix of the type I, and Fig. 2 with the coefficient matrix of type II and III, when applying the SGS3, the SGS2 and the PTFQMR method. For the matrix of type I, the PTFQMR method requires the shortest time to obtain the prescribed solution accuracy of $\varepsilon_{av} = 10^{-6}$. However, the SGS3 and SGS2 method show smoother average error reduction and faster error reduction to $\varepsilon_{av} = 2 \cdot 10^{-3}$. For the coefficient matrix of type III, all the methods are very good, and it is irrelevant which one would be applied. In the case of matrix of type II, the SGS3 method shows a slight advantage over the SGS2 and PTFQMR method. Again the SGS3 and the SGS2 show the better initial solution error reduction.

The new proposed SGS3 method is quite comparable to the PTFQMR method. When high solution accuracy is required the SGS3 method needs slightly more time than the PTFQMR method. The PTFQMR method has an unfavourable property of very slow average error reduction at the beginning of iterative process. In case of solving the set of nonlinear fluid flow equations a common practice is to reduce the average error just for an order of magnitude (the quicker the better) and then the recalculation of the matrix coefficients is performed. In this case the SGS3 method may show an advantage over the PTFQMR method, since it shows faster and smoother error reduction. This is a subject of further research.

References

- [1] Freund, R.W., 1993, A Transpose-free Quasi-Minimal Residual Algorithm for Non-Hermitian Linear Systems, SIAM Journal on Scientific Computing 14, 470-482.
- [2] Saad, Y. and M. Schultz, 1986, GMRES: A Generalized Minimal Residual Algorithm for Solving Nonsymmetric Linear Systems, SIAM Journal for Scientific and Statistical Computing 7, 856-869.

Dr. sc. Zdravko Virag

University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture
I. Lučića 5, 10000 Zagreb, Croatia, +385 1 6168137, zdravko.virag@fsb.hr

Dr. sc. Ivo Džijan

University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture
I. Lučića 5, 10000 Zagreb, Croatia, +385 1 61681241, ivo.dzijan@fsb.hr

Severino Krizmanić, dipl. ing.

University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture
I. Lučića 5, 10000 Zagreb, Croatia, +385 1 6168156, severino.krizmanic@fsb.hr

THE CALCULATION OF THERMPHYSICAL PROPERTIES FOR SOLIDS ON THE BASIS OF STATISTICAL MECHANICS

Jurij Avsec, Maks Oblak

Keywords: Statistical mechanics, thermomechanics, solids

1. Introduction

Billions of years ago, when enormous quantities of energy were released after the Big Bang, the fundamental particles followed by molecules were formed into more or less complex structures according to certain coincidental events [1]. In the period of several billion years of development, the Earth was also shaped as one of the planets in space with life on it. By some marvellous accident life was created on it, yet with a one flaw. In the process of evolution, Mother Nature formed human beings. By observing nature and through their own intelligence the human beings managed to subordinate all living beings in the process of evolution. At the same time, they also gradually learn how to exploit substances and materials. The ability of making tools and devices distinguished men from other living beings. Around 400,000 years ago, people were capable of making wooden spears and lances. They made tools and devices twice their own size. It has always been people's wish to make ever larger machines and devices. The reason is probably very simple, namely the leaders ruling at the time wanted to be ranked among the immortals. In Egypt, for example, the builders constructed pyramids in 2600 BC for the needs of the pharaohs, with the tallest being 147 m high Keops' pyramid. In 1931, the 449 m tall Empire State Building was built in New York. Currently, the last preparations are underway in Shanghai to construct a 1000 m high housing building. With the development of increasingly larger devices many inventors and scientists wished to reveal the secrets of micro cosmos. For centuries, clock makers were the most important representatives of those diminishing the size of devices. In the 17th century, the invention of the microscope opened up the way to the observation of microbes, plant and animal cells. Only in the late 20th century, micro-devices were technologically refined. Today, the size of transistors in integrated circuits is 0.18 micrometers and the transistors measuring 10 nanometres are already being developed in laboratories.

2. Results and discussions

The presented mathematical model was used to calculate the thermomechanical properties of state of some solids, namely:

1. Copper (Cu), as a typical representative of metallic bond.
2. Sodium chloride (NaCl), as a typical representative of ionic bond.

Tables 1 and 2 show the comparison between analytical calculations and experimental values [2]. The results obtained by analytical model show good agreement with the maximum AAD 5.2% for isentropic elastic modulus of copper.

Table 2 shows thermophysical properties for NaCl. The results obtained by analytical model show very good agreement with experimental data too. The maximum AAD is 6.2 % for isothermal elastic modulus.

The comparison of results showed a very good matching with experimental data, particularly for copper. Somewhat less optimum is the matching of analytical calculations with experimental data, such as NaCl. The reason for such a higher discrepancy between the measured and calculated results lies in the complex structure of crystals composed of diverse atoms. Figure 1 shows the relative deviation of the results for copper (Cu) in the wide temperature region between the analytical computation, and Eishah models obtained by correlated functions and comparison with experimental data. With the CF-1 is presented Eishah model with help of equation (21) and with CF.2 is presented Eishah model on the basis of equation (22) in the reference [6]. The results for all models obtained by statistical thermodynamics show relatively good agreement. The present analysis provides a good basis for further upgrading of this model allowing the calculation of a very accurate thermodynamic properties for calculation of thermal conductivity and electrical conductivity in molecular crystals.

Table 1: Isentropic and isothermal bulk modulus for copper

T K	V 10^6 m^3 /mol	K_S -an. N/mm ²	K_S -exp. N/mm ²	K_T -an. N/mm ²	K_T -exp. N/mm ²
250	7.04	134710	134640	130896	131064
300	7.06	134328	132801	128866	128532
500	7.12	139127	125982	132643	119189
800	7.26	109743	123552	102143	108339
1200	7.45	116142	112641	98520	96995
AAD		0.05184		0.03793	

Table 2: Isentropic and isothermal elastic modulus for sodium chloride

T K	V 10^6 m^3/mol	K_S -an. N/mm ²	K_S -exp. N/mm ²	K_T -an. N/mm ²	K_T -exp. N/mm ²
125	26.60	21536	25111	21317	24727
150	26.60	29025	25052	27621	24565
175	26.70	24722	24988	24327	24327
250	26.90	23699	24626	22818	23627
290	27.00	25003	24506	23583	23241
AAD		.0739		.0622	

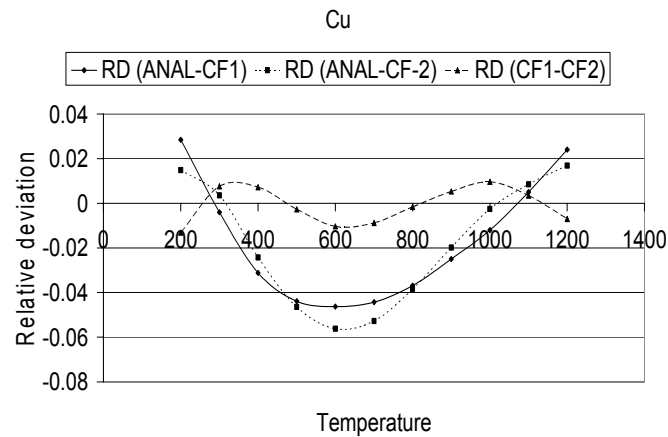


Fig 1: Relative deviation for thermal conductivity of Cu

Dr. Jurij Avsec, Assistant Professor

University of Maribor, Department of Mechanics, Smetanova 17, 2000 Maribor, P.O. BOX 224, SLOVENIA, E-Mail: jurij.avsec@uni-mb.si

Dr. Maks Oblak, Full Professor

University of Maribor, Department of Mechanics, Smetanova 17, 2000 Maribor, P.O. BOX 224, SLOVENIA, E-Mail: maks.oblak@uni-mb.si

CALCULATION OF CONTACT AREA OF SCREW – AND – NUT DRIVE THEARD SURFACES

Janis Rudzitis, Janis Shperbergs, Gints Konrads

Keywords: calculations, accuracy screw and nut drive, contact area, surface roughness

Abstract

Present researches in field surface roughness analyses and descriptions have a deep influence on practical manufacturing end engineering.

Target of present researches considering solution problems related with robotic moving part positioning and differential discrete location depending offered task. Robotic positioning accuracy mainly impact of kinematics pares manufactured tolerance and mating parts surface roughness on contact area.

The current research subject is contact area of screw and nut drive theared surfaces. Results of these researches are important in robotic transmission technology development.

This paper discusses methods for topography calculations and the correlation between the finishing processes parameters and the surface characteristics into high accuracy machine contact pairs: gear tooth's, bearings and screw and nut drive designs in general.

The present reference is focused on machine part's screw and nuts pare contact area spot calculations.

The mooving parts of contempoary robotic solutions often based on accurate screw and nut driving sistems. Screw and nut pares accuraccy depend of threaded surface manufactured quality. The contemporary technology toolworks allow high level surface roughness parameters quality but some valeys and picks already eksists if random spots are testing. The basic research problem is smoll size of screw and nut contact area spot and this area is covered and closed for mesuring. At workshops exists least of different mesurement devices and systems, but usually theese are suitable for plates or cilinders. These processes have a long time research history and one of mine treatment quality characterisation is surface roughness. Recently access has been trough 2D-measurement analysis and mainly observes circuit or planed structures as a simple and visible and easy measurement able.

The new quality related with 3D measurement assessing theory, measurement practice and calculation methodical. These theories observe surface texture us a valley and pick sum in space co-ordinate system having micro and macro measurement rate.

In recent research practice the schemes of contact between rough and ideal surfaces as well as between two rough surfaces are based on the assumption that the aspirates of mating parts are described by normal homogenous fields and the mean planes of both surfaces are parallel. The new quality related with 3D measurement assessing theory, measurement practice and calculation methodical. These theories observe surface texture us a valley and pick sum in space co-ordinate system having micro and macro measurement rate.

The present new view through random field theory and 3D assess is novelty of these references. The new access presents new mathematical method of surface roughness 3D parameters analysis never is used in engineering particle. The classics Gaussian theory allows create random waviness roughness field mathematics modelling compatible PC technology using.

In future these theory have predisposition create the new generation measurement units predictable to screw surface measuring directly on contact area. It will have important for robotic mating parts surface accuracy improving.

The calculations are based on using a criterion of contact between mating surfaces of friction, based on the application of standardised roughness parameters (EN ISO 12085:1997; EN ISO 4287: 19980)

Contemporary robotic driving systems have different transmissions including electromechanical units such as step motors with line or circuit screw and nut design. Screw and nut couple accuracy make summary error impact positioning pointing accuracy.

According adverse point of view these research has novelty in this research field. Patents and source analyse considering field task show little world wide achievement in this field and any international engineering summit always has great interest about reference and access explaining.

- [1] Janis Rudzitis, Professor Dr. sc. eng;. , Riga Tehnical University, 1 Kalku Street, Riga PDP, LV-1658, Latvia, Ph: +371 27089701; e-mail: aria@acad.latnet.lv
- [2] Janis Shperbergs M.sc.eng, Riga Tehnical Universit Liepaya Branch, Vanes Street 4, Liepaya PDP LV 3407, Latvia, Mob Ph +371 26564687; e-mail: ritti@tvnet.lv
- [3] Gints Konrads m.sc.eng, Riga Tehnical University, 1 Kalku Street, Riga PDP, LV-1658, Latvia, Ph: +371 27089701; e-mail: aria@acad.latnet.lv

Janis Rudzitis, Professor Dr. sc. eng;. , Riga Tehnical University, 1 Kalku Street, Riga PDP, LV-1658, Latvia, Ph: +371 27089701; e-mail: [**aria@acad.latnet.lv**](mailto:aria@acad.latnet.lv)

Janis Shperbergs M.sc.eng, Riga Tehnical Universit Liepaya Branch, Vanes Street 4, Liepaya PDP LV 3407, Latvia, Mob Ph +371 26564687; e-mail: [**ritti@tvnet.lv**](mailto:ritti@tvnet.lv)

Gints Konrads m.sc.eng, Riga Tehnical University, 1 Kalku Street, Riga PDP, LV-1658, Latvia, Ph: +371 27089701; e-mail: [**aria@acad.latnet.lv**](mailto:aria@acad.latnet.lv)

PROGRESSIVE FAILURE ANALYSIS OF ORTHOTROPIC PLATE LOADED BY TRANSVERSE LOW-VELOCITY IMPACT

T. Kroupa, R. Zemčík, V. Laš

Keywords: FEM, transient analysis, contact, composite, progressive failure analysis, stress waves, low-velocity impact

1. Introduction

This article is aimed at experimental and numerical investigation of the failure of composite material under impact loading. A small impactor made of steel or glass impacts on composite material and produces failure. A progressive failure analysis is used. Failure criterion LaRC04 (developed in NASA Langley research center) is used for failure analysis. Results from numerical simulations are compared with experimental results.

2. Transient dynamics

Impact event is transient dynamic contact problem. It is described by the set of 2nd order partial differential equations given below.

$$\begin{aligned}\frac{\partial}{\partial x} \sigma_{xx} + \frac{\partial}{\partial y} \sigma_{xy} + \frac{\partial}{\partial z} \sigma_{xz} &= \rho \frac{\partial^2 u}{\partial t^2}, \\ \frac{\partial}{\partial x} \sigma_{xy} + \frac{\partial}{\partial y} \sigma_{yy} + \frac{\partial}{\partial z} \sigma_{yz} &= \rho \frac{\partial^2 v}{\partial t^2}, \\ \frac{\partial}{\partial x} \sigma_{xz} + \frac{\partial}{\partial y} \sigma_{yz} + \frac{\partial}{\partial z} \sigma_{zz} &= \rho \frac{\partial^2 w}{\partial t^2},\end{aligned}\tag{1}$$

where x, y, z are coordinates, u, v, w are displacements, $\sigma_{xx}, \sigma_{yy}, \sigma_{zz}, \sigma_{xy}, \sigma_{yz}, \sigma_{xz}$ are stresses, t is time and ρ is density of the material. Initial and boundary conditions must be attached to these equations.

2. Progressive failure analysis

The stress analysis is performed in each time increment of the analysis. The chosen failure criterion finds out elements with cracked matrix or fibers at the end of each time increment. The properties of stiffness matrix of the damaged elements are changed. The type of degradation of the stiffness matrix depends on the type of failure (matrix, fibers). LaRC04 failure criterion was used for the failure analysis (for details see full paper).

3. Experiments and analyses

The model of stress waves propagation in a thin orthotropic plate was verified with the use of the experimental results presented in [3] (Figure 1.). The failure model was verified on model of the impact on thin strip due to size of model of the plate.

Table 1. Approach velocities of the bullet and damage of the strip

Velocity [ms ⁻¹]	2.215	2.426	2.525	2.6205
Damage (experiment)	NO	NO	YES / NO	YES
Damage (analysis)	NO	NO	NO	YES

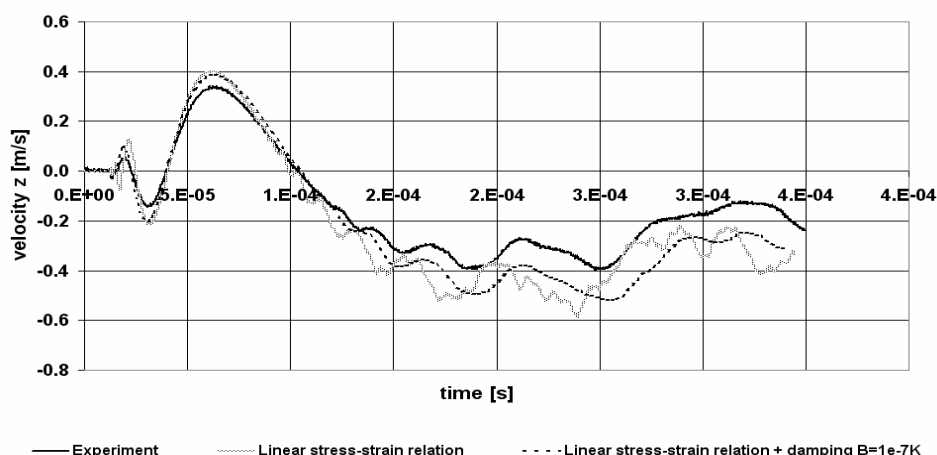


Figure 1. Velocity in transverse direction to the plate

6. Conclusion

The model of the stress waves propagation caused by impact event was verified and shows good agreement with the experimental data. Progressive failure analysis model was developed and verified on the impact event on thin orthotropic strip. Comparison of the experimental results and the results from the numerical analysis of the impacted plate with failure will be performed in close future.

Acknowledgement

This work was supported by the research project of the ministry of education of Czech Republic No. MSM 4977751303 and project GA AV IAA 200760611.

References

- [1] Hahn, T.,H., Tsai, S., W., "Nonlinear Elastic Behavior of Unidirectional Composite Laminae", Composite Materials Vol. 7., USA, 1972, pp. 102- 118.
- [2] Laš, V., "Mechanika kompozitních materiálů", ZČU Plzeň ČR, 2004
- [3] Kroupa, T.,Laš, V., "Transverse low-velocity impact on orthotropic plate", Proceedings 22nd DAS 2005, Monticelli Terme IT, 2005, pp. 254-255.
- [4] Kroupa, T., Zemčík, R., Laš, V., "Identification of composite material properties using non-linear stress-strain relation", Proceedings 8th AM2006, Srní ČR, 2006, pp.57-58.
- [5] Silvestre, T., P., Dávila, G., C., Camanho, P., P., Iannucci, L., Robinson, O., "Failure models and criteria for FRP under in-plane or three-dimensional stress state including shear non-linearity", NASA/TM-2005-213530, NASA Langley research center VA 23681 USA, 2005
- [6] Zemčík, R., Červ, J., Laš, V., "Numerical and experimental investigation of stress waves propagation in orthotropic strip", Proceedings 19th CM 2003, Nečtiny ČR, 2003, pp. 125-128.
- [7] Zemčík, R., Laš, V., „Identification of composite material properties using progressive failure analysis“, Proceedings 21st CM 2005, Nečtiny ČR, 2005, pp. 163-165.

Ing. Tomáš Kroupa

University of West Bohemia in Pilsen, Department of Mechanics, Univerzitní 22, 306 14, Plzeň, ČR, tel.: +420 377 632301, email: kroupa@kme.zcu.cz

Ing. Robert Zemčík, Ph.D.

University of West Bohemia in Pilsen, Department of Mechanics, Univerzitní 22, 306 14, Plzeň, ČR, tel.: +420 377 632301, email: zemcik@ntc.zcu.cz

Doc. Ing. Vladislav Laš, CSc.

University of West Bohemia in Pilsen, Department of Mechanics, Univerzitní 22, 306 14, Plzeň, ČR, tel.: +420 377 632301, email: las@kme.zcu.cz

BEAM MODEL FOR CREEP BUCKLING ANALYSIS

Domagoj Lanc, Goran Turkalj and Josip Brnić

Keywords: stability, buckling, beam structures, large rotations, large displacements, creep, material non-linearity

1. Introduction

In the field of structural engineering beams and frames constitute a very important class of load-carrying components, where they are applied both in their stand-alone forms and as stiffeners for some plate or shell assemblages. Because such structures, especially those of thin-walled cross-sections, could display very complex structural behaviour under a large displacement and rotation regime, the development of advanced non-linear beam models, which comprise both geometric and material inelasticity, has been a major activity of many structural engineering researchers in the recent years.

Columns under sustained loads are generally unstable in the regime of creep. That means that loss of stability may occur during a period of exploitation of structure even for loads lower than critical buckling load. Due to that reason stability is characterized by critical buckling time defined as load duration for which buckling deflections becomes infinitive.

The non-linear response of a load-carrying structure should be solved using numerical methods, e.g. the finite element method, and some of incremental descriptions like the total and updated Lagrangian ones, or the Eulerian description applied in this work. Each description utilizes a different structural configuration for system quantities referring and results in the form of a set of non-linear equilibrium equations of the structure. This set can further be linearized and should be solved using some incremental-iterative scheme, which consists of three main phases. The first or predictor phase comprises evaluating the overall structural stiffness and solving for the displacement increments from the approximated incremental equilibrium equations for the structure. Using the standard transformation process displacement increments of each finite element can be determined immediately. The second or corrector phase involves the geometry updating of each finite element and the determination of element nodal forces using some force recovery algorithm. The third or checking phase comprises checking if the adopted convergence criterion of iteration is achieved in the current increment by comparing with the pre-set tolerance value.

Standard Eulerian description is linear on element level and all geometrically non-linear effects are introduced through the transformation from local to global coordinate system [1]. Local Eulerian system follows the element chord during deformation and allows use of simplified strain-displacement relations on the local element level. Using such standard procedure it is unable to model pure torsional buckling because no geometric non-linearities due to cross-sectional deformation are taken into account to initiate it. From that reason an additional nonlinear part of stiffness matrix needs to be included which accounts for Wagner effect as a necessary condition for pure torsional buckling behaviour [2].

In this paper a one dimensional finite element for geometrically and materially nonlinear analysis of beam structures is presented. Beam members are supposed to be straight and prismatic. Spatial displacements and rotations are allowed to be large but strains are assumed to be small. Material is assumed to be homogeneous and isotropic.

Problem is approached through two phases. In first phase a pre-buckling behavior is modeled

through load deflection manner to reach appropriate instantaneous response of structure for applied load at zero moment while in second phase an explicit time integration scheme is used to reach critical buckling time. For modeling creeping material behavior Norton law have been applied. An original computer program BMCA is developed whose implementation is demonstrated through few test problems.

2. Example-Creep buckling of column

Column is loaded with constant axial force of 6000 N (it is about $0,89F_{cr}$). Instability is caused by material nonlinear behaviour due to creep. Norton power creep law is adopted with following constants: $n = 3$, $K = 0,58 \cdot 10^{-15} [\text{cm}^2/\text{N}]^n \cdot \text{h}^{-1}$. Results of analysis by computer program BMCA using 4 beam finite elements are shown on Figure 5 together with Walczak et al. [3] numerical results using 8 eight-noded plate finite elements as well as with NASTRAN results using 400 solid finite elements. Critical time is reached approximately after 15000 h what is in good agreement with reference results.

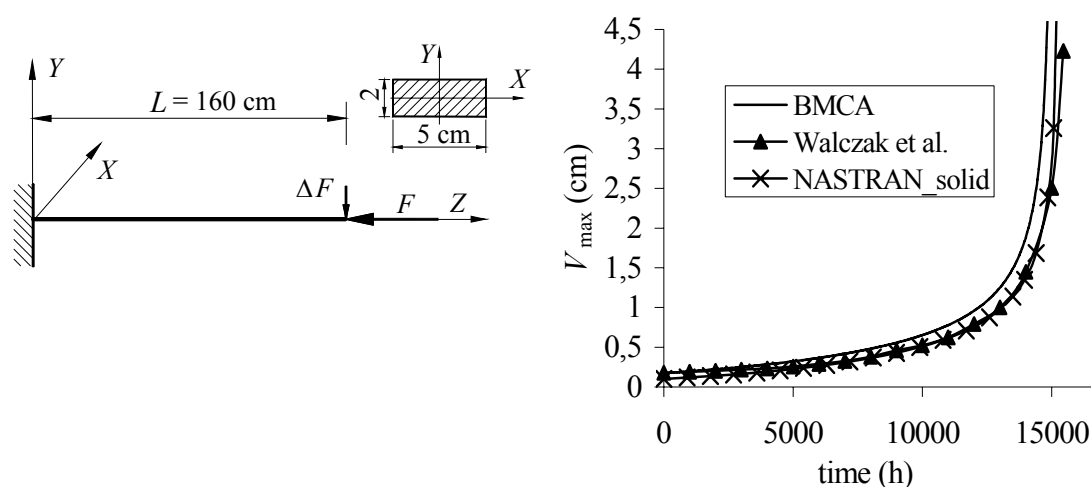


Figure 1. Creep buckling of axially loaded column

Acknowledgement

The research presented in this paper was made possible by the financial support of the Ministry of Science and Technology of the Republic of Croatia, under the project No. 0069-006.

References

- [1] Izzuddin, B.A. and Elnashai, A.S., "Eulerian formulation for Large-Displacement Analysis of Space Frames", *Journal of Engineering Mechanics*, Vol. 119, No. 3, 1993, pp 549-569.
- [2] Turkalj, G., Brnić, J. and Prpić-Oršić, J., "Large Rotation Analysis of Elastic Thin-Walled Beam-Type Structure using ESA Approach", *Computers & Structures*, Vol. 81, 2003, pp 1851-1864.
- [3] Walczak, J., Sieniawski, J., Bathe, K.J., "On the analysis of creep stability and rupture", *Computers & Structures*, Vol. 17, No.5-6, , 1983, pp 783-792.

Domagoj Lanc, Assistant, Ph.D.

University of Rijeka, Faculty of Engineering, Department o Engineering Mechanics, Vukovarska 58, Rijeka, Croatia, Phone: ++385 51 651 495, Fax: ++385 51 651 490, e-mail: dlanc@rijeka.riteh.hr

Goran Turkalj, Assoc. Professor, Ph.D.

University of Rijeka, Faculty of Engineering, Department o Engineering Mechanics, Vukovarska 58, Rijeka, Croatia, Phone: ++385 51 651 499, Fax: ++385 51 651 490, e-mail: turkalj@rijeka.riteh.hr

Josip Brnić, Full Professor, Ph.D.

University of Rijeka, Faculty of Engineering, Department o Engineering Mechanics, Vukovarska 58, Rijeka, Croatia, Phone: ++385 51 651 491, Fax: ++385 51 651 490, e-mail: brnic@rijeka.riteh.hr

STRENGTH ESTIMATION OF ADHESIVELY BONDED JOINT BY USING THE ADHESIVE STRENGTH LAW

Kozo Ikegami

Keywords: Adhesive resin, Bonded Joint, Deformation, Strength, Estimation, Strength law

1. Introduction

Adhesively bonded joints have many advantages for structural usages compared with mechanical fastenings such as bolts, nuts and rivets. However, they do not always have enough reliability as to the joint strength. This is caused by the fact that the strength of adhesively joint is difficult to estimate. A rational method is necessary for the estimation of the adhesive joint strength. In this paper, a method of estimating the strength of adhesively bonded joints is proposed and the method is applied to the strength estimation of adhesive double lap bonded joint under tensile-shear loading conditions.

2. Factors affecting the joint strength

Adhesive strength is affected by the many physical and chemical factors as well as bonded configurations. There is no way to evaluate the adhesive strength from bonding conditions. The adhesive strength necessary for strength design of adhesive joint has to determine experimentally.

3. Correlation between stress distribution and strength estimation

The joint strength is dominated by the critical loads of adherends, adhesive layer and adhesive interface. The critical load means the allowable load of the corresponding part without fracture. The critical load is able to estimate by correlating the stress distributions with the strength laws of the materials of adherends and adhesive layer. However what kind of strength law should be applied to the stress distribution of adhesive interface, is unknown solution. This is one of the most difficult matters to estimate the critical load of adhesive interface and to correlate the stress distributions of joints with estimating the joint strength.

4. Adhesive strength law to adhesive interface

A joint strength can be obtained by using the strength laws of the adherend and adhesive resin. But thus-obtained strength is usually higher than the actual strength. This is due to the lack of consideration of the strength of adhesive interface which is generally weaker than the adherend and adhesive resin. There are three kinds of critical loads of adhesive joints. The critical loads are the cohesive failure of adherends and adhesive layer, and the adhesive failure of adhesive interfaces. To include the strength of the interface for the estimation of joint strength, what kind of strength law is used for the interface, becomes of a subject of discussion.

The strength law of the adhesively boned elements is defined by the strength of the element under uniform combined stresses. The author proposes an experimental method to subject a butt joint of two thin-walled tubes to combined axial load, torsion and internal (or external) pressure [1].

Uniform combined stresses can be applied to adhesively bonded elements by this method. The author defines the strength of adhesively bonded elements determined by the method as the adhesive strength law.

5. Strength estimation by using the adhesive strength law

The author proposes a method of estimating the strength of adhesively bonded joints by using the adhesive strength law [2]. The stress conditions are determined for the adherends, adhesive layer and adhesive interfaces. The strength laws of the adherends, adhesive resin and adhesive interface are applied to the material elements of corresponding parts of the joint.

The critical loads or stresses of the elements, which causes the initial failure or yielding, are determined by correlating the strength laws with the stress conditions of the elements. The critical loads or stresses of adherends, adhesive layer and adhesive interface are determined by the minimum value of the critical loads or stresses of the material elements of adherends, adhesive layer and adhesive interface, respectively. The initial failure or yielding occurs at the position of material elements having the minimum value.

6. Strength estimation of double lap joint

The tensile-shear strength of double lap joints is estimated by using adhesive strength law. The stress distribution of the joint is analysed by the elastic finite element. The stress distribute more uniformly for thick adherends than for thin adherends. The Mises conditions are applied to the stress distributions of adherends and adhesive layers, and the adhesive strength law is applied to the stress distribution of the adhesive interfaces. The critical loads are determined for adherends, adhesive layers and adhesive interfaces. It is shown that the joint strength is dominated by the adhesive interfaces.

7. Conclusion

The method of estimating adhesively bonded joints is proposed on the basis of strength design. Three critical positions exist in adherends, adhesive layers and adhesive interfaces. These critical loads is estimated by substituting the stress distributions into the strength laws of corresponding parts. The joint strength is estimated by the minimum value among three critical loads of adherends, adhesive layers and adhesive interfaces. This method is applied to estimate the strength of double joints and the estimated strength fits for the experimental value. It is shown that the method of using strength laws of adherends, adhesive layer and adhesive interfaces is useful to estimate the adhesively bonded joint.

References

- [1] Ikegami, K., Kajiyama, A., Kamiko, S. and Shiratori, E., "Experimental Studies of the Strength of an Adhesive Joint in a State of Combined Stress", Journal of Adhesion, Vol.10, No.1, 1979, pp 25-38.
- [2] Ikegami, K., "Stress Analysis and Strength Design of Adhesively Bonded Joints", Transactions of Japan Society of Mechanical Engineers, Vol.50, No.457, 1959, pp 1557-1564.

Kozo Ikegami
Tokyo Denki University, Department of Mechanical Engineering
2-2 Kanda-Nishiki-cho, Chiyoda-ku, Tokyo, 101-8457 Japan
Tel/Fax: +81-3-5280-3370 e-mail: kikegami@cck.dendai.ac.jp

NON-STATIONARY STATE OF STRESS OF A VISCOELASTIC BEAM: DISPERSION AND ATTENUATION BEHAVIOUR

V. Adámek, F. Valeš

Keywords: non-stationary state of stress, viscoelastic, beam, dispersion, attenuation.

1. Introduction

When we study the propagation of waves in viscoelastic continuum we find out that their dispersion and attenuation occur, it means that the phase velocity of waves and their attenuation are frequently dependent. The phenomenon of dispersion is caused not only by their possible finite geometry as it is in the case of elastic bodies but also by their specific material behaviour (so called material dispersion). The second mentioned property, i.e. attenuation of wave amplitudes, is caused by dissipation of energy due to non-zero viscosity of material. This fact leads to wide application field of viscoelastic materials, e.g. impact absorber, dumpers etc.

Presented work focuses on analytical solution of non-stationary state of stress in an infinite thin viscoelastic beam, concretely, dispersion and attenuation properties are investigated. After the problem formulation in the first part of the paper, the brief description of deriving governing equations is given and the methods used for solving resulted motion equations are mentioned. Further the dimensionless form of dispersion equation is derived and solved and required dispersion and attenuation curves are presented. Finally, obtained results are compared with corresponding curves for the case of thin viscoelastic plate.

2. Problem description and initial assumptions

The geometry consists of an infinite thin beam with rectangular cross section of width $2b$ and height $2d$. The material of this beam is considered homogeneous, isotropic and linear viscoelastic and its properties are described by the discrete model of standard viscoelastic solid represented by Zener model. The upper surface of the beam is subjected to external pressure loading acting on the rectangular area of size $2h \times 2b$. This loading is uniformly distributed across the beam width and it changes according to cosine function along the beam length. The amplitude of the loading $\sigma_0(x)$ changes according to the Heaviside function in time.

The theory of small deformations and the Timoshenko beam theory are used for deriving final motion equations. Consequently, the total deflection angle of beam can be expressed as partial derivation of beam deflection $v(x,t)$ with respect to x and in addition to this, it is equal to the sum of deflection angle caused by the bending moment acting on an element and the angle that corresponds to the shear strain of this element.

3. Solving procedure

Motion equations of the problem are derived based on the assumptions mentioned above and analogously to [4] where the problem of elastic beam was solved. Transverse force and bending moment are taken into account according to the Timoshenko beam theory. The constitutive equations for the standard linear viscoelastic solid can be derived using the principle of superposition as in [1] where the problem of a thin viscoelastic plate was solved. Corresponding deformation equations for the nonzero normal and shear strain components implying from the theory of small de-

formation can be found e.g. in [3]. The final form of motion equations is represented by two dependent partial integrodifferential equations of the second order. With respect to experience acquired in [2], the method of integral transformations is used to solve this system of equations. Concretely, the Laplace transformation in time domain and the Fourier transformation in space domain were applied in sequence.

4. Dispersion and attenuation behaviour

Dispersion and attenuation properties are investigated based on the analysis of singularities of deflection and deflection angle integral transforms. All singularities will be poles and it can be easily shown that the first pole is $p = 0$. This pole has no physical sense in the case of infinite beam but for a finite beam it is non-zero and represents the motion of beam as solid. The other singularities can be found by solving dimensionless dispersion equation

$$(\omega d)^2 \xi^4 + \{ \kappa U_1(\omega d, \xi) [3 + (\omega d)^2] + U_2(\omega d, \xi)(\omega d)^2 \} \xi^2 + \kappa U_1(\omega d, \xi) U_2(\omega d, \xi)(\omega d)^2 = 0, \quad (1)$$

where the product ωd represents dimensionless wave number, ξ is dimensionless complex variable, κ is the Timoshenko shear coefficient and $U_i(\omega d, \xi)$ ($i = 1, 2$) are complex functions containing material and geometric parameters. The solving of this equation that was performed in system Matlab has shown that in fact there exist two different negative real roots and two pairs of complex conjugate roots for $\omega d \neq 0$. Real and imaginary parts of these roots describe the attenuation and dispersion behaviour of harmonic stationary stress wave propagated in this type of continuum, respectively. With respect to this fact, two real roots are physically inadmissible for the absence of dispersion in the solid with finite geometry. Owing to the complex conjugation of remaining roots, only two dispersion curves and two attenuation curves exist as in elastic case (see [4]). As it was found by comparing obtained curves with results stated in [1], both dispersion and attenuation curves are very similar to them for thin viscoelastic plates. Dispersion curves differ in asymptote positions, mainly the second curve, while the most marked difference between attenuation curves occur in the maximum of the first curve.

5. Conclusions

The main result of this work is the derivation of dimensionless compact form of dispersion equation for the case of an infinite thin viscoelastic beam. As it was shown (see the full version of paper), both dispersion equation and dispersion and attenuation curves are very similar to that in the case of thin plate.

Acknowledgement

This research was supported by the Research Project MSM 4977751303 and by the grant of GA CR No 101/06/0213.

References

- [1] Adámek, V., "Analytical, Numerical and Experimental Investigation of Non-stationary State of Stress in a Thin Viscoelastic Plate", Ph.D. thesis, FAS UWB in Pilsen, 2004 (in Czech).
- [2] Adámek, V., Valeš, F., Laš, V., "Application of Integral Transformations to Analytical Solution of Non-stationary States of Stress", Proceedings of Applied Mechanics 2003, Gliwice Poland, 2003, pp 7-12.
- [3] Graff, K. F., "Wave Motion in Elastic Solids", Dover Publications, New York, 1975.
- [4] Valeš, F., "State of Stress in a Thin Beam Transverse Non-stationary Loaded – Timoshenko Model", Z 514/76, Institute of Thermomechanics, IT CAS CR, 1976 (in Czech).

Ing. Vítězslav Adámek, Ph.D.

University of West Bohemia in Pilsen, Department of Mechanics, Univerzitní 22, 306 14, Pilsen, Czech Republic, +420377632335, +420377632302, vadamek@kme.zcu.cz

Ing. František Valeš, Csc.

Czech Academy of Sciences, Institute of Thermomechanics, Vešslavínova 11, 301 14, Pilsen, Czech Republic, +420377236415, vales@cdm.it.cas.cz

ELASTIC MODELING OF FUNCTIONALLY GRADED MATERIALS: ONE DIMENSIONAL CASE

Z. Sharif Khodaei and J. Zeman

Generally speaking, the ultimate goal of every design is a product which fully utilizes properties of materials used in its construction. This philosophy, in its vast context, naturally leads to the employment of multi-phase composites with microstructure adapted to operation conditions. Functionally graded materials (FGMs) present one important man-made class of such material systems. Since their introduction in Japan in 1984 as barrier materials for high-temperature components, FGMs have proved to be an attractive choice for numerous applications such as wear resistant coatings, optical fibers, electrical razor blades and biomedical tools. To provide a concrete example, consider a microstructure of $\text{Al}_2\text{O}_3/\text{Y-ZrO}_2$ ceramics (Fig. 1) engineered for the production of all-ceramic hip bearings. In this case, controlled composition and porosity to achieve better long-term performance and hence lower clinical risks when compared to traditional metallic materials [3] is considered.

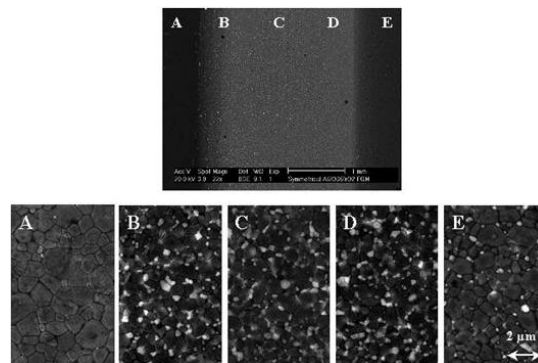


Figure 1. Graded microstructure of $\text{Al}_2\text{O}_3/\text{Y-ZrO}_2$ ceramics (Courtesy of J. Vleugels, K.U. Leuven)

As typical for all composites, the analysis of functionally graded materials is complicated by the fact that the explicit discrete modeling of the material microstructure results in a problem which is intractable due to huge degrees of freedom and its intrinsic randomness. As the most straightforward answer to this obstacle, models with a given smoothly varying material data are often employed. When the spatial non-homogeneity is assumed to follow a sufficiently simple form, this premise opens the route to very efficient numerical schemes, e.g., [6] and reference therein. Although this approach is very appealing from the computational point of view, its validity remains rather questionable as it contains no direct link with the underlying heterogeneous microstructure.

One possibility of establishing such a connection is to assert that the FGM locally behaves as a homogeneous composite characterized by a given volume fraction and use local effective media theories. Local averaging techniques have acquired a considerable attention due to their simplicity comparable with the previous class of models [4]. Still, despite substantial improvement in physical relevance of the model, local averaging may lead to inaccurate results. This was demonstrated by a study of Reiter et al. [5] which clearly shows that the averaging technique needs to adapt to detailed

character of the microstructure in a neighborhood of the analyzed material point. When considering the local averaging techniques, however, such information is evidently lost as the microstructural data has been lumped to volume fractions.

Another appealing approach to FGM modeling is an adaptive discrete analysis of the structure. Such a strategy has been, e.g., recently adopted by Vemaganti and Deshmukh [7] in the framework of goal-oriented modeling. Without a doubt, this approach yields the most accurate results for a given distribution of phases. However, its extension to include inevitable randomness of the microstructure seems to be an open problem.

The treatment of FGMs as random, statistically non-homogeneous, composites offers a possibility to apply a general machinery of statistical continuum mechanics. In this framework, overall response of the media is interpreted using the ensemble, rather than spatial, averages of the involved fields. This gives rise to a correct treatment of non-local effects when combined with appropriate techniques for estimating statistics of local quantities. Examples of FGMs-oriented studies include the work of Buryachenko and Rammerstorfer [1] who employ the multi-particle effective field method and the study by Luciano and Willis [2] based on the Hashin-Shtrikman energy principles. Both these studies, however, work with geometrically simple domains and neglect the finite size of the microstructure.

In this paper, we propose a modeling strategy for the linear elastic analysis of FGMs systematically based on a realistic random microstructural model. The overall response of FGMs is addressed in the framework of stochastic Hashin-Shtrikman variational principles. To allow for the analysis of finite bodies, recently introduced discretization schemes based on the Finite Element Method (FEM) and the Boundary Element Method (BEM) are employed to obtain low-order statistics of local fields.

Representative numerical examples are presented to compare the performance and accuracy of both schemes. To gain insight into similarities and differences between these methods and to minimize technicalities, the analysis is performed in the one-dimensional setting. It is demonstrated that the BEM formulation is computationally more efficient and accurate than the FEM. Nevertheless, the BEM-based analysis leads to smaller computational savings than for classical elasticity problem.

References

- [1] Buryachenko, V.A. and Rammerstorfer, F.G., "Local effective thermoelastic properties of graded random structure matrix composites". *Archive of Applied Mechanics*, Vol. 71, Nos. 4–5, 2001, pp 249–272
- [2] Luciano, R. and Willis, J.R., "Non-local constitutive equations for functionally graded materials", *Mechanics of Materials*, Vol. 36, No. 12, 2004, pp 1195–1206
- [3] Lukáš, P. and Vrána, M. and Šaroun, J. and Ryukhtin, V. and Vleugels, J. and Anne, G. and Van der Biest, O. and Gasik, M., "Neutron diffraction studies of functionally graded Alumina/Zirconia ceramics", *Materials Science Forum*, Vols. 492–493, 2005, pp. 201–206
- [4] Markworth, A.J. and Ramesh, K.S. and Parks, W.P., "Modeling studies applied to functionally graded materials". *Journal of Materials Science*, Vol 30, No. 9, 1995, pp 2183–2193.
- [5] Reiter, T. and Dvorak, G.J. and Tvergaard, V., "Micromechanical models for graded composite materials". *Journal of the Mechanics and Physics of Solids*, No. 45, Vol. 8, 1997, pp. 1281–1302.
- [6] Sládek, V. and Sládek, J. and Zhang, Ch., "Domain element local integral equation method for potential problems in anisotropic and functionally graded materials". *Computational Mechanics*, Vol. 37, No. 1, 2005, pp 78–85.
- [7] Vemaganti K. and Deshmukh, P., "An adaptive global-local approach to modeling functionally graded materials", *Computer Methods in Applied Mechanics and Engineering*, Vol. 195, 2006, pp 4230–4243.

Ing. Zahra Sharif Khodaei

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Fax:+420-2-24310775,
e-mail:zahra.sharif@gmail.com

Ing. Jan Zeman, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354482, Fax:+420-2-24310775,
e-mail:zemanj@cml.fsv.cvut.cz

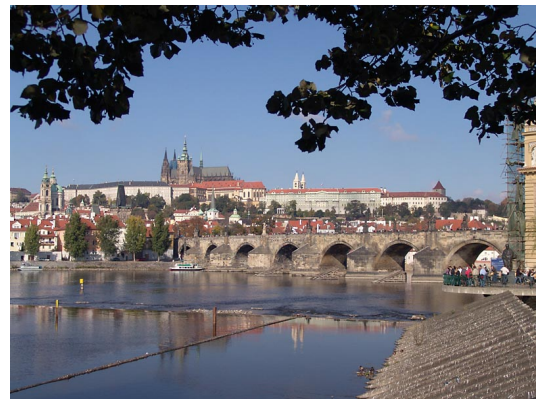
PRAGMATIC MULTI-SCALE AND MULTI-PHYSICS ANALYSIS OF THE CHARLES BRIDGE IN PRAGUE

Jan Zeman, Jan Novák, Michal Šejnoha and Jiří Šejnoha

The Charles Bridge in Prague is one of the listed monuments having the paramount historical importance. On 9th July 1357 in the morning, exactly at 5 hours and 31 minutes, which corresponds to the numerical sequence 135797531, Charles IV of the Louxembourg Dynasty, King of Bohemia and Germany and Italy Roman Emperor, laid the foundation stone and entrusted Peter Parlér and his stone mason's workshop to put up the bridge. The bridge itself had been completed almost after fifty years in 1406. Since that time, it has experienced several severe damages, mainly due to repeated floods and water erosion. These had been followed by reconstruction, rehabilitation and strengthening measures, which resulted in a large variability of materials as well as building techniques in various parts of the bridge.



(a)



(b)

Figure 1. The Charles Bridge in Prague (a) After the flood in 1890, (b) Current state

In the past three years, a lively debate on the load-bearing capacity and safety of the bridge had been underway among the Civil Engineering community in the Czech Republic. As one of the outcomes of this discussion, the research group of Prof. Jiří Šejnoha has been contacted to perform the computational assessment of the current state of the structure. There is no doubt that the repair of the Charles Bridge itself is extremely demanding professionally and carefully watched by the whole society. Therefore, it is natural that the final judgment should be carried out based on cogent material models and using progressive computational means. Of course, as typical for analysis of historical structures, a certain compromise, based on a careful engineering judgment, must be made between the theoretical appeal of the computational model and necessary simplifications due to limited data available; see, e.g., [1] for an excellent discussion on this subject. Even with these limitations, it is clear that advanced computational tools and methods can provide a reliable basis for diagnostics of historical structures as well as possible rehabilitation works, especially when compared with simplified linear elastic models.

In our particular case, the following actions should be included in the analysis to cover the most relevant situations:

- self-weight of the structure,
- loading due to spatially and temporarily variable temperature change,
- hydrostatic and hydrodynamic water pressure including possible uplift,
- floating vessel impact.

Strictly speaking, this list suggests that the analysis to be performed should be

- *multi-scale* due to heterogeneity of the structure on several spatial scales,
- based on *non-linear material* models taking into account quasi-brittle response of masonry materials,
- *multi-physics* to incorporate climatic effects on the structure as well as interaction of the bridge with water,
- *time-dependent* to cover the behavior of the bridge in the different parts of the year,
- *three-dimensional* as the bridge is a very massive structure and hence dimensionally reduced models are not appropriate,
- *dynamical* to properly describe a possible impact of a vessel to a bridge pier during flood.

Clearly, such an analysis is not currently feasible, not only from the viewpoint point of theoretical and numerical difficulties, but mainly due to very limited input data available. The engineering approach adopted in this work is to “decouple” the analysis into independent simplified parts, which are solved using appropriate specialized numerical tools. Outputs of these sub-problems then serve as inputs for a rather detailed non-linear mechanical simulation allowing us to assess their impact on overall behavior of the bridge.

The goal of the current contribution is to provide a brief but still complete overview of all relevant aspects of the performed analysis. Naturally, due to space restrictions, rather limited information is provided for each topic; more elaborate description can be found in a technical report [2].

References

- [1] Lourenço, P.B., “Computations of historical masonry constructions”, *Progress in Structural Engineering and Materials*, Vol. 4, No. 3, 2002, 301–319
- [2] Šejnoha, J., Bittnar, Z., Šejnoha, J., Zeman, J., Novák, J. and Janda, Z., “Stress and damage analysis of the Charles Bridge in Prague”. Technical report, Mott MacDonald Prague Ltd., 2005, 159 pages (In Czech).

Ing. Jan Zeman, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354482, Fax:+420-2-24310775,
e-mail:zemanj@cml.fsv.cvut.cz

Ing. Jan Novák

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354606, Fax:+420-2-24310775,
e-mail:novakj@cml.fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354494, Fax:+420-2-24310775,
e-mail:sejnom@fsv.cvut.cz

Prof. Ing. Jiří Šejnoha, DrSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354492, Fax:+420-2-24310775,
e-mail:sejnoha@fsv.cvut.cz

MULTI -AXIAL STATIC STRENGTH CRITERION FIT FOR DUCTILE QUASI-ISOTROPIC MATERIALS

J. Fuxa, R. Kubala

Keywords: Multi-axial strength criterion, testing equipment, Haigh's diagram, fracture strain diagram, methodology

1. Introduction

The work [7] contains 35 selected strength criteria suggested for static stress of isotropic (quasi-isotropic) materials. Plastic deformation is usually effected by *slip*. The slip is conditioned by movement of (free) dislocations. For realisation of slip in polycrystalline material it is necessary to activate *at least five different slip systems (Mises)*.

2. Strength criterion

Stress conditions in the general plane ρ (determined by its normal $n_\rho (\alpha_1, \alpha_2, \alpha_3)$ in respect to directions 1, 2, 3 of principal stresses) are known from any textbook of mathematical theory of elasticity:

$$\sigma_\rho = \sigma_1 \cos^2 \alpha_1 + \sigma_2 \cos^2 \alpha_2 + \sigma_3 \cos^2 \alpha_3, \quad (1)$$

$$\tau_\rho = [(\sigma_1^2 \cos^2 \alpha_1 + \sigma_2^2 \cos^2 \alpha_2 + \sigma_3^2 \cos^2 \alpha_3) - (\sigma_1 \cos^2 \alpha_1 + \sigma_2 \cos^2 \alpha_2 + \sigma_3 \cos^2 \alpha_3)^2]^{1/2}. \quad (2)$$

Based on these variables one can calculate the value of “reference normal stress σ_R ” and “reference shear stress τ_R ” using expressions [6, 7]:

$$\sigma_R = \lim_{R \rightarrow 0} [\int_{(S)} \sigma_\rho \cdot dS / (4\pi R^2)] = (\sigma_1 + \sigma_2 + \sigma_3) / 3, \quad (3)$$

$$\tau_R = \lim_{R \rightarrow 0} [\int_{(S)} \tau_\rho^2 \cdot dS / (4\pi R^2)]^{1/2} = (1/15)^{1/2} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} \quad (4)$$

Using the mentioned findings of physics of metals (see full text) as our guidelines, we expect the strength criterion in the form:

$$\tau_R = f(\sigma_R) \dots \text{for example in the simplest form:} \quad (5)$$

$$\tau_R = A - B \cdot \sigma_R \dots \text{with } \sigma_1, \sigma_2, \sigma_3 < \sigma_{MC}. \quad (5a)$$

The concept of the measurement and control of the testing machine has been described in [1, 3], torsion test evaluation has been depicted in [4], FEM applications have been described in [2] and probability analysis has been gathered in [10].

3. Application

Three applications of the *reference shear/normal stress strength criterion* have been described:

- Strength criterion of the *steel* marked *11 600* by CSN standard (Czech Republic) [5],
- *Fracture strain diagram* by strength criterion explanation [6, 8],
- Cast-iron strength criterion.

For detail - see full text in this publication.

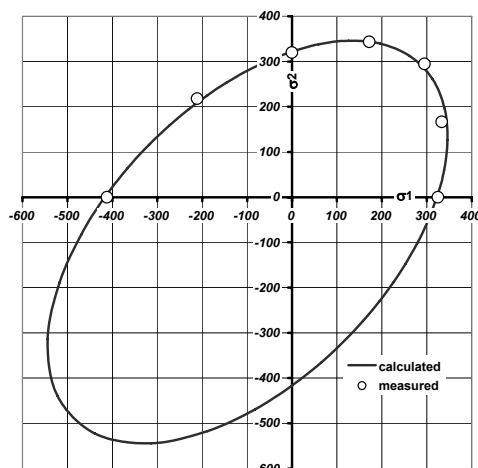


Figure 7. Haigh's diagram of the Malleable Cast Iron - data [9]

4. Conclusion

Three *applications* have been presented. All of them give very good correspondence among measured and computed data.

Acknowledgement

The paper was created within the frame of solution of the GAČR project No. 101/04/0475.

References

- [1] Fojtík, F., "The Adjustment of the Machine Schenck for the Conditions of ultiaxis Fatigue of the Material", Proceedings 2nd International PhD Conference on Mechanical Engineering 2004. Srní - Czech Republic, 2004, ISBN 80-7043-330-2, pp 23-24 (Book of Abstracts)
- [2] Frydrýšek, K., "The Torsion Test Simulating for Short Specimens", Sborník vědeckých prací VŠB-TU Ostrava, řada strojní, no.1, vol XLV, 1999, FS Ostrava, ISSN 1210-0471, pp 87-91
- [3] Fusek, M., "Řízení zkušebního stroje", Proceedings Aplikovaná mechanika 2002. Ostrava, 2002, pp 89-94
- [4] Fuxa, J., "Theory of the Torsion Test Evaluation", Proceedings 2nd Congress of Croatian Society of Mechanics 1997. Supetar - Croatia, 1997, pp 353-359
- [5] Fuxa, J., "Strength Criterion for Quasi-Isotropic Statically Loaded Material", Proceedings 18th DAS. Steyer - Austria, 2001, pp 77-78 (Book of Abstracts)
- [6] Fuxa, J., "Fracture Strain Diagram and Strength Criterion", Proceedings Engineering Mechanics 2003. Svratka - Czech Republic, 2003, pp 94-95 (Book of Abstracts)
- [7] Fuxa, J., "Výzkum kritérií pevnosti kvaziizotropních materiálů namáhaných v podmínkách víceosé napjatosti", Final report GAČR no 101/99/1245, VŠB-TU Ostrava - Czech Republic, 2001
- [8] Kolmogorov, V.P, Bogatov, A.A., "Plastičnost i razrušenie", Metallurgija Moskva, 1977
- [9] Lamaševskij, V.P., Makoveckij, I.V., "Děformirovanie i pročnosť kovkogo čuguna při složnom naprjaženom sostojanii", Strength of Materials, no 5 (377), 2005, ISSN 0556-171X, pp 71-83
- [10] Rojíček, M., "Probability analysis of problems leading to systém of linear equations", Proceedings Engineering Mechanics 2005. Svratka - Czech Republic, 2005, ISBN 80-85918-93-5, pp271-272

Prof. Ing. Jan Fuxa, CSc.

VŠB-TU Ostrava, Department of Mechanics of Material, Faculty of Mechanical Engineering, VŠB - Technical University Ostrava, 17. listopadu 15, 708 33 Ostrava - Poruba, Czech Republic, 420-732-4412, 420-69-6916490, jan.fuxa@vsb.cz

Ing. Rostislav Kubala, CSc.

VŠB-TU Ostrava, Department of Mechanics of Material, Faculty of Mechanical Engineering, VŠB - Technical University Ostrava, 17. listopadu 15, 708 33 Ostrava - Poruba, Czech Republic, 420-732-3495, 420-69-6916490, rostislav.kubala@vsb.cz

EXPERIMENTAL INVESTIGATION AND NUMERICAL ANALYSES ON AIRFOILS WITH HIGH LIFT DEVICE AND SPOILERS

Ing. N. Součková , Ing. L. Popelka, Ing. M. Matějka

Keywords: Airfoil, High Lift Devices, Airbrake, Spoiler, Particle Image Velocimetry, Visualisation, Fluent

1. Introduction

The reason of choice this problems were design methods of airfoils. The airfoils are designed for travel condition, but it isn't designed for landing and taking off. High lift devices and airbrakes (spoilers) are used for good behaviour of airplane in those flight regimes.

Nowadays it's still actual sailplane airfoils problems with fulfilment of required stalling speed and gliding angle required by existent regulations. Likewise the problem with designing sailplane airbrakes with sufficient effectivity exists [1].

2. Experiment

2.1 PIV - Particle Image Velocimetry

It was used PIV devices of Division of fluid dynamics and thermodynamics [2]. The measurement was carried out in closed circuit wind tunnel with open test section [3]. The experiment proceeded at Reynolds number $Re = 3,3 \cdot 10^5$, angle of attack $\alpha = 0^\circ, 6^\circ$ and turbulence intensity $Tu = 3,8\%$. The models with chord 300 mm and span 400 mm are airfoils MS (1)-0313 with split flap, NACA 23012 with tilting spoiler and FX66-17AII-182 with extending spoiler.

2.2 Pressure distribution measurement

Measurement of pressure distribution was accomplished for airfoil FX66-17AII-182 alone and with extending spoiler in open circuit wind tunnel with closed test section (1200 by 400 mm) of the same laboratory at the same Reynolds number, turbulence intensity $Tu = 1,2\%$ and in angles of attack range $\alpha = -2^\circ; 0^\circ; 2,5^\circ; 5^\circ; 7,5^\circ; 10^\circ; 12,5^\circ$. It was used model with span 400 mm, chord 400 mm and a single row of pressure orifices on upper and lower surfaces all over chord in the middle of span.

2.3 Smoke-wire visualization

This method was used for supplementation of information about flow in sections of leading and trailing edge on airfoil NACA 23012 with tilting spoiler. Measurement proceeds at zero angle of attack and with regard to its character at very small velocity approximately 2 m/s. It was used camera linked with computer for record of character of flow.

3. Numerical Analyses

In this case it was tried to find a solution via pre-processor GAMBIT 2.1 and solver FLUENT 6.1, which involves also post-processor, so the solution is limited by accuracy second order. It was used turbulent free stream flow. The geometry of profiles procured from web site [4], flaps geometry was measured from models used in experiment. It was applied combination of structured mesh near

the airfoil and unstructured mesh in the vicinity for airfoils NACA 23012 with tilting spoiler and FX66-17AII-182 with extending spoiler. Structured mesh was used for the same airfoil without spoiler. It was used k- ϵ model of turbulence. In respect of using mesh was selected discretization upwind second order. Setting of solution responded visualization by PIV method and measurement of pressure distribution, which has the same Reynolds number, but lower turbulence intensity. The computation was realized at angle of attack $\alpha = 0^\circ$.

4. Conclusion

From results of PIV experiment is obvious separation of flow on both spoilers, which is evoking considerable drag increase. Moreover there is back flow and vortices in area behind the flap. It is known from theory that split flap causes lift increasing by influence of camber change of airfoil [5]. From measurement is indicated wider wake, which causes drag increasing in additional from real velocity distribution behind the flap is evident vortices separation.

The numerical analyses provide image of flow field in larger area than experiment, so it is possible to determine size of section influenced by spoiler, also stagnation point on leading edge, decrease of velocity in upper side and increase at the bottom. There is occurred vortex separation on trailing edge and back flow of airfoil NACA 23012 with tilting spoiler. In both cases was generated negative lift force at zero angle of attack. The negative value of lift coefficient corresponds with previous mentioned measurement and measurement in Stuttgart [6]. Variance between measured and computed lift coefficient indicates problem of steady computation, because there are unsteady processes in flow for case with spoiler. The difference between experimental and computed value of lift coefficient for case without spoiler is caused applied discretization scheme upwind second order. The velocity profiles are in correspondence with experiment. The increment of drag coefficient of airfoil NACA 23012 is 2-times lower than the same coefficient of airfoil FX66-17AII-182 as well the increment of lift coefficient is 1,4-times lower. It is evoked smaller area of separation and back flow.

Position of stagnation point and vortex separation on trailing edge from numerical analyses was confirmed by smoke-wire visualization method.

For other research of spoiler or airbrakes is important target to determine drag force, which is significant parameter of airbrakes. Also it is necessary to measure a few kind of spoiler with the same airfoil for comprehension of airbrakes (spoiler) problems.

References

- [1] „Předpis JAR 22“, Available from WWW: <<http://www.lba.de/englisch/lba/org/t/t4/m3jar221.htm>>, 2001.
- [2] Nožička, J., Adamec, J., Novotný, J., „Aerodynamický výzkum proudových polimetodou PIV“, VXIII. Medzinárodná vedecká konferencia: Aplikácia experimentálnych a numerických metód v mechanike tekutín, Žilina, Žilinská univerzita, 2002, p. 114-119, ISBN 80-7100-955-5.
- [3] Antoš, P., Sulitka, M., „Úpravy aerodynamického cirklačného tunelu odboru mechaniky tekutín a termodynamiky ČVUT v Praze“, VXIII. Medzinárodná vedecká konferencia: Aplikácia experimentálnych a numerických metód v mechanike tekutín, Žilina, Žilinská univerzita, 2002, p. 126-131, ISBN 80-7100-955-5
- [4] “UIUC Airfoil Coordinates Database - Version 2.0“, Available from WWW: <http://www.ae.uiuc.edu/m-selig/ads/coord_database.html>.
- [5] Schlichting, H., Truckenbrodt, E., Ramm, H.J., „Aerodynamics of the Airplane“, USA, Osborne-McGraw-Hill, 1979, p. 541.
- [6] Althaus, D., Wortmann, F. X., „Stuttgarter Profilkatalog 1“, Braunschweig/Wiesbaden, Friedr. Vieweg & Sohn.

Natálie Součková Ing.

Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Fluid Dynamics and Power Engineering, Technická 4, 166 07 Prague, Czech Republic, +420 224 352 899, +420 224 352 579, **Natalie.Souckova@fs.cvut.cz**

Lukáš Popelka Ing.

Institute of Thermomechanics AS CR, Prague, Dolejškova 1402/5, 182 00 Prague, Czech Republic, +420 266 053 313, **popelka@it.cas.cz**

Milan Matějka Ing.

Institute of Thermomechanics AS CR, Prague, Dolejškova 1402/5, 182 00 Prague, Czech Republic, +420 266 053 441, **Milan.Matejka@it.cas.cz**

IMPORTANCE OF CRANKSHAFT TORSIONAL VIBRATION SIGNALS IN DIAGNOSTICS OF IC ENGINES

V. Doleček, I. Filipović, Dž. Bibić, M. Petrović

Keywords: IC engine, torsional vibration, diagnostics

1. Introduction

Measurement and control of sophisticated systems for use in thermo-energy systems are finding application at IC engines too. Advantages accomplished by their use every day are greater and potential of these systems is nearly not exhausted. Using such system, in the form of free-of-contact sensor for angular velocity and crankshaft position measuring, it is opening possibilities for observing and diagnostics of the IC engine work in relatively simple and inexpensive way. Observing changes of crankshaft angular velocity and its appropriate examination, reliable information about IC engine working state can be obtained. From one point of view it means that is possible to find out irregularity in its operation, while another point offers a definition of causes for such operation.

2. Equivalent dynamical systems of IC engines and equation of motion

Research topic of this work is in-line, six-cylinder, four-stroke, turbocharged, water-cooled diesel engine dedicated for heavy duty road vehicles. Because of complexity of the real system, its transformation in so called line equivalent system [1, 2, 3, 4, 5,] has done.

Using of Lagrange's equations of second type (by initiation of crankshaft angular motion around balance turning angle as variable, its explication in Taylor's row in surroundings of $\omega_i t$ point, and by rejection of high-order elements) equation of vibration of equivalent torsion-vibration system can be presented in form of differential equation system, here given in matrix form:

$$[I] \left\{ \ddot{\vartheta} \right\} + [K_s] \left\{ \dot{\vartheta} \right\} + [K_u] \left\{ \Delta \dot{\vartheta} \right\} + [c] \left\{ \Delta \vartheta \right\} = \{M\}, \quad (1)$$

where I is moment of inertia of the masses, c shaft stiffness, K_u internal damping, K_s external damping, ϑ torsional angle, $\dot{\vartheta}$ and $\ddot{\vartheta}$ appropriate derivations of the torsional angle, $\Delta \dot{\vartheta}$ difference of torsional velocities between neighboring discs, M excitation torque of the system.

3. Simulation of the causes of work irregularity in IC engine

The IC engine, as very sophisticated cyclic thermo-energy machine, transforms bundled chemical energy of fuel into mechanical work. From such given definition results that for its optimal work it is necessary to synchronize thermal as well as mechanical parameters of the whole system. Using free-of-contact incremental sensors for angular velocity and crankshaft position, it is offering a possibility of its operation diagnostics by comparison of such obtained and processed signals with previous simulated signals of its normal, as well as irregularity work. A break of the crankshaft caused by extreme torsional load, as an example of engine irregularity work, was analyzed and curve trends of crankshaft angular motion on pulley assembly position in case of normal and irregularity work are presented in the Figure 1.

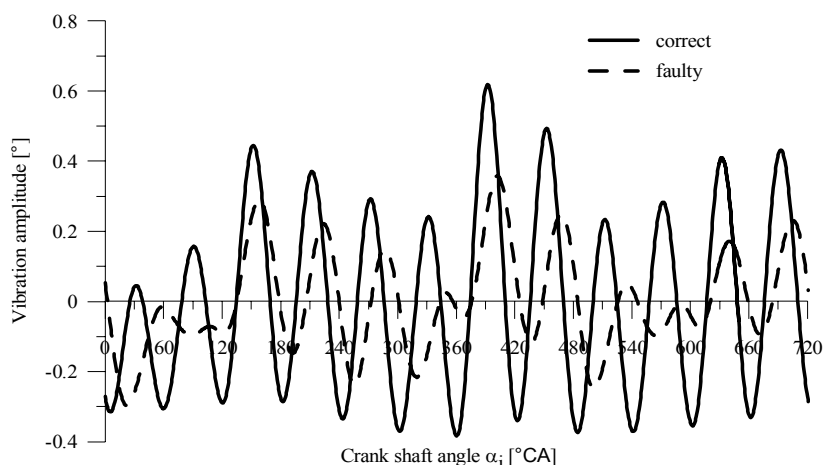


Figure 1. Vibration amplitude of pulley assembly of the considered IC engine with micro brakes on crankshaft at 2100 rpm

Deviations of torsional angle curve, presented in the Figure 1, in the case of crankshaft break in relation with original curve, are visible enough and it can be observed with very simple comparison that it is present some irregularity on the system which generated stiffness decrement, whereby characteristic vibration frequency is decreased, as well as amplitudes. This refers on moving of the resonant regime position of engine work.

Additionally, few examples of the diagnostics possibility of contingent irregularity in IC engine operation are presented in the paper. Specified approach possibilities of engine work diagnostics using angular acceleration and crankshaft position sensors are presented too, as well as limitations related with approach to the analyze.

References

- [1] Hafner E. K., Mass H., „Theorie der Triebwerksschwingungen der Verbrennungskraftmaschine“, Springer-Verlag Wien - New York A, 1984
- [2] Hafner K. E., Mass H., „Torsionsschwingungen in der Verbrennungskraftmaschine“, Springer-Verlag Wien - New York A, 1985
- [3] Doleček V., Filipović I., Bibić Dž., „Istraživanje karakteristika unutrašnjeg trenja viskoelastičnih elemenata u torziona – oscilatornom sistemu“, 04-39-3840-1/04, Mašinski fakultet Univerziteta u Sarajevu, 2005
- [4] Filipović I., Doleček V., Bibić Dž., “ Identification of Parameters for Torsional Vibrations Mathematical Model”, Proceedings TMT 2004. Neum BiH, 2004, pp 367-370.
- [5] Filipović I., Doleček V., Bibić Dž., “Modeling and the Analysis of Parameters in the Torsional - Oscillatory System Equivalent to the Diesel Engines in Heavy-Duty Vehicles”, Journal of Mechanical Engineering. Vol. 51, No. 12/05, iss. 488, 2005, pp 786-797

Vlatko Doleček, Full Prof. Dr.

University of Sarajevo, Department for Mechanics, Vilsonovo šetalište 9, 71000 Sarajevo, Bosnia and Herzegovina, Tel. +387 (0)33 65 65 62, Telefax +387 (0)33 650 841, dolecek@mef.ba

Ivan Filipović, Full Prof. Dr.

University of Sarajevo, Department of IC Engines and Vehicles, Vilsonovo šetalište 9, 71000 Sarajevo, Bosnia and Herzegovina, Tel. +387 (0)33 653 208, Telefax +387 (0)33 650 841, filipovic@mef.ba

Dževad Bibić, Teaching Assistant, M. Sc.

University of Sarajevo, Department of IC Engines and Vehicles, Vilsonovo šetalište 9, 71000 Sarajevo, Bosnia and Herzegovina, Tel. +387 (0)33 650 841, Telefax +387 (0)33 650 841, bibic@mef.ba

Marin Petrović, Assistant, B. Sc.

University of Sarajevo, Department for Mechanics, Vilsonovo šetalište 9, 71000 Sarajevo, Bosnia and Herzegovina, Tel. +387 (0)33 656 562, Telefax +387 (0)33 650 841, petrovic@mef.ba

THE RELIABILITY OF THE STEEL STRUCTURE UNDER WIND ACTION

Marta Sulyok-Selimbegović, Diana Šimić, Nataša Mrazović

Keywords: Reliability indices, specifications, FOSM theory, wind load, distribution density functions

1. Introduction

The structure of the steel chimney, which is examined in the examples of calibration, is subjected to the wind loads on two different locations (Zagreb and Slavonski Brod) and designed by our current specifications. The statistical parameters of the results are used concerning the random action of wind loading and dynamical response of the structures in the evaluation models. It is evident that the differences in the applied loads on the structures calculated by the same design requirements and global factors, give the different reliability index, which provide the regionalization of Croatia for the various intensity of wind load. The results of calibrations are compared and show that they are on the conservative side for the certain wind load influences.

2. Theoretical model for the design under wind loading

They are differences and similarities for various code-rules which are based on the essentially the same theoretical background and the purpose of this section is to use one of these methods. The use of the analytically exact solutions for the design of cross sections under wind loading and end conditions for steel chimney are empirically modified and accommodate to idealized conservative simplifications. The wind load is high-variable action of stochastic behaviour and the statistical parameters for load have to be specified in the connexion with the specific response of the structures. The response of the structure is specified concerning the evaluation model under the characteristics of wind load action during certain intervals. The principal differences of structures are in their strength towards various dynamic load actions so we can specify two main groups of the structures with high and low frequency.

The statistical parameters concerning the action of the average velocity of dominant wind determines wind load. This velocity has to be connected with average hourly wind velocity measured ten meters above ground. The basic data are the maximum of annually average wind velocity during an hour. The maximum annual wind velocities are measured during the mean recurrence intervals of 25, 50 and 100 years. The annual maximum wind speed distribution is assumed to be constant in time so the data are identically distributed with the cumulative distribution function based on the Type II Extreme Value probabilistic distribution. This probability distribution is defined by the parameters of mean and standard deviation or coefficient of variation. Wind speeds corresponding to the specified mean recurrence intervals may be converted to effective velocity pressures that vary with height and exposure. This effective velocity pressure has to take into account the dynamic response to gusts of structures in a direction parallel to the wind. They do not provide the effects of vortex shedding or instability due to galloping or flutter.

3. The representative model of the structure

For practical mean of describing the response of a structure under dominant wind load action, the freestanding steel chimney is used. The example of the chimney-tower is circular steel tube of constant section ($\varnothing$ 1000 mm) rigidly connected to the concrete footings below ground surface.

The high of the chimney is 30 meters and surrounding area is relatively free of obstructions. The chimney tower is located in continental part of RH on 2 locations for which the measured data for wind velocity is collected and which are Zagreb and Slavonski Brod.

4. The evaluation of the reliability indices for steel chimney under wind loads

In the probabilistic approach for the calibration of the structures the limit state design is used in which load action and resistance are the basic random variables mutually independent. In this case the basic random variables are the bending moment of dominant wind action (M_S) and plastic moment of the cross section (M_R) at the foot of the chimney evaluated by limit state design.

These two random variables have different distributions of probability density functions. The statistical parameters of the gradient wind velocity (v_G), are evaluated from the meteorological data from the measurement done in Croatia, from which the statistical parameters of the wind velocity (V_{30}) at the high $Z=30\text{m}$ above ground are evaluated. The basic wind pressure (q_{w0}) has the functional relationship with wind velocity at high 30m (v_{30}). The influence of the wind pressure and their statistical parameters through the high of the steel chimney defined by the shape and dynamic coefficients and the basic wind pressure are taken in the evaluation. From these parameters of the wind action on the chimney, the maximum bending moment is calculated and with its statistical parameters.

From the statistical parameters of the test results for the ultimate strength for model of the steel chimney, the evaluation of the reliability indices is performed and tabulated (Table 7 in full paper) for stated locations under wind loads.

7. Conclusion

The analysis of the steel structures with representative example of the steel chimney for our design models is obtained, by which the ultimate limit state of the bending strength is evaluated for the purpose of the calibration of these structures under the dominant wind loads from the measured data in Croatia on two locations. The results of the calibration varies with applied wind loads for the examined model, designed by our code criteria, in the range from realized reliability indices $\beta=6,65$ to $4,95$, for the dominant wind load in the continental parts of Croatia, and are quite on the target safety side. It is evident that there is necessity to take into consideration partial safety factors for the dominant wind loads concerning the measured data in Croatia, and the evaluation of the dynamic models for structures under dominant wind loads with resistance factors of the target reliability level in order to achieve uniform reliability with the proposed loads factors, concerning the applied loads in this case.

References

- [1] Eurocode 3 Design of Steel Structures-Part 1-1: General rules and rules for buildings, 1987
 - [2] ENV 1991-2-4, Basic of Design and Action on Structure-Wind Loads ,pp 153., 1995
 - [3] Melchers, R. E. Structural Reliability - Analysis and Prediction, John Wiley&Sons, 1987
 - [4] Sachs, P, *Wind Forces in Engineering*, Pergamon Press, Oxford, UK, 1978
 - [5] Sulyok, M. and Galambos T.V. ,The Evaluation of the Resistance Factors for the Laterally Unsupported Rolled and Welded Beams and Plate Girders Designed by LRFD of the AISC, pp 1-10, Steel Structures: Advances Design and Construction, Elsevier Applied Science, London & New York, UK, 1987
- Prof. Dr. sc.Marta Sulyok-Selimbegović, Ph.D. in Civil Engineering
University of Zagreb, Faculty of Architecture ,10000 Zagreb, Kačićeva 26, Croatia, +385/1/4639-348
+385/1/4828-079, marta.sulyok-selimbegovic@arhitekt.hr
- Ass. Prof. Diana Šimić, Ph.D. in Civil Engineering
University of Zagreb, Faculty of Civil Engineering, 10000 Zagreb, Kačićeva 26, Croatia, +385/1/4639-426
+385/1/4828-050, dianas@grad.hr
- Nataša Mrazović, research novice
University of Zagreb, Faculty of Architecture ,10000 Zagreb, Kačićeva 26, Croatia, +385/1/4639-312
+385/1/4828-079, natasa.mrazovic@arhitekt.hr

AN IDENTIFICATION OF BIFURCATION TYPE USING POSTCRITICAL MOTION ANALYSIS

Safet Isić, Vlatko Doleček, Isak Karabegović

Keywords: *nonlinear analysis, numerical analysis, stability, bifurcation, postcritical motions, phase orbits*

1. Introduction

It is well known that some elastic systems under a particular value of applied load can have, in addition to the primary equilibrium position, other equilibrium positions as well, thereby exhibiting the bifurcation of the equilibrium path [2], [3], [4]. In this paper the analysis of bifurcation of the equilibrium position using the postcritical motion of the observed system is presented. The analysis is performed on both conservative and non-conservative systems with two degrees of freedom, loaded with axial and follower force. Nonlinear equations of motion, which take large displacements into account, are presented. The solution is obtained numerically for different initial conditions. Numerical results show that the simplification of a nonlinear analysis in some cases can be based on the linear analysis of the eigenvalues of buckling. These results are used for the derivation of the equation that describes phase trajectories, the shape of which is dependant on the bifurcation type and can thus be used for the identification of the bifurcation.

2. The analysed models

Analysis is performed on the pendulum like conservative and nonconservative systems with two d.o.f., presented on Fig.1. These systems have similar shape and geometry, but exploits different behaviour when applied load is greater then critical. Systems on Fig. 1a and Fig.1c are supported by angular springs, while system on Fig. 1b is supported by springs, which remains horizontal during deformation of the pendulum.

3. Nonlinear equations of motion

On the basis of kinetic and potential energy, generalized forces and dissipation for the systems on Fig.1, using Lagrange's equation, we obtain equations of motion in the matrix form

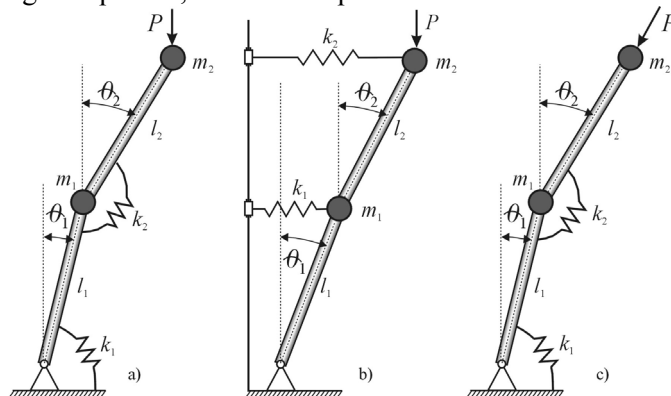


Figure 1. Analysed two d.o.f. systems.

$$[\mathbf{M}]\{\ddot{\boldsymbol{\theta}}\} + [\mathbf{K}]\{\boldsymbol{\theta}\} + [\mathbf{M}_1]\{\ddot{\boldsymbol{\theta}}\} + [\mathbf{K}_1]\{\boldsymbol{\theta}\} + [\mathbf{C}_1]\{\dot{\boldsymbol{\theta}}\} = 0. \quad (1)$$

where mass matrix $[\mathbf{M}]$ and stiffness matrix $[\mathbf{K}]$ contain linear part of equation, and $[\mathbf{M}_1]$, $[\mathbf{K}_1]$ and $[\mathbf{C}_1]$ are matrices of nonlinear effects. Matrices $[\mathbf{M}_1]$ and $[\mathbf{C}_1]$ are equal for all of three analysed system, while matrix $[\mathbf{K}]$ and matrix of nonlinear part of geometric stiffness $[\mathbf{K}_1]$ are dependent on the considered system, and cause their difference in dynamical behaviour.

4. Numerical and approximate analytical solutions

Equations of motion are transformed in standard form of first order equations and solved numerically for specified initial conditions and value of applied load. As method for solution forth order Runge-Kutta method is used. Some specific phase orbits for system on Fig.1a are presented on the Fig.2a, where initial position has shape of the buckling eigenmode of the linear part of equation (1).

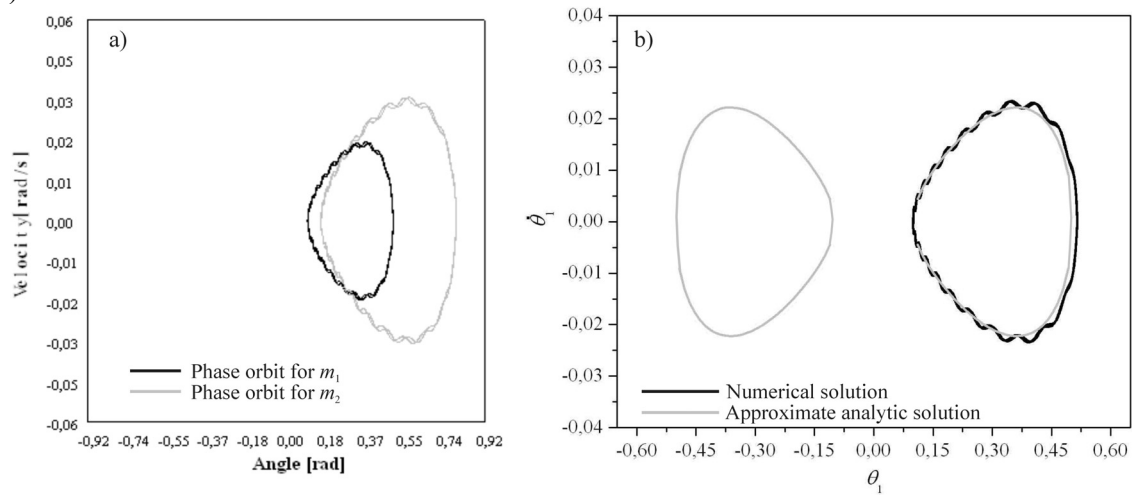


Figure 2. Phase orbits for $P = 1.05P_{cr}$ and initial position $\{\theta_1, \theta_2\} = \{0.1, 0.1618\}$.

According to the numeric solution for applied force greater then critical and initial position according to buckling mode, it is derived single equation of the phase orbit, shown on the Fig.2b, which agrees well with numeric results and identify stable symmetric bifurcation in this case. Such equations may be used for analysis of bifurcation type of the system instead numeric solution of equations for many different initial conditions.

References

- [1] J.J. Thomsen, "Chaotic Dynamics of Partially Follower-Loaded Elastic Double Pendulum", Journal of Sound and Vibration, Volume 188, Issue 3, Pages: 385-405, October 1994.
- [2] H. Ziegler, "Principles of Structural Stability", Blasidell Publishing Company, Watham, Massachusetts, Toronto, London, 1968.
- [3] J.M.T. Thompson, G.W. Hunt, "A General Theory of Elastic Stability", John Wiley & Sons, 1973.
- [4] J.M.T. Thomson, "Instabilities and Catastrophes in Science and Engineering", John Wiley & Sons Ltd., 1982.
- [5] J.J. Thomsen, "Vibration and Stability – Order&Chaos, DTU, Lyngby, 1994.

Safet Isić

Faculty of Mechanical Engineering, University Campus, Mostar, Bosnia and Herzegovina, e-mail: safet.isic@unmo.ba

Vlatko Doleček, Prof.Dr.

Faculty of Mechanical Engineering, Vilsonovo šetaliste 9, Sarajevo, Bosnia and Herzegovina, e-mail: vldolecek@yahoo.com

Isak Karabegović, Prof.Dr.

Technical Faculty Bihać, Ul. Irfana Ljubijankića, Bihać, Bosnia and Herzegovina, e-mail: isak1910@hotmail.com

FORCES ON A DISLOCATION NEAR A CRACK IN A PIEZOCERAMIC MATERIAL

Z. Kulenović

Keywords: force, dislocation, crack, piezoceramic material

Piezoelectric materials produce an electric field when stressed and deform when subjected to an electric field. Such an intrinsic coupling of course has attracted wide industrial applications of piezoelectric materials. The defects, such as dislocations and cracks, may be produced in piezoelectric materials during their manufacturing process. When the materials are subjected to electromechanical loads, stress concentrations due to these defects can give rise to critical crack growth and subsequent mechanical failure. Therefore, it is very important that the behaviors of these defects are investigated.

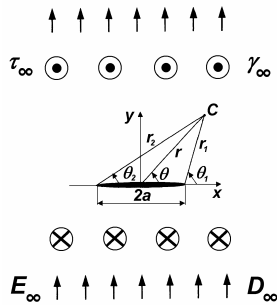


Figure 1. Geometry of the problem

In this work is analyzed a simple continuum model of a single screw dislocation near a crack in a piezoelectric crystal subjected to mechanical and electrical loads. The dislocation has a line force and a line charge along its core. The piezoelectric material is transversely isotropic and containing a charged screw dislocation around a finite crack of length $2a$, Figure 1. The material has a hexagonal symmetry with an isotropic basal plane of xy -plane and a poling direction of z -axes. The crack is situated along the plane $y = 0$. The material is subjected to far field antiplane mechanical and in plane electric loads. In this configuration, the piezoelectric boundary value problem is simplified considerably because only the out-of-plane displacement and in-plane electric fields exist.

The constitutive relations for the piezoelectric material are:

$$\tau_{zi} = c_{44}\gamma_{zi} - e_{15}E_i, \quad D_i = e_{15}\gamma_{zi} + \epsilon_{11}E_i \quad (1)$$

where $\tau_{zi}(x, y)$, $\gamma_{zi}(x, y)$, $E_i(x, y)$, and $D_i(x, y)$ ($i = x, y$) are the components of the shear stress, shear strain, electric field, and electric displacement vectors, respectively. Also, c_{44} , e_{15} and ϵ_{11} are the elastic modulus (constant electric field), piezoelectric constant, and dielectric permittivity (constant strain) of piezoelectric material, respectively.

Taking in consideration the boundary conditions, the governing equations can be expressed as:

$$\nabla^2 w = 0, \quad \nabla^2 \varphi = 0 \quad (2)$$

where $w(x, y)$ is mechanical displacement, and $\varphi(x, y)$ is the electric potential.

The solution is found by application of the complex analytic functions and conformal mapping method, and the forces acting on a dislocation are determined:

$$F_x = b_z \tau_{zy}^T + \Delta \varphi D_y^T + p_z^S \gamma_{zx}^T + q_z^S E_x^T, \quad F_y = -b_z \tau_{zx}^T - \Delta \varphi D_x^T + p_z^S \gamma_{zy}^T + q_z^S E_y^T \quad (3)$$

where b_z , $\Delta\varphi$, p_z , and q_z are the Burgers vector, electric potential jump, line force, and line charge, respectively.

The numerical analysis for piezoelectric PZT-5H ceramic material is obtained in MatLab and presented in 2D and 3D graphs. Basic values of material parameters are: $a = 1 \cdot 10^{-2}$ m, $b_z = 1 \cdot 10^{-9}$ m, $\Delta\varphi = 1$ V, $p_z = 10$ N/m, $q_z = 1 \cdot 10^{-8}$ C/m, $c_{44} = 2,3 \cdot 10^{10}$ N/m², $e_{15} = 17$ C/m², $\epsilon_{11} = 150 \cdot 10^{-10}$ C/Vm, $\gamma_\infty = 9,5 \cdot 10^{-5}$, $E_\infty = 2 \cdot 10^5$ (V/m).

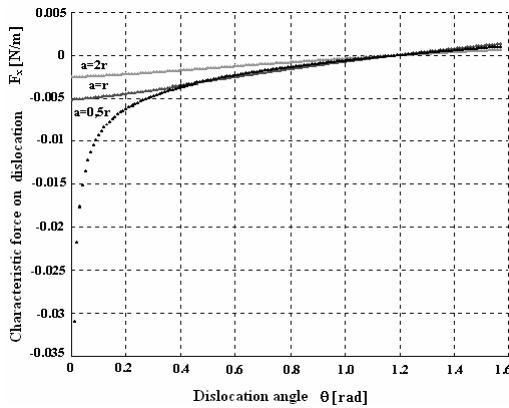


Figure 2. Characteristic force in x-direction vs. dislocation angle

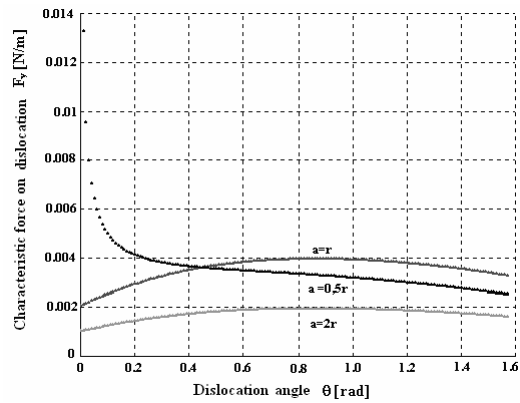


Figure 3. Characteristic force in y-direction vs. dislocation angle

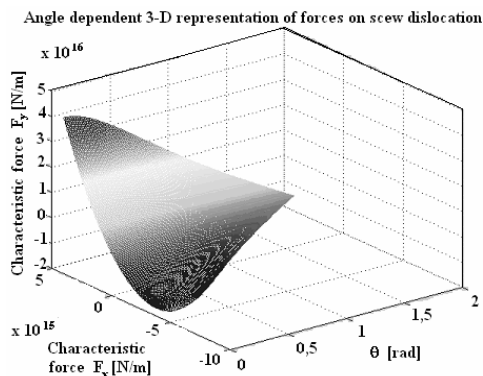


Figure 4. Characteristic forces on screw dislocation vs. dislocation angle

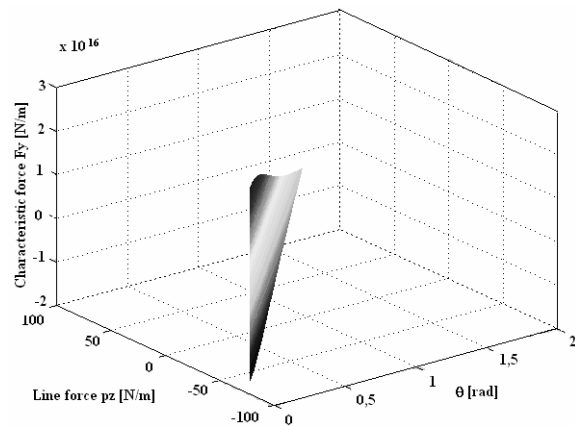


Figure 5. Characteristic y-axis force and line force vs. dislocation angle

Characteristic force F_x in x-direction versus dislocation angle θ for ratios: $a = 0,5r$, $a = r$, and $a = 2r$, is presented in Figure 2. Figure 3 shows the characteristic force F_y in y-direction versus dislocation angle θ for ratios: $a = 0,5r$, $a = r$, and $a = 2r$. Characteristic forces F_x and F_y on screw dislocation in dependence on dislocation angle θ , are shown in Figure 4. Figure 5 shows characteristic y-axis force F_y and line force p_z versus dislocation angle θ . The results of this investigation can be useful for producers and users of PZT materials.

Zlatan Kulenović, Dr. Sc. Professor

University of Split, Faculty of Maritime Studies, Zrinsko-Frankopanska 38, HR - 21000 Split, Croatia,
tel.: +385 (0)21 380 762, fax.: +385 (0)21 380 759, e-mail: zlatan@pfst.hr

DETERMINATION OF THE FRICTION PARAMETERS USING MATLAB INTERPOLATION SURFACES

Frane Vlak, Vedrana Cvitanić and Igor Duplančić

Keywords: friction, ring compression test, FEM, calibration surface

1. Introduction

The friction is inevitable background of the metal forming processes and primarily it increases energy consumption. It can be reduced by lubrication and thus, quality of the products can be improved, too. Therefore, in designing a metal forming operation, it is necessary to quantify friction. Many friction laws [1] are used for that purpose. Parameters, usually called coefficients or factors, used in that laws can be determined using various tests. Amongst them, ring compression test [2] has proven to be one of the best. In this work, it is shown how the *MATLAB* [3] predefined interpolation functions can be used to determine required friction parameters.

2. Determination of the friction parameters using MATLAB

During the ring compression test, the height h and the inner diameter d_i are measured and engineering strains ε_{d_i} and ε_h are calculated. This strains are then used to determine friction parameters using diagrams with calibration curves. In this work, rigid-plastic FEM [4],[5] implemented in *CAMPform* software [6] is used to obtain calibration curves using the constant friction model [5]. For that purpose, an axisymmetric finite element model is analysed under various friction factor values.

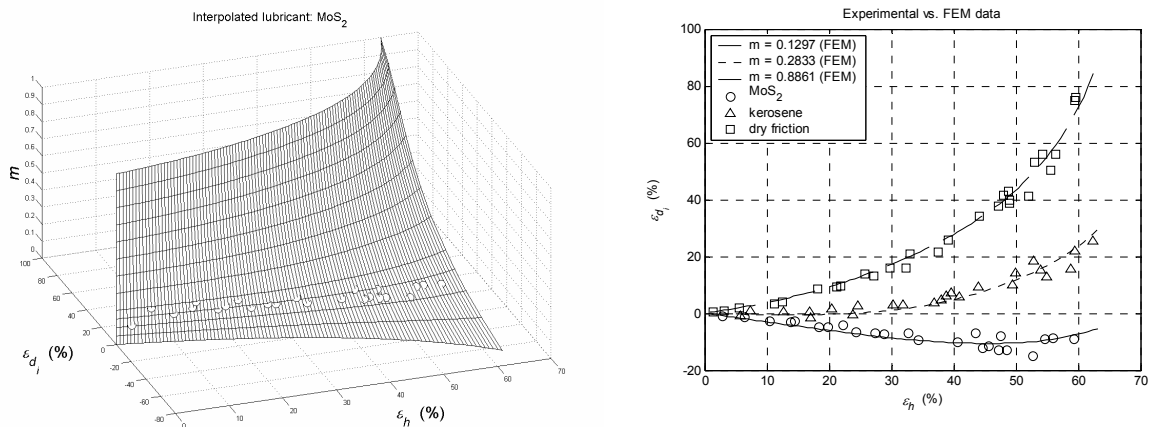


Figure 1. Calibration surface with interpolated MoS₂ lubricant data (left) and calibration curves for calculated friction factors (right)

The main idea of the proposed procedure for the friction parameter determination is 3D representation of the calibration curves, Figure 1 (left). Namely, once the calibration curves are determined, they can be used to generate 3D surface in $(\varepsilon_{d_i}, \varepsilon_h, m)$ coordinate system using *MATLAB*

predefined functions. Then, friction factors can be determined using algorithm that is analogous to the least square fitting method. This algorithm is implemented on the example given in [7]. Ring geometry used is $\varnothing 30 : \varnothing 15 : 10$ (mm) and material employed is E1CM commercially pure aluminium (series 1000). The experiment is performed using three different lubricants: MoS₂ paste, commercial kerosene and with no lubricant (dry friction). Proposed procedure is then implemented in order to obtain friction factor values for these lubricants. Experimental data for the MoS₂ paste, interpolated into calibration surface are shown in the Figure 1 (left). Determined friction factor values, represented as a calibration curves, are shown in the Figure 1 (right).

By numerical integration, it is proved that calibration curves interpolated from the calibration surface correspond almost exactly with calibration curves obtained by finite element solutions that are calculated by using determined friction factor values. Also, slightly discrepancies between these calculated friction values and those obtained in [7] can be observed. They are not consequence of wrong proposed procedure but the difference of the calibration curves obtained using different finite element software and different stress-strain relationships.

6. Conclusion

In this work, it is shown how the MATLAB capabilities can be used for determination of the friction parameters in the metal forming processes. The ring compression test is described and proposed procedure for the friction parameter determination is implemented for the aluminium specimens. For that purpose, calibration curves obtained by rigid-plastic finite element analysis are transformed into calibration surface. Using MATLAB interpolation functions, friction factors for three different lubricant conditions and for the constant friction model are obtained. These factors, which are supposed to be constant over whole range of deformation process, can be considered as sufficiently good for the use in the more complex analysis. Finally, comparison of the interpolated solution and finite element solution is performed and excellent agreement is found. Same procedure can be implemented for other friction models and these informations would give more detailed insight into influence of the lubricants on the metal forming processes.

References

- [1] Tan X., "Comparisons of friction models in bulk metal forming", Tribology International, Vol. 35, Issue 6, 2002, pp 385-393.
- [2] Male A.T., Cockroft M.G., "A Method for the Determination of the Coefficient of Friction of Metals Under Conditions of Bulk Plastic Deformation", Journal of Institute of Metals, Vol. 93, 1964, pp 489-494.
- [3] MATLAB v6.5 R13, *The language of Technical Computing*, The MathWorks Inc., Copyright 1984-2003
- [4] Zienkiewicz O.C., "Flow Formulation for Numerical Solution of Forming Processes", in Numerical Analysis of Forming Processes, Chapter 1, John Wiley and Sons, New York, 1984
- [5] Kobayashi S., Oh S.I., Altan T., "Metal Forming and the Finite Element Method", Oxford University Press, Oxford, 1989
- [6] CAMPform v1.5, "<http://camp.kaist.ac.kr/>", KAIST Institute, Korea, 2002.
- [7] Petersen S. B., Martins P. A. F., Bay N., "An alternative ring-test geometry for the evaluation of friction under low normal pressure", Journal of Materials Processing Technology, Vol. 79, Issues 1-3, 1998, pp 14-24.

Frane Vlak, dr.sc.

FESB/University of Split, Department of mechanical engineering and naval architecture, Ruđera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 864, Fax: 385(0)21/ 463 877, e-mail: fvlak@fesb.hr

Vedrana Cvitanić, dr.sc.

FESB/University of Split, Department of mechanical engineering and naval architecture, Ruđera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 865, Fax: 385(0)21/ 463 877, e-mail: vcvit@fesb.hr

Igor Duplančić, dr.sc.

FESB/University of Split, Department of technology, Ruđera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 853, Fax: 385(0)21/ 463 877, e-mail: iduplan@fesb.hr

THEORETICAL AND EXPERIMENTAL MODELING OF A PIEZOELECTRIC COMMON RAIL INJECTOR

Valdas Čaika, Mario Jelović and Dino Mandić

Keywords: fuel injection, common rail, piezoelectricity, numerical simulation

1. Introduction

The paper is devoted to the dynamic modeling and experimental analysis of the piezoelectric injector of a modern diesel common rail system. A detailed one-dimensional dynamic model of the injector is created using AVL software Hydsim [3]. It consists of multiple hydraulic and mechanical components for the modeling of the pressure wave propagation and motion of the solid parts. The injector model includes the electro-mechanical description of the piezoelectric stack actuator with the hysteresis behavior. Different operating conditions of the injector are simulated and tested on AVL common rail test rig. Theoretical and experimental results are compared.

2. Injector Model

The injector in question is a piezo-controlled common rail diesel injector from Bosch (CRI3.0). Its main components are the piezoelectric actuator, hydraulic amplifier, control valve and nozzle. The injection process is regulated via the energizing duration of the piezo actuator which consists of a dense pile of separately contacted layers (wafers) of electrically active ceramic material.

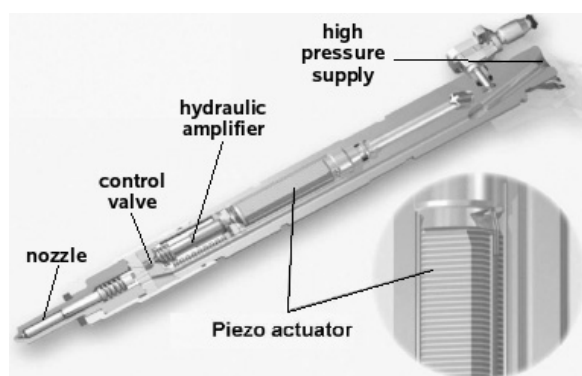


Figure 1. Schematic of Bosch CRI3 Piezo Injector

Main new features of the CRI3 injector are [1]: sharp injector rate, small PI-quantity (0.5 mm^3), no PI-plateau, reduced tolerances for EU IV emission standard, multiple injections (up to 5 per cycle), reduced static and dynamic forces (wear), no leakage, fail-safe-principle, low noise, compact design and upgrade of rail pressure to 180 MPa (CRI3.1) and 200 MPa (CRI3.2).

CRI3 piezo injector operates much faster than the traditional solenoid-controlled injector [2]. It can generate multiple pulses during one engine cycle. In combination with modern combustion processes, enhance engine performance, improve emission quality and reduce noise.

AVL HYDSIM model of the injector is shown in Figure 2. It consists of the fuel rail with high-pressure connection, injection nozzle, Bosch tube, throttle plate with the inlet, outlet and by-pass orifices, control valve, hydraulic amplifier (coupler) and piezoelectric actuator.

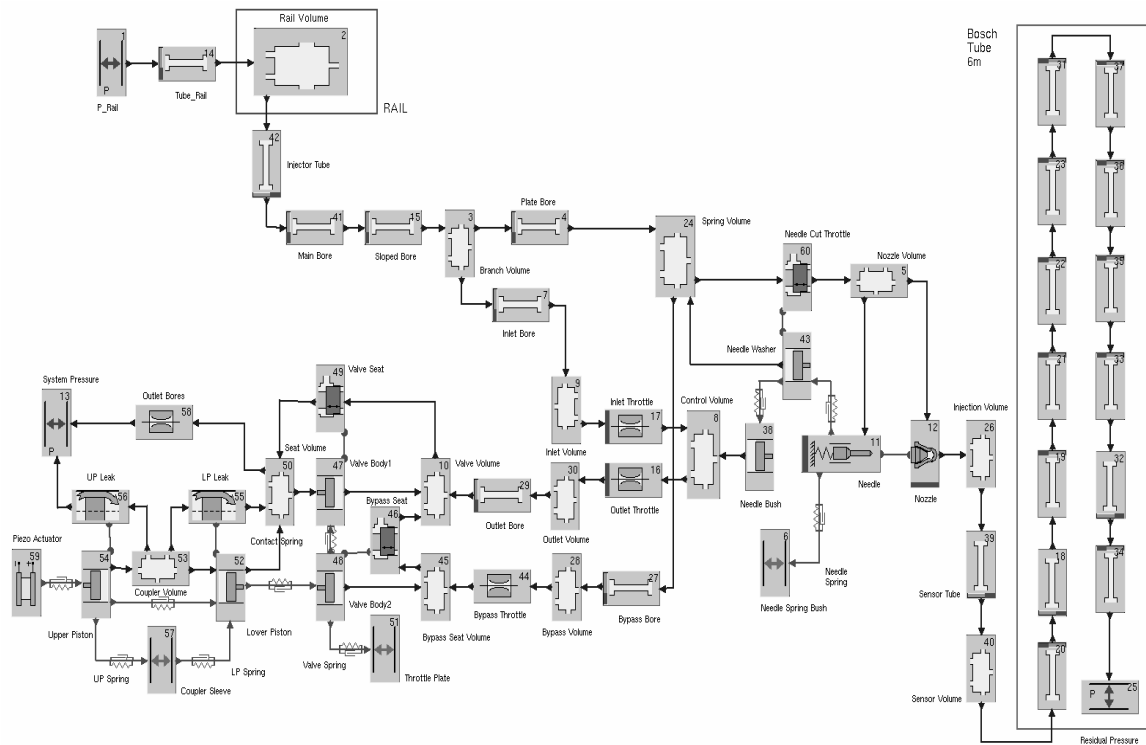


Figure 2. AVL HYDSIM Model of CRI3 Injector with Bosch Tube

3. Experiment

Experimental analysis is carried out on the AVL common rail test rig. It consists of the following components: high pressure pump, rail, Bosch tube, scale, injection quantity measuring device, individual adapters for injector clamping, engine speed simulator, control unit and indication system (rail pressure, current signal, injection rate, needle lift signal, line pressure). The injection rate measurements are carried out for 4 full load and 2 part load cases. Piezo actuator energizing time is used as initial condition. The injection pressure measurements are performed in the spiral Bosch tube. The main requirement for the Bosch tube is that its length is high enough to prevent the reflected pressure wave interaction with the forward-propagating pressure wave. Additionally, the fuel delivery (quantity) is measured for each individual injection event.

References:

- [1] Boecking F., Dohle U., Hammer J. and Kampmann S., "Passenger Car Common Rail Systems for Future Emissions Standards", MTZ 7-8/2005, Volume 66, pp 552-557.
- [2] Hummel K. et al, "Third-Generation Common Rail System with Piezo Inline Injectors from Bosch for Passenger Cars", MTZ 3/2004, Volume 65, pp 181-189.
- [3] AVL HydSim, Reference Manual, Graz 2005.

Dr. Valdas Čaika

AVL List GmbH, Advanced Simulation Technologies, Hans-List-Platz 1, A-8020 Graz, Austria,
Tel: +43 316 787 1748, Fax: +43 316 787 1923, E-mail: valdas.caika@avl.com

Mr. Sc. Mario Jelović

AVL-AST d.o.o., Av. Dubrovnik 10, 10020 Zagreb, Croatia, Tel: +385 1 6598602, Fax: +385 1 6598666,
E-mail: mario.jelovic@avl.com

Dipl. Ing. Dino Mandić

AVL-AST d.o.o., Av. Dubrovnik 10, 10020 Zagreb, Croatia, Tel: +385 1 6598603, Fax: +385 1 6598666,
E-mail: dino.mandic@avl.com

FATIGUE RESISTANCE OF MAGNESIUM ALLOY AZ 91D AT THE HIGH-FREQUENCY CYCLIC LOADING

V. Bella – M. Kuffová

Keywords: static mechanical characteristics, high frequency resistance, loading cycle; asymmetry, fatigue crack, fatigue life

1. Introduction

The presented paper brings the result of Mg-alloy AZ91D metallographic and fractographic fracture surfaces analysis, fatigue crack propagation and fatigue life measured under high frequency cyclic loading frequency $f \approx 20$ kHz, alternating symmetric pull-push, loading cycles asymmetry $R = -1$, temperature $T = 20 \pm 10^\circ\text{C}$, in the field of high number of cyclic from $N = 6.10^5$ cycles to $N = 1.10^9$ cycles.

Nowadays, magnesium alloys are very interesting material for practice. Low specific weight and ability to damp vibrations preordains them for applications in automobile, airplane and rocket industry, in telecommunication and instrument technology [1, 2, 3].

The fatigue resistance in the region of high number of cycles of the magnesium alloy AZ91D with known chemical composition, mechanical properties and microstructure after thermal processing T4 (melting annealing – temperature 413°C , holding stage 18 hours and cooling to water) has been carried out on the experimental equipment determining of the complete Wöhler's curves, dependence $\sigma_a = f(N_f)$ including endurance fatigue limit σ_c and timing fatigue limit σ_{CN} is realised on a test equipment CAFR (complete acoustic fatigue resistance) HFCL (high frequency cyclic loading).

Microstructure of magnesium alloy AZ 91D specimens after thermal processing is created by uniform polyedric grains of phase δ .

2. Results and Discussion

On the tested magnesium alloy AZ 91D, the dependence $\sigma_a = f(N)$ in the region from $N \approx 6.10^5$ to $N \approx 1.10^9$ of cycles to fracture has been studied. Experimentally measured values of the fatigue resistance show relatively high dispersion. This dispersion of the measured values has been caused, first of all, by the presence of cast defects (micropipes, structural heterogeneities). The occurrence of these defects, their character, size, orientation, amount and location on the surface and in subsurface layers influenced in a negative way and to a various degree of strength the fatigue resistance of the studied alloy. Experimental results show an obvious difference in the values measured on test bars taken from the plate centre and the plate edge. Different size of grains caused by different speed of hardening and cooling at the plate centres and at the plate edges in the course of casting, plays here a considerable role.

The fractographic analysis of fracture surfaces of test bars disturbed by fatigue has shown that the fatigue cracks, almost in all cases, have been initiated from the surface of the test bars. The cast defects were mostly microscopic small, nevertheless their occurrence was very frequent and sometimes they created rather a wide network. The disturbance character of test bars taken from the plate centre and plate edge has not disclosed any substantial differences from the fractographic point of view.

3. Conclusion

Experimental assessment of the fatigue resistance of the studied magnesium alloy AZ 91D in the region of high number of loading cycles ($N \approx 6 \cdot 10^5 \div 1 \cdot 10^9$ cycles), carried out under high-frequency cyclic loading ($f \approx 20$ kHz, $T = 20 \pm 10^\circ\text{C}$, $R = -1$) has shown that the fatigue behaviour and character of the fatigue fracture of the alloy AZ 91D are highly dependent on structural factors and the level of technical perfection of their production, the character of the fracture surface morphology depended on the size of amplitude of strain σ_a to a considerable degree,

due to strong structural heterogeneity and occurrence of large amount of cast defects (first of all, usually appearing microscopic cavities) it was not possible to show factually the influence of the amplitude size of the applied strain on the initiation and micro-mechanism of fatigue failure development.

The most substantial improving of the fatigue properties of the produced magnesium alloy AZ 91D can be obtained by means of improving the level of technological perfection of its production (above all by reducing the amount and size of micro-cavities and making the grain size more uniform).

References

- [1] Kainer K.U. a kol.: “Communications on the Edge of the Milleniums”, EDIS, Žilina, 1998.
- [2] Skočovský P., Bokůvka O., Konečná R., Tillová E.: “Náuka o materiáli pre odbory strojnícke”, EDIS, Žilina, 2001.
- [3] Kuffová, M.: “Únavová odolnosť vybraných horčikových zliatin”, Projekt dizertačnej práce, Žilina 2001.

Vladimír Bella, Prof. Ing., Ph.D.,

Academy of the Armed Forces Gen. M. R. Stefanik, Department of Mechanical Engineering, Liptovský Mikuláš, Slovakia, +421 960 423369, bella@aoslm.sk

Mariana Kuffová, Ing., Ph.D.,

Academy of the Armed Forces Gen. M. R. Stefanik, Department of Mechanical Engineering, Liptovský Mikuláš, Slovakia, +421 960 423252, kuffova@aoslm.sk

METHOD OF CONTINUOUS DIPOLES FOR MODELING COMPOSITE MATERIALS WITH MICROSTRUCTURE

Vladimír Kompiš, Marián Kompiš, Michal Kaukič

Keywords: Composite materials; fibre reinforcement; continuous dipoles; quasi-singular integral equations.

1. Introduction

In view of exceptional mechanical properties of carbon nanotubes (CNT), they have been considered as ideal reinforcements in composite structures. For nanotube composite materials, it has been shown that a carbon nanotube weight fraction of 1% results in the same increase in composite elastic modulus as a composite with a 10% weight fraction of carbon fibers. Nanoparticles with high aspect ratio have proven to be good reinforcing agents in polymeric materials. The tremendous mechanical properties of carbon nanotubes and other nano-reinforcements can be realized only if efficient load transfer exists between the matrix and reinforcement. When nanoparticles are added to a traditional composite, such as carbon nanotubes being added to a carbon-carbon fibre composite, the surface area of the matrix increases greatly, allowing for better mechanical bond and decreased crack propagation. Materials can be made anisotropic, giving not only differing strength properties throughout the material, but directional thermal and electrical properties. The scientists discovered that if all the nanotubes were lined up parallel to each other and a heat flux was applied parallel to the direction of the nanotubes, the material is an excellent thermal conductor. However, for the same material, if the heat is applied perpendicular to the direction of the nanotubes, then the material reflects the heat, and is an insulator. A material could even be made with properties that would vary with a changing electrical field, magnetic field, heat flux, or mechanical stress. These materials could change their colour, opacity, or conductive properties.

This paper deals with Continuum Mechanics (CM) models for simulation of behaviour of such composite materials.

2. Continuum Mechanics models

The continuum based methods primary include techniques such as the Finite Element Method (FEM), the Boundary Element Method (BEM), and the micromechanics approach developed for composite materials.

Simulation of composites reinforced with particles with large aspect ratio using known classical approaches such as FEM, BEM, etc. is not effective. The analytical models require simplification, which cannot simulate all effects necessary for accurate investigation of all problems. Special models based on potential theory [1] developed by our team increase considerably the efficiency of the models and can be a basis for development of computational models for interaction of electro-magnetic, thermal and mechanical fields for such materials. Interactions of all nano-fibres with matrix material, interaction of nano-fibres with other nano-fibres and with other inclusions as well as the interaction of the fibre with structure boundaries and the interaction of nano-particles with nano-fibres are important for evaluation of the local and global effects.

The interaction of a fibre with the matrix is simulated by continuous dipoles located along the axis of the fibre. The intensity of the dipoles is computed so that the boundary conditions for a rigid body displacement of the fibre along its boundary will be satisfied. The problem leads to the solu-

tion of quasi-singular integral equations. Numerical evaluation of the integrals is inefficient because of large gradients in the integrands; instead formulations with analytic integration were developed. The numerical behaviour of the models and results are discussed in the paper.

Acknowledgement

The authors appreciate the partial support for this research by the Slovak Grant Agency for Research and Development (the Grant No. APVT-20-035404) and by NATO RTO Grant agency (Grant No. 001-AVT-SVK, V. Kompiš).

References

- [1] Kompiš V., Štiavnický M., Kompiš M., Kireitseu M., “Continuum models for micro- and nano-fiber/tube-composites”, Proc. of the 5th Internat. Conf. on Comput. of Shells and Spatial Structures, June 1-4, 2005, Salzburg, Eds. E. Ramm, W. A. Wall, K.-U. Bletzinger, M. Bischoff, CD ROM and www.iassiacm2005.de.

Vladimír Kompiš, Prof. Dr.-Ing. PhD.

Academy of Armed Forces of General M. R. Štefánik, Department of Mechanical Engineerig, Demänovská 393, 03119 Lipt. Mikuláš, Slovakia, Tel. +421 960423021, Fax +421 960423071, **e-mail: kompis@aoslm.sk**

Marián Kompiš, PhD.

Building Test and Research Institute, A. Rudnaya 90, 01001 Žilina, Slovakia, Tel.+421 415683494 **e-mail: kompis@za.tsus.sk**

Michal Kaukič, PhD.

University of Žilina, Faculty of Management Science and Informatics, Veľký diel, 01026 Žilina, Slovakia, Tel. +421 415134259, **e-mail: mike@frcatel.fri.utc.sk**.

INFLUENCE OF YIELD FUNCTION ON MODELING OF MECHANICAL RESPONSE OF SOIL MATERIALS

Tomáš Janda and Michal Šejnoha

Apart from almost classical models such as Mohr-Coulomb or Drucker-Prager elasto-plastic models, there exist a number of more advanced models enable to predict fundamental aspects of grainy materials. The widely used approach for the prediction of mechanical response of fine grained soils is to follow the concept of critical state. This concept suggests that for every stress state there exists a certain critical void ratio the soil reaches after sufficiently large shearing. To arrive at the critical state the material can either contract or dilate depending on the value of the initial void ratio.

The concept of the critical state of soil material was first introduced by Hvorslev [1] and further developed by Roscoe and his co-workers [2] to eventually arrive at the formulation of modified Cam clay (MCC) model [3]. The model can address, at least qualitatively, most of the characteristics of the real soil behavior including hardening or softening and the final collapse of soil when reaching the critical state. Nevertheless, it has been suggested in the literature that the original formulation suffers from a number of drawbacks particularly on the dilation part predicting unrealistic softening behavior that occurs for heavily overconsolidated materials.

One of the objectives of this paper is to propose some modifications to the original formulation in attempt to reduce the above undesirable feature of the MCC model. It is expected that more realistic response of the soil on the dilation part can be provided by substituting the original yield function in the dilatancy domain by the formulation corresponding better with the classical Mohr-Coulomb model. In addition, the cohesion and the dependency of the yield function on the Lode angle, which appears crucial for the reliable prediction of collapse of soil, is adopted. This modification allows for approximate fitting the Mohr-Coulomb yield criterion while taking the critical state concept of the MCC model into account. It is worth noting that the generalization of the original MCC yield surface requires only to additional material parameters, namely the peak angle of internal friction and the cohesion. For further reference the proposed model is named the generalized Cam clay (GCC) model. The corresponding yield surface is plotted in Fig. 1.

To appreciate the predictive capabilities of the new formulation we present a number of examples. First, the conventional triaxial shear test is performed to highlight the major differences in the MCC and GCC models. The influence of shape of the yield function in the deviatoric plane is examined next by solving the problem of foundation strip settlement. A similar problem was also addressed in [4]. The result of this analysis appears in Fig. 2 showing the variation of the vertical displacement as a function of the applied load. The collapse load predicted by the GCC modeling is considerably smaller when compared to the one found with the MCC model. This is attributed to the fact that the Lode angle dependent yield surface of the GCC model turns on the plastic deformations at much early stages of loading since being essentially inscribed into the circular surface of the MCC model.

Acknowledgement

The financial support provided by the grant No. 1ET410430516 awarded by the Czech Academy of Sciences is gratefully acknowledged.

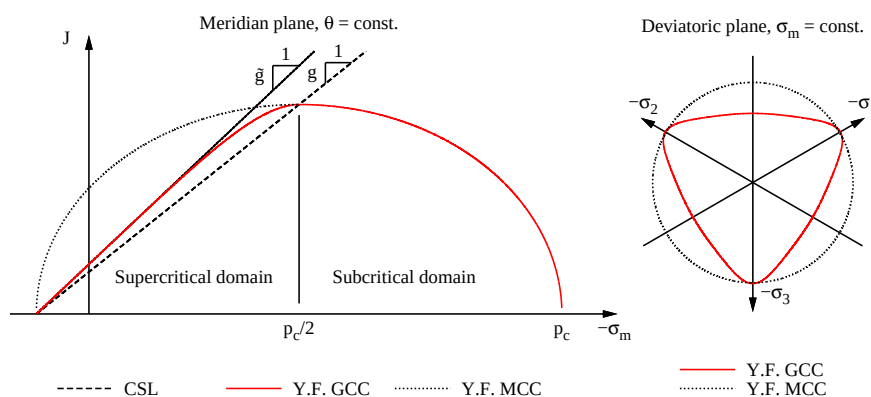


Figure 1. Yield function of the generalized Cam clay model

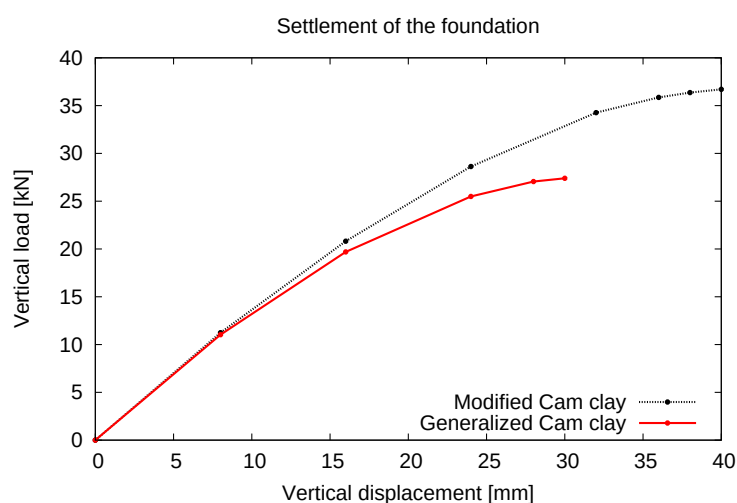


Figure 2. Comparison of settlement progress

References

- [1] Hvorslev, M. J., "On the physical properties of disturbed cohesive soils", Ingeniorvidenskabelige Skrifter. A, No. 45, 1958, pp. 159
- [2] Roscoe, K. H. and Schofield, A. N. and Worth, C. P., "On the yielding of soils", Geotechnique, Vol. 8, 1958, pp. 22–55
- [3] Roscoe, K. H. and Burland, J. B., "On the generalised behaviour of 'wet' clay", Engineering plasticity, Heyman, J. and Lechie, F. A., Cambridge University Press 1968, pp. 535–609
- [4] Potts, D. M. and Zdravkovič, L., "Finite element analysis in geotechnical engineering – theory", Thomas Telford, London, 1999

Ing. Tomáš Janda

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354498, e-mail:tomas.janda@fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420-2-24354494, Fax:+420-2-24310775, e-mail:sejnom@fsv.cvut.cz

Vortex Structures in Blade Cascades

D. Simurda, T. Hyhlik, P. Safarik

Keywords: blade cascade, horseshoe vortex, channel vortex, flow separation, tip-clearance vortex, vortex identification

1. Introduction

Capability of constructing more efficient turbine and compressor stages is based on understanding the features of a flow in the stages. Vortex structures being part of the flow features have evident influence on efficiency. Hence understanding principles of vortex generation and its development in the stages is essential.

2 Computational models

Numerical solution was performed for both 2D and 3D turbine and compressor blade cascade using commercial code Fluent 6.1.

Fluid was the air ($\rho = 1,225 \text{ kg/m}^3$, $\mu = 1,7894 \cdot 10^{-5} \text{ kg/ms}$).

3 Methods of vortex identification

3.1 Mapping of streamlines onto a plane normal to the vortex core

Structure is called vortex when instantaneous streamlines mapped onto a plane, normal to the vortex core, exhibit roughly spiral or circular pattern, when viewed from a reference frame moving with the centre of the vortex core [1].

3.2 Q-criterion [2]

Vortex exists in locations where rotation dominates over strain. The second invariant Q of velocity gradient is positive in such locations.

$$Q = \frac{1}{2}(\Omega_{ij}\Omega_{ij} - S_{ij}S_{ij}) \quad (1)$$

$$Q \geq Q_{th} \quad (2)$$

3.3 Normalized helicity [3]

Angle between velocity vector and vector of vorticity is zero in vortex core. Normalized helicity is defined as a cosine of this angle.

$$H_n = \frac{w_i \cdot \omega_i}{|w_i| \cdot |\omega_i|} \quad (3)$$

Hence the vortex core is defined as

$$|H_n| = 1. \quad (4)$$

4 Identified vortex structures

4.1 Horseshoe vortex

Vortex takes place due to existence of a pressure gradient on a leading edge. The pressure gradient originates from interaction of wall boundary layer and the leading edge. Upper branch of the vortex copies upper surface of a blade whereas bottom branch is pushed due to pressure gradient in the channel passage off the bottom surface of the blade.

4.2 Passage vortex

When a fluid flows through a curved channel, centrifugal forces acts on fluid particles in non-uniform velocity profile and passage vortex is generated.

4.3 Induced vortices

This vortex usually coexists with inducing contrarotating vortex structure. It takes place in locations where it is impossible for fluid to circulate in the sense of the inducing structure.

4.4 Flow separation

Due to inverse pressure gradient in compressor flow passage, the flow separates from the airfoil and side wall close to trailing edge.

4.5 Tip-clearance vortex

When there is a gap between casing and the blade tip, fluid flows from high pressure region on the bottom surface of the blade to the low pressure region on the suction surface at the tip of the blade. This flow is usually referenced as a leakage flow. This effect is also influenced by the wall movement relative to the blade.

5 Loss rise estimation

Loss rise due to vortex structures was estimated as a difference between loss coefficients in 2D and 3D cases for both blade cascades.

6. Conclusion

Principles of generation and further development of vortex structures in blade cascades are explained and it is shown that influence of vortex structures on the efficiency of blade stages is considerable. Vortex structures identified in this paper corresponds to model of the secondary flow in turbine blade cascades proposed in [4].

References

- [1] Robinson, S. K., "Coherent Motions in the Turbulent Boundary Layer", *Ann. Rev. Fluid. Mech.* Vol. 23., 1991.
- [2] Levy, Y., Degani, D., Seginer, A., "Graphical Visualization of Vortical Flows by Means of Helicity", *AIAA Journal*, Vol.28, 1990, pp. 1347-1352.
- [3] Sahner, J., Weinkauff, T., Hege, H. C., "Galilean Invariant Extraction and Iconic Representation of Vortex Core Lines", *EUROGRAPHICS-IEE VGTC Symposium on Visualization*, 2005.
- [4] Chen, N., Wang, Z., Wang, S., Feng, G., "A Study on Secondary Flow Pattern and Blade Bowing Effects in Turbine Bladings", *International Symposium on Experimental and Computational Aerothermodynamics of Internal Flows*, Gdansk, Poland, 2001, pp. 19-43.

Ing. David Simurda

Ing. Tomas Hyhlik, Ph.D.

Prof. Ing. Pavel Safarik, CSc.

CTU in Prague, Faculty of Mechanical Engineering, Department of Fluid Mechanics and Thermodynamics, Technicka 4, Prague, Czech Republic

d_simurda@hotmail.com, Tomas.Hyhlik@fs.cvut.cz, Pavel.Safarik@fs.cvut.cz

SOME REMARKS ON DISPLACEMENT BASED DYNAMIC MEASUREMENTS

Ivica Kožar, Tino Medvidović,

Keywords: dynamic displacement measurement, kinematical acceleration, moving load, structural eigenfrequency

1. Introduction

Dynamic measurements on structures are common engineering task providing one with relevant insight into structural behaviour. The most frequent task is determination of eigenfrequencies of a structure and sometimes modal shapes are also of interest. Eigenfrequencies are typically assessed using accelerometers and performing discrete Fourier transform (DFT) of the result. Lately new equipment for measuring displacement instead of acceleration is appearing with fairly large number of them being based on laser devices. The main reason for introducing such devices is their ability to acquire data without direct contact with the structure under consideration; only laser beam is focused on a receiver, sometimes from a distance of almost 400 meters. If the device is capable of dynamic data acquisition then one device could be used for static as well as dynamic measurements. Eigenfrequencies are extracted using DFT as in acceleration-based measurements. The question is whether there is any difference in results obtained performing DFT on dynamic displacements or on accelerations. In answering the question we will divide the structures into those excited with a dynamic force that has fixed position on the structure and those with a source that moves over the structure. The later one includes bridges, roads, railways etc. under the influence of a moving loading. In all the problems and examples considered it is assumed that the accuracy of the results presents no problems.

2. Structures with fixed position of dynamic force

This type of structures is characterized with artificial loading such as some kind of oscillating machinery. If we recall that the solution of the system of differential equations describing this problem consists of homogeneous and particular part and that the homogeneous one has the form $x = A \cdot e^{i\omega t}$ and that the acceleration is $\ddot{x} = -\omega^2 x$ then we could make some conclusions about the differences between the results from the dynamic measurements via displacements and via accelerations. Both groups of data contain the same information but accelerations are scaled with squared frequency so higher frequencies would have higher amplitudes in the power spectra. The consequence is that measurement of the dynamic displacements is better suited for large civil engineering structures where lower frequencies are of greater significance. In the following example power spectra for various dynamic forces are compared for both displacements and accelerations.

3. Structures with moving force

For a moving force it is not necessary to change in time in order to produce dynamic effects in structure. Actually we could separate dynamic behaviour into two parts: one belonging to oscillations of the structure due to structural masses and the other belonging to change of deflection due to speed of the moving force. We shall see that the fact that the moving force produces both static

and dynamic displacements has to be taken into account when considering data obtained from dynamic displacement measurements on structures with the moving load.

4. Conclusion

Dynamic displacements of structures resulting from a moving load contain the static component that pollutes the frequency domain and makes difficult determination of eigenfrequencies of the structure. In some instances it is possible to remove them by simple filtration of the frequencies below the half of the first eigenfrequency. However that procedure is not always effective and it is suggested that static displacement resulting from the moving load should be subtracted before proceeding with the frequency analysis. For most structures static displacement in the point of measurement could be numerically calculated. Results in the frequency domain obtained in that manner are typically superior to accelerations obtained through numerical derivation of displacements. For carefully selected loading speed elastic static component of the displacement can be extracted from the recording of dynamic displacements and used for other purposes.

References

- [1] Kožar I, Štimac I, 'Dynamic Analysis of Loads Moving Over Structures', in proceedings of 4th International Congress of Croatian Society of Mechanics, Bizovac, Hrvatska, 18-20.09.2003., p. 269 – 274.
- [2] Kožar I, Štimac I, 'Numerical Modelling of Beam Wave Equation', in Proceedings of 1st Symposium Computing in Engineering, Zagreb, Hrvatska, 4-6.12.2003., Faculty of Civil Engineering, Zagreb, p. 33 – 40.
- [3] Kožar I, 'Numerical Analysis of Structures under Moving Load', in Multi-physics and Multi-scale Computer Models in Non-linear Analysis and Optimal Design of Engineering Structures under Extreme Conditions, in Ibrahimbegović A, Brank B (eds.): proceedings of NATO Advanced Research Workshop, PST.ARW 980268, Bled, Slovenia, 13-17.06.2004., Faculty of Civil and Geodetic Engineering of University of Ljubljana, Slovenia, p. 527-530.
- [4] Kožar I, 'Nonlinear analysis of railway under moving load', in Soize, G.I. Schueller (eds), EUROLYN 2005: proceedings of the 6th International conference Structural Dynamics, Paris, France, 4-7. September 2005, Millpress Rotterdam Netherlands 2005, p.2071 – 2076.
- [5] Shames I: Engineering Mechanics Dynamics, Prentice Hall Europe, 1998.
- [6] Štimac I, Meštrović D, Kožar I, 'Analiza mostovnih konstrukcija pobuđenih pokretnim opterećenjem' (in Croatian), GRAĐEVINAR vol. 56, no. 6, p. 347-353, 2004
- [7] Wilson E: Three-Dimensional Static and Dynamic Analysis of Structures, CSI, Berkeley, California, USA, 2002

Ivica Kožar, Faculty of Civil Engineering Rijeka, Computer modelling of materials and constructions, V.C. Emina 5, Rijeka, CROATIA, tel: 352 178, fax:332 816, ivicak@gradri.hr

Tino Medvidović, Faculty of Civil Engineering Rijeka, Computer modelling of materials and constructions, V.C. Emina 5, Rijeka, CROATIA, tel: 352 170, fax:332 816, tinom@gradri.hr

A MODEL FOR CYCLIC FINITE STRAIN THERMOPLASTICITY

Marko Čanađija and Josip Brnić

Keywords: non-associated plasticity, kinematic hardening, thermoplasticity, finite strains

1. Introduction

This paper will present a model of nonassociated plasticity. Model is placed in thermodynamic framework, and a thermodynamically consistent model suitable for finite plastic strains is presented. The mechanical problem is augmented with a thermal problem what gives rise to the thermoplastic material behaviour. This can simulate experimentally noticed heating during plastic deformation and softening of materials due to the temperature increase. Model is based on hyperelastic formulation and involves multiplicative separation of deformation gradient into elastic and plastic part. Additional configuration, the so-called center configuration, is introduced to properly simulate movement of the yield surface center. The cornerstone of numerical simulation is the finite element method with displacements and temperatures as primary variables.

2. Kinematics

The principal measure of deformation is deformation gradient. In order to define elastic and plastic part of deformation, we introduce multiplicative decomposition of deformation gradient into elastic and plastic part:

$$\mathbf{F} = \mathbf{F}^e \mathbf{F}^p. \quad (1)$$

Plastic part of deformation gradient can be treated as mapping to stress-free intermediate configuration, Fig. 1. Mapping from intermediate configuration to the current configuration is governed by $\mathbf{F}^e$. In order to define kinematic hardening through movement of the center of yield surface, we introduce center configuration determined with deformation gradient $\mathbf{F}^k$, Fig. 1 [3, 5, 7].

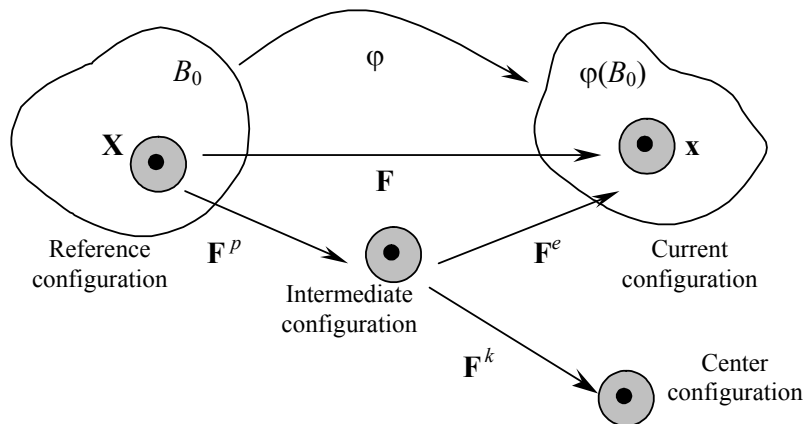


Figure 1. Configurations

3. Thermodynamical Preliminaries

To obtain evolution equations, we start from the second law of thermodynamics [4]:

$$\rho_0 N \frac{\partial \Theta}{\partial t} + \frac{\partial \Psi}{\partial t} - \frac{1}{2} \mathbf{S} : \dot{\mathbf{C}} + \frac{1}{\Theta} \mathbf{Q} \cdot \text{Grad } \Theta \leq 0. \quad (2)$$

Final form of evolution equations is:

$$\dot{\mathbf{F}}^p = (\lambda \partial_{\Sigma} g) \mathbf{F}^p \quad ; \quad \dot{\mathbf{F}}^k = (-\lambda \partial_{\mathbf{B}} g) \mathbf{F}^k \quad ; \quad \dot{k}_\alpha = -\lambda \partial_{K_\alpha} g. \quad (3)$$

4. Material model

Helmholtz's free energy function Ψ used in this problem is used in the following form:

$$\Psi = W(\bar{\mathbf{C}}^e) + U(J) + T(\Theta) + M(J, \Theta) + K^{iso}(k) + K^{kin}(\bar{\mathbf{C}}^k), \quad (4)$$

5. Numerical Procedures

The numerical procedure is based on the mixed finite element method. Problem is decoupled by application of the isothermal split. Consistent tangent operators in each phase are evaluated by numerical differentiation. Evolution laws, eqs. (3), are discretized in time and their increments are evaluated from following nonlinear system of equations by Newton-Raphson method:

$$\begin{bmatrix} \partial_{\mathbf{F}^e} \mathbf{R}^e & \partial_{\mathbf{F}^k} \mathbf{R}^e & \partial_k \mathbf{R}^e \\ \partial_{\mathbf{F}^e} \mathbf{R}^k & \partial_{\mathbf{F}^k} \mathbf{R}^k & \partial_k \mathbf{R}^k \\ \partial_{\mathbf{F}^e} R^{iso} & \partial_{\mathbf{F}^k} R^{iso} & \partial_k R^{iso} \end{bmatrix} \circ \begin{Bmatrix} \Delta \mathbf{F}^e \\ \Delta \mathbf{F}^k \\ \Delta k \end{Bmatrix} = \begin{Bmatrix} -\mathbf{R}^e \\ -\mathbf{R}^k \\ -R^{iso} \end{Bmatrix}. \quad (5)$$

Acknowledgement

This research is made possible through financial support of Ministry of Science, Education and Sport of Republic of Croatia within the research project no. 0069-006. This support is gratefully acknowledged.

References

- [1] Coleman, B. D., Gurtin, M. E., "Thermodynamics with Internal State Variables", The Journal of Chemical Physics, Vol. 47, 1967, pp. 597-613.
- [2] Čanađija, M., Brnić, J., "Associative coupled thermoplasticity at finite strains with temperature dependent material properties", International Journal of Plasticity, Vol. 20, 2004, pp. 1851-1874.
- [3] Håkansson, P., Wallin, M., Ristinmaa, M., "Comparison of isotropic hardening and kinematic hardening in thermoplasticity", International Journal of Plasticity, Vol. 21, No. 7, 2005, pp. 1435-1460.
- [4] Marsden, J. E., Hughes, T. J. R., "Mathematical Theory of Elasticity", Dover Publications, New York, 1994.
- [5] Menzel, A., Ekh, M., Runesson, K., Steinmann, P., "A framework for multiplicative elastoplasticity with kinematic hardening coupled to anisotropic damage", International Journal of Plasticity, Vol. 21, No. 3, 2005, pp. 397-434.
- [6] Simo, J. C., Miehe, C., "Associative Coupled Thermoplasticity at Finite Strains: Formulation, Numerical Analysis and Implementation", Computer Methods in Applied Mechanics and Engineering, Vol. 98, 1992, pp. 41-104.
- [7] Wallin, M., Ristinmaa, M., Ottosen, N. S., "Kinematic Hardening in Large Strain Plasticity", European Journal of Mechanics - A/Solids, Vol. 22, No. 3, 2003, pp. 341-356.
- [8] Wriggers, P., "Nichtlineare Finite-Element-Methoden", Springer-Verlag, Berlin, 2001.
- [9]

Assist. Prof. Marko Čanađija, D. Sc.

Faculty of Engineering, University of Rijeka, Department of Engineering Mechanics, Vukovarska 58, 51000 Rijeka, Croatia, Telephone ++385 51 651 496, Telefax ++385 51 651 490, markoc@riteh.hr

Prof. Josip Brnić, D. Sc.

COMPUTATIONAL SIMULATION AND VERIFICATION OF THE BOLT HEAD TRIMMING PROCESS

Jindrich Petruska, Jan Borkovec and Rudolf Foret

Keywords: FE simulation, material cutting, ductile fracture, bolt production

1. Introduction

Bolt head trimming is an important operation of bolts production, following the forward extrusion of shaft and compression of head [1]. It is realized by a trimming matrix, showed in Fig.1. Crucial problem of the bolt head cutting simulation is the material model, incorporating an effective model of ductile fracture. This should describe the damage cumulation, fracture initiation and its growth up to the stage of material separation in an effective and realistic way. The effectivity is very important here, as the local ductile fracture model is a part of a global macroscopic description of the whole complicated geometry of interacting parts of bolt, tool and other pieces. In this paper, we tested three different ductile fracture models in the specific conditions of bolt head trimming:

- limit strain model,
- modified Gurson model,
- model EWK.

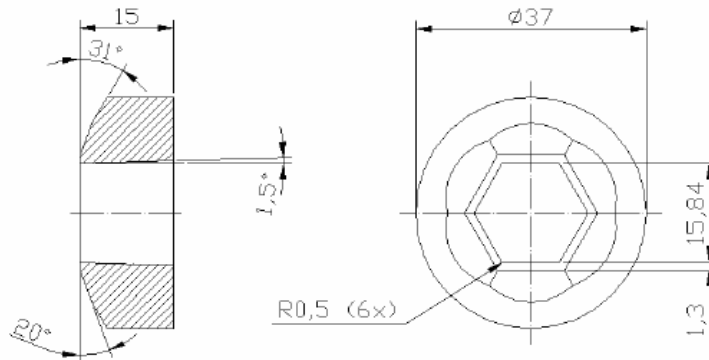


Figure 1. Trimming matrix geometry

2. Computational model and experiments

To cut the computational time, an axisymmetrical model was used for all the presented solutions. Explicit FE codes Ansys/LS-Dyna and PAM-CRASH were used for the simulations.

Elastic behaviour of the tool was supposed during the whole process simulation. Coulomb friction with friction coefficient 0.15 was applied between the tool and semi-product. Material of the semi-product is a special bolt steel 23MnB4 with flow curve obtained from compression tests. Bolt material was supposed to be elasto-plastic, isotropic, with von Mises yield condition. Three different fracture criteria were used.

To verify the numerical results, cutting experiments were realized on a mechanical press in the company J-VST Brno. Movement of the cutting tool was stopped in predefined positions, so that different stages of partially removed chips were obtained at different specimens. By metallographic analysis of these, full view of plastic deformation and fractures in the vicinity of the cutting region can be seen (Fig.2), and their comparison with the computational results can be done (Fig.3).

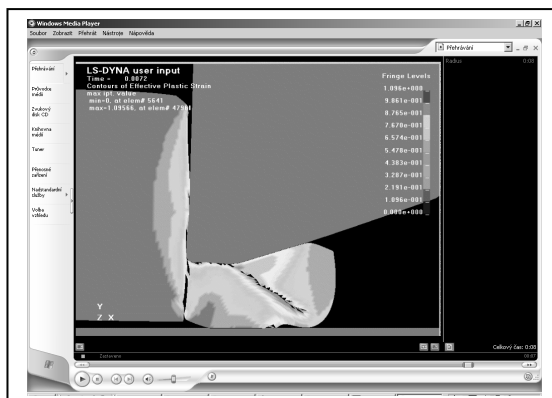
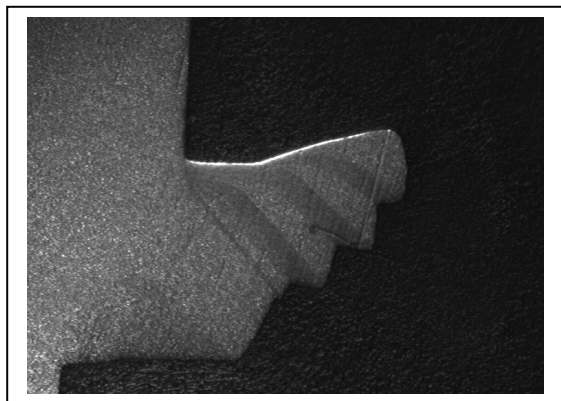


Figure 2. Cutting – experiment
Figure 3. Cutting – computational simulation

3. Results and conclusions

The results showed, that the simulation can be successfully used for prediction of important characteristics and process parameters. Especially the global power and force characteristics were in a good correspondence with the experiment. Also the orientation of shear band and fracture was correct, although the character of its distribution should be improved. Obtained results helped to suggest some changes of geometry of tooling to decrease the necessary cutting force and power and increase the trimming matrix durability.

Three different ductile fracture criteria were tested. The results show, that development of generally applicable and reliable fracture criteria is still an actual problem of future research, especially in the case of 3D problems. At the same time, such criteria should be simple enough, so that robust methods of material data calibration from basic material tests could be used.

Acknowledgement

This work is supported by the Grant Agency of the Czech Republic under the project No. 1-101/06/0914

References

- [1] Janíček, L., Petruška, J., Maroš, B., Rusz, S., "Cold forming of bolts without thermal treatment", J.Mat.Proc.Tech., Vol. 125-126, 2002, pp 341-346
- [2] Gurson, A. L., "Continuum theory of ductile rupture by void nucleation and growth", J.Eng.Mat.Tech., Vol. 76-Mat-CC, 1977
- [3] Needleman, A., Tvergaard, V., „An analysis of ductile rupture in notched bars“, J.Mech.Phys.Solids, Vol. 32, No.6, 1984
- [4] Laš, V., Řehounek, L., „Aspects of modelling ductile fracture“, Proc. UWB, Vol.5, 2001, pp 115-126
- [5] Wilkins, M. L., „Computer Simulation of Dynamic Phenomena“, Springer-Verlag, Berlin, 1999

Assoc. Prof. Jindrich Petruska

Brno University of Technology, Faculty of Mech.Eng., Technická 2, 616 69 Brno, Czech Rep., tel +420 54114 2858, fax +420 54114 2876, e-mail petruska@fme.vutbr.cz

Jan Borkovec, MSc.

Brno University of Technology, Faculty of Mech.Eng., Technická 2, 616 69 Brno, Czech Rep., tel +420 54114 2874, fax +420 54114 2876, e-mail jan.borkovec@centrum.cz

Prof. Rudolf Foret

Brno University of Technology, Faculty of Mech.Eng., Technická 2, 616 69 Brno, Czech Rep., tel +420 54114 3191, fax +420 54114 2876, e-mail foret@fme.vutbr.cz

Modeling of Long Structures on Infinite Soil

Ivica Kožar, Neira Torić

Keywords: Modeling of Long Structures on Infinite Soil

1. Introduction

There are many soil models that are used in calculation of structure - soil interaction but still the most used one is Winkler model. With the only advantage of being simple it has many drawbacks since the model localizes soil deformation. In the static analysis the consequence is that the ground deforms only where the force is present (no soil deformation outside the edges of a structure). In the dynamic analysis Winkler model does not allow propagation of transversal waves but through the structure so dynamic waves are reflected from below and from the edges of a structure [5].

In dynamic analysis one could easily improve the model behaviour by introducing either discrete or continuous damping in such a way that wave reflections are significantly reduced. On the other hand the same method can not be applied in static analysis. Intention of the paper is to introduce a model usable in both static and dynamic analysis with the obvious benefit of sharing of parameters.

The well-known approach to the problem is through use of some of more elaborate soil models like Boussinesq elastic half-space model. Simpler approach of improving the model of a structure on a soil would be by imposing endless boundaries on a soil which is the idea behind this paper.

In this paper, based on the idea of an endless beam we have developed endless boundary for Winkler soil model. The model behaves well in both static and dynamic analysis as it is demonstrated in numerical examples.

2. Model of infinite soil

The model is based on an idea of an endless beam lying on Winkler foundation. If we take the usual shape function for 3-point beam [1] and assume that the last point displacement is zero we obtain for the beam shape function

$$u = \begin{bmatrix} \frac{1}{2}(-\xi + \xi^2) & 1 - \xi^2 \end{bmatrix} \begin{Bmatrix} \Phi_1 \\ \Phi_2 \end{Bmatrix} \quad \text{or} \quad u = \underline{N} \cdot \underline{\Phi} \quad (1)$$

We have for stiffness matrix that represents soil with endless boundary

$$\underline{K}_{soil} = B \int_{-1}^1 \underline{N}^T \cdot \underline{N} \cdot \frac{2a}{(1-\xi)^2} \cdot d\xi = 2 \cdot B \cdot a \int_{-1}^1 \begin{bmatrix} \frac{1}{4} \cdot \xi^2 & -\frac{1}{2} \cdot \xi \cdot (1+\xi) \\ -\frac{1}{2} \cdot \xi \cdot (1+\xi) & (1+\xi)^2 \end{bmatrix} \cdot d\xi \quad (2)$$

Many analysis have been performed for different beam, soil and loading parameters (hard and soft soil modulus compared to the beam stiffness, high, moderate and low force velocities) and beam with simulation of the infinite soil has performed better in all the examples. Characteristic example for medium force velocity (2 seconds needed for force to travel the beam) and medium to hard soil modulus (10 MN/m spring stiffness in Winkler soil model for beam with $D=100000$ MN/m² stiffness and force $P=10$ kN) is presented in Figures 4 and 5.

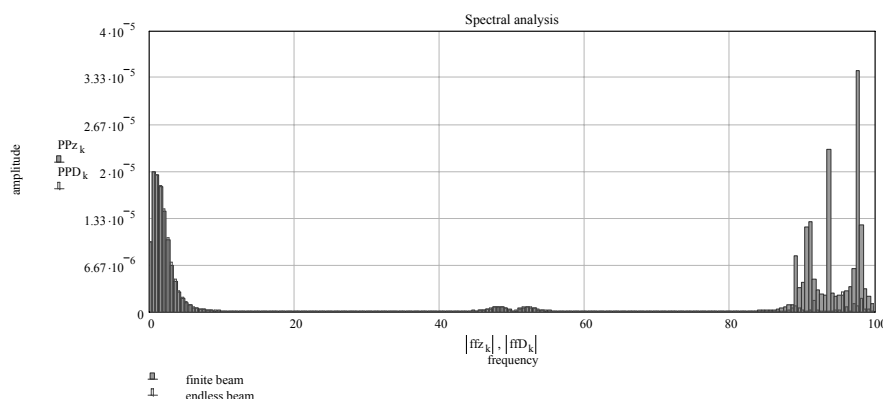


Figure 5. Comparison of frequency spectra for two beam models.

5. Conclusion

Two approaches have been compared for beam on elastic Winkler foundation: one without and the other with endless boundary conditions applied on soil. The advocated approach with simulation of infinite soil under the finite beam has significant advantages in both static and dynamic analysis: it gives more realistic results and is much more numerically stable due to highly suppressed wave reflections from the beam boundary. Presented model is considerably simpler than Bousinesq elastic half-space model.

References

- [1] Cook, R.D., Malkus, D.S., Plesha, M.E., Witt, R.J.(2002), "Concepts and Applications of Finite Element Analysis", Wiley
- [2] Štimac, I., Meštrović, D., Kožar, I., 2004, "Analiza mostovnih konstrukcija pobuđenih pokretnim opterećenjem", Građevinar, Vol. 56, No. 6, pp 347-353.
- [3] Kožar, I., 2005, "Nonlinear analysis of railway under moving load", in Soize, G.I. Schueller (eds), EUROLYN 2005: proceedings of the 6th International conference Structural Dynamics, Paris, France, 4-7. September 2005, Millpress Rotterdam Netherlands 2005, pp 2071 – 2076.
- [4] Kožar, I., Štimac, I., 2003, "Numerical Modelling of Beam Wave Equation", in proceedings of 1st Symposium Computing in Engineering, Zagreb, Hrvatska, 4-6.12.2003., Faculty of Civil Engineering, Zagreb, pp 33 – 40.
- [5] Kožar, I., 2004, "Numerical Analysis of Structures under Moving Load", in Multi-physics and Multi-scale Computer Models in Non-linear Analysis and Optimal Design of Engineering Structures under Extreme Conditions, in Ibrahimbegović, A., Brank, B., (eds.): proceedings of NATO Advanced Research Workshop, PST.ARW 980268, Bled, Slovenia, 13-17.06.2004., Faculty of Civil and Geodetic Engineering of University of Ljubljana, Slovenia , pp 527-530.

Ivica Kožar, Prof.dr.sc., Faculty of Civil Engineering, University of Rijeka, Department for computational modelling of materials and structures, V.C.Emina 5, 51000 Rijeka, Croatia,
Tel: +385 51 352 178, Fax: +385 51 332 816, E-mail: ivicak@gradri.hr

Neira Toric, Civil Engineer, Faculty of Civil Engineering, University of Rijeka, Department for computational modelling of materials and structures, V.C.Emina 5, 51000 Rijeka, Croatia,
Tel: +385 51 352 108, Fax: +385 51 332 816, E-mail: neira.toric@gradri.hr

MONITORING OF MODAL CHARACTERISTIC CHANGES OF A PRESTRESSED CONCRETE ELEMENT IN DEPENDENCE ON DYNAMIC CYCLIC LOADING

Tomáš Plachý, Michal Polák and Antonín Rezek

Keywords: modal analysis, natural frequency, mode shape, prestressed concrete, fatigue

1. Introduction

In practice, various kinds of damage on concrete structures can be found. Undetected damage can lead gradually to the failure of structural members. Therefore damage detection of a structure at the earliest possible stage is very important. The need for methods that can be applied to complex structures led to the development of methods that examine changes in modal characteristics of a structure. Methods, which use results of an experimental modal analysis for estimation of a degradation degree of a structure, are suitable to verify on simple structural elements where we know their damage. The experiment described in this paper is focused on monitoring of the influence of high intensity fatigue loading on the change of the modal characteristics of the fully prestressed concrete structures.

2. Description of the prestressed concrete slab

For the purpose of this project a prestressed concrete slab was made. The dimensions of the slab were 130 x 1155 x 4500 mm with ends expanded to the height 400 mm for tie of the prestressing cables. Slab was made from concrete C45/55 with 11 prestressing cables of diameter 15.7 mm. The slab was put on two bearings to be a simply supported with the span 3.5 m with cantilevered ends 0.5 m on both sides. The tested slab was designed as a fully prestressed concrete slab to not comply the safety condition in the lower part of the slab when the high intensive cyclic loading is applied.

3. Loading of the slab and data acquisition

Tests of the slab were carried out in laboratories of Faculty of Civil Engineering, CTU in Prague. The prestressed concrete slab was loaded in four point bending to get a constant bending moment in the mid-section of the slab. The cyclic load was applied to the slab in several steps. This fatigue load of the beam was induced by harmonic force with frequency 5 Hz. The amplitude of the dynamic load was chosen to not comply the safety condition for the fatigue loading of the concrete.

The experimental modal analysis was done after each load step. The Modal Exciter Type 2732 (Bruel & Kjaer) was used for the excitation of the slab during the modal analysis. The exciter produced a random driving force over the frequency range of 5 to 400 Hz, which was measured by the force transducer Endevco 2311-10. The response of the element onto forcing by the exciter was measured by three piezoelectric acceleration transducers Bruel & Kjaer 4507 B 005 in a chosen net of points on the upper face of the slab.

4. Results

The program MEscapeVES (Bruel & Kjaer) was used for modal characteristic evaluation from measured Frequency Response Functions. With regard to a frequency range of the dynamic analysis 5 to 400 Hz nine natural frequencies and mode shapes were evaluated. Natural mode shapes

evaluated after 125 000 loading cycles are shown in Figures 5-9. Modal characteristics of the slab, which were measured after each loading step, were mutually compared.

For comparison of natural modes, changes of a mode surface curvature $\text{CAMOSUC}_{(j),x}$ (Fig. 1), the changes of a modal flexibility matrix $[\delta]$ (Fig. 1) and the second derivative of changes of diagonal members of a modal flexibility matrix $\Delta[\delta]''$ (Fig. 1) were used.

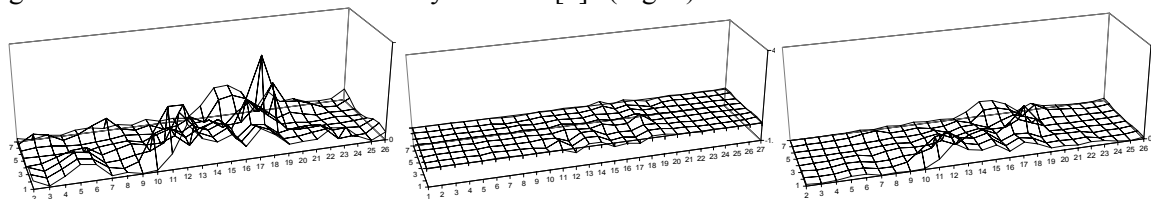


Figure 1. The $\text{CAMOSUC}_{(j),x}$, $\Delta[\delta]$ and $\Delta[\delta]''$ between the virgin state and the state after 625 000 loading cycles.

5. Conclusion

In this experimental dynamic study, the influence of the dynamic cyclic fatigue loading to the change of the modal characteristics of the fully prestressed concrete element was monitored. The modal characteristics after several loading steps (each step is equal to 125 000 loading cycles) were evaluated. Mode shapes were mutually compared using three different methods: $\text{CAMOSUC}_{(j),x}$ (Fig. 1), $\Delta[\delta]$ (Fig. 1) and $\Delta[\delta]''$ (Fig. 1). From the lower values of these methods it results that no significant changes of dynamic behavior of the slab occurred after first loading steps. After 625 000 loading cycles some values of previous functions a little bit increased, e.g. $\Delta[\delta]''$ (Fig. 1) but no more significant changes occurred in dynamic behavior of the prestressed slab. The experimental study still continues. More measurements will be done on this slab and two more identical slabs will be investigated in the same manner to make conclusions that will be more general.

Acknowledgement

First, I would like to thank SMP CONSTRUCTION a.s. for making and preparation of the slabs and for their financial participation. The investigation was supported by Czech Science Foundation as the project No. 103/05/P302 „Damage detection of a prestressed concrete element based on dynamic characteristics changes.”

References

- [1] Frýba, L., Pirner, M., Urushadze, S., “Localization of Damages in Concrete Structures“, Proceedings of Computational Methods and Experimental Measurements X, 2001, pp. 417-426.
- [2] Ndambi, J.M., De Visscher, J., Vantomme, J., De Wilde, W.P., Peeters, B., Wahab, M.A., De Roeck, G., “Material Property assessment in cracked reinforced concrete beams using dynamic system identification“, Proc. of the RILEM - IMEKO Int. Conf. Non - destructive Testing and Experimental Stress Analysis of Concrete Structures, Expertcentrum Bratislava, Bratislava, 1998, pp. 351-356.
- [3] Plachý, T., “Dynamic Study of Reinforced Concrete Beams Damaged by Cracks“, FCE CTU in Prague, PhD Thesis (In Czech), 2003.
- [4] Plachý, T.-Polák, M., “Influence of Damage of a Reinforced Concrete Beam on Change of Its Behaviour“, Proc. of the 5th Int. Conference on Structural Dynamics – EUROODYN2002, Munich, Germany, 2002, pp. 1451 –1456

Ing. Tomáš Plachý, Ph.D.

Department of Mechanics, Faculty of Civil Engineering, Czech Technical University in Prague, Thákurova 7, Praha 6, Prague, Czech Republic, +420 2 2435 4483, plachy@fsv.cvut.cz

doc. Ing. Michal Polák, CSc.

Department of Mechanics, Faculty of Civil Engineering, Czech Technical University in Prague, Thákurova 7, Praha 6, Prague, Czech Republic, +420 2 2435 4476, polak@fsv.cvut.cz

Ing. Antonín Rezek

Department of Mechanics, Faculty of Civil Engineering, Czech Technical University in Prague, Thákurova 7, Praha 6, Prague, Czech Republic, +420 2 2435 4483, antonin.rezek@fsv.cvut.cz

CONTACT SIMULATION DURING NAIL PENETRATING INTO A CONCRETE BLOCK

I. Kožar, J. Ožbolt, B. Irhan and V. Travaš

Keywords: contact problem, Signorini problem, 3D finite element, geometrical constraints

1. Introduction

In the present report a current status of the research project: “3D numerical simulation of the penetration of a nail into a concrete block” is given. Contact algorithm is employed to determine the degree of nail penetration into concrete. It is important that the algorithm is robust since it cannot be predicted what shape will concrete block take during crushing in front of nail. Large displacement and strain formulation is needed in order to realistically model behavior of concrete on the front of the nail where displacements and strains are large. Dynamic analysis is needed in order to account for the transformation of nail's kinetic energy into strain energy of concrete block (that then causes damage and crushing of concrete) as well as to take into account the influence of inertia forces on the concrete response.

2. NAIL PENETRATION

Contact problem is characterized by non-linear geometrical constraints imposed as boundary conditions. Formulation of that problem is known as Signorini problem for a solid body (Zhong, 1993).

Equilibrium:

$$\frac{\partial \boldsymbol{\sigma}}{\partial \mathbf{x}} + \mathbf{b} = \mathbf{0} \quad (1)$$

Prescribed boundary conditions:

$$\mathbf{u} = \bar{\mathbf{u}} \text{ on } \Gamma_d; \quad \boldsymbol{\sigma} \mathbf{N} = \mathbf{q} \text{ on } \Gamma_f \quad (2)$$

Contact conditions:

$$g = g_0 - \mathbf{u} \mathbf{N} \geq 0 \text{ on } \Gamma_c; \quad q_1 = \mathbf{q}_c \mathbf{N} \leq 0 \text{ on } \Gamma_c; \quad q_1 g = 0 \text{ on } \Gamma_c \quad (3)$$

Penalty method transforms constrained system of equations into unconstrained one (Cook et al. 2002). The resulting number of equation is the same for both constrained and unconstrained problem but the constraint conditions are obeyed only approximately. The parameter α is a measure of the enforcement of constraints that are described with matrix $\mathbf{C}$ (relationships among degrees of freedom) and vector $\mathbf{Q}$ (conditions to be imposed on the degrees of freedom).

$$(\mathbf{K} + \mathbf{C}^T \alpha \mathbf{C}) \mathbf{D} = \mathbf{R} + \mathbf{C}^T \alpha \mathbf{Q} \quad (3)$$

Penalty approach has been adopted in this work since it preserves the number of equations, which is already very high.

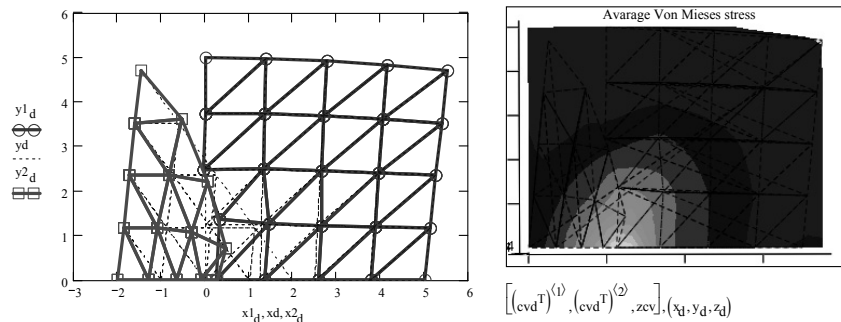


Figure 1. Numerical implementation in 2D

Conclusion

Several versions of implementation of the contact constraint have been devised and tested. Finally the penalty approach has been implemented into finite element code.

REFERENCES

- [1] Belytschko, T., Liu, W.K. and Moran, M., "Nonlinear Finite Elements for Continua and Structures". John Wiley & Sons Ltd., 2001
- [2] Cook, R.D., Malkus, D.S., Plesha, M.E., Witt, R.J., "Concepts and Applications of Finite Element Analysis", Wiley., 2002
- [3] Dilger, W.H., Koch, R. and Kowalczyk, R., "Ductility of plain and confined concrete under different strain rates". American Concrete Institute, Special publication, Detroit, Michigan, USA, 1978
- [4] Ibrahimbegović, A. and Kožar, I., "Nonlinear Wilson's Brick Element for Finite Elastic Deformation of Three-Dimensional Solids", Communications in Numerical Methods in Engineering (0748-8025), 1995, (11) pp 655-664.
- [5] Krausz, A. S. and Krausz, K., "Fracture kinetics of crack growth", Kluwer, Dordrecht, The Netherlands, 1988
- [6] Ožbolt, J., "Size effect in concrete and reinforced concrete structures". Postdoctoral Thesis, University of Stuttgart, Germany, 1995
- [7] Ožbolt, J., Li, Y.-J. and Kožar, I., "Microplane model for concrete with relaxed kinematic constraint". International Journal of Solids and Structures, 2001, 38: pp 2683-2711.
- [8] Ožbolt, J. and Reinhardt, H.W., "Three-dimensional finite element model for creep-cracking interaction of concrete". Proceedings of the sixth international conference CONCREEP-6, Ed. By Ulm, Bažant & Wittmann, 2001, pp 221-228.
- [9] Ožbolt, J., Eligehausen, R., Periškić, G. and Mayer, U., "3D FE Analysis of Anchor Bolts with Large Embedment Depths". Proceedings of the FraMCoS-5, Willam & Billington (eds.), Vail, USA, 12—16 April 2004, Vol. 2, 2004, pp 845-852.
- [10] Weerheijm, J., "Concrete under impact tensile loading and lateral compression". Dissertation, TU Delft, the Netherlands, 1992
- [11] Zhong, Z., "Finite Element Procedures for Contact-Impact Problems", Oxford University Press, 1993

Ivica Kožar, Ph.D.

Faculty of Civil Engineering University of Rijeka, Department for modelling of materials and structures, Viktora Cara Emina 5, Rijeka, Croatia, tel/fax: +385 51 352-111 / +385 51 332-816, ivicak@gradri.hr

Joško Ožbolt, Ph.D.

Institute of Construction Materials, Pfaffenwaldring 4, Universität Stuttgart, Stuttgart, Germany, tel/fax: 0711 685-63330 / 0711 685-63349, josko.ozbolt@gradri.hr

Baris Irhan, M.Sc.

Institute of Construction Materials, Pfaffenwaldring 4, Universität Stuttgart, Stuttgart, Germany, tel/fax: 0711 685-63330 / 0711 685-63349, barisirhan@yahoo.com

Vanja Travaš, M. Eng.

Faculty of Civil Engineering University of Rijeka, Department for modelling of materials and structures, Viktora Cara Emina 5, Rijeka, Croatia, tel/fax: +385 51 352-111 / +385 51 332-816, vanja.travas@gradri.hr

LATTICE MODELLING OF METALLIC FOAMS

M. Borovinšek, Z. Ren

Keywords: metallic foams, open-cell foams, lattice modelling, perturbation method, Voronoi tessellation

1. Introduction

Materials, combining lightweight and high mechanical energy absorption, are of great interest for a variety of automotive, locomotive, marine, and aerospace applications. Recent advances in materials fabrication and processing have led to emergence of metallic foams, which combine low density and high efficiency in absorbing external mechanical energy through its deformation. Thus, significant research effort has been focused on research of metallic foams characterisation and structural integrity [1].

Developed analytical and experimental models for predicting mechanical properties of cellular solids are usually based on idealized volume elements, i.e. unit cells. Properly defined unit cell (repeating unit) can capture the essential microstructural features of an actual cellular material.

Unit cell-based models can provide some insight into behaviour of cellular structures, but they are limited only to regular structures and can not account for microstructural imperfections which are present in real cellular materials. Since cellular structures of foams are typically non-periodic, non-uniform and disordered, more complex, statistical models are needed to obtain improved predictions. Different approach to computational modelling of cellular structures is therefore required to account for stochastic nature of cellular solids.

2. Metallic foams

Foams can be visualised as structures made of tightly packed polyhedra cells. If the base material is contained only in the cell edges (the cells connect through open faces), the foam is classified as open-cell foam. If the cell faces also contain the base material, the foam is called the closed-cell foam (Figure 1).

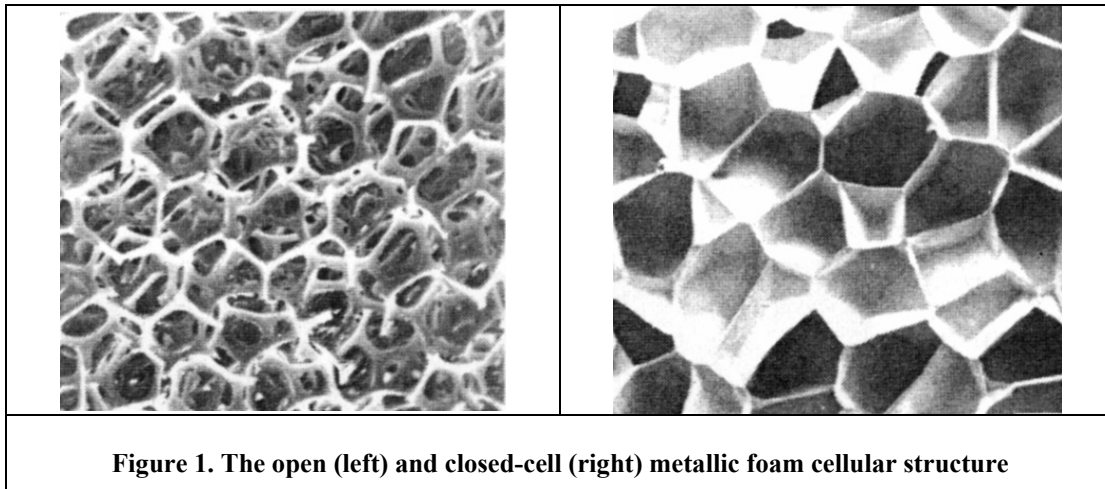
Metallic foams are a new class of materials, which were only recently developed to its full potential. They are produced with a range of novel processing techniques, many still under development. However, the production processes are imperfectly controlled, resulting in some variability in properties. This invariably leads to incomplete characterisation of metallic foams macroscopic material properties.

3. Modelling of irregular open-cell structures

Aluminium open-cell foams may be best represented by packed tetrakaidecahedral cells, since the tetrakaidecahedron is known to be the polyhedron that can pack with identical units to fill space by minimizing the surface energy [2].

A regular open-cell lattice model can thus be constructed by using the tetrakaidecahedral cells and replacing the edges of tetrakaidecahedral cells with the beam finite elements. The irregularity of such lattice models is introduced by controlled disorder of lattice vertices coordinates [3], while the cross-sectional area of beams and their shape remain identical on all edges.

Due to the similarity between the mathematical procedure of the Voronoi tessellation and the physics of foam production, the Voronoi tessellation technique is another choice for describing the structure of foams resulting from a bubble forming process [4].



4. Conclusions

Two different methods for 3D reproduction of irregular open-cell structures are presented. The first method is based on perturbation of the regular open-cell structure while the second method uses the Voronoi tessellation of three-dimensional space.

Both presented methods have certain advantages. The Voronoi tessellation method produces lattice models with different cell topologies while perturbation method preserves the cell topologies. At the same time, the perturbation method takes significantly less time for generation of larger lattice models and produces meshes with evenly distributed cell edge lengths.

A mesh smoothing technique was introduced to assure proper stability and accuracy of the presented lattice models for their efficient use in explicit dynamic simulations. Since it involves the removal of short beam elements from the lattice mesh and so introduces some minor changes to the mesh geometry, the influence of using this procedure on the simulation results has to be evaluated using comparative dynamic simulations of smoothed and unsmoothed meshes.

References

- [1] Gibson, L. J., Ashby, M. F., "Cellular solids, Structure and Properties, Second Edition", Cambridge University Press, Cambridge, United Kingdom, 1999.
- [2] Roberts, A. P., Garboczi, E. J., "Elastic properties of model random three-dimensional open-cell solids", " Journal of the Mechanics and Physics of Solids, Vol.50, 2002, pp 33-55.
- [3] Zheng , Z., Yu J., Li , J., "Dynamic crushing of 2D cellular structures: A finite element study", International Journal of Impact Engineering, Vol.32, 2005, pp 650-664.
- [4] Huang, J. S., Gibson L. J., "Creep of open-cell Voronoi foams", Materials Science and Engineering, Vol.339, 2003, pp 220-226.

Matej Borovinšek, Uni. Dipl. Ing. Mech. Eng.
 University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, Maribor, Slovenia,
 matej.borovinsek@uni-mb.si
 Prof. Dr. Zoran Ren, Uni. Dipl. Ing. Mech. Eng.
 University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, Maribor, Slovenia,
 ren@uni-mb.si

SIMULATION OF ROAD SAFETY BARRIER CRASH TESTS

M. Borovinšek, Z. Ren

Keywords: road safety barrier, computational crash simulations, explicit FE code

1. Introduction

Safety is becoming one of the most important aspects in modern road traffic. The car industry is developing new passive and active vehicle safety systems to increase the safety of vehicle occupants while at the same time the roads are made safer with installation of more effective road restraint systems.

Ensuring appropriate safety levels in road traffic is of significant importance. One way to improve road safety is to install appropriate road restraint systems. The purpose of such system is to prevent a vehicle from veering off the road and to restrain it from entering dangerous area to avoid or reduce the injuries of vehicle occupants and other road traffic participants and objects.

Road restraint systems installed on public roads in Europe have to fulfill the European EN 1317 standard [1], which mandates certain full scale crash tests to certify safety barriers for targeted containment levels. To reduce the development and testing costs of new safety barrier designs it is advisable to use computational crash simulations for early evaluation of safety barrier behaviour under test vehicle impact. That way the impact severity parameters and barrier deformation can be estimated well in advance of a real crash test and possible design changes can be easily evaluated.

2. Description of used computational models

The computational model for safety barrier crash simulation consists of the barrier and vehicle models. The barrier parts are modeled in compliance with their design using necessary simplifications. The vehicle models are purposely built or can be retrieved from publicly accessible vehicle libraries, like the one at the National Crash Analysis Center (NCAC) [2].

Different FE vehicle models (TB11, TB32, TB42, TB51 and TB61) which were developed and validated in order to perform crash simulations of road safety barriers are described. The FE models were developed using the vehicle library at NCAC and subjected to certain changes to fulfill the EN 1317 standard regulations.

The main safety barrier parts, guardrails, posts, distant spacers and additional reinforcements are modeled in detail, while the bolt connections are represented with linear beam elements. The required length of the modeled safety barrier depends on the type of the test. The position of impact point (the point of the first vehicle – barrier contact) depends on the length of the modeled barrier and the test type. Continuation of the guardrail and possible longitudinal barrier reinforcements are modeled with elastic spring elements while the soil influence is simulated by using the spring elements with elasto-viscoplastic characteristic varying with depth.

3. Computational simulations

Two different computational studies are presented to show the applicability of computer simulations for safety barrier evaluation. In the first study different reinforcements of the road safety bar-

rier are first introduced and then tested using computer simulations. At the end, the most suited reinforcement is chosen according to the EN 1317 requirements. In the second study the simulation results of computational simulations are compared to the large scale experiment of the same road safety barrier design (Figure 1).



Figure 1. The barrier deformation during large scale test (top) and simulation (bottom)

4 Conclusions

The paper describes computational simulations of large scale crash tests by using the finite element code LS-DYNA on a multiprocessor platform. Different vehicle models were developed and validated for crash simulations of road safety barriers. To reduce the costs of development and testing of new safety barrier designs the computer simulations were used to estimate the impact severity parameters and barrier deformation before the real crash tests were conducted. That way any design changes can be easily evaluated.

Computational crash simulations used for early evaluation of different safety barrier reinforcements are presented. According to EN 1317 required crash test were simulated to compare the behaviour of the reinforced safety road barriers under test vehicle impact. Using the simulation results the most suited safety barrier design was determined. At the end a good correlation of computational and experimental results is illustrated using two vehicle impact tests for chosen containment level.

References

- [1] "EN 1317-1 to 1317-5 Road restraint system", European Committee for Standardization, 1998.
- [2] "Finite Element Model Archive [World Wide Web]", FHWA/NHTSA National Crash Analysis Center, available at: <http://www.ncac.gwu.edu/vml/models.html>, 16.5.2006.

Matej Borovinšek, Uni. Dipl. Ing. Mech. Eng.
 University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, Maribor, Slovenia,
matej.borovinsek@uni-mb.si
 Prof. Dr. Zoran Ren, Uni. Dipl. Ing. Mech. Eng.
 University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, Maribor, Slovenia,
ren@uni-mb.si

CONVERGENCE OF EIGENVALUES OF MONODROMY MATRIX OF PIECEWISE LINEAR OSCILLATORS

H. Wolf, D. Semenski and A. Sušić

Keywords: dynamical stability, Floquet-Liapounov theorem, piecewise- linear oscillator

1. Introduction

Among the great number of various types of non-linear dynamic systems, a very specific group constitutes piecewise-linear oscillators. Responses of these strongly non-linear systems can be determined in the time domain by digital simulation, but such procedures can be exceptionally time consuming. Very efficient methods for solving this type of non-linear differential equations in the frequency domain are multi-harmonic balance methods [1]. They can be successfully applied to a wide range of strongly non-linear problems, and they are very well suited for parametric studies and bifurcation analysis. Since these methods enable obtaining both dynamically stable and unstable solutions, the determination of their dynamical stability should be reliable and numerically efficient. As the estimation of dynamical stability by Floquet-Liapounov theorem can be a very sensitive procedure [2, 3], the convergence of the eigenvalues of the monodromy matrix is considered in this paper in dependence on the number of harmonics (N) used in the approximate solution.

2. Short description of the applied method

Model of a simple mechanical system with clearance and corresponding piecewise-linear function are shown in Fig. 1(a) and Fig. 1(b) respectively.

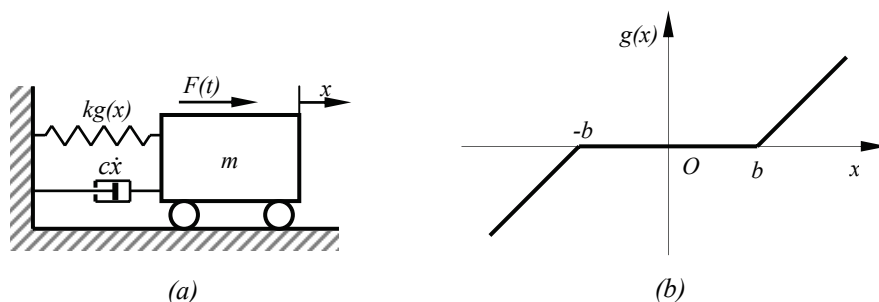


Figure 1. Model of the dynamic system (a) and corresponding piecewise-linear function (b)

By applying the incremental harmonic balance method (IHBM) the governing non-linear differential equation is transformed into the system of $2N+1$ linearized incremental algebraic equations with Fourier coefficients ($a_0, a_i, b_i, i=1, \dots, N$) as unknowns. In each incremental step, only linear (i.e. linearized) algebraic equations have to be formed and solved. The comprehensive description of the IHBM is given in [1]. The stability criteria are related to the eigenvalues of the monodromy matrix, which is defined as the state transition matrix at the end of one period. According to Floquet-Liapounov theorem [4], the solution is stable if all the moduli of the eigenvalues of the monodromy matrix are less than unity. Otherwise the solution is unstable. Bifurcation occurs when one

of the eigenvalues of the monodromy matrix crosses the unit circle in the complex plane (i.e. when one of the moduli of the monodromy matrix reaches unity).

3. Numerical example

Fig. 2(a) shows an effective amplitude-frequency plot obtained by IHB (both stable and unstable solutions). Fig. 2(b) shows a plot of maximum modulus of the eigenvalues of the monodromy matrix $|\lambda_{\max}|$ (branch a from Fig. 2(a)). One can see a frequency range with no convergence of $|\lambda_{\max}|$ with increasing N , what can lead to incorrect estimation of dynamical stability and can make bifurcation analysis difficult. Because of that, it is of great importance to detect frequency ranges inside which the procedure of evaluation of eigenvalues of the monodromy matrix does not converge with increasing the number of harmonics included in the supposed approximate solution. Absence of convergence can be explained by essential difference between IHB and the procedure of evaluating the monodromy matrix.

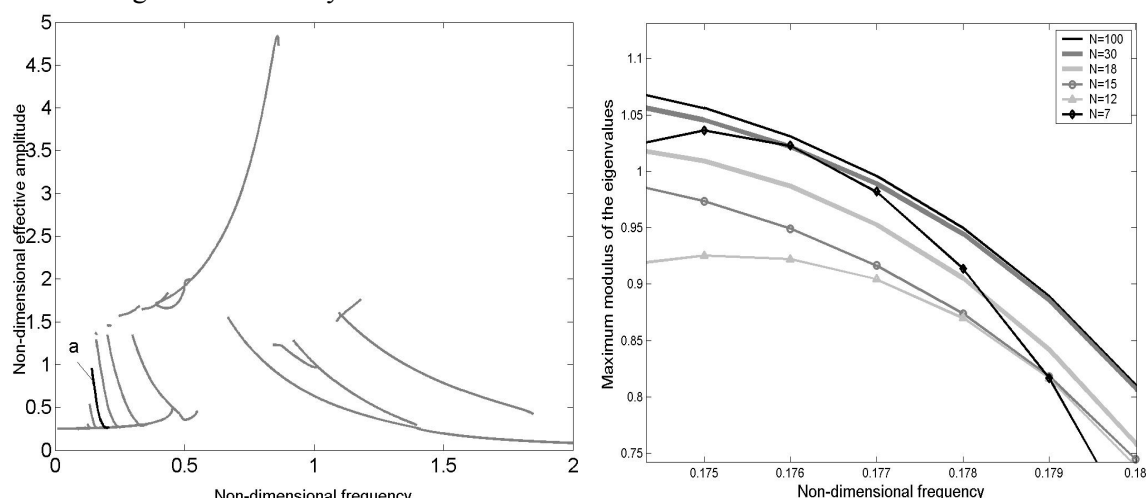


Figure 2. Effective amplitude-frequency plot (a) and a plot of $|\lambda_{\max}| = |\lambda_{\max}|(\eta, N)$ (branch a) (b)

References

- [1] Wong, C.W., Zhang, W.S., Lau, S.L., "Periodic forced vibration of unsymmetrical piecewise-linear systems by incremental harmonic balance method", Journal of Sound and Vibration, Vol. 149, 1991, pp 91-105.
- [2] Wolf, H., Kodvanj, J., Bjelovučić-Kopilović, S., "Effect of smoothing piecewise-linear oscillators on their stability predictions", Journal of Sound and Vibration, Vol. 270, 2004, pp 917-932.
- [3] Wolf, H., Terze, Z., Sušić, A., "Dynamical stability of the response of oscillators with discontinuous or steep first derivative of restoring characteristic", European Journal of Mechanics A/Solids, Vol. 23, 2004, pp 1041-1050.
- [4] Minorsky, N., "Nonlinear oscillations", D. Van Nostrand Company, Princeton, New Jersey, Toronto, London, New York, 1962

Hinko Wolf, Phd, Associated professor

Faculty of Mech. Eng. Nav. Arch./University of Zagreb, Dept. of Engineering Mechanics, Ivana Lučića 5, Zagreb, Croatia, ++38516168168, ++38516168187, e-mail: hwolf@fsb.hr

Damir Semenski, Phd, Professor

Faculty of Mech. Eng. Nav. Arch./University of Zagreb, Dept. of Engineering Mechanics, Ivana Lučića 5, Zagreb, Croatia, ++38516168211, ++38516168187, e-mail: damir.semenski@fsb.hr

Aleksandar Sušić Phd, Assistant

Faculty of Mech. Eng. Nav. Arch./University of Zagreb, Dept. of Engineering Mechanics, Ivana Lučića 5, Zagreb, Croatia, ++38516168570, ++38516168187, e-mail: asusic@fsb.hr

EXPERIMENTAL IDENTIFICATION OF NONLINEAR MATERIAL PARAMETERS OF REGULAR BRICK MASONRY

Jan Novák, Miroslav Vokáč and Michal Šejnoha

Nowadays a number of computational material models of quasibrittle materials, e. g. concrete, stone blocks, etc. are available. Nevertheless it is quite difficult to apply them when dealing with strongly anisotropic materials such as masonry. A realistic prediction of the behavior of such structures due to material or topological heterogeneity is often taken into account by multi-scale modeling. Two suitable approaches for a reliable assessment of nonlinear mechanical properties subsequently fed into constitutive models on both scales are available. Since numerical simulations suffer from certain uncertainties an approach of physical experiments is desirable and proposed in this paper.

The experimental program assumes several large scale specimens exhibiting topological anisotropy (bed joints inclination: 0, 45, 90 degrees) loaded by vertical force. As a result the homogenized loading paths are obtained. All of them can be used as comparative curves for numerical and stochastic simulations presented in accompanied paper [3]. A coherent problem stands in description of impaired properties of interfacial transition zone (ITZ) [4]. To that end the Mohr-Coulomb plasticity condition is proposed for various numerical simulations. To get the required data a number of carefully prepared shear tests were performed. While the displacement load control is assumed for compression tests, the stress control regime is considered when loading the specimens in shear.

Currently used computational procedures often quantify the resistance of masonry structures for various loadings in view of shear, pressure or tensile strength of blocks or mortar joints. In other words, only the strength of a component determines the bearing capacity of the whole structure. Geometrical arrangement of blocks inside the masonry wall is, therefore, not taken into account.

Considerably different approach to the analysis of masonry structures is offered through the application of multi-scale analysis combined with periodic homogenization exploiting the concept of periodic unit cell. In such a case the true geometry of the masonry structure can be reflected even for structures with irregular arrangement of bricks or stone blocks in case of quarry masonry, see [1, 3] for more details on this subject. Such an approach requires the knowledge of material properties, including fracture energies and interfacial properties, of individual phases - bricks and mortar.

Determination of these properties from carefully prepared laboratory tests is thus the main goal of this contribution. A special attention is paid to the determination of fracture energy of individual constituents following the procedure advocated in [2]. Apart from that, a set of large scale specimens is tested to provide macroscopic response used in turn for comparison with numerical simulation based on first order homogenization [3]. One specific example showing the macroscopic force-displacement curves and corresponding crack pattern at final stage of loading appear in Fig. 1.

The results obtained from a combination of numerical and experimental approaches should demonstrate the ability of multi-scale analysis when searching for mechanical response of the masonry structure. Up to now the completed laboratory experiments as well as the numerical simulations exhibit convenient application of this attitude in standard engineering practices even in exceptional conditions such as historical constructions (sacral and aristocratic buildings or engineering constructions such as bridges, aqueducts etc.).

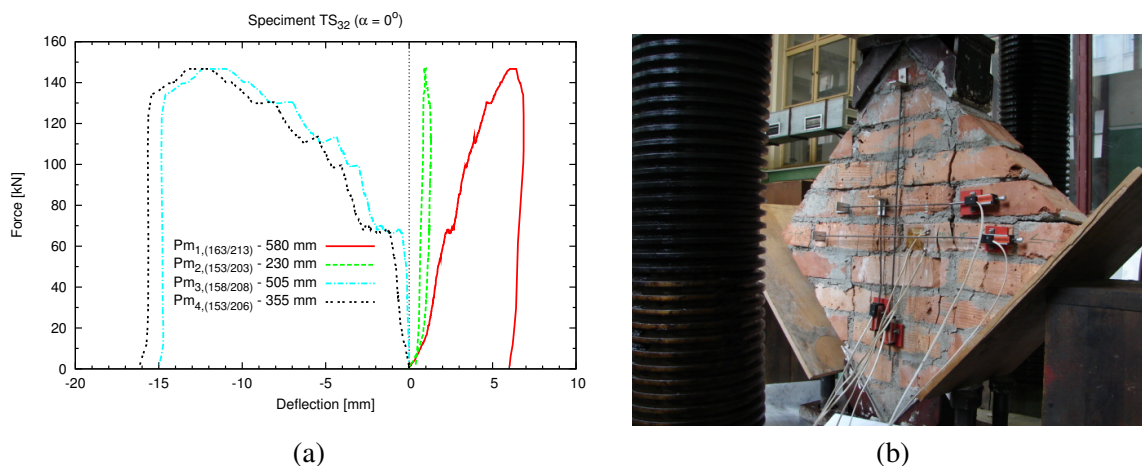


Figure 1. a) Resulting macroscopic loading curves, b) corresponding pattern of cracks

Acknowledgement

This outcome has been achieved with the financial support of the Ministry of Education, Youth and Sports, project No. 1M680470001, within activities of the CIDEAS research center. In this undertaking, theoretical results gained in the projects GAČR 103/04/1321 and 103/06/H150 were partially exploited.

References

- [1] Šejnoha, M., Zeman, J. and Novák, J., "Homogenization of random masonry structures - comparison of numerical methods", In: EM 2004 - 17th ASCE Engineering Mechanics Division Conference, Newark, University of Delaware, 2004
- [2] Karihallo, B.L. nad Abdalla, H.M., "A simple method for the determination of the true specific fracture energy of concrete", 2nd International symposium NON-TRADITIONAL CEMENT & CONCRETE, Edited by Bílek and Keršner, Brno, 2005
- [3] Vorel, J., Sýkora, J., Šejnoha, J. and Šejnoha, M., "Prediction of homogenized material properties of quarry masonry from micromechanics", presented at 5th International congress of Croatian Society of Mechanics, Split, September, 21-23, 2006
- [4] Šejnoha, J., Šejnoha, M., Sýkora, J. and Vorel, J., "An improved material model for quarry masonry", presented at a CIVIL-COMP Conference. Las Palmas de Gran Canaria, Spain 2006

Ing. Jan Novák

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354472, Fax: +420-2-24310775, e-mail: jan.novak@cml.fsv.cvut.cz

Ing. Miroslav Vokáč

Czech Technical University in Prague, Klokner Institute, Department of Experimental and Measurement Methods, Šolínova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24353509, Fax: +420-2-24353530, e-mail: vokac@klok.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Structural Mechanics, Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420-2-24354494, Fax: +420-2-24310775, e-mail: sejnomo@fsv.cvut.cz

NUMERICAL AND EXPERIMENTAL ANALYSIS OF DELAMINATION OF UNIDIRECTIONAL COMPOSITE MATERIAL

Robert Zemčík, Vladislav Laš and Petr Měšťánek

Keywords: composite material, delamination, experiment, finite elements, energy release rate

1. Introduction

One of the damage mechanisms in laminates is the origin and propagation of failure or crack between individual layers, so-called delamination. Delamination can be caused by imperfection during manufacturing process or by static or dynamic loads. Its existence in composite material degrades its stiffness and in certain cases it can degrade the stability to critical level.

The majority of works investigating the delamination on laminated structures use either the critical value of energy release rate G_c [4] or the stress intensity factor K_c [3] as the interlaminar fracture toughness. The early works were focused mainly on the experimental procedures and analytical solutions. These were later followed by works dealing with numerical simulations with the use of finite element analysis and works introducing new special element types for the modeling of delamination [2].

2. Experiment

Firstly, an experimental assessment of interlaminar toughness for Mode I delamination was carried out according to the ASTM standard [1]. The specimens were rectangular strips denoted as DCB (Double Cantilever Beam) as shown in Figure 1. The specimens were manufactured from 20 epoxy prepreps reinforced with continuous Toray T600SC carbon fibers. A non-adhesive aluminum foil, which served as the initial crack, was inserted in one half of the plate's midplane during manufacture.

The latter experiments were performed on specimens having slightly altered geometry. These were so-called Mixed Mode Flexure (MMF) specimens subjected to three-point bending as shown in Figure 2. The specimen is therefore subjected to a combination of Mode I and II.

The major problems identified during the experiments are the inhomogeneity of the pre-crack zone mentioned above and so-called fiber bridging between the two layers. The bridging of fibers occurs on composite specimens with single fiber orientation layers as in this case.

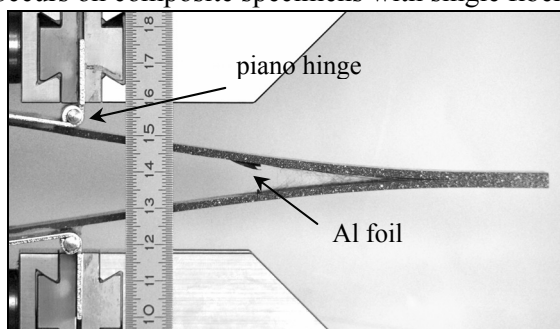


Figure 1. DCB specimen.

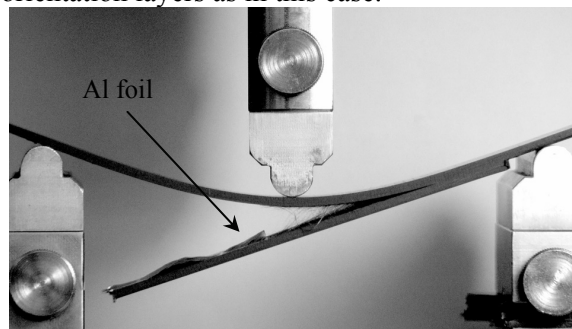


Figure 2. . MMF specimen.

3. Estimation of energy release rate and numerical simulation

The energy release rate G for the DCB specimen can be calculated a) from the definition, b) using Euler-Bernoulli theory of beams or c) using the corrected formula as suggested in ASTM standard for Mode I delamination:

$$\text{a) } G = -\frac{1}{b} \frac{dU}{da}, \quad \text{b) } G_I = \frac{12F^2 a^2}{b^2 h^3 E_{11}}, \quad \text{c) } G_I = \frac{3F\delta}{2b(a + |\Delta|)} \quad (1)$$

where dU is differential increase in strain energy, da is differential increase in delamination (or crack) length, F is force, b is specimen width, E_{11} is Young's modulus in the longitudinal direction (direction of fibers), h is half of the specimen thickness, and $|\Delta|$ is crack length correction [1].

The numerical simulation was performed using finite element method in MSC.Marc system. Herein, the analysis was solved as a plane strain problem using the equality between values of energy release rate and J-integral in the case of elastic analysis. The calculated values of G_{Ic} on DCB specimen are plotted in Figure 3 and an example of the reconstructed load-displacement curves are shown for the case of MMF specimen.

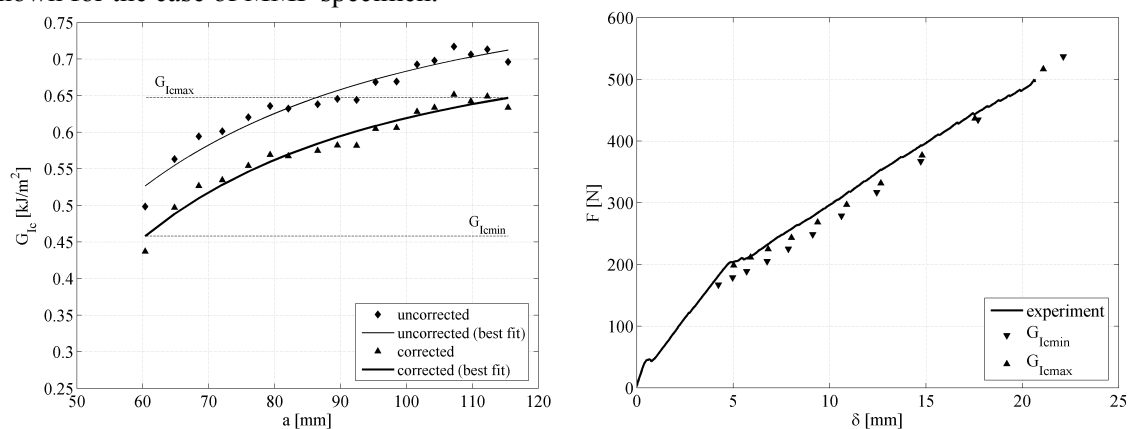


Figure 3. Calculated G_c values (left) and load-displacement curves on MMF specimen (right).

4. Conclusions

An experimental investigation of low-speed Mode I delamination was carried out on carbon fiber-reinforced epoxy specimens (Double Cantilever Beam) in order to estimate the critical energy release rate values G_{Ic} . The values were determined according to ASTM standard. Numerical simulation of the experiment was performed using FEA and the equality between G and J-integral in elastic case. The load-displacement curves were reconstructed for the above specimens as well as for the Mixed Mode Flexure specimens and compared with experimental data.

Acknowledgement

This work was supported by the research projects MSM 4977751303 and GA AV IAA200760611.

References

- [1] ASTM D5528-01, Standard Test Method for Mode I Interlaminar Fracture Toughness of Unidirectional Fiber-Reinforced Polymer Matrix Composites, Annual Book of ASTM Standard, 2002, pp. 249-260.
- [2] Borst R., Remmers J. J. C.: Computational modelling of delamination. Composites Science and Technology 66, 2006, pp. 713-722.
- [3] Chow W. T., Atluri S. N.: Stress intensity factors at the fracture parameters of delaminations crack growth in composite laminates. Composites Part B. 28B, 1997, pp. 375-384.
- [4] Pereira A. B., Morais A. B.: Mode I interlaminar fracture of carbon/epoxy multidirectional laminates. Composites Science and Technology 64, 2004, pp. 2261-2270.

Ing. Robert Zemčík, Ph.D. (corresponding author), zemcik@kme.zcu.cz, Doc. Ing. Vladislav Laš, CSc. and Petr Měšťánek, Department of Mechanics, Faculty of Applied Sciences, University of West Bohemia Univerzitní 8, 306 14, Plzeň, Czech Republic, tel.: +420 377 632 336, fax: +420 377 632 302

ACCOMPLISHMENT OF WATER TIGHTNESS AND VOLUME STABILITY OF CONCRETE SLABS

R. Valenta, J. Šejnoha, M. Šejnoha, T. Krejčí

Keywords: concrete structures, water tightness, volume stability, casting in layers, catalyst, coupled analysis, shrinkage, creep, damage model, heat and moisture transfer

In this paper the influence of casting procedure in layers together with application of H-Krystal on water tightness and volume stability of concrete structures of considerable thickness is investigated. Proposed procedures are verified by both the numerical and physical experiments.

The results of this research can be used for the design of foundation slabs and basically in all cases when designing all concrete slabs with large thickness. The similar methodology can be applied for diaphragm walls as well. These structures are often of large volumes (e.g., considerable thickness of the slab) that generate problems linked to the evolution of hydration heat and, subsequently, to the propagation of cracks. Reaching the volume stability results in significant reduction of crack evolution. The compact concrete thus provides protection for reinforcements, so that these structures can also be used in an aggressive environment.

The goal of this work is to propose a methodological approach ensuring the highest possible reduction of crack evolution and yielding watertight concretes. The main causes of crack evolution in an early stage of concrete are the hydration heat and shrinkage. The volume changes in concrete are responsible for tensile stresses leading eventually to formation of cracks [1].

To reach the water tightness and the volume stability of concrete, suitable casting procedures such as casting in layers and an application of H-Krystal in dilatation joints are produced, which allows us to control the hydration heat and the compact connection of layers is attained.

H-Krystal is the catalyst delaying evolution of hydration heat about by 7 hours and ensuring water tightness of concrete. It does so by taking advantage of the natural and inherent characteristics of concrete; concrete is both porous (capillary tract system) and chemical in nature. By means of diffusion, the reactive chemicals in H-Krystal use water as a migrating medium to enter and travel through the capillary tracts in the concrete. This process precipitates a chemical reaction between H-Krystal, moisture and the natural chemical by-products of cement hydration (calcium hydroxide, mineral salts, mineral oxides and unhydrated and partially hydrated cement particles). The result is crystallization and, ultimately, a non-soluble crystalline structure that plugs the pores and capillary tracts of the concrete. In this condition, the pores become discontinuous and the concrete is thereby rendered impenetrable by water and other liquids from any direction. In other words H-Krystal is a catalyst in sand-cementitious paste which can with the help of capillary water initiate the growth of crystals and provides the solid water tightness of concrete. Delaying of hydration and therefore delaying of the onset of hardening and consequently also shrinkage in the interface layers will cause non-uniform distribution of initial stresses in the concrete. This controlled non-uniform distribution of initial stresses may lead to prestressing of preselected concrete layers.

To verify both the basic assumptions and the numerical model of thermo-hydro-mechanical behavior of early age concrete the pilot physical experiment was prepared, where the thermo-hydro-mechanical properties of three concrete blocks were examined. The numerical analysis was performed using the SIFEL finite element code, which allows solving the associate problems of transport processes and the effects described by damage mechanics. To analyze the evolution and the

progress of concrete cracks it is essential to determine the temperature and moisture field history. Without stating the details the transport analysis is focused on a nonlinear coupled problem where transport laws, balance equations and material laws must be solved simultaneously. [2].

The temperature field obtained from the numerical analysis is consistent with the measured experimental results. The numerical analysis also confirmed the non-uniform distribution of stresses along the cross-section. According to the results of computational simulations, the shrinkage tensile stresses in surface layers almost do not occur and they are concentrated in internal layers. Compression zones are mainly situated in surface layers, which reduce the crack occurrence, see Fig. 1.. The amount of damage depends on tensile strength evolution and on the amount of reinforcement in the internal layers of the slab. The obtained experimental results satisfactory correspond to the analytical data.

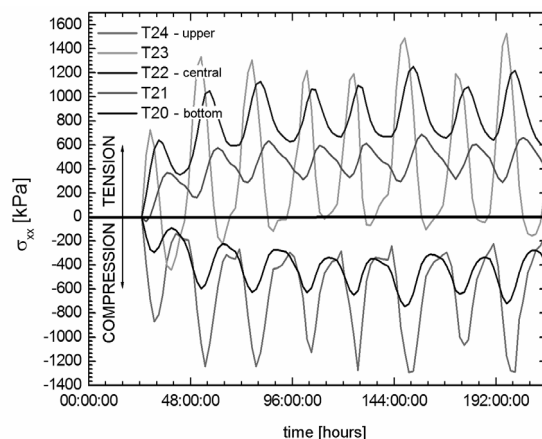


Figure 5. Graph of dependence of stress σ on time in the block K1 – simulation

Acknowledgement

This outcome has been achieved with the financial support of the Ministry of Education, Youth and Sports of the Czech Republic, project No. 1M6840770001, within activities of the CIDEAS research centre.

References

- [1] Mokarem, D. W., Meyerson, R. M., Weyers, R. E., Development of concrete shrinkage performance specifications, Final contract report, Virginia Transportation Research Council, Virginia, 2003.
- [2] Krejčí, T, Time – dependent behavior of concrete and other porous materials, CTU in Prague, Prague, Ph.D. Thesis, 2003.

Ing. Richard Valenta

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420224354606, Fax:+420224310775,
e-mail:richard.valenta@centrum.cz

Prof. Ing. Jiří Šejnoha, DrSc.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel: +420224354492, Fax:+420224310775,
e-mail:sejnoha@fsv.cvut.cz

Doc. Ing. Michal Šejnoha, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420224354494, Fax:+420224310775,
e-mail:sejnom@fsv.cvut.cz

Ing. Tomáš Krejčí, Ph.D.

Czech Technical University in Prague, Faculty of Civil Engineering, Department of Mechanics,
Thákurova 7, 166 29 Prague 6, Czech Republic, Tel:+420224354309, Fax:+420224310775,
e-mail: krejci@cml.fsv.cvut.cz

SHOT-PEENING TECHNIQUE FOR FATIGUE IMPROVEMENT OF HIGH STRENGTH STEEL GEARS

G. Olmi and A. Freddi

Keywords: Gear, Shot-peening, Residual stress, Experimental tests, Fatigue

1. Introduction

The object of this work was to evaluate the effect of shot-peening on the improvement of fatigue life [1] of high strength steel gears. Shot-peening treatments to be investigated were arranged in a factorial plan with two factors: peen diameter and Almen intensity. Residual stresses induced beneath the surface were experimentally evaluated. Then seven treatments were selected for fatigue testing and evaluation of dynamic strength. The effects of single and double peening were taken into account, while fractographic analyses made sure of proper bending fatigue failure.

2. Materials and Methods

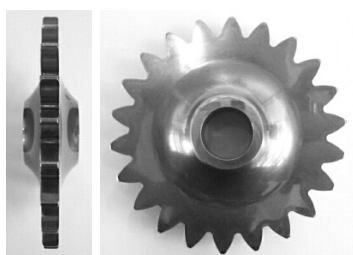


Figure 1. Plant and prospect views of one of the tested gears

During the development of this work eighteen gears were used to perform residual stress analyses and fatigue tests. Shot-peening treatments to be tested were arranged in a factorial plan with two factors consisting in the peen diameter varying from 0.3 mm (S110) to 0.8 mm (S330) and in Almen intensity varying on three levels corresponding to shot pressure from 2 bar to 4.5 bar. Both single and double peening treatments (additional peening process with small values for peen diameter (Z150, 0.15 mm) and Almen intensity (6/8A)) were considered. The aim of this double peening treatment was to reduce rugosity and to make it more homogeneous.

3. Residual stress distribution

Residual stresses were measured by the X-ray diffraction method and their values were estimated on the basis of the “ $\sin^2\Psi$ ” method with CrK α -radiation on the {211}-plane of the bcc phase (XStress3000, by Stresstech, Finland) [2].

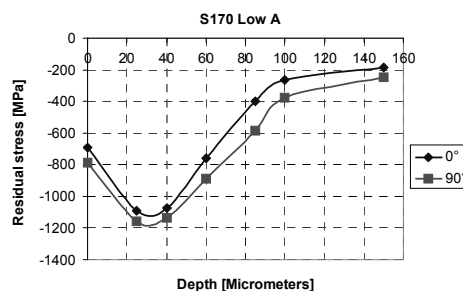
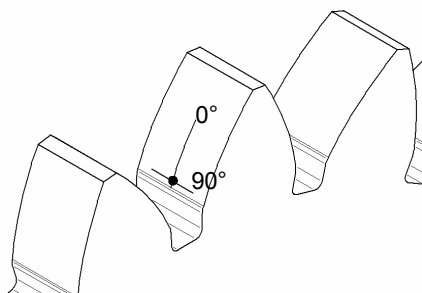


Figure 2. Evaluation of residual stress induced by shot-peening

The measurement of the residual stresses distribution was performed by progressive electrolytical removal of thin surface layers and subsequent X-ray measurements. With reference to a location at the tooth root residual stress distribution was investigated along two directions perpendicular and parallel to the gear axis. These directions are referenced as 0° and 90° directions (Fig. 2).

4. Experimental fatigue tests

A resonant machine (Rumul, Switzerland) was used for fatigue tests which were carried out at a load frequency of about 108 Hz. Results were then elaborated with the use of the Dixon Staircase method [3]. The aim of the tests was to estimate bending fatigue strength for selected treatments.

A particular installation was designed and constructed, aiming to subject just a couple of not adjacent teeth to a pulsating load. For each combination of parameters sets of 6 or more specimens (for specimen a couple of teeth under test must be intended) were tested in order to estimate the fatigue limit. The load levels were arranged according to [3]. The specimens were tested to failure or to 10 million cycles. Results are presented in Fig. 3.

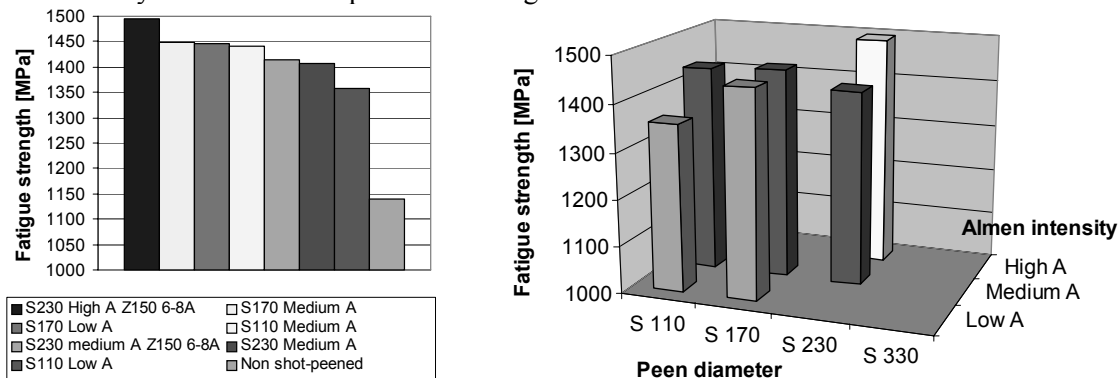


Figure 3. Fatigue strength for different treatments

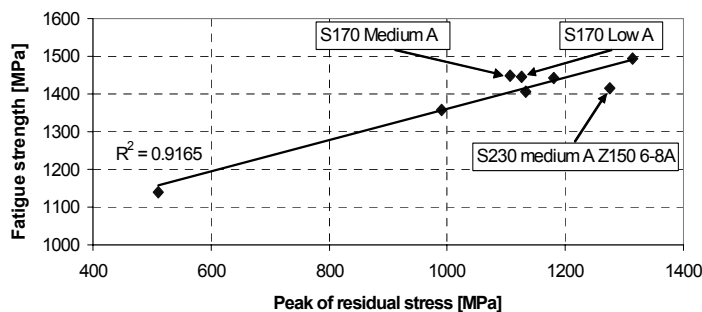


Figure 4. Relationship between residual stress and fatigue limit

well linearly interpolated, as confirmed by R^2 value, 0.92. Other relationships were investigated, by relating evaluated strengths to maximum stress at the failure section and to stress gradient.

Essential References

- [1] Croccolo D., Cristofolini L., Bandini M., Freddi A., "Fatigue strength of shot-peened nitrided steel", *Fatigue & Fracture of Engineering Materials & Structures*, Vol.25, No.7, 2002, pp. 695-707
- [2] Benedetti M., Fontanari V., Hohn B.R., Oster P., Tobie T., "Influence of shot peening on bending tooth fatigue limit of case hardened gears", *International Journal of Fatigue*, Vol.24, 2002, pp. 1127-1136
- [3] Dixon W.J., Massey F.J.Jr., "Introduction to Statistical Analysis", McGraw-Hill U.S., 1983

Giorgio Olmi, Fellow Researcher, and Alessandro Freddi, Full Professor
University of Bologna, DIEM Department, Viale Risorgimento, 2, Bologna, Italy, Telephone: +39 051 2093455, Telefax: +39 051 2093412, e – mail: giorgio.olmi@mail.ing.unibo.it, alessandro.freddi@unibo.it

FLEXIBLE MANUFACTURING TECHNOLOGY OF ASSEMBLY BASED ON COMPUTER-AIDED DESIGN SYSTEMS

Mozga N., Sudnieks F., Dreijja Z, Filipov A.

Keywords: flexible manufacturing systems, assembly, automation, batch fabrication.

1. Flexible Manufacturing Systems Of Assembly Is A Future

As during manufacturing of any complete product the technological operation of assembly always takes place and labour input of assembly works in some cases can reach 80 % from labour input of manufacturing of all product the automation of assembly processes and aspiration to complex automation is especially important problem.

At complex automation of assembly processes the highest technical and economic effect is reached at use of flexible manufacturing systems (FMS) of assembly. In turn the greatest effect from introduction FMS of assembly occurs only in conditions of multinomenclature manufacture when specialization is conducted not on one product, but on group or family of similar products. Practical value of group manufacture at creation FMS of assembly can be united in following substantive provisions:

- if the level of unification of products, technologies, the process equipment, program control systems is higher, then the creation of FMS is more easier and its operation is more effective;
- if the specialization of shops, sites, industrial modules within the limits of the nomenclature is more expanded, then the productivity of FMS is higher and, certainly, efficiency too. The nomenclature is limited to group of technologically similar details.
- if the result from introduction of FMS by groups of products is certain more particularly from the beginning of manufacturing details, then more purposefully it is possible to organize work on each site and at each stage, considering current and perspective requirements of manufacture.
- if the operating system of manufacture and system of planning more correspond to FMS requirements, then its level of accordance with products, materials, tools, programs is higher; and then as a result productivity also increases, and equipment downtime on the contrary decreases.

Therefore in case the organizational-technological structure mismatches principles of group manufacture then before creation FMS of assembly it is necessary to lead partial or full reorganization of the manufacturing firm.

2. Optimal CAD/CAE Decision For Assembly Manufacture

Recently the following tendency is observed: more and more than heads of the manufacturing firms, understanding, that that their manufacture has got greater "flexibility" and could react quickly to changing requirements of the market, make a decision on its reorganization and at once face other problem - a choice corresponding, suitable them, CAD/CAE systems. As competitiveness of the manufacturing firm substantially depends on a choice optimum CAD/CAE decisions and so in given article some essentially important problems which are necessary for solving to engineers and heads of the manufacturing firms at choice CAD/CAE of systems are considered.

Today in mechanical engineering there is a lot of positive technological changes. Constant perfection of CAD/CAE systems promotes the further progress in sphere of designing and development.

Essentially new equipment now can be created much more quickly, than ever earlier. Application of new methods leads to improvement of quality and reliability of products and, simultaneously, to decrease in charges on designing and manufacture. But, unfortunately the ideal CAD/CAE system equally suitable all companies, simply does not exist, since each manufacturer can have specific requirements. Therefore criteria of a choice for them can be a set of opportunities of system.

In given article opportunities of various systems in view of requirements of assembly manufacture are considered. It is possible to carry to such requirements:

- a system effectiveness in creation and editing of 3D-models - as details, and assemblies;
- opportunities of preparation of drawings from 3D-models;
- presence of additional opportunities of system, such as the structural analysis, modelling of dynamics, creation of programs for machine tools with CNC, etc.;
- presence of tools for management of data of the project (PDM);
- reliability of the software.

Conclusions

The maximum level of automation of assembly process is use of flexible manufacturing systems of assembly that allow to quickly react to changing requirements of the market, reconstructing the production equipment and technology on manufacturing of a new product. Flexible manufacturing system can be the "flexible system" and meet all requirements shown to flexible systems, only in the event that process of production will be organized on the basis of principles of batch fabrication, i.e. on manufacturing of the same details. Modelling of details of assembly, testing of their joining, drawings creations, modelling the kinematics and dynamics of the mechanism, designing of assembly procedure, preparing the programs for machine tools with CNC and many other things, i.e. projection the technological process of assembly from the beginning and up to the end it is possible by means of one of CAD/CAE systems.

References

- [1] Beljanin, P.N. Flexible manufacturing complexes. Mechanical engineering, Moscow, 1984 (in Russian).
- [2] Belousov, A.P. Bases of automation of manufacture in mechanical engineering. Higher school, Moscow, 1982 (in Russian).
- [3] Dachenko, F.I. Designing of automatic transfer lines. Higher school, Moscow, 1983 (in Russian).
- [4] Kosilova, A.G. Hand-book of the technologist on the transfer lines. Mechanical engineering, Moscow, 1982 (in Russian).
- [5] Majorov, S.F. Flexible fabrications. Mechanical engineering, Leningrad, 1983 (in Russian).
- [6] Melnikov, G.N. Designing of machine-assembly department. Mechanical engineering, Moscow, 1990 (in Russian).

Mozga Natalija, Dr.Sc.ing., lector

Riga Technical University, Mechanical Engineering Institute, Address: Ezermalas Str. 6, LV-1006, Riga, Latvia; Phone: +371 7089701, Fax: +371 7089701; e-mail: natalija.mozga@inbox.lv

Francis Sudnieks, Asoc.prof., Dr.Sc.ing., professor

Riga Technical University, Mechanical Engineering Institute, Address: Ezermalas Str. 6, LV-1006, Riga, Latvia; Phone: +371 7089701, Fax: +371 7089701; e-mail: arai@acad.latnet.lv

Dreija Zita, M.Sc.ing., student

Riga Technical University, Mechanical Engineering Institute, Address: Ezermalas Str. 6, LV-1006, Riga, Latvia; Phone: +371 7089701, Fax: +371 7089701; e-mail: zikina@one.lv

Filipov Aleksandr, Dr.Sc.ing., lector

Riga Technical University, Mechanical Engineering Institute, Address: Ezermalas Str. 6, LV-1006, Riga, Latvia; Phone: +371 7089701, Fax: +371 7089701; e-mail: arai@acad.latnet.lv

THE FACTOR ANALYSIS OF PRESTRESSING STEEL CABLES ONE COMPONENT OF PRESTRESSED CONCRETE BEAMS

T. Svaguša

Keywords: factor analysis, prestressed steel cable, beams, physical and mechanical properties, variables, factors, explorative analysis, confirmative analysis

1. Introduction

In the laboratories of Civil Engineering Institute of Croatia (IGH) tests are carried out in the fields of civil engineering area applying numerous testing methods.

This paper elaborates a part of the research where the factor analysis is preformed on one component of the prestressed concrete beam i.e. prestressing steel cables. Complete research included materials of reinforcing steel bars and concrete. The research embraced following mechanical and physiscal propreties of prestressing steel cables: S_0 , $F_{p0,1}$, $F_{p0,2}$, F_m , R_m , A_g , A_{500} , E and m/l .

2. Application of the factor analysis method

The factor anlysis method enables exploration of deeper relations between different properties of diverse materials. Observed measurable dimensions are only one form of appearance of some hidden dimension which is not directly measurable. By elaboration of measured results it is aspired to isolate one or more dimensions so called factors which may explain interrelated dependencies among measured dimensions. The factor analysis starts with calculation of the correlation matrix **R** and follows by its eigenvalues calculation. A diagram presenting eigenvalues is shown on the figure 1. Values of three eigenvalues are greater than one. It indicates that only 3 factors may be extracted.

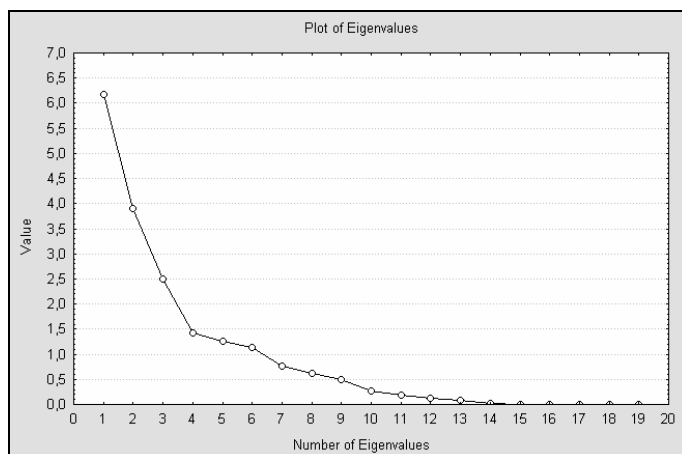


Figure 1. Eigenvalues presented by Scree-test

For factors extraction we use a principal components method and for improvement of results a method called varimax normalized rotation. On that basis three factors are extracted and corresponding factor loadings matrix **A** is created. Values of the factor loadings, factors and position of

corresponding variables are presented by 3D graph on the figure 2.

Further on for each variable we determine part of unit variance that these variable shears with common factors. Communalities h_i^2 for each variable is calculated as the sum total of belonging factor loadings to the power. Its value in our case is:

$$h_i^2 = a_{i1}^2 + a_{i2}^2 + a_{i3}^2 \quad (1)$$

Reduced correlation matrix $\mathbf{R}_h$ is calculated through multiplication of the factor loadings matrix $\mathbf{A}$ by its transposed matrix $\mathbf{A}'$ i.e. $\mathbf{R}_h = \mathbf{A} \cdot \mathbf{A}'$. Residual correlation matrix is the difference between correlation matrix and reduced correlation matrix $\mathbf{R}_{res} = \mathbf{R} - \mathbf{R}_h$. It shows how good are factor loadings and extracted factors. In this case extracted factors relatively good reproduce original correlation matrix. Finally, factor scores matrix $\mathbf{P}$ shows magnitude and type of influence of factors on examined variables.

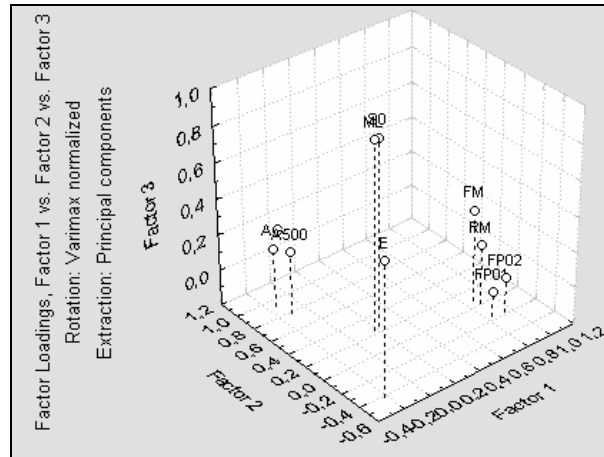


Figure 2. Factor loadings for factors 1, 2 and 3

The factor analysis discovered three factors. The first factor, named a Stress factor, consists of four variables with high factor loadings i.e. $F_{p0.1}$, $F_{p0.2}$, F_m , and R_m . The second is called a Strain factor and consists of two major variables with high factor loadings representing A_g and A_{500} . The third factor is named a Geometry and Elasticity factor since it consists of three variables: two with high factor loadings representing S_0 and m/l , and one with middle factor loadings representing E .

3. Conclusion

The factor analysis rendered new insight of the already known facts. The complete research comprised factor analysis of the reinforcing steel bars, prestressing steel cables and concrete simulating the case of prestressed concrete beam. Extracted factors comply with factors extracted through the independent analysis of each material. Thereby we conclude that with factor analysis from a bigger assemble of data we can extract factors that comprise related variables with great sense. Grouping of variables contributes to simplification of observed problem.

References

- [1] Fulgosi, A., "Factor analysis", Školska knjiga, Zagreb 1984
- [2] "Factor Analysis", Script for postgraduate students, FSB University of Zagreb, Zagreb 2001
- [3] Bosnar, R., Čala, I., Roketinec, I., "Application of Factor Analysis in Condition Based Maintenance", Maintenance and Exploitation 5 (3/4), 135-138, 2001
- [4] Živadinović, N. K., "Determination of Basic Properties of Products by Application of Factor Analysis", Economic review 55 (11/12), 952-966, 2004
- [5] Horvat, J., Katavić, S., Mikrut, M., "Segmentation of Domestic Continental Tourist Market Using Factor and Cluster Analysis", Tourism 50 (2), 067-176, 2002

Tomislav Svaguša, B.Sc.Mech.Eng.

Civil Engineering Institute of Croatia, Department for Steel Structures, J. Rakuše 1, Zagreb, Croatia, phone +385 1 6125 562, fax +385 1 6125 554, tomislav.svagusa@igh.hr

RESONANT SPECTRUM ANALYSIS OF SEISMIC AND WIND LOADING OF STRUCTURE

A. Mihanović, Ž. Nikolić and N. Živaljić

Keywords: Windows-wavelet, resonant spectra, wind, force spectra, average earthquake acceleration, fluctuating earthquake acceleration

1. Introduction

In this paper the resonant spectra of wind and earthquake action on the single degree of freedom system (SDOF) are shown. The earthquake load was taken from the literature [1], while the wind load was obtained from the measuring experiments [2], [3].

2. Resonant spectra of wind force action on SDOF

On the figure 1. the measured force of the wind with the resolution 0.002 seconds is shown.

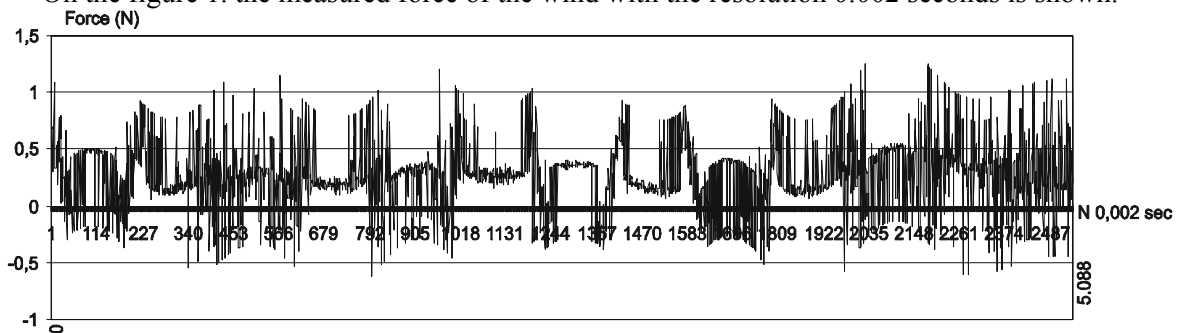


Figure 1. Measured wind force at the resolution of 0.002 sec

Figure 2. shows the result of averaging of wind force taken from the figure 1.

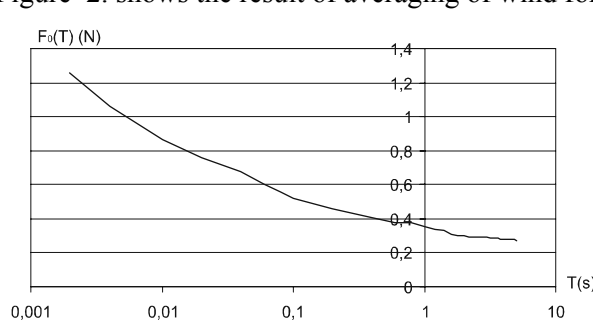


Figure 2. Result of averaging of wind force

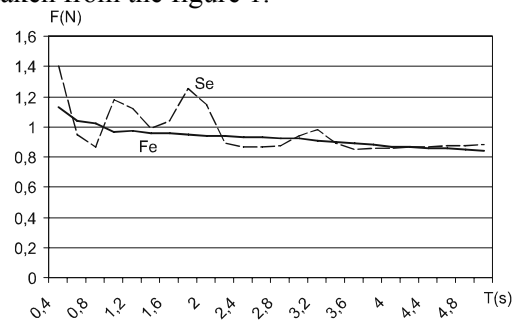


Figure 3. Resonant and classic elastic force spectrum

Figure 3 shows the classical spectrum S_e and the resonant spectrum F_e of the elastic force response for the given wind force load.

3. Resonant spectra of earthquake action

The input acceleration on SDOF for the constant component a_0 , the input acceleration for the fluc-

tuating component $a_{rs\alpha=1}$ and resonant spectra acceleration of SDOF mass RSa are shown in the figure 4.

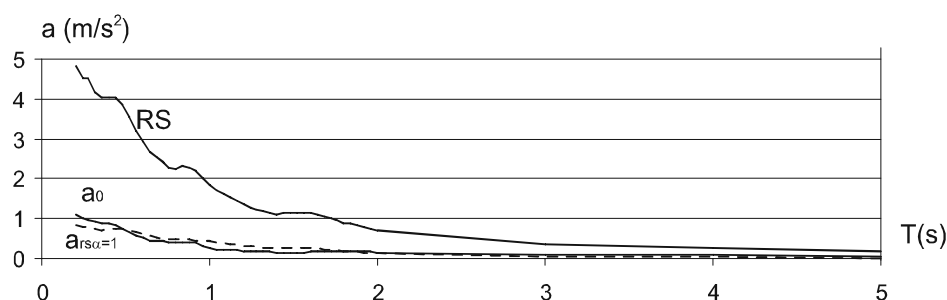


Figure 4. Resonant acceleration spectra for Northridge earthquake

4. Conclusions

The averaging interval is extremely important for the wind loading of the structure. A good approach is usage of the averaging interval equal to the duration of the period T of the analysed system. The significant increase of the intensity of the constant component by decreasing the period T of analysed system can be observed. The part of the fluctuating component presented as sinusoidal function of the duration of one wave is quantified. Its intensity is on an average one half of the intensity of the stable component. The total action, presented by the extreme elastic force, is quantified by the dynamic factor of the elastic force in relation to the force magnitude of the constant component. The value of thus defined dynamic factor is between 2.74 up to 3.26, on an average 3.0. The data in this paper are obtained for an example of a wind force record. The same conclusions are shown in reference [4] where the large number of the wind force records is analyzed.

The resonant spectra can be also applied on the earthquake action on SDOF. The constant and fluctuating component of the earthquake result from the analysis. The intensity of the constant and the fluctuating component is approximately equal. The overall action of both components gives a resonant spectrum of acceleration response. The dynamic factor obtained as a ratio of the response and the stable component action, has the value from 3 to 8, most often between 4 and 6.

The relation of the resonant spectrum and the classical acceleration spectrum for zero viscous damping shows that the ordinates of the classical spectrum are significantly higher, while for the period T from 0 to 2 seconds are about twice.

The comprehensions about the intensity of the action by several sinusoidal waves have the special importance in the conclusions for both actions. It's shown that the intensity of the waves with several wave lengths decreases inversely proportionate to the number of waves, i.e. duration time. This fact shows [5] that the resonant effect isn't characteristic of the analysed phenomenon.

References

- [1] [http://training.fema.gov/EMIWeb-EMI/Nonlinear Dynamics Time History Analysis](http://training.fema.gov/EMIWeb-EMI/Nonlinear%20Dynamics%20Time%20History%20Analysis).
- [2] Mihanović, A., Nikolić, Ž., Marović, P., "Waves Properties of Wind Actions on Structure". Intl. J. Engineering Modelling, (Submitted).
- [3] Mihanović, A., Nikolić, Ž., Živaljić, N., "Resonant Spectrum in Hazard Analysis of Structure Seismic Loading", NATO-ARW 981641: Extreme Man-Made and Natural Hazard in Dynamics of Structures, Opatija, May 27, 2006, pp. 1-7.
- [4] Mihanović, A., Gusić, G., "Resonant Spectra Obtained by Windowed-Wavelet Analysis", Proceedings of Int. Conf. COST TC Technology and culture as a basis for urban development, Ed. R. Žarnić, University of Ljubljana, 1995, pp 1-6.
- [5] Humar J.L., "Dynamics of Structures" Prentice Hall, New Jersey, 1990

Ante Mihanović, prof. dr. sc., Željana Nikolić, prof. dr. sc., Nikolina Živaljić, mr. sc.
University of Split, Faculty of Civil Engineering and Architecture, Matice hrvatske 15, HR-21000 Split,
CROATIA, tel: ++385/21/303-357, e-mail: ante.mihanovic@gradst.hr, web page: <http://www.gradst.hr>

IMPACT FATIGUE – RESEARCH FOR FURTHER DEVELOPMENT

Ion Dumitru, Nicolae Faur and Adrian Cipleu

Keywords: impact fatigue, durability, non-impact fatigue, cracks, Charpy specimens

1. Introduction

Many real elements from mechanical engineering are subjected to repeated shocks or, in some cases; these shocks can appear as additional loads. Repeated shocks define a fatigue phenomenon known under the name of Impact Fatigue. Because the strain rate changes the material characteristics it is to expect that the material properties at impact fatigue to be different in regard to those obtained at non-impact fatigue. Regardless to the fact the first studies on impact fatigue started earlier enough the extension of these line of research is very slow, no more than ten papers per year are published in this domain. The cause of this is the experimental difficulties and the lack of a unitary methodology regarding the impact fatigue durability approach.

2. Types of I.F. tests

The impact fatigue tests can be classified in:

- Direct impact tests that have the purpose of local degradation study under dynamic contact;
- Indirect impact tests when is taken under consideration the in volume degradations due to the wave shocks.

3. Particularities regarding the I.F. durability

The durability assessment parameter, in impact fatigue, is the one shock energy E , obtained from the mechanics of the load. In this case the durability plots are charted in $E-N$ coordinates. The purpose of drawing such diagrams is given by the research of relative impact toughness of diverse materials. Because the $E-N$ charts are directly depended of the equipment type, specimens type and restrains used the authors propose the use of Charpy specimens for impact fatigue durability assessment. In this purpose was realized an equipment for repeated impact tests that mimics the conditions of one shock bending test. The acceptance of the Charpy specimens as the probe type for impact fatigue tests gives in the first place the possibility for comparing from different point of view the behaviour of materials at impact fatigue in the presence of a sharp notch.

In the second place it can establish in which measure the durability at impact fatigue of a material is dependent or not of the energy of one IF, and in the third place the dynamic toughness parameters (K_{Id} and J_d) established on the same type of specimens can be correlated with the fatigue strength at multiple impacts.

Experimental studies, done on this machine, had allowed a comparative analysis of material behaviour at multiple impacts in the $10^3 - 10^6$ cycle's interval, the correlation equation was:

$$N \cdot E^m = D_0 \quad (1)$$

Where: n and D_0 are material constants.

The durability assessment at impact fatigue, of the Charpy specimens, based only on $E-N$ charts is not always enough. Starting from this aspect the authors had proposed, as assessment parameter

of the impact fatigue durability the rapport η defined as:

$$\eta = \frac{E_t}{E_d} \quad (2)$$

Where: E_d – the energy of one impact fracture of the Charpy probes;

E_t – total energy at impact fatigue test with an equal test energy:

$$E_t = N_f \cdot E \quad (3)$$

Where: N_f – number of impacts until fracture.

The drawing of $\eta=f(E)$ charts for different steels in diverse treatment states had allowed a more rational selection of the materials regarding the shock absorbent capacity of the strain energy at impact fatigue function of the impact energy.

4. I.F. cracks initiation and propagation

Was also proved that the crack velocity, at impact fatigue, can be correlated with the energy of one load through an equation of the following form:

$$\frac{da}{dN} = C_0 \cdot E^b \cdot a^c \quad (4)$$

The same researches, on different steels, had showed the existence of a relation between the energy of one shock fracture and the cumulative energy of the repeated impacts.

Another problem is in which way the impact fatigue tests brings new elements in regard to the non-impact fatigue tests.

If the comparison of the durability charts, in σ -N coordinates, of the smooth specimens doesn't allow making some certain conclusions, the tests on notched specimens clearly shows the reduction tendency of the impact fatigue durability in regard to non-impact fatigue one. Even if only a few papers approach the problem of crack initiation, a clear tendency is evident and that is the increased yield area near the notch tip at non-impact fatigue tests.

Many researches showed that the Paris equation can be successfully applied far the analysis of crack propagation at impact fatigue tests. Here was clearly proved the inequality $(da/dN)_{\text{impact fatigue}} > (da/dN)_{\text{non-impact fatigue}}$ for the same ΔK .

The experimental research done by the authors revealed the same conclusion, for steel used for mining chains. The IF tests were done on dynamic traction equipment using a hammer droved by a rotary disc.

5. Conclusions

By analyzing, in general, the impact fatigue durability can be draw the conclusion that this tests are more complete that the non-impact fatigue ones. This can be explain through the fact that in the domain of small cyclic numbers until fracture, the impact fatigue tests presents the shock behavior of the material and in the high cyclic number domain it's shows the ordinary fatigue behavior.

One of the principal problems in the impact fatigue durability assessment, still unsolved, is the taking under account the elastic wave's propagation in specimens.

In the end can be consider that the IF continues to be a domain with great development perspectives in the future, especially with the progress's done in Dynamic Fracture Mechanics.

Ion Dumitru, Professor

University "Politehnica" of Timisoara, Strength of Material Department, Bv. M. Viteazu no. 1, 300222, +40/256/403577, ion.dumitru@mec.utt.ro

Nicolae Faur, Professor

University "Politehnica" of Timisoara, Strength of Material Department, Bv. M. Viteazu no. 1, 300222, +40/256/403523, faur@mec.utt.ro

Adrian Cipleu, PhD

University "Politehnica" of Timisoara, Transport Engineering Department, Bv. M. Viteazu no. 1, 300222, +40/256/404301, acipleu@mpt.utt.ro

A COMPARISON OF MEASURED DATA, CODE AND FLUID-STRUCTURE INTERACTION SIMULATION OF THE WIND FLOW AROUND SOLAR PANELS

Lovre Krstulović-Opara and Ivan Perasović

Keywords: fluid-structure interaction, EUROCODE 1, coupled problem, wind flow

ABSTRACT

Within the initiative concerned with the sustainable energy systems (www.stens.fesb.hr), the 30kW electric power plant based on the grid-connected photovoltaic panels (i.e. solar panels) is planned to be constructed on the campus building. The presented paper is concerned with determining the wind load on the solar panels where specific requirements have to be met. Main questions pinpointed herein are: what wind velocities should be taken as the basis in calculus, and do the national code [1] or still undeclared Eurocode 1 [2], bring realistic results. Results based on codes [1,2], together with on-site measurements, are compared with the numerical model (Fig. 1) where fluid-structure response is used to simulate realistic conditions and to estimate the wind force coefficient for this specific case of solar panel. Over dimensioning the panel carrier frame results in too heavy structure. Such a structure is acceptable to be mounted on the roof that was not constructed to carry the additional equipment. The FEM FSI model (Fig. 2, 3) enabled designing carrier structure significantly lighter than it will result from classical calculations based on the code.

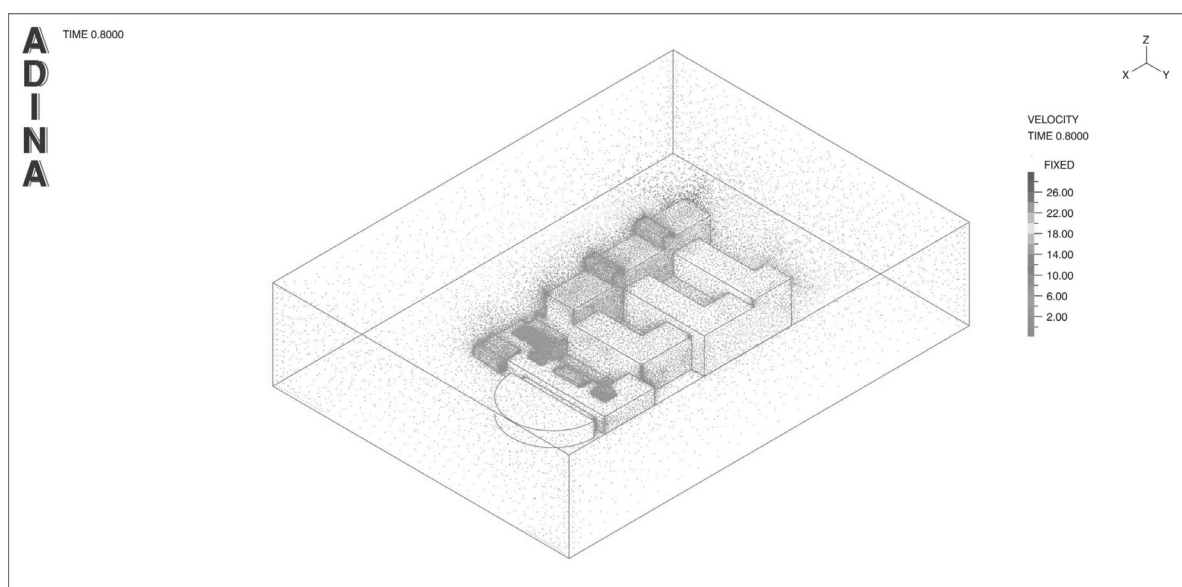


Figure 1. The tridimensional model of the wind flow around the campus building

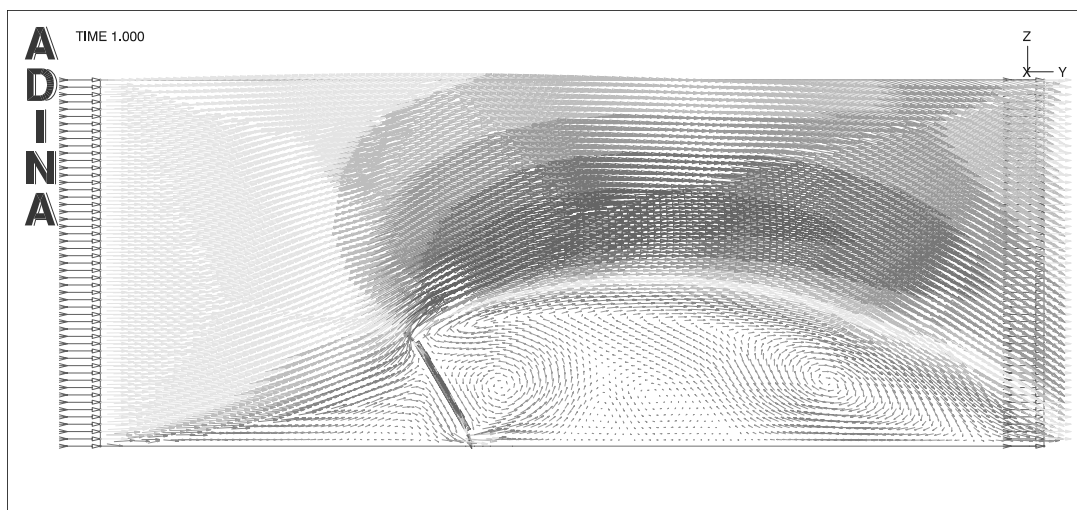


Figure 2. The two-dimensional model of the wind flow for the concept A

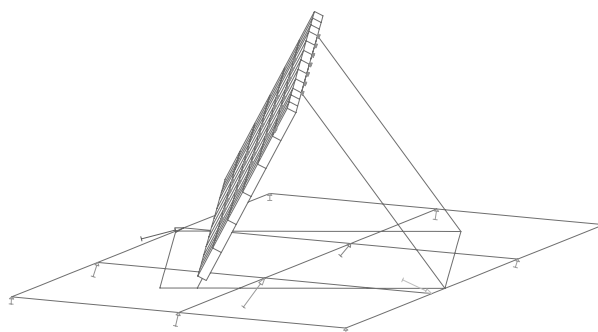


Figure 3. Reactions of the carrier frame loaded by the wind flow (FSI)

From the presented research it can be concluded that, as the wind flow on a panel with negative inclination is not described by the code, the code is not giving representative results. Therefore a force coefficient for this case has to be modified by the force coefficients evaluated in presented paper.

When comparing results obtained by the Eurocode 1 [2] to the still official national code [1], the Eurocode 1 can be recommended.

Considering representative wind velocity, it can be concluded that the Eurocod 1 is giving satisfying assumptions.

Acknowledgement

The financial support from the Ministry of Science and Technology project No.0023023: "Fatigue strength of machines and constructions" is gratefully acknowledged.

References

- [1] Furundžić, B., "Zbirka tehničkih propisa u građevinarstvu", Građevinska knjiga Beograd, 1977.
- [2] EN 1991-1-4 General actions - Wind actions (EUROCODE 1), 1991.
- [3] Perasočić, I. "Proračun nosača fotonaponskih panela na krovu FESB-a", diplom thesis, Faculty of Electr. Eng, Mech. Eng. and Naval Arch., University of Split, 2006.

Corresponding author:

Dr.-Ing. Lovre Krstulović-Opara

Faculty of Electrical Engineering, Mechanical Engineering and Naval Architecture / University of Split, R.Boškovića b.b., 21000 Split, Croatia, Phone: +385/21/305891, Fax: +385/21/463877

E-mail: Lovre.Krstulovic-Opara@fesb.hr

www.fesb.hr/kk

TIME- vs. FREQUENCY-DOMAIN ANALYSIS OF SINGLE/MULTIPLE-BOX GIRDER BRIDGE DECK SECTIONS

Bartoli G., Borri C., Costa C. and Mannini C.

Keywords: Bridges, Aeroelasticity, Wind Tunnel Tests, Flutter Derivatives, Indicinal Functions

1. Introduction

Flutter instability is a phenomenon due to fluid-structure interaction and it is one of the problems of major concern in long-span bridge design since it can lead to the collapse of the structure. At the present state of the art wind-tunnel tests are the only reliable tool to address flutter and other aeroelastic phenomena. Consequently, a vast experimental campaign was conducted at the CRIACIV (Inter-University Research Center on Building Aerodynamics and Wind Engineering) Boundary Layer Wind Tunnel, Italy (<http://windlab.ing.unifi.it>), shown in Figure 1a, on two section models: (i) a single-box trapezoidal girder with lateral cantilevers [2, 4], because of its common geometry for medium-span cable-stayed and suspension bridges and (ii) a series of multiple-box girder decks (Figure 1b), very similar to the one proposed for Messina Strait Bridge [1, 2], which is a very appealing innovative solution to design large-span bridges.



Figure 1. CRIACIV Boundary Layer Wind Tunnel, Italy (a); picture of one of the multiple-box girder deck section models tested in wind tunnel (b)

Static and aeroelastic tests are performed with the set-ups described in detail in Ref. [4]. Flutter derivatives (e.g. [5]) are measured via free-vibration technique [4, 5]. Particular attention is paid to the definition of the dependence of these aeroelastic coefficients on motion amplitude, mean angle of attack and geometric features such as the distance between the boxes and the presence of non-structural details and grid porosity.

Flutter derivatives can be used directly to calculate the critical wind speed in frequency domain through an eigenvalue analysis. Alternatively, indicial functions can be identified from flutter derivatives following the method outlined in Ref. [3] and time-domain simulations can be performed in order to calculate the pre-critical behaviour of the system as well as the flutter boundaries. Time-domain self-excited forces can be easily implemented in Finite-Element models and allow to account for structural nonlinearities and eventual presence of concentrated damping devices. They

also represent a powerful tool to calculate the response of the bridge under turbulent wind. The measured heaving and pitching free-decaying motion for different wind speeds is compared with the time-domain simulations performed via indicial functions (Figure 2) and good agreement is generally found except in one case, due to the local poor approximation of some flutter derivatives. The critical flutter wind speed and frequency are accurately reproduced both in frequency domain by means of flutter derivatives and in time domain with indicial functions. Conversely, as expected, the quasi-steady approach is found to overestimate significantly the critical condition.

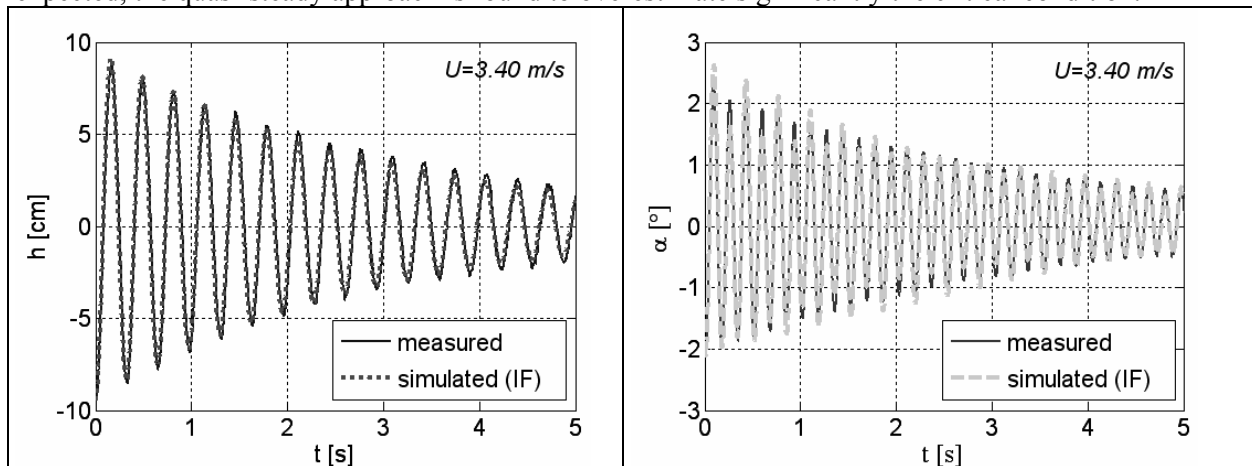


Figure 2. Measured heaving (h) and pitching (α) motions vs. indicial function simulations

Acknowledgement

The work has partially been supported by MIUR (Italian Ministry of University and Research) as a part of a National Research Program and within an International Ph. D. Course on “Risk Management on the Built Environment” (Università di Firenze – TU Braunschweig). The generous financial support of the Ente Cassa di Risparmio di Firenze (Research project on: “CRIACIV: a Center of Excellence in Research”) is also gratefully acknowledged.

References

- [1] Bartoli, G., D’Asdia, P., Febo, S., Mannini, C., Pastò, S. and Procino, L., “Analisi di sensibilità aeroelastica nella progettazione di ponti sospesi di grande luce”, Proceedings IN-VENTO 2006, Pescara, Italy, 2006
- [2] Bartoli, G., Ricciardelli, F., Saitta, A. and Sepe, V., “Performance of wind exposed structures. Results of the PERBACCO project”, Firenze University Press, Florence, Italy, 2006
- [3] Costa, C. and Borri C., “Application of indicial functions in bridge deck aeroelasticity”, J. Wind Eng. Ind. Aerodyn., 2006 (in press)
- [4] Mannini, C., “Flutter vulnerability assessment of flexible bridges”, Ph. D. Thesis, University of Florence, Italy - TU Braunschweig, Germany, 2006
- [5] Scanlan, R.H. and Tomko, J.J., “Airfoil and bridge deck flutter derivatives”, J. Eng. Mech. Div., ASCE, Vol. 97, 1971, pp 1717-1737

Gianni Bartoli, Associate Professor

CRIACIV-University of Florence, Dept. of Civil Engineering, Via S. Marta 3, 50139 Florence, Italy,
Telephone +39.055.4796218, Telefax +39.055.4796230, e-mail gbartoli@dicea.unifi.it

Claudio Borri, Full Professor

CRIACIV-University of Florence, Dept. of Civil Engineering, Via S. Marta 3, 50139 Florence, Italy,
Telephone +39.055.4796217, Telefax +39.055.4796230, e-mail cborri@dicea.unifi.it

Carlotta Costa, Ph. D., Post Doct. Research Fellow

University of Florence, Dept. of Civil Engineering, Via S. Marta 3, 50139 Florence, Italy,
Telephone +39.055.4796306, Telefax +39.055.4796230, e-mail carlotta@dicea.unifi.it

Claudio Mannini, Ph. D., Post Doct. Research Fellow

University of Florence, Dept. of Civil Engineering, Via S. Marta 3, 50139 Florence, Italy,
Telephone +39.055.4796306, Telefax +39.055.4796230, e-mail claudio.mannini@dicea.unifi.it

HYBRID BEAM ELEMENT FOR STABILITY ANALYSIS OF SEMI-RIGID FRAMES

Goran Turkalj, Goran Vizentin and Josip Brnić

Keywords: Stability, semi-rigid connections, semi-rigid frames, hybrid beam element, FEM

1. Introduction

The paper presents a numerical stability analysis of semi-rigid frames using finite element model by analyzing the influence of semi-rigid connections on the elastic buckling load value of frame structure.

In this paper a hybrid two node beam element is used. Semi-rigid connections effects are taken in account by means of the initial connection stiffness S_i for each DOF of the beam element, thus including translational connection stiffness as well as rotational. Hybrid beam element stiffness matrices are obtained by modifying conventional beam element stiffness matrices with an appropriate transformation matrix also described in this paper.

Stability problems are treated as a matrix eigenvalue problem, i.e. an initial stability analysis is performed. Equilibrium equations of a buckled beam member are derived applying linearized virtual work principle. Material is assumed to be linear elastic.

2. Hybrid S-R finite beam element

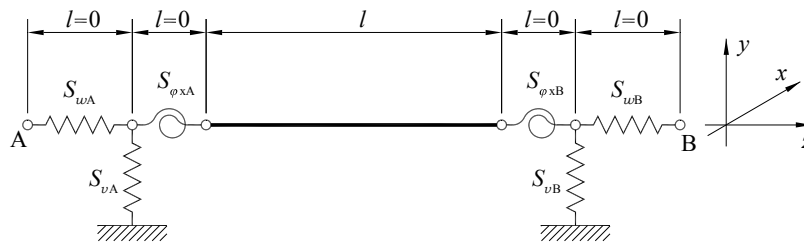


Figure 1. S-R finite beam element

Figure 1 shows the hybrid S-R finite element composed of a conventional straight beam finite element and added springs. The S-R finite element stiffness matrices can be obtained by modifying the appropriate conventional beam element stiffness matrices.

$$\mathbf{f}_{sr}^e = \mathbf{k}_{T, sr}^e \mathbf{u}_{sr}^e \quad (1)$$

$$\mathbf{f}^e = (\mathbf{T}_{sr}^e)^{-1} \mathbf{k}_T^e \mathbf{u}_{sr}^e \quad (2)$$

$$\mathbf{k}_{T, sr}^e = (\mathbf{T}_{sr}^e)^{-1} \mathbf{k}_T^e; \quad \mathbf{T}_{sr}^e = \mathbf{I} + \mathbf{k}_T^e (\mathbf{S}^e)^{-1} \quad (3)$$

where: $\mathbf{I}$ - identity matrix, $(\mathbf{T}_{sr}^e)^{-1}$ - transformation matrix for the S-R finite element, $\mathbf{S}^e$ - connection stiffness matrix, $\mathbf{k}_T^e$ - conventional finite element tangent stiffness matrix.

3. Example

A planar frame with pinned connections composed of three members is shown in Figure 2a. Each member length $l = 100$ cm. Cross-section characteristics are: $A = 100$ cm², $I_x = I_y = 833.3$ cm⁴, $J = 1000$ cm⁴.

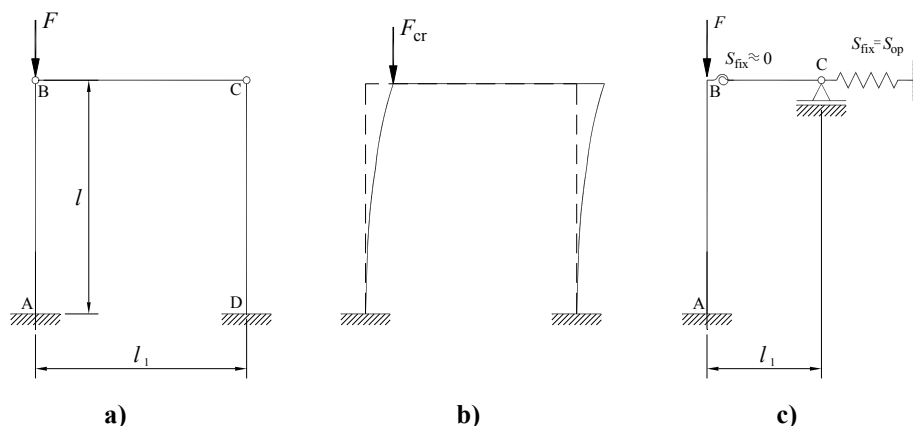


Figure 2. Portal frame with pinned connections: a) frame geometry; b) sway mode; c) frame model

The analysis was performed for sway buckling mode, Figure 2b, using the substituting model shown in Figure 2c, discretized by 1, 2, 4 and 8 elements per member. The right frame column influence is replaced by a linear spring of the stiffness $S_{sp} = 3EI_x/l^3$. Pinned connection B is modelled by flexural spring of negligible initial stiffness. Analysis results are shown in Table 1.

Table 1. Analysis results, F_{cr} (kN)

Analytical solution	Numerical analysis (EIGEN-SR)		Error, %
8060	1 el./memb.	8216	1,90
	2 el./memb.	8083	0,28
	4 el./memb.	8079	0,24
	8 el./memb.	8078	0,22

References

- [1] Yu, C. H., Shanmugam, N. E.: "Stability of Semi-Rigid Space Frames", Computers & Structures, Vol.28, No.1, 1988, pp 85-91.
- [2] Shakourzadeh, H., Gou, Y.Q., Batoz, J.L.: "Modelling of Connections in the Analyses of Thin-Walled Space Frames", Computers & Structures, Vol.71, No.4, 1999, pp 423-433.
- [3] Blandford, G.E.: "Stability Analysis of Flexibly Connected Thin-Walled Space Frames", Computer & Structures, Vol.53, No.4, 1994, pp 839-847.
- [4] Turkalj, G., Brnic, J., Prpic-Orsic, J.: "Large Rotation Analysis of Elastic Thin-Walled Beam-Type Structures using ESA Approach", Computers & Structures, Vol.81, No.18-19, 2003, pp 1851-1864.

Goran Turkalj, Ph.D., Assoc. Professor

Faculty of Engineering/University of Rijeka, Department of Engineering Mechanics, Vukovarska 58, HR-51000 Rijeka, Croatia, ++ 385 51 651 499, Goran.Turkalj@riteh.hr

Goran Vizentin, M.Sc., Junior Researcher

Faculty of Engineering/University of Rijeka, Department of Engineering Mechanics, Vukovarska 58, HR-51000 Rijeka, Croatia, ++ 385 51 651 503, Goran.Vizentin@riteh.hr

Josip Brnić, Ph.D., Professor

Faculty of Engineering/University of Rijeka, Department of Engineering Mechanics, Vukovarska 58, HR-51000 Rijeka, Croatia, ++ 385 51 651 491, Josip.Brnic@riteh.hr

DESIGN OF A DROP-WEIGHT MACHINE FOR COMPOSITE MATERIALS IMPACT TESTING

Giangiacoimo Minak and Daniele Ghelli

Keywords: CFRP, Low velocity impact, Drop-weight, Laser velocity measurement

1. Introduction

Composite materials, such as Carbon Fibre Reinforced Polymers (CFRP), are widely used in structural members when high stiffness, high strength and small mass are required. On the other hand, application of these materials is limited by their poor resistance to concentrated loads, particularly those produced by a foreign object impact [1, 2, 3, 4]. Since impact events are quite common during service life of mechanical parts and cannot be avoided, while the various failure modes of composites are not yet easily predictable, further research appears necessary to improve understanding of involved phenomena.

Suitable rigs are needed to reproduce impacts in controlled conditions, with different properties and operating principles depending on velocity and mass of the impactor [3]. This article describes the design, construction and set up of a drop-weight impact tester, which can simulate low velocity strikes (up to 8 m/s) on several types of CFRP specimens. An example of impact test on a CFRP laminate is also presented.

2. Drop-weight machine design

As can be seen in the drawing in figure 1, two vertical rods act as rails for the impactor, that consists of an aluminium alloy frame. Impactor drop is controlled with an electromagnet switched by the control unit of the machine. A thick steel plate holds the specimen and its clamping fixture. The appropriate equipment for laminated plates, which simulates clamp conditions along a circular boundary, has already been constructed; however other types of fixtures, specifically designed for different specimens, can also be mounted.

The machine is equipped with a piezoelectric load cell and a laser device for velocity measurement. The load cell is attached to the impactor in central position, so that its hemispherical head hits the target at the centre and contact force is measured as a function of time. The laser device is placed on the clamping fixture, and provides the actual value of impactor velocity just before collision. Four acoustic emission sensors can also be positioned on the laminate.

An electromagnetic braking system has been designed to prevent multiple impacts on the specimen when falling device rebounds after first strike. Two bars are lifted by electromagnetic devices after first contact ends, so impactor falls on them instead of hitting the target again. The control unit activates brakes some time after receiving an electric signal from a proximity sensor, which commutates when the impactor passes by it during drop. Maximum drop height is 3,20 m, corresponding to an impact velocity of 7,9 m/s approximately; overall impactor mass is about 1,25 kg, so in this configuration kinetic energy can reach 39 J, a typical value for low velocity impacts on

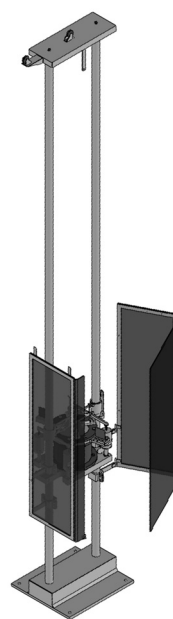


Figure 1. Drop-weight device

aircraft panels [5]. The braking system is able to stop impactor when rebound height is less than 400 mm. According to a static finite element analysis, the aluminium frame can withstand a contact force of 10 kN, while the maximum force admissible for the load cell is 22 kN. Impactor mass could be increased, provided operating limits of frame, load cell and braking system are not exceeded.

3. Drop-weight impact test

The procedure of an impact test is here described: the specimen is clamped into the grips and the acoustic emission sensors are placed on it while the mechanical safety system protects the operator; the laser device, the load cell and the magnets are turned on, the impactor is lifted by means of a manual device up to the desired height, corresponding to a prefixed impact energy, then the braking system is armed; the protective door is closed and in this way the safety system is disabled; the operator, by means of the PC, starts the acquisition and simultaneously turns off the electro-magnet holding the impactor; the impactor passes through the laser beam, then near the proximity sensor and finally hits the specimen; the brakes are released and the impactor is either pushed up or simply stopped if rebounding. Typical results are shown in figures 2 and 3 where load is plotted re-

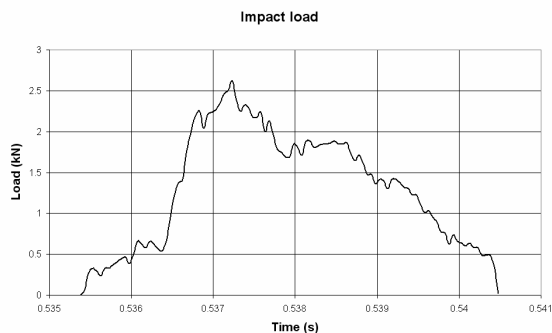


Figure 2. Impact load as a function of time

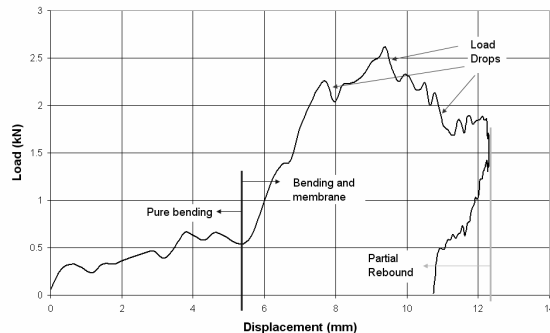


Figure 3. Load-displacement curve

spectively in function of time and displacement (obtained from impact load data by double integration of the Newton law). In figure 3 it is possible distinguish different slopes between flexural and membrane behaviour, three load drops due to laminate failures (delamination and fibre breaking) and the partial rebound due to the incomplete penetration into the laminate.

Acknowledgements

The authors thank by Dott. Federico Ricavi for his precious contribution in the design of the drop-weight machine. We are also grateful to Prof. Francesco Cesari for his support and revision of the manuscript.

References

- [1] Abrate, S., "Impact on laminated composite materials", Applied Mechanics Reviews, Vol. 44, No. 4., 1991, pp 155-190.
- [2] Abrate, S., "Impact on laminated composites: recent advances", Applied Mechanics Reviews, Vol. 47, No. 11, 1994, pp 517-544.
- [3] Abrate, S., "Impact on composite structures", Cambridge University Press, Cambridge, 1998.
- [4] Cantwell, W.J., Morton, J., "The impact resistance of composite materials: a review", Composites, Vol. 22, No. 5, 1991, pp 347-362.
- [5] Hosur, M.V., Murty, C.R.L., Ramamurthy, T.S., Shet, A., "Estimation of impact-induced damage of CFRP laminates through ultrasonic imaging", NDT&E International, Vol. 31, No. 5, 1998, pp 359-374.

Giorgio Minak, PhD

Alma Mater Studiorum Università di Bologna, DIEM, viale Risorgimento 2, 40136 Bologna, Italia,
+390512093266, +390512093412, giorgio.minak@unibo.it

Daniele Ghelli, MSc

Alma Mater Studiorum Università di Bologna, DIEM, viale Risorgimento 2, 40136 Bologna, Italia,
+390512093457, +390512093412, daniele.ghelli@mail.ing.unibo.it

ANALYTICAL AND NUMERICAL INVESTIGATION OF CRACK OPENING IN STRAIN-HARDENING MATERIAL

D. Pustačić and M. Lovrenić

Keywords: isotropic strain-hardening material, magnitude of plastic zone r_p , crack tip opening displacement (CTOD) δ_t , crack centre opening displacement (CCOD) δ_c , J -integral

1. Introduction

Exactly determination of the displacement and stress fields around a crack tip is an exceptionally complex task. In order to achieve a goal the different models (Barenblatt, Dugdale, Bilby, Cottrell, Svinden, Rice, Wnuk and many others) have been proposed. All those models have common property a complex physical picture of a real stress and strain state around crack tip they have tried to describe by the relative simple yield models. One of such assumption is an assumption about *constant cohesive stresses* within a plastic zone, or an assumption about *elastic-perfectly plastic material* of a solid body, respectively. Since the assumption incorporated in the Dugdale strip yield model is very rudely approximation of real elastic-plastic state around crack tip, in the present investigations we have assumed a *non-linear distribution of cohesive stress* within a yield zone. In order to determine a distribution law of those stresses by analytical way it would be necessary to carry out a certain incremental-iterative procedure while the equilibrium conditions are not satisfied. It is complex and required procedure. In order to obtain a distribution of the cohesive stresses we used the finite element method (FEM) and the commercial code ABAQUS.

2. Numerical modelling of the cohesive stresses in a yield zone

In order to determine the distribution of the cohesive stresses within a yield zone, a plate quarter was modelled, according to Fig.1. A plate quarter is divided all together on the 2381 plane finite elements. Those are isoparametric quadratic quadrilateral finite elements with eight nodes per each element. With the degenerated elements it is possible to describe a singularity of the type $1/\sqrt{r}$ at the crack tip [1].

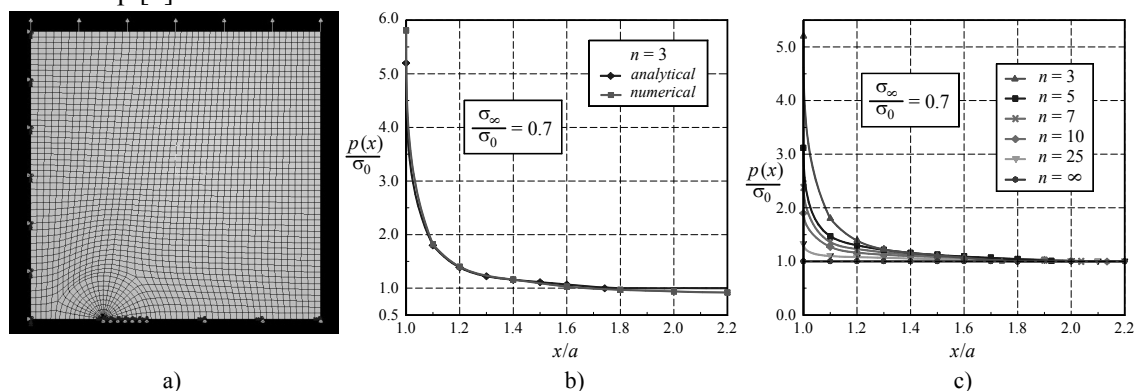


Figure 1. a) Discretization of a plate quarter on the finite elements and the boundary conditions, b) distribution of cohesive stress in a yield zone for $n = 3$, c) distribution of cohesive stress in a yield zone obtained on a base of analytical expression (1)

3. Analytical description of the cohesive stresses distribution

A distribution of the cohesive stresses obtained using the finite element method, Fig. 1.b) was tried to approximate by some analytical expression, i.e. by some function. It was shown that the best approximation is achieved by the analytical expression

$$p(x) = \sigma_0 \left(\frac{r_p}{x-a} \right)^{1/(n+1)}. \quad (1)$$

4. Determination of plastic zone magnitude r_p around a crack tip

The plastic zone length r_p in front of the crack tip, normalized to the initial crack length a is defined with the analytical expression

$$\frac{r_p}{a} = \frac{\pi \left(\frac{\sigma_\infty}{\sigma_0} \right)^2}{1 - \frac{\pi \left(\frac{\sigma_\infty}{\sigma_0} \right)^2}{\left[\Gamma \left(\frac{1}{2} + \frac{n}{n+1} \right) / \Gamma \left(\frac{n}{n+1} \right) \right]^2}} \cdot \frac{\left[\Gamma \left(\frac{1}{2} + \frac{n}{n+1} \right) / \Gamma \left(\frac{n}{n+1} \right) \right]^2}{\left[\Gamma \left(\frac{1}{2} + \frac{n}{n+1} \right) / \Gamma \left(\frac{n}{n+1} \right) \right]^2}. \quad (2)$$

5. Crack tip opening displacement (CTOD)

The crack opening displacement caused by cohesive stresses, distributed according to the expression (1), and under the small scale yielding condition will be

$$\delta_{\text{coh}}(a) = \sqrt{\frac{8}{\pi}} \frac{\sigma_0}{E} \frac{n+1}{n} \sqrt{b^2 - a^2} \cdot \sqrt{\frac{r_p}{b}} \cdot \frac{\Gamma \left(\frac{n}{n+1} \right)}{\Gamma \left(\frac{1}{2} + \frac{n}{n+1} \right)}. \quad (3)$$

6. Conclusion

It was indicated that the complex problems of elastic-plastic fracture mechanics in a case of isotropic strain-hardening of material could very successfully be solved by using the combination of the analytical and numerical methods. How much is in this matter an accuracy and a reliability of the obtained results? Such a high, how much the numerical results obtained by commercial package ABAQUS are accurate and how much accurate those results are approximated by the analytical expression (1), or by a certain similar expression.

References

- [1] ABAQUS Technological Brief, "Fracture Mechanics of Compact Tension Specimen Using ABAQUS/CAE", 2004, http://www.abaqus.com/solutions/tb_pdf/TB-04-FMCAE-1.pdf.
- [2] Chen, X. G., Wu, X. R. and Yan, M. G., "Dugdale Model for Strain Hardening Materials", Engineering Fracture Mechanics, vol. **41**, no. 6, 1992, pp 843-871.
- [3] Pustać, D. and Štok, B., "Crack Tip Plasticity Investigations using Dugdale Strip Yield Model Approach", In Proceedings of the 11th European Conference on Fracture (ECF-11), edited by J. Petit, J. de Fouquet, G. Henaff, P. Villechaise and A. Dragon, Engineering Materials Advisory Services (EMAS), Poitiers-Futuroscope, 1996, vol. I, pp 151-156.

[4]

Prof. Dragan Pustać, Ph.D.

Faculty of Mechanical Engineering and Naval Architecture / University of Zagreb, Ivana Lučića 5, Zagreb, Croatia, Tel: ++385 (1) 6168 178, Fax: ++385 (1) 6168 187, e-mail: dragan.pustaic@fsb.hr

Martina Lovrenić, B.Sc.

Faculty of Mechanical Engineering and Naval Architecture / University of Zagreb, Ivana Lučića 5, Zagreb, Croatia, Tel: ++385 (1) 6168 514, Fax: ++385 (1) 6168 187, e-mail: martina.lovrenic@fsb.hr

DYNAMICS OF FLEXIBLE MULTIBODY SYSTEMS USING A CO-ROTATIONAL APPROACH

V. Damić, M. Čohodar

Keywords: flexible multibody system, bond graphs, component model approach, differential - algebraic equations

1. Introduction

This paper presents general approach for modelling spatial multibody systems composed of rigid and flexible links which are modelled as slender bodies. Among several formulations which are used for analysis of multibody systems a finite element formulation based on co-rotational approach is adopted in this investigation.

This formulation provides nonlinear framework which enables that finite element method can be used in analysis of large motion problems of multibody system. The decomposition of the element motion into rigid body and pure deformation part through the use of a local co-rotational frame which rotates and translates with the element, provides that deformation can be described in the local frame using well-known constitutive relations, while rigid body motion can be analyzed in the inertial frame, that simplifies the treatment of inertial effects.

Also, traditional co-rotation formulation with some adaptations is very usable for application of bond graphs. This technique is used here to develop model of multibody system. Using bond graphs as powerful visual modelling language is provided by the *BondSim* (Damic) application.

2. Bond graph model of flexible link

Bond graph model of 3D beam element is developed (Fig.1a) as the base unit in the development of flexible links. It is represented by component **3DBeam** and realized on more hierarchical levels. Program *BondSim* provides the model development gradually by applying the components decomposition technique.

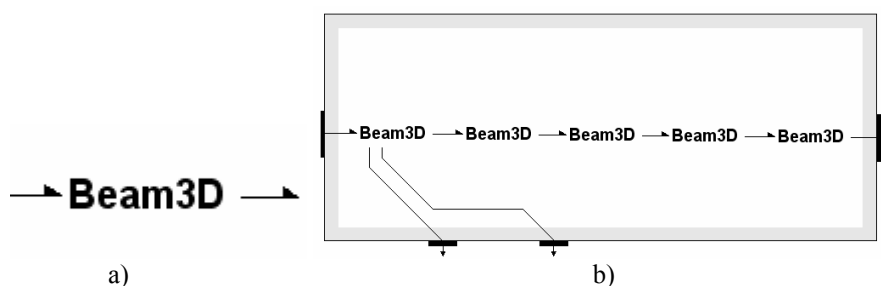


Figure 1. Bond graph model of: a) beam element, b) flexible link composed of five finite elements

Model component **3DBeam** can be put in *Library* of *BondSim* and reused in the development of flexible link. In the other words, first it's necessary to create word model which represents flexible link. After creating word model, it's necessary to insert of beam bond graph components (**3DBeam**) from library in the model workspace using procedure *Copy-Past*, for example five times as shown in Fig.1b if we want to discretize flexible link into five finite elements. These operations are not simple copy and insertion operations, but really involve copying of the complete model trees.

3. Numerical examples

To validate the algorithm presented in this paper two dynamic simulations are performed. As the first example a cantilevered beam attached to a cylinder is considered in order to show possibility of proposed procedure to capture stiffening effect. Spatial manipulator composed of flexible and rigid component connected with revolute joints is chosen as second example. In all cases the flexible links are discretized in ten 3D finite elements each. The results obtained in this investigation show excellent agreement with results obtained in reported references that validates the method presented in the paper.

4. Conclusion

This paper presents general and effective algorithm for modelling and simulation of spatial flexible multibody systems. The co-rotation approach provides non-linear framework for using finite element method. It can be used for systematic development of dynamical models in essentially the same way as when treating system as rigid owing to component model approach. Comparison of simulation results of two numerical examples with the other author results shows excellent agreement and the proposed approach can be recommended to solve this class of problems.

References

- [1] Avelo, A.; Garcia de Jalon, J.; Bayo, E., "Dynamics of flexible multibody systems using Cartesian coordinates and large displacement theory", *Int. J. for Num. Methods in Engineering*, 32, 1991, pp 1543-1563.
- [2] Crisfield, M.A.; Moita, G. F., "A Unified Co-rotational Framework for Solids, Shells and Beams", *Int. J. Solids Structures*, 33, 1996, pp 2969-2992.
- [3] Damić, V. (to appear), "Modelling flexible body systems: a bond graph component model approach", *Mathematical and Computer Modelling of Dynamical Systems*, 12, 2006, pp 175-187.
- [4] Damić, V., Čohodar, M., "A Bond Graph Approach to Modeling of Spatial Flexible Multibody Systems Based on Co-rotational Formulation", *ICBGM'05 (New Orleans, Louisiana Jan.23-27)*, 37, 2005, pp 213-218,
- [5] Damić, V., Čohodar, M., "Modeling Flexible Multibody Systems using Bond Graph Technique", *Extended Abstracts of the 4th International Congress of Croatian Society of Mechanics (Croatia, Osijek Sep.18-20)*, 2003, pp 69-70,
- [6] Damić, V., Montgomery, J., "Mechatronics by Bond Graphs: An Object-Oriented Approach to Modeling", Springer-Verlag, Berlin-Heidelberg, 2003,
- [7] Karnopp, D., "Understanding Multibody Dynamics Using Bond Graph Representations", *Journal of the Franklin Institute*, 334B, 4, 1997, pp 631-642,
- [8] Liu, J.; Hong, J., "Dynamics of three-dimensional beams undergoing large overall motion", *European Journal of Mechanics A/Solids*, 23, 2004, pp 1051-1068.
- [9] Nour-Omid, B., Rankin, C.C., "Finite Rotations Analysis and Consistent Linearization Using Projector", *Computer Methods in Applied Mechanics and Engineering*, 93, 1991, pp 361-384,
- [10] Pacoste, C., Eriksson, A., "Beam Elements in Instability problems". *Int. J. for Numerical Methods in Engineering*, 144, 1997, pp 163-197,
- [11] Rankin, C.C., Brogan, F.A., "An Element Independent Corotational Procedure for the Treatment of Large Rotations", *ASME Journal of Pressure Vessel Technology*, 108, 1986, 165-174.
- [12] Shabana, A.A., "Dynamics of Multibody Systems", 2nd Edn. Cambridge University Press, Cambridge, 1998.

Vjekoslav Damić, Prof. Dr. sc.,

University of Dubrovnik, Department of Mechanical Engineering, Cira Carica 4, 20000 Dubrovnik, Croatia,
+385 20 445 713, vdamic@unidu.hr

Majda Čohodar, Dr.sc.

University of Sarajevo, Faculty of Mechanical Engineering, Vilsonovo setaliste 9, Sarajevo, Bosnia and Herzegovina, +387 33 656 562, cohodar@mef.unsa.ba

TURBULENT FLOW SIMULATION IN KAPLAN DRAFT TUBE

Z. Čarija, Z. Mrša and L. Dragović

Keywords: CFD, Numerical simulation, Kaplan turbine, Turbulence models, Fluent

1. Introduction

This work was motivated by a need for validation of commercial CFD fluid flow solver for hydraulic turbine flow simulations. For the same reason, the ERCOFTAC draft tube has been used for many years as a base for evaluating the state-of-the-art CFD (Computational Fluid Dynamic) codes in the field of hydropower generation. Probably one of the most interesting parts, from the aspect of efficiency, is a hydraulic turbine draft tube. Its role is to increase the pressure head by converting outlet kinetic energy into static pressure and therefore impacts directly on the total efficiency of the hydropower plant. Draft tube fluid flow is difficult to predict numerically due the highly swirled, fluctuating and non-uniform flow that leaves the rotating runner and enters the draft tube. Without visualization of the fluid flow simulation results, only pure intuition existed of what the fluid flow in hydraulic turbines looks like. From the flow simulations results the fluid flow behavior inside draft tube can be understood in sufficient detail and this knowledge can be used to design new generation of the draft tubes or revitalize old ones.

2. Problem description

In this paper the turbulent fluid flow in the Hölleforsen hydropower draft tube model is simulated with the commercial fluid flow code Fluent 6.2. The results were compared with the available experimental data. Experimental measured database, conducted at 60% load of Kaplan turbine, correspond to near optimal point of operation (case T). Experimental settings were $H=4.5$ m (head), $n=595$ rpm (runner speed) and $Q=0.522$ m³/s (flow rate).

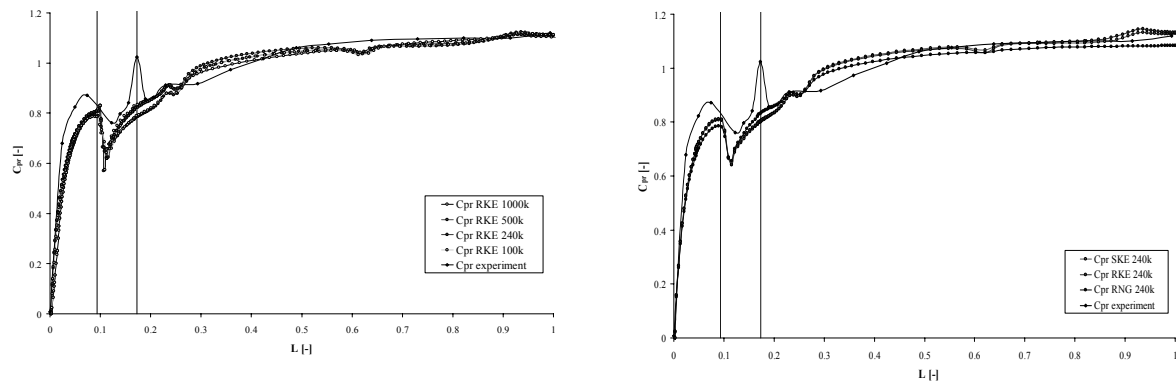
3. Results

Calculated pressure recovery distributions along upper and lower wall center line (Fig. 1.) compared with the experimental data are very good. Larger deviations from experimental results are found in elbow part of draft tube. This is reasonable because this is a region with highly complex fluid flow structure where RANS turbulence models are probably not enough appropriate for fluid flow prediction. After the elbow in horizontal draft tube part calculated pressure recovery almost equals to measurements.

RKE and SKE turbulence models have shown the robustness with suppressing level of accuracy predicting draft tube engineering quantities very close to measured one. RNG turbulence model appears bad, predicting lower value in whole domain and its application should be avoided for this and similar problems. Strong grid size influence was observed on calculated results especially on pressure distribution near inlet section where non realistic high pressure region are found when using coarser grids.

Flow behavior in draft tube can be visualized with path lines analyses performed by releasing fluid particles from the curves of constant radial-coordinate values at the inlet section (Fig. 2.). Path lines

released from inner inlet sections (lower radial coordinate values) are surrounded with path lines released from outer sections forming large dominant vortex in the whole flow domain.



a)

b)

Figure 1. C_{pr} distribution along upper wall centreline ; a) RKE turbulence model, b) 240k mesh size

Path lines released from inner section (closer to runner cone) contains higher swirl intensity, generated by rotating runner cone, which is propagated towards draft tube outlet section.

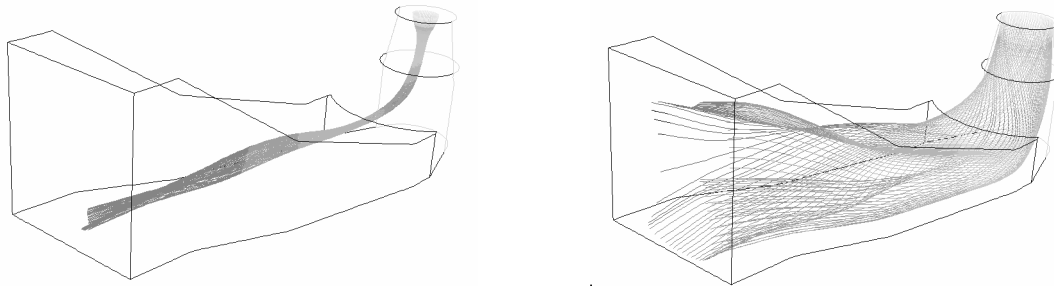


Figure 2. Path lines in draft tube released from different inlet radial sections

4. Conclusion

Fluid flow analyses in hydropower draft tube model were investigated using three $k-\epsilon$ turbulence models (standard, realizable and RNG) applied on several hexahedral meshes. Realizable and standard turbulence $k-\epsilon$ turbulence model results have shown suppressing level of accuracy predicting draft tube engineering quantities very close to measured one. RNG turbulence model predict lower pressure recovery value in whole draft tube domain and its application should be avoided for this and similar problems.

References

- [1] Fluent 6.2 User manual, Fluent.Inc, 2006.
- [2] Launder B. E., Spalding D. B. The Numerical Computation of Turbulent Flows. Computer Methods in Applied Mechanics and Engineering, 1974.

Zoran Čarija, mr.sc.

Faculty of Engineering, University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, Phone: 0038551 651554, e-mail: zcarija@rijeka.riteh.hr

Zoran Mrša, prof.dr.sc.

Faculty of Engineering, University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, Phone: 0038551 651554, e-mail: mrsa@rijeka.riteh.hr

Luka Dragović, dipl.ing., e-mail: lukad@rijeka.riteh.hr

APPLICATION OF NON-ASSOCIATED FLOW RULE TO SHEET METAL FORMING

Vedrana Cvitanić, Željko Lozina and Frane Vlák

Keywords: *anisotropic constitutive model, anisotropic yield function, non-associated flow rule, numerical integration*

1. Introduction

Sheet metals usually exhibit a certain initial plastic anisotropy mainly due to crystallographic texture that is caused by production rolling steps and annealing. Anisotropy should be considered carefully for accurate analysis of sheet metal forming operations. A common approach in metal plasticity theory is assumption that the yield function and plastic potential function are identical, an assumption known as the associated flow rule. Realistic elasto-plastic anisotropic constitutive models based on associated flow rule require rather complex anisotropic yield functions. In this article anisotropic model that is based on non-associated flow rule is proposed and analyzed on constitutive and algorithmic level considering material characteristics of aluminum alloy Al 2090-T3. It should be pointed out that application of non-associated flow rule refers for examination of theoretical considerations that are commonly interpreted as validation of associated flow rule [1], [2].

2. Orthotropic yield functions and plastic potential functions

In proposed model as yield function and plastic potential function, Hill (1948) [3] and Karafyllis-Boyce (1993) [4] functions are considered. Parameters of yield functions denoted as Hill-ID-1 and KB-ID-1 are defined in terms of measured yield stresses. Parameters of plastic potential functions, denoted as Hill-ID-2 and KB-ID-2, are defined in terms of plastic strain ratios (r -values). Predictions of planar anisotropy of these functions are compared for aluminum alloy Al 2090-T3, Figure 1.

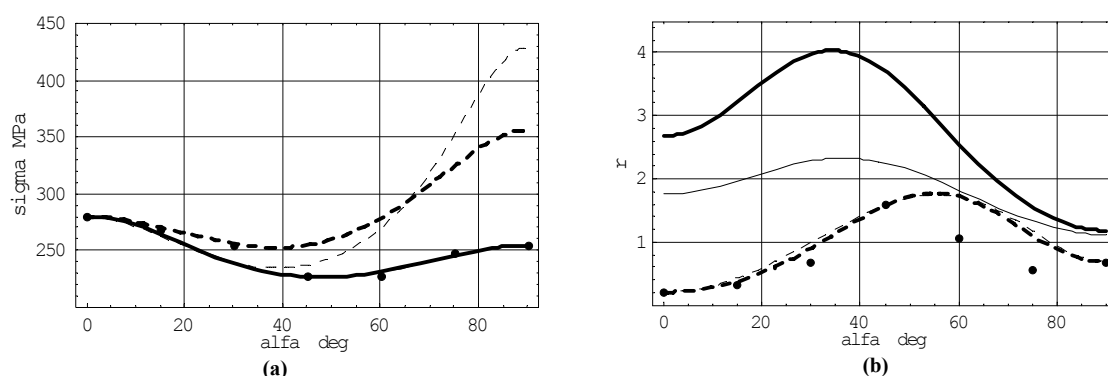


Figure 1. Comparison of experimental and predicted directional (a) yield stresses, (b) r -values for Al 2090-T3. (• - Measured values, — - Hill-ID-1, - - - Hill-ID-2, - · - KB-ID-1, - - - KB-ID-2)

3. Algorithmic formulation of constitutive model based on non-associated flow rule

Algorithmic formulation of model that assumes anisotropic yield function, isotropic hardening and non-associated flow rule is derived by application of implicit return mapping algorithm [5]. Evolution equation for hardening variable consistent with principle of plastic work equivalence is

proposed. Utilized evolution equations for plastic strain rate $\dot{\epsilon}^p$ and hardening variable $\bar{\epsilon}^p$ read

$$\dot{\epsilon}^p = \dot{\lambda} \frac{\partial f_p(\boldsymbol{\sigma})}{\partial \boldsymbol{\sigma}}, \quad \dot{\bar{\epsilon}}^p = \dot{\lambda} f_p(\boldsymbol{\sigma}) / f_y(\boldsymbol{\sigma}) \quad (1)$$

where f_p is plastic potential function, f_y yield function, $\boldsymbol{\sigma}$ stress tensor and $\dot{\lambda}$ consistency parameter. Calculation procedure for updating state variables is developed using *Mathematica* [6]. Accuracy of formulations that utilize Hill (1948) function and associated flow rule (AFR-Hill) or non-associated flow rule (NAFR-Hill) are estimated by calculating iso-error maps. Iso-error maps are calculated at three representative points on yield surface. As an example, iso-error maps of point A-uniaxial stress state, are shown at Figure 2.

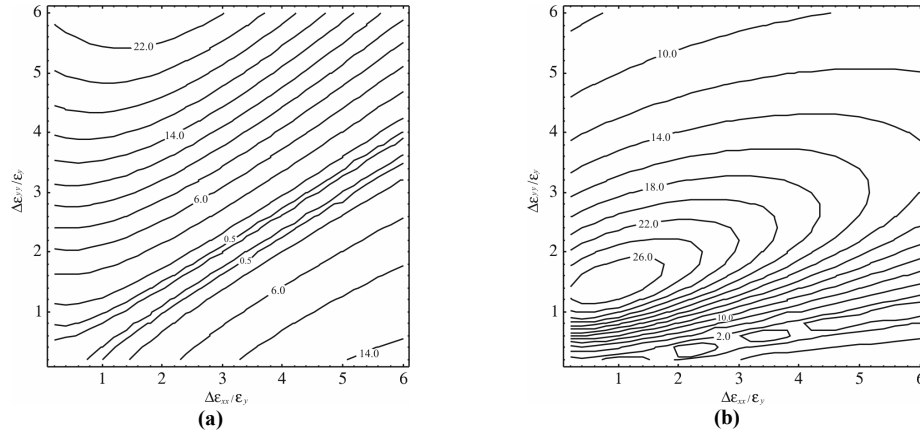


Figure 2. Iso-error maps of the point A: (a) AFR-Hill; (b) NAFR-Hill

4. Conclusion

In this article, it is shown how under assumption of non-associated flow rule simple anisotropic functions such as Hill (1948) and Karafillis-Boyce (1993) enable satisfactory description of planar anisotropy. Algorithmic formulation of constitutive model that assumes isotropic hardening and non-associated flow rule is derived by application of implicit return mapping algorithm. By calculating iso-error maps it has been revealed that reduction of strain increment doesn't entail reduction of error magnitude if non-associated flow rule is used.

References

- [1] Stoughton, T.B., Yoon, J.W., "Review of Drucker's postulate and issue of plastic stability in metal forming", *International Journal of Plasticity*, Vol. 22, 2006, pp. 391-433.
- [2] Cvitanic, V., "Anisotropic elasto-plastic model of aluminum alloys based on non-associated flow rule", (in Croatian), Doctoral Thesis, FESB/University of Split, 2006.
- [3] Khan, A.S., Huang, S., "Continuum Theory of Plasticity", Wiley-Sons Inc., New York, 1995.
- [4] Karafillis, A.P., Boyce, M., "A general anisotropic yield criterion using bounds and a transformation weighting tensor", *Journal of the Mechanics and Physics of Solids*, Vol. 41, 1993, pp. 1859-1886.
- [5] Simo, J.C., Hughes T.J.R., "Elastoplasticity and Viscoplasticity-Computational Aspects", Springer-Verlag, Copyright 1985, 1987, 1988.
- [6] *Mathematica 5.1*, Wolfram Media Inc.

Dr.sc. Vedrana Cvitanic

FESB/University of Split, Department of mechanical engineering and naval architecture, Rudjera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 865, Fax: 385(0)21/463 877, e-mail: vcvit@fesb.hr

Prof. dr.sc. Željko Lozina

FESB/University of Split, Department of mechanical engineering and naval architecture, Rudjera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 890, Fax: 385(0)21/463 877, e-mail: zeljan.lozina@fesb.hr

Dr.sc. Frane Vlsek

FESB/University of Split, Department of mechanical engineering and naval architecture, Rudjera Boškovića bb, Split, Croatia, Tel: 385(0)21/305 864, Fax: 385(0)21/463 877, e-mail: fvlak@fesb.hr

REGIONAL AIR POLLUTION MODELLING

M. Čavrak, Z. Mrša and G. Štimac

Keywords: air pollution, atmospheric modelling, emission sources

1. Introduction

Wide range of problems of atmospheric physics, climate, and environmental protection are solved using mathematical modeling methods. The biggest environmental concern nowadays is air pollution and the methods of its reduction whether the source is biogenic or anthropogenic. Coupling meteorological prognostic models, emission processing and chemical transport models an off-line air quality modeling system was created. Results were compared with observations from measuring stations located in the vicinity of wider Rijeka region. That modeling system offers ability to forecast potential pollution from newly created emission sources, getting insight into effects of emission reductions, or as a step toward obtaining daily air quality level through an, yet to be build, on-line air quality modeling system.

2. Methodology

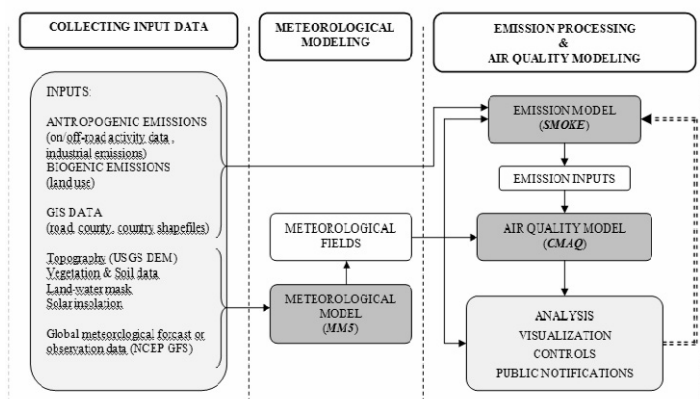


Figure 2. – Structure of an air quality modelling system

Air quality modelling system was created based on EPA's Community Multiscale Air Quality (CMAQ v4.5.1) model that incorporates *Model3* mathematical one-atmosphere 3D model with numerical routines for solving atmospheric chemistry.

Using meteorological preprocessor MM5 (*Mesoscale model 5*) meteorological fields (pressure, velocity vectors, temperature, humidity, etc.) were derived and used as input variables for CMAQ.

3. Emissions

Emission data were obtained from different sources (European National Emission Inventory (EMEP) – global european temporarily and spatially averaged emission data subdivided in *sectors* following SNAP-97 nomenclature, on-road activity data (obtained from Croatian road association measuring stations) and yearly averaged emissions from industrial point sources (obtained from INA's (petrochemical refinery) and Croatian NEI's databases). SMOKE v2.2 (*Matrix Operator Kernel Emissions Modelling System*) was applied for assigning chemical, temporal and spatial profiles on yearly averaged emissions.

4. Simulated episode description

Evaluation was performed for an episode of high temperatures from 27th till 31st of June 2005. According to measured data that weekend offered high concentration of ozone. Using nesting technique a simulation was performed on two domains. First, coarser, for North Adriatic region using rectangular 300km x 300km domain (100x100 nodes) and for a finer 50km x 50km domain (50x50 nodes) centered in the center of Kvarner bay. Vertical dimension consists of 18 sigma layers, i.e. it follows terrain shape up to the upper boundary layer the 100 mbar top pressure surface, hence covering whole troposphere. Coarse domain D1 boundary and initial conditions were constructed from EMEP expert emission data, hence for D2 local road activity and emissions from industrial sources were used.

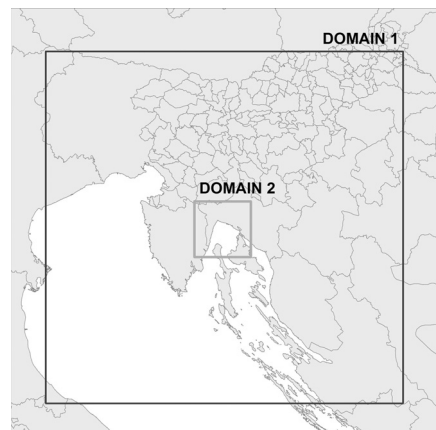


Figure 3. D1 & D2 domains coverage

5. Results and discussion

Analysis of multiple day simulation showed strong dependence of pollutants formation and transport to the horizontal advection through domain boundaries (orographic channelling) and to the stable meteorological conditions [Fig. 3.] leading to pollution accumulation in Kvarner basin in first 500m a.s.l. Comparison with measurements showed underestimation of concentration with all accounted pollutants as expected due to lack of biogenic emissions and fine resolution emission data in general.

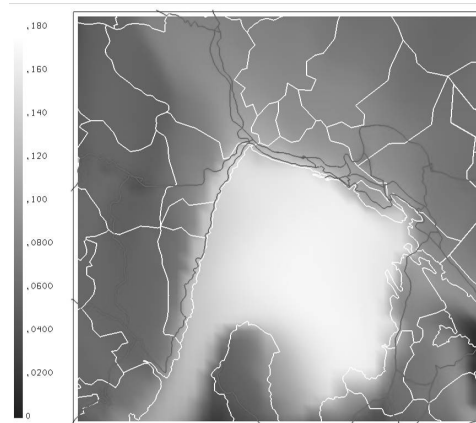


Figure 4. – O₃ concentrations

6. References

- [1] Guerrero, P. J., Air Quality Modeling in Very Complex Terrains: Ozone Dynamics in the Northeastern Iberian Peninsula, Ph.D. Thesis, Polytechnical university of Catalonia, 2005.
- [2] Guidelines for developing an Air Quality (Ozone and PM_{2.5}) forecasting program, EPA-456/R-03-002, 2003.
- [3] Thunis P., The influence of scale on modelled ground level O₃ concentrations, Research Report no. 57, EMEP/MS-C-W, Norwegian Meteorological Institute, 2001.

[4]

Marko Čavrak, mech. eng.

University of Rijeka, Technical faculty, Department for fluid mechanics and computational engineering, Vukovarska 58, Rijeka, Croatia, +385 (0)51 651-554, +385 (0)51 651-290, mcavrak@riteh.hr

Prof. dr. sc. Zoran Mrša, mech. eng.

University of Rijeka, Technical faculty, Department for fluid mechanics and computational engineering, Vukovarska 58, Rijeka, Croatia, +385 (0)51 651-500, +385 (0)51 651-290, mrša@riteh.hr

Goranka Štimac, mech. eng.

University of Rijeka, Technical faculty, Department for technical mechanics

Vukovarska 58, Rijeka, Croatia, +385 (0)51 651-444, +385 (0)51 651-290, goranka.stimac@riteh.hr

RIVER FLOOD LINES PREDICTION WITH 1D OPEN CHANNEL FLOW MODEL

Siniša Družeta, Jerko Škifić, Bojan Crnković, Luka Sopta, Danko Holjević, Nelida Črnjarić-Žic and Senka Maćešić

Keywords: open channel flow, flood lines, flood mitigation, environmental hydraulics

1. Introduction

Numerical modelling is becoming standard practice for river flood lines prediction among other procedures in flood mitigation planning. 1D open channel flow model has shown to be optimal for large-scale natural watercourses modelling thus it was used for flood lines prediction in the entire Croatian region of Gorski kotar where total length of modelled watercourses amounted to 108 km.

Software package *Stripp12* used for river flow and flooding simulation was employed for unsteady water wave propagation simulations, as opposed to classic steady-state simulations of peak discharge flow regime.

2. Mathematical model and numerical scheme

For the unsteady water wave propagation modelling 1D open channel flow model was used with first-order upwind Q-Scheme as implemented in the software package *Stripp12*. Previous research and practical projects demonstrated the first-order balanced upwind Q-Scheme as stable and reliable, especially for difficult channel geometries. Furthermore, the numerical scheme satisfies the exact conservation property i.e. quiescent flow test.

3. Flood wave propagation simulations

Before the simulations of the flood wave propagation could be performed, the model input data such as topographic data (maps, drawings, etc.), hydrologic analysis results and watercourse cross-sections were collected and prepared.

Essentially, the procedure of the flood wave propagation simulations was the same for each of the analyzed watercourses and went as follows:

Test simulations were performed for the 1000-year return period discharge hydrographs in order to determine the locations at which the water levels reach the top of the river banks. These locations were identified in the available drawings and maps and the watercourse cross-sections in these areas were extended so to include the flood areas. In this manner broadened 1D watercourse model thus included all potential flood areas.

Lateral inflows were adjusted through a series of test simulations to facilitate the agreement of the computed discharge values and the hydrological analysis results.

Flood wave propagation simulations were performed on the developed numerical models of the analyzed watercourses, for each of the return periods.

4. Flood lines construction

The results of the unsteady flood wave propagation simulations were filtered in a way that only the maximum water level values at each of the domain cells were preserved. Thus the longitudinal

maximum water level cross-sections were obtained for all analyzed watercourses for each of the different return period scenarios.

Based on the available geodesic information and maximum water level values at a specific watercourse reach, left and right water perimeter points were determined for the corresponding river cross-section. This procedure was performed along the entire watercourse length, yielding flood lines. The outcome of the process were the flood lines for all analyzed watercourses for each of the different return period scenarios.

Standard maps of the analyzed region overlapped with the flood lines were used as flood risk maps. Maps of this kind were afterwards overlapped with spatial planning maps, which made possible the implementation of flood damage analysis, cost-benefit analysis of the flood prevention measures, etc.

5. Conclusion

The 1D open channel flow model was successfully applied for the purpose of flood lines prediction for very long watercourses. First-order balanced upwind Q-Scheme has shown to be robust and reliable. It enables unsteady modelling of steep, irregular and discontinuous channel geometries. Therefore it was especially useful for the modelling of the watercourses such as the mostly canyon rivers of the Gorski kotar region.

Unlike traditionally practiced steady-state simulations of maximum discharge for the purpose of flood lines prediction, the procedure described here was based on unsteady simulations of the flood wave propagation in its whole time period. This makes the simulations far more realistic and usually results in smaller flood areas, since steady-state simulations systematically overestimate the size of the flood areas.

In order to further improve the explained methodology of the flood risk maps production, it is advised to maximize the quantity and quality of the geodesic and hydrological input data. Moreover, in the areas of potentially greatest flood-induced damage, it would be necessary to employ 2D modelling. However, 2D modelling is much more demanding in terms of required geodesic survey.

Siniša Družeta, Assist. M. Sc.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651497, 00385 (0)51 651490, sinisa.druzeta@riteh.hr

Jerko Škifić, Assist. M. Sc.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651497, 00385 (0)51 651490, jerko.skific@riteh.hr

Bojan Crnković, Young Assist.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651497, 00385 (0)51 651490, bojan.crnkovic@riteh.hr

Luka Sopta, Prof.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651493, 00385 (0)51 651490, luka.sopta@riteh.hr

Danko Holjević, M. Sc.

Croatian Waters, VGO Rijeka, Đure Šporera 3, Rijeka, Croatia, 00385 (0)51 666400, 00385 (0)51 336947, dholjev@voda.hr

Nelida Črnjarić-Žić, Assist. Prof.

Faculty of Engineering University of Rijeka, Department of Mathematics, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651481, 00385 (0)51 651490, nelida.crnjaric@riteh.hr

Senka Mačević, Prof.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651494, 00385 (0)51 651490, senka.vukovic@riteh.hr

IMPROVED IMPLICIT NUMERICAL SCHEME FOR ONE-DIMENSIONAL OPEN CHANNEL FLOW EQUATIONS

L. Kranjčević, B. Crnković and N. Črnjarić Žic

Keywords: open channel flow, implicit numerical scheme, balance laws, source term decomposition, channel with irregular geometry

1. Introduction

Implicit schemes are well known for the property of allowing numerical stability in the resolution of partial differential equations in presence of time steps not restricted by the Courant-Friedrichs-Lewy (CFL) condition. In this work upwind implicit scheme is used for resolution of the one-dimensional open channel flow equations driven by the interest in accurate and efficient numerical modeling of nonhomogeneous hyperbolic system of conservation laws emphasizing space dependency of the flux and significant geometrical source term variation. Numerically balanced approximations of flux and source terms including the flux space dependency have been introduced by some authors employing exclusively explicit numerical approach. In this paper we modified original finite volume Linearised Conservative Implicit (LCI) scheme in order to account for the spatially variable flux dependency, and consequently the source term was appropriately decomposed to balance the upwind flux decomposition. These numerical modifications enabled the use of implicit numerical scheme in modeling of the open channel flow equations involving non-prismatic channels with rectangular cross section geometry.

2. The one-dimensional open channel flow equations

The governing one-dimensional open channel flow equations [6] model the homogeneous, incompressible, viscous water flow in rivers and channels characterized by a hydrostatic pressure distribution. The one-dimensional open channel flow equations are based on conservation of mass and conservation of momentum that form a system of partial differential equations that written in vector form reads:

$$\frac{\partial \mathbf{w}}{\partial t} + \frac{\partial}{\partial x} \mathbf{f}(x, \mathbf{w}) = \mathbf{s}(x, \mathbf{w})$$

in which $\mathbf{w}$ is a vector of flow variables, $\mathbf{f}$ represents flux vector, $\mathbf{s}$ source term vector.

3. Improved upwind implicit scheme with source term decomposed

In this section implicit upwind TVD numerical scheme used is briefly presented, since primary goal was to construct implicit numerical scheme able to deal with channels with variable width and strong variation in bed geometry with friction term included. Implicit scheme used in this paper is based upon Roe's scheme solving Riemann problems approximately on the cell boundaries. Finite volume method is employed, with uniform space step Δx , and adjustable time step Δt . The general form of finite volume numerical scheme used can be written as:

$$\mathbf{w}_i^{n+1} + \beta \theta (\tilde{\mathbf{f}}_{i+1/2}^{n+1} - \tilde{\mathbf{f}}_{i-1/2}^{n+1}) - \theta \cdot \Delta t \mathbf{s}_i^{n+1} = \mathbf{w}_i^n - \beta(1 - \theta)(\tilde{\mathbf{f}}_{i+1/2}^n - \tilde{\mathbf{f}}_{i-1/2}^n) + (1 - \theta)\Delta t \mathbf{s}_i^n$$

where $\beta = \Delta t / \Delta x$, $0 \leq \theta \leq 1$ and $\tilde{\mathbf{f}}_{i\pm 1/2}^{n, n+1}$ numerical flux vector. For $\theta = 0$, the scheme is explicit and for $\theta = 1$ Euler implicit time integration is achieved, i.e., it is the fully implicit case. For implicit case numerical flux vector is evaluated

$$\tilde{\mathbf{f}}_{i+1/2}^{n+1} = \frac{1}{2}(\mathbf{f}_{i+1}^{n+1} + \mathbf{f}_i^{n+1}) - \frac{1}{2}|\tilde{\mathbf{Q}}_{i+1/2}|(\mathbf{w}_{i+1}^{n+1} - \mathbf{w}_i^{n+1}) - \frac{1}{2}\tilde{\mathbf{Q}}_{i+1/2}^{-1}|\tilde{\mathbf{Q}}_{i+1/2}|\mathbf{V}_{i+1/2}^{n+1}\Delta x \quad (1)$$

with matrix $\tilde{\mathbf{Q}}_{i+1/2}$ being approximate Jacobian of the flux at the cell interface, vector $\mathbf{V}$ in (1) accounts for space dependency of the flux. Source term is appropriately decomposed to balance the flux upwind decomposition in (1):

$$\mathbf{s}_i^{n+1} = \mathbf{s}_{i+1/2}^{n+1,L} + \mathbf{s}_{i-1/2}^{n+1,R}$$

where

$$\begin{aligned} \mathbf{s}_{i+1/2}^{n+1,L} &= \frac{1}{2}\left(I - \tilde{\mathbf{Q}}_{i+1/2}^{-1}|\tilde{\mathbf{Q}}_{i+1/2}|\right)\mathbf{s}_{i+1/2}^{n+1} \\ \mathbf{s}_{i-1/2}^{n+1,R} &= \frac{1}{2}\left(I + \tilde{\mathbf{Q}}_{i-1/2}^{-1}|\tilde{\mathbf{Q}}_{i-1/2}|\right)\mathbf{s}_{i-1/2}^{n+1} \end{aligned}$$

Some terms need to be linearised using Taylor expansion around w^n and finally we get the numerical procedure that produces block tridiagonal linear system of equations that is to be solved in each time step.

4. Numerical tests

Chosen test problems validate the correct balancing of the flux gradient and the source term. In order to illustrate the quality and the achieved improvements, these balanced schemes are compared with their versions with pointwise evaluated source terms.

5. Conclusion

Two versions of conservative implicit methods belonging to the family of linearized upwind schemes have been presented. Improved balanced implicit scheme was tested and compared with simple non-balanced pointwise version of the scheme on different open channel test problems which include friction, nonuniform bed slopes, strong channel width variations and dam break problem with analysis of transient solution. In unsteady case both methods proved to be nondispersive, showing good shock tracking capability, and with moderate CFL numbers produced acceptably dissipative results. The one-dimensional open channel flow equations numerical modeling approach proposed showed excellent results because successful balancing of the implicit upwind scheme was performed reaching exact conservation property and still keeping all the benefits that implicit numerical approach brings.

Lado Kranjčević, Ph.D., Assist.

Faculty of Engineering/University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, Phone 00385 (0)51 651444, Fax 00385 (0)51 651490, lado.kranjcevic@riteh.hr

Bojan Crnković, Assist.

Faculty of Engineering/University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, Phone 00385 (0)51 651444, Fax 00385 (0)51 651490, bojan.crnkovic@riteh.hr

Nelida Črnjarić Žic, Ph.D., Assist. Prof.

Faculty of Engineering/University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, Phone 00385 (0)51 651444, Fax 00385 (0)51 651490, nelida@riteh.hr

[5]

[6]

EFFICIENT IMPLEMENTATION OF ENO SCHEME IN WATER HAMMER WAVE MODELLING

J. Škifić, N. Črnjarić-Žic and B. Crnković

Keywords: Water hammer, wave modelling, ENO scheme, hyperbolic conservation laws

1. Introduction

Hydraulic transients can cause high pressures, which can be dangerous for the piping systems, especially in hydroelectric powerplants. During the simulation of transient pipe flow, friction losses are often estimated by using steady or quasi-steady friction terms. The velocity profiles in rapid transients significantly differ from the "steady" profiles in slowly varying transients. Since hydraulic transients can be described by the hyperbolic conservation laws and since the source term cannot be exactly evaluated due to increasingly more complicated expressions, it would be desirable to apply numerical schemes that are high order accurate. The numerical schemes satisfying the stated requirements are finite volume ENO schemes, which are used in this paper.

2. Mathematical model

Allievi's liquid pipe flow mathematical model is derived from mass and momentum conservation laws [2]. The friction factor used is considered as the sum of steady and the unsteady part. The steady part of the friction factor is known, while the unsteady part is related to the instantaneous temporal and instantaneous convective acceleration, that is the Vítkovský's formulation [2] of the original Brunone model is used, where the Brunone friction coefficient k is evaluated using the Vardy and Brown's shear decay coefficient C^* [2].

The Brunone unsteady friction model cannot produce the low frequency shift noticed in experimental results [2]. Therefore, the momentum correction factor β and the kinetic energy correction factor α equations were presented [1]. The momentum correction factor can be determined from either the log or power laws for the velocity distribution.

3. Finite volume ENO scheme

Finite volume ENO scheme was implemented taking into account the momentum correction factor and unsteady friction model. ENO (essentially non-oscillatory) reconstruction is based on high order polynomial approximations, where the main idea is to avoid the polynomials with spurious oscillations, which appear around the discontinuities. To achieve this effect, the Newton formulation of interpolating polynomial and the corresponding divided differences are considered [4].

4. Numerical results

Numerical tests were conducted with finite volume ENO scheme using the Lax Friedrichs flux, with time order 3 and space order 2 and compared with experimental data [3]. In order to investigate the robustness of the proposed method, different computational number of points was used, that is 256, 128, 64 and 32 points. Also different C_{eff} coefficients were used: 0,9, 0,75 and 0,5.

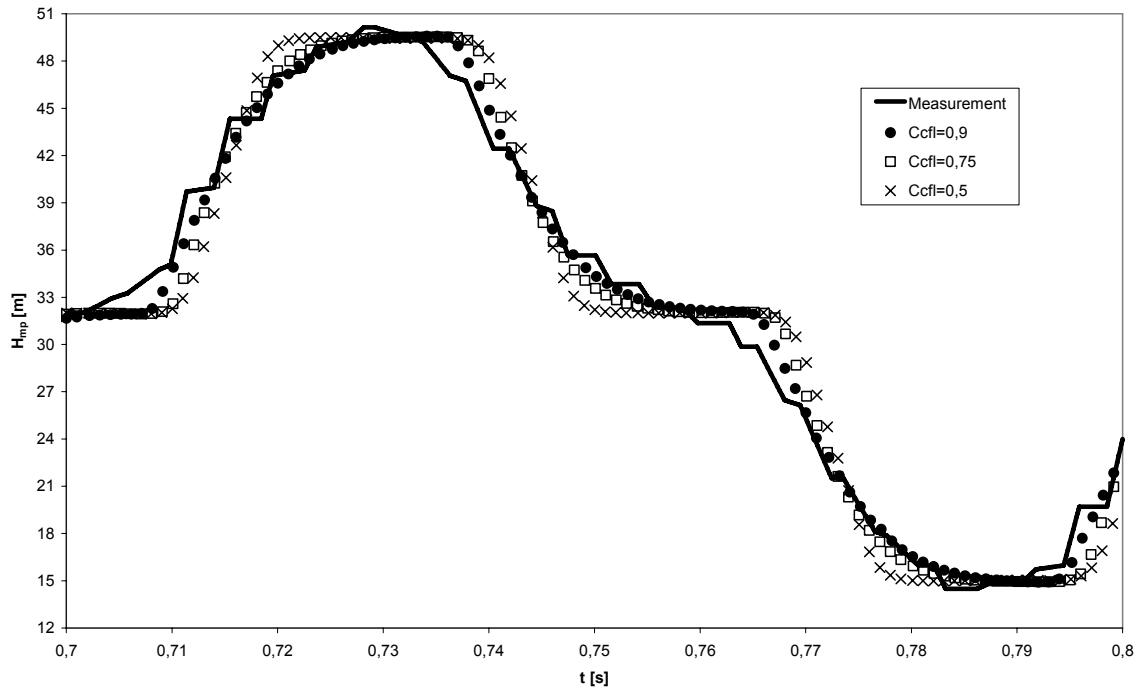


Figure 1. Numerical results with N=128 numerical points used at pipeline midpoint

The difference between the model results and the experimental data points towards a high frequency damping phenomena that is not currently being adequately modelled. Moreover, there is evident dependency upon the c_{fl} and number of computational points used. However, all simulations model water hammer's attenuation fairly correctly, while its shape is not so accurately modeled, which suggests model's lack of diffusion present in experimental data.

5. Conclusion

Implementation of finite volume ENO scheme to the hydraulic transients modeling was proven to be justified. Applied one dimensional assumptions fail to capture the full three dimensional nature of the resulting damped oscillatory flow. Unsteady friction is the primary manifestation of this effect. However, conducted numerical simulations have shown to be in a good agreement with measured data. It is also worth noting that coarse grids give adequately good results compared to denser grid. Moreover, c_{fl} number variation shows that finite volume ENO scheme produces very little numerical diffusion.

Jerko Škifić, Assist. M. Sc.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651497, 00385 (0)51 651490, jerko.skific@riteh.hr

Nelida Črnjarić-Žic, Assist. Prof.

Faculty of Engineering University of Rijeka, Department of Mathematics, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651481, 00385 (0)51 651490, nelida.crnjaric@riteh.hr

Bojan Crnković, Young Assist.

Faculty of Engineering University of Rijeka, Department of Fluid Mechanics and Computational Engineering, Vukovarska 58, Rijeka, Croatia, 00385 (0)51 651497, 00385 (0)51 651490, bojan.crnkovic@riteh.hr

NUMERICAL SOLVING OF PROBLEMS WITH HIGH GRADIENTS USING ADAPTIVE MULTIREOLUTION METHOD

Vedrana Kozulić, Blaž Gotovac, Hrvoje Gotovac and Tonči Radelja

Keywords: mesh free method, basis functions, collocation method, multilevel numerical model, high gradients, adaptive procedure

Extended abstract

In recent years, there has been an intensive development of meshfree methods for solving of complex physical processes and engineering problems ([1], [5], [7]). Meshfree methods belong to one of the most progressively developed part of Computational Mechanics. The concept of a mesh-free method is in establishing a system of algebraic equations for the whole problem domain without using a predefined mesh. Results obtained by using this new class of numerical methods are more accurate than those obtained by using FEM, as there are no stress discontinuity problems existing on interfaces between the finite elements.

In meshfree methods, the construction of basis functions is the central issue. The domain for field variable approximation (the support domain) should be small in comparison with the entire problem domain (compact support). Satisfaction of the compact condition leads to a banded system matrix that can be handled with good computational efficiency. Therefore, the choice shall be on finite basis functions with a small support, which do not depend upon the type and degree of the boundary-value problem. In order that the numerical solution is not too complex, it is necessary to calculate simply enough values of basis functions and their derivatives as well as the scalar products of the function with itself, its derivatives and elementary functions. Satisfaction of these requirements ensures both easy implementation of the meshfree method and accuracy of the numerical solutions.

Functions implemented in numerical analyses of this work are the Fup basis functions. Fup basis functions belong to the class of atomic or R_{bf} basis functions (Rvachev's basis functions), see [8], [2]. They are infinitely differentiable functions with compact support. Atomic functions are classified between classic polynomials and spline functions. However, in practice, their application as basis functions is closer to splines or wavelets. These basis functions have properties of good approximation as well as very important property of universality ([9]), which means that the vector space of the n -th dimension is contained within the vector space of the $(n+1)$ dimension. This property makes it possible to add basis functions hierarchically in the domain to the initial base of an approximate solution.

As it is possible to calculate derivation values of Fup basis functions of high degree in a precise yet simple way, so it is possible efficiently to apply strong formulation procedures. In this work, the collocation method is used to develop the numerical model. The differential equation of the problem is satisfied in collocation points of closed domain, while boundary conditions are satisfied exactly at the domain boundary. The domain of the problem and its boundaries are represented by a set of nodes rather than resorting to the traditional discretization into finite elements. The selection of nodes (collocation points) is important to obtain stable and accurate results. Spatial discretization and grid adaptation are obtained by Fup collocation transform. The distribution of collocation points within the observed area is changed adaptively, depending on the character of solution function and accuracy criteria. The process of node generation is fully automated by a computer.

Meshfree method developed in this work is named **A**daptive **F**up **C**ollocation **M**ethod (AFCM). Basic characteristic of the numerical model is an adaptive multi-resolution evolution of solution with all resolved spatial scales and with desired level of accuracy. The main advantages of the method are the ease of use and the ease of implementation of the algorithm. The method is very convenient for numerical simulations of processes with high gradients and concentrated loads. Efficiency of the proposed method in computational mechanics is provided by few examples of one dimensional problem, namely, the approximation of the test function characterized by narrow zones of front and wave, and the analysis of a beam on an elastic foundation subjected to a concentrated load. Results show that AFCM is well suited for dealing with general boundary conditions and sharp gradients. Hierarchic increase a number of basis functions in the model provides a simple way to increase the accuracy of an approximate solution in places where rapid changes of solution steepness occur. Simultaneously, computational efforts are reduced due to space adaptation.

The proposed mesh free method has very significant advantages: (1) predefined mesh and numerical integration are avoided; (2) values of the main solution function and all values derived from the main solution are calculated in the same points with the same level of accuracy; (3) the solution and all its derivatives are expressed by continuous functions on the entire domain due to properties of localized Fup basis functions.

An accuracy of numerical solution depends on prescribed threshold. The prescribed threshold active controls numerical error of an approximate solution. With a very small value of the prescribed threshold, numerical solution obtained by AFCM is practically exact solution of the considered engineering problem.

References

- [1] Atluri, S.N., "Methods of Computer Modeling in Engineering & the Sciences", Volume I, Tech Science Press, University of California, Irvine, 2005
- [2] Gotovac, B., "Numerical modelling of engineering problems by smooth finite functions", (in Croatian), Ph.D. Thesis, Faculty of Civil Engineering, University of Zagreb, Zagreb, 1986
- [3] Gotovac, B. and Kozulić, V., "On a selection of basis functions in numerical analyses of engineering problems", International Journal for Engineering Modeling, Vol. 12, No. 1-4, 1999, pp 25-41.
- [4] Gotovac, B. and Kozulić, V., "Numerical solving the initial value problems by R_{bf} basis functions", Int. J. Structural Engineering and Mechanics, Vol. 14, No. 3, 2002, pp 263-285.
- [5] Griebel, M. and Schweitzer, M.A. (Eds.), "Meshfree Methods for Partial Differential Equations", Springer-Verlag, Berlin, 2003
- [6] Kozulić, V. and Gotovac, B., "Numerical analyses of 2D problems using $Fup_n(x,y)$ basis functions", International Journal for Engineering Modeling, Vol. 13, No. 1-2, 2000, pp 7-18.
- [7] Liu, G.R., "Mesh free methods : Moving beyond the finite element method", CRC Press LLC, Boca Raton, 2003
- [8] Rvachev, V.L. and Rvachev, V.A., "On a Finite Function", DAN URSR, A(6), 1971, pp 705-707.
- [9] Rvachev, V.L. and Rvachev, V.A., "Non-classical methods for approximate solution of boundary conditions", (in Russian), Naukova dumka, Kiev, 1979
- [10] Rvachev, V.L., "Theory of R-functions and their application", Naukova dumka, Kiev, 1982

Vedrana Kozulić, professor

Faculty of Civil Engineering and Architecture University of Split, Matice hrvatske 15, 21000 Split, Croatia,
Tel: +385-21-303-365, Fax: +385-21-465-117, **e-mail:** vedrana.kozulic@gradst.hr

Blaž Gotovac, professor

Faculty of Civil Engineering and Architecture University of Split, Matice hrvatske 15, 21000 Split, Croatia,
Tel: +385-21-303-335, Fax: +385-21-465-117, **e-mail:** blaz.gotovac@gradst.hr

Hrvoje Gotovac, junior researcher

Faculty of Civil Engineering and Architecture University of Split, Matice hrvatske 15, 21000 Split, Croatia,
Tel: +385-21-303-354, Fax: +385-21-465-117, **e-mail:** hrvoje.gotovac@gradst.hr

Tonči Radelja, senior lecturer

Faculty of Civil Engineering and Architecture University of Split, Matice hrvatske 15, 21000 Split, Croatia,
Tel: +385-21-303-302, Fax: +385-21-465-117, **e-mail:** tonci.radelja@gradst.hr

DYNAMIC PROPERTIES OF STRUCTURES AND ADVANCED ALGORITHMS FOR VALIDATION

Zeljani Lozina, Damir Vucina and Damir Sedlar

Keywords: dynamics properties, modal analysis, changed elements, finite element, validation structures

1. Introduction

The methods for validation and calibration of structures that detect changes in structures' mass and stiffness properties would be developed. Finite element methods are typically used to model the structures in their original state as well as to extract information on the respective dynamic properties such as frequencies and mode shapes in the changed state. The force vector derived from the residual force will be investigated to locate changed regions in the structure. Frequencies difference between original state and changed state will be used to determine amount of change.

2. Structural changes detection technique

To use this technique dynamic properties such as natural frequencies and mode shapes of structure have to be obtained. This can be done by measuring or using finite element method to modeling original and changed structure. In this work natural frequencies and mode shapes are obtained using finite element method and all computations were done in MATLAB. Eigenvalues and eigenvectors are derived from characteristic equation

$$(\mathbf{K} - \lambda_i \mathbf{M}) \boldsymbol{\varphi}_i = 0 \quad (1)$$

where λ_i is the i th mode eigenvalue ($\lambda_i = \omega_i^2$, ω_i – natural frequencies), and $\boldsymbol{\varphi}_i$ is i th eigenvector. Equation (1) is used to obtain dynamic properties in original and changed state. Stiffness and mass matrices in changed state $\mathbf{K}_c$ and $\mathbf{M}_c$ are defined as

$$\mathbf{K}_c = \mathbf{K} + \Delta \mathbf{K}, \mathbf{M}_c = \mathbf{M} + \Delta \mathbf{M} \quad (2)$$

where $\Delta \mathbf{K}$ and $\Delta \mathbf{M}$ are change of original stiffness and mass matrices. Now, substituting equation (2) into equation (1) and rearranging residual force vector is obtained

$$\mathbf{R}_i = \Delta \mathbf{K} \boldsymbol{\varphi}_{ci} - \lambda_{ci} \Delta \mathbf{M} \boldsymbol{\varphi}_{ci} = (-\mathbf{K} + \lambda_{ci} \mathbf{M}) \boldsymbol{\varphi}_{ci} \quad (3)$$

Residual force for i th degree of freedom is zero if changed in stiffness and mass matrices are zero. To determine the amount of changes, natural frequencies and mode shapes in changed state are defined as

$$\lambda_{ci} = \lambda_i + \Delta \lambda, \boldsymbol{\varphi}_{ci} = \boldsymbol{\varphi}_i + \Delta \boldsymbol{\varphi} \quad (4)$$

Substituting equation (4) and (2) into equation (1) for changed state, neglecting higher order terms, pre-multiplying all remaining terms by $\boldsymbol{\varphi}_i^T$, doing necessarily mathematical operations, and if $\boldsymbol{\varphi}_i$ is normalized following equation is obtained

$$\boldsymbol{\varphi}_i^T \Delta \mathbf{K} \boldsymbol{\varphi}_i - \lambda_i \boldsymbol{\varphi}_i^T \Delta \mathbf{M} \boldsymbol{\varphi}_i = \Delta \lambda_i \quad (5)$$

Reducing $\Delta \mathbf{K}$, $\Delta \mathbf{M}$ and $\boldsymbol{\varphi}_i$ to $\Delta \mathbf{K}'$, $\Delta \mathbf{M}'$ and $\boldsymbol{\varphi}_i'$ matrices which are associated with changed elements, and if matrices $\Delta \mathbf{K}'$ and $\Delta \mathbf{M}'$ are expressed as function of original element stiffness and mass matrices $\mathbf{K}_j^e$ and $\mathbf{M}_j^e$

$$\Delta \mathbf{K}' = \sum_{j=1}^{n_c} \mathbf{K}_j^e \delta k_j, \Delta \mathbf{M}' = \sum_{j=1}^{n_c} \mathbf{M}_j^e \delta m_j \quad (6)$$

where n_c is the number of changed element and δk_j and δm_j are stiffness and mass proportional modification factors, equation (5) can be used to determine amount of change i.e. δk_j and δm_j .

3. Conclusion

Executed numerical examples validate presented technique. Pseudo force vector have all values less than 10^{-8} except for nodes which unique determine changed element, Table 1. Obtained results for amount of change are very near actual amount of change, as shown in Table 2.

Table 1. Pseudo force vector

Dof	15% stiffness reduction	15% mass reduction	15% stiffness and 15% mass reduction
22	$9.65 \cdot 10^4$	$1.74 \cdot 10^4$	$1.49 \cdot 10^5$
23	$8.39 \cdot 10^5$	$3.10 \cdot 10^4$	$8.91 \cdot 10^5$
24	$6.61 \cdot 10^4$	$2.19 \cdot 10^3$	$5.70 \cdot 10^4$
28	$9.65 \cdot 10^4$	$2.01 \cdot 10^4$	$1.33 \cdot 10^5$
29	$8.39 \cdot 10^5$	$3.55 \cdot 10^4$	$7.86 \cdot 10^5$
30	$6.07 \cdot 10^4$	$3.49 \cdot 10^3$	$6.10 \cdot 10^4$

Table 2. Modification factors – amount of change

Modification factors	15% stiffness reduction	15% mass reduction	15% stiffness and 15% mass reduction
δk	-0.161	0.0038	-0.156
δm	0.0053	-0.1503	-0.143

References

- [1] Farrar, C.R. and Doebling, S.W., “An Overview of Modal-Based Damage Identification Methods”, Euromech 365 International Workshop: DAMAS 97, Sheffield UK, 1997
- [2] Chandrupatla, T.R. and Belegundu, A.D., “Introduction to Finite Elements in Engineering“, Prentice-Hall Inc., New Jersey, 1991
- [3] Ge, M. and Lui, E.M., “Structural damage identification using system dynamic properties”, Computers and Structure, 83, 2005, 2185-2196.
- [4] Hatch, M.R., “Vibration Simulation using MATLAB and ANSYS”, Chapman & Hall, 2001
- [5] Meirovitch, L., “Elements of Vibration Analysis”, McGraw-Hill, 1986

Zeljko Lozina. Dr. sc., MechEng, Full Professor
University of Split, Faculty of Electrical and Mechanical Engineering and Naval Architecture, Rudjera Boskovic bb, HR-21000 Split, Croatia, tel: + 385 21 305890, fax: + 385 21 463877, e-mail: zeljan.lozina@fesb.hr

Damir Vucina. Dr. sc., MechEng, Full Professor
University of Split, Faculty of Electrical and Mechanical Engineering and Naval Architecture, Rudjera Boskovic bb, HR-21000 Split, Croatia, tel: + 385 21 305890, fax: + 385 21 463877, e-mail: damir.vucina@fesb.hr

Damir Sedlar, MechEng, Assistant
University of Split, Faculty of Electrical and Mechanical Engineering and Naval Architecture, Rudjera Boskovic bb, HR-21000 Split, Croatia, tel: + 385 21 305864, fax: + 385 21 463877, e-mail: damir.sedlar@fesb.hr

THE MULTISCALE MATERIAL BEHAVIOR MODEL

Ante Agić, Budimir Mijović and Veljko Filipan

Keywords: Multiscale modelling, quasicontinuum, nanoindentation

1. Introduction to quasicontinuum

The multiscale models generally use a coarse-fine grain decomposition for the physical domain under analysis[1]. The small fine grain subregion with strong deformation gradients, are captured by the individual atomistic simulation. The large peripheral coarse grain region where the deformation is to be homogeneous and smooth a continuum simulation is used [2]. The displacement of small fraction of the atoms (called representative atoms or 'repatoms') can be treated explicitly , with the displacements of the remaining atoms approximately found through finite element interpolation procedure. Deformed configuration defined by nodal coordinates system $x = \{x_i, i = 1, \dots, N\}$, while undeformed referential coordinate system defined by $X = \{X_i, i = 1, \dots, N\}$. The extended variational problem become [5]

$$\min_{x, X} \Pi(x, X) \quad (1)$$

where Π is total potential energy. The minimization procedure has the effect of equilibrating the body; whereas the later minimization has the effect of optimizing the nodal positions of the triangulation

2. Nanoindentation example

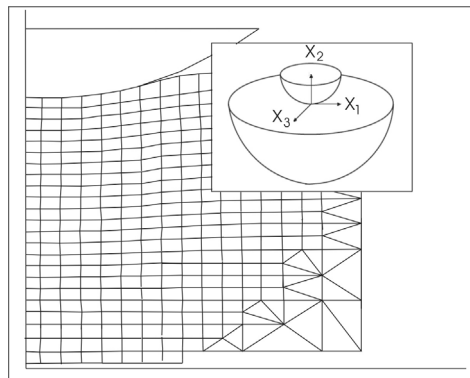


Figure 1. The indentation test model

A single crystal of Ni with dimensions 100 x 100 nm was indented by a spherical indenter of radius 10 nm, as shown by Figure 1. The model was implemented in the original version of the QC code described in [6], [7]. The procedure is based on the concept of a mean force potential for coarse-grain medium. The loading curves as result of simulation are presented in Figure 2. The results compared with data from reference [9]. The solid line in Figure 2 is a fit to the elastic solution of Hertz. The calculated and experimental measured elastic constants are about 20 % difference.

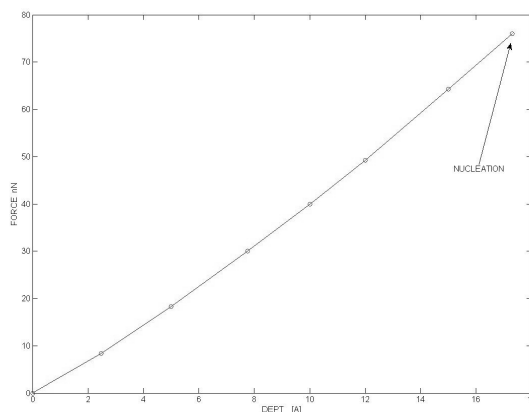


Figure 2 The force - indentation dept curve

References

- [1] Tadmor, E. B., Ortiz, M. and Phillips, R., Quasicontinuum analysis of defects in solids, *Phil. Mag. A*, Vol. 73, 1996, pp. 1529-1563.
- [2] Shilkrot, L. E. and Miller, R. E. and Curtin, W. A., Multiscale plasticity modeling coupled atomistic and discrete dislocation mechanics, *J. Mech. Phys. Solids*, Vol. 52, 2004, pp. 755-787.
- [3] Wagner, G. J. and Liu, W. K. Coupling of atomistic and continuum simulations using a bridge scale decomposition, *J. Comput. Phys.*, Vol. 190, 2003, pp. 249-274.
- [4] Medyanik, S. N., Karpov, E. G. And Liu, W. K. Domain reduction method for atomistic simulations, *J. Comput. Phys.*, Vol. xx, 2006, pp. xx-xx.
- [5] Thoutireddy, P. and Ortiz, M. A variational r-adaption and shape-optimization method for finite deformation elasticity, *Int. J. Numeric. Methods in engineering*, 61, 2004, pp. 1-21.
- [6] Miller, R. and Tadmor, E., Quasicontinuum Method Website, <http://www.qcmethod.com>
- [7] Miller, R. E. and Tadmor, E. B. QC-Reference Manual Version 1.2, Brown University, 2001
- [8] Tadmor, E. B., Miller, R. and Phillips, R., Wilkinson, P., Nanoindentation and incipient plasticity, *J. Mater. Res.* Vol. 14, 1999, pp. 2233-2250.
- [9] Dupuy, L. M., Tadmor, E. B., Miller, R. E. and Phillips, Finite temperature Quasicontinuum: Molecular dynamics without all the atoms, *Phys. Rev. Letters*, 95, 2005, pp-60202-60207.
- [10] Nanoindentation Simulation website, <http://www.gpm2.inpg.fr/axes/plast/MicroPlast/ddd/nano.html>

Ante Agić, Assistant Professor

Faculty of Chemical Engineering and Technology/University of Zagreb, Department of Surface Engineering, Marulicev trg 19, Zagreb, Croatia, ++385 1 4816970, aagic@fkit.hr

Budimir Mijović, Professor

Faculty of Textile Technology/University of Zagreb, Department of Fundamental, Natural and Technical Sciences, Baruna Filipovica 30, Zagreb, Croatia, ++385 1 3712 500, budimir.mijovic@tft.hr

Veljko Filipan, Associate Professor

Faculty of Chemical Engineering and Technology/University of Zagreb, Department of Energy Engineering, Marulicev trg 19, Zagreb, Croatia, ++385 1 4597109, vfilipan@fkit.hr

Numerical modelling of contact between elastic and rigid body

Budimir Mijović, Ante Agić

Keywords: Contact, augmented Lagrangian, finite element

1. Introduction

Most engineering products are made up of assembled parts and some of these parts are mechanically joined. In order to predict accurately the strength and reliability, or any other structural characteristic of the joints it is important to know the contact condition between them. Hertz first made a satisfactory analysis for the contact problem of two elastic bodies [1]. A lot of research effort is currently focussed on the development of modern numerical algorithms for analysis of contact problems, [2]. There are several improvements in attempts to solve the general contact problem using the finite element computations with penalty method or Lagrange multiplier methods, [3]. In this paper, we presented the augmented Lagrangian approach to contact problems involving contact with friction. minimization has the effect of optimizing the nodal positions of the triangulation this document from the Congress's web site. Figures, tables, photos etc. must be gray-scaled.

2. Numerical example

Non-conforming bodies have been used with small deformation. We suppose that the surfaces are topographically flat in micro and macro sense. Friction on the surface of two non-conforming bodies in contact has role only if elastic constants of the two materials are different. The global (Lagrangian) variational principal for the two-body contact system has the following form

$$\int_{\Omega} \sigma_{ij} : \delta \varepsilon_{ij} d\Omega = \int_{\Omega} \bar{f}_v \circ \delta \bar{u} d\Omega + \int_{\Gamma} \bar{f}_s \circ \delta \bar{u} d\Gamma + \int_{\Gamma_c} (-F_N \cdot \bar{n} \circ \delta \bar{u} - F_T \cdot \bar{t} \circ \delta \bar{u}_T) d\Gamma_c \quad (1)$$

Where are σ_{ij} and ε_{ij} stress and strain tensors, respectively. We denote the displacement field over all contacting bodies $\Omega = \Omega_c \cup \Omega_T$ by $\bar{u}$, (figure 1).

The contact problem between an long solid elastic cylinder resting on a rigid plate is analyzed. The cylinder is subjected to a downward concentrated load F on the top. Under plane simetry condition 3D (three dimensional) problem replaced by 2D plane strain problem. The half-model of the cylinder is discretized by triangular finite elements. The plate surface is discretized by set target segment elements. Contact element is used to represent contact and sliding between target surface and cylinderas contractor.

On Figure 2 are shown finite element solution, for the same load and boundary conditions as above. The visual inspection both two figures show good agreement mostly everywhere.

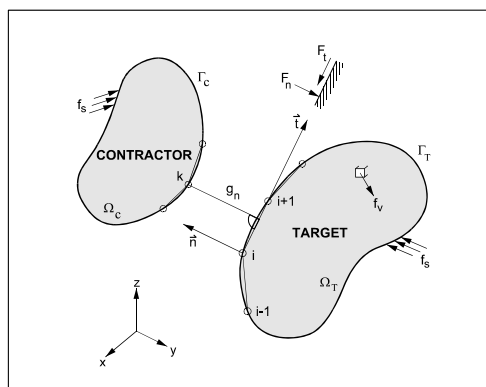


Figure 1 Contact between two solids

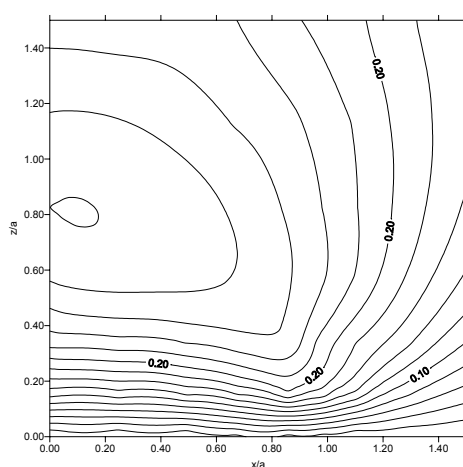


Figure 2. Contours of principal shear stresses in x-z plane – (Finite element solution)

References

- [1] Hertz H., Ueber die Berührung fester elastischer Körper, Journal für die reine und angewandte Mathematik, 92, 156-171, (1882).
- [2] Johnson, K. L., Contact Mechanics, Cambridge University Press, Cambridge, 1985.
- [3] Nackenhorst, U., Rollkontaktdynamik, Habilitationsschrift, Universität der Bundeswehr Hamburg, Hamburg, 2000.
- [4] Simo, J. C. and Laursen, T. A., An augmented Lagrangian treatment of contact problem involving friction, Computers & Structures, 42, 97-116, (1992).

Budimir Mijović, Professor

Faculty of Textile Technology/University of Zagreb, Department of Fundamental, Natural and Technical Sciences, Baruna Filipovica 30, Zagreb, Croatia, ++385 1 3712 500, budimir.mijovic@ttf.hr

Ante Agić, Assistant Professor

Faculty of Chemical Engineering and Technology/University of Zagreb, Department of Surface Engineering, Marulicev trg 19, Zagreb, Croatia, ++385 1 4816970, aagic@fkit.hr

AN EFFICIENT MODELLING OF ANISOTROPIC LARGE STRAIN ELASTOPLASTIC DEFORMATION RESPONSES

Igor Karšaj, Carlo Sansour, and Jurica Sorić

Keywords: multiplicative inelasticity, anisotropic elastic law, anisotropic yield function, exponential map

1. Introduction

In recent years, the topic of anisotropic inelasticity at finite strains has been attracting considerable attention because of its relevance to the deformation of structural components and the many still open questions encountered in its modelling and computation. Although considerable progress has been made in material modelling at meso and micro scales, it is not yet possible to develop efficient numerical algorithm for whole structural components. Hence, approaches based on phenomenological continuum mechanics are still attractive.

In this paper a formulation of multiplicative anisotropic elastoplasticity is presented. In order to achieve formulation invariant to finite rotations, $\mathbf{F}_p$ is assumed to be material-like tensor. An anisotropic elastic constitutive law is described by means of free energy that is function of elastic right Cauchy-Green tensor and structural tensors defining material orientation. This dependency is formulated by using so called integrity basis. An anisotropic yield function, described by non-symmetric Eshelby-like stress tensor and further structural tensors, is developed. In order to obtain yield function as an isotropic function of material invariants, integrity basis are made by using theorems for functions with non-symmetric arguments. The model considers non-linear isotropic hardening, as well. An accurate integration algorithm employing exponential map is developed. In order to ensure a high convergence rate in the global iteration approach, an algorithmic tangent operator for anisotropic multiplicative inelasticity is derived for the first time. Various numerical examples show robustness and efficiency of the proposed algorithm. The influence of anisotropy on finite elastoplastic deformation responses is also considered. Detailed description of this material model can be found in [1] and **Pogreška! Izvor reference nije pronađen..**

2. Formulation of anisotropy

The kinematics of large strain inelastic deformations is based on multiplicative decomposition of the deformation gradient on the elastic and plastic parts, $\mathbf{F}=\mathbf{F}_e\mathbf{F}_p$. $\mathbf{F}_p$ is assumed to be material tensor. Furthermore, expressions for right Cauchy-Green tensor $\mathbf{C}$ and its elastic part $\mathbf{C}_e$ are used. The time derivatives of the deformation gradient, $\mathbf{L}$, and its inelastic part, $\mathbf{L}_p$, are formulated.

The elastic behaviour of the body is fully characterised by means of free energy function, ψ , Table 1. The main idea is to develop irreducible set of material invariants, J_i and J_{i+3} . By using these invariants it is possible to describe anisotropic behaviour with isotropic function for elastic free energy. Having identified the free energy function, the evaluation of the dissipation inequality, D , follows, from which the thermodynamical forces for Eshelby-like stress tensor, Ξ , and isotropic hardening, Y are derived. Furthermore, dissipation can be written in reduced form, D_r .

Anisotropic inelastic material behaviour is described through a yield function, ϕ , which is given as function of material invariants I_i , I_{i+3} and I_{i+6} . By using the principle of maximum dissipation, evolution equations for inelastic right rate, $\mathbf{L}_p$, and rate of internal isotropic hardening variable, Z , are derived. Herein, λ is plastic multiplier and $\mathbf{M}_e^i$ and $\mathbf{M}_y^i$ are elastic and inelastic structural ten-

sors, respectively.

Table 1 Formulation of multiplicative anisotropy

$\psi = \psi_e(\mathbf{C}_e, \mathbf{M}_e^i) + \psi_h(Z)$ $J_i = \text{tr}(\mathbf{M}_e^i \mathbf{C}_e), J_{i+3} = \text{tr}(\mathbf{M}_e^i \mathbf{C}_e^2)$	$I_i = \text{tr}[\mathbf{M}_y \text{dev } \mathbf{\Xi}], \quad i = 1, 2, 3$ $I_{i+3} = \text{tr}[\mathbf{M}_y (\text{dev } \mathbf{\Xi})^2], \quad i = 1, 2, 3$ $I_{i+6} = \text{tr}[\mathbf{M}_y \text{dev } \mathbf{\Xi} (\text{dev } \mathbf{\Xi})^T + (\text{dev } \mathbf{\Xi})^T \text{dev } \mathbf{\Xi}]$
$D = \mathbf{\Xi} : \mathbf{L} - \rho_e \left(\frac{\partial \psi}{\partial \mathbf{C}_e} : \dot{\mathbf{C}}_e + \frac{\partial \psi}{\partial Z} \cdot \dot{Z} \right) \geq 0$	$\phi = \sqrt{\frac{2}{3}} \left[\sigma_{11}^o \sqrt{\chi} - (\sigma_{11}^o - Y) \right]$ $Y = -HZ - (\sigma_{\infty}^o - \sigma_{11}^o) [1 - \exp(-\eta Z)]$
$\mathbf{\Xi} = 2\rho_0 \mathbf{C} \mathbf{F}_p^{-1} \frac{\partial \psi}{\partial \mathbf{C}_e} \mathbf{F}_p^{-T}$ $Y = -\rho_0 \frac{\partial \psi}{\partial Z}$ $D_r = \mathbf{\Xi} : \mathbf{L}_p + Y \cdot \dot{Z} \geq 0$	$\mathbf{L}_p = \lambda \frac{\partial \phi}{\partial \mathbf{\Xi}}$ $\dot{Z} = \lambda \frac{\partial \phi}{\partial Y}$

3. Numerical example

3.1. Circular plate subjected to conservative load

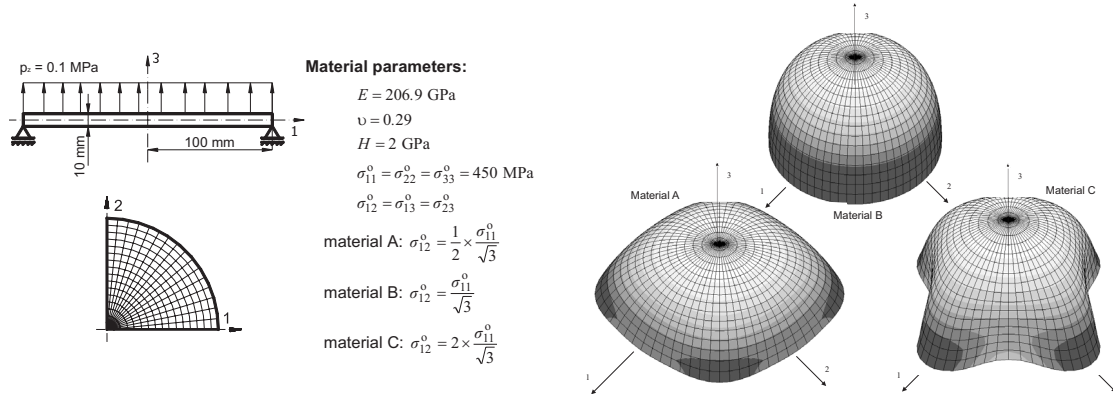


Figure 5 Geomety, material data and deformed configurations

Circular plate is subjected to a uniform conservative load. Geometry and material data are shown in Figure 1. Due to the geometrical and material symmetry one quarter of the plate is discretized by 16×16 elements. Figure 1 depicts the deformed configurations for the three different materials. From the deformed configurations, it can be noticed that the plastic strains in material A are concentrated in the direction of 45°, while in material C the concentration is in the directions 1 and 2, as expected

References

- [1] Karšaj I, Numerical modelling of finite strain elastoplastic deformations, doctoral dissertation, Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, 2006, Zagreb
- [2] Sansour C. Karšaj I. and Sorić J., "A formulation of anisotropic continuum elastoplasticity at finite strains. Part I: Modelling", International Journal of Plasticity, submitted

Igor Karšaj, Ph.D., Prof. dr.sc. Jurica Sorić, University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, Zagreb, Ivana Lučića 5, Croatia, +38516168514, igor.karsaj@fsb.hr, jurica.soric@fsb.hr, Prof. Dr.-Ing. Carlo Sansour, The University of Nottingham, School of Civil Engineering, University Park, Nottingham NG7 2RD, UK

A robust meshless formulation for numerical analysis of shell-like structural components

Tomislav Jarak, Jurica Sorić, Josip Hoster

Keywords: *meshless formulation, Meshless Local Petrov-Galerkin Method, plates, shells, three-dimensional solid concept, moving least squares approximation*

1. Introduction

The Meshless Local Petrov-Galerkin (MLPG) approach described in [1], which requires no elements or background cells for either interpolation or integration, is applied for analyses of plates and shells. The solid-shell concept is applied, allowing the use of a complete 3-D constitutive model. The local symmetric weak form (LSWF) of equilibrium equations over a local sub-domain is derived by assuming linear test functions in the thickness directions. Thereby, the essential boundary conditions are enforced by a penalty method. According to the FEM formulation in [2], simple polynomial functions are utilized to approximate the displacement field in the thickness direction, while the moving least squares (MLS) interpolation [1] is used in the in-plane directions. The exact shell geometry description may be performed. The thickness locking effect is eliminated by using the hierarchical quadratic interpolation over the thickness for the transversal displacement component [3]. The shear and membrane locking effects appearing in the case of thin structures and the sensitivity to the rigid body motion due to the exact geometry description are minimized by applying the MLS basis functions of sufficiently high order. The efficiency of the proposed formulations is demonstrated by numerical examples.

2. Meshless formulation

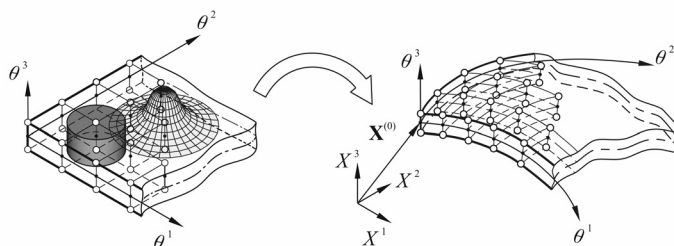


Figure 1. Discretization of shell continuum and mapping from parametric space to the Cartesian coordinate system

Discretization of the plate or shell continuum is carried out in the parametric space of convective coordinates θ^i by sets of nodes located on the upper and lower surface. The discretized continuum is then mapped in the global Cartesian coordinate system, as shown in Figure 1. Thereby, the plate continuum can be considered as a special case of shell structure. According to the MLPG method, a cylindrical local sub-domain Ω_s^I is constructed in the parametric space around a couple of nodes I , located on the upper and lower surfaces, over which a local weak form of 3-D equilibrium is written as

$$\int_{\Omega_s^I} (\sigma_{ij,j} + b_i) v_{ki} d\Omega - \alpha \int_{\Gamma_{su}^I} (u_i - \bar{u}_i) v_{ki} d\Gamma = 0, \quad I = 1, 2, \dots, N, \quad (1)$$

with $v_{ki} = \delta_{ki} v(\theta^j)$, where δ_{ki} is the Kronecker delta and $v(\theta^j)$ denotes an arbitrary test function. Γ_{su}^I is a part of the local sub-domain boundary with the prescribed displacement $\bar{u}_i$, and α denotes a penalty parameter, $\alpha \gg 1$. N is the total number of nodes used for the discretization. Linear test function in the thickness direction yields the LSWF over Ω_s^I consisting of six equations. Since deformation responses of the shell-like continuum are described with seven variables through the thickness, the additional equation is obtained by employing the collocation method inside Ω_s^I at the mid-point located on the middle surface between the nodes on the upper and the lower surfaces. A discretized system of linear equations is then obtained by employing both the MLS approximation and polynomial functions.

3. Numerical example

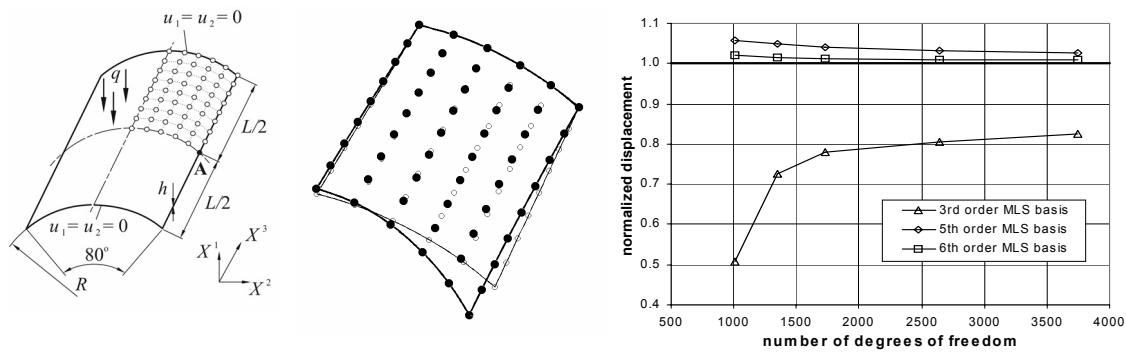


Figure 2. Scordelis-Lo roof: geometry and discretization (right), deformed shape (center) and convergence rate of the vertical displacement at the middle surface point A (right)

As an example, Scordelis-Lo roof subjected to the uniform vertical load of $q = 0.00625$ is analysed. The radius of the roof is $R = 3000$, the length is $L = 6000$ and the thickness is $h = 30$. Material data are Young's modulus $E = 30000$ and Poisson's ratio $\nu = 0.0$. Due to symmetry, only one quarter of the shell is discretized by uniform grids, as shown in the Figure 2. The convergence rate of the vertical displacement at the middle surface point A normalized by the analytical value is displayed in the diagram for different MLS basis functions. As expected, the shear locking effect is minimized by using the MLS basis function of a sufficiently high order.

References

- [1] Atluri S.N., Shen S., "The meshless local Petrov-Galerkin (MLPG) method", Tech Science Press, Encino, USA, 2002
- [2] Hauptmann R., Schweizerhof K., "A systematic development of solid-shell element formulations for linear and non-linear analyses employing only displacement degrees of freedom, Int. J. Numer. Meth. Engng., Vol. 42, 1998, pp 49-69
- [3] Sorić J., Li Q., Jarak T., Atluri S.N., "Meshless Local Petrov-Galerkin (MLPG) Formulation for Analysis of Thick Plates", CMES: Comput. Model. Eng. Sci., Vol 6, No. 4, 2004, pp 349-357

Tomislav Jarak, Dipl.ing.

Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Chair of Mechanics and Strength of Materials, Ivana Lučića 5, Zagreb, Croatia, Tel: +385 1 6168 514, E-mail: tomislav.jarak@fsb.hr

Jurica Sorić, Professor
Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Chair of Mechanics and Strength of Materials, Ivana Lučića 5, Zagreb, Croatia, Tel: +385 1 6168 103, E-mail: jurica.soric@fsb.hr

Josip Hoster, Dipl.ing.
Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Chair of Mechanics and Strength of Materials, Ivana Lučića 5, Zagreb, Croatia, Tel: +385 1 6168 115, E-mail: josip.hoster@fsb.hr

STRUCTURAL INTEGRITY ASSESSMENT OF T-JOINTS IN STEAM PIPELINES

N. Gubelj, D. Kozak, J. Predan, P. Konjatić,
M. Oblak, F. Matejiček, A. Sedmak, J. Vojvodič-Tuma

Keywords: Integrity assessment, steam pipeline, T-joint, finite element analysis

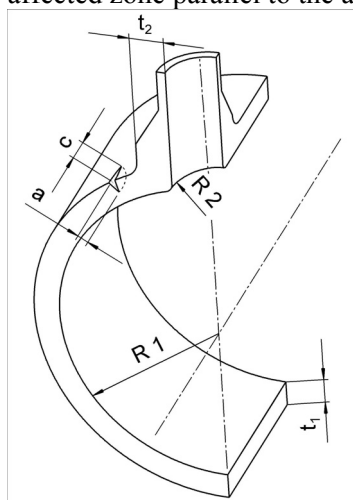
1. Introduction

Structure integrity assessment is issued to failure prediction of cracked structural component regarding to increasing applied load or crack size. In order to ensure safe use of cracked structural component the many standards and procedures are available (e.g. SINTAP), cited as ref. [1]. The results obtained by using of these procedures consist always certain portion of conservatism, what leads to underestimation of the real carrying capacity of engineering component.

In this paper a T-joint of 8 inch RHR nozzle on 29 inch RC hot leg as the part of a high pressured pipeline of the nuclear power plant has been considered as engineering component. Some experiences of operators show that a semi-elliptical surface crack can occur at the transition between two pipes of different diameters during exploitation of steam pipelines.

2. Problem definition

T-joint of pipes as the part of primary pipeline of the power plant (Fig. 1) is subjected to the operating pressure of 17,133 MPa and temperature of +350° C. Weld joint has been made from the material with higher strength than base material (yield strength $R_{p0.2} = 270$ MPa, ultimate tensile strength $R_m = 575$ MPa, elastic modulus $E = 185$ GPa, Poisson's number $\nu = 0,3$ and fracture toughness $CTOD_{mat} = 0,75$ mm) [2], what could be the reason for crack appearing in the base metal at the vicinity of the joint. The possible location of the semi-elliptical surface crack is in the heat affected zone parallel to the axis of symmetry of the pipe with greater diameter.



$R_1 = 370,2\text{mm}$
 $t_1 = 59,18\text{mm}$
 $R_2 = 88,875\text{mm}$
 $t_2 = 79,45\text{mm}$
 $a = 19\text{mm}$
 $2c = 100\text{mm}$

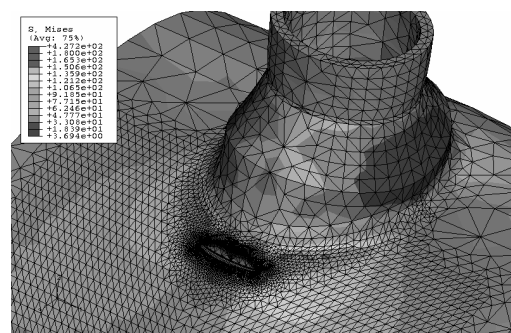


Figure 1. Geometry of the T-joint

Figure 2. Mises stress distribution in the T-joint

3. Determination of the carrying capacity of the T-joint of pipes

The aim of this paper was to compare three different approaches defining the carrying capacity of one cracked T-joint in steam pipes:

- Engineering approach based on the stress intensity factor (SIF) calculation by using equations given in the literature and an expression for the hoop stress
- Combined approach based on the analytical SIF calculation by using arbitrary stress distribution through the thickness obtained from the FEA for the structure with no crack and
- Numerical calculation of SIF from the FE model of joint with real crack geometry (Fig. 2).

Figure 3 shows comparison between numerically determined loading paths against failure assessment curve. Since, linear hoop-solution cross the failure curve at lower internal pressure value than FEM solution, it is obvious that hoop solution is most conservative compare to arbitrary stress solution and FE solutions, e.g. critical pressure by hoop stress is $p^*_{crit}=45$ MPa, arbitrary stress solution $p^{**}_{crit}=48$ MPa while by FEM analysis, the critical stress is $p_{crit}=53,6$ MPa for surface point.

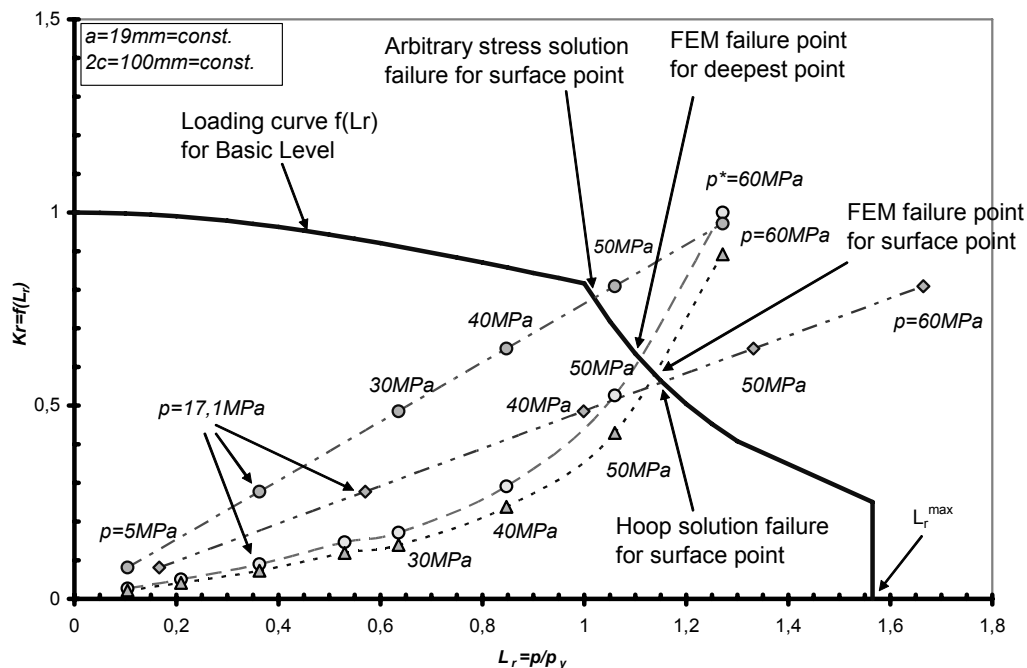


Figure 3. Comparison between hoop stress solution, arbitrary stress solution and FEM solution during increasing internal pressure

References

- [1] SINTAP: Structural INTEgrity Assessment Procedures for European Industry', Final Revision, EU Project BE 95-1462, Brite Euram Programme, 1999
- [2] Gubelj, N.; Kozak, D.; Vojvodić-Tuma, J.; Predan, J., Aplikacija mehanike loma pri revitalizaciji energijskih komponent, Končno poročilo, Elaborat za nuklearno elektrarno Krško, 2005

Nenad Gubelj, Assoc. Prof.; Jožef Predan, PhD and Maks Oblak, Professor
University of Maribor, Faculty of Mechanical Engineering, Smetanova 17, SI-2103 Maribor, Slovenia,
Telephone: +386 2 220 7661, Telefax: +386 2 220 7990, e-mail: nenad.gubelj@uni-mb.si
Dražan Kozak, Assoc. Prof.; Pejo Konjatić, BSc and Franjo Matejiček, Prof.
J. J. Strossmayer University of Osijek, Mechanical Engineering Faculty, Trg I. B. Mažuranić 2, HR-35000
Slavonski Brod, Croatia, Telephone: +385 35 446 188, Telefax: +385 35 446 446, e-mail: dkozak@sfsb.hr
Aleksandar Sedmak, Prof.
University of Belgrade, Faculty of Mechanical Engineering, Kraljice Marije 16, SER-11120 Belgrade 35,
Serbia, Mob: +381 64 816 6010, e-mail: asedmak@eunet.yu
Jelena Vojvodić-Tuma, Assist. Prof.
Institute of Metals and Technology, Lepi pot 11, SI-1001 Ljubljana, Slovenia, Telephone: +386 1 4701930,
Telefax: +386 1 4701800, e-mail: jelena.tuma@imt.si

Effect of asymmetric density-temperature relationship of water on natural convection in rectangular Enclosure

M. Živić, M. Šavar and S. Krizmanić

Keywords: Natural convection in water, Symmetric and Asymmetric density- temperature relationships

1. Introduction

The present study aims at revealing the effect of density inversion on transient natural convection in water in a rectangular enclosure [1,2], because it is relevant to freezing of water or to melting of ice. It is considered a rectangular enclosure with two vertical walls maintained at constant different temperatures, T_h and T_c and two horizontal, insulated walls filled with stagnant water at initial temperature T_{in} . The investigation is performed with three different initial temperatures $T_{in}=2, 4$, and $6\text{ }^{\circ}\text{C}$, with two density-temperature relationships of water, the symmetric [1], and the asymmetric [3]. Basic dimensionless parameters describing the problem are: the Prandtl number, $Pr = 11,57$, and the Rayleigh number, $Ra = 5 \cdot 10^5$.

The problem is governed by the basic laws represented by the continuity, momentum and energy equations and with the following assumptions: the flow is laminar and incompressible, the thermo physical properties are independent of temperature except for the density in the buoyancy term and the viscous dissipation is neglected [4]. The mathematical model is solved by the finite volume method on staggered grid and the SIMPLER algorithm is used [5].

2. Results and discussion

The difference between the cases of the symmetric and the asymmetric density-temperature relationship in terms of steady state temperature field is shown in Fig. 1 while the streamlines for the symmetric and the asymmetric cases are shown in Figs 2a and 2b. As expected the flow pattern and the temperature field are asymmetric for the asymmetric density-temperature relationship. In both cases the flow structure and the temperature field in a steady state are independent of initial temperature of water.

The behavior of the transient average Nusselt number at the hot vertical wall is displayed in Figs. 3a for symmetric and 3b for asymmetric density-temperature relationship. It shows that, at the beginning of the process, the Nusselt number decreases steeply within a short time for the cases $T_{in}=2\text{ }^{\circ}\text{C}$ and $T_{in}=6\text{ }^{\circ}\text{C}$. At the point where the heat transfer mode changes from conduction-dominated to convection-dominated at larger times, the slope of the Nusselt curve becomes flat, and at very long time reaches a constant value. In both cases the average Nusselt number at the hot wall in a steady state is practically the same $Nu_h=4.022$. There is a great difference in time variation of Nusselt number. This difference in reaching steady state is the subject of further investigations.

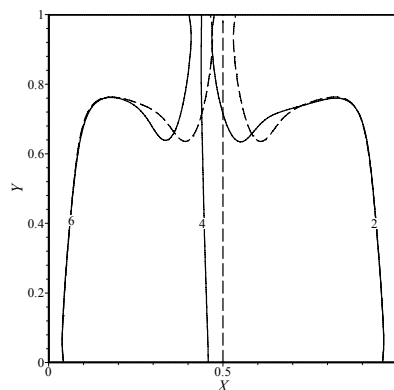


Figure 1. Isotherms (2, 4, 6 °C) for symmetric (dashed line) and asymmetric (solid line) density-temperature relationship of water in a steady state

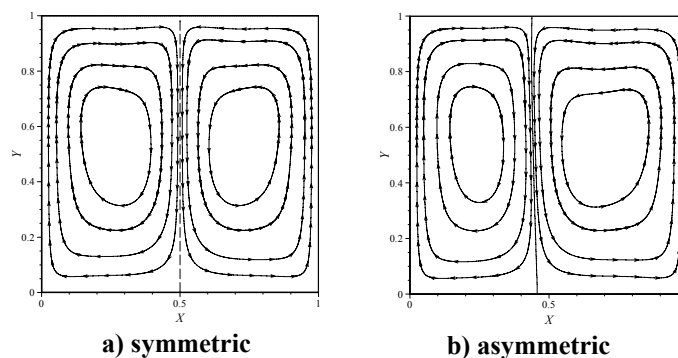
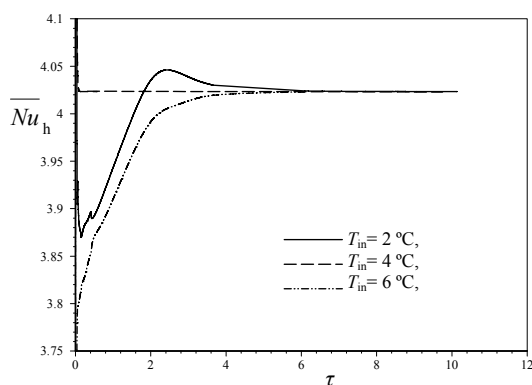
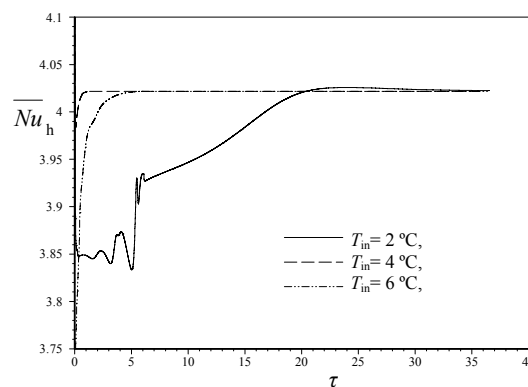


Figure 2. Streamlines for the considered density-temperature relationships of water in a steady state



a) symmetric



b) asymmetric

Figure 3. Average Nusselt number at the hot wall for the considered density temperature relationships

References

- [1] Vasseur, P., Robillard, L., "Transient natural convection heat transfer in a mass of water cooled through 4°C", Int. J. Heat Mass Transfer, Vol.23, 1980, pp 1195-1205.
- [2] Tong, W., Koster, J. N., "Density inversion effect on transient natural convection in a rectangular enclosure", Int. J. Heat Mass Transfer, Vol.37, 1994, pp 927-938.
- [3] Debler, W.R., "On the analogy between thermal and rotational hydrodynamic stability", J. Fluid Mechanics, Vol.24, 1966, pp 165-176.
- [4] Virag, Z., Živić, M., Galović, A., Influence of natural convection on the melting of ice block surrounded by water on all sides, Int. J. Heat Mass Transfer, 2006, (in press).
- [5] Patankar, S.V., "Numerical Heat Transfer and Fluid Flow", Hemisphere, McGraw- Hill, New York, 1980.

Marija Živić, Assistant Professor

J.J. Strossmayer University of Osijek, Mechanical Engineering Faculty, Trg I.B. Mažuranić 18, 35 000 Slavonski Brod, Croatia, tel.(035)446188, fax: (035)446446, e-mail: marija.zivic@sfsb.hr

Mario Šavar, Associate Professor

University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, I. Lučića 5, 10 000 Zagreb, Croatia, tel.(01)6168438, fax: (01) 61 56 940, e-mail: mario.savar@fsb.hr

Severino Krizmanić,

University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, I. Lučića 5, 10 000 Zagreb, Croatia, tel.(01)6168156, fax: (01) 61 56 940, e-mail: severino.krizmanic@fsb.hr

PLASTIC COLLAPSE ANALYSIS OF SURFACE CRACKED STEAM GENERATOR TUBES

Ivica Skozrit, Zdenko Tonković and Ivo Alfiredić

Keywords: *Finite element analysis; Steam generator tubes; Surface cracks; Limit pressure; Collapse pressure; Flow stress*

Summary

Operating experience with steam generators has shown that axial surface cracks present one of the most common causes of loss of steam generator tube integrity. An accurate computation of the plastic limit pressure represents a key for the prediction of structural integrity and reliability of pressurized tubes. A greater number of existing solutions for limit pressure of a cracked tube have been developed either analytically, based on a simple equilibrium stress field, or empirically, based on test data [1]. These solutions are generally shown to be too conservative for components with part-through-thickness defects. Recently, a finite element based plastic limit pressure expression for cylinders with external axial semi-elliptical surface cracks has been developed in Ref. [2]. However, the proposed expression is applicable to a very limited range of crack dimensions, and therefore the new extended solutions are desirable.

In this paper, three-dimensional non-linear finite element analyses have been carried out to determine the plastic limit and collapse loads for external axial semi-elliptical surface cracks in VVER steam generator tubes under internal pressure p . The tube geometry and loading are shown in Fig. 1. The outer radius of the VVER tube R_0 is 8 mm and the wall thickness t is 1,5 mm. The crack is assumed to have a semi-elliptical shape described by a length $2c$, depth a and normalized crack length ρ , defined as $\rho = c / \sqrt{R_m t}$. Six different crack lengths were considered, $2c = 5, 10, 20, 30, 40$ and 50 mm, and four values of the ratio of the crack depth to the tube thickness were selected, $a/t = 0.2, 0.4, 0.6$ and 0.8 .

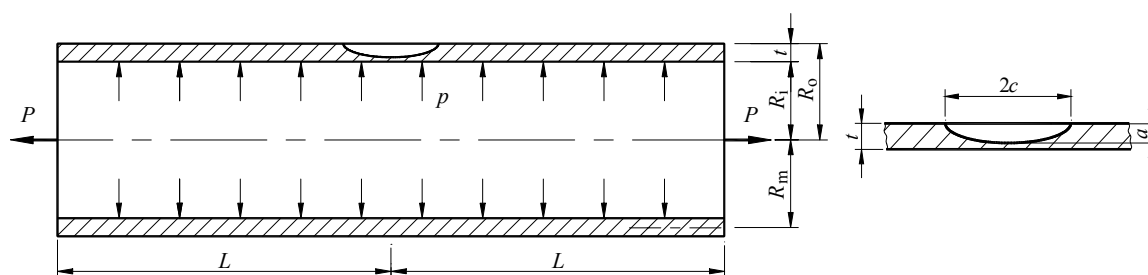


Figure 1. Geometry and dimensions of a tube subjected to internal pressure

These thick tubes are made of austenitic steel 08X18H10T that corresponds to AISI 321 grade. The material data used in the analysis determined experimentally in a standard tensile test are taken from [3]. The finite element analysis is performed by using the commercial FE package ABAQUS/Standard [4]. A typical finite element mesh applied in the analysis is shown in Fig. 2.

The plastic limit pressure solutions have been developed on the basis of the finite element limit load analyses employing elastic-perfectly plastic material behaviour. Using these solutions, a new analytical approximation of the plastic limit pressure has been developed for a wide range of cracks

$$p_L = (2/\sqrt{3}) \sigma_y \ln \frac{R_o}{R_i} \left[A_0 + A_1 (a/t) + A_2 (a/t)^2 \right]. \quad (1)$$

The values of the plastic limit pressure obtained by this study are compared with the empirical solutions presented in Ref. [1] and with finite element based plastic limit pressure expression developed in Ref. [2]. The proposed analytical approximation of limit pressure provides very useful tools for assessing the integrity of pressurized tubes.

Additionally, plastic collapse analyses with real strain hardening material model and large deformation theory were performed and the dependency of the flow stress on the crack geometry is investigated. It is assumed that tube is exposed to operating temperature of 300°C, so all material characteristics correspond to that value. Selected results of these analyses are presented in the form of load-displacement plots in Figure 3. The results demonstrate that choice of flow stress factor has a great influence in the estimation of a plastic collapse. The flow stress factor is more influenced by crack depth then by crack length. For larger and deeper cracks, flow stress factor values reach the values of about 0.4 which is in accordance with experimental results.

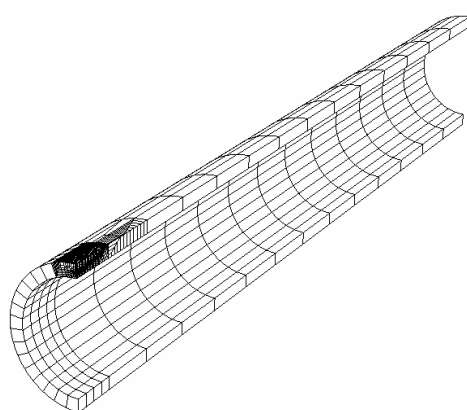


Figure 2. Typical FE mesh for a tube with external axial surface crack ($2c = 10$ mm, $a/t = 0.6$)

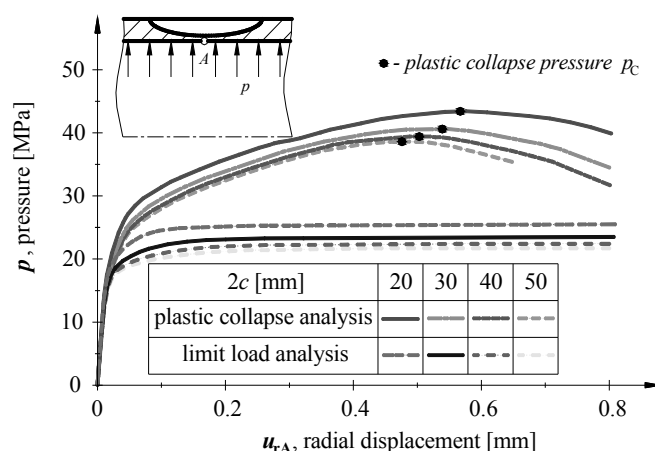


Figure 3. Variation of pressure with radial displacement of point A for a cracked SG tube with $a/t=0.6$ and $2c= 20, 30, 40, 50$ mm. Plastic limit and collapse load analyses.

References

- [1] Miller, A.G., "Review of Limit Loads of Structures Containing Defects", International Journal of Pressure Vessels and Piping, Vol.32, No.1-4, 1988, pp 197-327.
- [2] Kim Y.-J., Shim D.-J., Nikbin K., Kim Y.-J., Hwang S.-S. and Kim J.-S., "Finite element based plastic limit loads for cylinders with part-through surface cracks under combined loading", International Journal of Pressure Vessels and Piping, Vol.80, No.7-8, 2003, pp 527-540.
- [3] Norms for the calculation of strength of the equipment and the pipelines of nuclear power installations (PNAE G-7-002-86) / Regulatory authority for the control of nuclear power of the USSR. Moscow, Energoatomizdat, 1989, (in Russian).
- [4] ABAQUS User's guide and theoretical manual, Version 6.5, Hibbitt, Karlsson & Serensen, Inc., 2005.

Ivica Skozrit, assistant

Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, I. Lučića 5, 10000 Zagreb, Croatia, Tel: +385 1 6168115, Fax: +385 1 6168187, e-mail: ivica.skozrit@fsb.hr

Zdenko Tonković, associate professor

Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, I. Lučića 5, 10000 Zagreb, Croatia, Tel: +385 1 6168450, Fax: +385 1 6168187, e-mail: ztonkov@fsb.hr

Ivo Alfirević, professor

Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, I. Lučića 5, 10000 Zagreb, Croatia, Tel: +385 1 6168240, Fax: +385 1 6168187, e-mail: i.alfirevic@fsb.hr

AN INVESTIGATION INTO ATHEROSCLEROSIS USING THE FLUID-STRUCTURE INTERACTION TECHNIQUE

Niamh M. Quinn, Alojz Ivankovic and Aleksandar Karac

Keywords: Atherosclerosis, Fluid-structure interaction

1. Introduction

Atherosclerosis, also known as hardening of the arteries, is a disease of the large and middle-sized arteries. This is a chronic medical condition, which is characterised by the accumulation of cells at localised regions in the artery wall. Together with its related complications, this disease is the most common cause of death in western societies with the annual number of deaths exceeding those caused by cancer [1, 8]. It is also becoming increasingly prevalent in developing nations. The elasticity in affected arteries decreases with advancement of the disease. This adversely affects circulation. In the latter stages of the disease blood supply through the diseased artery may be cut off completely.

The fatty streak, the first stage in plaque development, is a stiffened hardened lesion in the artery wall, which should affect the deformation of the artery under the action of the cardiac cycle. It should therefore be possible to detect the disease whilst still in its early stages and prevent future complications. This work aims to determine the effect a stiffened lesion in an artery wall has on its deformation and on the stresses generated within it. The aim is to establish whether the deformation of an artery is measurably altered in the early stages of the disease.

2. Numerical Method

The finite volume method is used for both fluid and solid solution domains [2, 3]. Fluid and solid are discretised into separate meshes with a common interface between them. Each domain is solved simultaneously using a single code that passes information between the fluid and the solid. This code has been implemented in FOAM2.3.2 [10]. A simplified idealised geometry is used where the solid artery wall is modelled as a hollow tube, 100mm in length, internal diameter of 20mm and wall thickness of 3mm. Four arteries are modelled, one healthy and three with stiffened patches ranging in size and position.

A diseased lesion is modelled by altering the material properties in a region of the solid domain. The healthy mock artery has the following properties: density = 1000kg/m^3 , Poisson's ratio = 0.49995, Young's modulus = 0.5MPa. In the diseased region E is increased to 5MPa. A program assigns each cell a material index number based on its location within the mesh. In all simulations carried out blood is modelled as a compressible Newtonian fluid. The fluid properties are as follows: density = 1000kg/m^3 , bulk modulus = 2.2GPa and dynamic viscosity = 0.004kg/ms .

In order to simulate the effect of a static pressure on the healthy and diseased mock arteries constant fluid pressures of 5kPa and 10kPa were set. In the dynamic simulations a sudden step increase in pressure of 5kPa was applied at the fluid inlet and allowed to propagate along the length of the tube. The effect of an altered pressure profile has also been examined though this investigation focused on 2 models only. A 5kPa pressure pulse with $1.04\text{e-}04\text{s}$ duration was applied to the inlet, i.e. the pulse was allowed to propagate for $1.04\text{e-}04\text{s}$. The pressure inlet condition was then changed to zero and the simulation continued.

3. Experimental Method

Experimental testing is carried out on a mock artery made from polyurethane rubber tubing. The experimental work aims to provide clear evidence of the effects of the stiffened region on the deformation of the mock artery, and, at the same time, provide verification data for numerical predictions. Static pressure experiments are used to establish the relationship between internal pressure and tube diameter and based on these measurements the hoop stress and hoop strain are calculated.

4. Discussion

It has been demonstrated numerically that a stiffened region in the solid domain has a marked effect on the deformation and stress profiles of the artery wall. A constant fluid pressure produced a decrease in radial displacement but a sharp increase in hoop stress in the stiffened region compared to the un-stiffened region. Under a dynamic pressure the deformation profiles in patched and un-patched cases were distinctly different. As was the case in static simulations, the difference between patched and un-patched hoop stresses was of a greater magnitude than of the displacements. An investigation into the flow field revealed that the stiffened patch has relatively little effect on the flow patterns. This is true in all four models.

The results of the experimental tests indicate that the polyurethane, which from the tubing is made, is a non-linear material with Young's Modulus of 4.2MPa. Future work will include experimentally subjecting the mock artery to dynamic pressures, which will mimic the cardiac cycle. Stiffened patches will be applied to the tubing at prescribed locations. Future numerical work will use the exact geometry of the polyurethane tubing and will model its material properties accurately. In this way numerical and experimental results can be compared and validated.

Acknowledgements

This work is funded by the Irish Research Council for Science Engineering and Technology and Science Foundation Ireland.

References

- [1] Glass, C. K., Witztum, J. L., "Atherosclerosis: The Road Ahead", Cell Vol. 104, 2001, pp503-516.
- [2] Greenshields, C. J., Weller, H. G., Ivankovic, A., "The Finite Volume Method for Coupled Fluid Flow and Stress Analysis", Computer Modelling and Simulation in Engineering Vol. 4, No. 3, 1999, pp213-218.
- [3] Ivankovic, A., Karac, A., Dendrinis, E., Parker, K., "Towards early diagnosis of atherosclerosis: The finite volume method for fluid – structure interaction", Biorheology 39, 2002, pp401-407.
- [4] Kim, K., Weitzel, W. F., Rubin, J. M., Xie, H., Chen, X, O'Donnell, M., "Vascular intramural strain imaging using arterial pressure equalization", Ultrasound in Medicine and Biology, Vol. 30, No. 6, 2004, pp761-771.
- [5] Kleinstreuer, C., Nazemi, M., Archie, J., P., "Hemodynamics Analysis of a Stenosed Carotid Bifurcation and its Plaque – Mitigating Design", Journal of Biomechanical Engineering, Vol. 113, 1991 pp330-335.
- [6] Ku, D. N., "Blood flow in Arteries", Annual Review of Fluid Mechanics 29, 1997 pp399-434.
- [7] Libby, P., "Current Concepts of the Pathogenesis of the Acute Coronary Syndromes", Circulation 104, 2001, pp365-372.
- [8] Libby, P., "Atherosclerosis: the new review", Scientific American, 2002, pp51-59.
- [9] Steinberg, D., "Atherogenesis in perspective: Hypercholesterolemia and inflammation as partners in crime", Nature Medicine Vol. 8, No. 11, 2002, pp1211-1217.
- [10] Weller, H. G., Tabor, G., Jasak, H., Fureby, C., 1998. "A tensorial approach to computational continuum mechanics using object – oriented techniques", Computers in Physics, Vol. 12, No. 6, 1998, pp620-631.

Niamh M Quinn, PhD Student.School of Electrical, Electronic and Mechanical Engineering, University College Dublin, Dublin, Ireland. 353-7161880, niamh.quinn@ucd.ie

Alojz Ivankovic, Prof. Of Engineering Mechanics. School of Electrical, Electronic and Mechanical Engineering, University College Dublin, Dublin, Ireland. 353-7161994, alojz.ivankovic@ucd.ie

AN INVESTIGATION INTO DYNAMIC CRACK BIFURCATION IN PMMA USING A COHESIVE ZONE MODEL

N. Murphy and A. Ivankovic

Keywords: Dynamic Fracture, Cohesive Zone Model, Finite Volume Method.

1. Introduction

In this work, an investigation of the branching characteristics of small (63 by 25 mm) PMMA single edge notched tensile (SENT) specimens is presented. Tests were carried out at a rate of 2 mm/min with a variety of notch depths and specimen thicknesses. It was found that crack branching only occurred in thicker (8 mm) specimens and short notch depths of 0.1 and 0.15 mm, as shown in Figure 1.

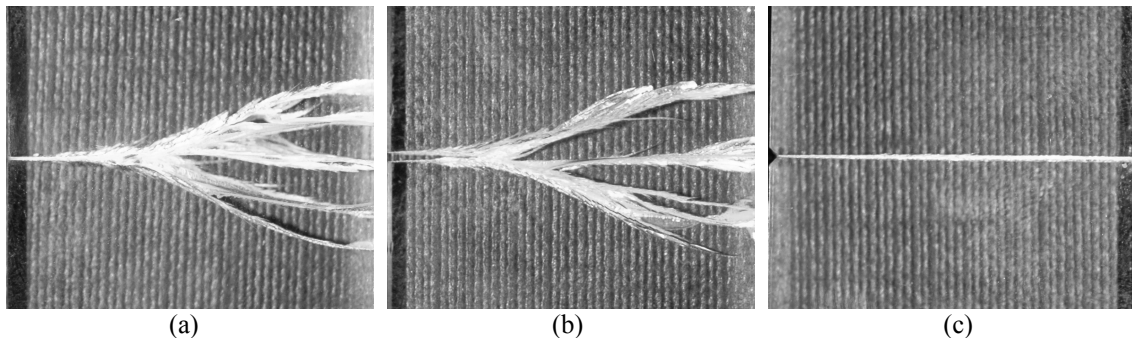


Figure 1. Branching patterns in specimens containing the 0.1 mm notches (a) and the 0.15 mm notches (b). Successful branching occurred in the majority of the 8 mm thick specimens tested at these shorter notch depths. Image (c) shows the macroscopically planar fracture surfaces for a specimen containing the 0.5 mm notch. The size of the images is 25 by 20 mm in each case.

The full-sized specimens were modelled using a fully-implicit finite volume / cohesive zone formulation [1-4]. A fine grid region was defined which extended 10 mm above and below the initial fracture plane and contained 20 by 20 micron cells. The number of continuum cells in these models exceeded 1.3 million. A cohesive strength and fracture energy of 80 MPa and 133 J/m² respectively were determined for this material and the shape of the cohesive law was adjusted until the static and dynamic behaviour of the model were in good agreement with experimental observations. The analysis procedure may be summarized as follows. At the end of a given time step, the tractions across the faces of each continuum cell were evaluated and if the failure criterion was satisfied anywhere in the model, cohesive cells were inserted between the continuum cells adjacent to this face. The time step was then repeated so that a converged solution was obtained for the model containing the recently added cohesive cells before proceeding to the next time step. In this investigation, a purely tensile failure criterion was assumed. Hence the cohesive cells were inserted if the

normal traction was greater than or equal to the cohesive strength. Further details may be obtained in [3-4].

Branching was successfully predicted for the 0.1 and 0.15 mm notches. Whilst the branching angles were higher than were observed in practice, the point of macroscopic branching was accurately predicted. The characteristic high frequency oscillations in crack velocity and the dramatic increase in fracture surface area as a function of notch depth were also predicted.

Finally, further insight was gained into the mechanism of macroscopic branch formation. It was confirmed that the evolving morphology of the expanding fracture process region plays a fundamental role in determining the macroscopic behaviour of the running crack. As shown in Figure 2, a cluster of micro-cracks was formed in the vicinity of the main crack tip which was propagating at velocities of the order of 800 m/sec. The event was then governed by the dynamic interaction between these micro-cracks and the main crack. Only the micro-cracks that formed far enough away from the main crack tip subsequently grew into a macroscopic branch, while those close to the main crack tip unloaded and did not propagate further. The successful branches then effectively grew backwards and coalesced with the main crack tip.

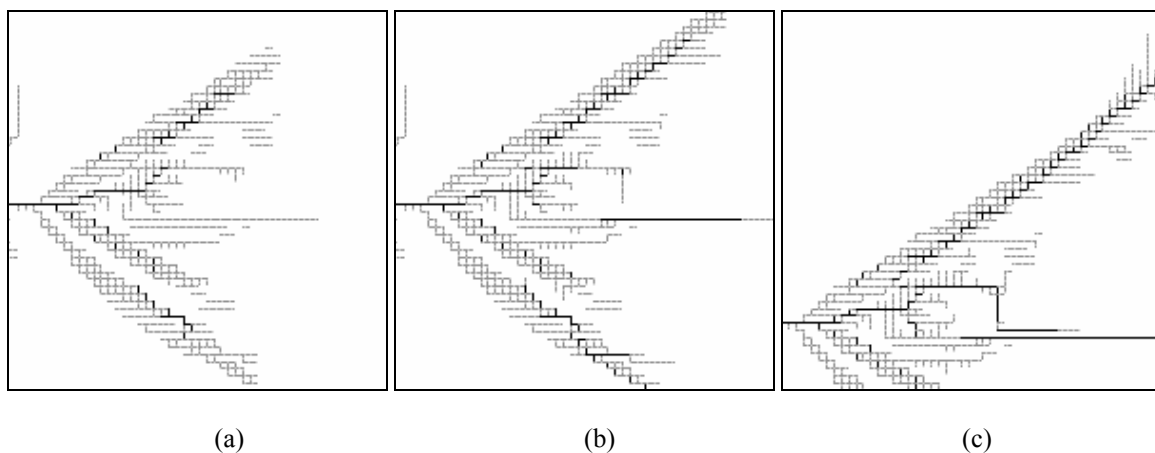


Figure 2. The development of a typical macroscopic branch during rapid crack propagation. Cohesive cells that are not fully separated are shown in grey, while fully separated cells are shown in black. Note that only the micro-cracks that form far enough away from the main crack tip subsequently grow into a macroscopic branch, while those close to the main crack tip unload and do not propagate further. As seen in (c), these cracks then effectively grow backwards and coalesce with the main crack tip. Size of windows: 1 mm by 1 mm.

References

- [1] Demirdzic I, Muzaferija S. "Finite volume method for stress analysis in complex domains". Int J Num Meth Eng, Vol. 37, 1994, pp 3751-3766.
- [2] Ivankovic, A., Murphy, N., Hillmansen, S. "Evolution of dynamic fractures in PMMA: experimental and numerical investigations". In: Aliabadi, M.H., Ivankovic, A., editors. Crack dynamics. Advances in fracture mechanics, vol. 9. Southampton, UK: WIT Press/Computational Mechanics Publications; 2005.
- [3] Murphy, N., Ivankovic, A. "The prediction of dynamic fracture evolution in PMMA using a cohesive zone model". Eng. Fract. Mech., Vol. 72, 2005, pp 861-875.
- [4] Murphy, N., Ali, M., and Ivankovic, A. "Dynamic Crack Bifurcation in PMMA". Eng. Fract. Mech., in press.

Neal Murphy,
School of Electrical, Electronic and Mechanical Engineering,
University College Dublin, Belfield, Dublin 4, Ireland. neal.murphy@ucd.ie

Professor Alojz Ivankovic,
School of Electrical, Electronic and Mechanical Engineering,
University College Dublin, Belfield, Dublin 4, Ireland. alojz.ivankovic@ucd.ie

SHAFTING ALIGNMENT CALCULATION AND VALIDATION CRITERIA

N.Vulić, A. Šestan, V. Cvitanić

Keywords: ship, propulsion, shaft line, alignment, transfer matrix method

1. Introduction

The main propulsion shaft line, as the essential part of a modern ship propulsion system, is exposed to various operating conditions throughout the ship's lifetime. Shaft line alignment procedure considers static and pseudo-static loading of the shafting in order to determine its static response. The main goal of this procedure is to determine and ensure achievement onboard the designed positions of the bearings in athwart direction in order to comply with certain criteria. The shaft line is usually modelled as a continuous multi-span beam on several supports. The supports may be considered as absolutely stiff, or linearly elastic, or even as real radial journal bearings.

The paper describes the conventional shafting alignment calculation procedure and how to obtain and validate the designed situation onboard. Calculation presumptions, modelling of shafting parts, material properties and loading are given in detail. The important issue is to establish the designed shafting elastic line onboard the ship, during the outfitting phase in the shipyard. The validation procedure of the achieved shafting position is described, with a real life example.

2. Shafting Alignment Calculation Procedure and Example

The shafting alignment calculation comprises the determination of support offsets, shafting elastic line and the determination of support reaction forces. In case of propulsion systems with gearboxes the scope of this analysis is restricted to shaft line from the propeller to the output shaft of the gearbox, together with its bearings and the bull gear.

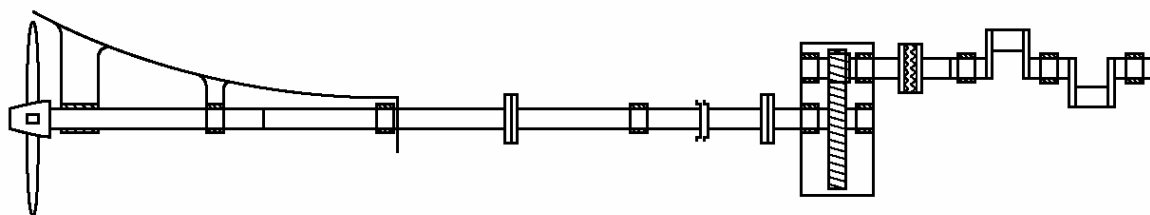


Figure 1. A typical marine shaft line involving a gearbox [1]

The data describing dimensions, material and loading of the shafts, as well as the bearings are to be available. The real system is modelled as a statically indeterminate system of variable section beams with multiple supports. Shaft line elements are modelled by means of circular section model elements. In general, the cross-section varies from one beam to another. The shaft line bearings are modelled by means of absolutely stiff or linearly elastic supports.

Method of initial parameters in its matrix form (so called: transfer-matrix method) is chosen as the most appropriate modelling and calculation procedure. It is a pure analytical method which describes the whole system by smaller matrices than the finite element method.

The calculation procedure has been implemented in the computer program *MarShAl* (Marine Shafting Alignment), specially developed for this purpose. For illustration a few characteristic diagrams obtained for an inland navigation ship are shown below.

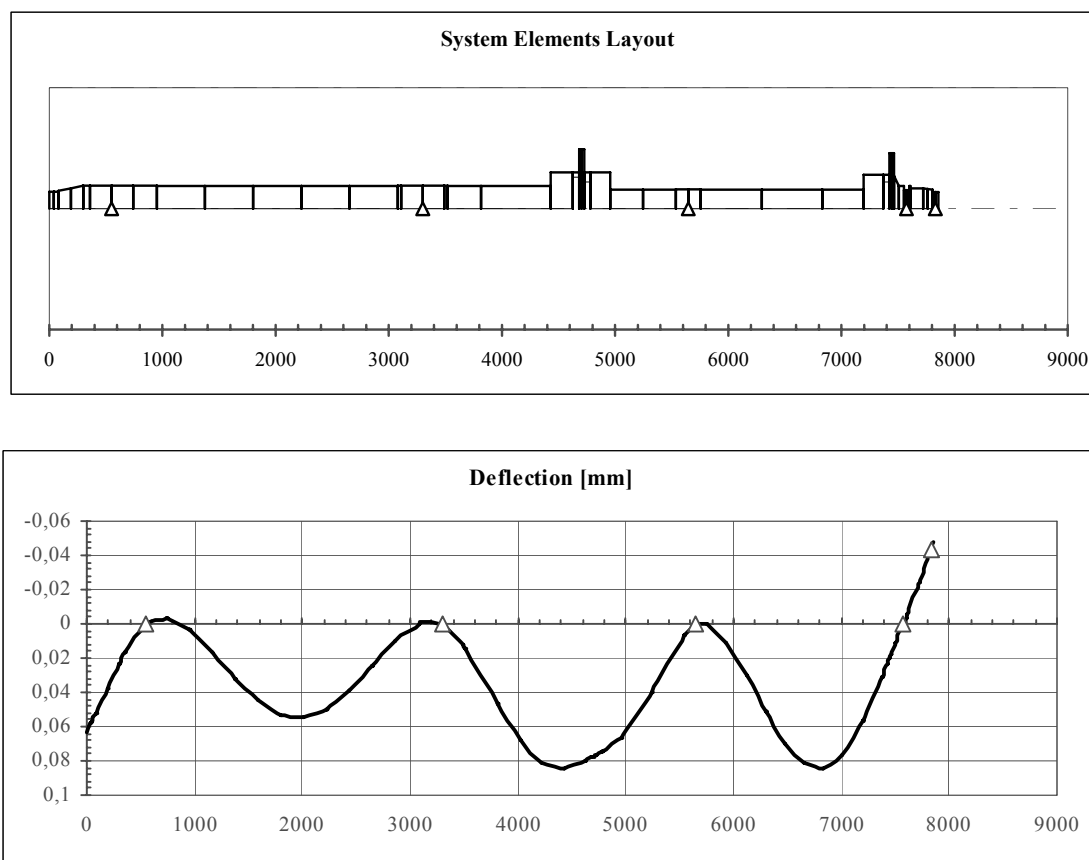


Figure 5. Example of calculated results

5. Conclusion

Shaft line is to be properly aligned in order to ensure its proper functioning throughout the complete ship lifetime. Careful calculation, as well as setting up of its results onboard (for the ship afloat) and their validation during the assembly and testing phase is essential. From the authors' experience it should be important to introduce the requirement for shafting alignment in the classification technical rules, even in the case of smaller ships. This will be the matter of further work.

References

- [1] Kozousek, W.M., Davies, P. G., "Analysis and Survey Procedures of Propulsion Systems: Shafting Alignment", LR Technical Association, Paper No. 5, London 2000
- [2] Vulić, N., "Advanced Shafting Alignment: Behaviour of Shafting in Operation", Brodogradnja, Vol. 52, No. 3, 2004, pp 203-212.
- [3] Vulić, N., "Shaft Alignment Computer Calculation (in Croatian)", Bulletin Jugoregistar, No. 1, 1988, pp. 1-13.

Prof. dr. sc. Nenad Vulić, Croatian Register of Shipping, Marasovićeve 67, HR-21000 Split, Croatia,
 Telephone: +385 21 408 163, Telefax +385 21 358 159, e-mail: **Nenad.Vulic@fesb.hr**
 Prof. dr. sc. Ante Šestan, University of Zagreb - FSB, Ivana Lučića 5, HR-10000 Zagreb,
 Telephone: +385 1 6168 378, Telefax +385 1 615 6940, e-mail: **ante.sestan@fsb.hr**
 Dr. sc. Vedrana Cvitanić, University of Split - FESB, Ruđera Boškovića bb, HR-21000 Split,
 Telephone: +385 21 305 865, Telefax +385 21 463 877, e-mail: **Vedrana.Cvitanic@fesb.hr**

PROJECTIVE CONSTRAINT VIOLATION STABILIZATION METHOD FOR MULTIBODY SYSTEMS ON MANIFOLDS

Zdravko Terze, Joris Naudet

Keywords: Constrained mechanical systems, Numerical integration on manifolds, Dynamic simulation of multibody systems

1. Introduction

During dynamical simulation of constrained multibody systems, a violation of system kinematical constraints is the basic source of time-integration errors and frequent difficulty that analyst have to cope with. The stabilized integration procedure, whose stabilization step is based on projection of the integration results to the underlying constraint manifold *via* post-integration correction of selected coordinates, is proposed and compared with similar integration schemes in [3]. As an extension of the previous work, a further elaboration of the projective stabilization method applied on holonomic and non-holonomic mechanical systems is reported in this paper.

2. Constrained MBS on manifolds

Unconstrained multibody system (MBS) is an autonomous Lagrangian system. By taking differentiable manifold approach, the configuration space $\mathcal{R}^n$ is considered to be a manifold $\mathcal{M}^n$. The manifold $\mathcal{M}^n$ and union of all tangent spaces make $2n$ -dimensional tangent bundle, $T\mathcal{M}: \bigcup_{\mathbf{x} \in \mathcal{M}^n} T_{\mathbf{x}}\mathcal{M}$, covered by the coordinates $\mathbf{x}, \dot{\mathbf{x}}: T\mathcal{M} = \{ (\mathbf{x}, \dot{\mathbf{x}}): \mathbf{x} \in \mathcal{M}, \dot{\mathbf{x}} \in T_{\mathbf{x}}\mathcal{M} \}$ [4, 5, 6].

$$\Phi(\mathbf{x}, t) = \mathbf{0}, \Phi(\mathbf{x}, t): \mathcal{R}^n \times \mathcal{R} \rightarrow \mathcal{R}^r \quad (1)$$

Holonomic constraints (1), which are imposed on the system, restrict system configuration space and impose constraints at velocity level: a trajectory $T: x^i = x^i(t)$ is ‘forced to move’ on the $n-r$ dimensional constraint manifold $\mathcal{S}^{n-r}(t)$, $\mathcal{S}^{n-r}(t) = \{ \mathbf{x} \in \mathcal{M}, \Phi(\mathbf{x}, t) = \mathbf{0} \}$, $t \geq 0$, $\mathbf{x}(t_0) \in \mathcal{S}^{n-r}(t_0)$ and linear constraint equation $\Phi_{\mathbf{x}}^*(\mathbf{x}, t)\dot{\mathbf{x}} = -\Phi_t = \boldsymbol{\tau}$ is induced at system velocities [1]. If additional nh non-holonomic constraints in the form $\Psi(\mathbf{x}, \dot{\mathbf{x}}, t) = \mathbf{0}$ are present in the system, they impose additional velocity constraints on holonomic constraint manifold tangent bundle $T\mathcal{S}$, $\dot{\mathbf{x}} \in T_{\mathbf{x}}^{n-r-nh}\mathcal{S}^{n-r} \subset T_{\mathbf{x}}^{n-r}\mathcal{S}^{n-r}$.

3. Constraint gradient projective method for stabilization of constraint violation

If system dynamical equations are based on the mathematical models in descriptor form, numerical violation of system constraints may appear during time integration and constraint violation stabilization procedure have to be applied. The stabilized integration procedure, whose stabilization step is based on projection of the integration results to the underlying constraint manifold *via* post-integration correction of selected coordinates, is proposed in [3].

The main problem that may occur during stabilization procedure is an inadequate coordinate selection that may have a negative effect on the integration accuracy along the constraint manifold (Fig. 1). By assuming that system exact position after integration step is ②, the optimized

ON THE USE OF CANONICAL MOMENTA FOR REAL-TIME SIMULATIONS

J. Naudet and D. Lefebvre

Keywords: Hamiltonian equations, Recursive formulation, Forward dynamics, Canonical momenta, Closed-loop systems

1. Introduction

There probably exist as many different methods to establish the equations of motion of multi-body systems as there are researchers in the field. The multitude of algorithms and the complexity of multibody system dynamics make it difficult, if not impossible, to decide which one is more efficient. It is however reasonably clear that the so-called $O(n)$ -algorithms are the better choice for systems with a large number of degrees of freedom. Most algorithms with linear complexity are, to some extent, a variation of the articulated body method, first introduced in a useful form by Featherstone [1]. A similarity all these methods share, is the use of accelerations. Kane [2] pointed out that it may be possible to reduce the number of required operations to establish the equations of motion by using adequate quasi-velocities, which are linear combinations of the coordinate velocities. In this presentation, we show that an efficient algorithm can be designed, based on the system's canonical momenta, which are, in fact, a specific kind of quasi-velocities. Furthermore, the use of these canonical momenta lead to a set of Hamiltonian equations of motion.

2. Equations of motion of a single rigid body

In multibody dynamics, it is common to use so-called spatial vectors to obtain an elegant way of formulating the equations of motion. A spatial vector contains information about the 6-dimensional space of a rigid body, translational and angular motion are incorporated in one single vector. Besides the well-known spatial velocity, acceleration and force vectors, one can define a spatial momentum vector, which contains the linear and angular momenta of a body. It is interesting to note that the projection of this spatial momentum vector into the joint subspace results in the set of canonical momenta conjugated to that joint. Expressing the Newton-Euler equations in terms of the spatial momentum vector, one can easily obtain the set of Hamiltonian equations for a single rigid body.

3. Equations of motion of open-loop multibody systems

To tackle open-loop multibody systems, the concept of articulated bodies as introduced by Featherstone [1] is used. The articulated momentum of an articulated body is defined as the sum of the momenta of the individual bodies, reduced to the base body. Here too, one can obtain the canonical momenta by projecting the spatial articulated momentum vector into the joint subspace. This allows to design a recursive algorithm to establish the equations of motion. In table 1, one can find the operations count for several recursive algorithms. Obviously, the method based on canonical momenta is more efficient for formulating the equations of motion. The global efficiency of the algorithm is however also dependent on the behavior during numerical integration. Numerical experiments showed that the effects of the numerical integration on a set of Hamiltonian equations or a set of acceleration-based equations are very similar.

Table 2: operations count

Algorithm	Additions	Multiplications	Total
Featherstone[1]	275n-18	336n-220	611n-238
Vukobratović[3]	231n-294	249n-272	480n-566
Valasek[4]	206n-345	226n-343	432n-688
Rein[5]	195n-247	216n-317	411n-669
Canonical momenta[6]	178n-230	185n-245	363n-475

4. Equations of motion of constrained multibody systems

The algorithm has been extended to cope with closed-loop systems as well. Two different approaches were developed. Both methods start with cutting joints in the system to obtain an underlying open-loop structure. In the first approach, the underlying open-loop structure is treated as a normal open-loop system, with that difference that the unknown reaction forces resulting from the additional constraints are propagated symbolically throughout the recursion steps. When the equations of motion are established, it is possible to eliminate the unknown reaction forces by using the constraint equations. A disadvantage of this method is that it loses its efficiency when heavily constrained systems are considered. The second approach is an $O(n+m)$ method, m being the number of constraints. It is based on the works of Valasek [4], Anderson and Critchley [7] and performs a coordinate reduction in combination with the principle of virtual power to achieve linear complexity. The resulting algorithm has much in common with the algorithm for open-loop systems. The articulated momentum is now replaced by a constrained momentum, which is a combination of the former and a contribution arising from the constraints. Besides the linear complexity, the algorithm has the additional advantage of introducing the constraint equations at the velocity level, such that the constraint violation errors do not grow rapidly.

5. Conclusions

Starting from the concept of spatial momentum vector, it is possible to formulate a set of Hamiltonian equations of motion for constrained multibody systems. The resulting algorithms are efficient regarding the number of required operations and the constraint violation errors.

References

- [1] R. Featherstone, *Robot Dynamics Algorithms*, Kluwer Academic Publishers (1987).
- [2] T. R. Kane and D. A. Levinson, *Dynamics: Theory and Applications*, McGraw-Hill (1985)
- [3] M.K. Vukobratovic and al. "A unified approach to mathematical modeling of robotic manipulator dynamics", *Robotica*, 12, 411-420, (1994).
- [4] V. Stjeskal and M. Valasek, *Kinematics and Dynamics of Machinery*, Marcel Dekker (1996)
- [5] U. Rein "Efficient Object Oriented Programming of Multibody Dynamics Formalisms", *Proceedings of the NATO-Advanced Study Institute on Computer Aided Analysis of Rigid and Flexible Mechanical Systems* 12, 59-69, (1993).
- [6] J. Naudet, D. Lefeber and Z. Terze "Forward Dynamics of Open-Loop Multibody Mechanisms using an Efficient Recursive Algorithm based on Canonical Momenta", *Multibody Systems Dynamics*, 10(1), 45-59, (2003)
- [7] K.S. Anderson and J.H. Critchley "Order N Performance Algorithm for the Simulation of Constrained Multi-Rigid-Body Dynamic Systems", *Multibody Systems Dynamics*, 9, 185-212, (2003).

Joris Naudet (dr. ir.)

Vrije Universiteit Brussel, Faculty of Applied Sciences, Department of Mechanical Engineering, Pleinlaan 2, 1050 Brussels, Belgium, tel:+32/(0)2/629.31.81, fax:+32/(0)2.629.28.65, joris.naudet@vub.ac.be

Dirk Lefeber (prof. dr. ir.)

Vrije Universiteit Brussel, Faculty of Applied Sciences, Department of Mechanical Engineering, Pleinlaan 2, 1050 Brussels, Belgium, tel:+32/(0)2/629.28.64, fax:+32/(0)2.629.28.65, dlefeber@vub.ac.be

Nonlinear Wall Models for Earthquake Response Analysis

Vladimir Z. Sigmund, Tanja Kalman and Ivica Guljaš

Keywords: Buildings, r-c walls, nonlinear models, earthquake response, experiment, analysis

1. Introduction

The use of nonlinear time history analysis for design of new buildings and for strength evaluation of the existing ones is allowed in the Eurocode8. Linear elastic analysis depends on few factors and gives more-or-less comparable results irrespective on the computer code used. The qualities of results obtained by the nonlinear analysis depend on many factors and the obtained results can have only a vague connection to the reality. Their quality, sometimes, questions the endeavor put into it. (If we know that we are going to be wrong we could be it the easy way).

There are various numerical models available for the analysis of wall nonlinear response during earthquake excitation. They could be divided into two main groups: microelement- (nonlinear-FEM, micro-fiber, etc.) and macro element-models (beam-column, structural-wall, etc.) models. In this paper, we modeled two reinforced/concrete cantilever walls tested on the shaking table during two European experiments and compared the measured results with the results obtained by using various numerical models.

The numerical models used are: micro fiber element model used in **ZeusNL** (Elnashai, 2004.), structural-wall element model with- and without shear spring used in **Ruaumoko** (Athol, 2003.) and one component beam-column (wall) element model used in **LarzWD** (Lopez, 1988.) with the modifications (Sigmund, 2000.). The quality of numerically calculated results have been verified against the measured ones in terms of the response waveforms, maximum displacements and base shears, inter-story drifts of the first story and stiffness degradation expressed as period elongation.

2. Experimental analysis

Large-scale RC cantilever structural walls were studied in the frame of the CAMUS (Conception et Analyse des Murs sous Séisme) program on the shaking table at the CEA, Saclay in France. Within the CAMUS3 (Combescure, 2000.), the wall was designed according to the EC8 standard. The project was supported by international benchmark studies (blind analytical predictions and after test numerical simulation).

Within the ECOLEADER Slo-wall test (Kante, 2005.) a shaking table test of a 1/3rd scaled specimen, representing relatively low, thin and lightly reinforced H-shaped structural wall with openings was performed. Seismic loading was applied in both horizontal directions at the same time.

3. Numerical models

The performed analysis has been limited to the plane in the excitation direction and out-of-plane deformations were neglected. Models were fixed to the shaking table and its deformations were neglected in the calculations. Masses were concentrated at the floor levels and equivalent viscous damping was assumed to be 2%. Geometry of the sections and initial mechanical characteristics of

the materials were used the same in all models.

The computer codes used were: Ruaumoko (Athol, 2003.) with the Structural-wall (R-wall) and Structural-wall with shear spring model (R-wall+spring), ZeusNonlinear (ZeusNL-wall, Elnashai, et al. 2004.) and LarzWD (LarzWD-wall, Lopez, et al. 1998.).

Table 1. Measured and calculated characteristic values

		Camus III					Ecoleader Slo-WALL				
		Exsp	Wall R	w+s R	Wall Z	Wall L	Exsp	Wall R	w+s R	Wall Z	Wall L
Correlation	dxr6	1	0.7	0.7	0.52	0.78	1.00	0.43	0.42	0.69	0.68
	IDR23	1	0.64	0.65	0.54	0.73	1.00	0.28	0.26	0.55	0.54
	BS	1	0.56	0.62	0.4	0.76	1.00	0.06	0.09	0.56	0.71
Max / Min values	MDR %	0.51	0.6	0.45	0.51	0.4	0.16	0.27	0.48	0.29	0.09
		-0.71	-0.25	-0.65	-0.77	-0.75	-0.21	-0.55	-0.52	-0.31	-0.11
	IDR %	0.87	0.1	0.28	0.45	0.6	0.33	0.12	0.20	0.12	0.06
		-0.5	-0.34	-0.18	-0.3	-0.34	-0.48	-0.27	-0.24	-0.13	-0.07
	BS (kN) Max/Min	148.4	107.5	159.5	122.5	154.7	63.50	34.42	35.37	94.40	52.21
		-151.5	-66.81	-155.4	-158.9	-100.8	-80.15	-30.71	-41.13	-110.04	-70.88
Period	T0 (s)	0.15	0.15	0.16	0.1	0.11	0.15	0.11	0.12	0.12	0.12
	T1 (s)	0.22	0.31	0.33	0.24	0.2	0.23	0.42	0.41	0.11	0.16

4. Conclusion

Though geometry, material characteristics, masses and input excitations were known before the calculations, different codes gave different quality of the calculated nonlinear response results. The general response values (maximum displacements and base-shear) were much better represented than local ones (inter-story drift). It seems that the first story waveforms are more sensitive to small differences in evaluated stiffness properties.

Nonlinear response of r/c walls to earthquake excitation gives a clue on what the true response might be. It still does not fully represent the true wall behavior during earthquake and must be observed statistically. Microelement model (as used in ZeusNL) have no visible advantages over macro element model. Considering many uncertainties involved in establishing input design motion, material characteristics and member behavior, simple macro-element model gives a good idea of expected lateral drifts and maximal responses during inelastic earthquake response of structures and is justified for engineers' usage.

References

- [1] Athol, J.C., 2003. RUAUMOKO-2D Inelastic Dynamic Analysis, University of Canterbury, NZ
- [2] Combescure, D., and Chaudat, Th., 2000. ICONS European program seismic tests on r/c bearing walls, CAMUS 3 Specimen, CEA Report DMT, Saclay, France
- [3] Elnashai, A. et al., 2004. ZeusNL, University of Illinois at Urbana/Champaign, USA, 2004.
- [4] Lopez, R. and Sozen, M.A., 1988. A Guide to Data Preparation for LARZWD and LARZWS Computer Programs for Non-linear Analysis of Planar Reinforced Concrete Structures Incorporating Frame and Walls, University of Illinois, Urbana, IL.

Vladimir Z. Sigmund, Ph.D., Tanja Kalman, MCE and Ivica Guljaš, Ph.D.
University of Osijek, Faculty of Civil Engineering, Drinska 16a, 31000 Osijek, Croatia, sigmund@gfos.hr

AUTHOR INDEX

- Adámek, Vítězslav [121]¹
 Agić, Ante [178], [179]
 Alfirević, Ivo [184]
 Avsec, Jurij [116]
 Bartoli, G. [161]
 Bella, Vladimír [132]
 Beyl, Pieter [192]
 Bibić Dževad [126]
 Borkovec, Jan [141]
 Borovinšek, M. [146], [147]
 Borri, Claudio [161]
 Brnić, Josip [119], [140], [162]
 Čanađija, Marko [140]
 Čarija, Zoran [168]
 Čavrak, Marko [170]
 Cipleu, Adrian [158]
 Čohodar, Majda [166]
 Costa, C. [161]
 Črnjarić-Žic, Nelida [171], [173], [174]
 Crnković, Bojan [171], [173], [174]
 Cvitanić, Vedrana [130], [169], [188]
 Damić, Vjekoslav [166]
 Del Piero, Gianpietro [104]
 Doleček Vlatko [126], [128]
 Domazet, Ž. [103]
 Dragović, Luka [168]
 Dreija Z. [155]
 Družeta, Siniša [171]
 Dumitru, Ion [158]
 Duplančić, Igor [130]
 Džijan Ivo [115]
 Elizabeta Hristovska [185]
 Faur, Nicolae [158]
 Filipov A. [155]
 Filipović Ivan [126]
 Freddi, A. [153]
 Fuxa, Jan [124]
 Gajdošík, Jan [105]
 Ghelli, Daniele [163]
 Gotovac, Blaž [175]
 Gotovac, Hrvoje [175]
 Gubeljak, Nenad [182]
 Guljaš, Ivica [191]
 Holjević, Danko [171]
 Hoster, Josip [181]
 Hyhlík, T. [137]
 Ikegami, Kozo [120]
 Irhan B. [144]
 Isić, Safet [128]
 Ivankovic, A. [186], [187]
 Janda, T. [136]
 Jarak, Tomislav [181]
 Jasak Hrvoje [193]
 Jelović, Mario [131]
 Kalman, Tanja [191]
 Kalousková, Marie [106], [107]
 Karabegović, Isak [128]
 Karac, A. [186]
 Karšaj, Igor [180]
 Kaukič, M. [133]
 Khodaei Z. Sharif [122]
 Kompiš, M. [133]
 Kompiš, V. [133]
 Kong, Chin-Hwa [102]
 Konjatić, P. [182]
 Konrads G. [117]
 Kozak, D. [182]
 Kožar, Ivica [139], [142], [144]
 Kozulić, Vedrana [175]
 Kranjčević, Lado [173]
 Krejčí, T. [151]
 Krizmanić, Severino [115] [183]
 Kroupa, T. [118]
 Krstulović-Opara, Lovre [103], [160]
 Kubala, Rostislav [124]
 Kuffová, Mariana [132]
 Kulenović, Zlatan [129]
 Lanc, D. [119]
 Lancioni, Giovanni [104]
 Laš V., [150], [118]
 Lefebvre, Dirk [190], [192]
 Liu, Chi-Min [102]
 Lovrenić, M. [164]
 Lozina, Željko [169], [176]
 Maćešić, Senka [171]

¹ Number of paper defined in contents

Mandić Dino [131]
 Mannini, C. [161]
 March, Riccardo [104]
 Matejiček, F. [182]
 Matějka, Milan [125]
 Medvidović, Tino [139]
 Měšťánek P. [150]
 Mihanović, Ante [157]
 Mijović, Budimir [178], [179]
 Minak, Giangiacomo [163]
 Mozga N. [155]
 Mrazović, Nataša [127]
 Mrša, Zoran [168], [170]
 Murphy, N. [187]
 Naudet, Joris [189], [190], [192]
 Nikolić, Željana [157]
 Novák Jan [105], [107], [123], [149]
 Oblak Maks [112], [116], [182]
 Öchsner, Z. [103]
 Olmia, G. [153]
 Perasović, Ivan [160]
 Petrović Marin [126]
 Petruska, Jindrich [141]
 Plachý, Tomáš. [143]
 Polák, Michal [143]
 Popelka, Lukáš [125]
 Predan, J. [182]
 Pušenjak, Rudolf [112]
 Pustaić, D. [164]
 Quinn, N. [186]
 Radelja Tonči [175]
 Ren, Z. [103], [114], [146], [147]
 Repčić, Nedžad [101]
 Rezek, Antonín. [143]
 Rosman, Riko [110]
 Rudzītis J. [113], [117]
 Šafařík, P. [137]
 Sansour, Carlo [180]
 Šavar, Mario [183]
 Sedlar, Damir [176]
 Sedmak Aleksandar [182]
 Šejnoha, Jiří [106], [108], [109], [123], [151]
 Šejnoha, Michal [105], [106], [107], [108], [109], [123], [136], [149], [151]
 Semenski, Damir [148]
 Šestan A. [188]
 Shperbergs J. [117]
 Sigmund, Vladimir Z. [191]
 Šimić, Diana [111], [127]
 Šimurda, D. [137]
 Škifić, Jerko [171], [174]
 Skoček, Jan [105]
 Skozrit, Ivica [184]
 Sopta, Luka [171]
 Sorić, Jurica [180], [181]
 Součková, Natálie [125]
 Štimacm, Goranka a170]
 Sudnieks F. [155]
 Sulyok-Selimbegović, Marta [127]
 Sušić, Aleksandar [148]
 Svaguša, Tomislav [156]
 Sýkora, Jan [108], [109]
 Terze, Zdravko [189]
 Tonković, Zdenko [184]
 Torić, Neira [142]
 Torims T. [113]
 Travaš, Vanja [144]
 Turkalj, G. [119], [162]
 Ulbin, M. [114]
 Valdas Čaika, [131]
 Valenta, R. [151]
 Valeš, František [121]
 Van Damme, Michaël [192]
 Van Ham, Ronald [192]
 Vanderborght, Bram [192]
 Verrelst, Björn [192]
 Versluys, Rino [192]
 Vesenjaj, M. [103]
 Virag Zdravko [115]
 Vizentin, Goran [162]
 Vlak, Frane [130], [169]
 Vojvodič-Tuma J. [182]
 Vokáč M. [149]
 Vorel, Jan [108], [109]
 Vučina, Damir [176]
 Vulić, Nenad [188]
 Wolf, Hinko a148]
 Zeman, Jan [105], [107], [108], [122], [123]
 Zemčík, R. [118], [150]
 Živaljić, Nikolina [157]
 Živić, Marija [183]